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THE ANALYSIS OF THE INFLUENCE OF A TORSIONAL VIBRATION DAMPER ON TRANSVERSAL DISPLACEMENT OF A CRANKSHAFT

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Abstract

The article presents a consideration of the influence of a damper of torsional vibrations (TVD) on transverse dynamics of the crankshaft. Dampers of this type work based on the dynamic eliminator of vibrations. The systems of this type in modern combustion engines are more often used to reduce torsional vibrations. It is strictly connected with the increase of fatigue life of the crankshaft in case when its amplitude of vibrations has a crucial meaning. Computation models used to choose a damper assume that transverse and torsional vibrations occurring in a crank system are not coupled. Modern trends in machine design make the new devices lighter and less rigid. Therefore, the couplings between small vibrations may be more important. Therefore, the effects observed in angular vibrations can be transmitted to transverse vibrations. In this context, the impact of dampers TVD on transverse dynamics is particularly interesting.

In the introduction, the article discusses the basic types of torsional vibration dampers and the method of selection of this type of systems. The next part of the article presents the model of a combustion engine with a dynamic eliminator of vibrations, which includes bending-torsional vibrations occurring in the system. For this purpose, the linear relation of the vector of generalized co-ordinates and generalized forces in the system was assumed. The next part shows numerical simulations, which were carried out in the paper. The paper presents obtained results with their further analysis in frequency domain. The whole paper is summarized with synthetic conclusions on the simulations, the impact of applied eliminator on the dynamics of the entire crank system, in particular transverse vibrations. In addition, there was indicated a possibility of using presented results in research and diagnostics used in automotive industry of torsional vibration dampers.

Keywords: *crankshaft systems, torsional vibrations damper system of differential equation, numerical simulations*

1. Introduction

Piston engines are characterized with a big vibroactivity. It is connected with the character of their work. The use of advanced technology and the need to „tighten" the construction results in the fact that currently produced engines are more susceptible to vibrations. Thus, constructors are required to be more precise during the process of construction of typical elements of the engine and use additional elements, which will influence a decrease of destructive vibrations of the engine [6, 7, 11, 14, 17].

In most cases of car combustion piston engines, rubber dampers are used as torsional vibration dampers. They consist of three parts: a steel hub, a steel flywheel and a rubber ring which connects these two elements. The ring may be pressed or vulcanized to these two elements. The second solution is better because in the case of a sudden overload of the flywheel, it rotates about a small angle in relation to the hub and it still has its function In the case of a vulcanized damper, devulcanization means that the damper is permanently damaged and it needs to be changed for a new one.

The aim of this paper is to make the model of a currently produced engine TwinAir used in Fiat, which will be the basis for proposing corrected elements of the construction, which already exist, and the new ones. Such elements will not only increase the vibroacoustic emission of this engine but they will also influence its decrease like torsional vibration dampers mentioned earlier [9, 12, 18, 20, 21].

2. Modelling torsional vibration damper

The main objective of applying torsional vibration dampers in drive systems is to reduce the amplitude of torsional vibration of such a system. Reduction of maximum values of oscillations directly influences the increase of strength of the elements, which transmit power. This is due to fatigue character of loads occurring in power units [2, 4, 5, 22].

The main cause of excessive vibration values is work of the device in the neighbourhood of critical velocities. These are the easiest methods of vibrations reduction:

- energy dissipation in an additional element, e.g. in a viscous damper,
- change in the frequency structure of the machine by modifying the stiffness and mass distribution in the proposed object.

A typical method of forming the natural frequencies of the system is adding the inert element to the system by elastic connector. Such an additional component is a dynamic vibration eliminator or torsional vibration damper.

The industry uses several kinds of dampers of vibrations, which work like a dynamic eliminator. Solutions with one degree of freedom, which allow to „deleting" only one frequency of free vibrations, are the most common. In case of systems with polyharmonic extortion working in the range of frequencies, it is necessary to use wideband dampers Such elements are usually the systems with a few degrees of freedom, whereby the range of minimized frequencies can be wider [1, 8, 9, 12, 15].

For most technical solutions used in practice, it is possible to use models with one or two degrees of freedom presented schematically in Fig. 1 and 2.

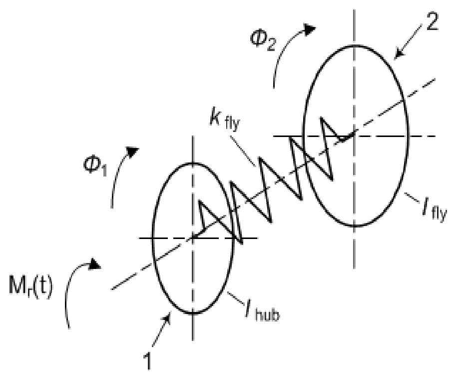

Fig. 1. Model of 1DOF torsional vibration damper

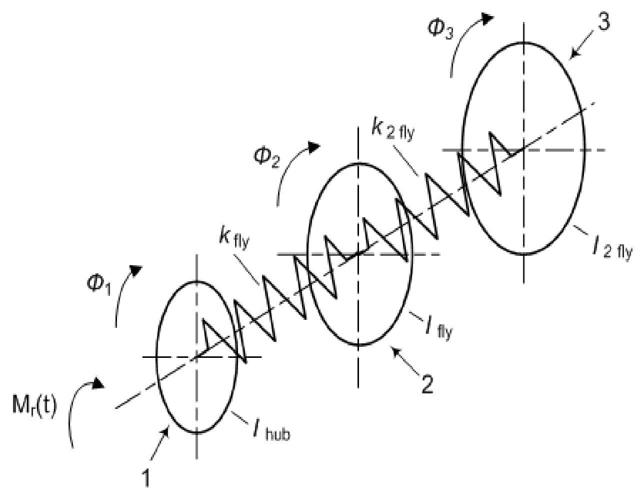

Fig. 2. Model of wideband torsional vibration damper

3. The object of model investigation

The methodology of modelling presented in the article and numerical simulations are supposed to be useful during the tests and diagnostics of the engine TwinAir and its equipment [10, 16, 23]. It is a two-cylinder turbocharged petrol engine. Model 3D and visualization of construction of this drive system is presented in Fig. 3 and 4.

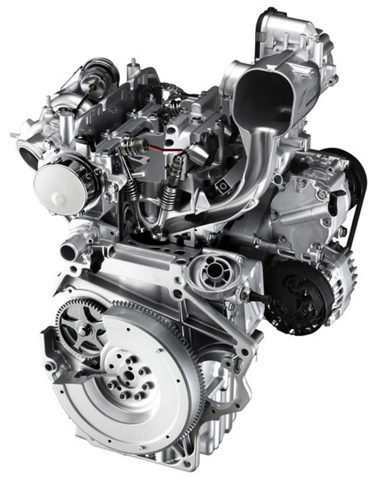

Fig. 3. Fiat's TwinAir engine piston

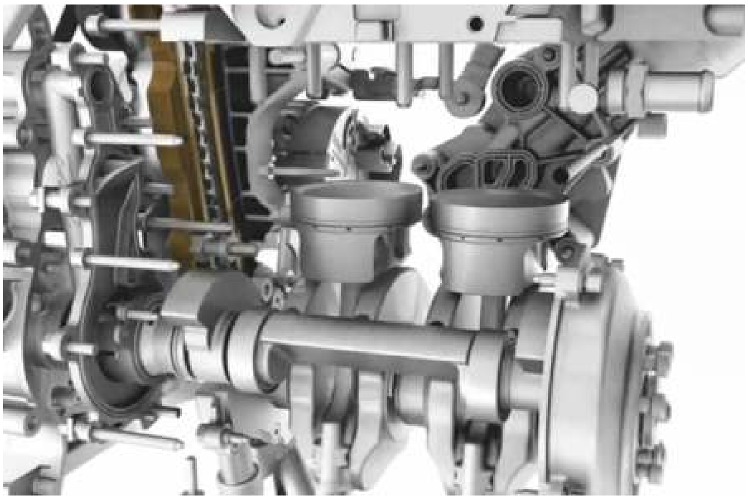

Fig. 4. The model of powertrain of TwinAir engine

Crankshaft of FIAT two-cylinder petrol engine TwinAir is a modelled object.

4. Linear-elastic model of deformable crankshaft

The crankshaft of the combustion engine is the object with a continuous mass distribution. It means that global dynamic and static properties result from spatial distribution of mass density and stiffness parameters. In this case, it is necessary to use the model of continuous system. Unfortunately, it is not feasible because of a very complex shape of the shaft and complicated material structure, which results from methods of forming process most commonly used in industry. Therefore, it is better to use the model of linear-elastic medium, for which the modelling results will coincide with the experiment. In this case, the parameters of the system of equations will have a generalized character and they will not be a direct representation of a continuous system. Obtained parameters can be interpreted as global stiffness coefficients of the shaft, resulting from its geometrical parameters and material properties. Such an approach will be characterized with a higher level of abstraction, where at the expense of difficult physical interpretation of parameters, the significant simplification of equations describing the object is obtained.

Unfortunately, such an approach is tantamount to separation of elastic properties of the system from its mass distribution. This means that such a model will not show significant characteristics for continuous systems, e.g. the number of natural frequencies will be finite, and their values and modes of vibrations will be different from the real ones. Despite this, such an approach allows for a more effective analysis of the problem. The inconveniences may be eliminated in the process of parametric identification of a dynamic model. Results obtained in this way will constitute reduced dynamic parameters of the model, including assumed structural errors of the model.

Assuming that the values of deformations occurring in the system are small and the materials used are linear-elastic in terms of acceptable displacements, the relationships between forces and generalized displacements are linear:

$$F = K \cdot u, \tag{1}$$

where:
K – stiffness matrix,
F – generalized force vector,
u – generalized displacement vector.

Modelled crankshaft consists of two cranks. Fig. 5 shows the system of loads and basic material parameters of the proposed model.

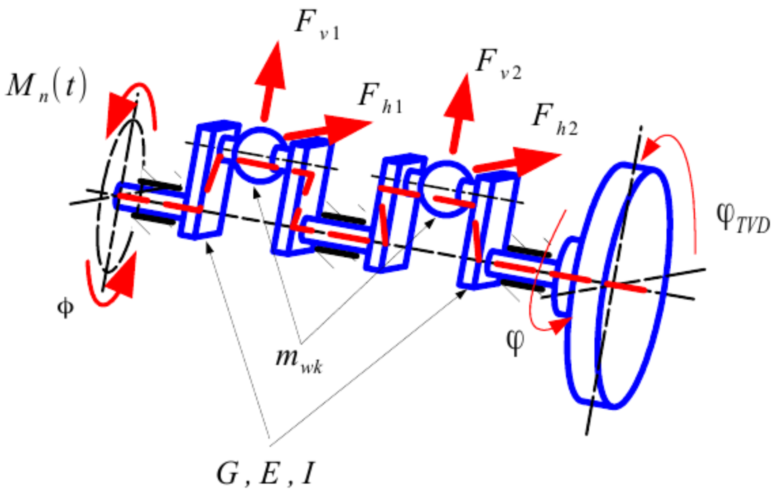

Fig. 5. The model of the crankshaft of one piston

The global stiffness matrix of the shaft shown in Fig. 5 has the following form:

$$K = \begin{bmatrix} k_{n1n1} & -k_{n1n2} & 0 & 0 & 0 \\ -k_{n1n2} & k_{n2n2} & 0 & 0 & 0 \\ 0 & 0 & k_{\tau1\tau1} & -k_{\tau1\tau2} & -k_{\tau1\theta} \\ 0 & 0 & -k_{\tau1\tau2} & k_{\tau2\tau2} & -k_{\tau2\theta} \\ 0 & 0 & -k_{\tau1\theta} & -k_{\tau2\theta} & k_{\theta\theta} \end{bmatrix}. \tag{2}$$

5. The equations of motion for a modelled crankshaft with a damper with one degree of freedom

There were used the Lagrange equations of the second kind in order to find the equations of motion of the crank mechanism with elastic shaft. Due to the expected character of motion, the system of coordinates presented in Fig. 6 was introduced.

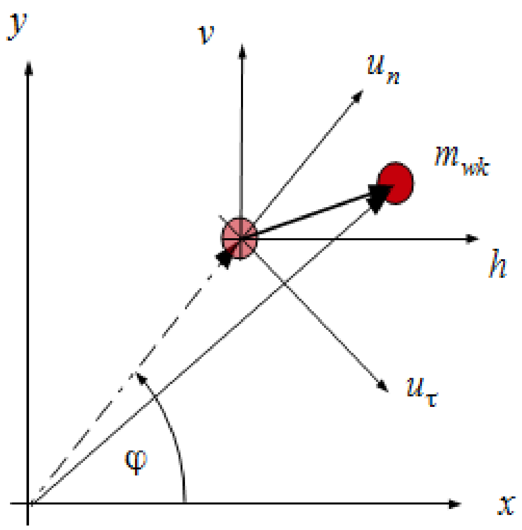

Fig. 1. The model of the single crank of the considered crankshaft in coordinate frame

The equations of linear-elastic model of the crankshaft in a proposed system of coordinates are as follows:

$$m_w \ddot{u}_{n1} - m_w R \dot{\varphi}_L^2 + k_{n1n1} u_{n1} - k_{n1n2} u_{n2} = P_{r1}, \tag{3}$$

$$m_w \ddot{u}_{n2} - m_w R \dot{\varphi}_L^2 - k_{n1n2} u_{n1} + k_{n2n2} u_{n2} = P_{r2}, \tag{4}$$

$$I_{kl} \ddot{\varphi}_L + m_{wk} R^2 \ddot{\varphi}_L + m_w R \ddot{u}_\tau - k_{\tau 1\theta} u_{\tau 1} - k_{\tau 2\theta} u_{\tau 2} + k_{\theta\theta}(\varphi_L - \varphi_P) = M_0, \tag{5}$$

$$m_w \ddot{u}_{\tau 1} + m_w R \ddot{\varphi}_L + k_{\tau 1\tau 1} u_{\tau 1} - k_{\tau 1\tau 2} u_{\tau 2} - k_{\tau 1\theta}(\varphi_P - \varphi_L) = P_{\tau 1}, \tag{6}$$

$$m_w \ddot{u}_{\tau 2} + m_w R \ddot{\varphi}_L - k_{\tau 1\tau 2} u_{\tau 1} + k_{\tau 2\tau 2} u_{\tau 2} - k_{\tau 1\theta}(\varphi_P - \varphi_L) = P_{\tau 2}, \tag{7}$$

$$I_{kP} \ddot{\varphi}_P - k_{\tau 1\theta} u_{\tau 1} - k_{\tau 2\theta} u_{\tau 2} + k_{\theta\theta}(\varphi_P - \varphi_L) - k_{TVD}(\varphi_{TVD} - \varphi_P) = 0, \tag{8}$$

the equation of motion of dynamic vibration eliminator:

$$I_{TVD} \ddot{\varphi}_{TVD} + k_{TVD}(\varphi_{TVD} - \varphi_P) = 0. \tag{9}$$

6. The analysis of transverse vibrations of the crankshaft

The mathematical model of the dynamics of the crankshaft is a system of differential equations (3-9). These are the equations of second order, whose results may be interpreted as vibrations of the crankshaft. For further analysis of the equations, there was found a number of numerical solutions. There were assumed different conditions of simulation i.e. without the coupling, the crankshaft without a damper and torsional vibration dampers, a damper fixed on the shaft.

Figure 7 and 8 present spectra of displacements of vibrations of the crankshaft with the omitted bending-torsional coupling.

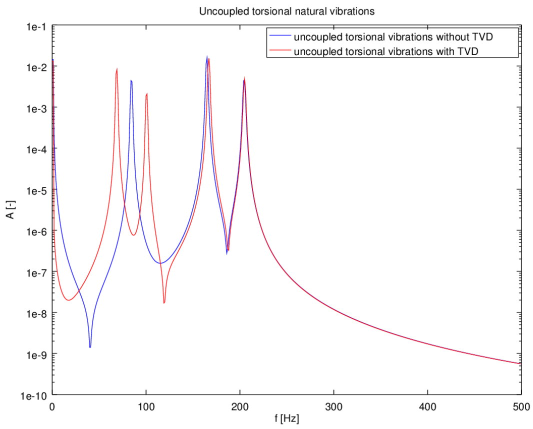

Fig. 7. The spectrum of displacements of torsional displacement of vibrations of the model of the crankshaft without the coupling

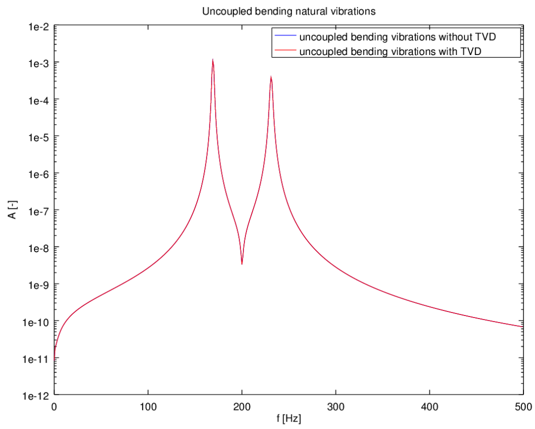

Fig. 8. The spectrum of natural of bending displacement of vibrations of the model of the crankshaft without the coupling

The use of torsional vibration damper has an influence on the frequency structure of the entire system. If the coupling and eliminator are taken into consideration, the changes in amplitude are also observed in bending oscillations. Spectra of bending and torsional vibrations in the case when the damper with one degree of freedom is used and in the case when it is not used are presented in Fig. 9 and 10.

Based on spectra (Fig. 7-10) it is clear that the results of numerical simulation correspond to structure of vibrations of piston combustion engine. This is confirmed by the research on the vibrations of engine systems carried out in Vibroacoutic Laboratory. Qualitative consistency of results obtained by the simulation and experiments confirms the usefulness of a proposed model and purposefulness of further work.

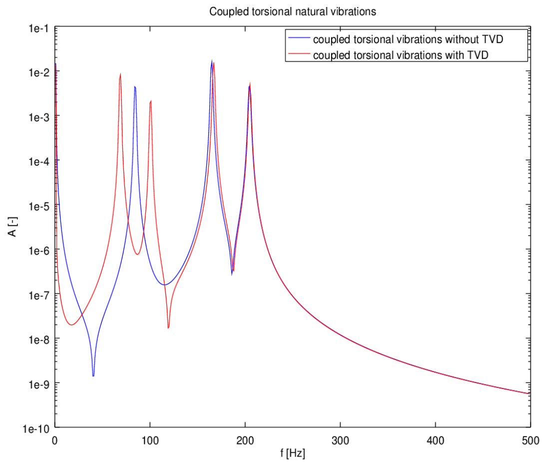

Fig. 9. The spectrum of natural displacements of torsional vibrations of the model of the crankshaft including the coupling

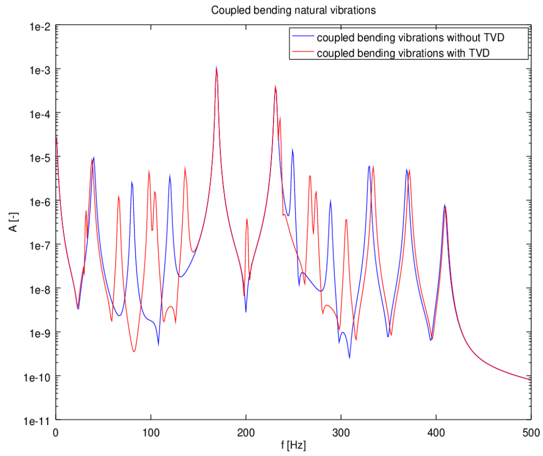

Fig. 10. The spectrum of natural displacements of torsional vibrations of the model of the crankshaft including the coupling

7. Conclusion

The model proposed in the article takes into account both geometry and load corresponding to the real drive systems of this type. Moreover, there was assumed the possibility to use a damper of torsional vibrations in order to reduce vibroactivity of the engine.

As the results of simulation presented in Fig. 7 and 10 show, the proposed model is sensitive to the parameters of the torsional vibration damper, remaining relatively simple. Thus, it is possible to make analysis of the influence of application of the vibration damper on vibroactivity of the crankshaft of the engine. Previous work carried out in the Laboratory of Vibroacoustics show that this model after the identification could be successfully used to describe the dynamics of the drive system [13, 19].

In addition, it can be said that the results of numerical simulations correspond qualitatively to the vibration structure, which leads to further work on the proposed solution.

The authors plan to use the identified model to reduce vibroactivity of currently produced Fiat TwinAir two-cylinder combustion engine by applying a torsional vibration damper. They also plan to make some changes in additional equipment of the engine [3].

References

[1] Batko, W., Dabrowski, Z., Kiciński, J., *Nonlinear Effects in Technical Diagnostics*, ITE-PIB, Radom 2008.

[2] Burdzik, R., Konieczny, Ł., *Research on structure, propagation and exposure to general vibration in passenger car for different damping parameters*, Journal of Vibroengineering, Vol. 15, Is. 4, pp. 1680-1688, 2013.

[3] Burdzik, R., Konieczny, Ł., Stanik, Z., Folęga, P., Smalcerz, A., Lisiecki, A., *Analysis of impact of chosen parameters on the wear of camshaft,* Archives of Metallurgy and Materials, Vol. 59, No. 3, pp. 957-963, 2014.

[4] Burdzik, R., *Monitoring system of vibration propagation in vehicles and method of analyzing vibration modes*, in: Mikulski J. (ed.), Telematics in the Transport Environment, CCIS 329, pp. 406-413, Springer, Heidelberg 2012.

[5] Burdzik, R., *Research on the influence of engine rotational speed to the vibration penetration into the driver via feet – Multidimensional analysis*, Journal of Vibroengineering, Vol. 15, Is. 4, pp. 2114-2123, 2013.

[6] Charles, P., Sinha, J.K., Gu, F., Lidstone, L., Ball, A.D., *Detecting the crankshaft torsional vibration of diesel engines for combustion related diagnosis,* Journal of Sound and Vibration, Vol. 321, pp. 1171-1185, 2009.
[7] Chiliński, B., *Analysis of disturbance torque influence on critical state in rotational systems*, Transportation Problems, Vol. 8, Is. 4, pp. 137-146, 2013.
[8] Chiliński, B., Pakowski, R., *Analysis of bending and torsional vibroactions of rotors with using perturbation methods*, in: Kleiber M., et al., (Ed.), 3rd Polish Congress of Mechanics and 21st International Conference on Computer Methods in Mechanics – PCM-CMM, Short papers, Vol. 1, 2, pp. 15-17, Polish Society of Theoretical and Applied Mechanics, 2015.
[9] Chiliński, B., Zawisza, M., *Modelling of lateral-torsional vibrations of the crank system with a damper of vibrations,* Vibroengineering Procedia, Vol. 6, No. 6, pp. 61-65, 2015.
[10] Czech, P., Wojnar, G., Burdzik, R., Konieczny, Ł., Warczek, J., *Application of the discrete wavelet transform and probabilistic neural networks in IC engine fault diagnostics*, Journal of Vibroengineering, Vol. 16, Is. 4, 2014, 1619-1639.
[11] Dabrowski, Z., Zawisza, M., *Investigations of the vibroacoustic signals mechanical defects sensitivity is not recognized by the OBD system in diesel engines*, Solid State Phenomena, Vol. 180, pp. 194-199, 2012.
[12] Dąbrowski, Z., Chiliński, B., *Identification of a model of crank shaft with a damper of torsional vibrations*, Vibroengineering Procedia, Vol. 6, No. 6, pp. 50-54, 2015.
[13] Dąbrowski, Z., Dziurdź, J., *Comparative analysis of torsional vibrations of the crankshaft and transverse vibrations of motor body*, Logistyka 6, pp. 2965-2972, 2014.
[14] Dąbrowski, Z., *Machine Shafts*, PWN, Warsaw 1999.
[15] Dąbrowski, Z., Madej, H*., Masking mechanical damages in the modern control systems of combustion engines*, Journal of KONES Powertrain and Transport, Vol. 13, No. 3, pp. 53-60, 2006, in Polish with an abstract in English, Retrieved June 19, 2013.
[16] Dąbrowski, Z., Zawisza, M., *Diagnostics of mechanical defects not recognised by the OBD system in self-ignition engines*, Combustion Engines – Silniki Spalinowe 3, 146, 2011.
[17] Desbazeille, M., Randall, R.B., Guillet, F., El Badaoui, M., Hoisnard, C., *Model-based diagnosis of large diesel engines based on angular speed variations of the crankshaft*, Mechanical Systems and Signal Processing, Vol. 24, pp. 1529-1541, 2010.
[18] Deuszkiewicz, P., Pankiewicz, J., Dziurdź, J. Zawisza, M., *Modeling of powertrain system dynamic behavior with torsional vibration damper*, Advanced Materials Research, Vol. 1036, pp. 586-591, 2014.
[19] Grządziela, A., *Modeling of propeller shaft dynamics at pulse load*, Polish Maritime Researches, Vol. 15, No. 4, pp. 52-58, 2008.
[20] Homik, W., *Broadband Torsional Dampers*, Wydawnictwo Naukowe Instytutu Technologii Eksploatacji – PIB, Rzeszów, 2012, in Polish.
[21] Homik, W., Pankiewicz, J., *Examinations of torsional vibration dampers used in reciprocating internal combustion engines*, Polish Journal of Environmental Studies, Vol. 20, No. 5A, pp. 108-111, Olsztyn 2011.
[22] Klekot, G., *Application of vibro-acoustic energy propagation measures to monitor status of the object and as a tool in the manage of noise*, ITE, Radom 2012.
[23] Peruń, G., Warczek, J., Burdzik, R., Łazarz, B., *Simulation and laboratory studies on the influence of selected engineering and operational parameters on gear transmission vibroactivity*, Key Engineering Materials, Vol. 588, pp. 266-275, 2014.