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Mitigating friction induced limit cycles by an intermediate flexure stage

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Abstract

Friction-induced limit cycling, termed hunting, bounds the positioning performance of precision systems. A friction isolator mitigates this issue through an intentionally passive compliance between the friction-inducing bearing and the actuator. This compliance omits the sudden change in friction force felt by the controller close to standstill. Many design parameters influence the performance of such a friction isolator, including its compliance and damping, system mass, travelled path, and control parameters. Currently, a general design guideline for these friction isolators is missing. This research presents a simulation environment with a metric for identifying hunting cycles, from which a design guideline is distilled. The simulations demonstrated, and experiments confirmed, that a limited controller bandwidth and a significant gap between static and dynamic friction forces can lead to hunting limit cycles. With a friction isolator in place, these hunting cycles are avoided under specific conditions. A parameter study revealed that the friction isolator' drive stiffness should be kept as low as possible for optimal hunting cycle mitigation. On the other hand, the design of the friction isolator is constrained by parasitic frequencies of this mechanism and the stiffness in supporting directions. The experimental setup demonstrated that hunting cycles could be prevented with a friction isolator even with a control bandwidth of only 5 Hz, whereas the non-isolated system necessitated 3 times higher controller bandwidth. A 0.3 mm stroke of the friction isolator proved sufficient to prevent hunting. These experiments validate the suitability of the friction isolator as a solution for systems exhibiting hunting behavior.

Friction, Friction isolation, Hunting, Limit cycles, Flexure mechanism

1. Introduction

Contact-based bearings are widely used in positioning mechanisms for their relatively low cost and high support stiffness. However, nonlinear friction effects in rolling of sliding linear guides can limit the positioning performance, especially over time as wear deteriorates the system [1]. Typically, an integral action of a Proportional-Integral-Derivative (PID) controller will try to push the carriage of a linear guide through the friction towards a setpoint. However, the stick-slip effect introduces a discontinuity in the friction force, resulting in a sudden transition from standstill to movement. This stick-slip effect can lead to hunting limit cycles, which negatively affect the positioning performance of a servo system [2, 3]. This hunting behavior, also called friction-induced limit cycles, is a result of the combination of a controller with an integral action and a system containing a sudden transition between the static friction and smaller dynamic friction, the stick-slip effect [4, 5].

To prevent hunting from occurring in a positioning system, the integral action could be removed from the controller, but this could lead to tens of microns steady-state position error of the servo system. Reduction of friction is another option by using aerostatic bearings, but this type of bearing are not easily integrated into clean-room environments and significantly increases the cost. An alternative method to decrease the friction in mechanical bearings has been presented by Dong et al. [6], in which high frequency vibrations are exerted on the bearing rail to mitigate the undesirable nonlinear friction effects like stick-slip. This method called vibration assisted nano-positioning (VAN), shows a improvement in settling time up to 52% without a significant increase in heat and wear in the system. Introducing vibration into a high-precision positioning system, however, might be unwanted due to parasitic resonances which might jeopardize performance. Dong et al. later introduce the Friction Isolator (FI) concept [7–10], in which an intentional compliant joint - allowing limited movement in one direction - is inserted between the bearing and the actuator of the system. Figure 1 illustrates this principle.

This passive second stage atop the standard linear stage introduces a smooth force-position relationship close to the setpoint, at potentially low cost. Experimental results showed mitigation of hunting cycles and improved settling times.

Many parameters are of influence on the working of a friction isolator, such as the friction isolator's compliance and damping, system mass, travelled path, and control parameters. In recent studies [7-10] some of these have been investigated, however the influence of these parameters on the hunting behavior and the performance of a friction isolator is not always clear and a

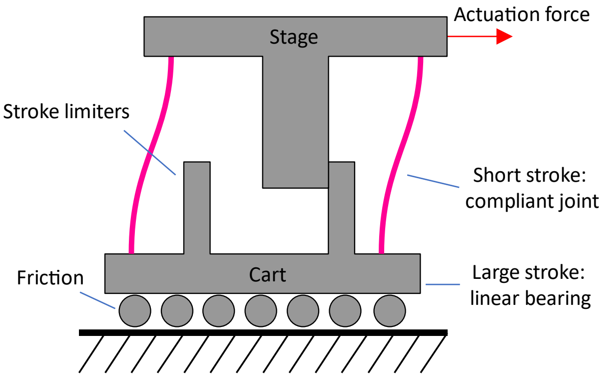


Figure 1. A schematic representation of a friction isolator. The actuation force positions the 'stage', which is connected through a compliant joint to the linear bearing cart. Stroke limiters are used to limit the maximum stroke of the friction isolator.

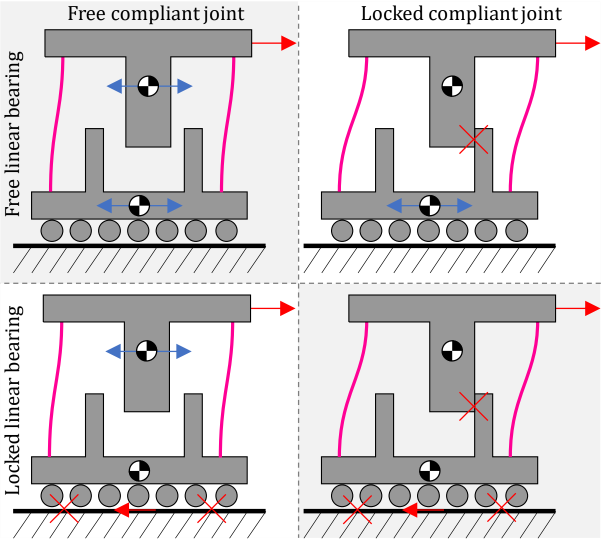


Figure 2. The controller experiences four limit cases, depending on whether the stroke limiter engages and locks the compliant joint (left vs right) or whether stiction at the linear bearing locks the cart (top vs bottom)

general guideline on designing a friction isolator is still missing and therefore the aim of this research [11].

Here, a general design strategy will be presented for a friction isolator system. This strategy is based on a parameter study in combination with simulations, from which it is identified which system parameters influence the hunting behavior. A setup is built to verify the model and showcase the improved settling behavior.

2. Methods

2.1. *Simulations*

The effects of friction in controlled systems have been extensively studied in literature, using various friction models. A sufficiently complex and computationally feasible model is the LuGre friction model [12,13], which is accurate in modelling presliding friction and the stick-slip effect. In the LuGre model, the contact between two bodies at asperities is modelled as elastic bristles. These bristles will deflect like springs when a tangential force is applied, leading to a friction force. If the tangential force is large enough, bristles will start to slip.

In our simulations, the friction isolator is modeled as two masses, which are connected through a spring damper (the compliant joint). On one of the masses LuGre friction forces are acting while on the other the actuation forces act, and the position is measured. The stroke limiters introduce reaction force between the two masses when engaged.

2.2. *Controller design*

The friction isolator is controlled by a PID controller. Here we use the cross-over frequency as main tuning parameter since it determines the bandwidth of the system and with that the response time of the controller. Using the moving mass of the system, the P, I, and D gains are computed [1].

The system to be controlled contains 4 limit cases (Figure 2) ; whether the cart is moving or stopped due to friction and whether the compliant mechanism is moving or stopped as it engages the stroke limiters. Between these cases, the moving mass and thus system dynamics differs significantly, it is either only the mass of the shuttle or the combined mass of the shuttle and the cart. Since a single controller is used to control all cases, the controller should be stable and have satisfying performance and stability in all cases. It was found that using the maximum mass for tuning the controller yields the best results, as is shown by Dong et al. [8].

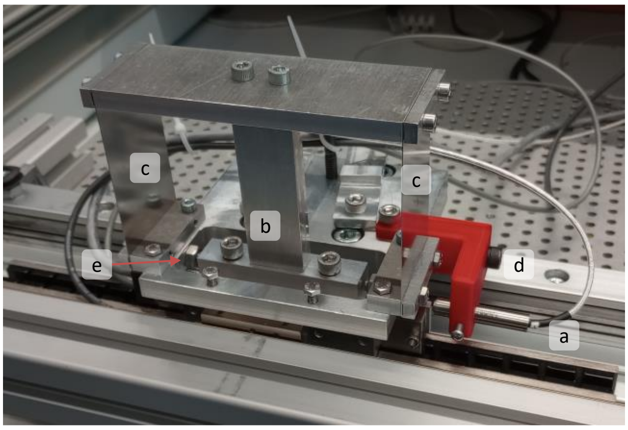


Figure 3. The experimental setup of a friction isolator. The linear actuator (a) positions the stage (b), which is connected through the compliant joint (c) to the linear bearing (d). Stroke limiters (e) are used to limit the maximum stroke of the compliant joint. A linear encoder and a capacitive sensor measure the displacement of the stage and the friction isolator, respectively.

2.3. *Hunting metric*

To quantify hunting, we propose here to count the number of oscillations in a fixed duration after expected settling time. Previous research [4] showed that hunting cycles have a duration in the order of seconds, with an amplitude in the order of millimeters. To find out whether a system shows stable hunting cycles the number of peaks in the position signal is counted after 10 seconds and over a period of 10 seconds (Figure 5). This provides sufficient time for the system to settle and if hunting cycles are observed, it can be stated that the system is affected by stable hunting cycles. The peaks are identified with a minimum time between the peaks of 0.5 seconds and a minimum peak height of 0.1 mm. Each hunting cycle will correspond to one peak.

2.4. *Parameter study*

A parameter study is conducted to find optimal parameters for the friction isolator and to study the effect of multiple parameters on the hunting behavior. Specially, two parameters are varied over a grid while the remaining parameters are kept constant. Firstly, the effect of static and coulomb friction parameters on a non-isolated system is investigated. Secondly, the influence of controller bandwidth and system mass parameters are varied for the non-isolated system. Thirdly, the stroke and stiffness of the friction isolator is changed. Lastly, the controller bandwidth and stroke of the friction isolator is changed to see if limit cycles will appear. Note that not all parameter variations are presented in this paper due to space limitations.

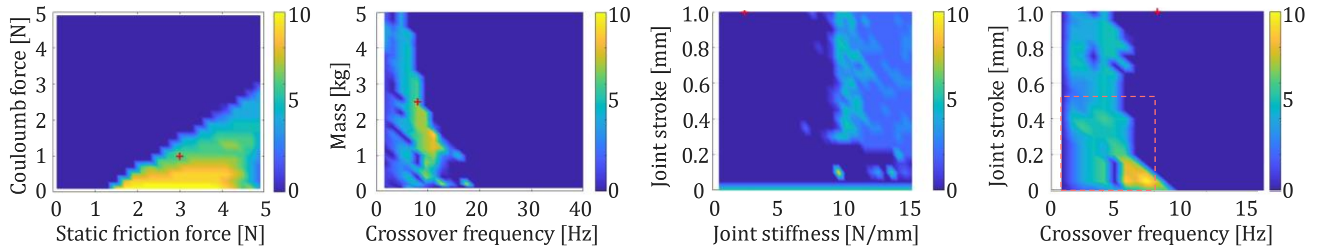
2.5. *Experimental setup*

A test setup is designed and built to experimentally verify the simulations and design method (Figure 3). On this set-up the friction isolator can be engaged or disengaged to see its effect on hunting. Furthermore, the controller bandwidth and friction isolator stroke are varied in a grid search to compare to the analytic results.

3. Results

3.1. *Parameter study*

Four pair-wise sets of parameters where varied to investigate their combined influence on hunting as illustrated by Figure 4..



A) B) C) D)

Figure 4. The resulting number of hunting cycles according to the metric for different combinations design parameters. The red marker indicates the evaluation point for the other studies. A) and B) are for the non-isolated case whereas C) and D) are for the isolated case. Be aware of the different horizontal scale when comparing B) to D). The box indicates the area of the experimental tests in D).

Here we selected a baseline parameter set, indicated with a cross and vary the parameter pair. The number of hunting cycles within the given time frame was selected as a hunting metric.

In Figure 4.A., it can be seen that the static friction force must be significantly larger than the Coulomb friction force for hunting to occur, approximately 1N in this case. This is logically explained by the stick-slip conditions, in which the static friction must be larger than the dynamic friction. A larger gap between the static and coulomb friction force leads to more hunting cycles. At the point of breakaway of the bearing, the control force is therefore also significantly larger than the dynamic friction force, resulting in a larger overshoot. Due to a larger error, the proportional part of the controller generates a larger force, and the build-up rate of the integral action is increased as well. This results in the breakaway force being reached sooner.

In Figure 4.B, it can be seen that with a higher cross-over frequency, larger than around 20 Hz in this case, no hunting occurs. With a higher cross-over frequency, the controller responds faster which can prevent hunting cycles from occurring. A higher required cross-over also places more stringent requirements on the system dynamics such as parasitic dynamics and time delays.

Figure 4.C shows that the stiffness of the compliant joint should be limited for the isolator to work. This is to be expected when taking the principle of the friction isolator into account. At a smaller stroke of the compliant joint, the stiffness of the joint can be higher, since the force applied to the bearing at the maximum stroke is then still smaller than the static friction force.

From this study it is found that the cross-over frequency also affects the hunting behavior of the friction isolator system (Figure 4.D). At smaller cross-over frequencies, hunting can still occur even in the isolated system. The boundary values of the cross-over frequency below which hunting occurs is affected by the stiffness of the compliant joint. A lower drive stiffness lowers the cross-over frequency at which hunting occurs.

3.2. *Experimental validation*

The friction isolator is tested for its ability to mitigate the hunting effect. Figure 5 shows a typical result of moving the setup back and forth over 100 mm, with a prescribed acceleration of 100 m/s^2. Results of both the isolated and non-isolated system are plotted in this graph, from which it can be seen that hunting cycles are present in the non-isolated system, while the friction isolator settles to the reference position. The simulations are not exactly equal to reality but show hunting in the same order of magnitude.

The same motion has been prescribed to the non-isolated system for a range of cross-over frequencies of the controller. The results of these experiments can be seen in Figure 6. This experiment shows that increasing cross-over frequency increases the performance of the system, since the response time to errors becomes smaller and the overshoot in the hunting cycles decreases. As expected from the parameter study, with increasing cross-over frequency the hunting cycles eventually are mitigated. Though the required cross-over frequency to mitigate the hunting behavior is found to be 14.3 Hz in this experiment. While for the isolated system 5 Hz is sufficient. Also, observe that no hunting seems to occur at the cross-over frequency of 9.5 Hz. This indicates that hunting is a partly understood behavior, that depends on a multitude of stochastic factors, as in other runs hunting was introduced with the same controller settings.

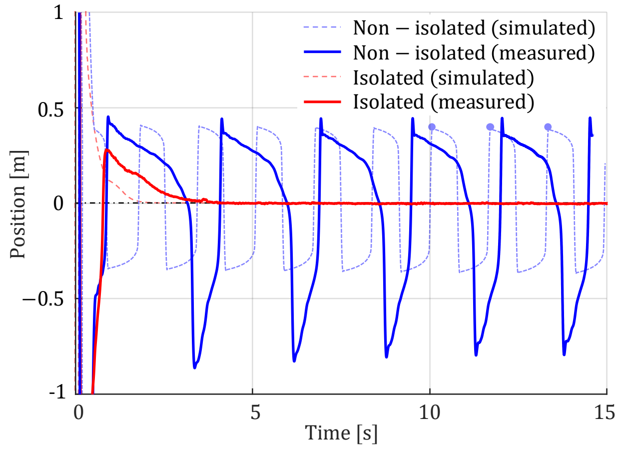


Figure 5. The simulated and measured position of the non-isolated and isolated system after settling from a step movement. The same cross-over frequency is used in all cases. The blue circles indicate the counting points as to compute the metric (section 2.3).

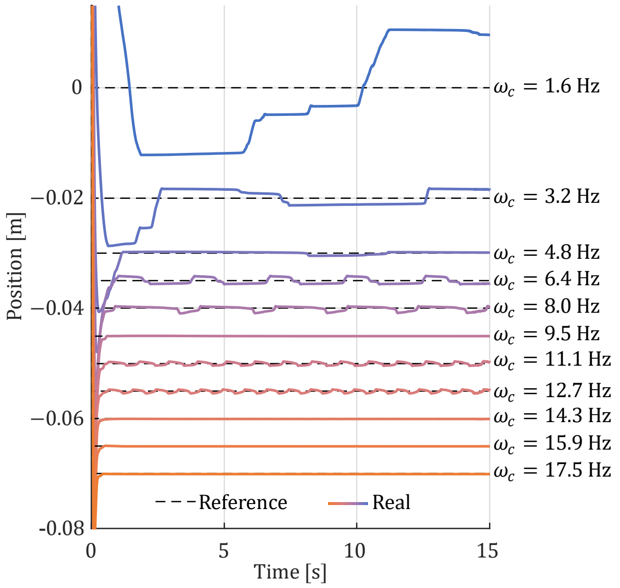


Figure 6. The settling behavior for a range of cross-over frequencies for the non-isolated system. To make a clear distinction between the different lines a small virtual offset is introduced while the actual set-point was the same for all experiments.

The stroke of the compliant joint and the cross-over frequency of the controller have been varied for experiments on the test setup. The results of this experiment can be found in Figure 7. At higher cross-over values, hunting is not present in the system at all. The trend is similar to the friction isolator cross-over parameter study of Figure 4.D, albeit grainier. This validates both the simulation setup and the finding that for a sufficiently high cross-over frequency no hunting cycles appear. It should be noted that the amount and frequency of hunting differs significantly between simulations and set-up due to the time and position varying nature of the friction parameters.

4. Design guidelines

To determine if a friction isolator is needed and what design parameter are to be chosen, we propose the following four step approach:

1. Determine the friction values of the bearing. If the difference between static and coulomb friction of the bearing is small, in this case < 1N, stick-slip might not occur and thus hunting cycles will not be an issue.
2. Determine the desired and reachable cross-over frequency of the system, if this frequency is limited, a friction isolator might be useful. A higher possible cross-over frequency can mean hunting cycles do not occur in the system and the benefit of a friction isolator is limited. This desired cross-over also gives minimal value for the parasitic frequency as used to design a compliant joint.
3. From the static friction value, combined with the maximum stroke of the compliant joint, a maximum drive stiffness can be determined. Based on this, a compliant joint is designed such that the parasitic frequencies do not interfere with the desired cross-over frequency.
4. Finally, the PID controller parameters can be determined when the mass of the system is known [1].

5. Conclusions

The conducted parameter study on the main parameters of a friction isolator system showed that a limited bandwidth of the applied PID controller and a significant gap between static and dynamic friction forces may result in hunting cycles to occur.

Introducing a compliant joint in a system can prevent these unwanted hunting cycles. It is found that the drive stiffness of this compliant joint should be as low as possible, for the best mitigation of hunting cycles. The design of the compliant joint is limited by the stiffness in supporting directions and the parasitic resonance frequencies that are introduced by this mechanism. The parameter study showed that the friction isolator is a robust method to mitigate hunting cycles against various friction values.

The experimental setup showed mitigation of hunting cycles using a friction isolator for a control bandwidth of only 5 Hz, while the non-isolated system requires a higher bandwidth of 15 Hz. A compliant joint stroke of 0.3 mm is sufficient to prevent hunting from occurring. These experiments verify that the friction isolator is a suitable solution for systems that show hunting behavior.

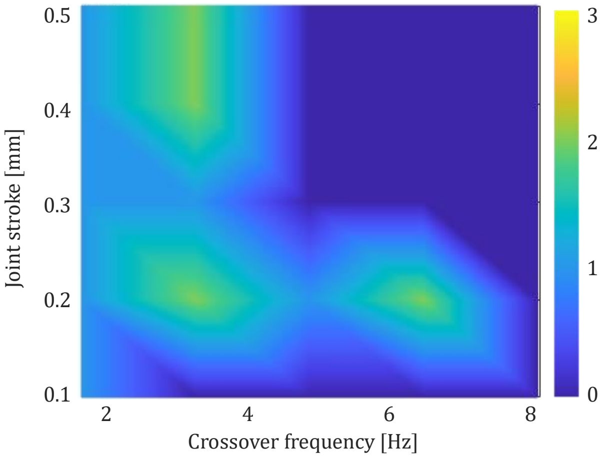


Figure 7. The measured number of hunting cycles for a range friction isolator joint strokes and cross-over frequency. These test results parallel measured results in the boxed area of Figure 4.D.

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