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DILATANT-FLUID TORSIONAL VIBRATION DAMPER FOR A FOUR-STROKE DIESEL ENGINE CRANKSHAFT

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ABSTRACT

This paper presents a study of a viscous torsional vibration damper for a crankshaft of a four-stroke diesel engine. The reliable operation of a widely used silicone-type viscous damper depends on the ability of the silicone oil to absorb the energy of torsional vibrations. The non-Newtonian shear flow of the silicone oil interlayer, characterised by a reduction in the shear-rate-dependent viscosity and a moment of the drag forces, negatively affects damping characteristics. A torsional vibration damper, filled with a shear-thickening fluid, was considered and a rheological approach, based on viscosity growth with the shear rate increase, was applied. For such a damper, larger velocity gradients correspond to the higher values of a viscous force, which decreases torsional vibration. The parameter of damper effectiveness (defined by the fluid flow index, values of the damper gaps, torsional vibration amplitude and frequency) was implemented. It has been established that the efficiency of the torsional vibration damper filled with a dilatant fluid does not depend on the damper dimensions and gaps and significantly increases when a shear-thickening fluid is used instead of silicone oil or a Newtonian fluid. At higher values of the flow index, when the non-Newtonian flow becomes distinct, torsional vibrations are damped more effectively. Critical vibration amplitudes at high-velocity gradients, in turn, increase the damping effect as the moment of the drag forces and flow index are power-law related.

Keywords: r torsional vibration damper, dilatant fluid, non-Newtonian flow, flow index, parameter of effectiveness

INTRODUCTION

The current trend of increasing power output in internal combustion engines has lead to the significant influence of dynamic loads on the engines' main elements. Shafting is one of the most stressful components, where energy is mainly concentrated. Shaft axial, torsional and bending vibrations may be excited by forces and moments that cause the rotational motion of the shaft. The main goal of shaft vibration control is to ensure the reliability of diesel engines at different operating modes. Significant amplitudes of torsional vibrations affect the operational life of the shafting elements, such as the crankshaft, gear train, clutch elastic elements and shaft joints [1, 2]. A considerable number of modern diesel engines are equipped with damping mechanisms that reduce the negative effect of high-amplitude torsional vibrations. The common feature of all existing types of dampers is a frictional element, located between the inertia ring and the housing, which absorbs part of the vibrational energy.

Because of their simplistic construction, viscous torsional vibration dampers are frequently installed at the free end of marine four-stroke diesel engines used as prime movers of diesel generators. These dampers effectively decrease torsional vibrations, although they are ineffective in powerful diesel engines, where other types of dampers are used [3]. A damper consists of a balanced freely rotating inertia ring inserted into the balanced outer housing [4]. The gap between the housing and the ring is filled with a viscous fluid. Oxygen-containing high-molecular organic-silicon compounds are widespread damping media [5], due to the high values of their viscosity and viscosity index [6], as well as their satisfactory lubricity [7]. When the damper housing rotates and executes torsional vibrations together with the engine crankshaft, the inertia ring due to the viscous friction in the silicone oil is drawn into co-rotation and torsional vibrations together with the damper housing. The large secondary mass – the inertia ring – tends to mechanically maintain the ultimate angular velocity. Relative motion between the inertia ring and the housing occurs as a result of their periodic rotational motion and the moment of viscous friction appears. This leads to the dissipation of part of the vibrational energy in a thin interlayer of a viscous fluid, leading to the torsional vibration amplitude contraction.

In the simplest case [8], it is assumed that the resulting frictional force F_{fr} (N), acting on a surface element dS (m^2) is proportional to the applied velocity gradient dv/dz (s^{-1})

$$dF_{fr} = -\eta \frac{dv}{dz}\, dS, \tag{1}$$

where the coefficient of proportionality η (Pa·s) is the coefficient of the dynamic viscosity. Fluids that obey this law are called Newtonian fluids. In theoretical research, approximations are limited by the constant value of the dynamic viscosity.

The analysis of a silicone-type damper was conducted by the authors in [9]. Their approach was based on the reduction of the shear-rate-dependent viscosity of the silicone oil in bulk [6] and structurally inhomogeneous micron interlayers of lubricating fluids [10, 11]. It has been shown that, in contrast to Newtonian fluids, the moment of the drag forces decreases when the velocity gradient increases. This, along with the fact that the viscosity of the silicone oil lowers within the damper's performance life [3], leads to the reduction of silicone-type damper efficiency [12] at the most dangerous high amplitudes of torsional vibrations.

PROBLEM STATEMENT

The purpose of this paper is to analyse the effectiveness of a torsional vibration damper, filled with a dilatant fluid, whose viscosity grows when the velocity gradient between the housing and the inertia ring increases. To the best of the authors' knowledge, research on the utilisation of shear-thickening fluids for vibration problems has only been conducted for civil engineering applications [13, 14].

The rheology of non-Newtonian fluids can be described by the nonlinear equation [8]:

$$dF_{fr} = K_n \left(\frac{dv}{dz}\right)^n dS, \tag{2}$$

where K_n is the consistency coefficient ($kg\, m^{-1}s^{n-2}$) that depends on the dimensionless flow index n.

The coefficient K_n can be constant within a certain range of the velocity gradients and is known as apparent viscosity. The deviation from Newtonian viscosity is characterised by the flow index $n \neq 1$. Dilatant fluids are characterised by an increase in viscosity when shear stress is set up. The flow index corresponds to $n > 1$. Most suspensions with high solid particle contents are examples of dilatant fluids [15]. For a damper filled with a dilatant fluid, the higher the value of the applied velocity gradient, the larger the viscous force that will decrease the torsional vibrations of high amplitudes.

CALCULATION OF THE EFFECTIVENESS PARAMETER OF THE DILATANT-FLUID TORSIONAL VIBRATION DAMPER

When a crankshaft rotates, the moment of forces M (Nm) is given by [16]

$$M = I \cdot \frac{\omega_{max} - \omega_{min}}{t}, \tag{3}$$

where I ($kg \cdot m^2$) is the reduced moment of inertia of the shaft, ω_{min} and ω_{max} (rad/s) are the respective minimum and maximum angular velocities of the shaft over a period, and t is the time (s).

Excess work W (J) spent on increasing the kinetic energy $\Delta Ek(J)$ of rotating masses is

$$W = \Delta Ek = I \cdot \frac{\omega_{max}^2 - \omega_{min}^2}{2} = I \cdot (\omega_{max} - \omega_{min})\omega_{mean} \tag{4}$$

where $\omega_{mean} = (\omega_{max} + \omega_{min})/2$ is the mean angular velocity (rad/s) of the shaft at the steady state conditions.

The kinetic energy created is converted into heat that is transferred to the environment and causes the heating of the viscous fluid and the damper. This continues until the temperature of the damper reaches the so-called saturation temperature [3], beyond which the overheated damper may seize up.

The internal moment of resistance M_{el} (Nm) in the shaft twisted through an angle φ is given by [16]

$$M_{el} = -\frac{GI_p}{l}\varphi, \tag{5}$$

where I_p (m^4) is the polar second moment of area, G (Pa) is the modulus of rigidity, l (m) is the length of the shaft, φ (rad) is an angle of twist.

At equilibrium, when the shaft is twisted through an angle φ:

$$|M_{el}| = M. \tag{6}$$

Substituting Eq. (3) into Eq. (4), taking Eq. (6) into account, gives

$$W = Mt\omega_{mean} = M\varphi = \Delta U_{el}, \tag{7}$$

where ΔU_{el} (J) is the increase in elastic strain energy stored in the twisted shaft.

The torsional vibration damper partially absorbs vibrational energy due to heat dissipation Q (J) as a result of the work of viscous friction forces:

$$W_{viscous} = Q = \int M_{viscous} d\varphi = <M_{viscous}> \varphi. \tag{8}$$

Following Eq. (8), the greater the moment of viscous forces $M_{viscous}$ in the damper, the more effective it is because an increase in a moment $M_{viscous}$ leads to a decrease in the angle of twist φ which can be considered as one of the criteria of damper effectiveness.

When the damper housing rotates together with the crankshaft, the inertia ring co-rotates around the housing with the same angular velocity, due to the viscous friction. If the rotational speed of the shaft changes, as long as the moment of inertia forces acting on the secondary mass is less than the moment of viscous friction, the housing and the ring rotate as a single whole. At high vibration amplitudes, as soon as the moment of inertia force becomes larger than the moment of viscous friction, the inertia ring turns relative to the housing. Such motion in the secondary mass is repeated with the frequency of torsional vibrations.

The moment of viscous friction, generated when a fluid flows through a precision shear gap between the inertia ring and the housing, creates damping drag. It should be noted that the viscous friction force for dilatant fluids at low shear rates is lower compared to Newtonian fluids and, as a result, the secondary mass starts to move at lower values of inertia force, i.e. at lower amplitudes of the torsional vibrations. However, the viscous friction force rises sharply when the torsional vibration amplitude (the velocity gradient) increases. Hence, it is expected that the viscous damper, filled with a dilatant fluid, should dampen the vibrations more effectively than dampers filled with silicone oil or a Newtonian fluid.

The assessment of the contribution of dilatant fluids to the effectiveness of the damper is presented below. The inertia ring 2 (Fig. 1), has the form of a ring of width L, and an inner r and outer R radii relative to the shaft axis, positioned inside the outer housing 1. Between the inner, outer and end surfaces of the inertia ring and the housing, there are small gaps h_i filled with a dilatant fluid.

The moment M (Nm) of the drag forces, arising in the damper as a result of torsional vibrations, is given by [17]:

$$M = \int \rho dF_{fr} = \int_s \rho K_n \cdot \left(\frac{dv}{dz}\right)^n dS, \tag{9}$$

where integration is carried out over the surface elements dS which are at a distance ρ from the axis of rotation.

A change in the shaft angle of twist φ (rad) can be expressed as a sinusoidal oscillation

$$\varphi = \varphi_0 \sin \omega t,$$

and the velocity gradient is defined as

$$\frac{dv}{dz} \approx \frac{\rho d\varphi}{dz} = \frac{\rho\varphi_0\omega}{h} \cos \omega t, \tag{10}$$

where φ_0 is the amplitude of the torsional vibrations.

Since gaps h_i between the surfaces of the housing and the ring are very narrow, compared to the damper dimensions, a change of dz for h_i is carried out in Eq. (10).

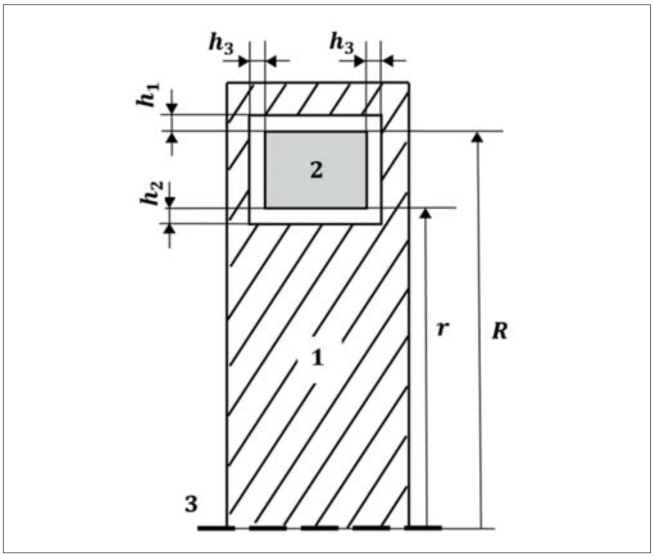

Fig. 1. The viscous damper: 1 – housing; 2 – inertia ring; 3 – shaft axis; – inner radius of the inertia ring; – outer radius of the inertia ring

Eq. (9) takes Eq. (10) into account to give

$$M_n(t) = K_n|\cos^n \omega t| \cdot \int_s \rho \cdot \left(\frac{\rho\varphi_0\omega}{h}\right)^n \cdot dS. \tag{11}$$

The absolute value of the harmonic function is considered in Eq. (11), as drag appears at negative and positive function values.

In a conventional damper design, there are two cylindrical surfaces between the ring and the housing, including the lower S_L, upper S_U and two side surfaces S_S. Their elements are determined as follows:

$$dS_L = 2\pi \cdot r \cdot dL, \quad dS_U = 2\pi \cdot R \cdot dL, \quad dS_S = 4\pi \cdot \rho \cdot d\rho. \tag{12}$$

The gaps h_i (Fig. 1) between the inertia ring and the damper housing are different [18]:

h_1 is the upper diameter clearance;
h_2 is the lower diameter clearance;
h_3 is the axial end clearance.

Calculation of the drag forces moment, taking into account Eq. (12), leads to

$$M_n(t) = K_n|\cos^n \omega t| \cdot f(n), \tag{13}$$

where

$$f(n) = \left(\frac{\varphi_0\omega}{h_1}\right)^n \cdot 2\pi \cdot L \cdot R^{n+2} + \left(\frac{\varphi_0\omega}{h_2}\right)^n \cdot 2\pi \cdot L \cdot r^{n+2} + \left(\frac{\varphi_0\omega}{h_3}\right)^n \cdot 4\pi \cdot \frac{R^{n+3} - r^{n+3}}{n+3}. \tag{14}$$

Time averaging over a period gives

$$<|\cos^n \omega t|> = \frac{1}{2\pi} \int_0^{2\pi} |\cos^n \varphi| d\varphi. \tag{15}$$

The ratio of the drag force moments that dilatant ($n \neq 1$) and Newtonian ($n=1$) fluids create will be of the form

$$\frac{<M_n(t)>}{<M_1(t)>} = C \frac{f(n) <|\cos^n \omega t|>}{f(1) <|\cos \omega t|>}; \tag{16}$$

here $C=K_n/\eta$ determines the dilatant fluid constant, in comparison with Newtonian fluid.

Parameter E defines the effectiveness of the dilatant-fluid damper, compared with the Newtonian-fluid analog, and is implemented in the following form:

$$E = \frac{1}{C} \cdot \frac{<M_n(t)>}{<M_1(t)>} = \frac{f(n)<|cos^n\ \omega t|>}{f(1)<|cos\ \omega t|>}. \quad (17)$$

Effectiveness depends on the flow index n and gap values h_i between the housing and the inertia ring and such parameters of torsional vibrations as amplitude φ_0 and frequency ω.

Calculation of parameter E at different values of the flow index n was carried out for representative values [18] of torsional vibrations: φ_0=0.02 rad and ω=0.5 rad/s. Two dampers were chosen for the calculations.

The dimensions and clearances between the inertia ring and the housing of the Holset [18] damper №1 were: R=340 mm, r=217 mm, L=105 mm, h_1=1.5–2.0 mm, h_2=0.2–0.25 mm, and h_3=0.4–0.5 mm. The dimensions [19] of damper №2 were: R=302.7 mm, r=191.0 mm, L=25.3 mm, and the clearances between all surfaces were considered to be equal to h=0.5 mm. The results of the calculation are summarised in Table 1 and depicted in Fig. 2. Gap values for damper №1 were averaged: h_1=1.75 mm, h_2=0.225 mm and h_3=0.45 mm. Errors of calculation of the integrals, functions and the parameter of effectiveness did not exceed ~ 0.1%.

Tab. 1. The dependence of the parameter of effectiveness E on the flow index n for the dilatant-fluid torsional vibration damper

n		1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00
E	№1	1.00	1.35	1.93	2.84	4.27	6.52	10.10	15.90	25.20
	№1	1.00	1.30	1.74	2.40	3.35	4.73	6.76	9.72	14.07

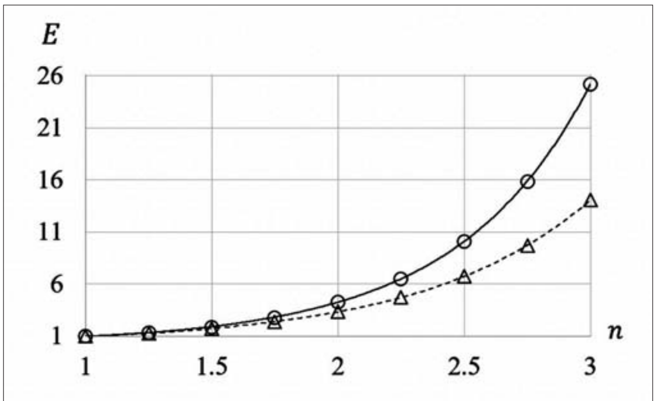

Fig. 2. The dependence of the parameter of effectiveness on the flow index for the dilatant-fluid torsional vibration damper. Dampers: ○ – № 1, Δ – № 2

CONCLUSION

The presented theoretical research on the increase of the effectiveness of the torsional vibration dampers has been conducted for shear-thickening fluids whose viscosity grows with the velocity gradient. For this purpose, the effectiveness parameter, characterised by the moment of viscous forces that depends on the apparent viscosity, frequency and amplitude of oscillations, has been introduced. Calculations have shown that it is expected that the operational reliability of the torsional vibration damper (filled with a dilatant fluid and regardless of the gaps and dimensions) will significantly grow, compared to the equivalent damper filled with silicone oil or a Newtonian fluid. Torsional vibrations will be dampened more effectively when the flow index increases by n>1 and the non-Newtonian flow becomes distinct. However, it is expected that high values of the flow index n will result in a decrease in damper effectiveness due to the large viscous friction forces, which will prevent the relative motion of the housing and the inertia ring. Critical vibration amplitudes appearing at the high-velocity gradients result in a significant increase in the damping effect, as the moment of the drag forces and the flow index are power-law related: $M\sim(\varphi_0)^n$.

The conducted theoretical research requires experimental verification using different dilatant fluids as operating media of viscous dampers. Along with the experimental assessment of damper effectiveness, the lubricating properties and thermal stability of the dilatant fluids also need to be investigated.

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