THE SIMULATION RESEARCHES ON THE WEAR FOR ELEMENTS OF THE SEAT INSERT-VALVE-VALVE GUIDE ASSEMBLY

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Abstract

The simulation researches have been carried out, where the object has been the elements of the insert seat – valve – valve guide assembly. The analysed valve has been driven magnetoelectrically. The aim of the researches has been to investigate the wear of elements of the assembly after $2*10^6$ cycles. The analysed valve has been made of the TiAl6Zr4Sn2Mo2 alloy. The seat insert and the valve guide have been of cast iron. The special simulation model has been elaborated and presented in the article. It has been used the same geometrical and material parameters as for elements in the real assembly, in simulation model. During simulation, the needed valve lift profile has been introduced for valve drive. Than, basing on the control algorithm, the total force acting on the valve – drive assembly has been obtained and next parameters of motion for that assembly have been calculated. The special wear model for valve, valve guide and for seat insert has been elaborated and presented in the paper. It has been used the mentioned earlier parameters of motion. The other parameters of the model, such as the wear intensity has been obtained from short experimental series of $48*10^4$ cycles made in the research stand. Results of such series have been presented in the paper. The results of simulation have been presented in the paper, either. The wear of valve guides has increased almost linearly and the wear of valve and of its seat insert has increased nonlinearly with the increase of cycle's number.

Keywords: *combustion engine, valve timing, lightweight valve, seat insert, valve guide*

1. Introduction

The camless valvetrain can be realized with the help of magnetoelectric drives for valves [1]. The specific feature for such valve drive is usually the different valve lift profile, than in case of the valvetrain driven by camshaft. It can result from the needed valve lift profile, oriented for the maximal efficiency of combustion or from the control algorithm. In result the different values of valve velocity have been obtained and different values of the wear for valve, its valve guide and its seat insert, in comparison to the case of valves driven by camshaft. In combustion engines with magnetoelectric valve drives, the valves made of steel or TiAl alloys can be used. Such valve can mate with valve guide and seat inserts made of cast iron. In the paper has been presented research results for the case, when they both have mated in the conditions of oil absence.

The wear values for elements of the seat insert – valve – valve guide assembly can be obtained by long-standing experimental researches or estimated in simulation model, using wear parameters obtained from short-standing researches. Because of very high costs of long-standing researches, it has decided to perform mentioned simulation researches.

The aim of the researches has been to investigate the wear of elements of the assembly after $2*10^6$ cycles. The parameters of motion have been obtained from simulation model. The wear parameters for elements of the assembly have been obtained from short-standing research series, performed in the special stand, where valves have been driven by camshaft with cams of special shape. Using elaborated wear model, the wear values for elements of the assembly have been estimated, after $2*10^6$ cycles. The results of the simulation have been presented in the article.

2. Simulation model

The simulation model has been elaborated with the help of FEM. The scheme of such model has been presented in Fig. 1.

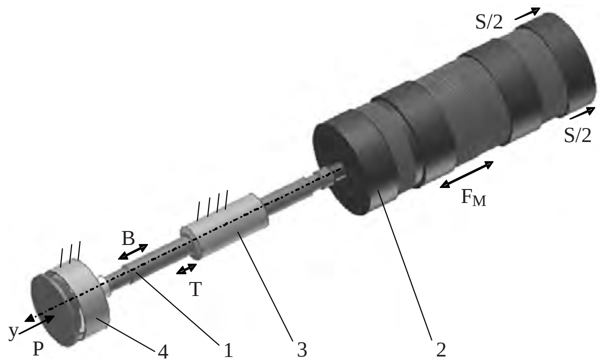

Fig. 1. The scheme of simulation model

The valve 1 has been driven by magnetoelectric drive 2. The valve stem has been bearing in its valve guide 3. The valve had has impacted into the seat insert 4. The moving coil of magnetoelectric drive has been actuated by applied voltage, which has allowed generating current in coil winding, positioned in the magnetic field from the drive magnet circuit. It has resulted in generation of the electrodynamical force F_M. The drive coil – valve assembly has been loaded by following forces [9]: electrodynamical force F_M, gas force *P*, inertia force *B*, spring force *S*, weight of assembly *G*, damping force *T*, described later in the article The generated electrodynamical force has accelerated the coil - valve assembly of mass *m*, with acceleration *a*. The motion of the coil – valve assembly has been controlled by special control algorithm, described later in the article.

3. The forces loading valve- drive assembly

During analysis the weight of assembly G has been neglected. It has been assumed for simplicity, that spring force S has increased linearly up to the maximal value 50 N with the increase of valve lift (displacement) y. In real such spring is of the nonlinear characteristic. The damping force T for coil form made of aluminium alloy has been equal 100 N, because of generated eddy currents during the motion of coil in the magnetic field.

The motion of drive coil – valve has been described by equation (1):

$$B(t)+T(t)+S(y)=F_M(t,y)+P(t), \tag{1}$$

The inertia force *B* has been calculated from equation (2):

$$B=m\cdot a\ , \tag{2}$$

where:
m = 113 g - mass of coil – valve assembly, when coil form is made from aluminium alloy.

The gas force *P(t)* has been calculated from equation (3):

$$P(t) = p_g(t) \cdot A = p_g(t) \cdot 0.25 \cdot \pi \cdot d^2, \tag{3}$$

where:
d = 30 mm - mean diameter of valve,
p_g = 0.5 MPa - gas pressure in the cylinder of combustion engine, loading exhaust valve during opening phase [1].

The example course of the modelled gas force vs. time has been shown in the Fig. 2.

The acceleration of the assembly can be calculated from equation (4) [1]:

$$a = \frac{F_M - P - T}{m}. \tag{4}$$

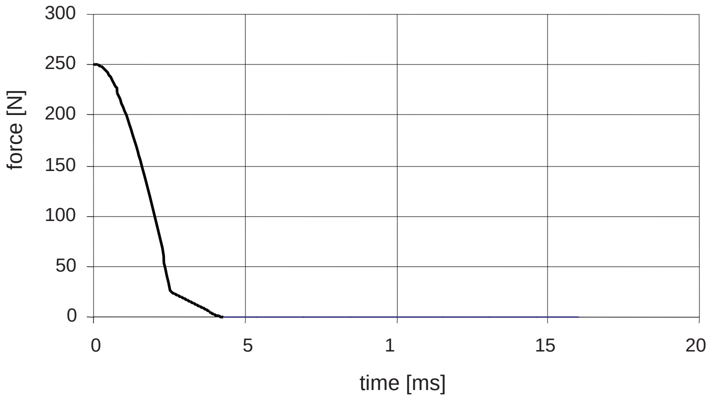

Fig. 2. The example course of the modelled gas force P(t) vs. time (t)

4. The control algorithm

The mathematical model has been elaborated to calculate valve lift vs. coil lift for different coil currents and dynamic parameters of the coil.

For the known phase angles of valve timing and the rpm of combustion engine the modelled course of valve lift y(CAD) has been first elaborated – for example its shape can be the same as for camshaft case or it can be of trapezoid shape. The shape could be constant with respect to the time and as result it depended on rpm - so it could be scaled when rpm changes to obtain the same phase angle vs. CAD. The shape of valve lift could be independent on the rpm of engine, either, as it is more often met. In such situation the phase angles vs. time should be chanced, to obtain the same phase angles vs. CAD.

Then the first $\dot{y}(t)$ and second $\ddot{y}(t)$ derivative have been calculated and the needed acceleration of the drive coil –valve assembly is obtained. Then the electrodynamical force is calculated from the equation (5):

$$F_M = P + T + ma + S, \tag{5}$$

Next, from the start point, the motion of coil has been calculated from the model (Fig. 1). Assuming that the rigid element connecting valve and assembly is assumed, the equation of motion has been following (6):

$$(m_1 + m_2)\frac{d^2 y_1}{dt^2} + c_1 \frac{dy_1}{dt} + k_1(y_1) y_1 = F(t, i, y_1), \tag{6}$$

where:

$m_1 + m_2 = m$ - mass of the valve – coil assembly,

y_1 - displacement of drive coil, y_2 – displacement of valve,

c_1 - damping coefficient; it can be assumed that $c_1\left(\frac{dy_1}{dt}\right) \cong T(sign(y_1))$,

$k_1(y_1)$ - stiffness of the spring connecting the coil and the basis.

From the simulation model, the dynamical parameters of the coil - valve assembly have been obtained.

Because the electrodynamical force has been the function of the current and of the coil position, the current has been estimated in each point of valve position. Then the pulse width has been assumed and value of current in that period have been constant – so we could obtain the needed current pulse train. The current in each pulse has been of the same absolute value, but could differ with respect to the direction. Next from the start point the motion of coil has been calculated from the model.

Next the calculated position of valve has been compared with the valve position from modelled course of valve lift vs. time. If the difference between actual and needed valve position has been positive, then current in the coil winding have started flowing in the opposite direction, if negative, the current started flowing in the same direction. If the maximal value of difference started overflowing the 5% of valve lift then the current has been decreased or increased of 1% and for the next cycle of valve motion the procedure has been repeated until the closest position of valve to the one from modelled course has been obtained in any moment.

5. The wear model

The wear W of valve guide has been calculated from the simple equation (7):

$$W = N \cdot w \cdot 2 \cdot H \,, \tag{7}$$

where:

N - number of cycles,

w_g - wear velocity of valve guide.

The wear velocity has been calculated from the equation (8):

$$w_g = \frac{W_{N\exp}}{N_{\exp} \cdot 2 \cdot H_{\exp}} \cdot \frac{2 \cdot H}{\sum_{i=1}^{i=n} v_i(t) \cdot \Delta t}, \tag{8}$$

where:

W_{Nexp} - wear of valve guide, obtained from experimental researches,

N_{exp} - number of cycles during single experimental series,

H_{exp} - valve lift during single experimental serie,

H - valve lift for the other analysed case then during experimental series,

v_i - valve velocity in *i*-th time period Δt.

The wear of the seat insert and valve seat have been resulted from impacts of valve into its seat insert and from sliding the seats initiated by the gas force P. In the model the changes of seat contact area A during following cycles have been taken into consideration.

The wear of the seat insert has been calculated from the equation (9):

$$W = \left(\frac{k\overline{P}Nx}{H_{si}} + KNe^n \frac{H_v}{H_{si} + H_v}\right)\left(\frac{A_i}{A}\right)^j, \tag{9}$$

where:
k - slide wear factor,
K, n - impact wear factors,
j - wear constant,
A - contact area after N cycles,
A_i - initial contact area,
P - mean gas force,
H_{si} - hardness for seat insert [HV],
H_v - hardness for seat insert [HV],
e - kinetic energy during valve impact into its seat insert.

The wear of the valve has been calculated from equation (10):

$$W = \left(\frac{k\overline{P}Nx}{H_v} + KNe^n \frac{H_{si}}{H_{si} + H_v} \right) \left(\frac{A_i}{A} \right)^j . \tag{10}$$

The values of constants n = 1, $j = 5$ have been obtained from the literature and of constants k, K from short-standing experimental series made in the research stand.

6. Research stand

The scheme of research stand for wear of valve, its guide and its seat insert has been shown in Fig. 3. The mentioned stand has been connected of the valve 4 mating with its valve guide 3 and its seat insert 1. The valve guide has been mounted in the sleeve 2 possessed in the pomp body. The valve has mated with its spring 5 through the cone wedges 6 and the cup 7. The valve has been driven by camshaft through the valve lifter with the regulation screw 8 and nut 9. The camshaft has been driven through the elastic coupling by the electric motor. Such research stand have been equipped with the microphone C1, the valve lift sensor C2, the valve acceleration sensor C3, the sensor for the rotational velocity of electric motor C4, the insert seat temperature sensor C5, the heater C6. The rotational velocity of the camshaft has been controlled by the control cassette C7.

The wear of valve and of seat insert has been measured basing on their geometry before and after series of 480000 cycles. The wear of valve guide has been obtained from the measurement of its mass before and after research series.

The mass of analysed valve has been equal 25.7 g. The camshaft has rotated with 1000 rpm.

The obtained values of wear after 48000 cycles, for the elements of the insert seat – valve – valve guide assembly have been presented in Tab. 1 and 2. From the short-standing series, it has been obtained following parameter values of the wear model for the analysed seat insert: K= 5.3E-14, k=0.0000275. From the short-standing series, it has been obtained following parameter values of the model for the analysed valve: K= 5.3E-14, k=0.0000105. The wear velocity of the analysed valve guide has been equal $2.67 \cdot 10^{-6}$ mm^3/cycles.

Tab. 1. Wear of analyzed valves and seat inserts

Valve material	Seat insert material	Valve wear [mm3]	Seat insert wear [mm3]
TiAl6Zr4Sn2Mo2	Cast iron	2.921	6.655

Tab. 2. Wear of analyzed valve guides

Valve guide material	Wear [g]	Wear [mm^3]
Cast iron	0.08	10.26

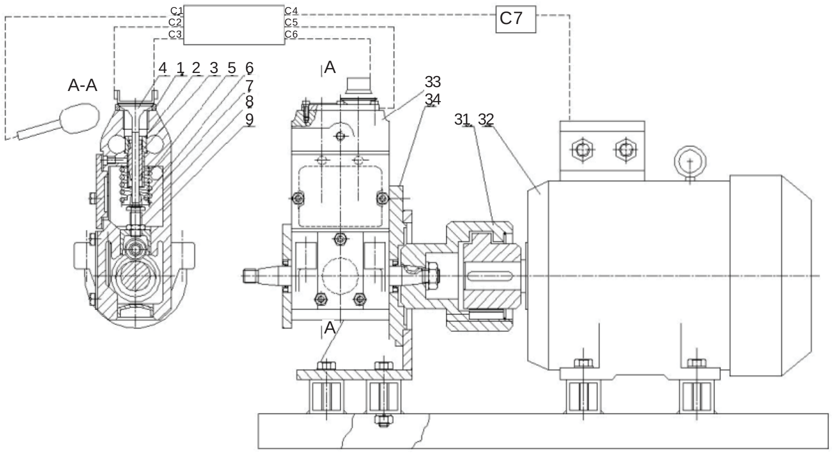

Fig. 3. The research stand for a measurement of the valve lift and acceleration, of the seat insert temperature, of the sound level generated during valve impacts into its seat insert, and of the wear for valves driven by the camshaft. 1 - seat insert, 2 - sleeve, 3 - valve guide, 4 - valve, 5 - valve spring, 6 - cone wedge, 7 - cup, 8 - regulation screw, 9 - nut, 31 - elastic coupling, 32 - electric motor, 33 - modified injection pomp, 34 - right side deck; C1 - microphone, C2 - valve lift sensor, C3 - valve acceleration sensor, C4 - sensor for the rotational velocity of electric motor, C5 - insert seat temperature sensor, C6 - heater, C7 - control cassette

7. The results of simulation researches

The results of simulation have been presented in the Fig. 4-11, for the valve driven magnetoelectrically, with the valve lift equal 5 mm and with the valvetrain related to that of camshaft rpm equal 1000.

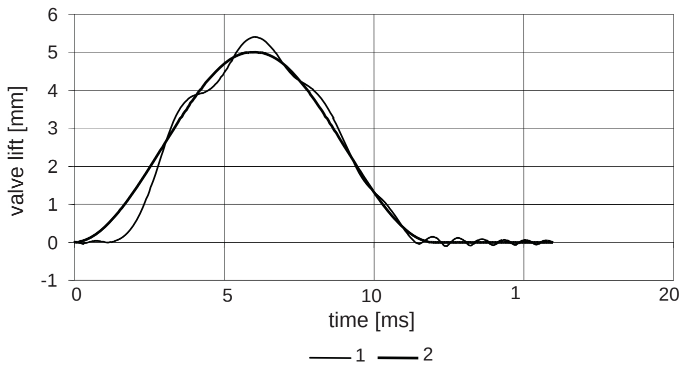

Fig. 4. Valve lift vs. time, 1 – values calculated from control algorithm, 2 – target values

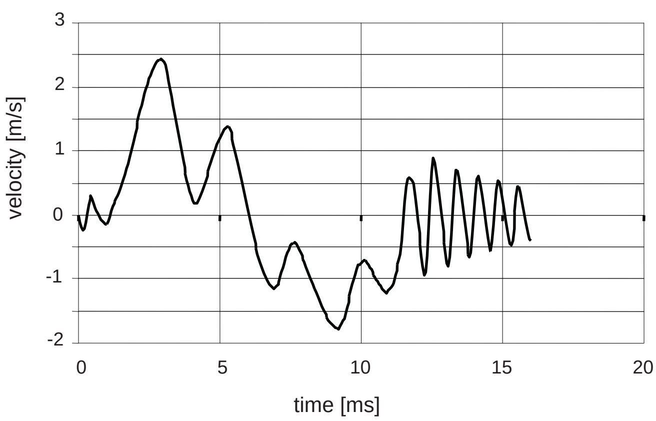

Fig. 5. Velocity of valve vs. time

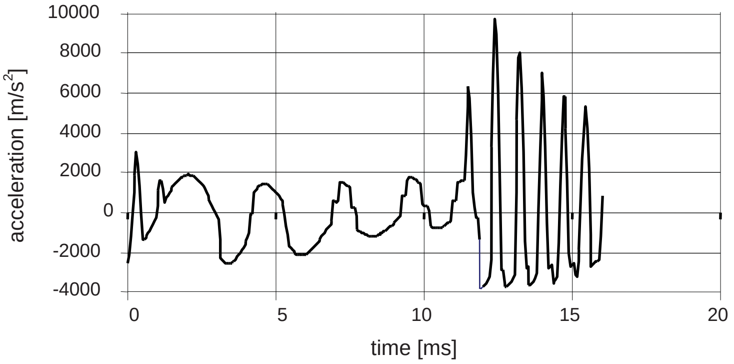

Fig. 6. Valve acceleration vs. time

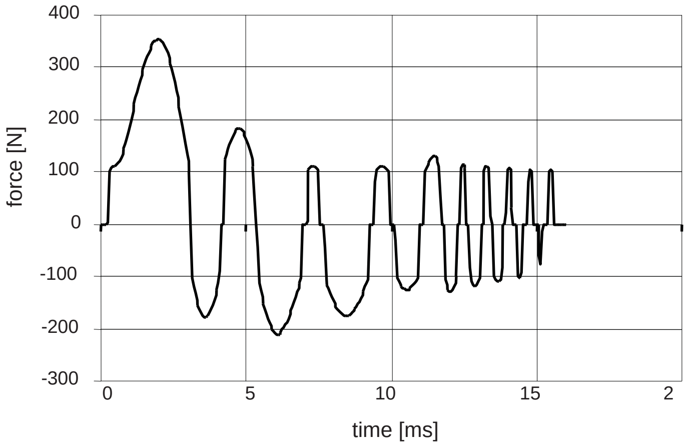

Fig. 7. Total force vs. time

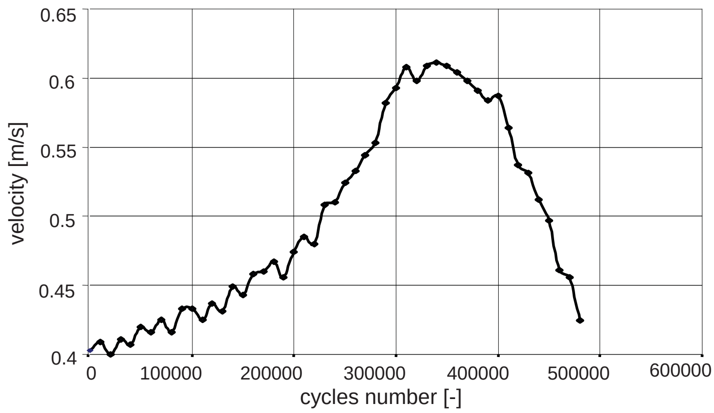

Fig. 8. Calculated valve seat velocity vs. cycles number

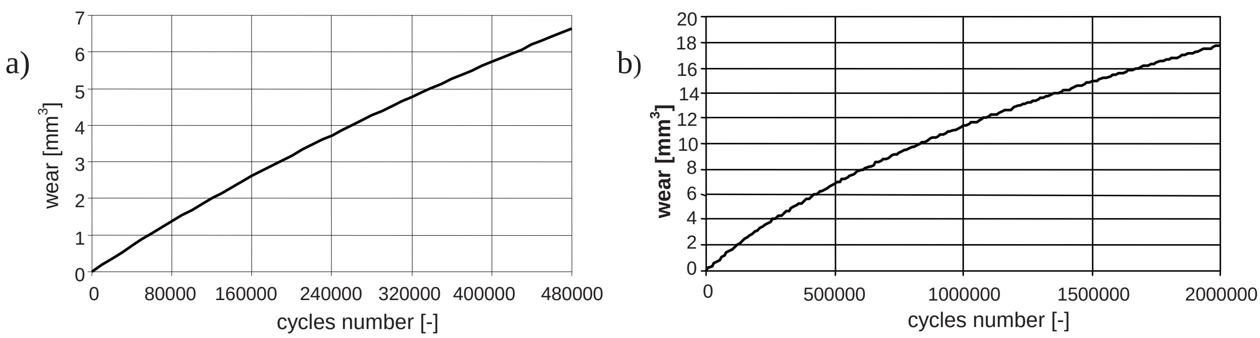

Fig. 9. Calculated wear of seat insert vs. cycles number, a – for N=Nexp=480000, b –predicted for N=2000000 cycles

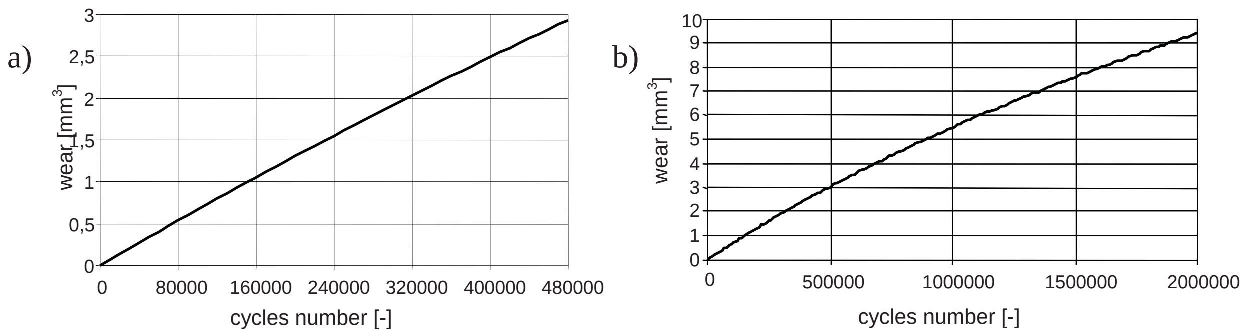

Fig. 10. Calculated wear of the valve vs. cycles number, a – for N=Nexp=480000 , b – predicted for N=2000000 cycles

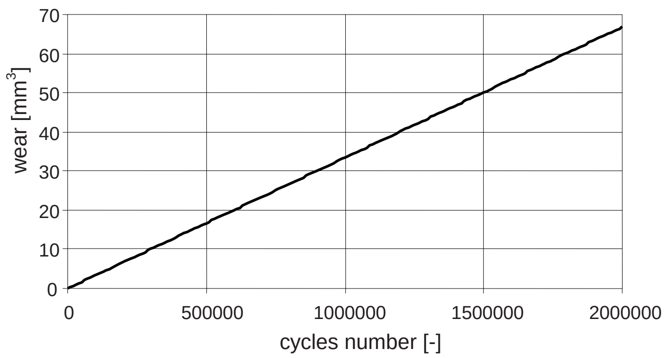

Fig. 11. Calculated wear of the valve guide vs. cycles number, predicted for N=2000000 cycles

8. Conclusion

1. Calculated values of valve lift obtained from the control algorithm have been different from those values of the target valve lift, up to 25%.
2. Calculated values of valve settle velocity have varied with the increase of cycle's number. After the initial velocity increase, up to 400000 cycles, it has decreased to about 0.2 m/s with the increase of cycle's number up to 200000.
3. Calculated values of valve velocity and of valve acceleration have had picked character because of the control algorithm influence. There have been no rigorous schemes for the position of such picks vs. time for different numbers of cycles
4. The wear of valve guides has increased almost linearly with the increase of cycle's number, although the picked values of valve velocity. The wear of valve and of its seat insert has increased nonlinearly with the increase of cycle's number, but the wear of seat insert has been twice greater than the valve wear during analyzed 2000000 cycles.

References

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Acknowledgement

The financial support of Ministry of Science and Higher Education by the grant No. NN502394535 is greatly appreciated