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Simple and robust feedforward compensation of quadrant glitches using a compliant joint

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ABSTRACT

Quadrant glitches commonly occur when rolling/sliding bearings are used in precision motion stages. Compensating them accurately using feedforward control is challenging due to the need for complex friction models, and the variability of on-machine friction dynamics. This paper shows that quadrant glitches can be accurately and robustly compensated with simple friction models by attaching the bearing to the moving table using a joint that is very compliant in the motion direction. Design of a prototype joint that achieves high compliance by combining a positive-stiffness flexure with negative-stiffness permanent magnets is presented. Large reductions in quadrant glitches are demonstrated in experiments.

1. Introduction

Mechanical bearings (i.e., sliding and, especially, rolling bearings) are widely used in precision motion stages because of their low cost, large motion range, high off-axis stiffness and ruggedness [1]. They are also finding increasing use in ultra-precision motion stages, for instance, as vacuum- and cleanroom-compatible substitutes for aerostatic and hydrostatic bearings, respectively [2]. However, the use of mechanical bearings usually results in quadrant glitches, which are position errors that occur as the stage tries to overcome pre-motion (i.e., pre-rolling/pre-sliding) friction during motion reversals [3]. Quadrant glitches can only be partially suppressed by feedback control, because a feedback controller must wait for errors to develop before it takes corrective actions. Therefore, feedforward compensation is most often used to suppress quadrant glitches beyond what is achievable using feedback control [4–9].

Feedforward compensation of quadrant glitches is achieved by predicting and preemptively canceling out the frictional forces experienced by a stage, using a model of pre-motion friction. To be effective, the friction model must be sufficiently accurate; and to be practical, the model should have a minimal number of parameters that can be identified easily without need for frequent re-calibration. The problem is that simple models of pre-motion friction often do not have sufficient accuracy, so complex models are needed. Moreover, both simple and complex models of pre-motion friction are not robust to changes in on-machine friction dynamics that frequently occur as a function of time and/or stage position [4,10]. Thus, they need frequent re-calibration or adaptation to maintain their effectiveness [10].

This paper shows that quadrant glitches can be accurately and robustly compensated using simple models of pre-motion friction by attaching the bearing to the moving table of the stage using a compliant joint. Section 2 presents the principle behind the proposed compliant joint solution, and the design of a prototype compliant joint for a single-axis ultra-precision motion stage. Section 3 experimentally demonstrates the effectiveness, simplicity and robustness of feedforward compensation of quadrant glitches using the designed compliant joint prototype, followed by conclusions and future work.

2. Compliant joint concept and prototype design

2.1. Concept of compliant joint

Fig. 1(a) illustrates a typical motion stage equipped with a mechanical bearing that is rigidly attached to the moving table. Before gross motion (i.e., full sliding or rolling) begins, the stage experiences pre-motion friction due to the bearing. Pre-motion friction can be modeled as an equivalent spring of stiffness k_f connecting the table to ground (as shown in Fig. 1(c)) [4]. Initially, k_f is very large, but as more driving force is applied to counteract friction, k_f rapidly reduces and eventually becomes zero, thus allowing gross motion of the stage. Quadrant glitches are servo errors developed as a motion stage tries to overcome k_f when starting from rest (e.g., after motion reversals). Pre-motion friction is a highly nonlinear phenomenon which is difficult to model accurately. Thus, when it is characterized using simple models, significant modeling errors are introduced into k_f (relative to the actual frictional stiffness of the stage). Modeling errors can be reduced by employing more accurate models of pre-motion friction, but such models are often complex, with lots of parameters that must be identified. Moreover, pre-motion friction changes with time and/or stage position; so even with complex models of friction, errors are introduced into k_f as the actual frictional stiffness varies.

Fig. 1(b) shows the proposed compliant joint concept for addressing the errors in k_f due to inaccurate modeling or variation

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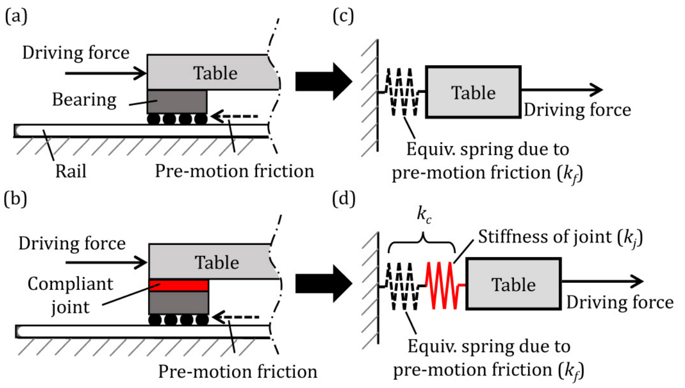

Fig. 1. Schematic of motion stage with mechanical bearing (a) rigidly attached to table (typical), and (b) attached to table using compliant joint (proposed); (c) and (d) are spring models of (a) and (b), respectively.

of friction. Rather than being rigidly attached to the moving table, the bearing is attached using a joint of stiffness k_j in the motion direction. Accordingly, the stage with the compliant joint can be modeled as a series combination of k_f and k_j, as shown in Fig. 1(d), with combined stiffness $k_c = k_f k_j/(k_f + k_j)$. The sensitivity of k_c to errors in k_f is therefore given by

$$\frac{\partial k_c}{\partial k_f} = \left(\frac{\eta}{1+\eta}\right)^2; \quad \text{where } \eta = \frac{k_j}{k_f} \tag{1}$$

Notice that if $k_j \ll k_f$, $\eta \to 0$ and the sensitivity of k_c to errors in k_f becomes very small. In other words, a very small k_j dominates the combined stiffness felt by the stage at the very beginnings of motion (reversals) when k_f is very high, and quadrant glitches occur. Therefore, if $k_j \ll k_f$, and k_j is precisely known, accurate compensation of quadrant glitches can be achieved even when a significant amount of error exists in k_f.

2.2. Design of compliant joint prototype

In this section, the design of a compliant joint prototype for the single-axis ultra-precision motion stage [2] shown in Fig. 2 is presented (as one possible realization of the proposed concept). The stage has 1.35 kg moving mass and 40 mm travel range. It is guided by a pair of high-rigidity preloaded linear ball bearings (THK, SR-15SB), riding on a super-precision grade rail. An air core linear motor (Aerotech, BLMUC-95) with 162 N peak and 23 N continuous force limits, powered by a linear amplifier (Trust Automation, TA-310), is employed to drive the stage. The table position is measured using a linear encoder system (Renishaw, T1000 read head and RGSZ20 scale) with post interpolation resolution of 4.88 nm. The designed compliant joint prototype, whose design is discussed in the rest of this section, is used to attach each bearing to the table. Based on the requirement that $k_j \ll k_f$, the compliant joint must

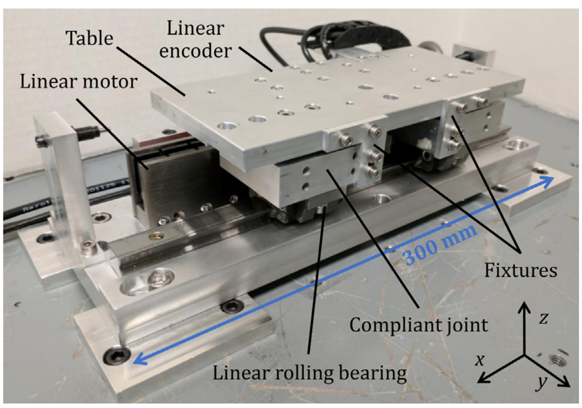

Fig. 2. Single-axis ultra-precision motion stage equipped with the designed compliant joints. (Note: the linear encoder is mounted on the distal side of the table and the fixtures are used to deactivate the compliant joint).

have orders of magnitude less stiffness than the initial stiffness of the pre-motion friction experienced by each bearing. However, the joint must also maintain the same order of magnitude of off-axis stiffness as the bearing, so as not to unduly compromise the rigidity of the stage.

Flexure hinges (or flexures, for short) are commonly used to design compliant joints because of their non-contact and friction-free nature; therefore, they are adopted for the designed prototype, as shown in Fig. 3(a). The outer platform of the flexure is connected to the bearing and the center platform is connected to the table. The stiffness of flexures is coupled in all directions. Therefore, it is challenging to get very low stiffness in the axial (x) direction without overly compromising off-axis stiffness (in the y and z directions). A double parallelogram configuration with an intermediate platform is adopted to connect the center and outer platforms of the flexure to help reduce its axial stiffness. Moreover, the positive stiffness of the flexure is combined in parallel with a negative stiffness mechanism to keep the net axial stiffness positive but smaller than that of the flexure alone. As highlighted in Fig. 3(b), the negative stiffness mechanism is realized using a pair of repelling permanent magnets (PMs) attached to the center and intermediate platforms of the flexure. Given relative motion δ from equilibrium, the PM pair is repelled further apart from each other. Note that eight N42 grade NdFeB PM blocks (each of dimension 1/2″ × 1/8″ × 1/16″) are used to construct opposing halves of the PM pair. As shown by north-pole-pointing arrows in Fig. 3(b), the blocks are arranged with alternating polarity to increase the repulsion force (hence the negative stiffness) of the PM pair [11]. Fig. 3(c) shows the manufactured compliant joint prototype. The flexures are manufactured from Aluminum 7075-T6 using wire EDM.

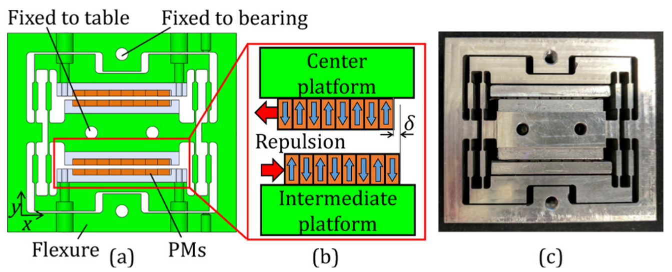

Fig. 3. (a) CAD schematic of compliant joint (section view), (b) negative stiffness mechanism with repelling permanent magnets, and (c) manufactured compliant joint prototype.

Table 1 summarizes the stiffness values of the designed flexure and PMs of Fig. 3 (obtained via finite element analysis using SolidWorks® and COMSOL®) together with the total stiffness of the compliant joint. Notice that the addition of the PMs reduces the axial stiffness of the compliant joint by more than 50% compared to the flexure alone, with virtually no change to its vertical and lateral stiffness. Table 1 also provides the stiffness values of each bearing. The vertical and lateral stiffness values are obtained from the bearing catalog; the reported axial stiffness represents an order-of-magnitude estimation of the bearing's initial frictional stiffness, based on experiments (as explained in Section 3). Notice that the axial stiffness of the compliant joint is two orders of magnitude less than, while the vertical and lateral stiffness values are of the same order of magnitude as that of the bearing. The combined stiffness of the bearing and compliant joint are computed as shown in the table. The combined axial stiffness is virtually the same as that of

Table 1
Designed stiffness values of compliant joint, bearing and their combination [N/m].

	Axial (x)	Lateral (y)	Vertical (z)
Flexure	9×10^4	3×10^7	3×10^7
PMs	-5×10^4	2×10^5	-2×10^4
Compliant joint (PMs & flexure)	4×10^4	3×10^7	3×10^7
Bearing	$\sim 10^6$	2×10^7	6×10^7
Combined (joint & bearing)	$\sim 4 \times 10^4$	1×10^7	2×10^7

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