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# A novel magnetic actuator design for active damping of machining tools

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ABSTRACT

Parts and cutting tools with large structural flexibility experience both forced and chatter vibrations during machining, resulting in poor surface finish or damage to the machine. This paper presents the design principles of a novel 3 degrees of freedom linear magnetic actuator which increases the damping and static stiffness of flexible structures during machining. The proposed actuator can deliver 248 N force in two radial ($x$, $y$) directions and 34 N $\times$ m (torque) in torsional ($\theta$) direction up to 850 Hz. The force and torque reduces to 107 N and 14.5 N $\times$ m at 2000 Hz, hence it can actively damp a wide range of structural modes. The magnetic force is linearized with respect to the input current using magnetic configuration design strategy. Loop shaping controllers are designed for active damping of boring bar vibrations. The static and dynamic stiffnesses of the boring bar were considerably increased with the designed actuator, leading to a significant increase in chatter-free material removal rates during cutting tests.



## 1. Introduction

Flexible parts and tools are often found in the boring of large cylinders and the turning of long and slender shafts. Because of the excessive flexibility of the tool or shaft, static deflections and forced and chatter vibrations limit productivity [1]. It is desired to increase both the static and dynamic stiffnesses of the structure so that higher material removal rates with better surface finish and tolerances can be achieved. The static stiffness is traditionally increased by adding stiffening elements to the setup, which may not always be possible as in the case of rotating shafts. The dynamic stiffness can be increased by improving the damping of the structure using passive or active devices. For example, Sandvik Coromant used carbide reinforcement and a built-in passive damper to improve the performance of a boring bar [2]. A variety of passive vibration absorbers have been proposed for chatter suppression in the literature [3,4]. Passive dampers are low cost and are easy to use, but the achievable damping is quite limited. Active damping systems can achieve higher damping, but they require actuators. In [5–7], piezo actuators have been installed inside boring bars to improve boring bar dynamics. Andrén et al. [14] installed a piezo actuator into a turning tool holder for vibration control . Pratt and Nayfeh [8] installed two Terfenol-D actuators outside the boring bar for active damping. Piezo actuators have been installed behind the outer rings of spindle bearings for active damping [15]. However, both piezo and Terfenol-D actuators have nonlinear hysteresis, which complicates the controller design. Unlike these actuators, electromagnetic actuators can have a large load capacity and almost no hysteresis. These advantages make it a better choice for active damping devices. Electromagnetic actuators have been implemented in milling and turning for active control of chatter [9,10]. However, past designs had a highly nonlinear relationship between the applied forces and input currents. Hence, the force output must be linearized before they can effectively be used.

In this paper, a novel noncontact linear magnetic actuator is developed to increase both the static and dynamic stiffnesses of long flexible parts and tools. The actuator has two radial and one rotational degrees of freedom, hence it can damp lateral and torsional vibrations simultaneously. Fiber optic sensors are installed into the actuator to measure the displacements of the armature. The basic concept of the actuator with a simple derivative controller was first presented by the authors in [17]. In this paper, full details of the actuator design principles are given with advanced loop shaping controllers designed for all three directions. The loop shaping controllers increase not only the dynamic stiffness, but also the static stiffness of the long flexible parts and tools. The actuator is experimentally tested on a boring

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bar application. The static and dynamic stiffnesses of the boring bar were increased significantly with the designed actuator, resulting in a higher chatter-free depth of cut.

## 2. Magnetic configuration and working principle of the actuator

### 2.1. Magnetic configuration of the actuator

The magnetic configuration of the designed actuator is shown in Fig. 1. It has four identical magnetic actuating units. Each magnetic actuating unit is comprised of two excitation coil windings, one permanent magnet, two stator side cores, one stator middle core, and an armature core. Both the armature and stator cores are made of soft magnetic material. The holes on the armature core are used for assembling. The force generated by each magnetic actuating unit is linearized with respect to current by the electromagnetic structure design strategy [11]. The designed magnetic actuator can be moved in $x$, $y$ and $\theta$ directions. The detailed working principle is explained in Section 2.2.

### 2.2. Working principle of the actuator

Fig. 2 shows the flux paths of the magnetic actuating unit. The air gaps on the lower and upper sides of the armature are denoted by $x_0+x$ and $x_0-x$, respectively, where $x_0$ is the air gap when the armature is centered, and $x$ is the armature displacement taken as positive for upward movement. The length of the permanent magnet is denoted by $L$; and $y_0$ is the air gap between the permanent magnet and the armature core. The permanent magnet plus the excitation coils generates a total flux $B_1$ on the upper side of the armature and a total flux $B_2$ on the lower side. The reference directions of both total fluxes are the same as those of the displacement $x$. These fluxes contain both biasing fluxes $\overline{B}_1$ and $\overline{B}_2$ generated by the permanent magnet, and excitation flux $\tilde{B}$ generated by the coil windings. If the actual current in the two coil windings is in the reference current direction as shown in Fig. 2, the biasing and excitation fluxes will add together on the armature top surface and subtract on the bottom surface. The normal force on the top surface will be larger than that on the bottom; and a net force in the $+x$ direction will be generated. When the excitation current reverses its direction, the net force direction will change accordingly. By properly driving the current directions for each magnetic actuating unit, actuating force/torque can be generated on the armature in $x$, $y$ and $\theta$ directions, as shown in Fig. 3.

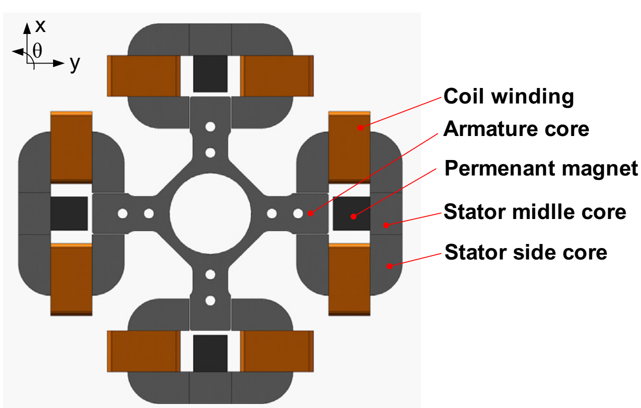


**Fig. 1.** Electromagnetic configuration of the designed actuator.

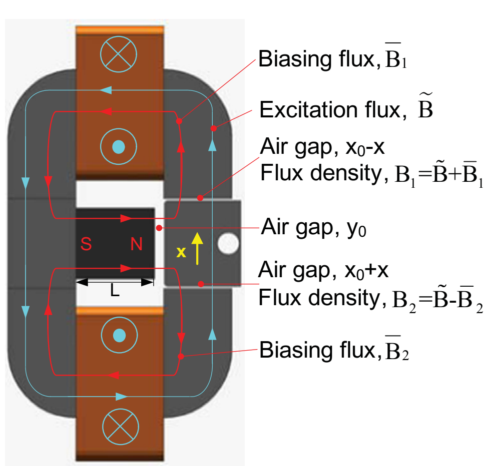


**Fig. 2.** Working principle of the magnetic actuating unit.

### 2.3. Lumped parameter force analysis

For this analysis, the permeability of the chosen soft magnetic material for the stators and armature is assumed to be infinite compared to the permeability of air. Thus, the main magnetic reluctance in the actuator is that of the air gap.

Fig. 4 shows a section of the magnetic actuating unit and the magnetic circuit model containing only the components related to the biasing flux generation. The reluctances of the system can be expressed as

$$R_1 = \frac{x_0 - x}{\mu_0 A} \tag{1}$$

$$R_2 = \frac{x_0 + x}{\mu_0 A} \tag{2}$$

$$R_a = \frac{y_0}{\mu_0 A_{pm}} \tag{3}$$

$$R_{pm} = \frac{L}{\mu_0 A_{pm}} \tag{4}$$

where $R_1$ and $R_2$ are the reluctances across the air gaps between armature and stator, $R_a$ is the reluctance of the air gap between armature and permanent magnet, $R_{pm}$ is the permanent magnet reluctance, $A$ is the stator pole area, $A_{pm}$ is the permanent magnet pole area, $\mu_0$is the vacuum permeability, and $\overline{\Phi}$ is the flux flowing into the armature.

The total flux generated by the permanent magnet is

$$\Phi_{pm} = B_r A_{pm} \tag{5}$$

where $B_r$ is the remanence of the permanent magnet. According to the magnetic circuit model

$$\frac{\overline{\Phi}}{\Phi_{pm}} = \frac{R_{pm}}{R_{pm} + R_a + \frac{R_1 R_2}{R_1 + R_2}} = \frac{L}{L + y_0 + \frac{A_{pm}}{2A} x_0 - \frac{A_{pm}}{2A} \frac{x^2}{x_0}} \tag{6}$$

Since the actuator is designed to damp the vibration of the armature and hold it at its zero position, $x$ will be much smaller than $x_0$. We can neglect the fourth part in the denominator of Eq. (6) which results in

$$\overline{\Phi} = \frac{L}{L + y_0 + \frac{A_{pm}}{2A} x_0} \Phi_{pm} \tag{7}$$

The fluxes need to satisfy the following equations from Gauss's law:

$$\begin{cases} \overline{B}_1 A + \overline{B}_2 A = \overline{\Phi} \\ \overline{B}_1 A R_1 = \overline{B}_2 A R_2 \end{cases} \tag{8}$$