Simulation of the force factors causing engine idling to vibrate

P V Safronov¹, I V Alekseev¹

¹ Moscow Automobile and Road Construction State Technical University (MADI), 64 Leningradski prospekt, Moscow 124319, Russia

E-mail: pavel_safronov@mail.ru

Abstract. The article provides a diagram and a description of the physical reasons that cause the engine to vibrate on the suspension at idle. The substantiation of the assumption about the inertial nature of these reasons is given. The main dependences for modeling the force effects on the engine casing at idle modes are given. The above theoretical aspects of modeling oscillations at idle speed are confirmed by the results of experiments to determine the influence of the moment of inertia of rotating engine parts on the amplitude of its oscillations on the suspension.

1. Introduction
Internal combustion engine is active source of vibrations in wide frequency range due to cyclical working processes and kinematic features of crank mechanism. As you know, the main force factor causing vibrations of the engine on the suspension is the variable overturning moment, which is the result of the force interaction between the engine and the external load. Variable inertia forces are another significant force factor. These factors take place at high speed and load conditions [1]. In the modes of minimum idle speed, these factors are minimal or equal to zero [2]. However, as practice shows, the vibration amplitudes of the engine on the suspension at idle speed can reach significant values. The analysis and modeling of the effect of inertial forces and torque on the vibrations of the engine on the suspension is widely presented in the works of many authors. In this case, the influence of friction forces in the main bearings of the crankshaft and the moment of inertia of rotating parts on engine vibrations on the suspension is practically not considered. However, at idle and low loads, this force factor is the main one, disturbing the vibrations of the engine on the suspension. This article is devoted to the consideration of the factors causing vibrations of the engine on the suspension at idle modes.

2. Modeling of inertial action
For consideration and further mathematical modeling of oscillations of the engine on the suspension, consider the scheme of a single-cylinder engine and the forces and moments acting on its elements, shown in Fig. 1. Stages of creation of mass-inertial model of engine on suspension are considered in detail in operation [3], [4].
Figure 1. Forces and moments acting on the elements of the internal combustion engine.

The equation of angular motion of the engine body (Fig. 1) can be written in the following form:

\[ M_R = M_{\text{overtur}} - M_{\text{i.body}} \]  (1)

where \( M_R = R_{\text{support}} \cdot A \) - moment from the elastic forces of the supports \( R_{\text{support}} \), \( M_{\text{overtur}} \) - overturning moment, \( M_{\text{i.body}} = I_{\text{body}} \frac{dy}{dt^2} \) - instantaneous inertial moment of the case, \( I_{\text{body}} \) - reduced to the crankshaft axis moment of inertia of the housing, \( \gamma \) - angular movement of the body around the longitudinal axis \( Z \).

The overturning moment acting on the body is the result of the action of two moments of forces:

\[ M_{\text{overtur}} = N \cdot h - M_{\text{m.bear}}. \]  (2)

where \( M_{\text{m.bear}} \) - moment from friction forces in the main bearings. Its value is approximately 1.5 ... 1.9 Nm for one main bearing at a speed of 1000 rpm at idle [5], \( N \) - lateral force, which is a projection onto an axis perpendicular to the cylinder axis, reactions from the connecting rod to forces acting on the piston \( N = (P_g + P_j + P_{fr}) \cdot tg(\beta) \), [6] \( h \) - arm of force \( N \) (distance from the axis of the crankshaft to the axis of the piston pin), \( \beta \) - is the angle of deflection of the connecting rod.

Substituting expression (2) into (1) and carrying out transformations, we obtain:

\[ M_R = N \cdot h - M_{\text{m.bear}} - I_{\text{body}} \cdot \frac{dy}{dt^2} \]  (3)

Usually, when analyzing the dynamics of an internal combustion engine, friction forces are ignored. This approach is justified when considering the nominal regime. For this mode, the cycle average value \( M_{\text{m.bear}} \) is only a few percent of the average \( Nh \) value and practically does not affect
the results of calculating the forces and moments acting on the motor housing. In this case, the instantaneous values of the loads on the engine mounts will depend on the value of the overturning moment \( N_h \), as well as the moment of inertia of the body. As the engine load decreases, the average \( N_h \) value decreases and becomes equal to the average value \( M_{m, bear_{\text{idle}}} \) at idle. However, the instantaneous value of the load on the engine mounts, in this mode, will depend on the ratio of \( N_h \) and \( M_{m, bear_{\text{idle}}} \) at a particular moment of time, as well as the value of \( I_{\text{body}} \). Despite the equality of the cycle mean values of \( N_h \) and \( M_{m, bear_{\text{idle}}} \), at an arbitrary time instant \( N_h \neq M_{m, bear_{\text{idle}}} \). This is due to the difference in the main factors influencing them. The magnitude of the frictional moment in the main bearings is mainly influenced by the angular velocity of the crankshaft \( (M_{m, bear_{\text{idle}}} = f(\omega)) \). The value of \( N_h \) is mainly influenced by the force of gas pressure \( P_g \) and the angle of rotation of the crank. Figure 2. presents the results of experimental determination of the indicated lateral force and angular speed of the crankshaft for a four-cylinder gasoline engine operating at idle speed. The lateral force was calculated from the results of measuring the pressure in each cylinder after one degree of rotation of the crankshaft and the calculated values of the inertia forces from the masses of parts moving back and forth. The measurement of the angular velocity and pressure in the cylinders was synchronized with each other and with the position of the pistons.

\[ M_{cr} = M_{rez} - I_{cr} \frac{d\omega}{dt} \]  

(4)

where \( M_{cr} \) – moment created by force \( T \) (Fig.1) on the crankshaft, \( M_{rez} \) – total moment of resistance to crankshaft rotation, \( I_{cr} \) – the moment of inertia of the moving parts of the crank mechanism reduced to the axis of rotation of the crankshaft. Considering that the moment of inertia of

Figure 2. Results of simultaneous determination of the indicated lateral force \( N_i \) and the instantaneous angular velocity of the crankshaft \( \omega \).

Presented in Fig. 2. \( N_i \) force is called indicator force, since when determining it, the friction of the piston and rings on the cylinder wall was not taken into account.

The graph shows that the maximum lateral force is reached 40...50 degrees of crankshaft rotation earlier than the maximum angular velocity.

The movement of the crankshaft is generally described by the following equation:

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rotating parts is usually 90...95\% [2] of the total reduced moment of inertia, \( I_{cr} = \text{const.} \) can be considered. 

\[ M_{cr} = T \cdot r, \]  
where \( r \) – crank radius, \( M_{rez} = M_{ext.rez} + M_{m \cdot \text{bear.}} \), where \( M_{ext.rez} \) – moment of external load.

Transforming expression (4), we obtain an expression for the moment of friction forces in the main bearings:

\[ M_{m \cdot \text{bear.}} = T \cdot r - M_{rez} - I_{cr} \cdot \frac{d\omega}{dt}, \]  
(5).

Substituting the resulting expression in (3) we get:

\[ M_{R} = N \cdot h - T \cdot r + M_{rez} + I_{cr} \cdot \frac{d\omega}{dt} - I_{\text{body}} \cdot \frac{dy}{dt^2}, \]  
(6).

From expression (6) it follows that the magnitude of variable loads on the engine supports, when operating under load, will be determined by the amplitude of the value \( Nh \). When the engine is idling \( (M_{rez} = 0) \), the magnitude of variable loads on the engine mounts will be determined by the acceleration of the crankshaft \( \frac{d\omega}{dt} \), as well as the ratio of the moments of inertia of the engine body and the reduced moment of inertia of the moving parts \( I_{cr} \).

3. Results of experimental determination of engine vibration parameters at different flywheel inertia moments

The influence of the flywheel moment of inertia on engine vibrations is confirmed by the experiments carried out to measure the vibration accelerations of the supports of a VAZ-2111 in-line four-cylinder engine installed on a test bench when it is idling with and without a flywheel. In this case, the moment of inertia of all moving parts, including the flywheel, reduced to the crankshaft axis, is \( I_{cr} = 0,091 \) kg\( \cdot \)m\(^2\), and the moment of inertia of the flywheel itself is \( I_{\text{flywheel}} = 0,075 \) kg\( \cdot \)m\(^2\). Moment of inertia of engine housing parts was \( I_{\text{body}} = 0,128 \) kg\( \cdot \)m\(^2\). In order to separate the frequencies of the disturbing factors caused by the overturning moment and other force effects, the unevenness of the rotational speed was artificially increased by eliminating the ignition in one of the cylinders. At an average crankshaft rotation speed of 1000 rpm, the period of the disturbing factor caused by a misfire is two crankshaft revolutions and has a frequency of 8,4 Hz (Fig. 3 and Fig. 4). From the graphs in these figures it can be seen that the value of vibration accelerations \( j \) of the attachment point of the left engine support in the direction of the cylinder axis (\( Y \)-axis in Fig. 1), at the indicated frequency, in the absence of a flywheel, is almost 2 times less than in its presence. Thus, the moment of inertia of the flywheel creates an external inertial load on the engine. The variable nature of this load, which is a consequence of the unevenness of the rotational speed, causes the engine to vibrate on an elastic suspension. The amplitude of these vibrations is directly proportional to the moment of inertia of the moving parts, reduced to the axis of rotation of the crankshaft, and the value of the angular velocity irregularity.
Figure 3. The spectrum of accelerations of the left engine support in the direction of the Y axis when the engine is running without ignition in one of the cylinders, complete with a flywheel ($I_{cr} = 0.091 \, \text{kg} \cdot \text{m}^2$).

Figure 4. The spectrum of accelerations of the left engine support in the Y-axis direction when the engine is running without ignition in one of the cylinders in the configuration without a flywheel ($I_{cr} = 0.016 \, \text{kg} \cdot \text{m}^2$).

4. Conclusion
1. The main reason for the occurrence of engine oscillations at idle is a mismatch in the time of change in the moment from the lateral force and friction coil in the main bearings.
2. The value of engine oscillations on the suspension in this case is directly proportional to the ratio of moments of inertia of the engine and its flywheel.
3. Reducing the reduced moment of inertia of the engine rotating masses from $I_{cr} = 0.091 \ \text{kg} \cdot \text{m}^2$ to $I_{cr} = 0.016 \ \text{kg} \cdot \text{m}^2$ resulted in a decrease in vibration acceleration of the engine support by almost two times (from $8.9 \ \text{m/s}^2$ to $4.6 \ \text{m/s}^2$).

4. When simulating engine oscillations on the suspension, it is necessary to take into account not only the frequency characteristics of power effects on the engine, but also their relative phase displacements.

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