The spring of internal combustion engine valve mechanism

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ABSTRACT: This article deals with computational research of piston engine valve train including helical spring with variable number of active coils. This engine is destined for passenger car and therefore is featured by wide range of operational speed.

Keywords: spring; valve spring; valve linkage; IC engines

Performing dynamical response analysis of low-speed internal combustion engine valve trains by means of discrete models the masses of moving return spring active coils are either neglected, or their relatively small reduced part is added, together with spring cap mass, to valve mass itself. As a matter of fact, in that case influence of valve spring as design component comes through its stiffness only.

Coming up to fast-running engines inertia forces of moving machine parts gain ground strongly. Return springs designed suitably in due proportion to these forces have insofar massive coils, that their generated vibration may affect significantly the operation of whole valve mechanism.

Now assume three-mass (three degree-of-freedom) simplified model of OHV or OHC type valve mechanism, which was subjected to modification described recently in HONCŮ (2004), so that the model obtained straight form. Similar discrete model of valve spring can be appended to the end of the mass point chain (Fig. 1). Substitute masses marked as m_1, m_2 and m_3 represent reduced parts in the place of camshaft, tappet and valve,

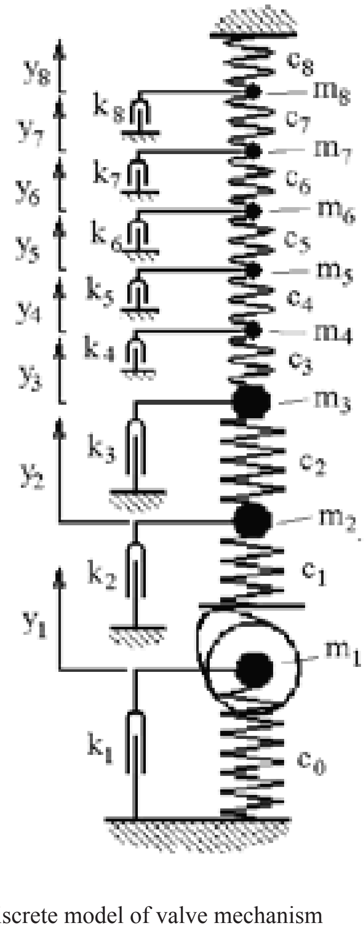

Fig. 1. Discrete model of valve mechanism

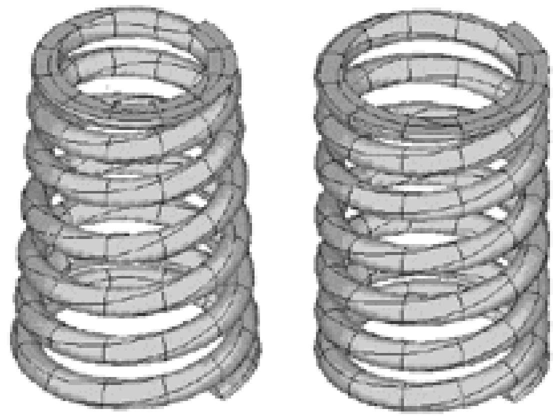

Fig. 2. Volume models of valve springs

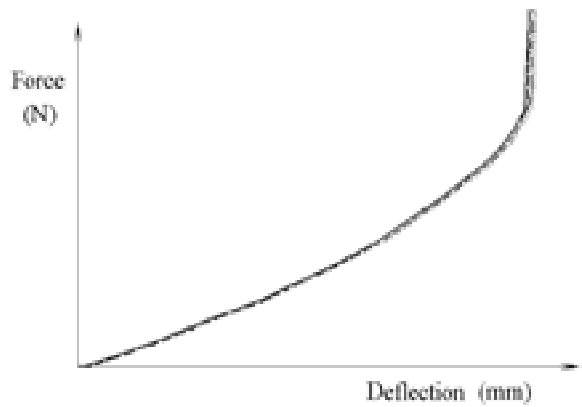

Fig. 3. Measured characteristics of four valve springs

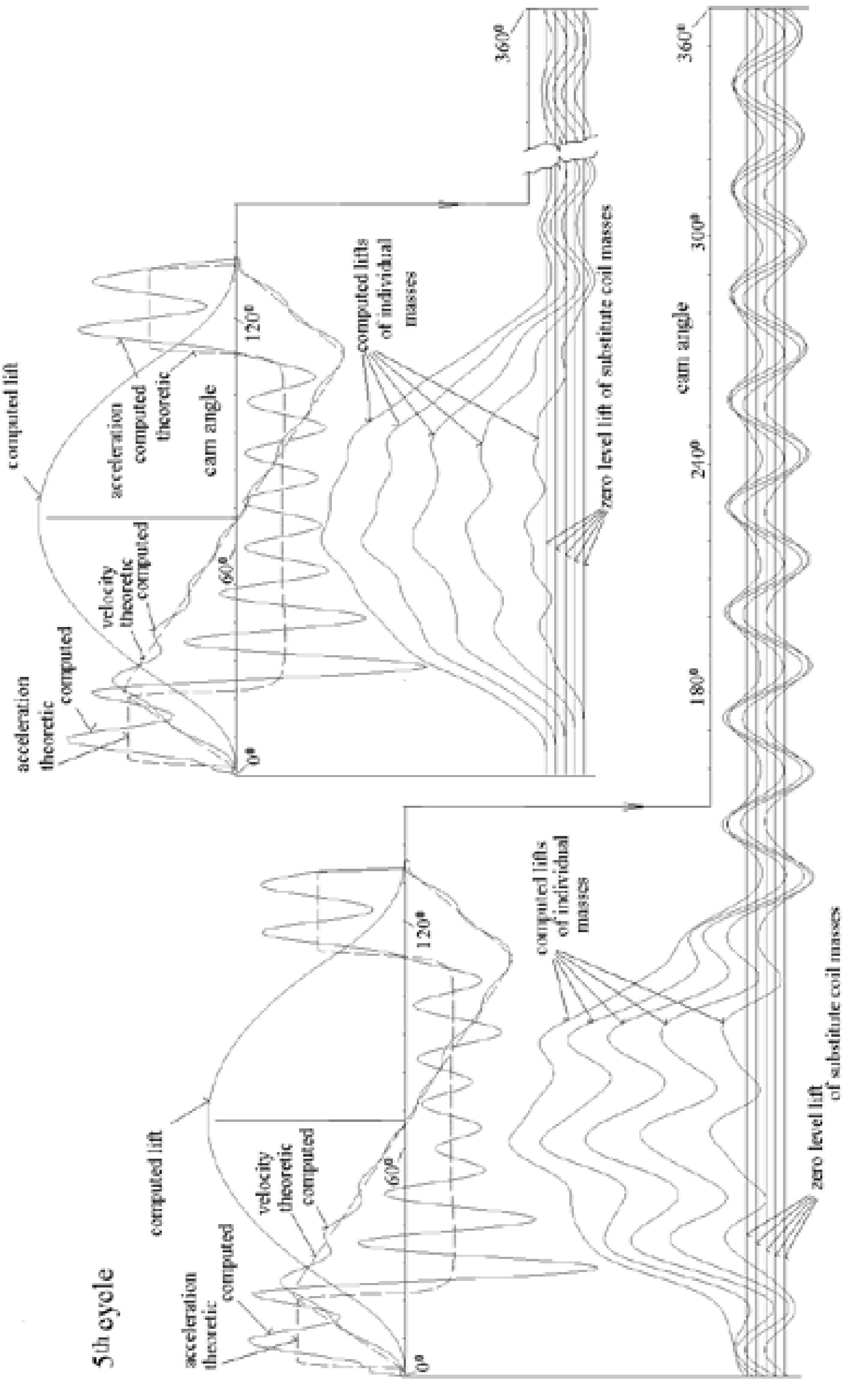

Fig. 4. Computed curves of substitute masses motion quantities

respectively, reduced intermediate stiffnesses are c_0, c_1 and c_2, starting with frame. In differential equations of motion are all kinds of passive resistance during oscillation substituted only by coefficients k_1, k_2 and k_3, that are in due proportion to velocity of these three masses. Such approach usually resulted in adequate accordance with measurement.

Two layouts of helical springs, conical and cylindrical, coiled using the wire of the same cross section and grade, were considered. Their volume models are shown in Fig. 2. Both have the same semi-finished product, the same number of active ($5\ ^2/_3$) and locking (≈2) coils. Cylindrical spring is purposely derived from conical one in order to achieve desired parity as for the ratio force/compression is concerned. Both springs have unequal spacing of active coils, so that active coils gradually abut (starting from the bottom) in the course of compression. Fig. 3 shows measured characteristics of four conical springs chosen accidentally. Spread of traces apparently did not exceed the rate usual for characteristics of cylindrical springs with fixed spacing of active coils.

In the course of research active part of both springs was divided into eight segments belonging to circumferential angle of identical size. Seven substitute masses placed in the segment boundaries of cylindrical spring have equal weights and equal are also stiffnesses of all segments. Analogous quantities for conical spring are

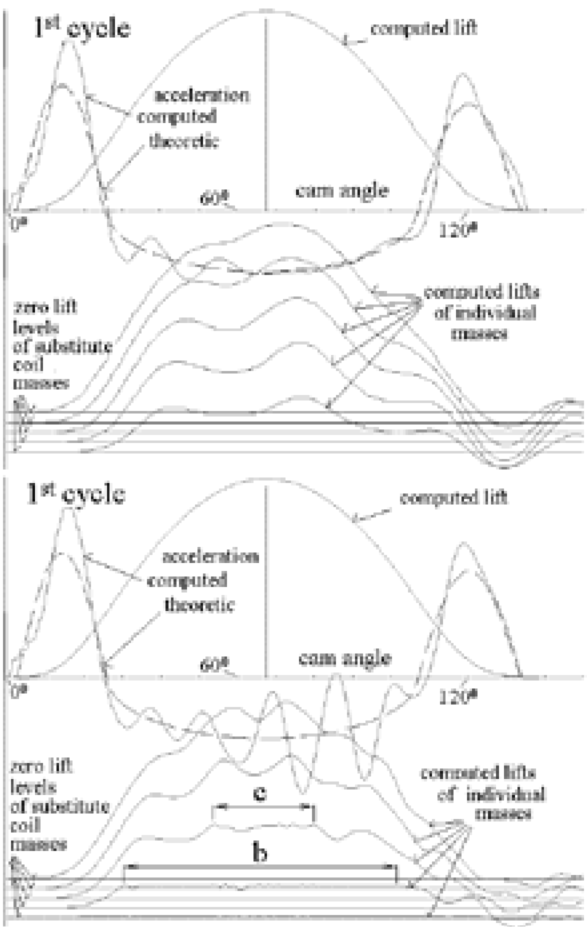

Fig. 5. Results of two comparative calculations

naturally variable. Such dividing was deliberately chosen in order to achieve the built-in condition, when just two spring segments abut to lower locking coils, thereby bringing down the number of vibrant masses about two.

Linking remaining five substitute masses of spring denoted from m_4 to m_8, stiffnesses from c_3 to c_8 and damping coefficients from k_4 to k_8 to original simplified three-mass system full model respecting the impact of spring is completed. It may be expected, that environment surrounding vibrating coils of spring in working engine generate also the resistance proportional to their velocity. Verification measurement proved that this damping is negligible, as well as is the damping caused by hysteresis of material.

In actual working valve mechanism the valve accomplish its return lift in good or worse compliance with cam profile after roughly one-third of camshaft revolution, sometimes after short jump as a result of rebound away from valve seat. Likewise other parts of valve linkage are halted owing to various passive resistance factors for remaining two-third of revolution. Slightly damped active spring coils, however, get on in their oscillating motion until the next exciting impulse caused by rotating cam (next cycle) take place. Allocation of additional masses representing return spring to valve linkage model thus makes the research considerably more complicated.

In the course of work using models without implementation of valve spring mass effect the validity of computed results generally ends in an instant when the valve closes to zero lift, because impact conditions of mass m_3 to seat are usually not modelled. As soon as the mass of coils is to be implemented, it is impossible to interrupt the computational process to determine valve linkage behaviour, because results of this process in following cycles may be affected by spring coil vibration. Therefore it will be necessary to observe continually the entire set of result cycles until steady state come into being. Models of various valve linkage mechanisms were examined and these computer tests demonstrated, that calculated results are already stabilised after the fourth cycle.

Impact of mass m_3 to valve seat can be admissible simulated by conditional command embedded to equation system computer program solver (Runge-Kutta-Gill Method), namely by means of an artificial holdback for a short time. In the course of calculation using this advanced model time dependent force acts on the spring cap, while, when the valve is closed, constant one (spring preload) acts using the older model. The value of variable force deviation from the force in built-in condition is, of course, dependent on engine operating mode.

Differences between computed curves of motion quantities belong to substitute mass on the place of valve and substitute masses of five spring active coil segments, obtained in the first and fifth computing cycles, can be traced in Fig. 4. Driving cam possesses continuous acceleration, formed by roughly rectangular positive and negative parts. Cam rotation speed was adjusted close to upper limit of engine operation range. Dashed lines denote the curves of theoretical velocity and acceleration, continuous lines show the curves of computed lift, velocity and acceleration. From the first to the fifth computing cycle in this operating mode vibration of spring coils grows, while its influence on valve acceleration is featureless.

Valve lift causes sequential abut of active spring coils. If we introduce commands of constrained (and temporary) stop for motion of spring substitute masses into model processing algorithm in the case that they reach the mutual distance limit (given by coil pitch of middle helical curve), we can also examine the coils abut by means of such modified program.

Compare accordingly the results of the first computing cycle from original program calculation (see top of Fig. 5) and the results valid for modified program option (bottom of Fig. 5). In this case the valve lift was produced by cam having another acceleration curve and cam rotation speed was slightly altered too.

This time successive abut of spring active coils impress the valve acceleration heavily. Recorded motion of the mass m_6 is halted for a time space proportional to line length c, motion of mass m_7 for a time space proportional to line length b and motion of mass m_8 is quieted shortly after the valve lift begins.

Rapid reinforcement of vibrations close to maximum valve lift activated by abut effect may have adverse consequences for all parts of mechanism. Legitimate process of coil abut, however, is not a sudden feature, but continuous one. Implementation of another model, quite different than is a discrete type, may develop the research and improve obtained results.

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Pružina ventilového mechanismu spalovacího motoru

ABSTRAKT: Článek se zabývá výpočtovým výzkumem ventilového rozvodového mechanismu spalovacího motoru se šroubovitě vinutou pružinou s proměnným počtem činných závitů. Motor je určen pro osobní automobil, a proto se vyznačuje širokým rozsahem pracovních otáček.

Klíčová slova: pružina; ventilová pružina; ventilový rozvod; pístové spalovací motory

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