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INVESTIGATION OF THE EFFECTS OF THE MAGNETIC CIRCUIT DESIGN PARAMETERS ON THE ELECTROMECHANICAL VALVE ACTUATORS

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Abstract: The paper describes the suitable type of magnetic circuits used in the electromechanical valve actuators (EVAs). Two different types of EVA models with the disc type of magnetic circuits were designed and the effects of various design parameters such as spring constant, moving mass, supply voltage, holding force on the system were investigated. The static and dynamic equations of the system were derived and their numerical solutions were obtained with MATLAB/Simulink program. The detailed theoretical analysis and experimental tests were carried out on the manufactured different prototypes of the EVAs. Simulation and experimental results showed a good agreement with each other.

Keywords: Electromechanical valve actuator (EVA), static and dynamic characteristics, magnetic circuit design parameters of magnetic circuit.

Manyetik Devre Tasarım Parametrelerinin Elektromekanik Supap Mekanizması Üzerindeki Etkilerinin İncelenmesi

Öz: Bu çalışmada elektromekanik supap mekanizmalarında kullanılabilecek uygun manyetik devre tipleri incelenmiştir. Disk tipi manyetik devreye elektromekanik supap mekanizması iki farklı tipte tasarlanmış ve yay sabiti, kütle, besleme gerilimi ve tutma akımı gibi çeşitli tasarım parametrelerinin sistem üzerindeki etkileri incelenmiştir. Sistemin statik ve dinamik denklemleri çıkarılarak MATLAB/Simulink programı ile sayısal çözümleri elde edilmiştir. Prototip iki sistem üzerinde deneysel çalışmalar yürütülmüş ve teorik sonuçlarla karşılaştırılmıştır. Elde edilen sonuçlardan benzetim ve deneysel çalışmaların birbiriyle uyumlu olduğu görülmüştür.

Anahtar Kelimeler: Elektromekanik supap mekanizması, statik ve dinamik karakteristikler, manyetik devre, manyetik devre tasarım parametreleri

1. INTRODUCTION

A significant improvement of engine performance can be realized by the electronic control of valve timing. As the filling taken into the cylinders changes continuously with the piston velocity of the engine, the valve timing should also changes continuously in order to have a better engine performance. In gasoline engines, fuel consumptions and emissions ratios can be reduced by the EVA which is also known as camless valve mechanism. By the use of EVAs, optimum fuel consumption for transient and idle operations and minimum pumping losses are claimed. EVAs also provide cylinder de-activation, elimination of external exhaust gas

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recirculation an improvement of engine transient behavior as well as higher maximum torque output (Park et.al., 2003; Nitu et.al., 2005; Clark et.al., 2005; Cope and Wright, 2006; Chlandy et.al., 2005; Kamis, 2005; Xrang, 2002; Eyabi and Washington, 2006; Peterson, 2005; Mohamed, 2012; Velasco, 2011; Chukwuneke et.al., 2013).

Various studies were carried out on the EVAs from the different point of views. Design features and experimental characteristics of these systems were analyzed in some of relevant topics. Static and dynamic characteristics of the EVA and the effect of design and operating parameters on the EVA were investigated in Park et.al., 2003; Nitu et.al., 2005; Clark et.al., 2005; Cope and Wright, 2006; Kamis, 2005; Xrang, 2002; Mohamed, 2012. E-core structure were used in the most designs (Park et.al., 2003; Nitu et.al., 2005; Clark et.al., 2005; Xrang, 2002; Velasco, 2011). Disc and plunger types of magnetic circuits have also been used in the design of the EVA. The usefulness of disc and plunger types of electromagnets in the EVA design was investigated in Cope and Wright, 2006; Kamis,2005. Chlandy et.al., 2005 used an elliptical cross section in the design of EVA and modeled the system.

The other works done on the EVAs are generally classified as the modeling, simulation and control of the designed system. The Finite Element Method (FEM) was used in predicting actuator characteristics in Park et.al., 2003; Nitu et.al., 2005; Clark et.al., 2005; Chlandy et.al., 2005; Xrang, 2002. Besides, MATLAB/Simulink environment were also used to analyze the system (Kamis, 2005; Eyabi and Washington, 2006; Velasco, 2011). As EVA is a nonlinear system and has instability for small air gaps, it has been claimed that nonlinear control techniques are necessary in order to provide lower contact velocities (Eyabi and Washington, 2006; Peterson, 2005; Velasco, 2011; Chukwuneke et.al., 2013).

This paper describes two different EVA models with disc types of magnetic circuits. Static and dynamic characteristics are investigated for different design parameters such as spring constant, moving mass, supply voltage, holding force on the system are investigated. The governing equations of the system are derived and their numerical solutions are obtained by the program that runs in MATLAB/Simulink environment. Experimental tests were carried out on the manufactured prototypes of the EVAs in order to verify the simulation.

2. ELECTROMECHANICAL VALVE ACTUATORS (EVAs)

The main components of the EVA are two magnetic circuits- one for opening and one for closing of the valve-, two opposing spring coils and a disc moves with the valve (Fig. 1). The disc and the magnetic circuits are made of ferromagnetic material and the disc moves between two electromagnets under the effect of the spring and the magnetic forces. Two different EVA models using the disc type of the magnetic circuits, namely Model 1 and Model 2, are shown in Fig. 1. While the springs are located to the outside of the magnetic circuit in Model 1, they are located inside of the magnetic circuit in Model 2 so that a more compact configuration could be obtained.

Although there are various types of design configurations for the EVAs, their operating principles and main components are similar. The valve can be brought to open or closed position by energizing one of the coils and at the same time de-energizing the other via the signal sent from the electronic control unit. When two coils of the magnetic circuits are de-energized, the moving element is held at balancing position by the springs. The springs provide primary force to open and close the valves and the electromagnetic force from the magnetic circuits provides control force.

Without magnetic force and frictional force, the dynamic behavior of the EVA is equivalent to a simple mass-spring system. The disc is accelerated by the aid of the spring force and it completes main part of its motion by the energy stored in the springs. The magnetic force created by electrical energy applied to the coils becomes effective at the completing of the movement and controls the opening and closing of the valves.

The motion time of the EVA must be as short as possible in order to provide a sufficient performance at the high engine speeds. The response time of the EVA does not depend on the speed and the load of the engine as it is mechanical valves. But it depends mainly on the natural frequency of the mass-spring system and dynamics of the electromagnets to a certain extent. The natural frequency must be high for providing a sufficient performance at the high engine speeds. The valve motion follows a law similar to a sine function. The motion time of the valve is equal to the half - period of the oscillation of a harmonic motion. Thus

$$t_c = \frac{T}{2} = \frac{\pi}{\omega_n} = \pi\sqrt{\frac{m}{K}} \tag{1}$$

As it can be seen from Eq. (1), either the mass of the moving elements must be reduced or the spring constant must be increased in order to reduce motion time of the valve.

The response time of the EVA also depends on the dynamic characteristics of the electromagnet to a certain extent. Therefore, most suitable magnetic circuit must be chosen and its design must be carried out accordingly for each specific EVA. In this study, two different types of the EVA models are designed with the disc type of magnetic circuits. The geometric configurations of the disc type of the magnetic circuits are shown in Fig. 2. The disc type of magnetic circuits are intended to produce a large force acting through a relatively short stroke (Kamis, 2005; Şefkat, 2010). They are characterized by having a magnetic circuit of extremely short length and large sectional area with two working gaps. These gaps are mechanically in parallel and magnetically in series; hence they produce a large effective holding surface area.

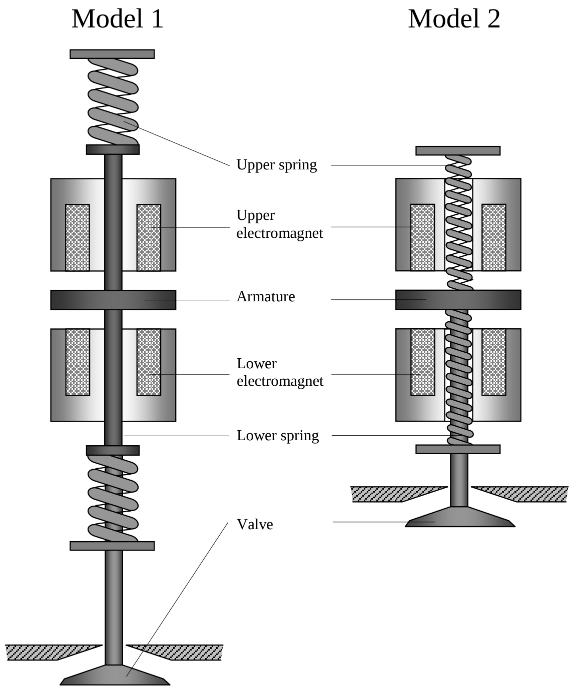


Figure 1:
Magnetic actuator configurations used in the design of the EVAs

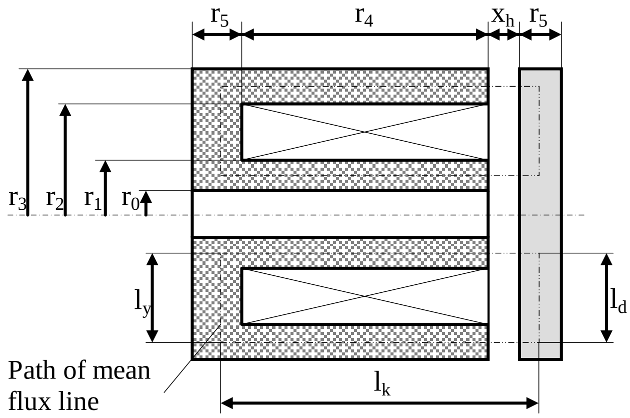


Figure 2:
Disc type of magnetic circuit used in the EVAs

3. MATHEMATICAL MODELING

Referring to Fig. 1 and Fig. 2, the model of the EVA can be considered as two distinct sub-systems, namely the electromagnetic actuator with electromagnetic relationships (or equations) and the mechanical part, consisting of a spring-mass system.

3.1 Electromagnetic subsystem

Static and dynamic characteristics of electromagnetic subsystems are governed by inherently nonlinear electromagnetic relationships. In a practical case, the main electromagnetic relationships constitute electrical circuit equation, magnetic circuit equation and magnetic force equation. Referring to Fig. 2, these equations are expressed as follows:

Electrical circuit equation: In the electromagnetic circuit, the constant input voltage across coil windings gives rise to an electrical current and then flux linkage of coil turn that can be described by the following set of coupled equations (Şefkat, 2010). After some simplifications, the electrical circuit equation can be expressed as

$$e = Ri + N\frac{di}{dt} \quad (2)$$

Magnetic circuit equation: This equation describes the exciting magnetomotive force NI, which must provide the flux required for given force and displacement and it is of the following form

$$NI = z\frac{B_g x_t}{\mu} + z\sum H_i l_i \quad (3)$$

Force equation: A simplified force equation can be expressed as a function of the flux density in the working gap as follows

$$F_m = \frac{z}{2}\frac{B_g^2 A}{\mu} \quad (4a)$$

where z is 2 for the disc type of magnetic circuit.

The force equation can also be expressed as a function of the total air gap and the current as follows

$$F_m = \frac{\mu A (NI)^2}{2 z x_t^2} \quad (4b)$$

This equation can be shown as a result of substituting B_g from Eq. 3 into Eq. 4a. Here the magnetomotive force of the iron part $z\sum H_i l_i$ is neglected which is about %30 of total MMF. Therefore, saturation characteristics of the magnetic material are neglected in equation 4b. But this equation gives an insight of force relationship with the working air gap and the current.

3.2 Mechanical subsystem

By the application of the Newton's second law of dynamics to the moving elements which move with the effect of the magnetic force is:

$$F_{me} = m\ddot{x} + F_{fr} \pm F_{spr} + F_{pr} + F_{mde} \quad (5)$$

The resultant spring force is expressed as:

$$F_{spr} = K\left(\frac{x_m}{2} - x\right) \quad (6)$$

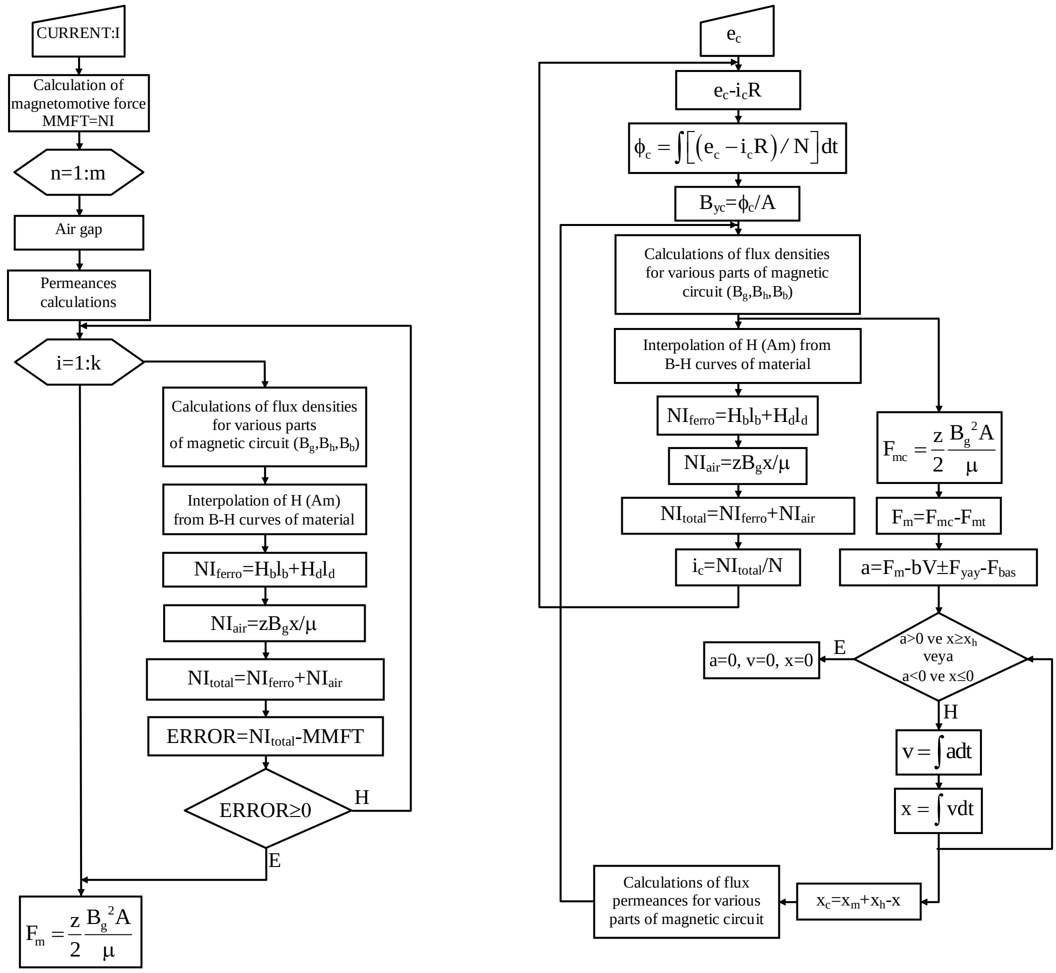
The pressure force is critical only at the beginning of the valve movement. Therefore the effect of the pressure is neglected for sake of simplicity of solutions of dynamic equations.

4. NUMERICAL SOLUTION OF THE SYSTEM EQUATIONS AND SIMULATION RESULTS

The accurate solutions of electromagnetic coupled equations are very difficult owing to the effects of magnetic leakage and the nonlinear relationship between magnetic flux and magnetic intensity in ferromagnetic materials. Therefore, they cannot be solved in closed form to determine dynamic characteristics of the electromechanical system. Approximate solutions of the equations may be obtained using small perturbation techniques, finite element methods or numerical computation methods. MATLAB/Simulink environment have been used to solve nonlinear equation of the system due to their ease of usage.

In this work, a MATLAB program is used to solve magnetic force equation with the magnetic circuit equation in order to obtain the static characteristics of the electromagnets (Kamis, 2005; Şefkat, 2010). The program is based on permanence formulae and leakage coefficient of the dimensioned magnetic circuits given in Roters, 1941. The flow chart of the program is shown in Fig. 3a. The program estimates various characteristic values by using the data taken from the magnetization curve (B-H curve) of the material used in the design of the magnets. It calculates/estimates the necessary magnetic force at each step of incremental air gap for a given electrical current. The output of the program can be organized for various static characteristics of the magnetic circuits such as force-displacement, magnetomotive force–flux density and also 3-D graphics of displacement-current-flux density characteristics is possible.

The dynamic characteristics of the electromechanical system based on the nonlinear equations (2-7) are obtained from a simulation program created by Simulink (Kamis, 2005; Şefkat, 2010). The flow chart of the simulation program is shown in Fig. 3b. As the upper and the lower magnets are symmetrical, only one side of the EVA is given in the figure. As it can be seen from the figure the simulation starts to estimate the flux distributions with the data taken from the B-H curve. The outputs of the program are dynamic characteristics of the EVA such as current, displacement, velocity, force, etc.



a. Static solution b. Dynamic solution

Figure 3:
Flow charts of numerical solutions

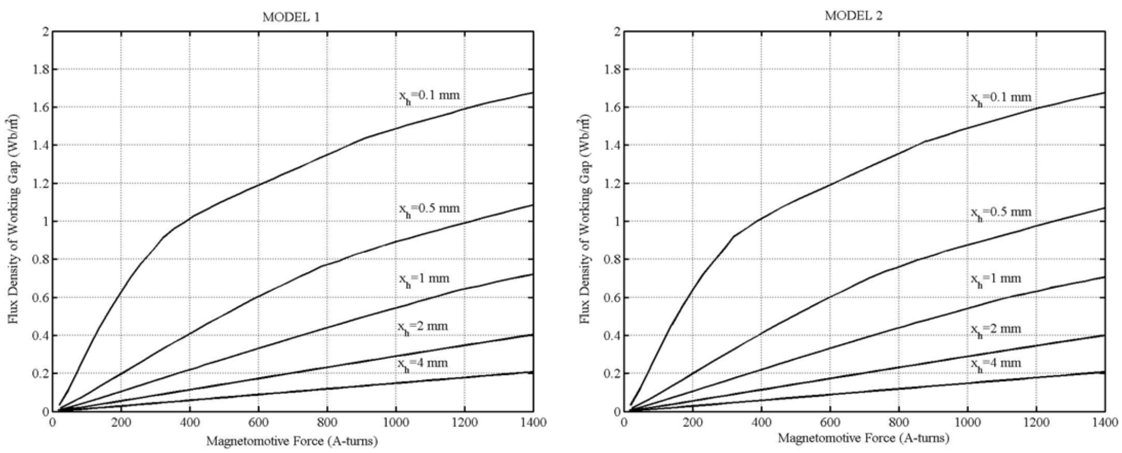


Figure 4:
Magnetomotive forces versus flux density

Fig. 4 shows magnetomotive force-flux density characteristics of the models. As it can be seen from the figure, there is a proportional relationship between input (magnetomotive force, NI) and output (the flux density of working gap, B_g) of the magnetic circuit in a certain region. Increasing of the flux density with magnetomotive force is linear in a certain region. Then it reaches saturation at a bending point for especially short working gaps. After the saturation point, if the magnetic circuit is forced to drive with more flux, the magnetic circuit will warm up and the magnetic efficiency will decrease. In practice short working distance corresponds to the holding position of the valve; hence, the current should be reduced to a sufficient level just enough to hold the moving parts.

In the designed models, as it can be seen from Fig. 4, the magnetization characteristics reach to the saturation point at approximately 0.5 mm and at less of working distances. On the other hand, the saturation does not occur at long working distances (approximately 2 mm and more). In practice, this situation corresponds to the beginning of motion of the moving parts. Therefore, high currents can be applied to the system without having any saturation problems at the beginning of the motion in order to get a short response time. After the moving part was held, the current can be reduced to an unsaturated region so that an optimum working condition of the magnetic circuit is determined.

Fig. 5 shows 3-D of magnetic force variations versus current and working distance for the models. The force-displacement curves are hyperbolic in shape and show very low force at relatively high distances and very high force at very short distances.

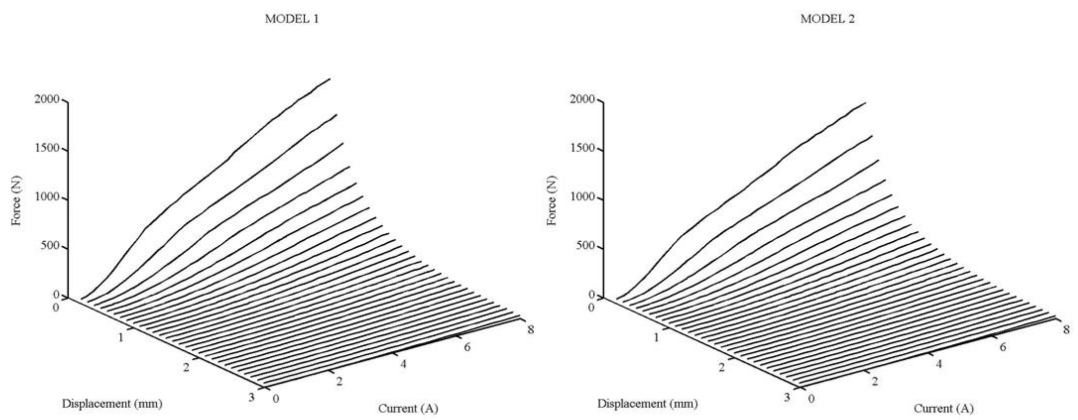
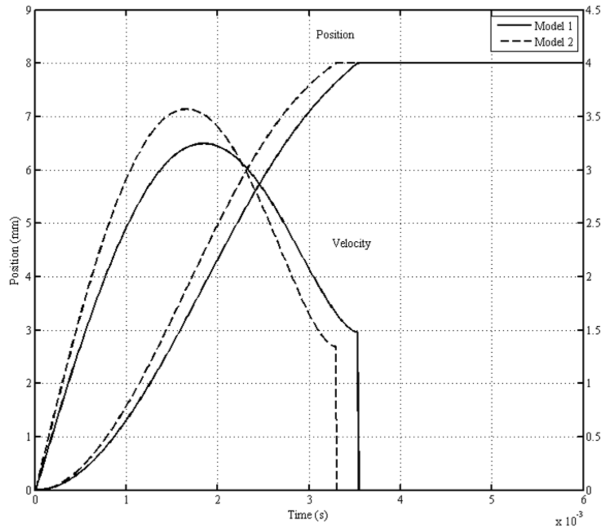


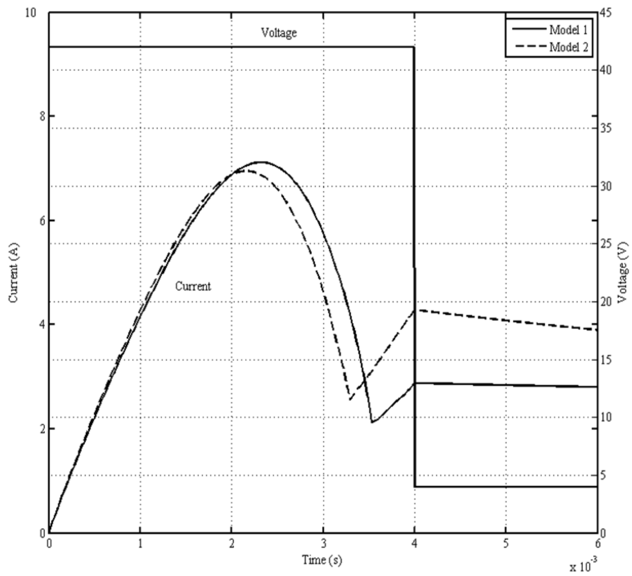
Figure 5:
3-D of magnetic force variations versus current and working distance

Fig. 6 shows the dynamic characteristics of the models obtained by the application of two levels of input voltage signal. The effects of friction force, the pressure force in the cylinder and holding current are neglected for the sake of obtaining ideal solutions.

The total moving mass of Model 1 constitutes a disc type of armature, linkage rods, valve and springs and it is approximately 246 g. As the springs are accommodated inside the magnet, the size of linkage rods is reduced in Model 2; hence, the total moving mass is 177 g. When the position variations shown in Fig. 6a are examined, it can be seen that the motion time of the valve is 3.7 ms in Model 1 and 3.2 ms in Model 2 for ideal conditions. As the total moving mass of Model 2 is less than Model 1, the motion time is shorter.



a



b

Figure 6:

Comparison of dynamic characteristics for the models

As far as the velocity changes of the system is concerned as shown in Fig. 6a, the impact velocities of Model 1 and Model 2 are 1.88 m/s and 1.97 m/s respectively. The impact velocities of the models are very high according to the cam driven valve system. Therefore, control techniques, such as feedback control, are required in order to reduce the impact velocity of the EVA.

Fig. 6b shows typical coil current characteristics of the models when a constant supply voltage is applied. Characteristic shape of current curves is due to change of inductance with the movement of the disc. As the disc starts to move, the inductance of the coil starts to increase due to the decrease gap between the disc and the attracting magnet. As the gap decreases, the effective time constant increase thereby causing the current to decrease to its local minimum when the disc reaches its final position. The disc in Model 1 and Model 2 finish its motion at the local minimum as it is expected.

5. EFFECTS OF THE DESIGN PARAMETERS ON THE DYNAMIC CHARACTERISTICS

5.1 Spring Constant

Fig. 7 shows effects of the spring stiffness on dynamic characteristics of the models. The response time decreases with the increase of spring constant. But beyond 154 N/mm of resultant spring constant, a jump occurs in the motion of the mass. In Model 2, the jump is much more remarkable. It is obvious that the jump will increase with the increase of the spring constant for the models.

Increasing stiffness of the springs is one way of improving response time of the EVA. But if the spring stiffness is increased too much the response time of free oscillating system decreases to a level that the magnetic force cannot develop in a sufficient level during the free response time. Hence the magnetic circuit cannot hold the moving mass and the moving mass starts to move in opposite direction. If the magnetic force continues to develop to a sufficient level during reverse movement, the magnet attracts the moving mass. Otherwise the moving mass oscillates until it reaches its equilibrium position. In this case size of magnetic circuit must be increased in order to get a reliable valve performance. In return this makes magnetic circuit very bulky and increased mass of moving parts of the system. Therefore, the response time of the system cannot be reduced continuously to increase the spring stiffness.

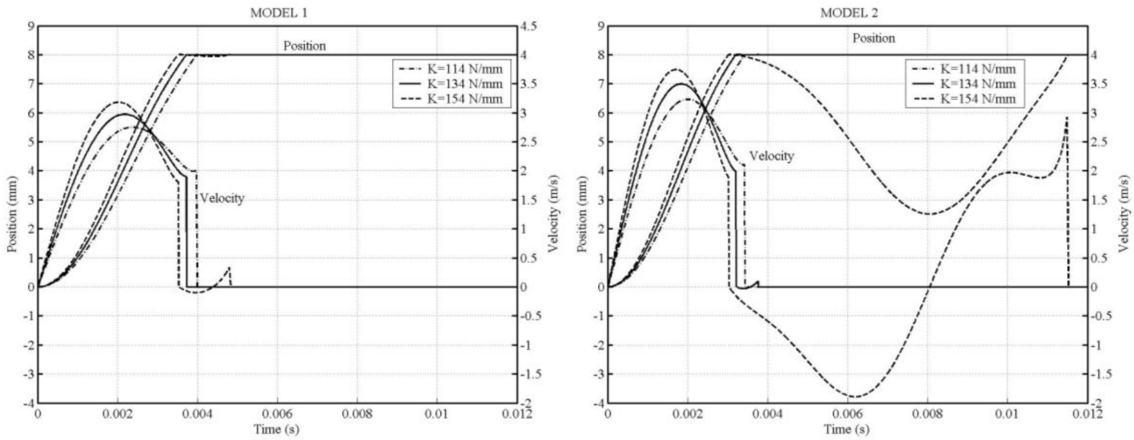


Figure 7:
Effect of spring stiffness on the dynamic behavior

If the size of the magnetic circuit and the moving mass are held constant, an increase of spring constant up to a certain value can reduce the response time. Hence, firstly the spring constant must be determined with consideration of the pressure inside the cylinder of the engine. The size of the magnetic circuit can be calculated according to this spring constant in the design of the EVA. Thus, the size of the magnetic circuit and the spring stiffness must be consistent with each other.

5.2 Moving mass

Fig. 8 shows the effect of mass variation on dynamic characteristics of the models. The highest values of mass shown in Fig. 8 belong to nominal size of magnetic circuits. Mass reduction is realized by reducing the thickness of disc. It is seen clearly from Fig. 8 that the response time decreases with the reduction of the mass.

Size of moving mass is determined according to magnetic circuit. If the size of the magnetic circuit is increased, the weight of the moving mass increases. Natural frequency of free oscillating mass-spring system can be increased by reducing weight of mass for a certain spring constant (see equation (1)). A main part of the moving mass of the EVA constitutes disc of the magnetic circuit.

5.3 Input (supply) voltage

Fig. 9 shows effects of supply voltage variations on dynamic characteristics of the models. Increasing of the supply voltage decreases the response times of the models. High supply voltage drives a high current and hence high magnetic force at the beginning of motion. Although it reduces the response time, high supply voltage increases energy consumption and heat in the magnetic circuit. As it can be seen from Fig. 9, increasing in the supply voltage is not particularly effective on the response times. This is due to the main motion energy is taken from the springs, not from the magnet. The magnetic force controls the motion and keeps the valve at open or closed position.

When the models are compared from energy consumption and response time points of view, it seems that Model 2 is most advantageous.

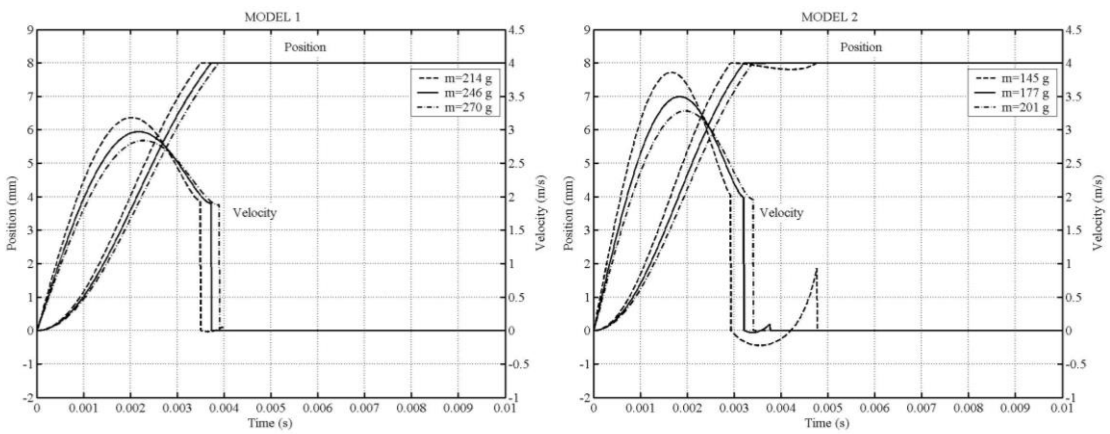


Figure 8:
Effect of moving mass on the dynamic behavior

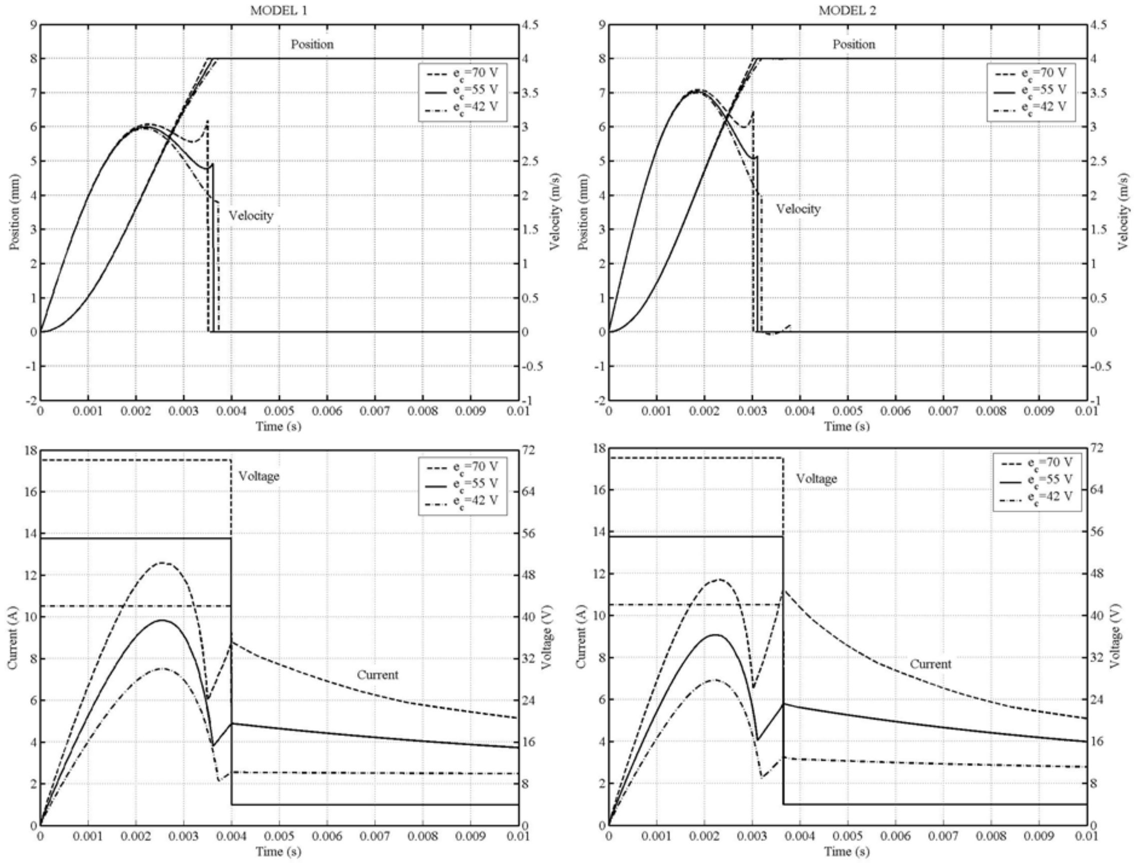


Figure 9:
Effect of the supply voltage on the dynamic behavior

5.4 Holding force

Dynamic behavior of the EVA is affected by current decay in de-energized magnet as well as spring force, magnetic force and pressure force. The current and the holding force decay in the de-energized magnet are not instantaneous but they follow a transient state. The holding current slows down at the beginning of motion. Fig.10 shows the effects of holding forces corresponding to holding currents on the response time of the models. As it can be seen from Fig. 10, the increase in the holding force is remarkably effective on the response times. Thus, the holding current should be adjusted to a value just enough to drive a magnetic force to hold the moving mass against to the spring force.

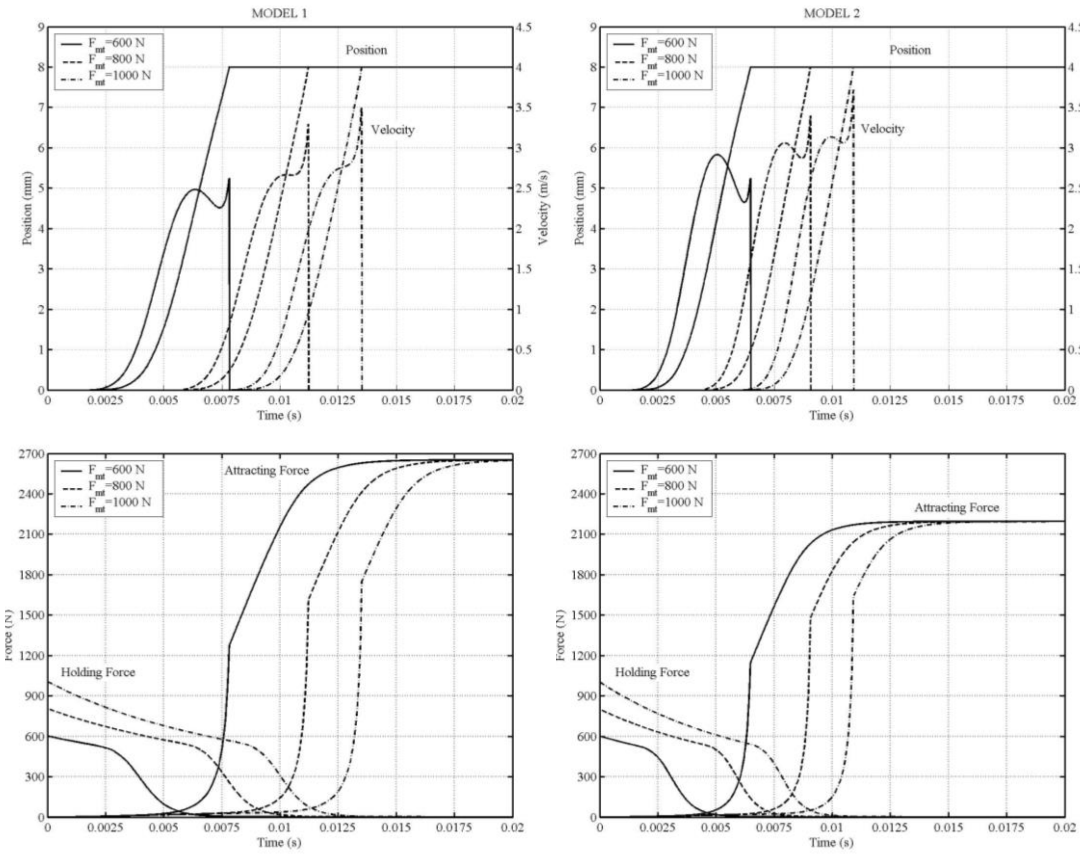


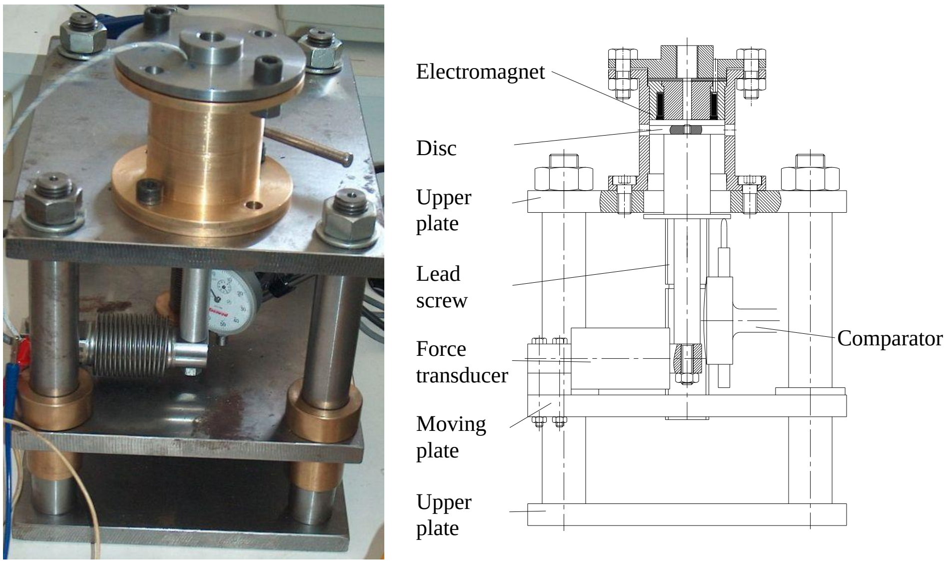
Figure 10:
Effect of the holding force on the dynamic behavior

6. EVA PROTOTYPES AND TESTING

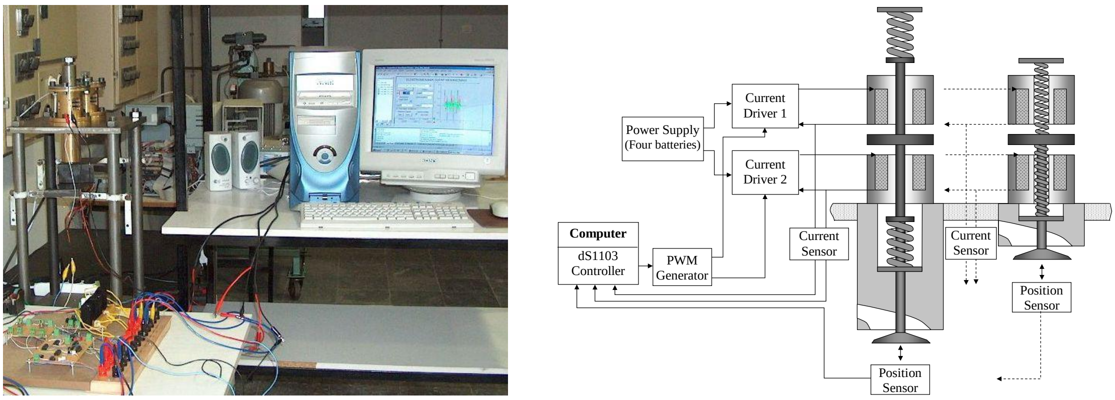
6.1 Static tests and results

The prototypes of the models are manufactured and static and dynamic tests are carried out on them. The outer size of Model 1 and Model 2 are the same. The main difference between two models is the size of hole drilled inside the magnet. Diameter of the hole is 6 mm in Model 1 and is 14 mm in Model 2. 0.5 mm of diameter wire is used with 156 turns on the bobbin for both models. The resistances of bobbins are 1.7Ω.

Static test setup of the EVA consists of an upper plate where the magnetic actuator is fixed and a moving plate where a force transducer is located (see Fig. 11a). Displacement of disc type of armature is adjusted by a screw mechanism and is measured by a comparator. During the measurements, the disc is fixed at a given distance and then the input voltage is applied causing a current through the coil. The resultant force at a certain displacement is measured by the force transducer.



a. Static test setup



b. Schematic diagram and picture of dynamic test setup

***Figure 11**:*
Experimental setups

The experimental results are shown in Fig. 12 where force versus displacement is plotted for both models. The results indicate good agreement with the results obtained from the simulations. However, a slight diversity occurs at short distances. This is mainly due to the difficulty to hold the disc steady at a distance close to the coil.

Fig. 13 shows simulation and experimental results for different disc thicknesses. Comparison is made with respect to 7.5 mm of nominal disc thickness of the designed magnet. When the disc thickness is reduced to 6 mm (about 8.8-8.9 % of reduction in whole weight of the two models), there is hardly any decrease in the magnetic force. But this reduction improves about 5-6 % of dynamic performance of the system (see Fig. 8). When the disc thickness is reduced to 4 mm which is almost 47 % of less than nominal thickness, static and dynamic performances are appreciably deteriorated.

6.2 Dynamic tests and results

A dynamic test setup was arranged for Model 1 and Model 2 as shown in Fig. 11b. The prototypes of Model 1 and Model 2, a PC computer with a controller board (ds1103) and the relevant software packages, four batteries connected in series for driving power source, two current driving circuits and a simple optic position sensor constitute the test setup.

Model 1 and Model 2 were tested seperately and the results are shown in Fig. 14. Two-level of step input voltages were applied to the coils of the EVAs and the changes of the valve position and coil currents were recorded as shown in Fig. 14.

The total switching time of the valve is evaluated as the sum of the valve delay time and the valve motion time. The total response time of opening or closing of Model 1 was recorded as 8.8 ms whereas the motion time is 4.7 ms.

The total response times of Model 2 were about 15.7 ms for opening of the valve and 13.4 ms for closing of the valve which were rather high comparing with the response time of Model 1. On the other hand, the motion time of Model 2 is 4.4 ms which is even less than the motion time of the Model 1. Therefore the high delay time of Model 2 can be interpreted as the result of extra friction forces due to some problems encountered during the manifacturing and the assembling of the prototype of Model 2. The simulation results confirm this situation as shown in Fig. 14 where higher friction forces than Model 1 were taken account. Therefore, it is envisaged that this problem can be overcome with a careful manufacturing and assembling of the prototype.

The simulation results are compared with the experimental results and they show a good agreement as it can be seen in Fig. 14. The friction forces, the retarding effect of holding current and resistance change in coil due to heating effects were taken account in the simulation models. From these results it can be asserted that when the friction forces are reduced a minimum level and the effect of the holding current are eliminated, the response time of the valve can be improved.

7. CONCLUSION

In this study, two EVA models using the disc type of magnetic circuits are investigated. Mathematical models of the systems are established and static and dynamic characteristics are obtained by the aid of MATLAB/Simulink programs. The effects of various design parameters on the dynamic behavior of the models are investigated. The prototypes of the models were manufactured. Static and dynamic tests that were carried out on the prototypes showed a good agreement with simulation results.

The tests done with various disc thicknesses showed that excessive reduction in disc thickness decreased the magnetic force. Dynamic tests and simulation results have shown that friction forces remarkably effected on the response time of the EVA.

When Model 1 and Model 2 are compared with each other, Model 2 seems theoretically more favorable from the points of view of response of the motion time, impact velocity and energy consumption. Also the length of Model 2 is shortened since the springs can be accommodated inside the magnets. Therefore this makes the configuration of Model 2 more compact.

It should be realized that the phenomenon of high impact velocities occurring in electromechanical valve actuators is an important problem awaiting solution. The work is continuing and in the future work it is going to be concentrated this phenomenon.

REFERENCES

1. Park, SH., Lee, J., Yoo, J., Kim, D., Park, K. (2003) Effects of design and operating parameters on the static and dynamic performance of an electromagnetic valve actuator. Journal of Automobile Engineering, 217: 193-201. doi: 10.1243/09544070360550480
2. Nitu, C., Gramescu, B., Nitu, S. (2005) Application of electromagnetic actuators to a variable distribution system for automobile engines. Journal of Materials Processing Technology, 161: 253-257. doi:10.1016/j.jmatprotec.2004.07.032
3. Clark, RE., Jewell, GW., Forrest, SJ., Rens, J., Maerky, C. (2005) Design features for enhancing the performance of electromagnetic valve actuation systems. IEEE Transactions on Magnetics. 41-3: 1163-1168. doi:10.1109/TMAG.2004.843342.
4. Cope, D., Wright, A. (2006) Electromagnetic Fully Flexible Valve Actuator, SAE Technical Paper 2006-01-0044, doi:10.4271/2006-01-0044.
5. Chladny, RR., Koch, CR., Lynch, AF. (2005) Modeling automotive gas-exchange solenoid valve actuators. IEEE Transactions on Magnetics 41-3: 1155-1162. doi:10.1109/TMAG.2004.841701
6. Kamis, Z. (2005) The investigation of design and control of electromechanical actuator for engine valves (in Turkish). PhD Dissertation. University of Uludağ.
7. Xiang, JY. (2002) Modeling and control of a linear electro-mechanical actuator (LEMA) for operating engine valves. Industry Applications Conference, 37th IAS Annual Meeting. Conference Record of the (Volume:3). doi: 10.1109/IAS.2002.1043799
8. Eyabi, P., Washington, G. (2006) Modeling and sensorless control of and electromagnetic valve actuator. Mechatronics. 16:159-175. doi:10.1016/j.mechatronics.2005.11.008.
9. Peterson, KS. (2005) Control methodologies for fast & low impact electromagnetic actuators for engine valves. Dissertation. The University of Michigan.
10. Mohamed, E. (2012) Modeling and performance evaluation of an electromechanical valve actuator for a camless IC engine. International Journal of Energy and Environment. 3-2: 275-294. ISSN: 2076-2895
11. Velasco, CIH. (2011) Modelling, analysis, control and experimental validation of electromechanical valve actuators in automotive systems. Dissertation. University of Naples Federico II.
12. Chukwuneke, JL., Achebe CH, Okolie PC, Anisiji OE (2013) Modelling of electromechanical control of camless internal combustion engine valve actuator. International Journal of Science and Engineering Investigations. 2-15: 123-136. ISSN: 2251-8843
13. Topcu, EE., Kamis, Z., Yüksel, I. (2008) Simplified numerical solution of electromechanical systems by look-up tables. Mechatronics. 18:559–565. doi:10.1016/j.mechatronics. 2008.05.006
14. Roters, HC. (1941) Electromagnetic devices. John Wiley, USA.
15. Sefkat, G. (2010) Investigating static and dynamic characteristics of electromechanical actuators (EMA) with MATLAB GUIs. Computer Applications in Engineering Education. 18-2:383-396 (2010) doi: 10.1002/cae.20279