ISSN 1392 - 1207. MECHANIKA. 2007. Nr.4(66)

The arrangement scheme and its influence on vibration process of the vehicle piston engine

A. Vasilyev, K. Dolgov
Volgograd State Technical University, Lenin Avenue 28, 400131 Volgograd, Russia, E-mail: vasilyev@vstu.ru

1. Introduction

As it is known, engine balancing is defined either by cylinder and cranks arrangement or by using special balancing mechanisms, which increase mass dimension parameters of the construction. The cylinder arrangement in space relative to the vertical plane is defined by the vee cylinder angle, which has direct influence on the balancing parameters and, consequently, on vibrations of the vehicle piston engine as well.

In the work [1] on the basis of a generalized calculation method of nominal balancing and mass parameter engine and a computer program has been created, on the basis of which a six-cylinder engine has been analyzed and the calculation of its main mass dimension indices has been performed. The obtained results permit to choose the most acceptable arrangements according technical task. However it is of great importance to define the influence of vibrations excited by the given power units on vehicle suspension. In connection with this analysis of vibrations of six-cylinder arrangements, installed on a vehicle frame, is executed.

2. Mathematic model and results

This article deals with a generalized mathematical model of engine vibrations of a vehicle (Fig. 1), which has a number of specific peculiarities: it takes into account the foundation flexibility; unbalanced force factors are brought to the centre of engine masses and are presented in a generalized analytical form in accordance with [1, 2]. Thereby the definition of forces, acting on its supports is provided.

The following main assumptions are accepted: actual masses in a power unit and Coriolis inertia forces, generated by its moving parts are not taken into account; there is no elastic deformations of the power unit the vehicle frame, on which the engine is installed, is absolutely rigid; elastic characteristics of power unit supports and support resistance forces, damping the power unit vibrations are linear. Small steady state harmonic vibrations of the engine under the effect of unbalanced force factors are considered.

The dynamic model «engine – vehicle frame» has twelve DOF (linear and angular to the axes C_1X_1, C_1Y_1, C_1Z_1 and C_2X_2, C_2Y_2, C_2Z_2). X_1, Y_1, Z_1, φ_1, ψ_1, χ_1 and X_2, Y_2, Z_2, φ_2, ψ_2, χ_2 are the generalized coordinates – the engine and the moving frame correspondingly.

The case of line engine arrangement relative to the vehicle foundation is considered. If it is turned (relatively to the axis C_1Y_1 or C_1Z_1), that is, the coordinate system connected with the engine, which does not coincide with the coordinate system connected with the frame, then the corresponding force factors can be easily determined according the well-known dependencies in the new system of coordinates.

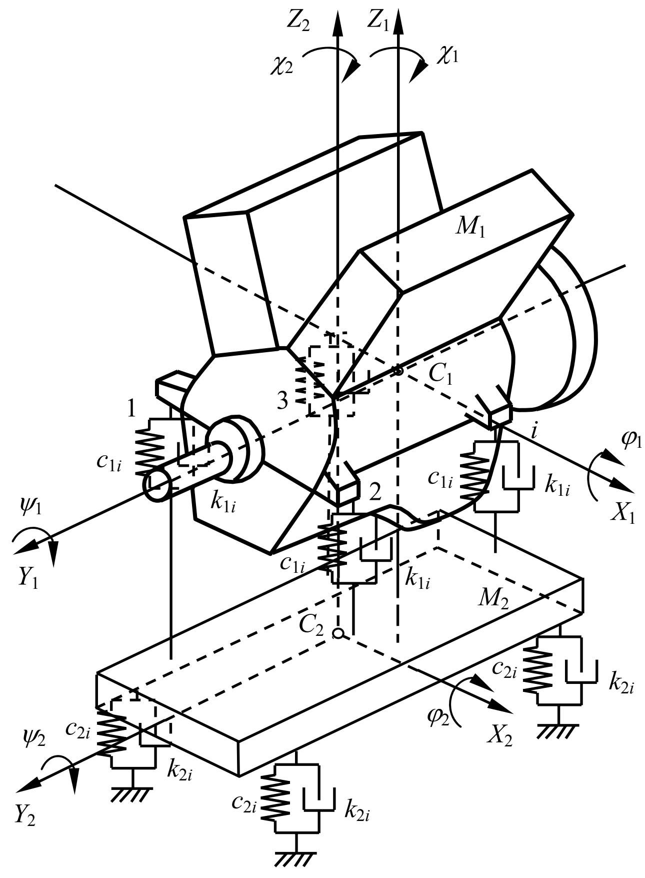

Fig. 1 Dynamical model of an engine on elastic frame

Taking into account the d'Alembert principle and the expressions of acting unbalanced force factors, differential equations vibrations of the arrangement on moving vehicle frame have been composed, the solution of which is performed by the numerical method of Runge Kutta

$$\left.\begin{aligned}
&M_1\ddot{Z}_1 = R_{z_1} + P_z - M_1 g;\\
&M_1\ddot{Y}_1 = R_{y_1} + P_y;\\
&M_1\ddot{X}_1 = R_{x_1} + P_x;\\
&J_{x_1}\ddot{\varphi}_1 = M_{x_1}^{c_1} + M_{x_1} = M_x + M_{x_1}^{c_1}(P) + M_{x_1};\\
&J_{y_1}\ddot{\psi}_1 = M_{y_1}^{c_1} + M_{y_1} + M_{\text{кр}} = M_y + M_{y_1}^{c_1}(P) + M_{y_1} + M_{\text{кр}};
\end{aligned}\right\} \quad (1)$$

$$\left.\begin{aligned}
&J_{z_1}\ddot{\chi}_1 = M_{z_1}^{c_1} + M_{z_1} = M_z + M_{z_1}^{c_1}(P) + M_{z_1};\\
&M_2\ddot{Z}_2 = -R_{z_1} + R_{z_2} - M_2 g;\\
&M_2\ddot{Y}_2 = -R_{y_1} + R_{y_2};\\
&M_2\ddot{X}_2 = -R_{x_1} + R_{x_2};\\
&J_{x_2}\ddot{\varphi}_2 = M'_{x_2} + M_{x_2};\\
&J_{y_2}\ddot{\psi}_2 = M'_{y_2} + M_{y_2};\\
&J_{z_2}\ddot{\chi}_2 = M'_{z_2} + M_{z_2},
\end{aligned}\right\} \qquad (1)$$

where M_1, M_2 are the engine and the frame masses correspondingly; g is acceleration of gravity; R_{x_1}, R_{y_1}, R_{z_1}, R_{x_2}, R_{y_2}, R_{z_2} are components of the reactions, arising at the engine and the frame supports; P_x, P_y, P_z, M_x, M_y, M_z are the main vector and the main moment components of the system; J_{x_1}, J_{y_1}, J_{z_1}, J_{x_2}, J_{y_2}, J_{z_2} are the main central inertia moments of the engine and of the vehicle frame; $M_{x_1}^{c_1}(P)$, $M_{y_1}^{c_1}(P)$, $M_{z_1}^{c_1}(P)$ are the moment components $M^{c_1}(P)$ of the main vector P about the masses centre C_1, applied in the old reduction center O; M_{x_1}, M_{y_1}, M_{z_1}, M_{x_2}, M_{y_2}, M_{z_2} are projections of the moments from total reactions at the engine and the frame supports; M_{kp} is the engine torque, acting in its longitudinal axis direction; $M_{x_2}{}'$, $M_{y_2}{}'$, $M_{z_2}{}'$ are projections of the moments from total reactions in the engine supports, influencing on the vehicle frame.

In Fig. 2 the ith engine support, located in the positive quadrant, with the acting reaction components is shown. They are defined as the difference of elasticity forces (product of rigidity and deformation) and internal friction in the supports and in the system of coordinates $C_1X_1Y_1Z_1$ are expressed as

$$\left.\begin{aligned}
&R_{x_{1i}} = \{c_{x_{1i}}[(-X_1 - z_{1i}\psi_1 + y_{1i}\chi_1) - (-X_2 - z_{2i}\psi_2 + y_{2i}\chi_2)] -\\
&-k_{x_{1i}}[(\dot{X}_1 + z_{1i}\dot{\psi}_1 - y_{1i}\dot{\chi}_1) - (\dot{X}_2 + z_{2i}\dot{\psi}_2 - y_{2i}\dot{\chi}_2)]\};\\
&R_{y_{1i}} = \{c_{y_{1i}}[(-Y_1 - x_{1i}\chi_1 + z_{1i}\varphi_1) - (-Y_2 - x_{2i}\chi_2 + z_{2i}\varphi_2)] -\\
&-k_{y_{1i}}[(\dot{Y}_1 + x_{1i}\dot{\chi}_1 - z_{1i}\dot{\varphi}_1) - (\dot{Y}_2 + x_{2i}\dot{\chi}_2 - z_{2i}\dot{\varphi}_2)]\};\\
&R_{z_{1i}} = \{c_{z_{1i}}[(-Z_1 - y_{1i}\varphi_1 + x_{1i}\psi_1) - (-Z_2 - y_{2i}\varphi_2 + x_{2i}\psi_2)] -\\
&-k_{z_{1i}}[(\dot{Z}_1 + y_{1i}\dot{\varphi}_1 - x_{1i}\dot{\psi}_1) - (\dot{Z}_2 + y_{2i}\dot{\varphi}_2 - x_{2i}\dot{\psi}_2)]\},
\end{aligned}\right\} (2)$$

where $c_{x_{1i}}, c_{y_{1i}}, c_{z_{1i}}$ are linear rigidities of the ith suspension element along the axes C_1X_1, C_1Y_1, C_1Z_1; x_{1i}, y_{1i}, z_{1i} are the distances from the ith engine support to its mass centre; x_{2i}, y_{2i}, z_{2i} are the distances from the ith vehicle frame support to its masses centre; $k_{x_{1i}}, k_{y_{1i}}, k_{z_{1i}}$ are damping factors of the ith engine suspension element along the axes C_1X_1, C_1Y_1, C_1Z_1.

If number of the engine suspension supports is equal n, then rigidities and each element coordinates are designated by indexes from 1 to n.

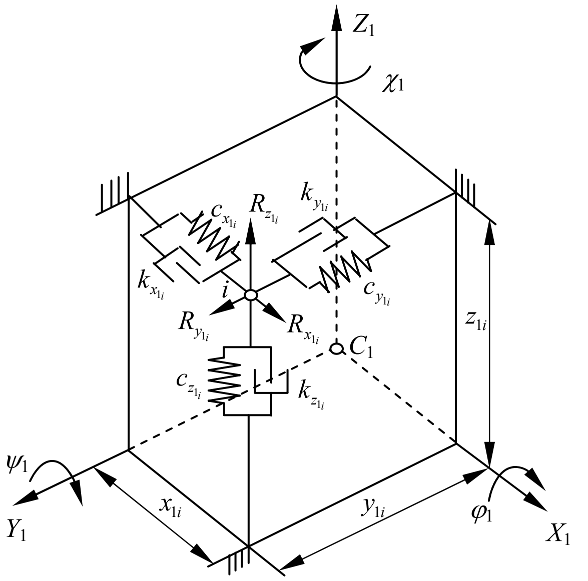

Fig. 2 Definition of the support reaction components in ith point of the engine suspension

Then the components of reactions resultant, originating in the engine supports, are

$$\left.\begin{aligned}
&R_{x_1} = \sum_{i=1}^{n} c_{x_{1i}}[(-X_1 - z_{1i}\psi_1 + y_{1i}\chi_1) - (-X_2 - z_{2i}\psi_2 + y_{2i}\chi_2)] -\\
&-k_{x_{1i}}[(\dot{X}_1 + z_{1i}\dot{\psi}_1 - y_{1i}\dot{\chi}_1) - (\dot{X}_2 + z_{2i}\dot{\psi}_2 - y_{2i}\dot{\chi}_2)]\};\\
&R_{y_1} = \sum_{i=1}^{n} \{c_{y_{1i}}[(-Y_1 - x_{1i}\chi_1 + z_{1i}\varphi_1) - (-Y_2 - x_{2i}\chi_2 + z_{2i}\varphi_2)] -\\
&-k_{y_{1i}}[(\dot{Y}_1 + x_{1i}\dot{\chi}_1 - z_{1i}\dot{\varphi}_1) - (\dot{Y}_2 + x_{2i}\dot{\chi}_2 - z_{2i}\dot{\varphi}_2)]\};\\
&R_{z_1} = \sum_{i=1}^{n} \{c_{z_{1i}}[(-Z_1 - y_{1i}\varphi_1 + x_{1i}\psi_1) - (-Z_2 - y_{2i}\varphi_2 + x_{2i}\psi_2)] -\\
&-k_{z_{1i}}[(\dot{Z}_1 + y_{1i}\dot{\varphi}_1 - x_{1i}\dot{\psi}_1) - (\dot{Z}_2 + y_{2i}\dot{\varphi}_2 - x_{2i}\dot{\psi}_2)]\}
\end{aligned}\right\} (3)$$

Similarly projections of the moments from total reactions in the engine supports and the frame are defined, and also projections of the moments from total reactions in the engine supports, acting on the vehicle frame.

Theoretical research of forced vibrations of six-cylinder and four-stroke V-type engines with the vee angles of $\gamma = 10°$, $20°$, $30°$, $60°$, $90°$, $120°$, $150°$, $180°$ on a steady vehicle frame has been carried out with the aim to obtain a comparative estimation of the vibrations excited by these engines from the torque variation and the moments of inertial forces of progressive actual masses (actual masses of the first M_{j1} and the second M_{j2} orders. It is known, that in those arrangements the moment of inertia forces of revolving masses M_c is not balanced either. However this given force factor is not considered in the calculations due to its balancing by the system of counterweights.

Besides that, in the six-cylinder V-engine with $\gamma = 90°$ balancing M_{j1} is obtained together with M_c, that is why it is not in the equations of forced vibrations; in the six-cylinder V-engine arrangement with $\gamma = 180°$ the moment of inertia forces of progressive moving masses of the second order is balanced. From the point of view of nominal balancing the arrangement with the vee angle $\gamma = 60°$ is of interest because in it the expression M_{j2} does not depend on the angle of action of the crankshaft.

As a basic arrangement, when calculating the vibrations, a V-type engine with the vee angle $\gamma = 90°$ and the cranks, arranged at an angle of 120°, has been used. It is common, that all the analyzed arrangements with various vee angles have the same power, mass, crankshaft construction, space arrangement of suspension elements and its main parameters are the same as those of the basic engine. The values of the main central inertia moments of the engines have been defined according to the calculation method. The given method presents calculation of inertia moments of the engine at different vee angles.

In Fig. 3 the force moment curve of gas pressure is presented for one cylinder. When analyzing the torques from the gas pressure force and inertia forces a great influence on the total torque curve have gas pressure forces.

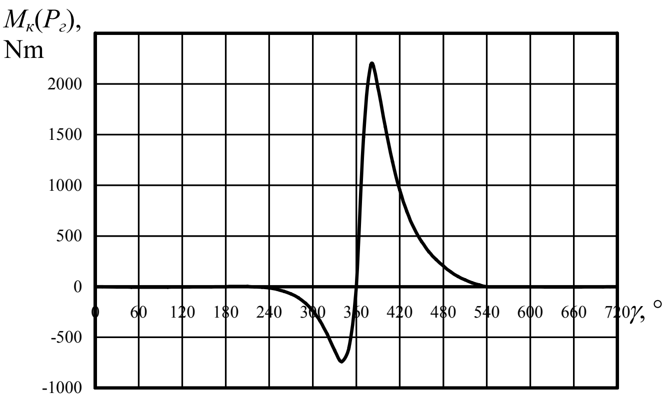

Fig. 3 Moment of gas pressure forces for one cylinder

A harmonious analysis of the torque curve of the basic engine at the nominal regime showed, that the components of the first and a half, the third and the fourth and a half orders are the most intensive. When carrying out the harmonious analysis of the torque curve of the six-cylinder V-engine with $\gamma = 60°$ it has been exposed that, in the spectrum of resolution the harmonics of the first and a half, the fourth and a half and the sixth motor orders are the most intensive.

In Table 1 cosine and sine harmonic factors A_k, B_k of the expansion torques total curves from the gas pressure force for six-cylinder V-type engines with $\gamma = 60°$ and 90°, and also amplitude values corresponding harmonics C_k and phase angles δ_k are shown.

From the analysis of the moments from vertical and horizontal components of inertia forces due to moving masses of the first order, acting in the six-cylinder V-engines with the vee angle $\gamma = 10°$, 20°, 30°, 60°, 120°, 150°, 180°, it has been shown, that in contrast to the six-cylinder V-engine with $\gamma = 90°$, they depend on the angle of the crankshaft action. Consequently, it is impossible completely to balance them by the counterweights on the crankshaft. In this case (e.g. the arrangement with $\gamma = 60°$) the moments, acting in vertical and horizontal planes, are included into resultant moment, not depending on the angle of crankshaft action, and the moment, acting in vertical plane. For their balancing two systems are used. One system presents usual counterweights on the crankshaft, forming the moment of centrifugal forces, and the other consists of two shafts, rotating in opposite direction with angular frequency of the crankshaft, and creating a balancing moment, acting in vertical plane opposite in phase of the component of inertia forces moment of progressive actual masses of the first order.

Table 1

Parameters of the arrangements harmonious analysis

γ, °	The order of the motor harmonic	A_k, N·m	B_k, N·m	C_k, N·m	δ_k, °
60	1.5	667.8	-1037.9	1234.2	147.2
	4.5	-232.8	-436.8	495	208.1
	6	-197.1	363.8	413.7	331.5
90	1.5	548.6	-380.4	667.6	124.7
	3	534.2	622.8	820.6	40.6
	4.5	409.9	-499.6	646.2	140.6

In Tables 2, 3 the data, necessary for the calculation of basic arrangement vibrations are shown, part from which is taken from the reference [3].

The vibrations of investigated power units are double coupled in pairs, since the suspension supports are located symmetrically relative to the planes $Y_1C_1Z_1$ and $X_1C_1Y_1$ (Fig. 1). In this case the centre of rigidity and the centre of gravity lie on the same axis (the axis of the least inertia moment of the power unit C_1Y_1). Thereby uncoupled are vibrations of the engine about and along the axis C_1Y_1. The vibrations along the axis C_1X_1 and about the axis C_1Z_1 are coupled in pairs, as well as about the axis C_1X_1 and along the axis C_1Z_1.

The engines, installed on the suspension, consisting of four supports, have a sufficiently low frequency of vibrations in the direction of torque effect. The incoherence of vibrations about the axis C_1Y_1 are a favourable factor for them, since the frequency decrease of vibrations about this axis does not cause the change of other frequencies of engine vibrations [3].

As a result of carrying out research of vibrating process of six-cylinder engines with vee angles $\gamma = 10°$, 20°, 30°, 60°, 90°, 120°, 150°, 180° it has been established, that according to the arrangement scheme of supports, the chosen rigidities and damping factors, the given power units perform linear transversals X_1 and angular vibrations (ψ_1 and χ_1) relative to the longitudinal Y_1 and the vertical Z_1 axes.

By the analysis of the presented diagrams it is shown, that maximum amplitudes of forced vibrations (Figs. 4, 5) are obtained in the six-cylinder arrangements with $\gamma = 60$, 90° and 150°, for which the greatest unevenness of flash alterations is characteristic. So, the resonance amplitude of transversal vibrations X_1 is 0.58 mm (the arrangement with $\gamma = 90°$), but the maximum amplitudes of angular ones (ψ_1 and χ_1) are 0.495 and 0.108 degrees correspondingly. Vibroactivity of six-cylinder engines with vee angles $\gamma = 10°$, 20°, 120°, 180° is sufficiently low. For instance, the resonance amplitude of transversal vibrations X_1 of six-cylinder arrangement with $\gamma = 10°$ is 0.54 mm,

Table 2

The main characteristics of the arrangements

The main characteristics	The basic arrangement (γ = 90°)			The six-cylinder engine (γ = 60°)		
1. Engine mass, kg	1220			1220		
2. Position of inertia centre a) in vertical plane	0.11 m from the forward support up; 0.672 m from the forward support along of the axis of the crankshaft			0.128 m from the forward support up; 0.674 m from the forward support along of the axis of the crankshaft		
b) in horizontal plane	0.345 m from the axis of the lateral supports of the engine			0.345 m from the axis of the lateral supports of the engine		
3. Inertia moments of the engine about axes, intersecting the inertia centre, kg·m^2	J_{x_1} = 258 J_{y_1} = 71.6 J_{z_1} = 226			J_{x_1} = 268.22 J_{y_1} = 65.3 J_{z_1} = 221.14		
4. Coordinates of elastic supports of the engine in the reference system with the origin in its inertia centre, m	X_1	Y_1	Z_1	X_1	Y_1	Z_1
a) forward	0	0.586	-0.286	0	0.586	-0.286
b) lateral right	0.345	-0.19	0.061	0.345	-0.19	0.061
c) lateral left	-0.345	-0.19	0.061	-0.345	-0.19	0.061
d) back	0	-1.054	0.122	0	-1.054	0.122

Table 3

The base parameters of engine suspension

Rigidities, N/m	The factors of damping suspension, N·cm/m
$c_{x_{11}} = 300\cdot10^3$	$k_{x_{11}} = 13.6\cdot10^3$
$c_{x_{12}} = 250\cdot10^3$	$k_{x_{12}} = 13.6\cdot10^3$
$c_{x_{13}} = 250\cdot10^3$	$k_{x_{13}} = 13.6\cdot10^3$
$c_{x_{14}} = 170\cdot10^3$	$k_{x_{14}} = 13.6\cdot10^3$
$c_{y_{11}} = 250\cdot10^3$	$k_{y_{11}} = 13.6\cdot10^3$
$c_{y_{12}} = 800\cdot10^3$	$k_{y_{12}} = 13.6\cdot10^3$
$c_{y_{13}} = 800\cdot10^3$	$k_{y_{13}} = 13.6\cdot10^3$
$c_{y_{14}} = 170\cdot10^3$	$k_{y_{14}} = 13.6\cdot10^3$
$c_{z_{11}} = 1500\cdot10^3$	$k_{z_{11}} = 13.6\cdot10^3$
$c_{z_{12}} = 800\cdot10^3$	$k_{z_{12}} = 13.6\cdot10^3$
$c_{z_{13}} = 800\cdot10^3$	$k_{z_{13}} = 13.6\cdot10^3$
$c_{z_{14}} = 500\cdot10^3$	$k_{z_{14}} = 13.6\cdot10^3$

but maximum amplitudes of angular ones (ψ_1 и χ_1) are 0.44 and 0.091 degrees. The given circumstance is accounted for a high evenness of the torque for the schemes with γ = 10° and 20° and a uniform flash alterations in the arrangement with γ = 120°.

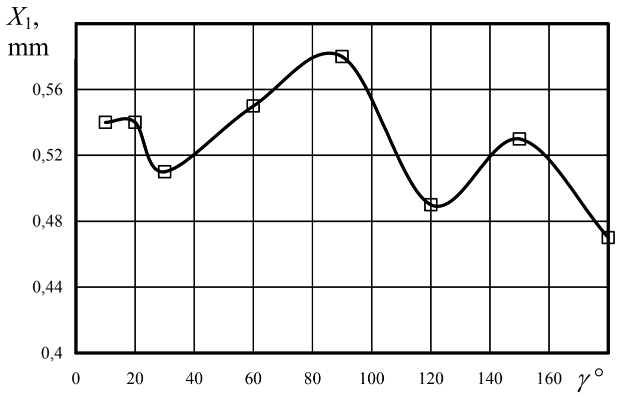

Fig. 4 The dependence of resonance amplitude of transversal vibrations X_1 of six-cylinder arrangements on vee angle of cylinders

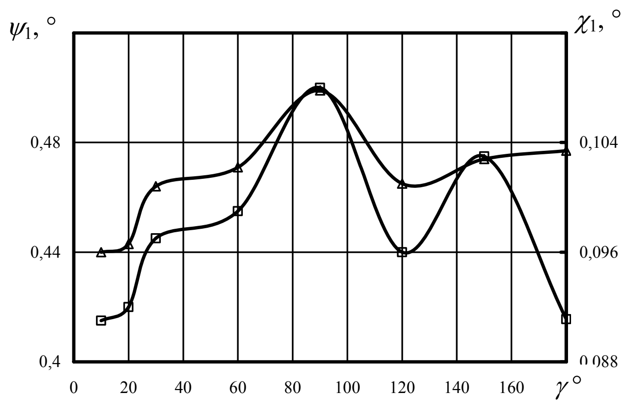

Fig. 5 The dependence of resonance amplitude of angular vibrations of the six-cylinder arrangements relative longitudinal ψ_1 (Δ), vertical χ_1 (□) axes on vee angle of cylinders

3. Conclusions

The method of influence analysis of the arrangement scheme on balancing and the level of engine vibrations of a vehicle is presented in this article. It permits to find out the arrangements with vee angular cylinders, having not only good data on balancing and acceptable mass dimension indices, but a minimal vibroactivity as well. On the example of six-cylinder V-type engines with the given parameters of the suspension the vee cylinder angles $\gamma = 10°$, $20°$, $120°$ satisfying the listed requirements have been defined. On the base of a generalized model of the engine system vibration in a vehicle, taking into account the modification of suspension parameters it is possible to carry out similar research for other engine sets.

References

1. **Grigoryev, E.A.** The influence of the number of cylinders and cranks on balancing and mass dimension parameters of engines /Grigoryev E.A., Vasilyev A.V., Dolgov K.O.//Enginebuilding, 2004, N 3, p.9-12 (in Russian).
2. **Grigoryev, E.A.** Periodic and Casual Forces Working in the Piston Engine.-Moscow: Machinebuilding, 2002.-272p. (in Russian).
3. Vibrations of a vehicle power unit /Tolskiy V.E., Korchemniy L.V., Latishev G.V., Minkin L.M.; edited by V.E. Tolskiy//Moscow: Machinebuilding, 1976.-266p. (in Russian).

A. Vasilyev, K. Dolgov

KOMPONAVIMO SCHEMA IR JOS ĮTAKA STŪMOKLINIO VARIKLIO VIRPESIAMS

R e z i u m ė

Straipsnyje pateikiama variklio komponavimo transporto priemonėje schemos įtakos jo darbo stabilumui ir virpesiams analizės metodika. Ji leidžia nustatyti V formos variklio komponuotę, kuri užtikrintų ne tik stabilų darbą, bet ir gerus masės ir matmenų indeksus bei minimalų virpesių lygį.

A. Vasilyev, K. Dolgov

THE ARRANGEMENT SCHEME AND ITS INFLUENCE ON THE VIBRATION PROCESS OF THE VEHICLE PISTON ENGINE

S u m m a r y

The method of influence analysis of the arrangement scheme on balancing and the level of engine vibrations of a vehicle is presented in this article. It permits to find out the arrangements with vee angular cylinders having not only good data on balancing and acceptable mass dimension indices, but a minimal vibroactivity as well.

А. Васильев, К. Долгов

КОМПОНОВОЧНАЯ СХЕМА И ЕЕ ВЛИЯНИЕ НА КОЛЕБАТЕЛЬНЫЙ ПРОЦЕСС ПОРШНЕВОГО ДВИГАТЕЛЯ ТРАНСПОРТНОГО СРЕДСТВА

Р е з ю м е

В данной статье представлена методика анализа влияния компоновочной схемы на уравновешенность и уровень колебаний двигателей в составе транспортного средства. Она позволяет выявить компоновки с углами развала цилиндров, обладающие не только хорошими данными по уравновешенности и приемлемыми массогабаритными показателями, но и минимальной виброактивностью.

Received May 15, 2007