THE CONTROL OF VALVE TIMING WITH MAGNETOELECTRIC ACTUATORS

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Abstract

The new control algorithm for the valve timing in camless combustion engine is presented in the article. The magnetoelectric actuators have been used in the valve timing for analyzed popular combustion engine. Each titanium valve has been connected with the coil of actuator. Such coil can move in the magnetic field, generated in magnet circuit containing sintered Nd-Fe-B magnets, the core, pole shoes and air gaps. The movement of coil can occur when current flows in the coil winds and then electrodynamic force is generated. The nonlinear curves of generated electrodynamic force versus displacement of coil - valve assembly and vs. current in the coil have been computed using FEM method and presented in the article. The new mathematical model has been elaborated to calculate valve lift vs. coil lift for different coil currents and dynamic parameters of the coil. The values of current in coil should be controlled by the elaborated algorithm with the feedback. This algorithm has been based on tracing of position points of valve, during valve movement. The modelled course of valve lift vs. time has been first elaborated, for the needed rpm of engine. Next the first approximated current pulse train has been generated, the movement of the coil - valve assembly could be obtained. The calculated position of valve has been compared with the valve position from modelled course of valve lift vs. time. Basing on obtained difference of valve lift, the next current pulse train has been corrected in such a way, to obtain in any moment the closest position of valve to the one from modelled course. The obtained courses of valve lift vs. time for different modelled courses of valve lift vs. time and vs. rpm of engine have been presented in the article.

Keywords: *combustion engine, camless valve timing, valve timing control, magnetoelectric drive*

1. Introduction

The camless valve train is more and more popular in the modern combustion engines. Such valve train can be realised using electromechanic [1, 2, 4, 7], electrohydraulic [5, 6, 8], magnetoelectric [9] and electropneumatic [3] actuators. The popular combustion engine used in automobile has been analysed in the article. The magnetoelectric drives (actuators) have been used in the valve timing for such engine, to replace the standard camshaft. The scheme of such magnetic valve drive has been presented in the figure 1. Each valve has been connected with the coil of drive in such modified valve timing. One of the head problems connected with such valve timing has been its control. One of the important aims of the control has been to provide the small values of the valve velocity near the valve seat position. It has been decided to use titanium valve in the analysed engine, because using of the light valve can help to obtain small values of valve velocity in the set position. The small valve mass can facilitate to brake the moving valve before its impact into valve seat, with small value of force generated in the coil of thr drive. Such coil can move in the magnetic field, which is generated in magnet circuit containing sintered magnets, the core, pole shoes and air gaps. The Nd-Fe-B magnets have been used in the magnet circuit. The movement of coil can occur when current flows in the coil winds and then electrodynamic force is generated.

The curves of generated electrodynamic force versus displacement of coil - valve assembly and vs. current in the coil has been computed using FEM method and presented in the article. Such curves have been nonlinear. The elasticity of the coil - valve assembly has had an effect on the electrodynamic force - valve movement dependency either. So, the control of valve movement in such valve timing has been rather difficult.

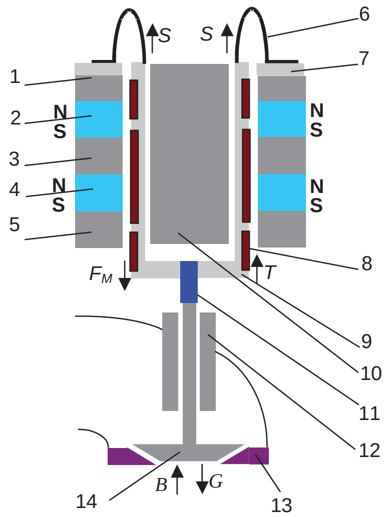

Fig. 1. The scheme of the magnetoelectric valve drive
1, 3, 5 - the pole shoe, 2, 4 – sintered magnet, 6 – base, 7 - spring, 8 - coil winds, 9 – coil form, 10 – core, 11 – element connecting valve with the coil 12 – valve guide, 13 - valve set, 14 – valve

2. The Forces Loading the Coil – Valve Assembly

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

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

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. During analysis the weight of assembly G has been neglected. The damping force T in case, when coil form has been made of Rezotex [10] equals 50 N and in case, when coil form has been made of aluminium alloy equal 100 N, because of generated eddy currents during the motion of coil in the magnetic field.

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,
m = 103 g – mass of coil – valve assembly, when coil form is made from Rezotex [10].

The time t of the assembly motion can be estimated from equation (3):

$$t = 4 \cdot \sqrt{\frac{h}{a}}, \tag{3}$$

where: h = 7 mm – maximal valve lift [3].

The gas force $P(t)$ has been calculated from equation (4):

$$P(t) = p_g(t) \cdot A = p_g(t) \cdot 0.25 \cdot \pi \cdot d^2, \tag{4}$$

where:
d = 27 mm – mean diameter of valve; such value is characteristic for combustion engine in automobile,
p_g = 0.5 MPa – gas pressure in the cylinder of combustion engine, loading exhaust valve during opening phase.
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 (5) [9]:

$$a = \frac{F_M - P - T}{m}. \tag{5}$$

The spring force S is the nonlinear function of assembly motion but is rather small and can be neglected. If necessary, it can be calculated with the FEM.

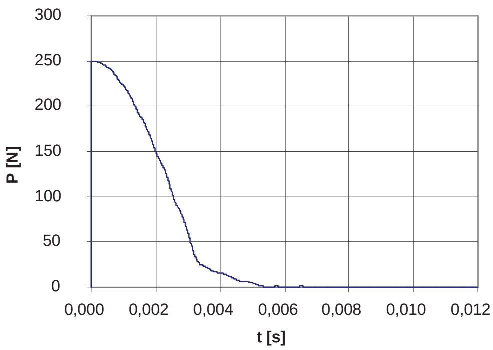

Fig. 2. The example course of the modelled gas force P(t) vs. time (t)

3. The Model of the Drive

The electrodynamic force F_M is nonlinear and should be calculated numerically. The values of the force depend on magnetic field changes during the coil motion. Such changes can occur because of the magnet field generated during current flow in moving coil winding and because of eddy currents, especially when coil form is made from aluminium alloy. Such phenomena can change generated electrodynamic force about 10% in comparison to the case, when the coil is fixed. To simplify the calculation, it has been assumed, that electrodynamical force is the function only of coil position. As it has been mentioned, the model of drive (actuator) has been elaborated using FEM. The scheme of magnetoelectrical valve drive has been shown in the figure 3 – for inlet valve and in the figure 4 – for exhaust valve. The electrodynamical force generated during coil motion in the magnet field from magnet circuit has acted in the direction of the coil axis. To simplify the calculation the axisymmetrical model of actuator has been assumed. Computation has been made with commercial programme. The plain eight node element PLANE53 (Fig. 3, 4) has been used. The degree of freedom in each node has been the value of vector magnetic potential A_Z – in the 0Z axis direction The values of electrodynamical force have been computed basing on obtained values of vector magnetic potential A_Z in the nodes of finite elements in the model. The grid of finite elements has been made automatically by the programme. The grid of elements and boundary conditions has been shown in the Fig. 3 and 4.

The obtained curves of generated electrodynamic force versus displacement of coil - valve assembly have been shown in the Fig. 5.

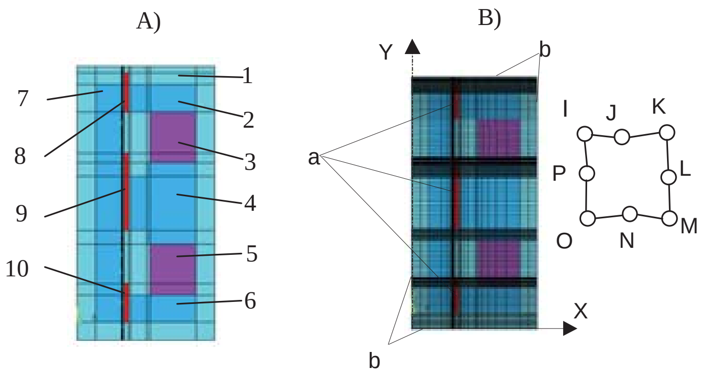

Fig. 3. A) The scheme of the modelled magnetoelectric drive; for inlet valve. B) The grid of finite elements and boundary conditions; for inlet valve. 1- air, 2, 4, 6 – pole shoe, 3, 5 – sintered magnet, 7 – core, 8, 9, 10 – coil; a – areas containing nodes of elements, where homogeneous electric density ρ has been introduced, b – lines containing nodes, where value of component of vector magnetic potential has been introduced A_z=0

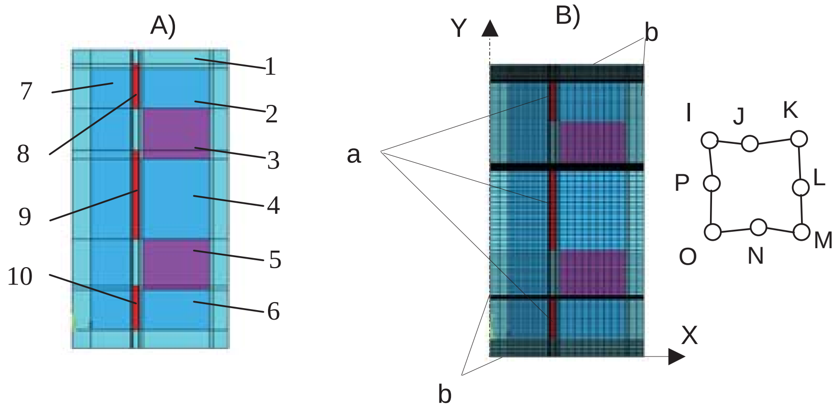

Fig. 4. A) The scheme of the modelled magnetoelectric drive, for exhaust valve. B) The grid of finite elements and boundary conditions; for exhaust valve. 1- air, 2, 4, 6 – pole shoe, 3, 5 – sintered magnet, 7 – core, 8, 9, 10 – coil winding; a – areas containing nodes of elements, where homogeneous electric density has been introduced, b – lines containing nodes, where value of component of vector magnetic potential has been introduced A_z = 0

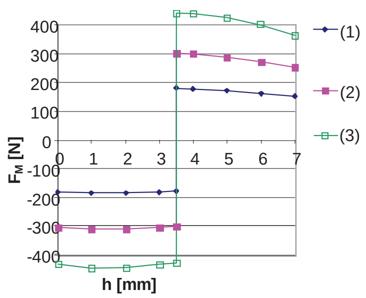

Fig. 5. Electrodynamic force F_M vs. valve lift h; (1) - for inlet valve drive, current in the coil i = 20 A; (2) - for exhaust valve drive - current in the coil i = 20 A; (3) - for exhaust valve drive - current in the coil i = 30 A.

It has been used the following boundary conditions:

The homogenous electric density has been introduced in the nodes of finite elements, which lay in the cross area of coil winds.

The values of the vector magnetic potential component $A_z=0$ have been introduced in the nodes, which lay on the outer lines restraining the air area, which one encloses actuator.
The material properties of coil elements are presented in the Tab. 1.

Tab. 1. Material properties of coil elements

Material	Density ρ [kg/m$_3$]	Young modulus E [MPa]	Poisson number [-]	R_m [MPa]	R_e [MPa]	$R_{bending}$ [MPa]
Cupper	8900	117000	0.35	227	57	
Aluminium alloy	2700	71000	0.27	420	290	
Rezotex [10]	1300	7000	0.3			100

4. The Control Algorithm

The new 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 changed, to obtain the same phase angles vs. CAD.

Then the first $\dot{y}(t)$ and the 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 (6):

$$F_M = P + T + ma + S. \tag{6}$$

Next, from the start point, the motion of coil has been calculated from the model (Fig. 6).

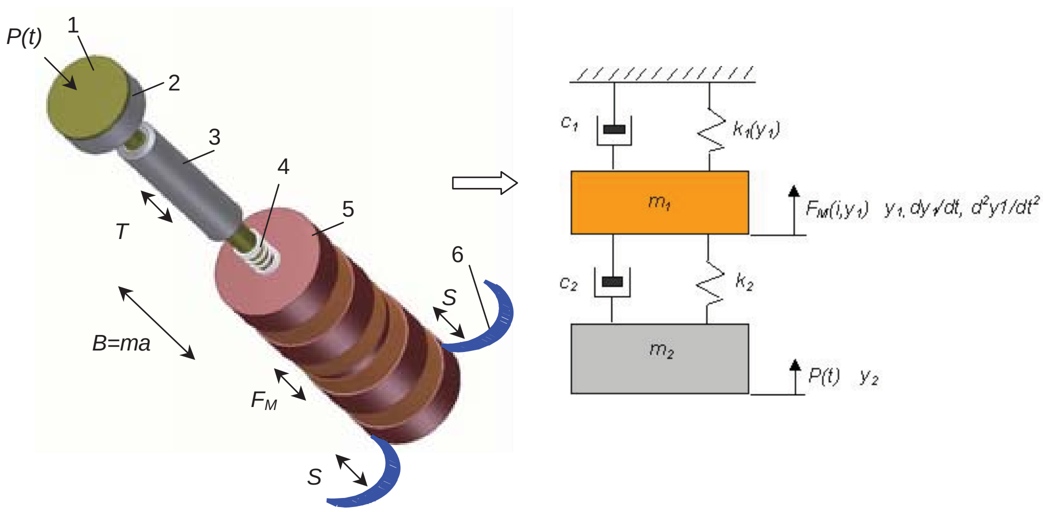

Fig. 6. The model of drive coil, valve assembly. 1 – valve (m_2), 2 – valve set, 3 – valve guide, 4 – element connecting valve end coil form, 5 – coil (m_1), 6 – spring

The equations of motion (7), (8) are following:

$$m_1 \frac{d^2 y_1}{dt^2} + c_1\left(\frac{dy_2}{dt} - \frac{dy_1}{dt}\right) + k_1(y_1)(y_2 - y_1) = F(t, i, y_1), \tag{7}$$

$$m_2 \frac{d^2 y_2}{dt^2} + c_2\left(\frac{dy_1}{dt} - \frac{dy_2}{dt}\right) + k_2(y_1 - y_2) = P(t), \tag{8}$$

where:

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

c_1, c_2 – dumping coefficient; it can be assumed that $c_1\left(\frac{dy_2}{dt} - \frac{dy_1}{dt}\right) \cong T(sign(y_1))$,

$k_1(y_1)$ – stiffness of the spring connecting the coil and the basis,

k_2 – stiffness of the element, connecting the valve and the coil.

I f rigid element connecting valve and assembly is assumed, then equations (7), (8) can be simplified to equation (9):

$$(m_1 + m_2)\frac{d^2 y_1}{dt^2} + c_1 \frac{dy_1}{dt} + k_1(y_1)y_1 = F(t, i, y_1), \tag{9}$$

where:

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

The model (Fig. 6) has been the simply FEM model, generated by CAD program. Geometry and material parameters are nearly the same as in the real. From the model, the dynamical parameters of coil and of valve have been obtained. The course of valve lift has become the new modelled course of valve lift. Such course can be unchanged when rigid element connects valve and coil.

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 such pulses could be of different values in each pulse, but in such case the control algorithm is very difficult to realize. In the simpler case 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.

For small and middle values of engine rpm the influence of elastic element between valve and coil is rather small and such algorithm with small number of current pulses in the current pulses train can be sufficient. For high value of engine rpm the number of current pulses should be greater about of order.

5. The Results of Calculations

The obtained courses of valve lift vs. time for different modelled courses of valve lift vs. time and vs. rpm of engine have been presented in the Fig. 7, 8 and 9.

The influence of maximal valve lift upon course of valve lift vs. time has been shown in Fig. 10. The influence of engine rpm upon course of valve lift vs. time has been presented in Fig. 11. The influence of sampling width upon course of valve lift vs. time has been presented in Fig. 12.

The influence of sampling width upon the maximal difference between valve lift and target curves, upon the difference between maximal valve lift and target curve and upon the mean settling velocity vs. rpm has been presented in the Fig. 13 – 15.

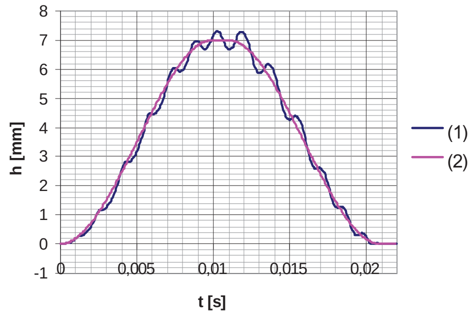

Fig. 7. The valve lift h vs. time (1) and target curve – modelled course of valve lift vs. time (2), for h = 7 mm, 2000 rpm

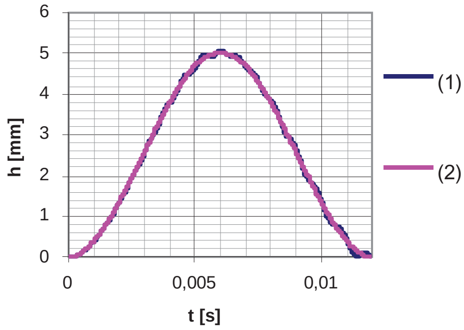

Fig. 8. The valve lift h vs. time t: target curve – for coil displacement (2), the new target curve - for valve lift (1), for h = 5 mm, 3500 rpm

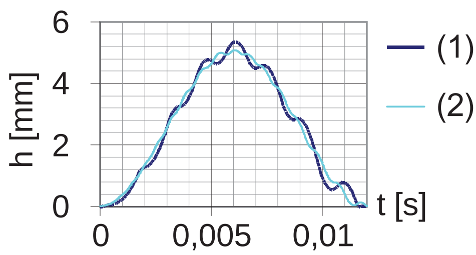

Fig. 9. The valve lift h vs. time t (1) and the new (modified) target curve - for valve lift (2), for h = 5 mm, 3500 rpm

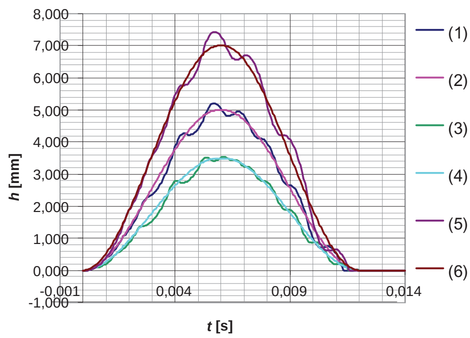

Fig. 10. The valve lift vs. time for maximal valve lift: h = 7 mm (5, 6), h = 5 mm (1, 2), h = 3.5 mm (3, 4). Values computed for modelled target curves (2, 4, 6) of valve lift and from elaborated algorithm (1, 3, 5); engine rpm = 3500

6. Conclusions

1. The values of electrodynamical force, generated in coil of actuator changes nonlinearly with the coil displacement. Such force can increase near linearly with the increasing of current flowing in the coil

2. The difference between maximal valve lift and target curve increases nonlinearly with the increasing of engine rpm and valve lift, for all cases of sampling.
3. The maximal difference between valve lift and target curve increases nonlinearly with the increasing of engine rpm and valve lift for all cases of sampling
4. With the increasing of engine rpm the calculated mean valve settling velocity increases nonlinearly for all cases of sampling.

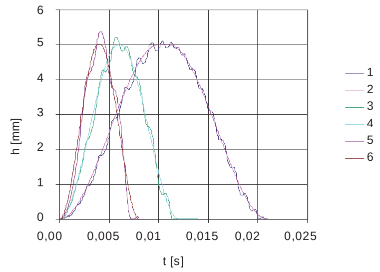

Fig. 11. The valve lift h vs. time t for engine rpm: n = 2000 (1, 2), n = 3500 (3, 4), n = 5000 (5, 6). Values computed for modeled target curves of valve lift and from elaborated algorithm (1, 3, 5); with maximal valve lift h = 5 mm

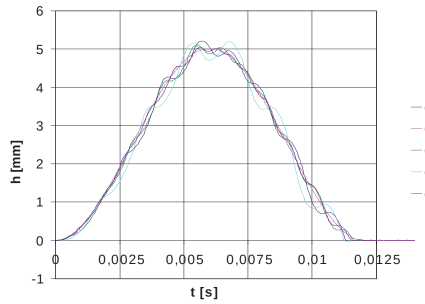

Fig. 12 . The lift of valve vs. time for four cases of sampling width: (1) Δ = 12.5 ms, (3) Δ = 25 ms, (4) Δ = 50 ms, (5) Δ = 100 ms. Values computed for (2) modelled target curve of valve lift and from elaborated algorithm; with maximal valve lift h = 5 mm and engine rpm = 3500

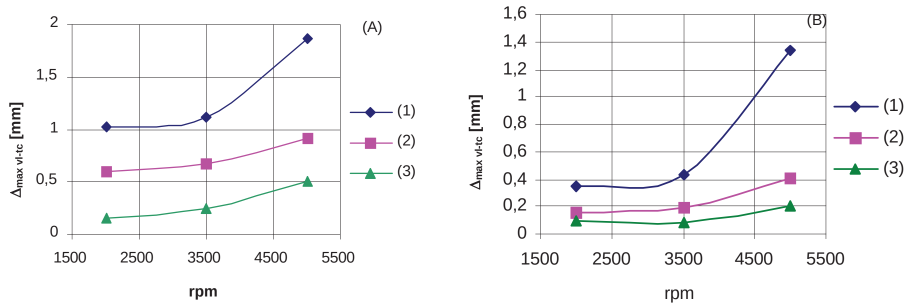

Fig. 13. Maximal difference between valve lift and target curve $\Delta_{max\ vl\text{-}tc}$ vs. engine rpm; sampling: (A) 100 μs, (B) 12.5 μs; Valve lift: (1) – 7 mm, (2) – 5 mm, (3) – 3.5 mm

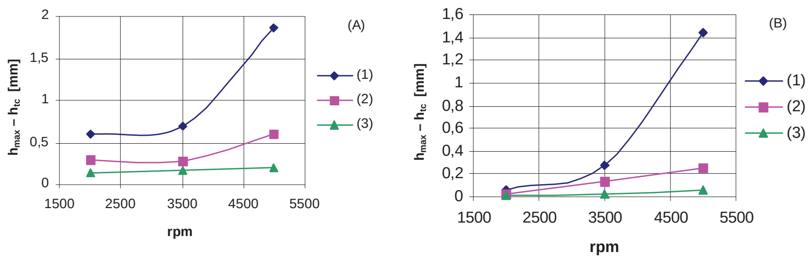

Fig. 14. Difference between maximal valve lift and target curve (h_{max} - h_{tc}) vs. engine rpm, sampling: (A) 100 μs, (B)

12.5 μs; Valve lift h: (1) – 7 mm, (2) – 5 mm, (3) – 3.5 mm

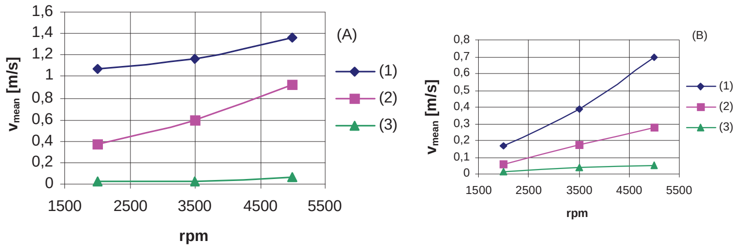

Fig. 15. Mean valve settling velocity v_{mean} vs. engine rpm, sampling: (A) 100 μs, (B) 12.5 μs; Valve lift h: (1) – 7 mm, (2) – 5 mm, (3) – 3.5 mm

5. The influence of stiffness of element connecting the drive coil and the valve is more visible in case of using of elaborated algorithm, than in case of ideal moving of coil according to target curve. The difference between valve lift and coil lift in case of algorithm are twice greater than in case of ideal moving of coil.

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