

Modeling and Investigation of Electromechanical Valve Train Actuator at simulated Pressure conditions.

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Abstract

In an electromechanical valve actuated engine, the valves are driven by solenoid-type actuators and cam-shaft is eliminated. Control of each valve provides flexibility in valve timings over all engine conditions and achieves the benefits of variable valve timing (VVT).

This paper is about investigation of Electro-mechanical actuator at simulated pressure conditions for a single cylinder engine. For this purpose, a scaled down actuator with reduced armature lift and high stiffness springs are being used.

Experiments are conducted to measure valve release timings, transition times and contact velocities. Furthermore, discussion about the spring, magnetic, exhausts gas forces and their ability to actuate the system as desired.

Keywords: *Electromechanical actuator (EMA), Variable valve timing, electromagnets.*

Introduction

The fixed valve motion by camshaft engines compromises the fuel economy, combustion stability and maximum torque performance at different loads. The conventional camshaft is replaced by electromechanical actuator in order to improve the performance of a combustion engine with a flexible scheme in valve timing at all engine operating conditions.

Electromechanical actuators are increasingly becoming the actuator of choice in industry, due to their ruggedness, low cost, reduced complexity, relative high force density and ease of control. VVT reduces or eliminates many of the tradeoffs between low and high speed torque, fuel economy, idle quality, and emissions that are currently made with fixed valve timing [1][2].

Several variable valve actuation schemes have recently been studied and reported in the literature. The examples range from more flexible cam-based systems, such as variable or dual cam timing, to totally camless engines for which the valves are independently operated by means of specifically designed valve actuators. Electro-hydraulic actuated systems and electro-magnetic actuated systems are two most common examples of camless actuation technology for achieving variable valve timing [3].

After multi-valve technology became standard in engine design, Variable Valve timing becomes the next step to improve engine output. With electromechanical valve train (EMVT) systems valve timings are fully independent from crankshaft position and with flexible valve timings, cylinder air charge and residual gas can be optimized. By controlling the intake valve events the throttled operation is eliminated in the gasoline engine and by doing so reduce the pumping loss which results high fuel efficiency [4][5].

Most electromagnetic systems in the literature use a spring system to accelerate and decelerate the valve. Solenoids or motors are used to hold the valves in the end positions and to compensate for friction losses, as well as combustion forces [6].

In EMVT actuator, speed and friction of moving parts results into high energy consumption. Scaled down actuator at higher speed need stronger springs, that require higher currents but the issue of magnetic saturation arises. Similarly at higher pressures especially at valve opening more catching current is required to open the valve against the air pressure.

Prototype electromechanically actuated VVT systems have been proposed by several companies in the automotive industry, the first being proposed by FEV Motorentechnik [7] [8]. Other companies that have worked on this technology include BMW, GM, Renault and Siemens.

This paper is about a scaled down electromechanical actuator designed for motorcycle applications. The actuator is investigated at a reduce armature lift and at higher speed for single cylinder engine with experimental results. Apart from design changes the effects of spring rate, armature lift and exhaust gas forces on valve are discussed.

Actuator Model

Main part of the system is an electromechanical actuator, which operates as a free oscillation system with electromagnets holding the valves in both final positions. The actuator consists of lower magnetic coil for opening the valve and an upper magnet for closing the valve. Actuator and valve spring push on armature and valve stem through spring retainers. At mid position the armature is centered between lower and upper magnets.

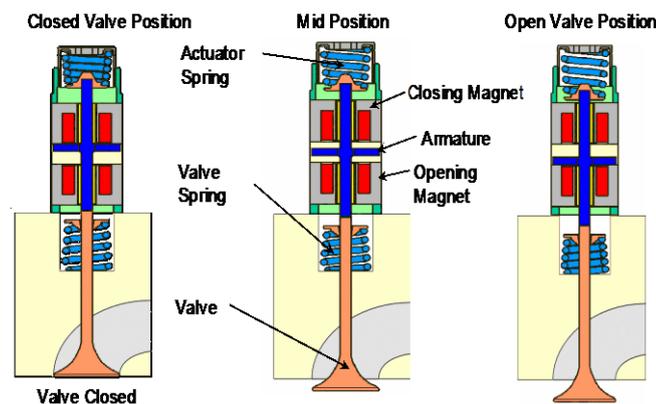


Fig1. Valve position by the action of magnetic and spring forces [9] .

At start, voltage is applied to one of the electromagnets to move the armature from Actuator middle position to the fully open position. A holding current is then maintained to holds the armature in place against the spring force. The mechanical spring force and magnetic force determine the actuator and valve operation.

At valve closing the armature moves to the upper magnet and a holding current is applied to hold the armature at closing magnet against the actuator spring force.

Actuator system comprising electrical, mechanical and magnetic system, Electrical energy is transferred to excite the electromagnets, due to which useful mechanical work is obtained as shown in Fig 2,

Subsystem of EMVT:

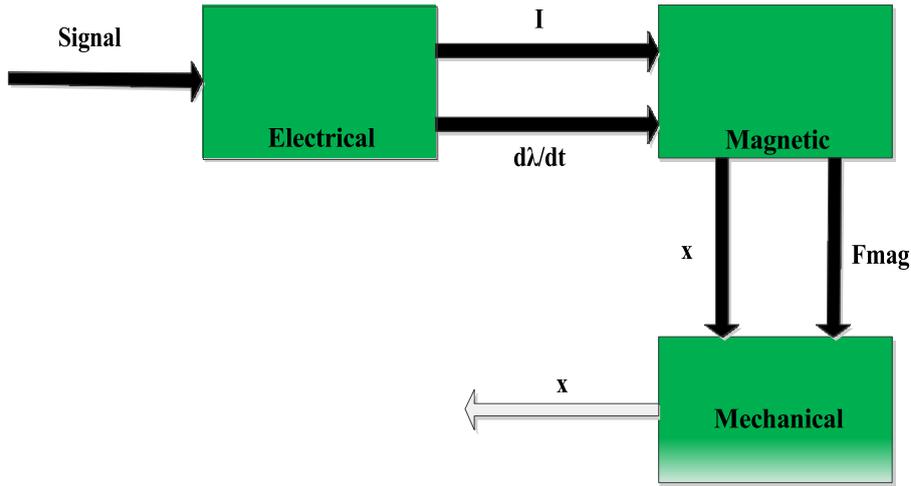


Fig 2. Model of actuator subsystem.

Where, I = current, x = armature position, $d\lambda/dt$ = change in flux and F_{mag} = magnetic force, Magnetic energy of the core is

$$W_{mag} = \iiint HdB.V = 4[a.b.d] \frac{B^2}{2\mu_0}. \quad (1)$$

The volume of the magnetic core is shown in Fig 3, while the electrical and magnetic power in the core is

$$P_{mag} = 4ab \frac{B^2}{2\mu_0} d' + 4abd \frac{2BB'}{2\mu_0}, \quad (2)$$

$$P_{el} = V.I = \frac{U - 2n\Phi'}{R} = \frac{U}{R} - \frac{2n}{R} B'a.b, \quad (3)$$

Mechanical power in the system is

$$P_{mech} = F.d' \quad (4)$$

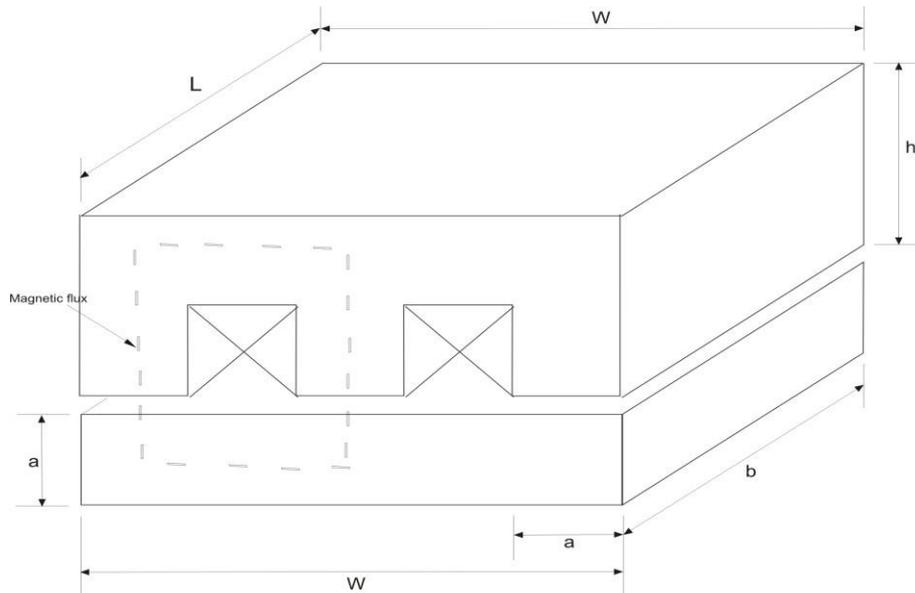


Fig 3. Magnetic core with moving part.

$$P_{el} = V.I - I^2 R = (2n\Phi' + IR)I = 2n\Phi'.I \quad (5)$$

$$= 2n\Phi' \frac{\Phi}{n \mu_0 A} \frac{2d}{\mu_0} = 2AB' \frac{2d}{\mu_0} B = 4ABB' \frac{d}{\mu_0}. \quad (6)$$

The sum of the electrical and magnetic energy will result in mechanical force,

$$P_{mag} + P_{el} = -Fd' \quad (7)$$

$$P_{mag} = P_{el} + Fd' \quad (8)$$

$$4ab \frac{B^2}{2\mu_0} d' + 4abd \frac{2BB'}{2\mu_0} = Fd' + 4ABB' \frac{d}{\mu_0}, \text{ a.b = A,} \quad (9)$$

that will become

$$F_{mag} = 4A \left[\frac{B^2}{2\mu_0} \right]. \quad (10)$$

Cylinder pressure

The variation of pressure difference between the cylinder side and on the port side of the valve produces a resultant force on the valve which is transmitted to the actuator. The computational fluid dynamic (CFD) program calculates the pressure difference and the forces at every moment for a circular valve.

$$F_{gas} = \left[\Delta P \left(\frac{\pi d^2}{4} \right) \right] \quad (11)$$

Spring force

As by linear law that is representative for the electromagnetic valve train actuator (EMVT) system, the spring force is given by,

$$F = c.x \quad (12)$$

Methods

A scaled down actuator is investigated for the effect of changing armature lift, spring rates and exhaust force on valve at varying diameters.

Table 1. Boundary conditions of the actuator.

	Standard actuator	Scaled down actuator
Oscillation time intake/exhaust	6.12 ms	4.8 ms
Speed	6000 rpm	8000 rpm
Transition time	2.9 ms	2.3 ms
Cylinder pressure at exhaust valve opening	0-7 bar	0-7 bar

Cylindrical spring with average spring constant	2x75 N/mm	2x120 N/mm
Valve lift	8 mm	6.5 mm
Operating voltage	42-55 V	42-55 V

Experimental results at valve opening and closing are carried through a test rig in the same way as real engine with the following parameters,

The test rig situation is:

- Combustion chamber volume: to be calculated
- One exhaust valve on cylinder
- Gas temperature at about ambient (293 C°)

Finding the compression volume:

$$T_{\text{engine}} = 1223 \text{ K}, \quad T_{\text{compression}} = 293 \text{ K}$$

$$\text{With } a = \sqrt{KRT} \text{ results } \frac{a_{\text{engine}}}{a_{\text{compression}}} = \sqrt{\frac{T_{\text{engine}}}{T_{\text{compression}}}} = \sqrt{\frac{1223}{293}} = 2.043$$

$$\frac{V_{\text{engine}}}{V_{\text{compression}}} = \frac{2.A_{\text{valve}} a_{\text{engine}}}{A_{\text{valve}} a_{\text{compression}}} = 4.087$$

with a cylinder of $V_{\text{cylinder}} = 200 \text{ cm}^3$ results $V_c = 200/4.807 = 50 \text{ cm}^3$

For this actuator the optimum pressure chamber volume is calculated as 50.25 cm^3 .

Experimental Results

The results obtained through oscilloscope and the data processed in Matlab [10] for the valve opening phase at 1bar absolute pressure (in the pressure chamber), the armature lift time (from valve close to fully open) is 2.8 ms. As the back pressure increases the lift time also increases due to the fact that valve is pushed against more pressure.

The transition time for the standard actuator with a spring rate of 150 N/mm and moving mass of 142 g is 2.9 ms, while the transition time for a scaled down actuator (used in project) with a spring rate of 240 N/mm and moving mass of 141 g is 2.3mm.

Since a stronger spring rates (240 N/mm, light springs) is used in this project the oscillation time is considerably reduced and a high speed for the actuator is reached.

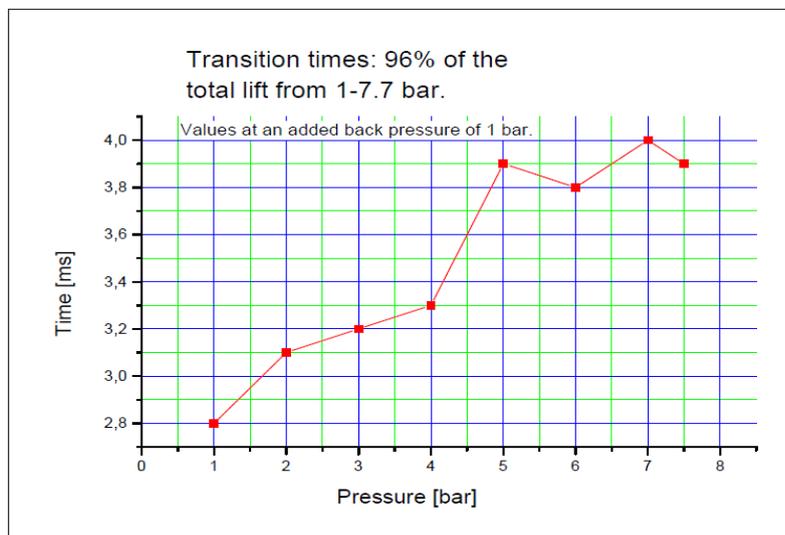


Fig 4. Transition times (2% up to 98% valve lift) from 1.0 bar to 7.5 bar absolute pressure inside the

pressure chamber.

Armature lift curve as it moves from upper magnet to lower magnet or the valve opening event. Figure 5 demonstrates that, the instant the armature starts to lift; the holding current comes to zero and the catching current starts to rise till the armature reaches its maximum lift. Catching current is more than holding current in order to overcome the friction losses.

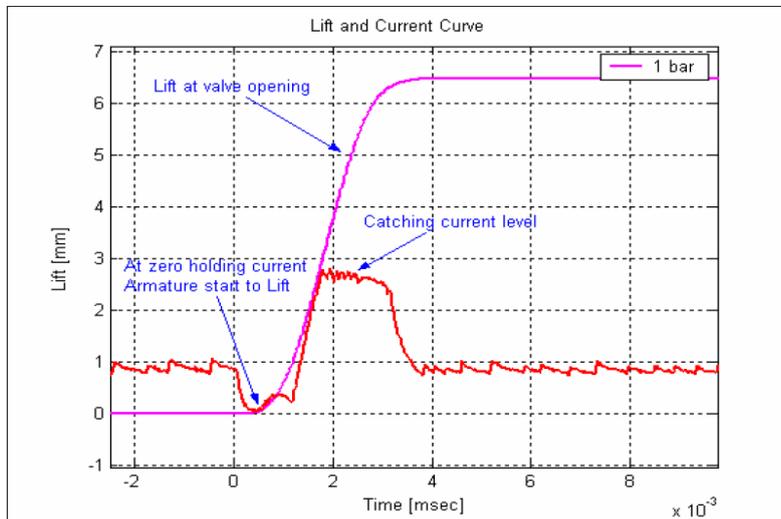


Fig 5. Starting event of the lift curve.

More catching current is needed as the pressure on the valve increases as shown in figure 6. At the start of the valve lift, holding current is almost same for all pressures because the upper magnet is at holding phase always working against the upper spring force and not against the pressure, the upper magnet holding current is independent to the pressure, more catching current is needed as the force on the valve increases while opening.

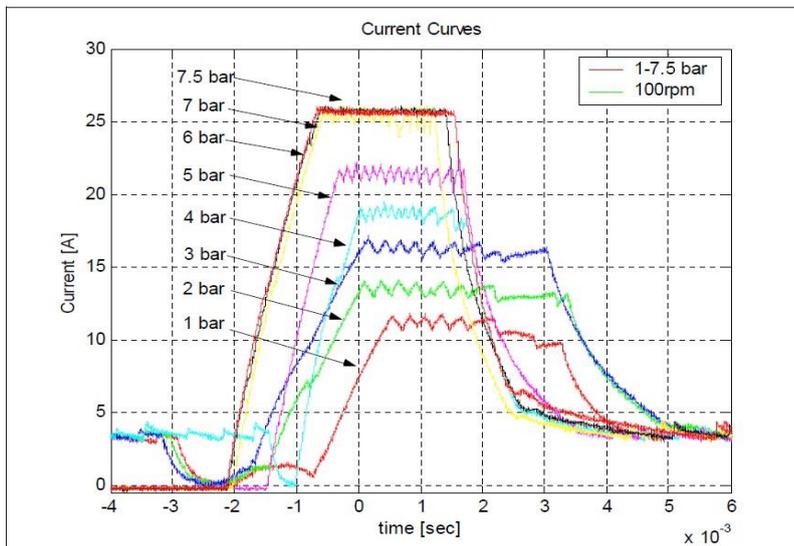


Fig 6. Current curves at a pressure from 1.0 up to 7.5 bar.

Maximum velocity of the armature reduces as pressure increases. At 1 bar pressure the maximum velocity is 3900 mm/sec while at 7.5 bar pressure the maximum velocity reduces to nearly 2000 mm/sec as shown in figure 7. A speed reduction of approximately 48% is observed. At the valve opening ,exhaust gases pressure present in combustion chamber exerts

force on the valve, the amount of this force increases as the exhaust gas pressure increases eventually resulting in a reduction of armature velocity.

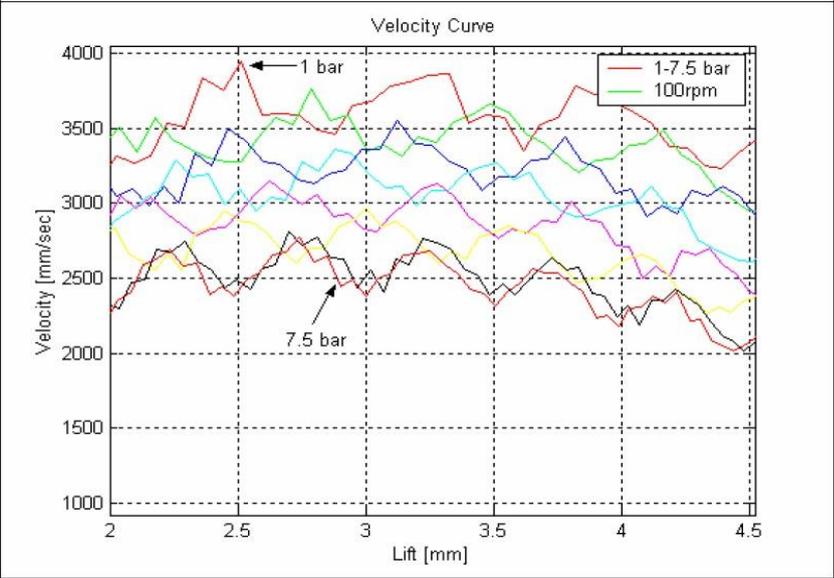


Fig 7. Velocity trace at a pressure from 1.0 up to 7.5 bar.

The kinetic energy ($1/2 mv^2$) of the system is more at the centre, due to it the system will move faster at centre as compare to the ends, the reduction in velocity would be more at the centre as compare to the ends as pressure increases, as shown in figure 8.

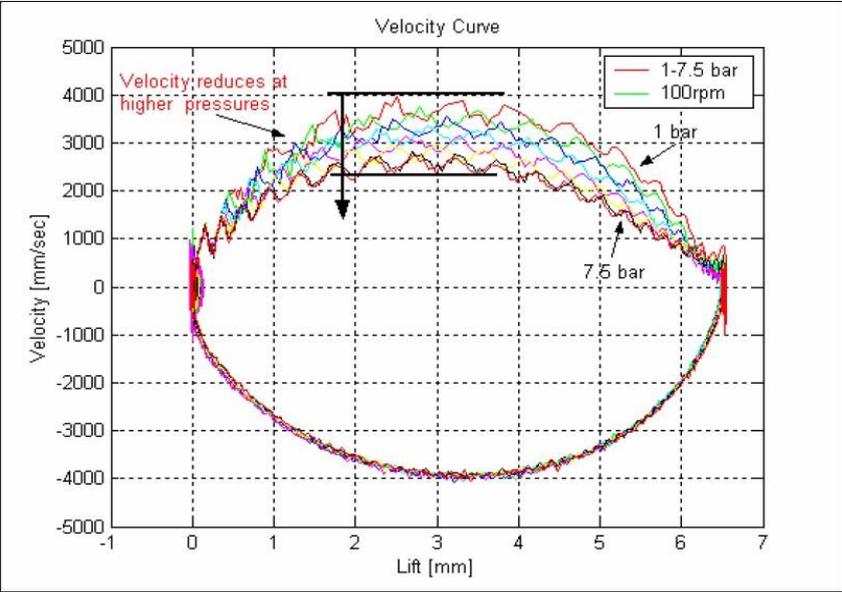


Fig 8. Velocity trace at a pressure from 1.0 to 7.5 bar.

Swings out curves are shown from 1 to 7.5 bar pressure in figure 9. Higher pressures forces on the valve causes the armature to settle down (at the centre) quickly as compare to low ones, the settling time at 7.5 bar is 0.071ms while at 1 bar the settling time is 0.15 ms. Armature lift reduces at higher pressures.

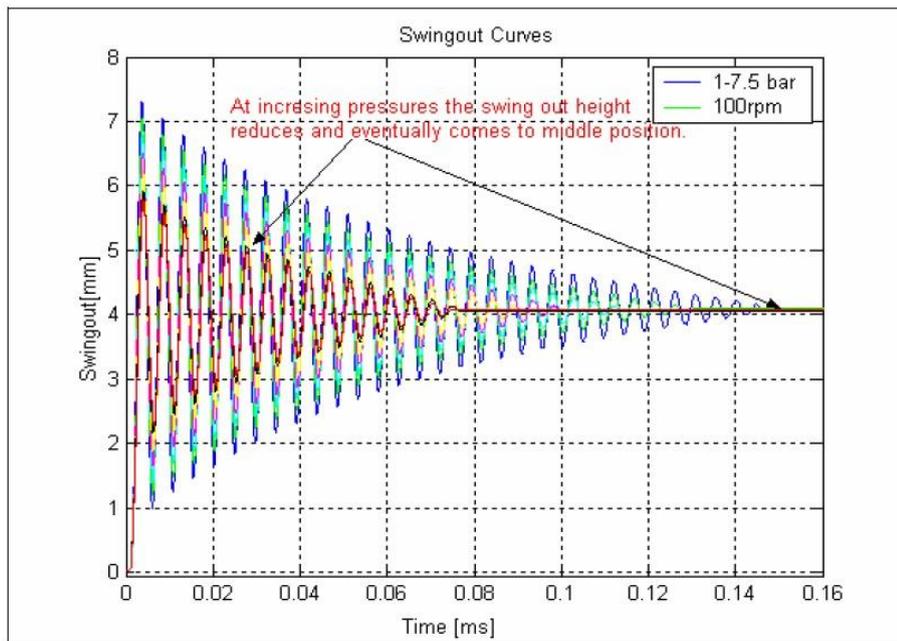


Fig 9. Swing out curves from 1 to 7.5 bar pressure.

Discussion

Higher end stop forces are needed when higher spring rates are used; it means more magnetic forces will be needed against this spring rate at end positions due to which more current is needed, resulting higher energy consumption. Magnetic force can be maximized by increasing the current at higher spring rates, but there is a limitation, a saturation point will reach beyond which the magnetic induction will not increase appreciably by giving more current. Furthermore, speed and friction of moving parts will increase results into high energy consumption.

By increasing the armature lift requires more catching current (to overcome the friction losses) that is also a loss of energy. Another issue with increasing armature lift is the limitation of magnetic induction. The greater the amount of current applied, the stronger the magnetic field in the component. But a point is reached that an additional increase in the current will produce very little increase in the magnetic flux; the material has reached a point of saturation.

As a result of induction eddy currents are built up in the core called eddies, they tend to flow in closed paths within the magnetic material and depend on the frequency, amplitude of the current and the permeability of the core material. It also generates as the flux varies due to the change in the air gap. This leads to heat losses and to a delay of the built up and decrease of magnetic field [11]. The copper losses depend on the resistance of the coil and increase with the square of the current.

The reduction of these losses is carried out through the suitable material selection and an assembly of thin insulated sheet metal which must be oriented in a direction parallel to the flow of magnetic flux.

The forces in the end positions depend on the neutral position, which is the place where the equilibrium of spring forces occurs. The existence of the valve lash produces an increase of the stored potential energy in the close position in comparison with that in open position. When the neutral position is the geometrical centre. The existence of the valve lash causes a large holding force in closed position due to the fact that only the actuator spring force is acting on the armature. The aerodynamic force, work against the armature motion due to the

air resistance, its value is negligible as compared to the main forces. The gravity force can be neglected as well for the same reason.

Conclusions

A fully variable valve train actuator is designed for motorcycle applications. Actuator is investigated at a reduce armature lift and at higher speed for single cylinder engine. Apart from design changes the effects of spring rate, armature lift and exhaust gas forces on valve are discussed.

Experimental results at valve opening and closing are carried through a test rig in the same way as real engine with a reduced chamber volume of 50 cm³, which is able to operate up to 8000 rpm engine speed, and is investigated on a test rig having a lift of 6.5 mm. The experimental results include swing out curves, velocity trace at valve opening and closing, lift curves, transition times and current trace at a pressure from 1 to 7 bar.

The transition times for opening event increases with higher pressures. The time for closing valve event is smaller than opening valve event and stays constant due to the fact that at valve closing the pressure has already disappeared. There is an appreciable loss of energy due to friction and eddy current. In the swing out curve the armature comes to its mean position as the power is switched off.

The magnetic force is sufficient to hold the armature at end positions against the spring forces, and also able to open the valve against the gas force.

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