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Technical note

THE IMPACT OF COMPLEX FORCING ON THE VISCOUS TORSIONAL VIBRATION DAMPER'S WORK IN THE CRANKSHAFT OF THE ROTATING COMBUSTION ENGINE

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The numerical calculations results of torsional vibration of the multi-cylinder crankshaft in the serial combustion engine (MC), including a viscous damper (VD), at complex forcing, were shown. In fact, in the MC case the crankshaft rotation forcings spectrum is the sum of harmonic forcing whose amplitude can be compared with the amplitude of the 1st harmonic. A significant impact, in the engine operational velocity, on the vibration damping process of MC, may be the amplitude of the 2nd harmonic of a forcing moment. The calculations results of MC vibration, depending on the amplitude of the 2nd harmonic of the forcing moment, for the first form of the torsional vibration, were shown. Higher forms of torsional vibrations have no practical significance. The calculations assume the optimum damping coefficient VD, when the simple harmonic forcing is equal to the base critical velocity of the MC crankshaft.

Key words: viscous damper, torsional vibration, crankshaft, combustion engine

1. Introduction

In multi-cylinder combustion engines installed in buses, trucks, on ships, etc., to dampen the torsional vibration of the crankshaft (MC), the Holsett type viscous dampers (VD) [1] are installed.

The VD damper is simple in construction and during operation requires virtually no adjustment and maintenance. It effectively dampens the torsional vibrations in the whole operating velocity range, provided it is properly matched to the type of engine (MC crankshaft). The disadvantages include: the relatively large mass, especially governor weight and possibility of generating significant flexural vibrations in the case of incorrect MC crankshaft bearing. For the system: engine-damper (E-VD), the appropriate optimal damper (OVD) can be selected, for which a value damping coefficient α is equal to the optimum value α_{OPT}. This system is called an optimal system (OS). For the OS the maximum amplitude is less than all maximum amplitudes from the set, for which $\alpha \neq \alpha_{OPT}$ in the whole operating velocity range of MC crankshaft. The optimal coefficient for the E-VD system can be calculated by using the theoretical formulas given in the literature [2, 3, 4]. They have been developed on the basis of a simple dynamic E-VD model of harmonic forcing. Gas pressure force and internal force are those ones forcing the torsional vibrations of the MC crankshaft.

The influence of inertial forces, in comparison to the gas pressure forces, on the torsional vibrations is small and in the dynamic calculation of the MC crankshaft can be omitted. The dynamic calculation takes into account only the periodic force from the pressure in the engine cylinder wherein the "shape" is similar to time characteristic of pressure wave caused by the rapid combustion of air-fuel mixture [5]. Theoretically, the forcing torsional vibration moment $M(t)$ can be represented as a Fourier series. Practically, it is possible to limit to the first few harmonics as higher harmonics with high frequencies and relatively small amplitudes do not matter. The work is limited to the 2nd-harmonic, where the amplitude, in comparison to the 1st one, is approximately *50*% reduced [1]. The aim of the study was to check what the impact is of the 2nd harmonic on the torsional vibrations of the crankshaft, and thus the VD crankshaft work, wherein the damping is assumed equal to α_{OPT} and calculated for harmonic forcing. The present work is part of the general issues related to the calculation issues of torsional vibration dampers which the author work on [6, 7, 8].

2. The dynamic model of the E-VD system

A dynamic model of the E-VD system is shown in Fig.1.

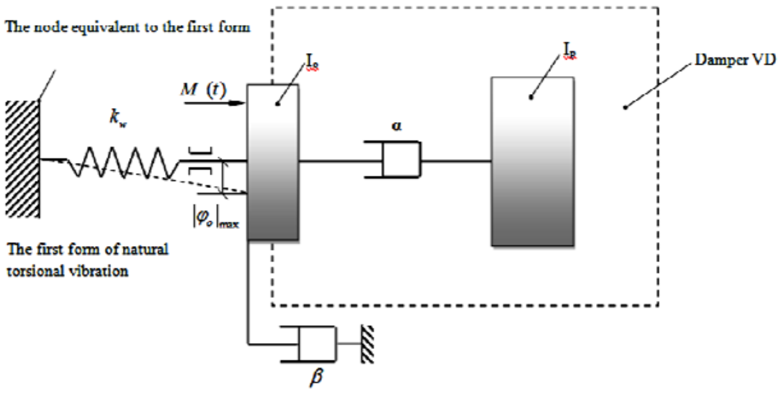

Fig.1. The dynamic E-VD model.

This is a one-frequency model with two degrees of freedom, and described by a dynamic equations system (2.1).

$$I_o\ddot{\varphi}_o + k\varphi_o + \alpha\left(\dot{\varphi}_o - \dot{\varphi}_p\right) + \beta\dot{\varphi}_o = M\sin\nu t,$$
$$I_p\ddot{\varphi}_p + \alpha\left(\dot{\varphi}_p - \dot{\varphi}_o\right) = 0 \tag{2.1}$$

where I_o - reduced mass inertia moment of the damper housing connected to the MC crankshaft. The reduction of the moment I_o should be carried out taking into account the form of torsional vibrations of the MC crankshaft. Most often this is the 1st torsional self-form, which corresponds to the 1st critical torsional crankshaft velocity (n_{I-kr}). Generally, this velocity is much higher than the maximum operating velocity of the crankshaft. This is due to the crankshaft torsional rigidity. The six-in-line and four-stroke engine can be the example [9], in which the resonant velocities (in the operating range) for the 1st form might occur when h (min) = *4.5*; and for the 2nd form embodiment when h (min) = *10.5*; where h - is the order of a harmonic. For this engine: n_{I-kr} = *18470* [*r/min*]; n_{II-kr} = *41900* [*r/min*]; Higher harmonics of the forcing moment, due to the smaller amplitude, the impact of damping of VD damper and engine natural damping (β), do not pose a threat to the work of MC. The specified moment I_o - should be treated as the average value for the full rotation of the MC crankshaft. In fact, the moment I_o - is a function of the crankshaft angle rotation $\phi(t)$, with the result that the system may cause additional parametric torsional vibrations. When the flywheel and damping are properly selected, the parametric vibration levels can be assessed on the noise level (as compared to the moment deriving from the gas forces). Similarly, in the case of moment coming from unbalance.

$I_p -$ moment of inertia of VD damper ring;

$k_w -$ equivalent crankshaft rigidity equals MC torsional rigidity, corresponding to the 1st form;

$\alpha -$ equivalent (linearised) damping factor of VD damper ;

β – - Includes natural damping (without damper) of MC vibration coefficient; β damping is small and its impact on the crankshaft dynamics is preferred.In the paper, $\beta = 0$.

M (t, T) - periodic moment forcing the torsional vibrations from gas forces of all the cylinders; T - basic forcing period.

$$T_{(2)} = \frac{60}{n_{wk} \cdot i_c},$$
$$T_{(4)} = \frac{120}{n_{wk} \cdot i_c} \tag{2.2}$$

$T_{(2)}$ - for the two-stroke engine;

$T_{(4)}$ - for the four-stroke engine;

n_{wk} [rev / min] – MC rotational speed;

i_c - the number of cylinders.

Solving the system of Eqs (2.1), for the harmonic ν and $\beta = 0$, the characteristics represented by the amplitude in the α damping coefficient function and ν frequency can be determined [2, 3]

$$\lambda_d(\text{def}) = \frac{|\varphi_0|_{\max}}{\frac{M_0}{k_w}} = f(\alpha, \nu),$$
$$f(\alpha = const; \nu) = \lambda_d(\nu), \tag{2.3}$$

$\lambda_d(\nu)$ - the amplitude - frequency (A-F) dimensionless characteristics

$\{ \max\{\lambda_d(\nu)\}$ for $\nu \in < \nu_{MIN}; \nu_{MAX} > = \lambda_{DYN}$ - E-VD system dynamic factor;

$$f(\alpha, \nu) = \lambda_d(\alpha, \nu) = \sqrt{\frac{I_p^2 \cdot \nu^2 + \alpha^2}{\left(1 - \frac{I_o}{k} \cdot \nu^2\right)^2 \cdot I_p^2 \cdot \nu^2 + \alpha^2 \cdot \left[\frac{I_o + I_p}{k} \cdot \nu^2 - 1\right]^2}}. \tag{2.4}$$

The A-F characteristics, developed on the basis of formula (2.4) for different values of the damping coefficient α were shown in the Fig.2.

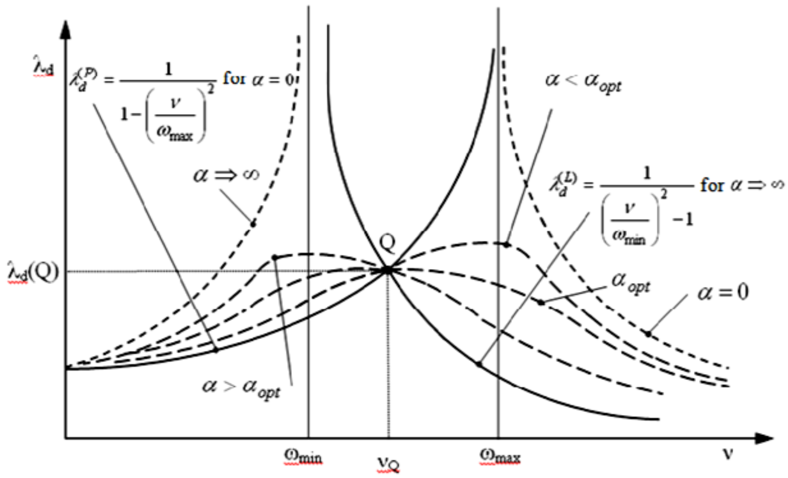

Fig.2. Characteristics $\lambda_d(\nu)$: special cases: $\alpha = 0$ - the lack of damping; $\alpha < \alpha_{opt}$ - "less damping"; The resonance zone moves towards higher velocities; $\alpha = \alpha_{opt}$ - the optimal damper; $\alpha > \alpha_{opt}$ - "bigger damping"; The resonance zone moves in the lower velocities direction; $\alpha \to \infty$ - the locked damper; The extreme cases are practically impossible.

The charts show that all the A-F characteristics intersect at the Q point, (the coordinates of the Q point do not depend on α)

$$\nu_Q = \sqrt{\frac{k_w}{I_0 + 0,5 \cdot I_p}}, \tag{2.5}$$

$$\lambda_{d(Q)} = 1 + \frac{2 \cdot I_0}{I_p}, \tag{2.6}$$

α_{OPT} can be determined from the condition of the function extremum (2.4) [2, 3]

$$\alpha_{opt} = 2 \cdot I_p \cdot \kappa_{opt} \cdot \sqrt{\frac{k_w}{I_0}} \tag{2.7}$$

where

$$\kappa_{opt} = \frac{1}{\sqrt{2 \cdot \left(2 + \frac{I_p}{I_0}\right) \cdot \left(1 + \frac{I_p}{I_0}\right)}}. \tag{2.8}$$

3. Numerical calculations

The calculations are made assuming that $\alpha = \alpha_{OPT}$ determined according to formulas (2.7) (2.8). The forcing took the form of

$$M(t) = M_{01} \cdot \sin(\nu_1 \cdot t) + M_{02} \cdot \sin(\nu_2 \cdot t) \tag{3.1}$$

where

$M_{01}; M_{02}$ - the first and second complex harmonic forcing amplitude, respectively;

$\nu_1; \nu_2$ - the first and second complex harmonic forcing frequency, respectively;

For calculations, the following quantities (in SI units) were assumed

$$I_O = 0.035\,;$$

$$I_p = 0.05;$$

$$k_w = 2.5 \cdot 10^4\,;$$

$$(2.5) \rightarrow \nu_Q = ok.645.5\,;$$

$$(2.6) \rightarrow \lambda_{d(Q)} = 2.4\,;$$

$$(2.7), (2.8) \rightarrow \alpha_{OPT} = 20.7104205;$$

$$\frac{M_{01}}{M_{02}} = 2\,; \quad \nu_1 = \nu\,; \quad \nu_2 = 2 \cdot \nu\,.$$

Figure 3 shows an exemplary graph of crankshaft torsional vibrations.

$$\zeta(t) = \frac{\varphi_0(t)}{M_{01}/k_w}\,; \qquad t \in \langle 0; 0.1 \rangle\,, \tag{3.2}$$

for $\nu = 0.5 \cdot \nu_Q (322.75)$; other computing data as described above.

At this frequency the 2nd harmonic is clearly noticeable.

In the relation to the value $\lambda_{d(Q)} = 2{,}4$ the vibration increased by approximately *4.21*%.

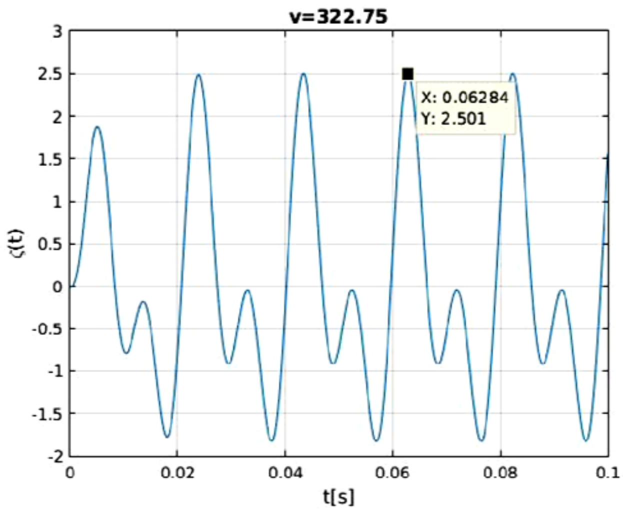

Fig.3. The crankshaft torsional vibrations chart when $\nu = 0{,}5 \cdot \nu_Q$.

The A-F characteristics by applying biharmonic forcing and formula (2.4) are shown in Fig.4. In addition, calculations were made for: $M_{02}/M_{01} = 0.25$ and $M_{02} / M_{01} = 0.1$.

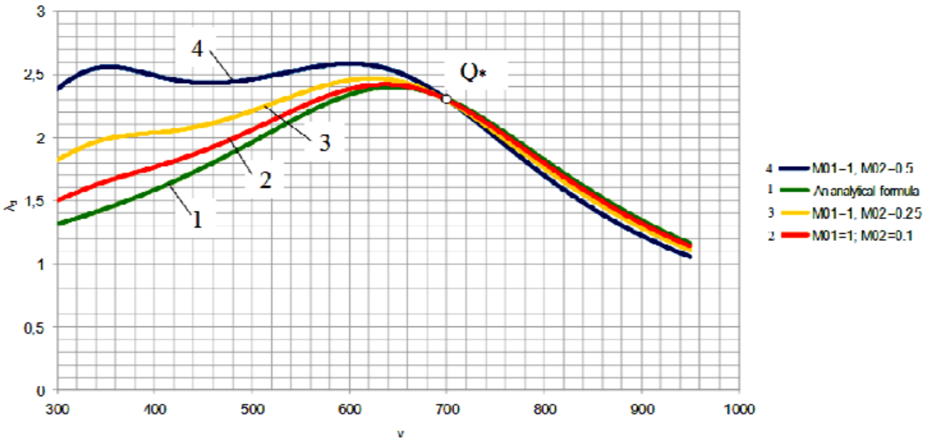

Fig.4. The calculation results in the frequency range <*300*; *950*>: 1 - of formula (2.4); 2 – $M_{02} / M_{01} = 0.1$; 3 – $M_{02} / M_{01} = 0.25$; 4 – $M_{02} / M_{01} = 0.5$.

Figure 5 shows the calculation results of λ_{DYN} coefficient and ν_* (the frequency when the vibrations are maximum) depending on the M_{02} / M_{01} ratio. The result that the resonance zone (with the M_{02} increase) decreases quasi-linearly (in the range of tested values) in the lower velocities direction is Noteworthy.

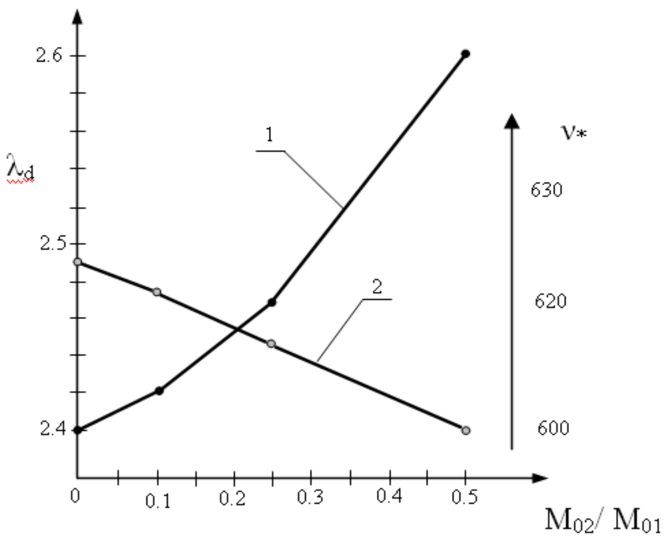

Fig.5. λ_{DYN} factor (1) change and frequency ν * (2) depending on M_{02}/M_{01} in the range of <*0*; *0.5*>.

4. Conclusion

The second forcing moment harmonic (at the level of *0.5*), in comparison to the 1st harmonic, increases vibration by *8*%, which is not a high fatigue risk. Of course, there are other higher harmonics but their impact should be appropriately small. This was partly confirmed by the calculation at the 2nd harmonic at levels of *0.25* (the vibration increased by approx. *3*%) and *0.1* (the vibration increased by approx. *1*%). However, this must be checked in further work. The resonance zone clearly increased. The vibrations are close to maximum (level of approx. *2.5*> *2.4*) and remain in the frequency range from approx. *320* to approx. *660*. It can be a threat to the crankshaft fatigue strength. The second local maximum was observed at a frequency close to the 2nd harmonic.

The runs of all designated characteristics in the area of the Q_* point change the relative position on the reverse ($\nu_Q = ok.700$). The Q point (Fig.2) moved (on right) by approximately 55 which represents approximately *8.5*%. For frequency $\nu > \nu_Q$, the second harmonic has practically no effect on the vibration level, and particularly with the increase of M_{02}, the vibration slightly decreased. The optimal damping, determined with formulas (2.7), should be replaced by a "new" value, where at the Q_* point (Fig.4) the characteristics of A-F (2.4) reach a maximum. This issue is currently the subject of the author's work.

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