A 10 kHz Short-Stroke Rotary Fast Tool Servo

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Introduction

This paper presents an overview of the design of our 10 kHz rotary fast tool servo (FTS) with an emphasis on the trade-offs made in this design. The heart of the design is a normal-stress variable reluctance actuator which allows the 1000 g tool tip acceleration required to follow a 5 micron PP sinusoidal trajectory at 10 kHz. At the time that this paper is written (August 2004) the design is complete, and the mechanical, magnetic, and electrical components for a 500 g version of the FTS are being fabricated.

The use of a fast tool servo with a diamond turning machine for producing non-axisymmetric or textured surfaces on a workpiece is well known. A rotary fast tool servo produces an in-and-out motion of the tool relative to a workpiece by swinging the tool along an arc having a fixed radius. Achieving this motion by applying a pure torque to the moving mass of the FTS leads to a reaction torque that is imparted to the machine supporting the FTS. In certain machining applications, for instance, producing a textured surface on a spherical workpiece, a reaction torque on the machine is preferable over the reaction force that a linear type FTS produces. The undesired lateral motion of a tool travelling along an arc can be made negligibly small by choosing an appropriate combination of tool travel and swing radius, and by setting the geometry of the tool arm so that the tangent plane on the workpiece containing the tool tip passes through the axis of rotation of the FTS.

Being able to control a fast tool servo at 10 kHz requires that the mechanical resonant modes be well above that frequency. Achieving a 1000 g tool tip acceleration requires a rotor and payload combination having a high torque-to-inertia ratio. A typical servo system usually has an actuator with its own moving mass connected to a payload through a coupling. In that case, the finite stiffness of the coupling results in an uncoupled resonant mode for the two lumped masses. We have merged the moving masses of our actuator and the payload into a single moving mass by attaching the tool arm directly to the rotating element of the actuator. By getting rid of the uncoupled mode, the first mechanical resonance is at a relatively higher frequency non-rigid body mode of the moving mass structure. Additionally, the merged mass approach results in a lower overall moving mass and a higher torque-to-inertia ratio. Placing the feedback sensor directly behind the tool and careful design of the support structure allows hiding certain flexible modes of the system to further improve high frequency performance.

Following the flexure design used in our earlier 2 kHz rotary fast tool servo [1], the 10 kHz rotary FTS uses an over-constrained flexure suspension for guiding the rotary motion and providing the stiffness needed for diamond turning a workpiece. The power amplifier and control system designed for the new actuator is also similar to the one used in our 2 kHz system [2], and will not be discussed further in this paper.

Normal-Stress Variable Reluctance Actuator

Our normal-stress variable reluctance actuator is designed to produce a magnetic force density an order of magnitude higher than the force density of a typical Lorentz-force actuator or shear-stress variable reluctance actuator. Figure 1 shows a model of the magnetic circuit for this actuator.

The rotor core is a rectangular prism suspended between two opposing C-shaped stator cores with nominal air gaps of 50 μm at the four pole faces. The rotor and stator cores are made of laminated soft magnetic material to reduce the hysteretic and eddy current losses associated with the AC flux. The DC coil provides a biasing magnetic flux in the four air gaps. Each stator core with its two AC coils forms a magnetic circuit with the rotor core. Considering one of the stator cores, energizing its AC coils in a manner that produces flux in a common direction creates a steering flux that circulates around that stator core and the rotor core via the two air gaps between the two. The bias and steering fluxes add at one of those gaps and subtract at the other. The same thing occurs between the other stator core and the rotor core. With proper phasing of the AC fluxes in the two stator cores and balanced magnitudes of DC and AC fluxes, the fluxes can be made to add in diagonally opposed air gaps and subtract in the opposite diagonally opposed air gaps, resulting in a net torque on the rotor with nominally no net force on it. One benefit of the DC magnetic bias is that it linearizes the torque versus current relationship for the actuator. Another benefit is that it reduces the reactive electrical power needed to drive AC flux in the air gaps by a factor of four compared to a non-biased magnetic circuit. The use of a DC magnetic bias can be found in other actuator designs, for instance, in certain electric engraver designs dating back to at least 1937.

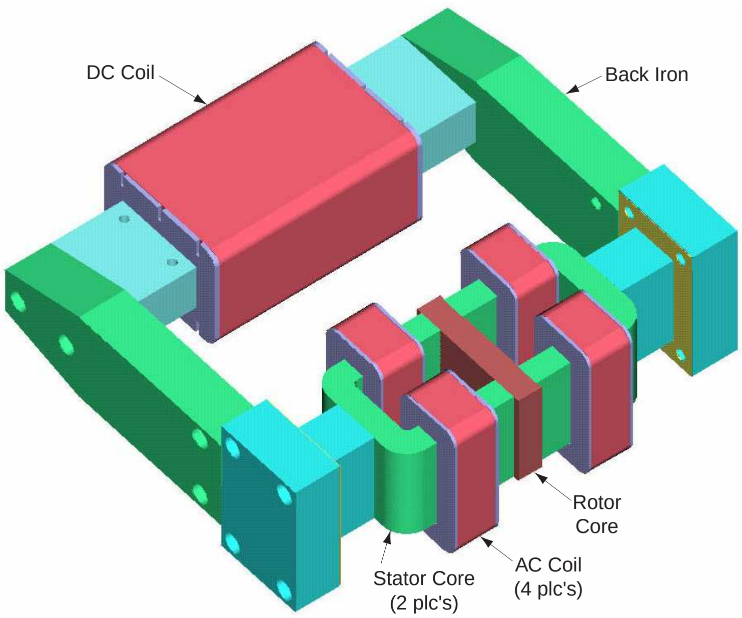

Figure 1: Model of the magnetic circuit for the normal-stress variable reluctance rotary actuator.

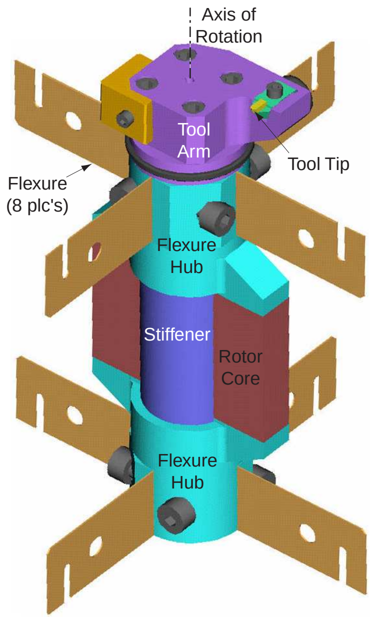

Figure 2: Model of the rotor with a tool arm and flexure suspension.

Figure 2 shows a model of the rotor with an attached tool arm and flexure suspension. The rotor core lamination layers are perpendicular to the axis of rotation. The rotor core is supported by stiffeners on its front and back, and by the flexure hubs at its top and bottom. The flexures are constrained at their inner ends by the flexure hubs and at the outer ends by a stationary support structure, and establish the axis of rotation for the rotor. Referring to Figure 3, the two stiffeners and the two flexure hubs encircle the rotor core. Making the stiffeners from a ceramic material provides high stiffness and avoids having a conducting loop that would produce eddy current power loss by linking the AC magnetic flux in the rotor core. The rotor core, stiffeners, and flexure hubs are joined together with an adhesive. The adhesive joints are designed so that the volumetric shrinkage of the adhesive upon curing causes the stiffeners to be preloaded in tension and the rotor core to be preloaded in compression. This is particularly important for the rotor core because tensile stress in the axial direction, and especially cleavage stresses, could lead to delamination of the core.

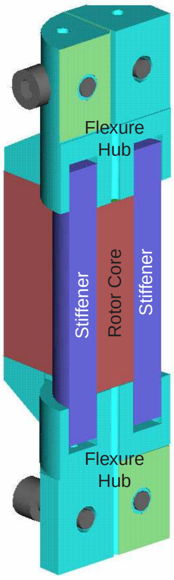

Figure 3: Model of a cross-section of the rotor.

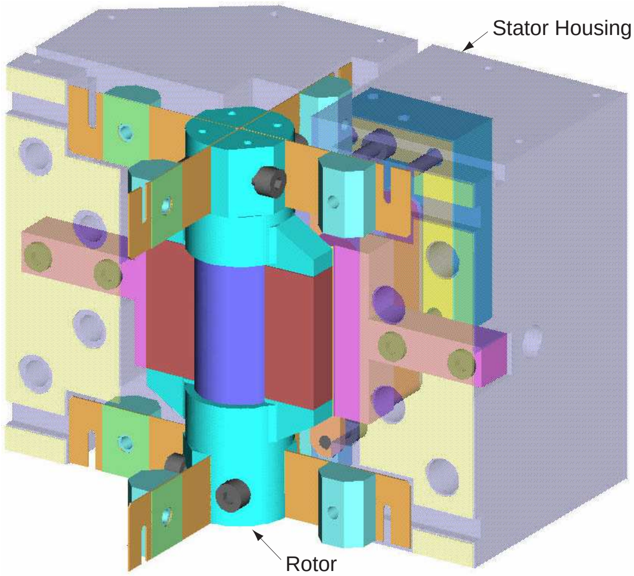

Figure 4: Model of the rotor mounted to one of two stator halves with flexures.

Overview of the Design

Figure 4 shows a model of the rotor mounted to one of the stator housings via the outer ends of four of the eight flexures. In a previous assembly step the stator pole faces are machined relative to the flexure mating surfaces on the stator housing to allow establishing the 50 μm air gaps between the stator and rotor cores. Shims are provided to accommodate any stack-up of manufacturing and assembly errors. The other stator half is added to what is shown in Figure 4 to complete the magnetic circuit and provide a stiff support structure for the rotor, which is essentially encased in a solid block of metal with openings provided for the stator cores, AC coils, flexures, and the flow of cooling fluid.

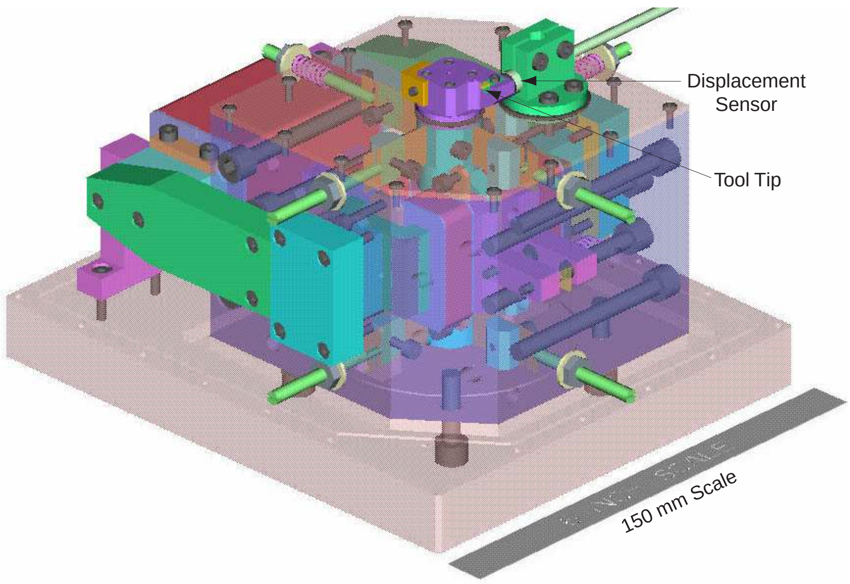

Figure 5: Model of the 10 kHz rotary fast tool servo.

Figure 5 shows a built-up model of the 10 kHz rotary fast tool servo. A capacitance gauge measures the displacement of the tool arm at a location just behind the tool. The FTS is designed for a maximum stroke of 10 μm peak-to-peak (PP) at low frequencies and was initially designed for a 5 μm PP stroke at 10 kHz (1000 g). The maximum stroke was made smaller than the 50 μm peak-to-peak (PP) stroke of our 2 kHz rotary FTS to allow tailoring the flexure suspension stiffness and fatigue stress level for 10 kHz operation. The designed stroke at 10 kHz was reduced to 2.5 μm PP (500 g) during a trade-off between having a more mechanically robust system versus the more fragile light-weighted rotor needed for 1000 g. We felt it more important to have a proof of principle that was more likely to survive anticipated tests for this first prototype, and expect to adjust design parameters in future versions to achieve 1000 g and beyond.

Referring back to Figures 3 and 4, the dimensions of the rotor core are determined by a trade-off analysis driven by closed-form equations relating those dimensions and the matching dimensions of the stiffeners and flexure hubs to the power losses in the rotor and stator cores, reactive electrical power required to drive the AC flux in the air gaps, current density and cooling requirements for the AC coils, rigid body and non-rigid body first modes of the rotor, and tool acceleration for a 1.5 Tesla peak flux density. Finite element analysis is used to determine the frequency of the first non-rigid body modes of a selection of rotor geometries to provide a bounding analysis for the design space, and agreed with the closed-form equations. Also folded into the trade-off was the attractiveness of using the same design for the flexure mounting hardware that proved successful in our earlier 2 kHz rotary FTS.

For the 500 g version of the FTS that we are currently building, the first non-rigid body mode of the rotor is a torsional mode at 22 kHz that drops to 18 kHz when the tool arm is added. This makes our planned operation at 10 kHz challenging, but with provisions in the design for adding squeeze-film damping and in the spirit of keeping the control system challenging enough for this PhD research work we decided it was acceptable.

Summary

This paper presents an overview of the design of our 10 kHz rotary fast tool servo (FTS) as well as some of the trade-offs made in that design. The novel normal-stress variable reluctance actuator is presented in detail to provide insight into the highly integrated electromagnetic and mechanical designs needed for achieving 1000 g tool tip acceleration and to operate at frequencies up to 10 kHz.

As this work progresses, we will assemble the mechanical, magnetic, and electrical hardware; develop and integrate the control system; characterize the FTS performance with bench tests; and integrate the FTS with a diamond turning machine to perform cutting tests on a workpiece.

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References

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