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Modeling and Experimental Verification of a Tubular Actuator for 20 g Acceleration in a Pick and Place Application

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Abstract—This paper presents the modeling and the experimental verification of a tubular actuator for a pick and place application. To increase the throughput of a placement robot for printed circuit boards, very fast linear motion is required. A moving magnet tubular actuator with axially magnetized magnets is selected. Using a semi-analytical magnetic field description coupled to thermal models, a design is created that potentially could achieve a translator acceleration of 20 g. A prototype of the designed actuator is built and coupled with a Simulink dSpace system to perform extensive measurements to validate the models and investigate the achievable acceleration within a pre-determined motion profile. The electro-motive force is measured, and the disturbance forces are identified. The position error is measured during the motion profile with an acceleration of 20 g and a stroke of 30 mm. Furthermore, thermal measurements are performed to check the achievable duty cycle. The built design shows good agreement with the models, and the specified acceleration of 20 g is achieved.

I. INTRODUCTION

In robotic applications there is an increasing demand for fast actuation with high precision and bandwidth capabilities. One particular application is the pick and place (P&P) robot that places surface mounted devices on printed circuit boards, which are picked from a feeder. The complete P&P action requires a four degrees of freedom (4-DoF) robotic motion, however, this paper focusses on the linear motion in the vertical direction to pick and place the components. To increase the throughput of the total P&P cycle, currently maximum 8000 components per hour, a high acceleration level is necessary. The aim of this paper is to design an actuator with a very high translator acceleration of 200 ms^{-2} on a stroke of 30 mm for a duty cycle of 30%. As can be seen in the figure, the motion profile consists of an acceleration phase, resulting in a velocity of 1.5 ms^{-1} , followed by a deceleration phase to be able to standstill at $x = 30 \text{ mm}$.

To achieve this motion profile, a three phase slotless tubular permanent magnet actuator (TPMA) is selected as shown in Fig. 1, since it has a high force density and very good servo characteristics [1]. The tubular actuator consists of a stator and a translator, where a moving magnet concept is preferred, since it does not require winding connections to the moving part. Further, the stationary part contains coils and is chosen to be

Fig. 1. A moving magnet slotless tubular permanent magnet actuator.

slotless as the actuator is designed for a precision positioning application, which requires a smooth force characteristic. An additional advantage is the manufacturability of a slotless structure especially in low volume actuators since the stator consists solely of a soft-magnetic tube.

II. MODELING

To model the actuator, a multi-physical model is created where magnetic and thermal behavior is coupled resulting in a very fast and accurate analysis and design tool. The magnetic properties are described using a semi-analytical description of the magnetic fields.

A. Magnetic model

Several papers have been written on the subject of designing tubular actuators using semi-analytical field equations. In [2]–[4] semi-analytical solutions for the magnetic fields in different tubular actuator topologies are presented, and [5] compares the force density of three different topologies. Although these papers are very extensive, the conclusions from [5] cannot be used in this research as the force is maximized instead of the acceleration. These two quantities are strongly connected but show different optima. Therefore, the semi-analytical model described in [2]–[4] is used to design an actuator for high translator acceleration.

B. Electrical model

The electrical model of the actuator is defined by the geometric parameters of the windings and the electrical properties of these windings. Slotless actuators exhibit no cogging force due to stator slotting, and the flux density in the coil in contains a negligible amount of higher harmonics. Therefore, there is no need for fractional coil pitch to pole pitch ratio to reduce EMF harmonics. To obtain a high winding factor, full pitch BLAC windings with a coil pitch equal to the pole pitch are implemented. As a current amplifier with an output filter is used to drive the actuator, there is no need for an inductance to filter higher harmonics in the current. Therefore, the inductance value is not taken into account in the model.

While the magnetic fields are primarily limited by material properties, the electrical properties of the actuator are mainly restricted by thermal constraints. Naturally, the current density in the coil region defines the heat produced in the windings. Due to this heat, the temperature difference between the armature and the ambient rises to

$$\Delta T = J^2 l_{coil} \rho_{Cu} \left(R_s - \frac{l_{coil}}{2} \right) \frac{1}{k_p R_{out} h}, \quad (1)$$

where J is the current density in the coil, k_p is the packing factor of the coil region, and h is the heat transfer coefficient. In this expression, the coil and armature are assumed to be perfect heat conductors which is a good approximation [6]. The heat transfer coefficient can have values of approximately $20 \text{ Wm}^{-2}\text{K}^{-1}$ for natural cooling, up to $70 \text{ Wm}^{-2}\text{K}^{-1}$ when forced air cooling is used [7].

C. Force and acceleration

As the considered actuator has no slots, the thrust force can be calculated by applying the Lorentz force equation (2) in the coil region

$$\vec{F}_z = \int_{V_{coil}} (\vec{J} \times \vec{B}) dV, \quad (2)$$

The translator acceleration is defined by the produced force divided by the total translator mass which depends on the stroke and the active length.

III. DESIGN

The models presented in the previous sections are coupled to create a multi-physical design tool. Using this tool, topologies with different magnetization patterns are investigated viz.

- 1) radial magnetized topology
- 2) quasi-Halbach magnetized topology with soft-magnetic core
- 3) quasi-Halbach magnetized topology with non-magnetic core
- 4) axial magnetized topology

The analysis showed that the quasi-Halbach topology is favorable when a small translator radius is used. Albeit that from manufacturing point of view, the axially magnetized topology is preferable because this topology contains less permanent magnet material, and all magnets are magnetized in the

Fig. 2. Third order motion profile with a peak acceleration of 200 ms^{-2} , a peak velocity of 1.5 ms^{-1} and a stroke of 30 mm.

Fig. 3. Different regions in an axially magnetized slotless tubular actuator.

(relatively easy) axial direction. The difference in performance of the two topologies is a few percent and therefore, the axially magnetized topology is chosen, which is shown in Fig. 3.

A new parametric search is performed where the geometric parameters are varied and the current density is updated to maintain a constant $\Delta T = 40^\circ\text{C}$ using (1). The acceleration and force density levels are now mainly limited by the achievable heat transfer coefficient and the temperature constraint. The heat transfer coefficient is fixed to $25 \text{ Wm}^{-2}\text{K}^{-1}$, as the actuator is fixed to a robot arm and moves through the air. Hence, the expected convection is higher than the natural convection due to an increased air flow. The mean copper losses (which determine the maximum current density) in the coils are calculated for the motion profile shown in Fig. 2 which has a duty cycle of 30 %.

As the components in the application are attached to the translator using vacuum, a hollow core is required, therefore $R_r = 2.0 \text{ mm}$ and $R_{in} = 1.0 \text{ mm}$. The translator core as shown in Fig. 1 is modeled as aluminum.

The geometric parameters of the final design are given in Table I and the material properties are given in Table II.

TABLE I
GEOMETRIC PARAMETERS PROTOTYPE.

Geometric parameter		Description
R_{out} (mm)	9.25	Outer radius
l_{stator} (mm)	1.00	Radial stator length
l_{coil} (mm)	2.00	Radial coil length
l_g (mm)	0.25	Radial airgap length
l_m (mm)	4.00	Radial magnet length
τ_p (mm)	8.33	Pole pitch
α_p	0.75	pole pitch to magnet pitch ratio
l_{active} (mm)	100.0	Axial active length
l_{ax} (mm)	100.0	Axial armature length
$m_{translator}$ (g)	151.0	Translator mass

TABLE II
MATERIAL PROPERTIES PROTOTYPE.

Property	Value	Description
Magnet	BM35H	Permanent magnet material
B_{rem} (T)	1.175	Remanent flux density permanent magnet
μ_r	1.08	Relative permeability permanent magnet
Steel	AH1010	Steel type
ρ_{Cu}	$1.7e^{-8}$	resistivity of copper
k_p	0.6	Coil packing factor

IV. EXPERIMENTAL VERIFICATION

A. Setup

Of the design presented in the previous section a prototype is built as shown in Fig. 4. At the left side of the picture, the optical position encoder (Renishaw RGH22) is visible. The actuator contains 36 coils which are all separately soldered to a 36 pins connector on the connection board. This provides the possibility to create different winding and phase configurations using a second printed circuit board. For the results presented in this paper, the coils of the three phases are connected in series while all three phases are separately driven by a current controller, hence, not wye or delta connected. The coils and armature are encased by an aluminum tube to enable the fixation of the bearings. Two nylon sliding bearings are used at both end of the armature. As can be seen, a slit is milled in the aluminum tube and the armature to provide a feed through for the windings of the coils to the connection board. The width of the slit in the armature is 1.0 mm, which is less than 2 %

Fig. 4. Prototype of the designed slotless permanent magnet tubular actuator.

Fig. 5. Radial flux density near the slit at $r = 8.20$ mm.

of the total armature material.

A 3D finite element analysis is performed to investigate the influence of the slit on the magnetic loading. To simplify the model, only one pole pitch is modeled and symmetry in the axial direction is used. In Fig. 5, the flux density in the radial direction is shown near the armature at $r = 8.20$ mm for an angle of 25 degrees to both sides with respect to the center of the slit. As can be seen, close to the edges of the slit the curve shows a peak while in the middle of the slit the flux density is decreased. The flux density in the center of the coil is integrated over the whole circumference is compared to solution obtained for a model without the slit. The difference in flux density in both models is 0.11 % in the center of the airgap and 0.15 % close to the armature. Consequently, the slit results in a negligible decrease of the magnetic loading. An advantage of the slit is the cut in the path of the eddy currents created by the movement of the translator and the armature reaction. To verify the analysis tool and to check the design specifications, several static and dynamic tests are performed.

B. Electro-motive force

Two magnets are characterized to obtain the correct remanent flux density and relative permeability. Measurements show that the remanent flux density is slightly lower than the value used in the modeling, i.e. 1.11 T instead of 1.15 T. The models are updated with the measured remanent flux density to be able to make a good comparison.

The electro-motive force (EMF) is measured by moving the translator with a certain speed through the actuator. In Fig. 6, the EMF of the three phases is shown, where the solid line shows the modeled EMF and the marks represent the measured values. As can be seen, the measurements are not exactly in phase with the model which can be caused by several effects. For example, the magnets might be not identically magnetized, the pole pitch varies, or the coil pitch varies. Although the results show good agreement with the model, the small deviation directly affects the performance by

Fig. 6. Electro motive force, both the measured and the modeled waveform.

Fig. 7. Cogging force due to finite translator length for a translator length of 100.0 mm

means of a smaller motor constant and a possible position error.

The figure shows that additional to the first harmonic, the third harmonic is present in the EMF. Because each single phase is connected to the amplifier and not wye connected, the third harmonic is not canceled, hence, the commutated current will not be completely sinusoidal.

C. Cogging force

Although the actuator is slotless and consequently has no slot cogging, the finite stator length results in a cogging force component which can be significant. Due to the abrupt change in permeance of the magnetic path at the stator back-iron ends, attraction forces in the axial direction occur. Varying the axial length of the back-iron is one of the methods that can be used to minimize this force [8]. The prototype back-iron has an axial length of a multiple of the pole pitch, which results in an end-effect cogging force up to 15 % of the rated force.

Fig. 8. Cogging force due to finite translator length for a translator length of 104.2 mm

Fig. 7 shows the measured end-effect compared to the results from models.

Two methods are used to measure the end-effect force. First, a load-cell is placed between the translator and the fixed world. Using a setscrew, the translator is moved to the left and to the right. The obtained waveform contains information about the end-effect force and the friction. The offset between moving left or right is equal to twice the friction. A disadvantage of this method is that the load-cell has to be aligned very accurately for a reliable measurement and additional equipment is required. The second method is based on a measurement of the current required to compensate the end-effect forces. A controller is implemented with a high integrator gain resulting in a small position error. A trajectory with a constant velocity results in constant friction, hence, the resulting current contains a constant friction component and the end-effect cogging. Note that here the assumption is made that the friction is not position dependent. The two methods show good agreement, and the latter one is used in Fig. 7. As can be seen, the waveform of the measurement and the simulation model are in good agreement.

Initially, the cogging force is measured and compensated by using feed-forward control at the cost of additional losses due to the increased current. Therefore, the back-iron is adapted to decrease this force component. As mentioned before, the end-effect cogging force is a function of the axial length of the translator, hence can be minimized by choosing a proper length. An additional model is created to find the optimal length. A soft-magnetic ring is manufactured with the correct axial length and fixed inside the bearing directly onto the back-iron. Although the ring and the back-iron are not in contact over the whole circumference, from magnetic point of view the space between the two is negligible as the effect airgap between the back-iron and the magnets is significantly larger.

A new end-effect measurement is undertaken to validate

Fig. 9. Position error during third order motion profile, as shown in Fig. 2.

the improvement. The results in Fig. 8 show that the peak values and the waveform show similar behavior. However, there is a significant error between the measurements and the model. Several changes in the model are made to account for mechanical inaccuracies to identify the difference between the measurements and the model but no definite agreement was found. Although the relative error between the modeled and measured value in Fig. 8 is significant, the absolute error between the model and the measurements of Fig. 7 and Fig. 8 are of the same order. Hence, the error is probably caused by other effects.

One expects that the end effect force caused by the permanent magnets of the translator is periodic with the pole pitch, however, the results in Fig. 7 and Fig. 8 do not show this periodicity. The larger period of the force is caused by the finite length of the translator, i.e., the end-effects in this actuator are caused by both the finite armature and the finite translator length. The total length of the translator is 143.7 mm and the armature is 100.0 mm while the stroke is 30.0 mm

D. Performance

To obtain the actuator performance, a controller is implemented in a Simulink dSpace environment and coupled to a three phase current amplifier with a 3 dB bandwidth of 6 kHz. A linear optical encoder with an accuracy of $1\mu\text{m}$ is used to obtain the position. A third order trajectory for a stroke of 30 mm and a peak acceleration of 20 g is generated as shown in Fig. 2. Using a feed-forward and PD controller, the position error, as shown in Fig. 9, is obtained. The structure of the controller is depicted in Fig. 10. The feed-forward controller contains end-effect compensation, acceleration-, friction- and gravity compensation, and the PD controller has a sensitivity bandwidth of 250 Hz. As can be seen, the error during the trajectory is smaller than $15\mu\text{m}$ while the final error at the end of the trajectory is only $1\mu\text{m}$.

During the trajectory, the current setpoints for the amplifier are obtained as shown in Fig. 11. The commutation uses current on the q-axis to drive the actuator. This current setpoint is depicted by the dashed line in Fig. 11, while the resulting phase currents are shown by the line with the markers.

The profile of the acceleration in Fig. 2 is directly visible by i_q in Fig. 11. The jerk is clearly recognizable as the slope of the current in the initial phase. The current increases up to almost 3 A followed by an oscillation of the current originated by the end effects of the actuator. Between $t = 10\text{ ms}$ and $t = 20\text{ ms}$, the current is oscillating around a value of

Fig. 10. Controller structure used to drive the actuator where d is the viscous friction, k the Coulomb friction and m is the translator mass.

Fig. 11. Three phase currents during motion profile shown in Fig. 2.

approximately 0.5 A. As in this phase the acceleration of the translator is zero, i.e., the translator moves at constant speed, the current is required to overcome friction and gravity as the movement is upwards. The whole current waveform is symmetric around 0.5 A and hence in the deceleration phase less current is required to have zero velocity at $t = 30\text{ ms}$. Due to the friction and the gravity, the deceleration phase of an upwards movement requires less power.

E. Thermal analysis

From the current waveforms shown in Fig. 11 and the measured resistance of the windings, the total power dissipation is calculated. The resistance is measured at $20\text{ }^\circ\text{C}$, however, the actuator is designed for an operating temperature of $\Delta T = 40\text{ }^\circ\text{C}$. Therefore, the measured resistance is compensated using

$$R_{60} = R_{20}(1 + \alpha\Delta T), \quad (3)$$

where α is the temperature coefficient of the resistance, for copper equal to $\alpha = 0.004\text{ K}^{-1}$. Using this value for the resistance, the mean power during one stroke is 25.2 W. As the duty cycle is 30 %, the mean power during operation is 7.5 W.

Fig. 12. Photo of the setup with three thermocouples for thermal measurements.

The actuator is assembled to an aluminum frame to prevent vibrations. As aluminum is a very good heat conductor, this frame will act as a heat sink for the actuator. Therefore, the actuator is separated from the frame to enable static thermal measurements. As shown in Fig. 12, three thermocouples are placed at the actuator, two between the coils (T1 and T2) and one on the outer surface of the actuator (T3). A current is applied to the windings and the voltage over the windings is measured. During the measurement, the value of the current is varied to dissipate a constant power of 7.5 W in the coils. The measured temperature of the three thermocouples is shown in Fig. 13, as can be seen, the temperature difference between the coils and the outer surface is approximately 2 °C. The ΔT , temperature difference between the environment and the actuator is approximately 27 °C. This value is lower than the value used in the design procedure which is caused by the increased outer surface of the actuator due to the aluminum tube around the actuator. The surface of the actuator is 6.89 cm², while the effective surface of this prototype is 11.50 cm². Using

$$\Delta T = \frac{P_{Cu}}{Sh} \quad (4)$$

where S is the outer surface of the actuator, the heat transfer coefficient, h , of the lab environment is found to be approximately 25 Wm⁻²K⁻¹ without active cooling. This is higher than considered during the design in Section III and is partly caused by the effective cooling surface. The model considers only the armature as surface for the heat transfer while the translator will also transfer some heat to the environment. If the aluminum tube is removed from the setup and the same heat transfer coefficient is taken into account, the value of ΔT is approximately 44 °C which is 10 % higher than the modeled value. This is mainly caused by the decreased remanent flux density of the magnet which is not taken into account in the model. The friction in the actuator is also slightly higher than modeled resulting in higher copper losses.

V. CONCLUSIONS

A tubular actuator has been designed using analytical models and built to validate. The actuator has been optimized to

Fig. 13. Temperature measured by the three thermocouples shown in Fig. 12, while dissipating 7.5 W in the coils.

obtain the highest translator acceleration. Several static and dynamic measurements have been conducted and presented in this paper. The results show good agreement with the models used for the design. To minimize the disturbance forces, the armature is adapted. The required acceleration has been achieved within a dynamic position accuracy of 15 μm. Thermal measurements are performed to check the feasibility of the translator acceleration with a duty cycle of 30 %. Due to a lower realized remanent flux density of the magnets and an increased friction force, the temperature of the actuator is 10 % higher, i.e. 44 °C instead of 40 °C, than expected from the models.

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