Scientific Journals
Maritime University of Szczecin

Zeszyty Naukowe
Akademia Morska w Szczecinie

2014, 40(112) pp. 9–16
ISSN 1733-8670

Torsional vibration silencers used in vessels propulsion systems

Wojciech Homik

Rzeszow University of Technology, The Faculty of Mechanical Engineering and Aeronautics
Dept of Mechanical Engineering
35-959 Rzeszów, al. Powstańców Warszawy 12, e-mail: whomik@prz.edu.pl

Key words: crankshaft vibration, longitudinal vibration, torsional vibration of crankshaft, torsional vibration damping

Abstract
The main source of vibration in the engine piston is the work of the piston – crank, and swapping reciprocating motion to the rotary motion. In this paper are described among others the crankshaft vibration which result in the occurrence of cyclic forces such as forces pressure of gas and inertia forces. In addition were made the analysis of longitudinal vibration of the crankshaft, which are these vibrations are an important problem in high-power vessels' engines. Regardless of the dynamic system in which the engine is running the greatest threat for crankshaft are the torsional vibrations. The possibilities of vibration damping of the crankshaft in the engine also were discussed.

Introduction

The main source of vibration in the engine piston is the work of the piston – crank, and more specifically, swapping reciprocating motion to the rotary motion. As we know all solutions of reciprocating internal combustion engines regardless of the number and manner of the spatial distribution of the cylinders rotation of gain through the piston crank mechanism. Kinetic analysis of the crankshaft can be accomplished by replacing the actual arrangement diagram shown in figure 1.

The basic geometrical quantities characterizing this pattern include [1, 2, 3, 4, 5]:

- connecting rod length $l = AB$ (measured from the axis of the piston pin to the axis of crankpin);
- radius of crank throws $r = BO = S/2$;
- piston stroke S (measured between the non-return piston TDC positions and DMP);
- the ratio of the radius of the crank, the crank length $\lambda = r/l$.

Depending on the type of engine used in vessels λ is as follows:

$$\lambda \in \left(\frac{1}{4.2} \div \frac{1}{5}\right) \text{ – low-speed engine;}$$

$$\lambda \in \left(\frac{1}{3.5} \div \frac{1}{4}\right) \text{ – high-speed engine.}$$

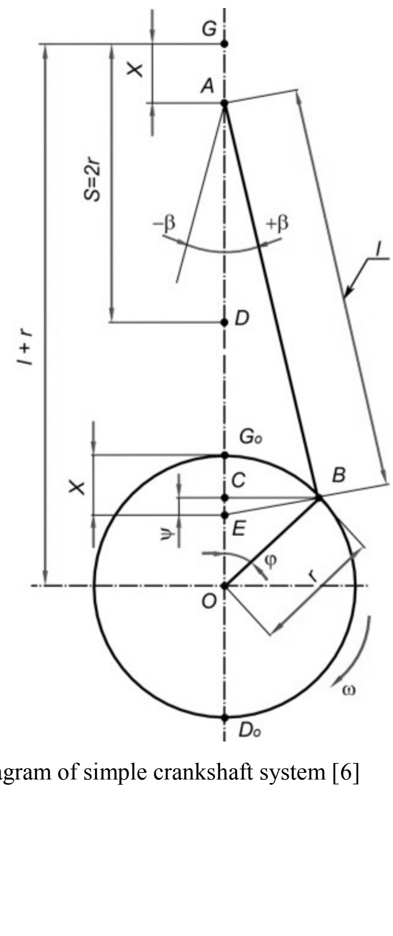

Fig. 1. Diagram of simple crankshaft system [6]

In the real system the crank motion of the piston is not exactly harmonic. The piston speed c in function of the angle of rotation of the shaft φ is shown by the formula:

$$c = r\omega\left(\sin\varphi + \frac{\lambda}{2}\sin 2\varphi\right) = r\omega\sin\varphi + r\omega\frac{\lambda}{2}\sin 2\varphi \tag{1}$$

From this formula it follows that the velocity of the piston substantially consists of first-order rate $c' = r\omega\cdot\sin\varphi$ of argument φ and speed of the second class $c'' = r\omega(\lambda/2)\cdot\sin(2\varphi)$ of argument 2φ.

Periodicity changes the piston speed makes the piston experiences both positive and negative acceleration (delays) [1, 5, 6, 7], and the largest acceleration values achieved in turning positions, i.e. while $\varphi = 0°$ and $\varphi = 180°$.

The crankshaft vibration

Working piston combustion engine is the source of vibration, which result in the occurrence of cyclic forces.

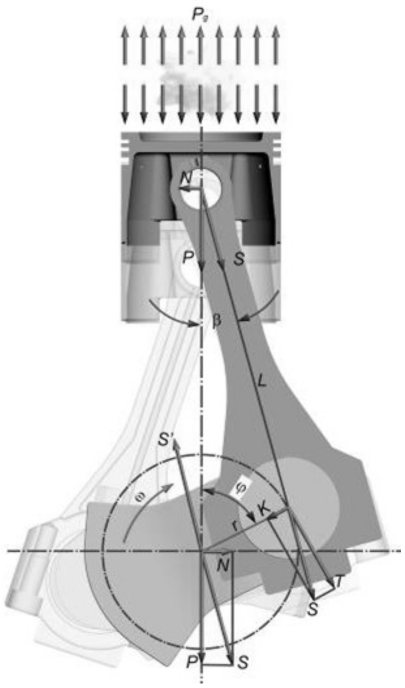

Fig. 2. Distribution of forces in the crankshaft – piston [6]

For the forces that acting on the motor crankshaft causing a vibration of the motor crankshaft include (Fig. 2) [1, 2, 6, 7, 8, 9, 10, 11, 12, 13]:

- gas pressure forces generated in the combustion process the mixture Pg;
- inertia forces originating from the masses in motion and reciprocating rotary motion (sliding force, and centrifugal force) Pb.

Periodic changes in gas pressure Pg forces and inertial forces Pp generate the following types of vibration of the crankshaft [1, 6, 12]:

- buckling vibrations;
- longitudinal vibrations;
- torsional vibrations.

Vibrations are a kind of defense factor, which have machine parts made of elastic materials, which involves giving up the applied load and absorption gradually transferred this energy in the form of vibrations. Many parts of machines including those crankshafts could be very quickly destroyed if not for their ability to absorb the energy by elastic deformations (Fig. 3).

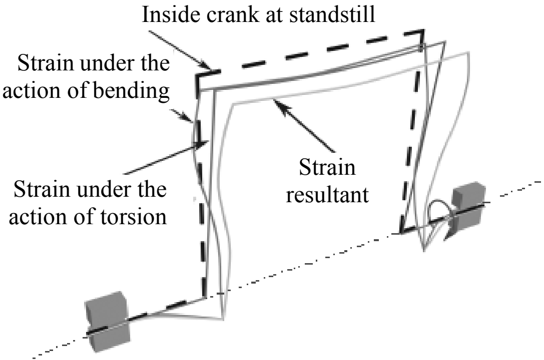

Fig. 3. Examples of the elastic deformation of the crank throw induced by the force T [13]

Engine vibration can also be the result of interference, e.g. working its ignition system. It is mentioned, for example, in studies [4, 6]. Of course, the engine is a vibration-damping element. The issues are widely described in [6, 14].

Longitudinal vibration of crankshaft

Longitudinal vibrations of the crankshaft are directly related to its buckling vibration. Any shaft deflection causes the axial displacement (Fig. 4). These vibrations in most cases do not interfere with the motors and do not constitute much of a threat to the stability of the crankshaft. This is due to the fact that the engine crankshaft has a high longitudinal stiffness, and thus the frequency of the vibrations are greater than the buckling. It should be clear that these vibrations are an important problem in high-power vessels' engines. They make the whole system (Fig. 5) composed of the engine crankshaft, flywheel, shaft lines and the propeller moves periodically along its axis. Longitudinal vibrations amplitude of the system depends practically on the design of the propeller, rather than the number of blades [5] and the damping bearing the resistance and the clutches.

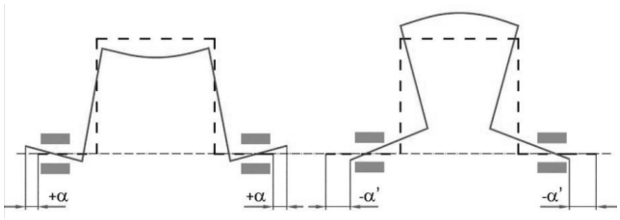

Fig. 4. The longitudinal vibration of the crankshaft [6, 13]

Torsional vibration of crankshaft

Regardless of the dynamic system in which the engine is running the greatest threat for crankshaft are the torsional vibrations [1, 2, 6, 11, 13, 15, 16]. Among a number of forces acting on the piston-crank system, the rotational motion of the crank-shaft causes the force T tangent to a circle made by the crank throw (Fig. 2).

$$M(\varphi)=T\cdot r=\frac{P}{\cos\beta}r\sin(\varphi+\beta) \tag{2}$$

where: $M(\varphi)$ – shaft torque, T – tangential force, R – the double shaft, φ – the angle of rotation of the crankshaft.

Variability of force T causes acceleration in the rotational motion of crankshaft in the engine cousing the torsional vibration that are vary with the change of shaft rotational speed. Process of tangential force T as a function of crank angle φ of the crankshaft presents mostly in the form of a graph called as a graph of tangential force (Fig. 5) [1, 2, 6].

Experience has been shown that in the harmonic analysis of a tangential force T is sufficient to

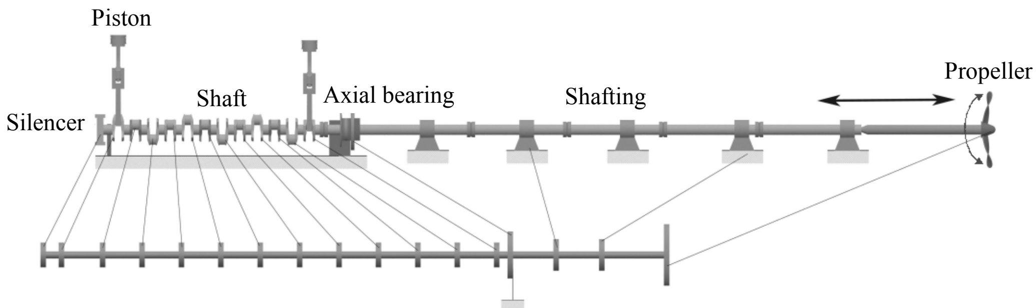

Fig. 5. The actual and replacement the vessels' drive system [9, 15]

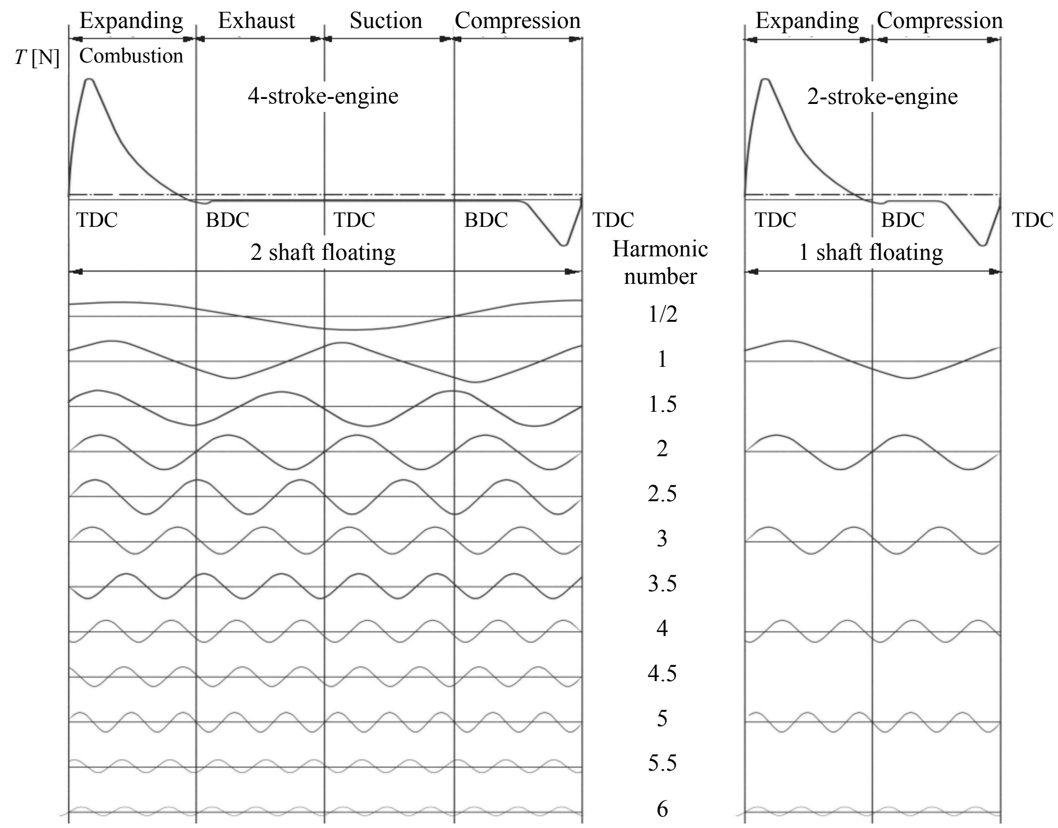

Fig. 6. Process of tangential force T and its successive harmonics in four- and two-stroke engine [5, 9]

designate only a certain number of harmonics K. Generally, it is about 12–18 (first) harmonic, because higher harmonics of high frequency and small amplitudes do not significantly affect to the torsional vibrations [2, 5, 6, 12]. It is worth emphasizing that the crankshaft's torsional vibrations are only limited by torsional stiffness of the shaft, and the amplitude of torsional vibrations exceed the limit values. In the absence of suppression of the amplitude of vibration tends to the infinity theoretically, for each rotational speed equal to another harmonic. Destruction (twisted) shaft with variable stiffness, and such is the crankshaft, occurs at the moment when the limit value is exceeded φ_{dop} amplitude (maximum torsion angle) [6]:

$$\varphi_{\mathrm{rz}} > \varphi_{\mathrm{dop}}$$

$$\varphi_{\mathrm{dop}} = M_s \sum_{i=1}^{n} \frac{L_i}{G I_{oi}} \tag{3}$$

where: M_s – torque, L_i – reduced length of the shaft, G – torsional Modulus G, I_{oi} – polar moment of inertia.

Admissible value φ_{dop} maximum torsion angle depends on the machine and set the overall tolerance of the geometrical parameters. The concept of unit angle of torsion is often using and easy to compare [6]:

$$\varphi' = \frac{\varphi}{l} = \frac{M_S}{GI_0} \quad [\mathrm{m}^{-1}] \tag{4}$$

For steel shafts with a load unilaterally of variables:

$$\varphi'_{\mathrm{dop}} = 0.004 \quad [\mathrm{m}^{-1}] \tag{4a}$$

and the loadings of variables on both sides:

$$\varphi'_{\mathrm{dop}} = 0.0025 \quad [\mathrm{m}^{-1}] \tag{4b}$$

Frequently torsion angles of the limit values are in the range:

$$\varphi'_{\mathrm{dop}} = (0.002 \div 0.01) \quad [\mathrm{m}^{-1}] \tag{4c}$$

The propulsion system of vessels additional source of torsional vibration is having a large waterlessness, mounted on the free end of the propeller shaft line (Fig. 5, Photo. 1) [6]. The moment in which followed the excitation of vibration coming from the propeller describes the relationship:

$$M_{ws} = 0.12(0.1 I_{ss} + I_{sw}) \frac{\omega_s^2}{h} \tag{5}$$

where: I_{ss} – mass moment of inertia of the screw; I_{sw} – mass moment of inertia of the screw absorbed by the water; ω_s – angular velocity of the screw; h – order of harmonic compatible with a multiplicity of propeller blades [17].

Keep in mind that the screw immersed in water is also damped. This issue is very important for the operation of the drive system of vessels and have reached a number of studies [6, 14, 18].

Torsional vibration damping

Torsional vibrations of crankshaft the engine are more difficult to detect than other vibrations. Imposed on the rotation of the shaft usually does not cause major backling vibration of neighboring parts, they are not a source of noise, and therefore may not be seen until the moment in which occurs the shaft's fatigue cracking. Their existence can often indicate the lack of uniformity engine's work, which timing system using a mechanical transmission (belt, chain, gear) is driven by the crankshaft torsional vibrating [1, 12]. The variety of modes of vibration and the polyharmonic nature of the tangential force T which forcing the vibration cause that the crankshaft can work in the area of resonance at different engine speeds.

In simple cases it is sufficient to take into account the first harmonic but high susceptibility of modern structures and impact of the propulsion system may make it necessary to take into account the higher harmonics.

In multi-cylinder engine, each family of harmonic excited by a single cylinder is applied to the harmonic excited by the other cylinders. Thus, the harmonics of the order h may be in phase. There are then so-called harmonic "strengthened" called harmonic major.

For the engine, in which the ignitions occur at equal intervals, the most dangerous are the critical rotational speeds at which the magnitude "h" harmonic "k" represents the number of ignitions per one rotation of crankshaft, so two-stroke engine – a multiple number of cylinders and engines four-stroke – half a multiple of the cylinder number [5].

Engine operation in the fields of critical (resonance) rotation speed can be avoided by:

- changing speeds;
- changing the natural frequencies of the whole system;
- change the course of excitation forces;
- use of dampers (eliminators) vibrations.

In most cases, the first three solutions may be impossible to implement in view of construction – consumables, and therefore apply torsional vibration dampers (TVDs) (eliminators), which most often placed at the free end of the engine crankshaft

(Fig. 5, Photo. 1). Their mission is to decrease the amplitude of torsional vibration of the engine crankshaft.

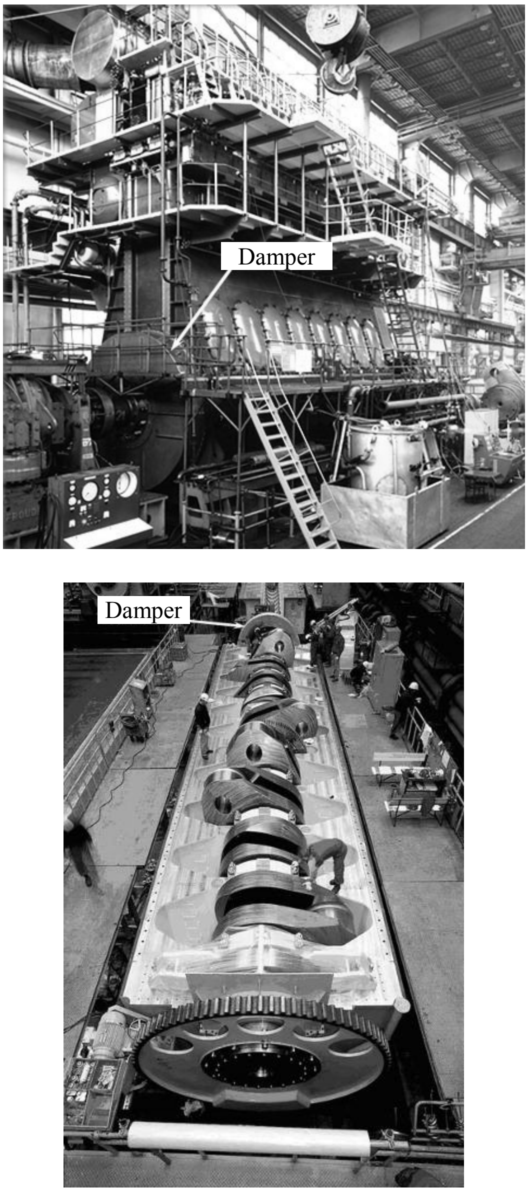

Photo. 1. Examples of the location of torsional vibrati on silencer on the engine crankshaft

Properly designed (selected, “tuned”) torsional vibration damper can reduce the resonance amplitude torsional vibration as much as 10-fold as well as shift and reduce the resonance zone. However, that each damper absorbs the output power of the engine [6, 15, 16].

Types of torsional vibration silencer

In practice, the commonly used terms “silencer” regardless of structures from the viewpoint of mechanics. Over the years, in order to minimize the risks derived from the torsional vibration were applied following types of silencer [6, 15, 16]:

- frictional;
- viscous (Fig. 3);
- rubber (Fig. 4);
- coupling (Fig. 5).

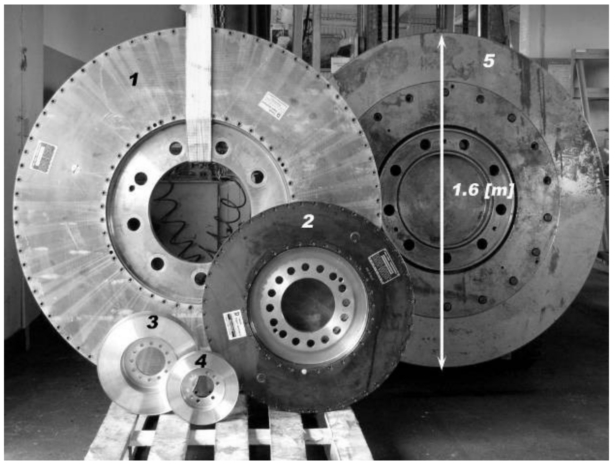

Photo. 2. Torsional vibration dampers: 1, 2, 3, 4 – viscous dampers torsion, 5 – coupling torsional vibration damper

These dumpers are typically tuned tensional vibration damper, in which the reduction of torsional vibration is used the inertial forces. Despite the common name, the dynamic dampers are very each other not only design solution but above all characteristic [6]. Currently, in propulsion system of vessels are used in practically three types of dampers:

- viscous (Fig. 3);
- rubber (Fig. 4);
- coupling (Fig. 5).

Noteworthy is the fact that it is also carried out research on a new generation of viscous dampers called active dampers where physical characteristics change with the change of extortion.

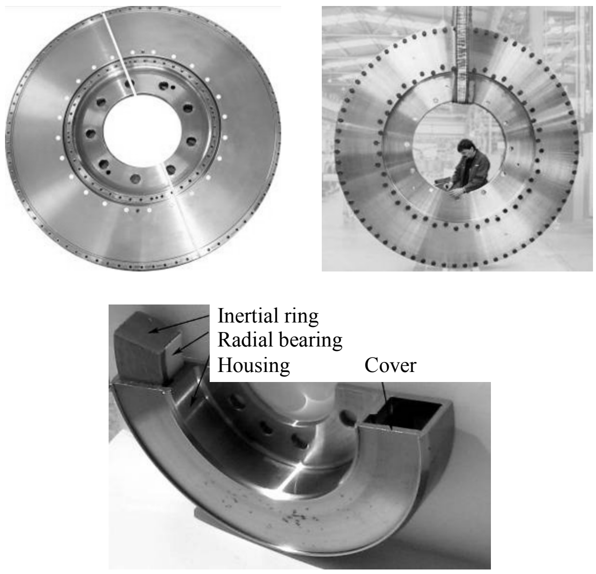

Photo. 3. Viscous torsional vibration dampers

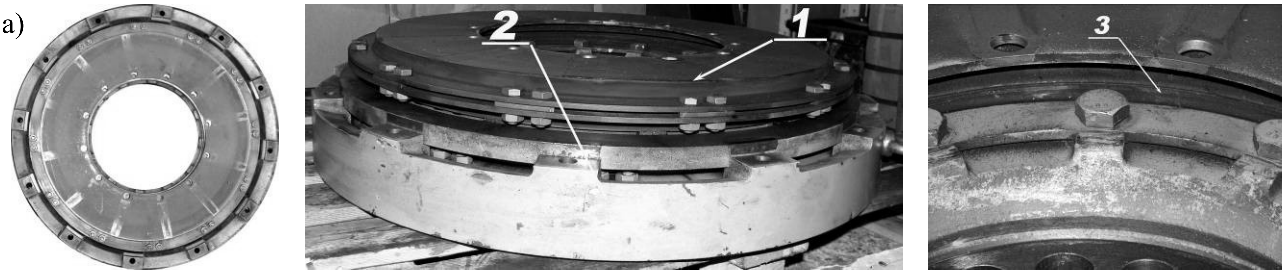

Photo. 4. Rubber torsion damper: a) car muffler, 1 – ring inertia, 2 – hub, 3 – rubber damping element

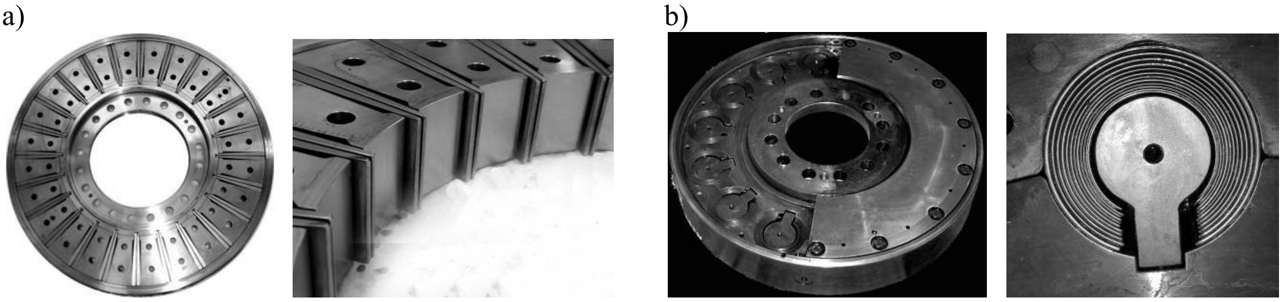

Photo. 5. Spring loaded torsional vibration dampers: a) spring damper company Geislinger (MAN), b) spring torsional vibration damper with double torsion springs package company Pielstick

As already mentioned, the greatest threat to the engine shaft is work in the speed range in which there is a strong resonance (Fig. 7, 8).

It seems that the installation on the free end of the shaft torsional vibration damper, significantly reduces its resonance vibration, not only in that area a "strong" resonance, but practically the whole speed range of engine operating. Prove the truth of this assertion is not only the results of theoretical research but most of all, the results of real objects (Fig. 10, 11).

Viscous damper, even if the damping is far from optimal damping reduces vibration of the engine crankshaft to a safe value. It could be argued that the viscous dampers well dampen torsional vibrations throughout the range of speed of rotation (Fig. 7, 10). It is obvious that the highest efficiency of the torsional vibration damping is characterized by the muffler, where $\alpha = \alpha_{opt}$. It must be remembered that this proposition is true, if the damper reaches the saturation temperature.

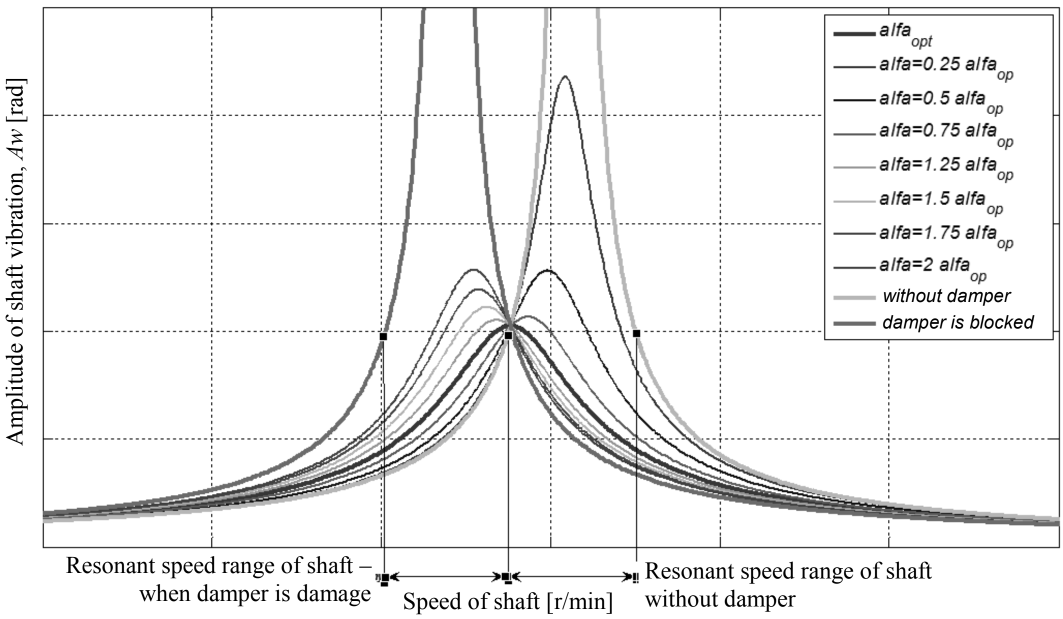

Fig. 7. Characteristics of shafts amplitude *Aw* on which is mounted a viscous damper – damping α (alfa = α)

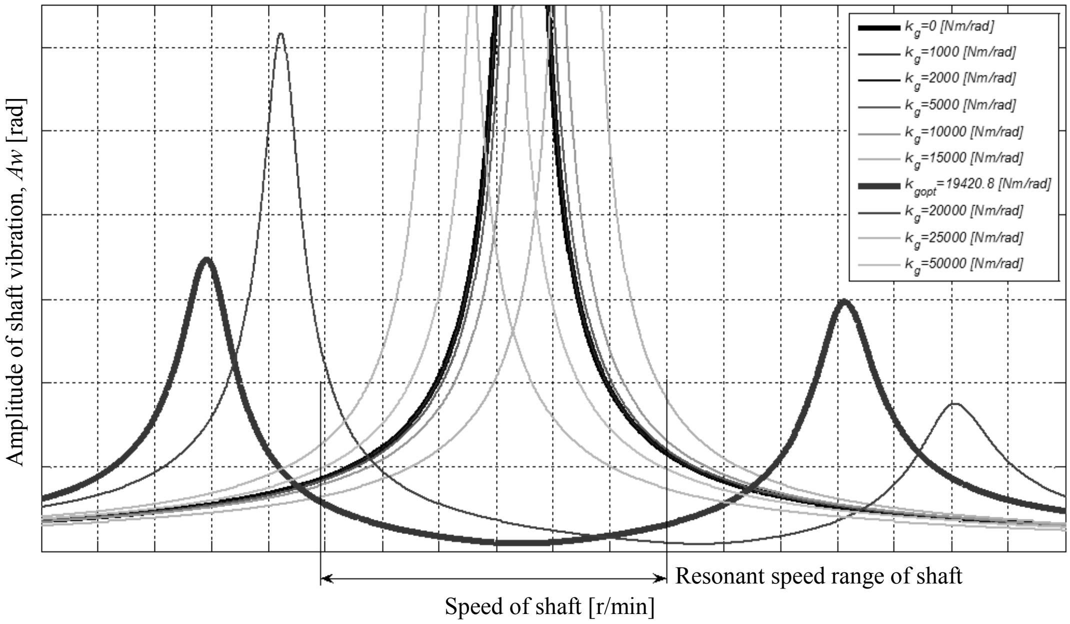

Fig. 8. characteristics of shaft's amplitude A_w on which is mounted rubber damper (k_g – dynamic stiffness)

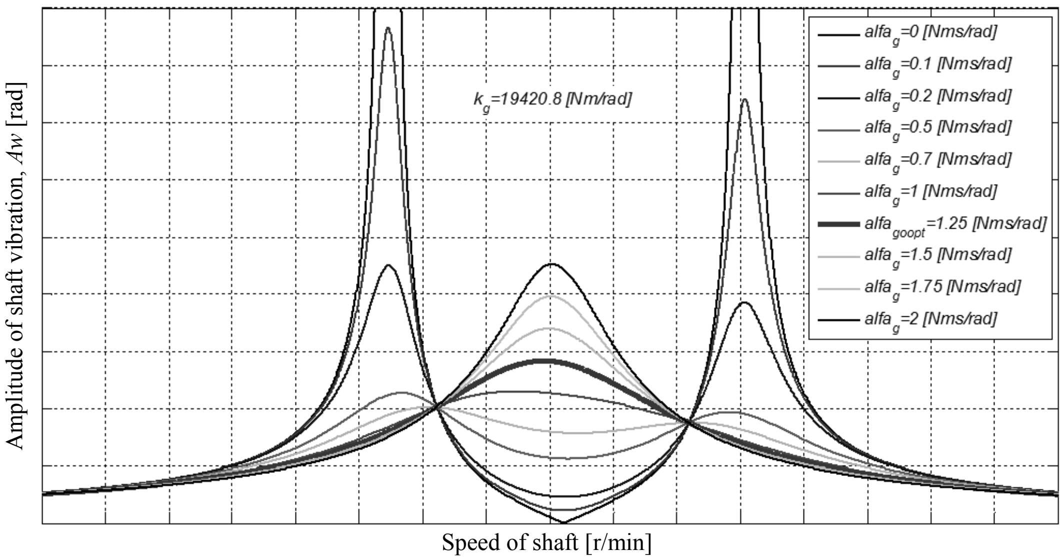

Fig. 9. characteristics of shaft's amplitude A_w on which is mounted rubber damper (alpha Attenuation inner Rubber)

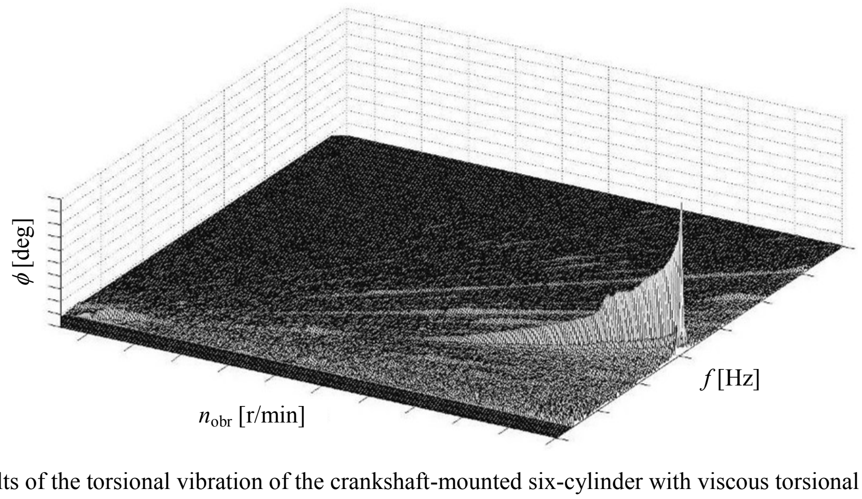

Fig. 10. Sample results of the torsional vibration of the crankshaft-mounted six-cylinder with viscous torsional vibration damper

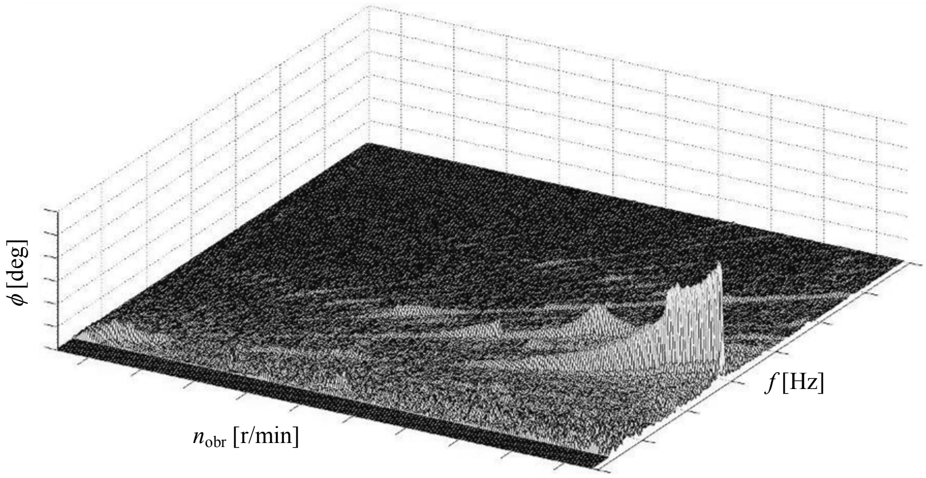

Fig. 11. Sample results of the torsional vibration of the crankshaft-mounted six-cylinder with rubber torsional vibration damper

The use of the rubber damper vibration causes a marked reduction in the resonance zone (Fig. 8, 9, 11), however, there are two zones resonant speeds respectively larger and smaller in relation to the critical speed of the shaft without an attenuator. However, the vibrations in these areas are considerably smaller. The results of numerical calculations and experimental studies also show that the overall level of vibration, and especially the vibration in the resonance zone is influenced not only by the dynamic stiffness of rubber k_g, but also the internal damping of rubber α_g. Analysis of the results shows that both for the dynamic stiffness of rubber k_g and for internal damping rubber α_g, there are optimum values k_{gopt} i α_{gopt}, for which the maximum amplitude of the resonance of the shaft reaches a minimum (Fig. 8, 9).

References

1. Brun R.: Szybkobieżne silniki wysokoprężne. WKiŁ, Warszawa 1973.
2. Jędrzejowski J.: Mechanika układów korbowych silników samochodowych. WKiŁ, Warszawa 1986.
3. Leyko J.: Dynamika układów materialnych. PWN, cz. I Warszawa 1959, cz. II Warszawa 1961.
4. Leyko J.: Mechanika ogólna. Warszawa 1976.
5. Niewiarowski K.: Tłokowe silniki spalinowe. WKiŁ, Warszawa 1968.
6. Homik W.: Szerokopasmowe tłumiki drgań skrętnych. Wydawnictwa Naukowe Instytutu Technologii Eksploatacji – PIB, Radom 2012.
7. Bernhardt M., Dobrzyński S., Loth E.: Silniki samochodowe. WKiŁ, Warszawa 1969.
8. Bernhardt M.: Badania trakcyjnych silników spalinowych. WKiŁ, Warszwa 1970.
9. Bernhardt M.: Drgania skrętne wałów korbowych. Nowa metoda określania częstości drgań własnych. Technika Motoryzacyjna 8, 1966.
10. Bernhardt M., Szczeciński S.: Obciążenia wałów korbowych. Przegląd mechaniczny 14, 1966.
11. Goliński J.: Wibroizolacja maszyn i urządzeń. WNT, Warszawa 1979.
12. Wajand J.A, Wajand J.T.: Tłokowe silniki spalinowe średnio i szybkoobrotowe. WNT, Warszawa 1993.
13. Werner J., Wajand A.: Silniki spalinowe małej i średniej mocy. WNT, Warszawa 1971.
14. Maciotta R., Saija Merlino F.: Research on damping of torsional vibrations in the Dieselengined propelling plants. FIAT Technical Bulletin 2, 1966.
15. Homik W.: Diagnostyka, serwisowanie i regeneracja tłumików drgań skrytych wałów korbowych okrętowych silników spalinowych. Polish Maritime Reserch 1, 2010.
16. Homik W.: Damping of torsional vitration oh ship engine crankshafts – general selection methods of viscous vibration damper. Polish Maritime Reserch 3(70), 18, 2011.
17. Zygmuntowicz J.: Metodyka obliczeń drgań skrętnych – metodyka obliczeń tłumików wiskotycznych drgań skrętnych. Materiały firmy, Warszawa 2010.
18. MacPherson D.M., Puleo V.R., Packard M.B.: Estimation of Entrained Water Added Mass Properties for Vibration Analysis. The Society of Naval Architects & Marine Engineers, New England Section, June 2007.

Other

19. Zygmuntowicz J.: Obliczeniowy model wiskotycznego tłumika drgań skrętnych. Silniki Spalinowe 4, 1989.