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ANALYSIS OF AN AUTOMOTIVE VEHICLE ENGINE MOUNT BASED ON A SQUEEZE-MODE MAGNETORHEOLOGICAL DAMPER

ANALIZA SEMIAKTYWNEGO ZAWIESZENIA SILNIKA SAMOCHODOWEGO Z WIBROIZOLATORAMI MAGNETOREOLOGICZNYMI

A b s t r a c t

Methods are outlined that can be employed to reduce the dynamic components of interaction forces between the vehicle engine and the car body, taking into account the main sources of vibration in the engine. The transmitted forces arising due to the reciprocating motion of the pistons in a two-cylinder engine are calculated. The amplitude and frequency characteristics are used to develop a method for controlling the engine mount stiffness and damping depending on the angular velocity of the crank shaft motion. The study explores the feasibility of damping control in the mount through the use of magnetorheological (MR) dampers operated in the squeeze mode.

Keywords: *vibration, engine mount system, MR damper, control*

S t r e s z c z e n i e

W artykule omówiono metody obniżenia wartości składowych dynamicznych sił oddziaływania silników spalinowych na karoserie samochodów biorąc pod uwagę podstawowe źródła drgań silników. Przeprowadzono obliczenia przenoszenia sił powstających w wyniku ruchu posuwisto-zwrotnego układu tłoków dla prostego silnika spalinowego. Na podstawie analizy otrzymanych charakterystyk amplitudowo-częstotliwościowych zaproponowano ogólną metodę sterowania sztywnością i tłumieniem zawieszenia silnika w zależności od prędkości obrotowej wału korbowego. Przedstawiono możliwości sterowania sztywnością i tłumieniem zawieszenia przy zastosowaniu specjalnie skonstruowanych tłumików magnetoreologicznych (MR).

Słowa kluczowe: *drgania silników spalinowych, zawieszenie silnika spalinowego, charakterystyka tłumika MR*

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1. Introduction

An automotive vehicle engine is a rigid body with a complex geometry. It is fixed to the car body on supports whose placements and parameters are derived from calculations of the static and dynamic behaviour of the entire driving system. The static calculations involve the displacements of the main sub-assemblies and deformation of the applied supports under static equilibrium conditions. The dynamics calculations are much more complex. They take into account the fundamental reasons for the drive unit vibrations such as inertia forces produced by the motions of unbalanced elements of the crank shaft system, the periodically fluctuating torque transmitted via the engine to subsequent components of the driving system in the vehicle and some random forces. A vibrating engine is one of the sources of vibrations of the car body. These vibrations are induced by forces transmitted via the engine mount elements onto the frame. The minimisation of the dynamic components of these forces is a major problem in vibration control. As the reasons of engine vibrations cannot be wholly eliminated, it is of particular importance that the structural design of the engine mount is such that the engine position should be stabilised and the dynamic components of forces between the engine and the car body should be minimised.

Elastic mounts of the automotive vehicle driving systems date back to the 1930s [10]. The first mount systems would be made of relatively small and cheap rubber elements. In the 1960s, the mount system was introduced in which the engine was fixed with hydraulic elements. In the following years, these elements were modified and improved. Currently, research efforts are focused on semi-active and active engine mount systems, enabling more effective reduction of undesired interaction forces between the engine and the vehicle body.

This study presents the results of the calculations of forces acting upon the car body as a result of the reciprocating piston motion. The model of a two-cylinder engine was used in calculations. Based on the obtained frequency response functions, a method is developed for controlling the engine mount vibrations (stiffness and damping control) depending upon the angular velocity of the crank shaft motion. The potentials of using dedicated MR dampers for mount damping and stiffness control are explored.

2. Design structure of key components of the vehicle engine mount systems

The reason why the selection of the engine mounts has received such a great deal of attention from design engineers is the current trend in vehicle development to fabricate lighter front-wheel drive vehicles having light engines with low idle speeds. The most stringent requirements could be satisfied only by mounts with controllable stiffness and damping characteristics. The first components developed in consideration of this requirement were hydraulic elements patented in 1962 [10]. When vehicle mounts are fitted with such components, the amplitudes of car body vibrations during the motion are decidedly lower in comparison to when using mounts with rubber components. The cross-section of a typical hydraulic element is shown in Fig. 1.

The top housing (3) connected to the engine via the fixing system (5), is made of rubber. It closes the top chamber (1) containing oil. Oil flows from the top to the bottom chamber

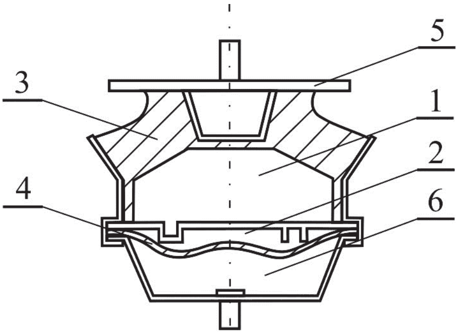

Fig. 1. Cross-section of a hydraulic pressure element [10]

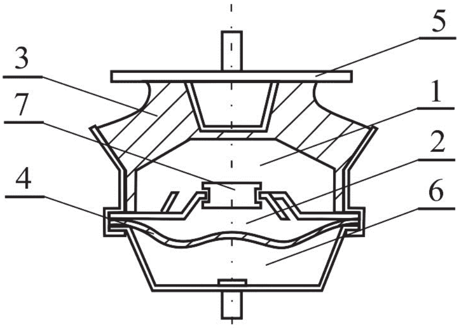

Fig. 2. Cross-section of a hydraulic pressure element with the de-coupler system [10]

(2) via an orifice or through a conduit with a precisely controlled diameter. The bottom chamber is limited from below by a rubber housing (4), which can expand due to the liquid pressure increase. It expands to the inside of the restricted volume (6) where the atmospheric pressure will prevail. The rubber housing deformed under the weight of the engine will generate pressure in both chambers. Dynamic displacement components give rise to further deformations of the upper housing (made of rubber), causing the oil flow through an orifice or a calibrated pipe. Mount stiffness and damping can thus be effectively controlled over a wide range of movement.

The most advanced version of hydraulic elements is shown in Fig. 2. Here, the solution incorporates an inertial system for decoupling the flow between the chambers (7). The aim of this system is to make the damping amplitude-dependent and frequency-dependent. The system controls the liquid flow between the chambers, such that the orifice should be bypassed at precisely selected amplitudes – thus, damping control is provided. In relation to the engine vibration parameters the mount characteristics can be better fitted to the engine performance. The operating principles of hydraulic elements employed in engine mounts are investigated, based on linear and nonlinear models [1, 3, 7]. The model parameters can be determined using simulation data or calculation procedures based on optimisation methods, alternatively, they can be determined experimentally.

Hydraulic mounts are tunable under the applied sine variable inputs of precisely controlled frequency. In the case of more complex inputs with a wide frequency range, the mount is no longer effective, this is a feature of passive mounts. Improved mount performance in one frequency range often leads to its deterioration in other frequency ranges. Therefore, the selection of model parameters will always involve a certain trade-off.

Optimal mount performance in the entire frequency range can be achieved with semi-active or active mounts, where the adequate control algorithm is applied [2, 4]. The literature on the subject abounds in reports on mounts in which MR and electrorheological (ER) fluids are employed. The property of these fluids is utilized – their apparent viscosity changes significantly under the influence of a variable electric field. The design structure of semi-active elements used in mounts is similar to that of hydraulic components. Around the flow path from the top to the bottom chamber, there are electrodes generating the electric field acting upon the ER fluid. Thus, the mount damping can be controlled and the optimal damping can be ensured in any frequency range. An integral part of any semi-active mount is a control system to implement the control algorithm.

3. Engine – frame interactions

To determine the interactions between the engine and the frame in the car body, the forces are calculated that appear in the engine mount due to the reciprocating motion of pistons in a two-cylinder four-stroke combustion engine. The schematic diagram of the engine is shown in Fig. 3.

Two pistons (1) are in concurrent operation. During the upward motion, the compression stroke is implemented in one cylinder and the exhaust stroke in the other. During the downward motion, the respective power stroke and intake stroke are performed. On account of the pistons' concurrent operation, a counterweight (5) is provided on the crank shaft (4) to partially balance the first harmonic of the inertia force associated with the pistons' reciprocating motion and the inertia force of the crankshaft web. The diagram also illustrates the flywheel (6), connecting rods (3) and the bolt (2) fixing the piston on the connecting rod.

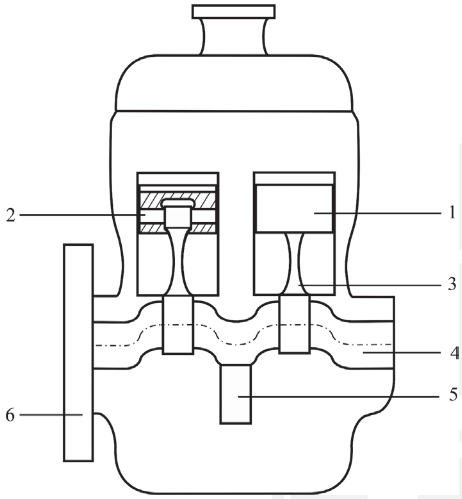

Fig. 3. Cross-section of the engine

It is assumed that the calculation procedure should involve only the engine motion in a vertical direction in the plane perpendicular to the crank shaft, passing through the crank centre of gravity. The simplified diagram of the crank shaft and piston mechanism in the engine block is shown schematically in Fig. 4.

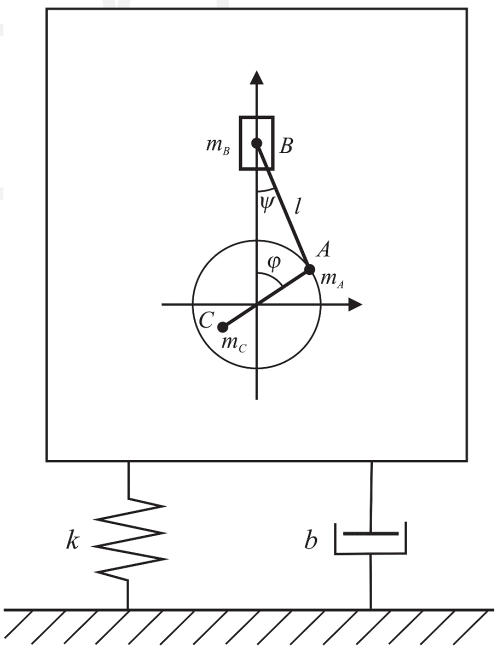

Fig. 4. Schematic diagram of a crank shaft and piston system inside the engine block

This diagram indicates all masses taken into account in the calculation procedure. The mass of the connecting rod is reduced to two points: point A, representing the connection of the rod and crank shaft and point B, representing the piston-rod connection. This is a widely employed method of reducing the mass of the connecting rod. Finally, m_A stands for the sum of the reduced crankshaft web mass and the reduced mass of the bottom part of the connecting rod, m_C is the mass of the counterweight, reduced to its centre of gravity.

The frame to which the engine is attached is assumed to be fixed and the engine mounting system is modeled by a spring and a viscous damper connected in parallel. During the steady-state operation of the engine, the angle φ is linearly dependent on time ($\varphi = \omega t$, where ω − angular speed of the shaft rotation). The equation of engine vibration is given as:

$$\begin{aligned} M\ddot{y} + b\dot{y} + ky = & \frac{1}{2} m_B r_A \omega^2 \cos(\omega t) + \\ & + m_B r_A \omega^2 \left[\left(\lambda + \frac{1}{4}\lambda^3 + \ldots \right) \cos(2\omega t) - \left(\frac{1}{4}\lambda^3 + \ldots \right) \cos(4\omega t) + \ldots.. \right] \end{aligned} \tag{1}$$

M stands for the engine mass, its components m_A, m_B, m_C are indicated in Fig 4. The coefficient *k* expresses the mount stiffness and *b* is the equivalent mount damping coefficient. The coordinate *y* represents the engine frame displacement with respect to its static equilibrium position. The assumption of model linearity allows the vibration equation to be separated from that governing the static equilibrium: $ky_{eq} = Mg$. The right-hand side of Eq (1) contains the term expressing the fundamental harmonic and the subsequent two harmonics of the piston inertia forces. The remaining harmonic components are neglected. The fundamental harmonic of the inertia forces is partly balanced by providing a counterweight. The most popular solution has been adopted here, whereby half of the first harmonic of the piston inertia force is balanced in such a way that the first harmonic of the resultant inertia force of the whole crankshaft-piston mechanism should become a vector with a constant modulus and rotating at the angular speed ω in the direction opposite to the crank shaft rotation [5, 6]. In such a configuration, the higher harmonic components of the inertia force will not be balanced. The coefficient λ is expressed as the quotient of the crank web length and the length of the connecting rod. Since the value of λ is always less than one, it is a widely adopted parameter used when expanding the inertia forces into the power series.

When analysing the vibration transmission from the engine onto the car body, the amplitude of the force of the mount-frame interaction is of particular importance. This interaction force can be derived as the resultant force of the force in the spring and the force in the damper:

$$S = b\frac{dy}{dt} + ky \tag{2}$$

Since Eq (1) and (2) are linear, the force $S(t)$ has the component $S_1(t)$ with the frequency ω associated with the first harmonic inertia force of the crankshaft-piston mechanism and the higher order components $S_2(t)$, $S_3(t)$ associated with higher harmonics of the inertia force. The amplitudes of all force components are functions of the angular speed of the crank. The amplitudes of the first and second harmonics can be derived from the formulae:

$$S_{10} = \frac{1}{2}\frac{m_B}{M} r_A k \tilde{\omega}^2 \sqrt{\frac{1+4\zeta^2\tilde{\omega}^2}{\left(1-\tilde{\omega}^2\right)^2+4\zeta^2\tilde{\omega}^2}}$$

$$S_{20} = \frac{m_B}{M} r_A k \tilde{\omega}^2 \left(\lambda + \frac{1}{4}\lambda^3 + \frac{15}{128}\lambda^5 + \ldots\right)\sqrt{\frac{1+4\zeta^2\left(2\tilde{\omega}\right)^2}{\left(1-\left(2\tilde{\omega}\right)^2\right)^2+4\zeta^2\left(2\tilde{\omega}\right)^2}} \tag{3}$$

where $\tilde{\omega}$ is the dimensionless angular speed of the crank shaft and ζ is a dimensionless engine mount damping coefficient. These quantities are defined by the formulas:

$$\tilde{\omega}^2 = \omega^2 \frac{M}{k}$$

$$\zeta = \frac{b}{2\sqrt{kM}} \tag{4}$$

The relationship between the first and second harmonic amplitudes of the engine – car body interaction force and angular speed is shown in Fig. 5. Dimensionless amplitudes are computed by dividing the force amplitudes in Eq (3) by the expression $m_B r_A k/M$. The values of the dimensionless damping coefficient ζ used when plotting the graphs are 0.01, 0.205, 0.305, and 0.8.

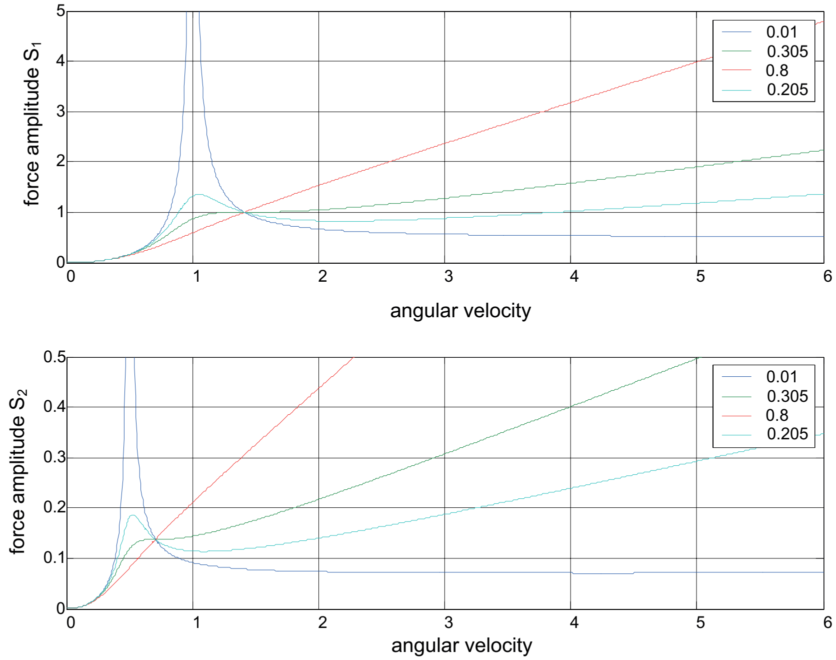

Fig. 5. Dimensionless amplitudes of the first S_1 and second S_2 harmonics of the engine-frame interaction force as functions of dimensionless angular velocity of the crank shaft motion

The function expressing the amplitude of the first harmonic becomes zero for zero frequency and tends to infinity when the angular velocity of the shaft is tending to infinity. There are two points where all graphs of the functions will intersect, regardless of the actual value of ζ. The coordinates of these two points are: (0; 0), ($\sqrt{2}$; 1).

The function governing the amplitude of the second harmonic follows a similar pattern, yet its values are significantly smaller. Resonance vibrations of the engine body cannot be observed due to the action of the second harmonic, since the maximum of the analysed function is placed for the angular velocity less than the idle speed.

4. Engine mount vibration control

The formula (3) defining the fundamental harmonic amplitude and graphs shown in Fig. 5 are utilised to develop the algorithms for engine mount stiffness and damping control. It is readily seen that:

– for the dimensionless angular velocity equal to $\sqrt{2}$, the value of the force's first harmonic becomes $m_B r_A k/M$, no matter what the actual value of the dimensionless damping coefficient,
– the force reaches its maximum for the dimensionless angular velocity being slightly more than 1 and less than $\sqrt{2}$,
– for the dimensionless angular velocity tending to zero, the force will also tend to zero,
– when the dimensionless damping coefficient is more than zero, then for the dimensionless angular velocity tending to infinity, all harmonic components of force will also tend to infinity, increasing at the rate 20 dB/decade,
– when the dimensionless damping coefficient is equal to zero, then for the dimensionless angular velocity tending to infinity, the amplitude of the force's first harmonic will approach 0.5 $m_B r_A k/M$, which is a finite value and greater than zero,
– for a dimensionless angular velocity smaller than $\sqrt{2}$, further increasing the dimensionless damping coefficient will cause the force value to be reduced, whereas for a dimensionless angular velocity greater than $\sqrt{2}$, an increase of the dimensionless damping factor will cause to the force value to increase,
– the force value goes up when the stiffness coefficient is increased and decreases when the stiffness coefficient decreases,
– the larger the ratio m_B/M, the larger the force value.

4.1. Damping control

The impacts of damping vary with the changed angular velocity of the shaft, this is why damping control algorithms have to utilise the instantaneous velocity value. This does not present any major difficulty as the angular velocity of the shaft is measured with sensors installed in the dedicated control systems provided in modern combustion engines. A damping control algorithm need not take into account the static analysis, since damping does not affect the static equilibrium position of the engine. Assuming that damping in the mount system is exclusively the result of providing an actuator placed parallel to the mount spring, the equation of vibration can be rewritten as:

$$M\ddot{y}+F_{con}(t)+ky=F_{ind}(t) \tag{5}$$

where $F_{ind}(t)$ is the inertia force inducing the vibration and $F_{con}(t)$ stands for the interaction force of the actuator in the control system.

The research investigation indicates that the simplest damping control algorithm is that switching between a large value of damping coefficient for angular velocities of the shaft less than the characteristic value (a product of $\sqrt{2}$ and natural frequency of the engine vibration) and a smaller value of the damping coefficient after a characteristic value of angular velocity is exceeded. It is worthwhile mentioning that the smallest angular velocity that can be maintained in the steady-state is the idle speed. It is slightly higher than the natural frequency. According to the control algorithm, the damping coefficient should assume a large value in the interval of crank shaft speed, starting from the idle speed to the characteristic value (being the product of $\sqrt{2}$ and natural frequency of the engine vibration).

Such an algorithm can be implemented by installing a spring and an MR damper connected in parallel in an engine mount. The spring should maintain the engine in static equilibrium and ensure the adequate natural frequency, slightly below the idle speed of the shaft. The design structure of the MR damper should be such that its stiffness is low and the damping ratio is easily switched between a very large and very small value.

4.2. Stiffness control

Stiffness reduction results in a decrease of the amplitudes of particular harmonics. However, when the total stiffness is reduced, the static equilibrium position of the engine will be changed which is unacceptable in the context of the drive system performance. Variation of the dynamic stiffness (i.e. the change of the stiffness coefficient in the vibration equation only, without changing the coefficient in the static equilibrium equation) can be accomplished by an indirect method, by connecting an actuator of the control system, parallel to the spring. Accordingly, the vibration equation will be rewritten as:

$$M\ddot{y}+b\dot{y}+ky=F_{ind}(t)+F_{con}(t) \tag{6}$$

In order that the control algorithm should lead to a change in the mount's dynamic stiffness, the actuator interaction force has to be dependent on the coordinate y, representing the engine vibration with respect to the static equilibrium position and be expressed by the formula: $F_{con} = \alpha\, y$. The vibration equation will be rewritten as:

$$M\ddot{y}+b\dot{y}+(k-\alpha)y=F_{ind}(t) \tag{7}$$

Accordingly, the effective stiffness coefficient being the difference of $(k - \alpha)$ will decrease in comparison to a stiff spring as long as the gain factor α in the control system is positive. This means that a system is required with positive feedback from the coordinate of displacement with respect to the static equilibrium position. This algorithm can only be easily implemented in active systems.

5. Application of squeeze-mode MR damper to engine vibration control

Further considerations are focused on applications of dampers filled with MR fluid and operated in squeeze mode when the damper is to be used as an actuator in a semi-active vibration control system for a combustion engine. The conceptual design of a MR damper operated in the squeeze mode is outlined in [8, 9]. In this operation mode of the MR fluid in the damper, its characteristic will differ from that of commonly employed MR dampers operating in shearing mode. The family of MR damper characteristics obtained for various current levels in the control coil inducing the magnetic field is shown in Fig. 6.

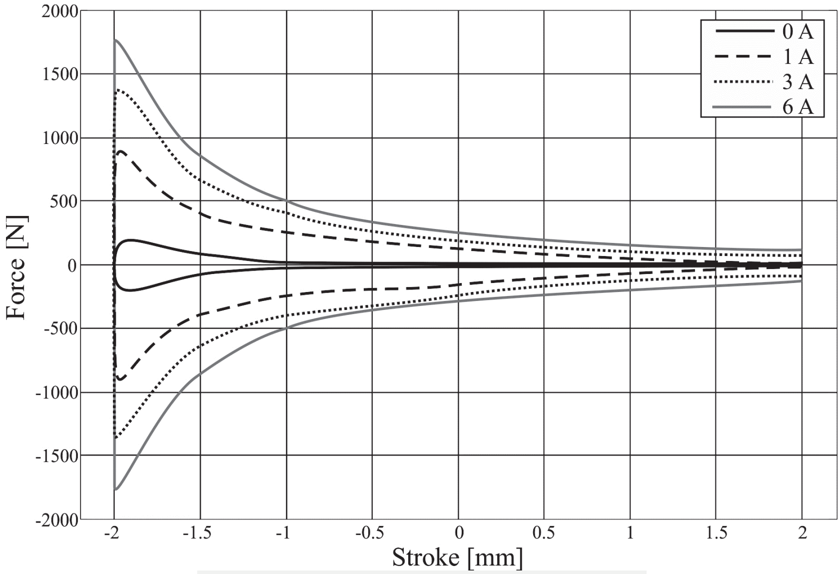

Fig. 6. Characteristics of an MR damper in the squeeze mode

Characteristics plotted in Fig. 6 show the damping force during the preset piston movement with respect to the housing under the applied sine displacement input with an amplitude of 2 mm and a frequency of 50 Hz. The selected input is similar to that experienced during steady-state damper operation in engine mounts. It is apparent that the MR damper is an energy-dissipating element and does not exhibit elastic properties. The distinctive feature is a major asymmetry of the damper performance pattern under compression and tension. This is a most undesirable feature in the context of the projected damper's performance in vibration control systems. The application of an asymmetric spring or damper causes that the vibrating system oscillates around the position shifted with respect to the static equilibrium point To avoid this effect, the mount design has to be modified such that it incorporates two dampers connected in parallel and operated in such a way that the piston motion towards the inside of the first damper should coincide with the piston motion in the opposite direction in the second damper, and vice versa. Such operation of dampers is possible when dampers are located in opposite sites of the vibrating element. As a result, the characteristic for the pair of dampers will be symmetrical (Fig. 7).

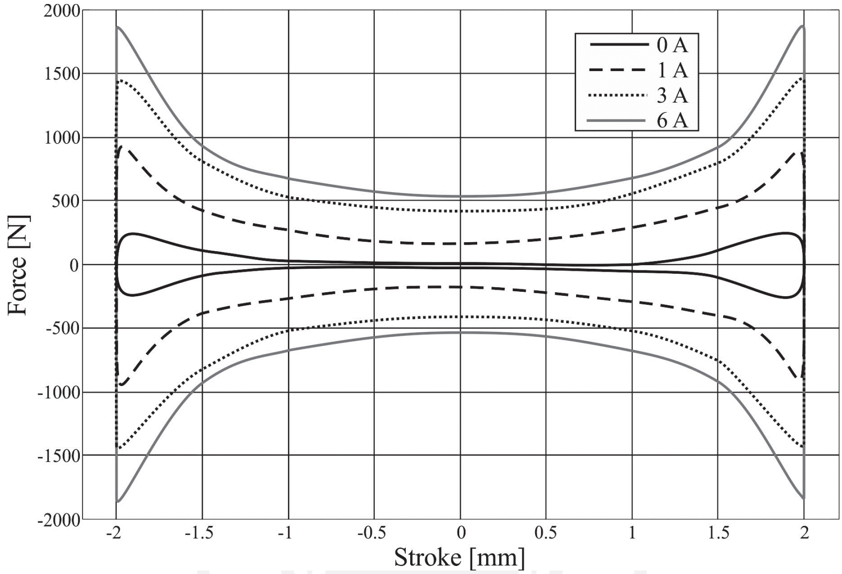

Fig 7. Resultant characteristics of two dampers connected in parallel

The characteristics shown in Fig. 7 are used for estimating the feasibility of damping control in the engine mount with the use of the two-damper system. Equivalent, dimensionless damping coefficients were calculated for the following current levels: 0, 1, 3, 6 A. The equivalent dimensionless coefficients were determined through comparing energy dissipated by an MR damper within one period of vibration with energy dissipated by a viscous damper operating under the same conditions. The calculations were performed for a two-cylinder engine with a mass of $M = 65$ kg. The stiffness coefficient of the mount was $k = 4 \times 10^5$ N/m. The natural frequency was slightly lower than the angular idle speed. The application of the viscous dampers whose characteristics are shown in Fig. 7 yields the equivalent dimensionless damping coefficients in the range 0.1 to 1.2. Graphs in Fig. 5 suggest that such dimensionless damping coefficients enable an effective damping control in accordance with the objectives of the control algorithms mentioned in earlier sections.

6. Conclusions

Research investigations summarised in this study underpin the feasibility analysis of using MR dampers in engine mounts. The calculations were restricted in the case where the engine vibrations were induced by inertia forces of the crankshaft-piston mechanism. Based on the results, in particular the formulas expressing the amplitudes of engine-car body interaction force, the principle of vibration control is formulated. Damping control can be successfully effected with the use of MR dampers. However, the quality of control depends on the damper design structure. It appears that effective control requires an angular velocity signal from the crank shaft in the engine. This signal is measured with sensors in the engine control system and can be effectively used in the MR damper control system. This first analysis of MR damper characteristics obtained for the squeeze mode and in the given configuration reveals their adequacy for use in the semi-active vibration isolation of the engine. The application of semi-active elements to mount stiffness control is a more complex task and active systems seem to be the recommended solutions. However, they might not prove cost-effective due to the high power demand of actuator systems while implementing the positive feedback control algorithms.

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