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Open-Loop Velocity Planning to Mitigate the Stiction Effect in Pushing Positioning

Laine Mears
Clemson University, mears@clemson.edu

Thomas Kurfess
Clemson University

OPEN-LOOP VELOCITY PLANNING
TO MITIGATE THE STICTION EFFECT IN PUSHING POSITIONING

Laine Mears, Ph.D., P.E.
Assistant Professor, Mechanical Engineering
International Center for Automotive Research
Clemson University
Clemson, South Carolina

Thomas R. Kurfess, Ph.D., P.E.
Professor and BMW Chair, Mechanical Engineering
Director, Campbell Graduate Engineering Center
International Center for Automotive Research
Clemson University
Clemson, South Carolina

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ABSTRACT

Actuation by pushing has been studied as both an economic and flexible alternative to traditional pick-and-place part positioning. Pushing actuation reduces the need for specialized end effector tooling, and can simplify the underlying automation mechanism (e.g., a 2-DOF rotational positioner in place of a 6-DOF robot). However, the interaction of the positioning command with the nonlinear behavior of sliding friction introduces a new source of position error. The stick-slip effect of friction (i.e., stiction) reduces local controllability in precision positioning applications. In this paper, we describe a method for identification of dominant stiction frequency across a range of actuation velocities and compensation of the input command signal to avoid actuating the part at or near the stiction frequency. The result when compared with simple constant-velocity actuation is a reduction in variation of actuating force and an improvement in achievable position accuracy through more consistent control of part sliding at end-of-actuation-stroke.

INTRODUCTION

Actuation by pushing has been studied as both an economic and flexible alternative to traditional pick-and-place part positioning. Using pushing actuation, a number of system improvements can be realized. Foremost, the complexity of positioning systems can be reduced. In a traditional pick-and-place system, space at least 3 degrees of freedom are typically employed: two for Cartesian positioning and one for changes in elevation. Employing a pushing actuating system allows for movement of a part along its current plane. This allows the part to not only be precisely positioned, but it can also be programmed to follow a predetermined actuation path; for example to clear obstacles. Additionally, pushing actuating can be employed using a single fixed pusher tip rather than the part-dependent tooling commonly found in pick-and-place systems. This reduces system complexity cost and improves flexibility in multipart operations.

However, actuation by pushing can introduce a stick-slip effect (*stiction*), which can degrade system accuracy. The system friction and stiction effect can be simply modeled, and system behavior predicted. We propose to use this information to aid in velocity planning for the

push actuation, in order to mitigate the stiction effect.

ACTUATION BY PUSHING

In the past 20 years, there have been numerous research efforts in the field of precision positioning by sliding the target object across a surface. Peshkin and Sanderson describe the motion of a sliding workpiece for all possible pressure distributions on the support surface [Peshkin and Sanderson 1988]. Zesch and Fearing explore force-controlled pushing for microparts with positional results in the 1µm range [Zesch and Fearing 1998]. Lynch and Mason have done extensive work on planning and control for stable pushing in the application of robotic manipulation as an alternative to pick-and-place positioning, including feasibility studies through both kinematic and force analyses [Lynch and Mason 1995; Lynch and Mason 1996]. Lynch also explores friction estimation for pushed objects and open-loop control for pushing the general polygonal shape, characterized by the "maneuverability" property [Lynch 1993; Lynch 1999].

A number of research efforts have been directed at positioning parts using impact or single-contactor pushing actuation. Benefits are a more inexpensive and flexible actuation system that can be designed for very large or very small parts. Research in application of impact to positioning has mainly been focused on static initial and end conditions and single impact system input. That is, a part initially at translational and rotational rest is struck once to impart a velocity, and then allowed to come to rest under environmental conditions (typically friction).

Application of these concepts to impact-based static positioning systems is treated separately by [Mendes, Nishimura et al. 1996] in the printed circuit board positioner, by [Liu, Higuchi et al. 2003] in their piezoelectric positioning table, as well as by [Siebenhaar 2004] in electromechanical hammer control.

FRICTION MODELING APPROACHES

Friction is present in all mechanical systems, and contributes significantly to force analysis and control of motion systems. In this case, it is important to fully understand and accurately model friction when developing an idealistic model of the physical system.

There exists substantial research on modeling of static and dynamic friction, both in the idealized linear case and the nonlinear case. Both [Olsson, Astrom et al. 1998] and [Canudas-de-Wit, Olsson et al. 1995] give a comprehensive overview of the major static and dynamic friction models utilized in practice. These ideas are extended to the special case of low velocity friction compensation by [Adams and Payandeh 1996].

The classical friction model was derived by Coulomb and is of the linearized form

$$F = F_C = \mu F_N \qquad (1)$$

This model has been successfully applied in the literature, and is the basis for generalized idealistic friction modeling. It has been successfully augmented by adding a linear viscous component of the form

$$F_C = \mu F_N + k_v v \qquad (2)$$

where k_v is the proportionally constant of force resistant to velocity. Simultaneous identification of μ and k_v through decrement analysis is treated by [Feeny and Liang 1996].

These model forms only apply to moving objects $\left(\dfrac{dx}{dt} \neq 0 \right)$. However, when velocity is at or near zero, there occurs distinct discontinuous and nonlinear behavior as shown in Figure 1.
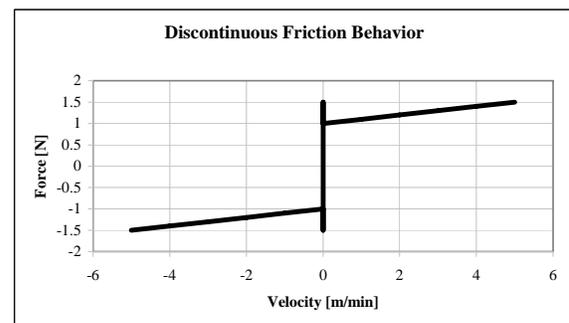

FIGURE 1. LINEAR VISCOUS COULOMB FRICTION (UNDEFINED AT V=0).

When a stationary object is excited by a force, it acts as a spring, resisting the force until the magnitude overcomes its static friction, a phenomenon known as stiction. After the static friction is overcome and the object begins to move, there is a decidedly nonlinear force-velocity relationship during the transient phase. One simple representation is to augment the Coulomb model with a specification at zero velocity:

$$F(0) = F_e , \quad v = 0 \; and \; |F_e| < F_S$$
$$F_e \equiv applied \; force$$
$$F_S \equiv static \; friction \; force = \mu F_N$$

(3)

This model presents problems near $v = 0$ due to discontinuity and localized nonlinear behavior. Stribeck developed a model which separately defines the nonlinear portion of friction force in the neighborhood of $v = 0$:

$$F = \begin{cases} F(v), & v \neq 0 \\ F_e, & v = 0 \; and \; |F_e| < F_S \\ F_S, & otherwise \end{cases}$$

(4)

where $F(v)$ is the force required to maintain a constant velocity. Stribeck empirically determined [Olsson, Astrom et al. 1998]

$$F(v) = F_C + (F_S - F_C)\exp\left[-\left|\frac{v}{v_s}\right|^{\delta_s}\right] + k_v v$$
$$F_C \equiv \mu_k F_N$$
$$F_N \equiv normal \; force$$

(5)

Note that this is the augmented Coulomb model with a transient decay component to account for the discontinuity between definitions at v=0.

For manufacturing systems, Lee addresses frictional energy of contact with respect to the hot rolling process [Lee, Choi et al. 2004]. [Tao and Lovell 2002] also address friction modeling in manufacturing through material removal process modeling. Additionally, new models of friction are being developed that lend themselves well to manufacturing system control due to their continuously differentiable nature

[Canudas-de-Wit, Olsson et al. 1998; Makkar, Dixon et al. 2005].

EFFECT OF STICTION

At low velocity, the actuating force is less than the static friction force F_S (the force required to break static friction and begin movement). This disparity between the breakaway force and the force required to maintain velocity can lead to jerky movement or stick-slip motion, as shown by simulation and experiment in Figure 2.
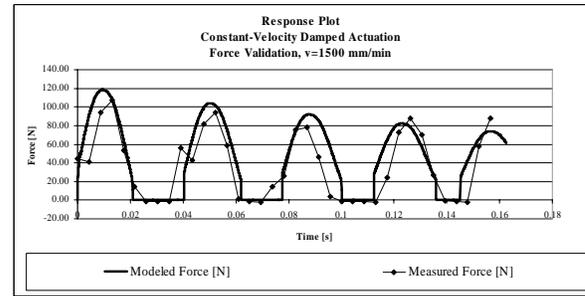

FIGURE 2. SIMULATED AND MEASURED FORCE RESPONSE OF PUSHING ACTUATION (V=1500MM/MIN, M=18.9KG).

This nonlinear motion, if not properly controlled, can lead to limit cycles or nonconvergent behavior in fine positioning systems [Olsson and Astrom 2001]. However, as explained by [Hirschorn and Miller 1999], the more accurately and completely friction is modeled, the better performance achievable by the compensating controller.

A number of augmented friction models exist which provide accurate results in different application domains. Bliman and Sorine developed a group of dynamic friction models to account for velocity-dependent behavior [Canudas-de-Wit, Olsson et al. 1995]. The LuGre model extends the model of Dahl to capture frictional properties such as stick-slip (known as stiction) and frictional time lag [Canudas-de-Wit, Olsson et al. 1995]. [Dupont, Armstrong et al. 2000] have developed a dynamic model that captures both stiction and observed presliding displacement. The model of [Canudas-de-Wit, Olsson et al.1995] brings together most experimentally observed effects: the Stribeck effect, hysteresis, the spring-like

behavior of stiction, and variation in the static friction force. [Song and Kumar 2003] analyze available rigid body dynamic models, and [Kraus, Kumar et al. 1998] propose a method for switching between rigid and compliant contact models in frictional systems to avoid discontinuities.

Recently, there has also been work to capture frictional effects for small displacement actuation of rigid bodies. [Ferrero and Barrau 1997] specifically study friction under small displacement and near-zero velocity. This is a highly nonlinear regime not modeled by Coulomb.

FRICTION MODELING OF CONSTANT-VELOCITY PUSHING

The studied pushing occurs beginning with zero relative velocity between the part and the surface. Moving the part requires a discontinuous transition between static and kinetic friction. After breakaway, the required force drops, often causing overshoot of the desired position, and possible return to zero velocity if the part loses contact with the pusher and comes to rest. This stick-slip motion is a common phenomenon occurring in frictional systems, and affects system accuracy and stability. The idealized one-dimensional system is shown in Figure 3.
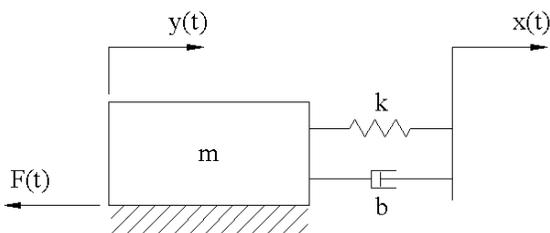

FIGURE 3. IDEALIZED RELATIVE MOTION SYSTEM.

The system behavior is governed by the output y(t) observed in response to the input x(t). The simulated force response using F(t) as described in (5) is shown in Figure 4.
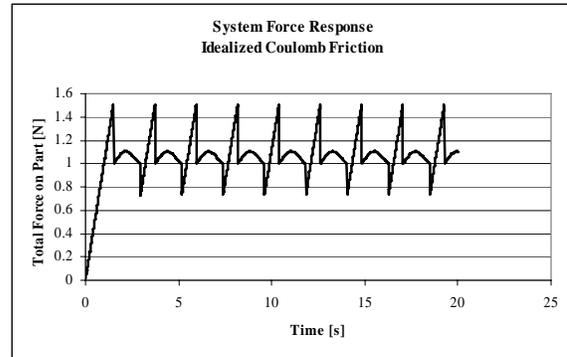

FIGURE 4. SIMULATED IDEALIZED SYSTEM FORCE RESPONSE TO RAMP INPUT.

STICTION FREQUENCY

The modeling and actuation described and demonstrated in the previous sections is fundamentally based on the assumption that actuation of the pushing element occurs at a fixed velocity. Though this simplifies motion control programming, it may not be optimal for controlling position of the sliding object. A new method of generating the actuation input command is proposed, based on analysis of the highest frequency of the stiction effect.

Resonance of Stiction

Fixed-velocity actuation can give rise to the stiction condition described previously, whereby upon impact the part accelerates, loses contact with the actuator, and comes to rest. This process is repeated, producing large nonlinear fluctuations in applied force and part velocity. In general, this condition is detrimental to precise and accurate centering.

In our model of part actuation by pushing, it is observed not only that the stiction cycle described in previous sections occurs, but also that it occurs at a relatively constant frequency over multiple contacts of the actuator with the part. For the simulated and observed force data in Figure 5, the modeled and observed damped natural frequencies were determined to be 29.9 Hz and 27.8 Hz, respectively.
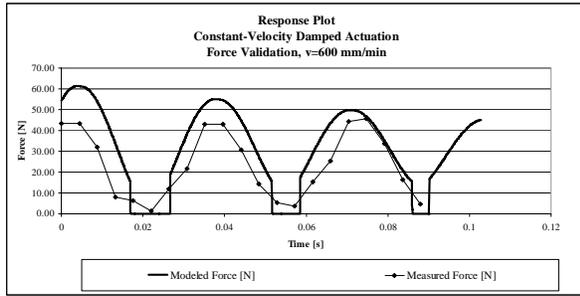
FIGURE 5. SIMULATED AND MEASURED FORCE
RESPONSE OF PUSHING ACTUATION
(V=600MM/MIN, M=18.9KG).

It is proposed that the input signal be filtered in the frequency domain to remove a band of frequencies around this *stiction resonance* value in order to avoid excitation of this frequency.

Invariance to Velocity

Prior to developing a filtering algorithm, the resonance frequency is analyzed over a range of input velocities. The dominant frequency of the reaction model data is calculated over the range of actuation velocities. Specific results of force modeling for several velocity cases of a steel part (m=18.9 kg) sliding on a carbide surface are given in Figure 6.
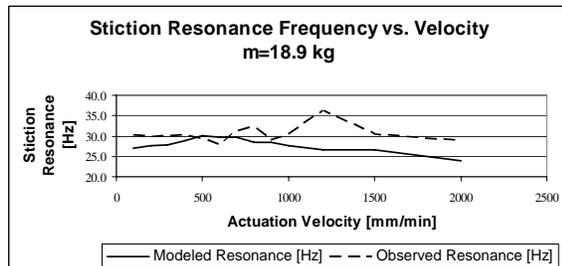

FIGURE 6. LARGEST FREQUENCY COMPONENT
OF POSITION VS. CONSTANT ACTUATION
VELOCITY.

Two conclusions are drawn from this data:

- Error of the simulation with respect to dominant frequency is relatively low across the range of frequencies tested. The maximum absolute error is 9.7 Hz and the average error is 3.1 Hz. The

simulated system is valid with respect to stiction resonance frequency prediction.

- The resonance frequency is relatively constant across the domain of actuation velocity. The absolute range of the simulated data is 6.2 Hz and range of the observed data is 8.5 Hz.

Due to the insensitivity of the stiction resonance frequency to changes in velocity over the applicable range of the system, the validated model can be used to predict the stiction resonance of the system for a given part. The system can then apply a fixed limit bandstop filter to the input velocity command signal around the resonance frequency rather than using a velocity-specific filtering algorithm.

COMMAND FILTERING APPROACH

The data and analyses of previous sections are based on constant-velocity actuation. Once a resonance frequency is identified, the constant-velocity input $\dot{x}(t)$ is filtered to eliminate actuation near resonance.

Using a 3rd order Butterworth filter, the frequency band from 20 Hz to 40 Hz is removed from the constant velocity signal to create the *anti-resonance* input signal for this part. The signal is attenuated in the frequency range around the resonance frequency. The resultant bandlimited filtered velocity command signal (filtered square wave step) is shown in Figure 7 with the original constant velocity input for reference.
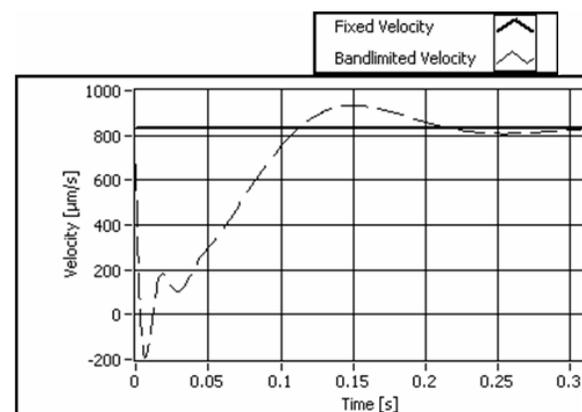

FIGURE 7. BANDLIMITED VELOCITY SIGNAL
(20HZ-40HZ REMOVED).

This actuation command results in the slide position contour shown in Figure 8. To better visualize the comparison of the system responses, the bandlimited position is normalized to the constant velocity ramp command position.
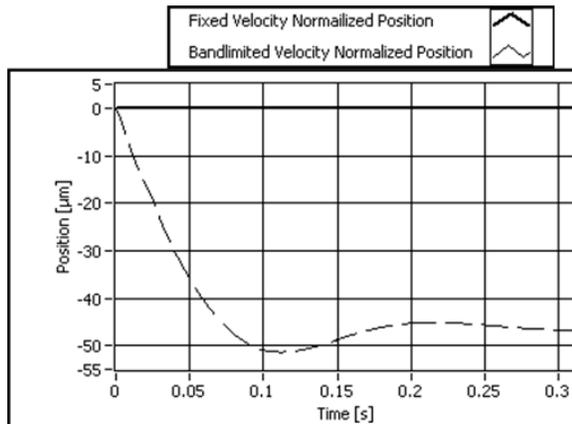

FIGURE 8. SIMULATED COMMAND POSITION NORMALIZED TO CONSTANT VELOCITY INPUT.

The filtered input command expectedly causes a lag in the system which must be compensated for in trajectory planning. In this simulation example, the contour falls away from the fixed-velocity contour near the beginning of actuation, then settles to a steady-state actuation that lags the constant-velocity signal by 47 µm. The net effect of this profile is acceleration of the part after contact near the beginning of the actuation, up to the steady-state velocity. This profile dynamically reduces the resonant effect of stiction.

SIMULATION OF MODIFIED SYSTEM

The profile of Figure 8 is used as the input to the part sliding simulation under the conditions m=18.9 kg, v=600 mm/min. Contact stiffness and damping were determined by experiment for inclusion in the simulation. The modeled force responses for the constant velocity and bandlimited velocity inputs are shown in Figure 9.
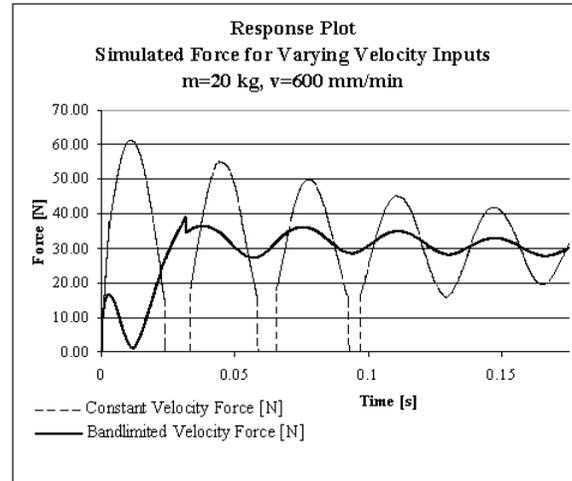

FIGURE 9. SIMULATED FORCE RESPONSE WITH CONSTANT AND BANDLIMITED VELOCITY INPUTS.

For actuation at constant velocity, the force fluctuates from a maximum of over 60 N to 0, indicating loss of contact with the part being pushed. In this example, the actuator loses contact with the part three times before settling into a slowly decaying resonant pushing mode. When actuated with the bandlimited velocity signal, the force achieves an approximate steady-state level in 0.05 seconds, fluctuates by a maximum value of only 17 N, and maintains contact throughout the actuation (no loss of control).

VALIDATION OF MODIFIED SYSTEM

It is expected that parts of higher mass are more susceptible to actuation problems arising from the large variations in force and position caused by stiction. A part of m=18.9 kg is tested using the frequency bandlimiting method. This part is run across a range of velocities from 100 mm/min to 2000 mm/min, both at constant velocity and frequency-bandlimited velocity command input. The force results of the case for v=600 mm/min are given in Figure 10.
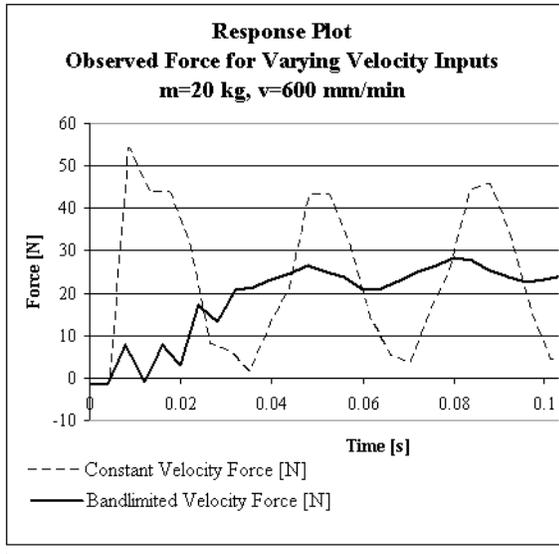
Response Plot
Observed Force for Varying Velocity Inputs
m=20 kg, v=600 mm/min

FIGURE 10. FORCE OBSERVATION FOR CONSTANT AND BANDLIMITED VELOCITY ACTUATION (V=600MM/MIN, M=18.9KG).

Response Plot
Observed Position (Normalized) for Varying Velocity Inputs
m=20 kg, v=600 mm/min

FIGURE 11. NORMALIZED POSITION OBSERVATION (V=600MM/MIN, M=18.9KG).

As predicted by the simulation, the constant velocity input results in a resonant "tapping" of the part, whereby the force periodically drops to zero, indicating loss of contact. Alternatively, the bandlimited velocity profile results in only a single acceleration to steady-state actuation, with less than 10 N of variation at steady state.

Additional experimental result cases for force over a range of base actuation velocities from 100 mm/min to 2000 mm/min were performed. In all cases, the force rises to the average pushing value and exhibits less fluctuation than with constant velocity pushing. In addition, the force is never reduced to zero during the actuation, indicating that contact with the part is never broken.

The position data for this experiment yield similar results when compared with simulation. The constant-velocity actuation shows a periodic free sliding effect, while the bandlimited-velocity actuation shows a smoother approach to steady state with less fluctuation. The data are normalized to the constant velocity (straight line) slide position in Figure 11.
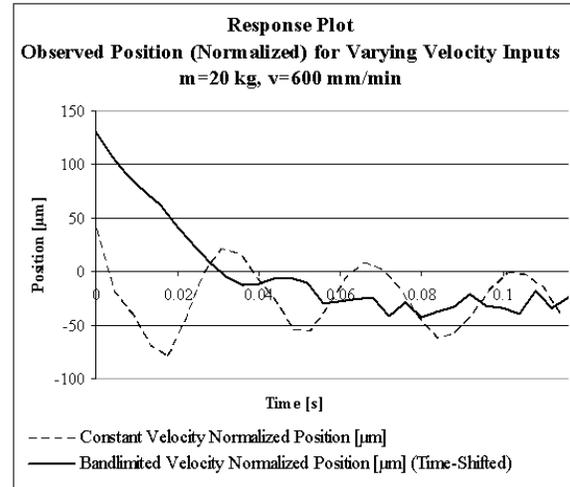
Additional experimental result cases over the range tested confirm the force result findings that the part undergoes reduced fluctuation with bandlimited velocity actuation, and contact between actuator and part is maintained.

CONCLUSION

Up to 80% reduction in positional fluctuation was observed using the modified input scheme for pushing positioning rather than constant-velocity pushing, with complete elimination of loss of contact in all tested velocities.

This input signal augmentation requires additional calculation time, and should be applied only where significant benefit can be achieved to avoid preemption of higher priority tasks in the controller, such as. It is anticipated that this benefit is more valuable for heavier part actuation.

Results of this work have been applied to both magnetic-chuck center-based material removal equipment and to prepositioning for precision roundness metrology.

Additional work is planned to

- Improve friction modeling for simulation of system performance
- Investigate alternative input schemes for comparison

- Create a system identification scheme to avoid having to model individual cases for the velocity planning rule
- Use the system identification scheme to predict resonant frequencies of stiction at different actuation velocities, and
- Apply and validate the algorithm in a velocity-specific form (relax the velocity-dependent frequency assumption).

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