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ANALYSIS OF BENDING-TORSIONAL VIBRATIONS FOR DIAGNOSTICS OF THE CRANK SYSTEM

Rafał BURDZIK*, Jarosław PANKIEWICZ**, Mariusz WĄDOŁOWSKI**
*Silesian University of Technology, Poland
Krasińskiego 8, 40-019 Katowice, e-mail: rafal.burdzik@polsl.pl
** Warsaw University of Technology, Poland
Narbutta 84, 02-524 Warsaw, e-mail: jaroslaw.pankiewicz@simr.pw.edu.pl; mariusz.wadolowski@simr.pw.edu.pl,

Summary
The article presents the model of the crankshaft taking into account the coupling of bending and torsional vibrations. In engineering practice the simplification which is omitting this phenomenon is commonly used. Authors using more complicated model show the influence of bending torsional coupling on the frequency structure of the crank system.At the beginning of the article the problem of vibrations in combustion engine is generally discussed. Then there is presented the analysis of forces distribution in a crank system with the assumption that there is no friction in bearing nodes. Further chapters present two approaches to modeling torsional and bending vibrations in crank systems and indicate the application area of the proposed methodology. In the last but one chapter there were discussed the results of numerical simulations of the proposed model of the crank system. The whole was summarized with synthetic conclusions and plans regarding further research.

Keywords: torsional vibrations of the crankshaft, modelling crank system, analitycal solutions, numerical simulations

ANALIZA DRGAŃ GIĘTNO-SKRĘTNYCH DLA POTRZEB DIAGNOSTYKI UKŁADU KORBOWEGO

Streszczenie
Artykuł przedstawia model wału korbowego uwzględniający sprzężenie drgań giętnych i skrętnych. W praktyce inżynierskiej powszechnie stosuje się uproszczenie polegające na pominięciu tego zjawiska. Autorzy wykorzystując bardziej rozbudowany model, pokazują wpływ sprzężenia giętno-skrętnego na strukturę częstotliwościową układu korbowego. Na wstępie artykułu ogólnie omówiono zagadnienie drgań w silnikach spalinowych. Następnie przedstawiono analizę rozkładu sił w układzie korbowym przy założeniu braku tarcia w węzłach łożyskowych. Dalsze rozdziały prezentują dwa podejścia do modelowania drgań skrętnych i giętnych w układach korbowych oraz wskazują na obszar zastosowań proponowanej metodyki. W przedostatnim rozdziale omówiono wyniki symulacji numerycznych zaproponowanego modelu układu korbowego. Całość podsumowano syntetycznymi wnioskami oraz planami odnośnie dalszych badań.

Słowa kluczowe: drgania skrętne wału korbowego, modelowanie wału ,rozwiązania analityczne, symulacje numeryczne

1. INTRODUCTION

Crank piston engines are the basis for modern technique. Most commonly used devices are driven by them. A multitude of solutions used in practical applications means that they form a very extensive field of knowledge. Despite many applications and types of construction, the crank system is a common element. The purpose of this mechanism is to change the translational motion into rotational motion of the shaft.

The dynamics of the crankshaft with connecting rods and pistons is a very important engineering problem. The reliability of the engine is directly related to working of this basic system. Whereas crankshaft working in conditions of a constant rotational speed is a well known problem, the dynamics in transient motion, is a problem which requires a lot of research.

From the operational point of view, vibrations in crank systems are very important. Due to the complex geometry of rotor and "complex" construction of the crank mechanism, there is coupling between transverse and torsional displacements . As a result, torsional and bending vibrations occur simultaneously in crank systems. Unfortunately, due to their complex nature, it is often necessary to omit this coupling.

This approach, of course, significantly simplifies the problem, but it is the cause of changes in spectral structure of the model of piston engine. In many situations a simplified modeling is desirable due to easy calculations e.g. in the case of engine design. However, the final calculations should take into account the phenomenon of coupling of vibrations in crank systems . This problem is important due to the fact that together with the coupling of bending-torsional vibrations there may occur new critical areas. Moreover, torsional vibrations affect transverse vibrations. Therefore, there can be observed a shift of particular frequencies of natural vibrations of uncoupled system. What is more, there are additional effects having non-linear or parametric character.

In practical applications, this phenomenon can be used to analyze torsional vibrations of the engine on the basis of its transverse displacements. This problem is so important that many of faults that may occur in combustion engines, significantly affect the angular vibrations of the drive system.

Thus, it is possible to use bending-torsional couplings of vibrations to analyse and assess technical condition of the device equipped with piston combustion engine. For this purpose, it is necessary to make a detailed analysis of the equations governing the dynamics of the crank system (equations of motion) and the results of simulations and experiments. The research conducted so far shows a clear relationship between the particular spectra. This demonstrates the possibility of using bending-torsional coupling of vibrations for diagnostic purposes 1.

2. FORCES IN A CRANK SYSTEM OF THE COMBUSTION ENGINE

The variable force which comes from the pressure of the combusted fuel has an effect on the crank system. This force is transmitted to the crank of the shaft via a connecting rod connected to the piston. The course of pressure is shown in the indicator diagram Fig 1 and 2.

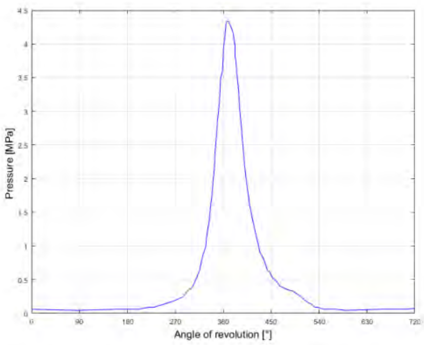

Fig.1. Indicator diagram.4

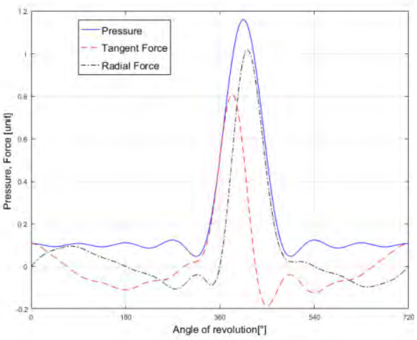

Fig.2. Expanded indicator diagram of the studied engine.

The kinematics of such a system is quite complex, the piston makes reciprocating motion, the connecting rod makes a complex movement, and the shaft makes a rotational movements. (Fig. 3). To calculate the torque acting on the shaft it is important to know the forces acting in particular elements depending on the angle of its rotation.

Knowing the instantaneous values of the pressure in the cylinder and dimensions of the piston it is possible to compute the force acting on the connecting rod. Assuming masses of individual elements and assuming the rotational speed of the crankshaft it is possible to calculate the inertia forces in the system. In practice, the inertia forces are partly balanced. These unbalanced forces cause additional variable torque acting on the crankshaft.

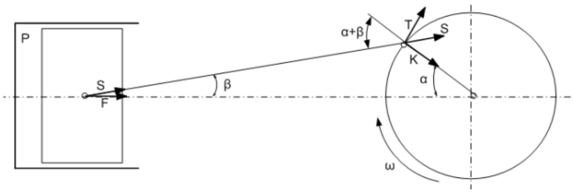

Fig.3. The diagram presenting the distribution of forces in the crank-piston mechanism.

where:

$$S = \frac{F}{cos\beta} \tag{1}$$

$$T = S \cdot \sin(\alpha + \beta) \tag{2}$$

$$K = S \cdot \cos(\alpha + \beta) \tag{3}$$

For further numerical analysis there was omitted a variable moment of inertia and there was modelled only the pressure course, from which the forces were calculated (1) - (3). The courses of forces in the system are presented in Fig.2.

3. THE CONVENTIONAL MODEL OF THE CRANKSHAFT OF ONE-CYLINDER ENGINE

There was assumed the model with three degrees of freedom (Fig. 4) where in the crankshaft, flywheel and pulley are reduced to the mass moments of inertia equal respectively I_1, I_2, I_3, interconnecting the spring elements with stiffness *k*. The crankshaft is loaded with torque *M1* which results from the kinematics of the system and modelled variable of gas force. This model allows to designate the changes of the angle of rotation of the shaft, which correspond with the frequency structure to the gas force. Only rotational speed and torsional vibrations connected with it are considered in the model.8

$$I_1\ddot{\varphi}_1 + k(\varphi_1 - \varphi_2) = 0 \tag{4}$$

$$I_2\ddot{\varphi}_2 + k(\varphi_2 - \varphi_1) + k(\varphi_2 - \varphi_3) = M_1 \tag{5}$$

$$I_3\ddot{\varphi}_3 + k(\varphi_3 - \varphi_2) = 0 \qquad (6)$$

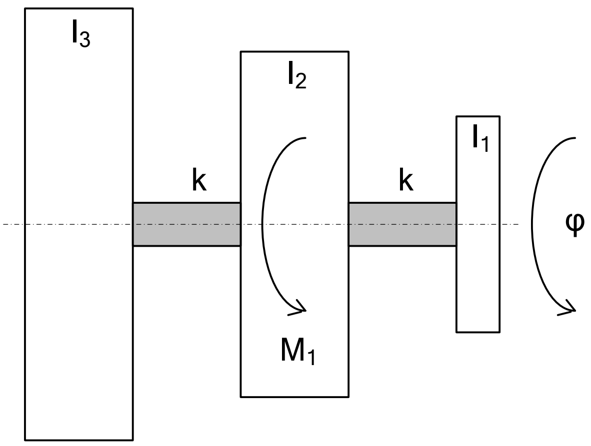

Fig. 4. Shaft model.

Calculations done for the speed of 3000 RPM. In the presented spectrum of vibration acceleration there are visible frequencies of natural vibrations and the frequencies of vibrations coming from rotational speed (Fig. 5).

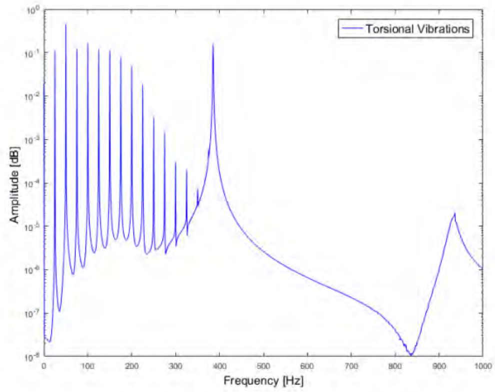

Fig. 5. The amplitude of torsional vibration acceleration of the model

4. THE DYNAMIC MODEL OF PISTON ENGINE WITH AN ELASTIC CRANKSHAFT

The crankshaft of the combustion engine is a continuous system. This means that at any point, it is possible to define physical parameters such as density, stiffness tensor, etc. These quantities depend mainly on the materials from which the object is made, and they directly influence the dynamics of the whole system. Using the dependencies known from the mechanics of continuous system, it is possible to write equations describing vibrations of such an object. Unfortunately, such a model would be very complicated, what is more, in practice, it would be very difficult to solve by methods other than numerical ones. Therefore, it is possible to propose a simplification which is the replacement of the continous system with the set of material points. Such inert elements arranged in arbitrarily chosen structural nodes are connected via weightless deformable elements representing elastic structure of the crankshaft.

Model of the system of point masses is a significant simplification of the continuous system. Even the simplest analysis of the continuous system proves that they are characterized by infinite set (although countable) of eigenvalues. In contrast, discrete systems have a number of free vibration frequencies equal to the number of degrees of freedom. In practice, however, high frequencies are strongly damped. Therefore, the spectral structures of the continuous system and a substitute model with point masses are very similar. This means that this simplification does not cause any serious errors. At the same time it must be emphasized that this method significantly simplifies conducted computations.

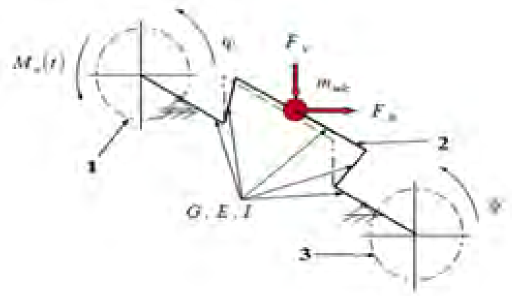

Fig.6. Model of the cranshaft

A fragment of the crankshaft of the piston engine is shown schematically in Fig. 6.

Crank mechanisms need to be very precise. Even small changes in the angular position of the crank can have an impact on the dynamics of the combustion process. Therefore, it is necessary to design very rigid crankshafts. In addition, in vibrating systems there is a risk of resonance with a basic harmonic of extortion which comes from gas forces. Of course, there is no practical possibility of complete avoidance of engine work in critical states. The oversize of the drive system allows only to make shift of the frequencies of natural vibrations into the area of higher harmonic elements of the torque.

As already noted, deformations in crank systems are very small. This allows for the calculations of the linear-elastic model. Fig. 7 shows the displacement ur of the crank described in the moving coordinate system.

In fig. 6 and 7 there were used the following generalized coordinates describing the dynamics of the analyzed model of the crank:

ϕ- rotation angle of the flywheel of the engine,
φ- rotation angle of the disc of the torsional vibration damper,
h-horizontal deformation of the crank,
v- vertical deformation of the crank.

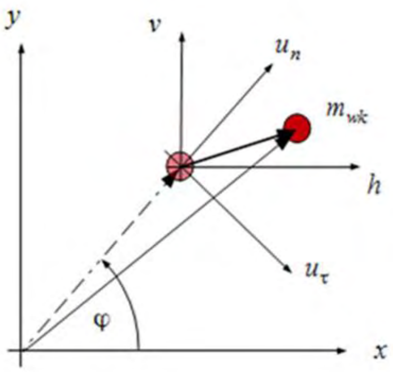

Fig.7. Displacment of crankshaft crank.

Generalized forces of certain structural nodes in the discussed problem may be determined on the basis of:

$$F=K\cdot u. \tag{7}$$

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

$$u=\begin{bmatrix} u_n \\ u_\tau \\ \varphi_L-\varphi_P \end{bmatrix}, \quad \theta=\varphi_L-\varphi_P$$

$u_n -$ radial deformation of the crank,
$u_\tau -$ tangential deformation of the crank,
$\theta -$ difference between rotation angle

Due to the symmetry conditions and the load system, the stiffness matrix has a simplified form:

$$K=\begin{bmatrix} k_{nn} & 0 & 0 \\ 0 & k_{\tau\tau} & -k_{\tau\theta} \\ 0 & -k_{\tau\theta} & k_{\theta\theta} \end{bmatrix} \tag{8}$$

It is possible to find motion equation for the system presented in Fig. 6 with the use of any formalism of analytical mechanics. Due to the linearity of the model and the presence of only holonomic constraints, the Lagrange equations of the second kind were used. On this basis, the following dynamic model is determined:

$$I_{kl}\ddot{\varphi}_L+m_{wk}R^2\ddot{\varphi}_L+m_wR\ddot{u}_\tau+k_{\tau\theta}u_\tau+k_{\theta\theta}(\varphi_L-\varphi_P)=M_0 \tag{9}$$

$$m_w\ddot{u_\varphi}+m_wR\ddot{\varphi}_L+k_{\tau\tau}u_\tau-k_{\tau\theta}(\varphi_P-\varphi_L)=P_\tau, \tag{10}$$

$$I_{kP}\ddot{\varphi}_P+[-k_{\theta\tau}u_\tau+k_{\theta\theta}(\varphi_P-\varphi_L)]=0. \tag{11}$$

$$m_w\ddot{u}_r-m_wR\dot{\varphi}_L^2+k_{rr}u_r=P_r. \tag{12}$$

The analyzed system (8)-(12) is a system of four coupled and "apparently nonlinear" ordinary differential equations with constant coefficients. The fourth equation comprising one of the variables in non-linear form is the evidence of this (12) However, the remaining equations are linear and do not belong to radial displacement ur. This means that equation (8) - (11) can be solved independently, and then the expression $m_wR\dot{\varphi}_L^2$ can be treated as the right side of the equation (12) and its solution can be found. Despite the apparent complexity the considered system, in practice, can be considered as linear 2123.

Further analysis shows that the first three equations describe torsional vibrations of the system. As already mentioned, they do not depend directly on displacements in the plane of the shaft ur. This means that the application of a disc model for description of torsional vibrations is justified. It is only necessary to determine the best methodology (from the point of view of the results) of weight reduction and stiffness of the crank-connecting rod-piston system to the shaft axis 828.

5. NUMERICAL SIMULATIONS OF BENDING-TORSIONAL VIBRATIONS OF THE CRANK SYSTEM

The proposed mathematical model of the dynamics of the crank system (8) - (12) can be solved sequentially. Qualitative analysis of solutions of equations (8) - (11) (based only on the equations of motion) shows that the solution gets all frequencies coming from the gas forces (by the presence of elements of gas forces in the right sides of the equations of motion) and 3 frequencies of natural vibrations (relatedto the structure of the analyzed system).

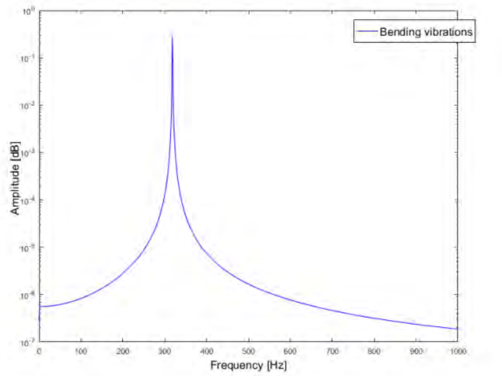

Fig. 8.Bending vibrations of the crankshaft of the system without coupling.

Solution to last equation (12) depends on the projection of gas force on the direction of crank and frequency structure of torsional vibrations. Expression $m_wR\dot{\varphi}_L^2$ from last equation depends on the square of the speed of torsional vibrations of the whole system. It means that in the spectra of simulation results of transverse vibrations one may expect frequency of torsional vibrations, their doublings and auto-modulation (due to the occurrence of the variable in the second power) and all frequencies of extortion.

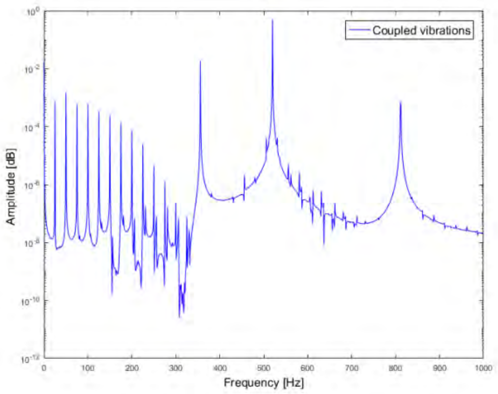

Fig. 9. Bending vibrations of the crankshaft of the system without coupling.

6. CONCLUSIONS

The proposed model of the crankshaft takes into account an important phenomenon of bending-torsional coupling of vibrations, and still remains analytically unsolvable. The presented considerations confirm the usefulness of the proposed model and the quality compliance of the results. The analysis of numerical simulations (figures 8-9) indicates that the spectral structure of the model with a coupling of vibrations is more complex than in the case of omission of this phenomenon. In addition, the main component of bending vibrations remains unchanged. This means that the proposed description of the phenomenon is consistent with an assumption of independencies of transverse and angular deformations previously used in practice. Because the mechanism of coupling of vibrations is very natural (vibrations caused by changes in the centrifugal forces of reaction) and one-sided, it is possible to use previously developed models effectively. It means that additional effects arising from bending-torsional coupling of vibrations can be considered later. This approach may be particularly convenient for design of drive systems, where preliminary calculation is performed with the use of model (4) - (6). However, further checking calculations need to be carried out including couplings of vibrations, because as it was proved in the article, they'll have more complex spectral structure.

In addition, such an approach to the problem of vibrations in the crank system is necessary in engines diagnostics. The evolution of "differences" of the frequency system is very often a useful diagnostic symptom during the observation of vibrations [83].

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Rafał BURDZIK is employed as an professor at the Silesian University of Technology, Faculty of Transport, Department of Automotive Vehicles Construction. Currently he is vice-dean for Education at this Faculty. He received PhD (2006 Faculty of Transport, Silesian University of Technology), DSc (2014 Faculty of Transport Warsaw University of Technology). His research interests are: vibration phenomena, safety and comfort in transport, machinery diagnostic, signal processing, transport, logistics and forwarding. He is member of Bureau of the Committee on Transport Katowice Branch of the Polish Academy of Sciences, Polish Societies of Technical Diagnostics and Katowice Regional Committee of Polish Maintenance Societies. He is member of few editorial boards of high level scientific journals and active reviewer of impacted journals.

Jarosław PANKIEWICZ is an assistant professor at the Institute of Machine Design Fundamentals, Faculty of Automotive and Construction Machinery Engineering, Warsaw University of Technology. In 1995 he received M.Sc. in Mechanics and Machine Design from WUT. In 2002 he received Ph.D. in Machine Design and Maintenance, also from WUT. He has authored over 30 journal and conference publications, mainly on diagnostics of machine technical condition and on vibrations and noise damping. His major fields of interest are vibroacoustics, machine dynamics and technical diagnostics.

Mariusz WĄDOŁOWSKI received Msc degree in Faculty of Automotive and Construction Machinery Engineering from Warsaw University of Technology, Poland, in 2005. Now he works at the same place. His current research interests include detection of mechanical defects in internal combustion engines.