

On the Development of a Micro-Scale Magneto-Static Actuator for a Planar Fuel Valve

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Abstract

A development effort for a liquid fuel delivery system for small-scale engines is ongoing whereby a magneto-static actuator is integrated with a silicon 'hole-in-wall' valve for portable power generation applications. The magneto-static actuator is designed to accommodate approximate 50mg/sec of fuel flow (2200 W) by moving a linearly actuated armature in a planar micro-valve. FEM analysis demonstrates linear force of 1.4mN in the valve moving direction while 96mN in gap closing direction. As such, two pairs of folded suspensions are adapted high stiffness ratio (860) between the valve moving direction and gap closing direction. Results of FEM analysis indicate 190 μ m of displacement with a peak stress of 106MPa of maximum stress exerted within the suspension.

Keywords: Fuel Delivery, Magneto-static linear actuator, Magneto-static forces, Folded- suspension,

1 - INTRODUCTION

The performance of small-scale internal combustion engine is greatly impacted by the performance of fuel delivery system [1]. Delivering accurate amounts of fuel is critical to engine performance because the air/fuel mixture determines the stoichiometry, which in turn determines the kinetics of the ignition event as well as heat release rate. Additionally, droplet size is critical for engine performance due to limitations in evaporation and mixing times at low turbulence levels [2].

Previous work has examined a fuel delivery system which consists of two main components: a fuel injector and a pressurized fuel tank [1]. In this system, for a given delivery pressure ranging from 5-30psig, the fuel injector determines droplet size (200-350 μ m) and the fuel valve determines stoichiometry of air/fuel mixture ($0.4 < \Phi < 1.2$).

Due to the presence of fuel in and around the valve, as well as large actuation distances, a magneto-static design was selected over a latch and release electro-static drive mechanism. Further, electro-static actuation is known to not work in a conductive fluid medium [3].

Since the armature is designed to freely move, there is some gap between the top and bottom sealing face. This fluid will not affect the sealing design except to limit the maximum operation speed due to film damping. The current development effort will create an electronically controlled, reduced scale, magneto-static, metering valve based on the previously demonstrated, passive, silicon 'hole-in-wall' valve, (Figure 1) [4, 5].

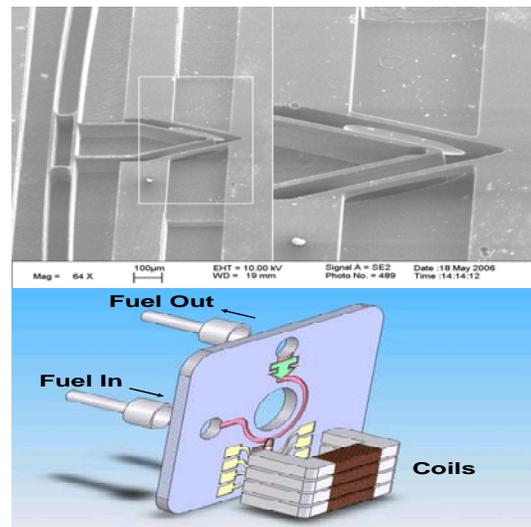


Figure 1 – (TOP) SEM image of a recently fabricated 'hole-in-wall' valve, showing a rectangular fluidic channel with a fluidic control valve. **(BOTTOM)** Schematic of an integrated magneto-static actuator and a valve

The 'hole-in-wall' designs demonstrated feasibility of a micro fluidic pumping system with two integrated valves in silicon and comparability with Isopropanol using thermal bubbles for fluid delivery [4].

Similar to the previous design [5], the device is expected to be fabricated on a SOI wafer with a device layer of 100 μ m and channel width of 400 μ m. In addition, electroforming of

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NiFe(50/50) to form the magneto-static actuator is to be carried out subsequent to the ‘hole in wall’ process. Fluidic delivery pressure is limited in this design to 40kPa to prevent fluid leakage and safety issues. For this delivery pressure, the valve is designed to accommodate a maximum of 50mg/sec of fuel.

2 - LINEAR FORCE ESTIMATION

Linear force of a magneto-static actuator is approximated as the following.

$$F = \frac{\partial W}{\partial x} = \frac{1}{2} \frac{\partial L}{\partial x} i^2 \quad (1)$$

where L is the self inductance and i is input current

The self inductance can be approximated as $L = N^2 P$ where N is number of coil turns and P is permeance [Weber per ampere-turns].

Linear actuation force, therefore, becomes a function of $\frac{\partial P}{\partial x}$, and it can be shown as the following.

$$F = \frac{1}{2} N^2 i^2 \frac{\partial P}{\partial x} \quad (2)$$

The permeance, P, as a function of an armature location, x, is therefore:

$$\sum P(x) = \frac{1}{4} n_t \mu_0 h [C_0 + C_1 \cos n_t x + C_2 \cos 2n_t x + \dots] \quad (3)$$

Expressions (2) and (3) give the linear force as a function of distance, x:

$$F = -\frac{1}{8} N^2 n_t^2 i^2 \mu_0 h [C_1 \sin n_t x + 2C_2 \sin 2n_t x + \dots] \quad (4)$$

where N is total number of coil turns in the phase

3 - DESIGN OF A MAGNETO-STATIC ACTUATOR

The proposed design of a magneto-static linear actuator is shown in figure 2 where the four coils, shown in red, are simplified in structure. The overall length and width of the actuator is designed as 14mm and 7mm respectively. The design has not been optimized either for die layout area or coil integration. The actuator is designed to have an actuation distance of 200 μ m in the forward and reverse positions. The distance between the armature and flux guides is set as 20 μ m to maintain mechanical strength with minimal magnetic flux losses due to fringing fields.

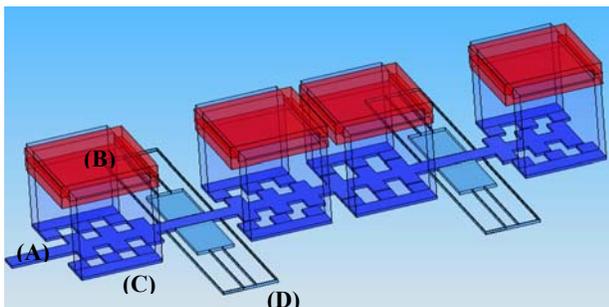


Figure 2 – Proposed design of a magneto-static linear actuator. (A) Armature, (B) Coil, (C) Flux guide and (D) Suspension

Current design of the actuator controls fluid flow in different modes (closed, half-open and wide-open) depends on a location of an armature (Figure 3). It is noted that the design can effectively create fine motions along the direction of travel with controls on the phasing of the current.

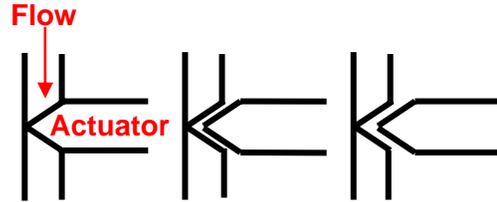


Figure 3 – Different positions of the magneto-static linear actuator (Left: fully closed, Center: half-open, Right: wide-open) in the three static positions. Active control enables any position to be maintained.

4 - FINITE ELEMENT ANALYSIS

4.1 - Magneto-Static Forces Analysis

A model of micro-scale actuator has been built with a CAD tool (Solidworks), and simulated with a FEM package (Cosmoses). This model is utilized material properties of Silicon (Si) for suspensions, Copper (Cu) for coils, Nickel Iron (NiFe) for the armature and flux guides, and the surrounding medium as air (Table 1). It is assumed that the properties of materials are independent of orientation (i.e. isotropic). Despite the unsupported hype regarding treating magnetic properties of fuel as same as air, most fuels contain no ferromagnetic elements and the approximation of relative permeability of 1 is considered valid for this analysis.

Table 1 - Electro-magnetic properties of a micro-scale device simulation. Fuel filled gaps in the vicinity of the valve are treated as air in this model.

	Relative Permeability	Conductivity (S/m)
Si	1	0
Air	1	0
Cu	0.99991	5.7×10^7
NiFe	2000	5×10^6

A finite element grid size of 20 μ m has been used to give a grid scale compatible with the minimum device length scale. Also, an total grid computation volume (total model volume), which indicates the maximum volume of existing magnetic flux lines, is set as 1.5 times of actuator volume respectively for each of the simulations. Input current of 0.06A with 1000 turns of wire (NI = 60) is set as a default input parameter.

As results of a simulation, magnetic flux density (B), applied current density (J) and magnetic field intensity (H) and linear force exerted on armature in three directions are obtained (Figure 4). Force directions in an armature moving direction,

gap closing direction and out of plane direction are denoted as subscription of x, y and z.

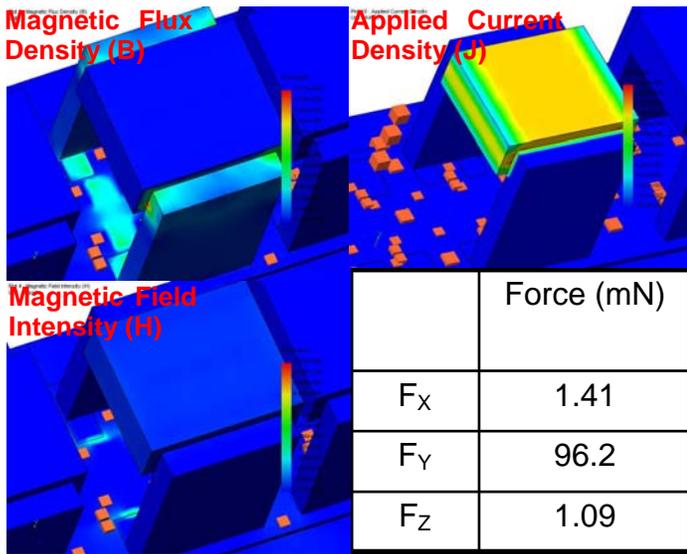


Figure 4 – The results of FEM simulation for prediction of force fields for the micro-scale electro-magnetic actuator

The results of the simulation indicated that the force in gap closing direction is significantly larger (70x) than the force exerted in the armature moving direction. With consideration of desired armature displacements (200 μ m for an armature moving direction and less than 15 μ m for a gap closing direction), the required stiffness of the guiding suspensions are determined as 7N/m for the armature moving direction (x-direction) and 6000N/m for the gap closing direction (y-direction). In addition, force acting on out of plane (z-direction) has forces on the same order of magnitude as force of the armature moving direction. This result implicates the design of suspension requires to be 860 times stiffer in the gap closing direction than the armature moving direction. This high ratio of stiffness in directions suggests suspension design of folded flexures.

4.2 - Suspension Analysis

The suspensions are designed to provide compliance along the direction of the armature moving motion, while restricting the gap closing motion to prevent the armature from touching with flux guides. In order to accommodate the high ratio of stiffness between a gap closing direction and the armature moving direction, four folded suspensions are implemented onto the armature (Figure 5). Even though the force acting on out of plane direction (z-direction) indicates considerable amount (same order of magnitude as the force in the valve moving direction), a stiffness in z-direction of the suspension beams is 44 times higher than the stiffness of the armature moving direction due to difference of moment of inertia (I) in x and z direction. The force acting on out of plane motion, therefore, has been neglected for this quasi-static analysis, but the force acting on out of place must be considered for high frequency operation.

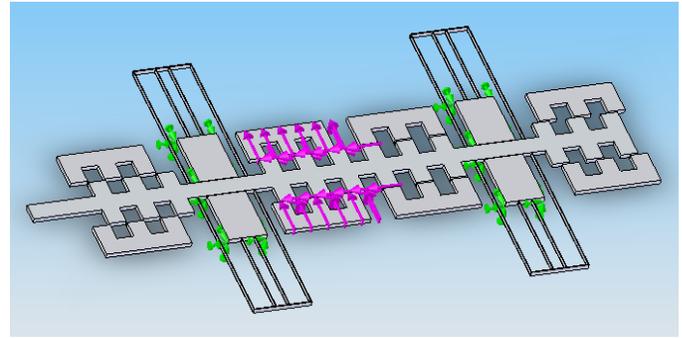


Figure 5 – Four Folded suspensions are adapted into the design to provide large deflection in the armature moving direction while providing high stiffness in the gap closing direction.

Linear elastic beam theory has been applied to estimate displacement of the armature, and stress distribution along the suspensions. Each suspension is decomposed into seven cantilever beams, and the dissected beams are evaluated for shear, bending moment, slope and displacement by the beam theory [6]. Cantilevers are superposed by solving a system of equations with boundary conditions. The superposition yields stiffness, the maximum stress, and deflection of the system. The validity of applying elastic beam theory is examined by comparing the maximum bending stress on the system with silicon yield stress.

The behavior of the suspensions is predicted by a MATLAB simulation, and dimensions of suspension, which satisfy desired stiffness in both directions, are estimated. The MATLAB code determines spring constants and bending stresses while varying lengths of two long beams (L2 and L3 on Figure 6). The thickness of the suspension is constrained by thickness of an armature (100 μ m) as determined by fabrication methodology, and the width of beam is an additional parameter to be optimized. The beam width is set by a realistic fabrication tolerance (aspect ratio) optimized for high process yield. These computed spring constants and stresses are compared with its given values of stiffness, which are determined from magneto-static force simulation, and with yield stress of Si. The code displays satisfied pairs of L2 and L3 as outputs. This suspension design includes a large anchor, located 10 μ m apart from an armature, to provide a limit stop in the gap closing direction (albeit limited by squeeze film damping by the fuel between the armature and anchor) as well as to serve as an armature guide to prevent in plane rotation. In addition, the large anchor prevents deformation of the armature due to high gap closing bending force.

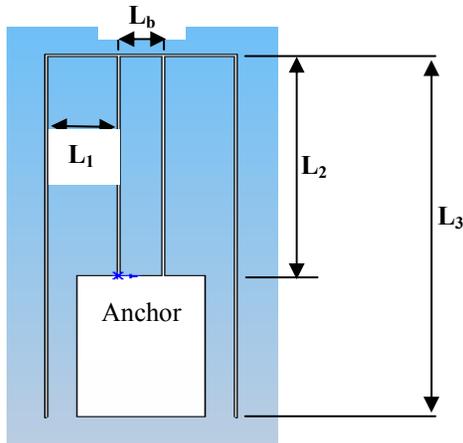


Figure 6 – The design of a folded suspension

FEM simulation is performed to determine the width of beams and optimum lengths of L2 and L3, as determined from the MATLAB simulation. Four folded flexure suspensions are included in this actuator system design (Figure 5) and the resulting stress and displacement profiles of the armature are determined by symmetry solutions for the four suspensions. The boundary condition of the suspension anchors is fixed and predicted linear forces from a magneto-static simulation are implemented as input forces. The end condition for each of the suspension members is guided, resulting in a fixed-guided beam solution. Outputs of this simulation are actuator displacement, and stresses acting on suspensions.

The width of the beams has been parametrically varied to determine how the armature displacement and stress of the suspensions are affected. With 15 μ m beam width, the simulation produces 190 μ m of armature displacement (Figure 7 -Left). In addition, the sixteen suspension beams of the 4 folded flexures suppress armature movement in a gap closing direction by allowing less than 10 μ m displacement, and prevents contact between an armature and flux guides without touching of the armature to suspension anchors. Overall the parametrically optimized dimensions of the suspension for this simulation are given in table 2. It is noted that the distance between sidewall of anchor and beam L3 is 210 μ m, and beam L3 will not have make contact with anchor when the armature has maximum displacement. Again, this is for a quasi-static analysis.

Table 2 - Calculated dimensions for a folded suspension

	L1	L2	L3	Lb	Width	Thickness
Length (μ m)	500	2000	3305	328	15	100

A stress analysis is subsequently performed on the optimized suspension design. The maximum stress within the structure for this design is determined to be 106MPa (Figure 7-Right), which is significantly below the silicon yield stress. The suspension design, therefore, prevents any brittle fracture due to applied stress.

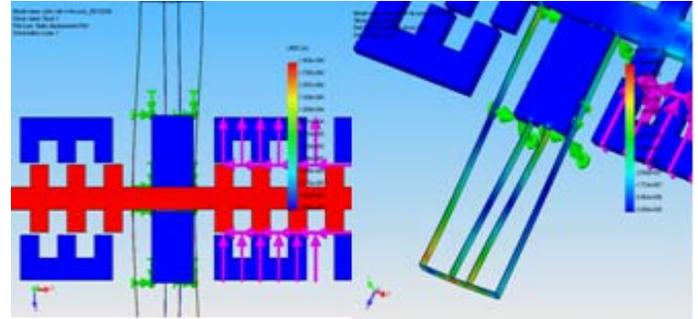


Figure 7 – Results of FEM simulations for suspensions (Left: Displacement, Right: Stress distribution)

5 - CONCLUSION

A development effort for a liquid fuel delivery system for small-scale engines is ongoing whereby a magneto-static actuator is being integrated with ‘hole-in-wall’ valve. This type of actuator appears to be the most appropriate for this application because the magneto-static actuator produces large force, and is most compatible with liquid fuels. A series of simulations have been performed to determine magneto-static forces using optimized designs as input parameters. An order of magnitude higher force has been estimated on the gap closing direction than the armature moving direction. As such, suspensions are designed to accommodate high stiffness ratio between an armature moving direction and a gap closing direction. Optimized suspensions create 190 μ m of the armature movement while having the maximum stress as 106MPa. Work is ongoing to implement these actuator designs into the planar fuel valve design. Fabrication in SOI is ongoing.

ACKNOWLEDGMENTS

Authors would like to acknowledge the Army Research Laboratory (ARL) for their support and Professor Dennis Lieu in U.C Berkeley for his helpful discussion regarding magnetic actuators.

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