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EXPERIMENTAL RESEARCH AND IMITATION SIMULATION OF THE DYNAMICS OF GENERAL-PURPOSE VALVE MECHANISM OF CAR ENGINE

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Abstract. The author specifies the technique of estimating internal combustion engine of valve mechanism on the example of VAZ engine, obtained by means of developed simulation model of studying valve train dynamics research. Experimental research of valve spring coil oscillations by high-speed motion-picture technique is being considered as well.

Key words: valve spring coil oscillations, stress loading, mathematical model, gas distribution mechanism, internal combustion engine, experimental research, high-speed motion-picture technique.

1. Introduction

In modern engines the presence of elastic deformable links in valve train contributes to its oscillatory processes. Variable nature of loading, compression, tension, bending and torsion stresses that take place in the system, reduce reliability of the components. Valve train dynamics depends significantly on stiffness and damping properties of valve train elements and contact points of these elements. This effect is most tangible in valve spring being the element with the lowest stiffness and having the lowest natural frequency (as compared to other valve train parts).

At resonance (with respect to camshaft operating speeds) engine operating modes stress surges in valve spring occur which affect not only the loading of the spring itself, but can also be the cause of other poor performance of the valve train itself.

Thereby, development of general-purpose simulation model for valve train dynamics research adjusted for valve spring coil oscillations, that will allow us to describe all the processes existent inside valve train in a most accurate way and estimate its loading seems to be urgent.

2. Simulation model

The proposed valve train dynamics simulation method is based on generalized dynamic model (“Dynamics”, *D*) that was developed at Automobile and Tractor Engines Department of Volgograd State Technical University. Determination of forces in valve train, displacement values of its parts is based on representation of the latter in the form of discrete masses connected with inertialess elastic elements and then the numerical integration of differential equations describing displacement of each mass.

In base model each valve spring is described by 6-mass discrete model (one spring mass is fixed and the other is attached to valve mass) (Fig. 1, a). The developed simulation model provides for possibility to vary presentation of valve springs. In addition to multi-mass approach presentation of equivalent flexible rods model – concentrated masses with distributed parameters (so called “surge-mode approach model”) – able to perform longitudinal oscillations is realized as well. The admissibility of such a representation follows from a more precise determination theory of springs parameters, where coil spring is considered as a thin curved space bar. This approach takes advantage of the fact that the stress wave propagation through valve spring elastic media can be described in terms of normal modes that are decisive in valve train loading estimation. To model these oscillations by longitudinal vibrations of the rod the equality of spring mass and stiffness values corresponding to those of the rod should be maintained. This scheme application, apart from valve displacement specification, allows to estimate valve springs loadings themselves in a more accurate way [2].

Valve springs vibrations and their influence on valve train dynamics calculating program was implemented as a separate calculating module (***SPR***) that works jointly with base dynamics simulation model.

Valve spring forces affecting on valve were determined during valve spring coils oscillation numerical integration [1; 2] with corresponding initial and boundary conditions within ***SPR*** module. Valve spring coils oscillation equation has the form of wave equation with initial friction damping term:

$$\frac{\partial^2 U(\xi;\varphi)}{\partial \varphi^2} + \frac{2\mu}{\omega}\frac{\partial U(\xi;\varphi)}{\partial \varphi} = \left(\frac{a}{\omega}\right)^2 \frac{\partial^2 U(\xi;\varphi)}{\partial \xi^2}, \quad (1)$$

where U – longitudinal displacement of elastic rod’s cross section equivalent to that of the real valve spring, mm; μ – viscous friction damping coefficient (takes the value of 20…30 if not using external friction damper); ξ – effective length (ratio of distance from valve spring active length start point to the coil section under consideration to total valve spring length); φ – camshaft angle, rad; ω – camshaft operating speed, rad/s; a – stress wave propagation speed, s^{-1}.

It’s easy to use zero initial conditions when valve is closed and sits on valve seat; valve spring cross section displacements are also equivalent to zero. Boundary conditions are as follows: valve spring fastened end displacement is equivalent to zero and displacement of its moving end is determined by valve motion.

Developed valve train dynamic simulation technique is based on method of successive approximations that allows us to research valve spring dynamics at steady-state engine operating mode. Calculation of the first iteration began from static equilibrium state of valve spring coils. Camshaft angle origin corresponded to valve lift start time point provided valve train expansion gap is completely eliminated. The original variable $U(\xi;\varphi)$ – valve spring coil cross sections displacements – was determined by numerical integration of strain values $\eta(\xi;\varphi)$ along effective length ξ of the valve spring.

$$U(\xi;\varphi) = \left(\frac{\omega}{a}\right)\int_0^{\xi_i} \eta(\xi;\varphi_j)\,d\xi. \quad (2)$$

Calculation can be extended until it reaches the steady-state valve train operating mode, implemented within the iteration cycle, and periodic solution is found. The solution was considered steady and iteration process was stopped as soon as the difference between the initial iteration data and outcome dropped below the errors of calculation. Valve spring forces affecting on valve were determined upon reaching steady-state valve train operating mode

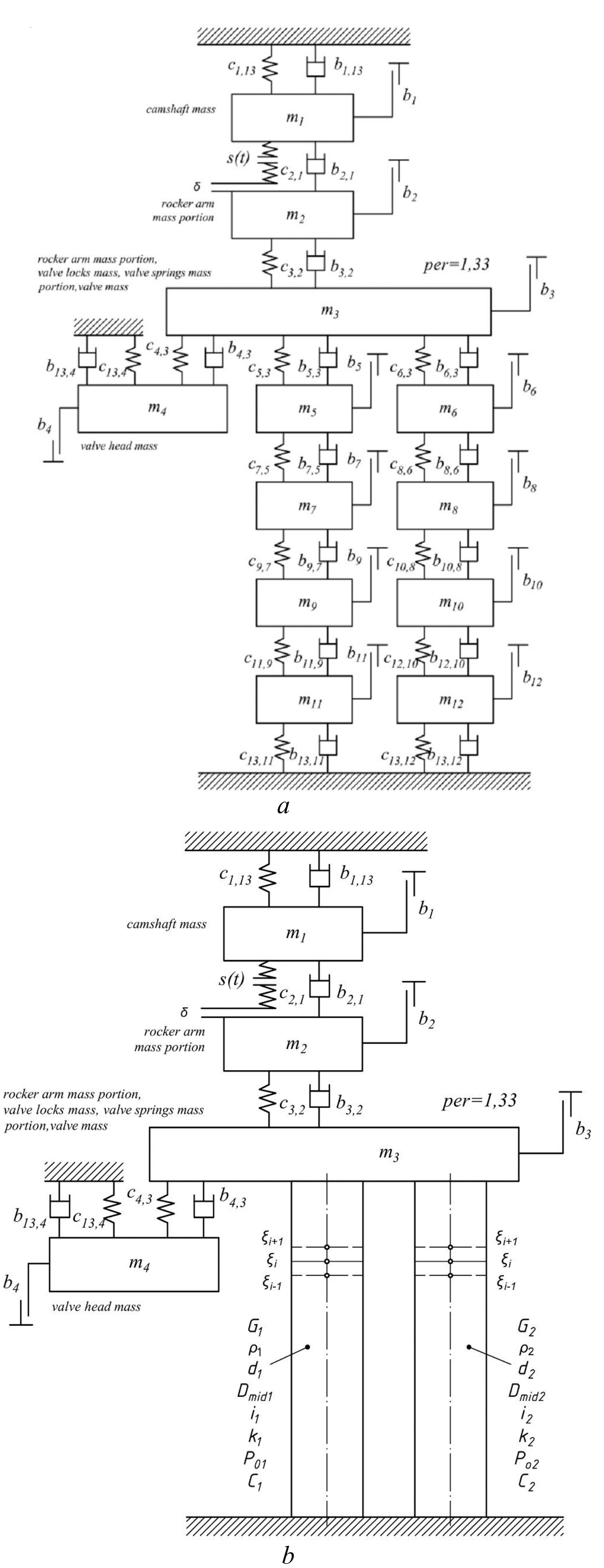


Fig. 1. Valve train dynamics simulation model design schemes:

a – base simulation model (valve spring are represented by discrete-mass chain);
b – proposed simulation model (valve spring are represented as elastic equivalent rods)

$$P_{ext.} = P_{ext.}(\xi,\varphi) = P_{0ext.} + c_{ext.}\frac{\omega}{a_{ext.}}\eta_{ext.}(\xi,\varphi);$$

$$P_{int.} = P_{int.}(\xi,\varphi) = P_{0int.} + c_{int.}\frac{\omega}{a_{int.}}\eta_{int.}(\xi,\varphi), \qquad (3)$$

where $P_{0ext.}$ and $P_{0int.}$ – outer and inner valve spring preload force values respectively, N; $c_{ext.}$ and $c_{int.}$ – outer and inner valve spring stiffness values respectively, N/mm; $a_{ext.}$ and $a_{int.}$ – outer and inner stress wave propagation speeds respectively, s^{-1}; $\eta_{ext.} = \left(\frac{a_{ext.}}{\omega}\right)\frac{\partial U_{ext.}}{\partial \xi}$ and $\eta_{int.} = \left(\frac{a_{int.}}{\omega}\right)\frac{\partial U_{int.}}{\partial \xi}$ – outer and inner valve springs strain value respectively, mm.

Thus, in relation to the foregoing task the interaction between base simulation model (“Dynamics”, ***D***) and developed calculating module (“Spring”, ***SPR***) within one calculation step of camshaft angle φ is based on determination of valve acceleration, velocity and displacement values as well as all the forces affecting on valve by Runge-Kutta numerical integration method applied for discrete masses displacements definition, assigning valve velocity value as a boundary condition of valve spring moving end to solute the wave equation (2) within ***SPR*** module, valve spring coils cross-sections displacement $U(\xi;\varphi)$ and its strain η $(\xi;\varphi_j)$ values definition, definition of valve springs forces P_{ext} and P_{int} affecting on valve and then transmitting them back to “Dynamics” (***D***) to calculate the actual valve train forces and estimate its loading.

Fig. 2 shows as an example obtained by calculation outer and inner valve spring first normal mode forces affecting on valve on camshaft rotating speed n = 2068 rpm. It’s easy to see that stress disturbances do not completely damp after valve seating and runs until the next valve lift.

3. Experimental research

In order to identify the simulation model proposed and examine adequacy of the ***SPR*** module, high-speed motion-picture technique series of experiments were made. We researched outer valve spring of VAZ engine. While lighting the external surface of the coil due to its cylindricity, light glare from lamp was focused providing clear reception of valve spring vibration process. Filming was done with the help of VS-FAST/G6 camera, and its signal was transmitted to a computer. Filming was done at 2000 fps

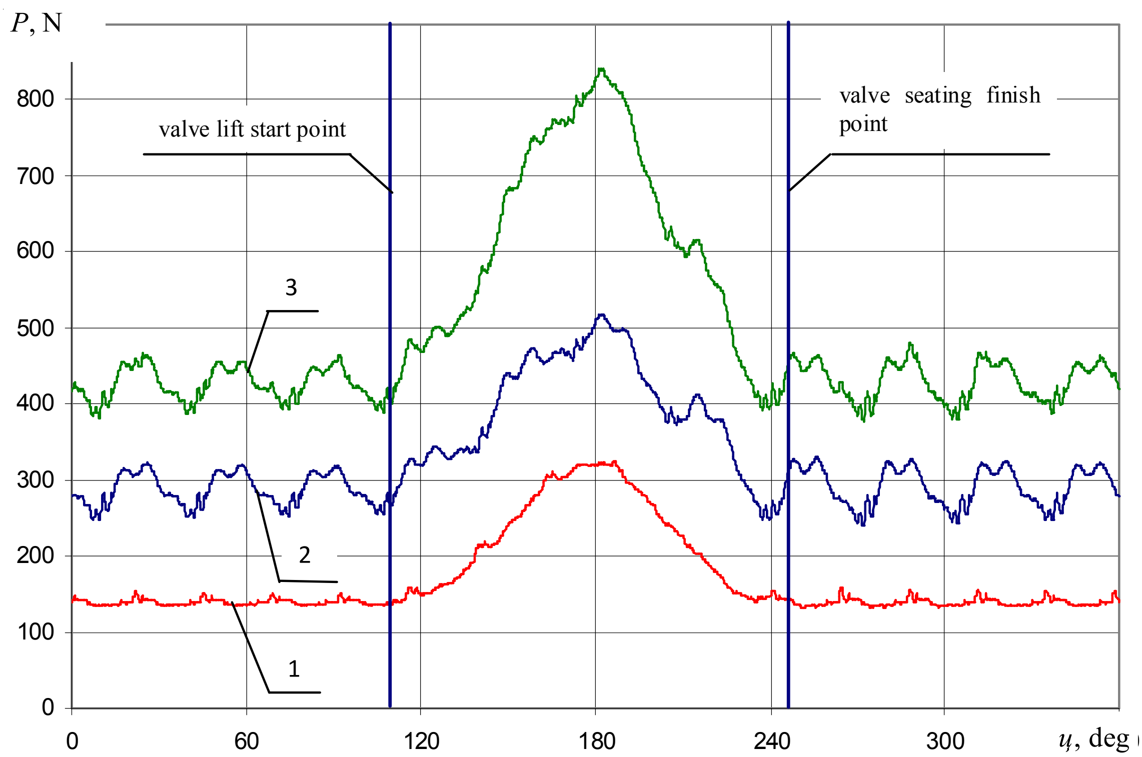


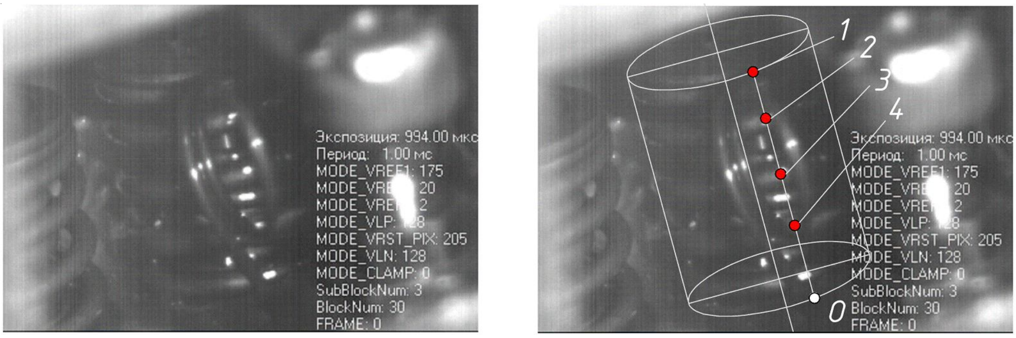
Fig. 2. Valve spring forces affecting on valve on camshaft rotating speed n = 2068 rpm:
1 – inner spring force; *2* – outer spring force; *3* – resultant valve spring force

(frames per second), that enabled to ensure the required accuracy of the results on the one hand and to take the most out of the laboratory equipment available on the other.

Valve spring coil oscillation charts were obtained through a storyboard of results filmed within Virtual Dub video editor, and processing each frame individually with measuring displacements of valve spring coil sections within AutoCAD subsequently. In each case, displacements of spring moving end and three active coils were determined; these measurements were specified by a fixed distance from the reference coil (point O) to the points on the surface of the coils (points 1, 2, 3 and 4) defining the coil section being researched. Their locations were determined by light reflection positions on the coil surface (see Fig. 3).

Fig. 4 illustrates VAZ engine outer valve spring examples on several engine operation modes. The numbering goes from valve spring fastened end. Engine operation modes are given as camshaft rotating speeds.

To assess experiment reproducibility the outer spring coil section average peak maximum of displacement amplitudes <*U*> statistical analysis with valve train expansion gaps 0 and 0.1 mm. was carried out. The research was done on 3rd, 4th and 5th outer valve spring coils for convenient observation of oscillation process (the 1st coil is fixed and the 6th coil movement is determined by valve displacement). Processing was performed at 11 engine operation modes in



a *b*

Fig. 3. Processing circuit and researched valve spring coil displacement measurements:

a – obtained through processing film frame example; *b* – processing circuit; valve spring coil displacements measurement; *1*, *2*, *3* и *4* – valve spring coil sections under research positions (*1* – moving end; *2*, *3* and *4* – valve spring active coils), *O* – reference coil section position

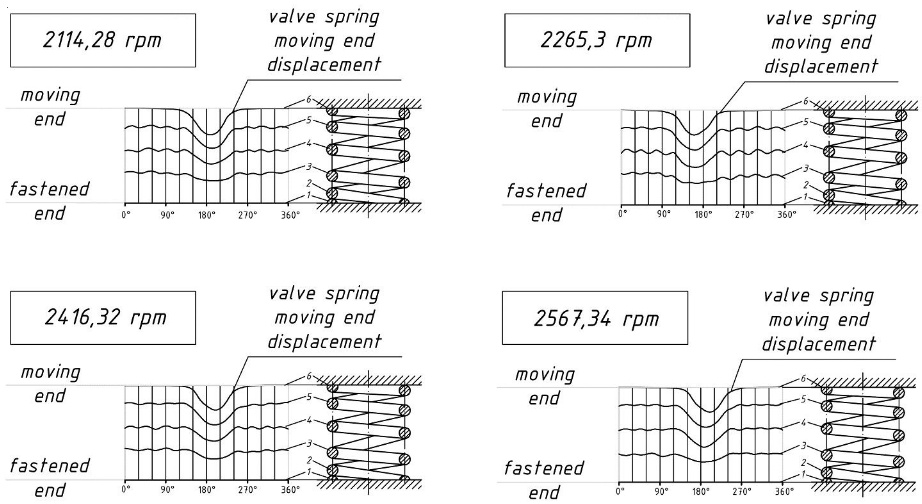
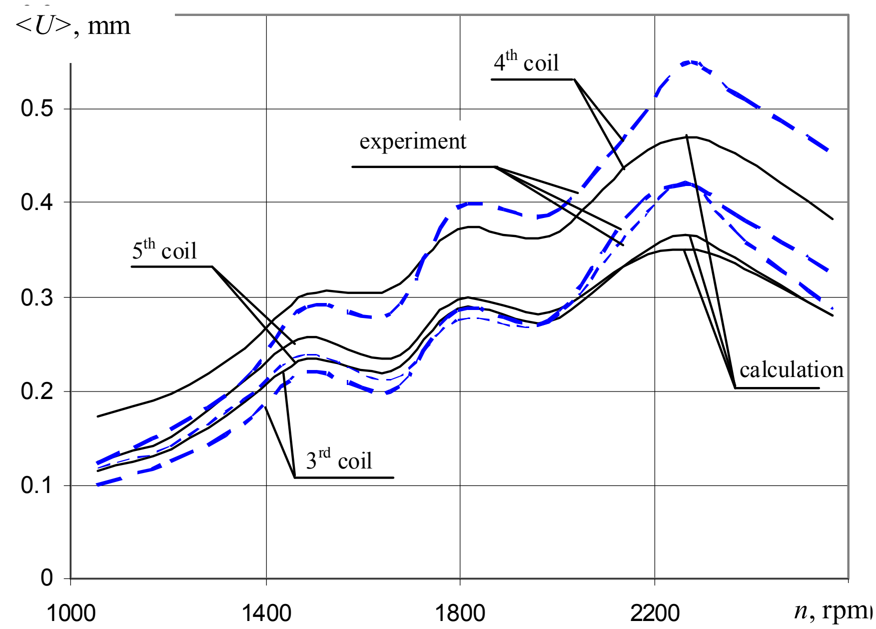


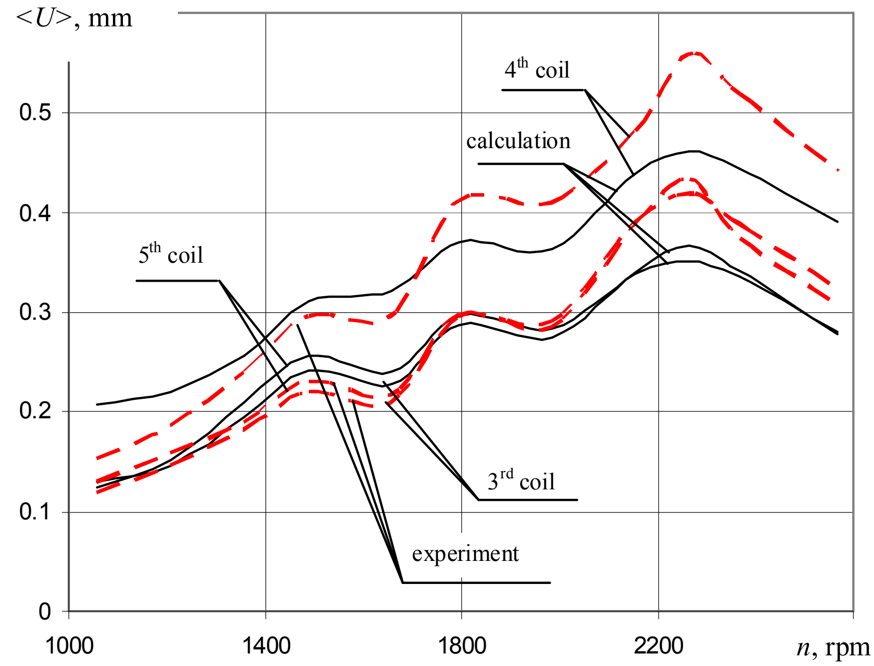
Fig. 4. Valve spring coil oscillation charts of the outer spring under research:

1 – reference coil (fastened end) *2, 3, 4, 5* – active coils; 6 – moving end camshaft rotating speed range of 1057…2567 rpm. Cochran's Q-test was an adequacy criterion.

Developed valve train dynamics simulation model and valve spring oscillation research calculation module ***SPR*** adequacy estimation was based on adequacy and reproducibility dispersion ratio and was made by comparing of average peak outer spring coil displacement values <*U*>obtained by experiment and calculation with 0 mm and 0,1 mm expansion gap provided. Fisher's F-test was an adequacy criterion. The proposed valve train dynamics research technique demonstrated good convergence with experimental data (Fig. 5).



a



b

Fig. 5. Average peak outer spring coil displacement values (calculation):
a – 0 mm expansion gap; *b* – 0,1 mm expansion gap

4. Discussion

Since valve spring forces affecting on valve were defined, valve train loading was estimated and the valve springs representation method effect on results obtained during valve train dynamics research was analyzed on the basis of developed simulation model.

Fig. 6 diagram illustrates rocker arm force affecting on valve under different valve spring representation methods: each valve spring is defined by 6 discrete masses (Fig. 6, curve 1) that corresponds to 12 mass dynamic simulation model (Fig. 1, a); valve springs are represented as equivalent elastic rods (Fig. 6, curve 2). The calculated data were compared to that obtained experimentally by valve train strain gauging [4] (Fig. 6, curve 3).

It should be noted that it has been found previously by researchers that the way you divide valve spring into discrete masses and their amount effects significantly on validity of results obtained by valve train dynamics simulation. For adequate representation of valve train loading valve spring coil single mass representation is sufficient, and increasing the number of masses complicates only the simulation model, but does not lead to improvement of results obtained [3]. It allows to take into account all valve spring harmonics and influence on valve train dynamics. In [4] it is shown that discrete 12-mass simulation model, where each valve spring is presented by 6 discrete-mass chain, provides the most exact and close to experimental data results.

According to diagram (Fig. 6) the proposed combined technique provides better experimental data approximation by reducing the difference between design data and that obtained experimentally. It is especially evident in the second half of the chart, when valve closes and finally sits on valve seat. This is due to the fact that equivalent rod approach allows to determine valve springs loading itself and the forces affecting on valve much more precisely, and therefore makes it possible to assess valve train technical condition more adequately.

Developed simulation model demonstrated good agreement between calculated and experimental data in camshaft rotating speed range of n = 157…2100 rpm (Fig. 7, curves 2, 3).

Fig. 7 illustrates correlation between valve train average peak force and camshaft rotating speed. The maximum discrepancy between valve train average peek force values <P> obtained by proposed technique (Fig. 7, curve 2) with that obtained experimentally (Fig. 7, curve 3) was about 7 % (n = 1368 rpm), while using discrete-mass technique (where valve spring were presented in a form of discrete-mass chain), it has reached 12 % (n = 2068 rpm).

Average valve train peak force specification value under proposed technique is 2 %. It was noted that the highest specification values take place at higher camshaft speeds (4…6 % for n = 1538...2100 rpm).

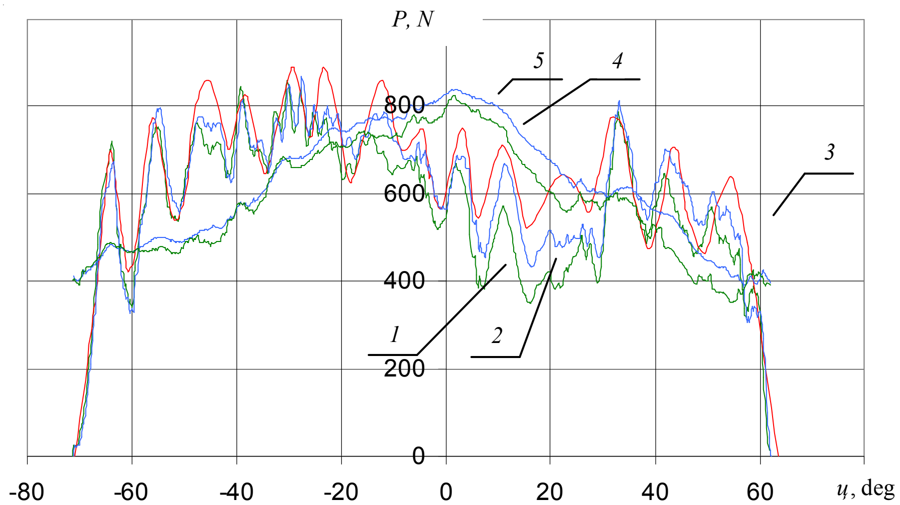


Fig. 6. Valve force; camshaft rotating speed n = 2068 rpm:

1 – discrete-mass model; *2* – proposed combined technique; *3* – experiment; *4* – resultant valve spring force (discrete-mass model); *5* – resultant valve spring force (equivalent rods approach model)

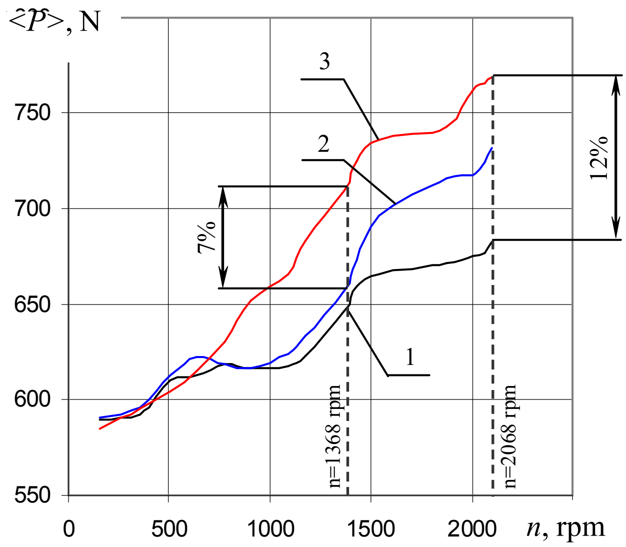


Fig. 7. Valve train average peak force:

1 – discrete-mass model (valve spring 6-mass chain representation);
2 – proposed combined technique (valve spring equivalent rod representation); *3* – experiment

5. Conclusion

Fig. 7 shows that approximation to experimental data improves with increasing camshaft rotating speed. Hence, the proposed valve train dynamics simulation technique can be extrapolated to engine high-speed operating modes and recommended for high-speed modern automobile engines dynamics research.

Thus, the developed simulation model providing a variety of ways to represent valve springs, is a type of complex technique that combines advantages of both multi-mass (simplicity; valve train dynamics higher vibration modes impact assessment with increase of discrete mass number) and equivalent rod (more accurate assessment of valve springs stress loading) approaches. It allows to make changes to design scheme structure and change its parameters efficiently, with simplicity to be kept and specify loads in the valve train.

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ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ И ИМИТАЦИОННОЕ МОДЕЛИРОВАНИЕ ДИНАМИКИ КЛАПАННОГО МЕХАНИЗМА АВТОМОБИЛЬНОГО ДВИГАТЕЛЯ

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Аннотация. В данной работе уточнена методика расчета двигателя внутреннего сгорания клапанного механизма на примере двигателя ВАЗ, полученная с помощью разработанной имитационной модели исследования динамики клапанного механизма. Также приведена методика экспериментального исследования колебаний витков клапанных пружин, реализованная посредством высокоскоростной киносъемки.

Ключевые слова: колебания витков клапанной пружины, нагруженность, математическая модель, механизм газораспределения, двигатель внутреннего сгорания, экспериментальное исследование, высокоскоростная киносъемка.