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Vibration Isolation of a Feedback Linearized Model for a Contactless Electromagnetic Isolator by Virtually Varying Mass Control

Chenyang Ding*, A.A.H. Damen*, and P.P.J. van den Bosch*
*Department of Electrical Engineering, Eindhoven University of Technology
Den Dolech 2, P.O.Box 513, 5600 MB Eindhoven, The Netherlands

Abstract—Two active vibration isolation methods, Virtually Increasing Mass (VIM) control and On-Off Mass (OOM) control, based on absolute acceleration feedback are proposed for a contactless Electro-Magnetic Isolator (EMI) being designed for heavy payload. They are applied to a stabilized feedback-linearized model for the vertical DOF of a candidate EMI design. The control objectives of this application are seeking, vibration isolation and force disturbance rejection. The vibration isolation performance (transmissibility) up to 1000 [Hz] is evaluated by simulation with both white noise vibrations and sweep sine vibrations. Simulation shows that the transmissibility magnitude peak is shifted to lower frequency by the proposed vibration isolation methods. The high-frequency nonlinear behaviors are analyzed. Besides, the step response to a constant force disturbance is not compromised and the seeking performance is acceptable. Above all the simulation results, the proposed control methods are feasible for the presented application.

Index Terms—Vibration Control, Permanent Magnets, Magnetic Levitation.

I. INTRODUCTION

In a micro-lithographic machine, referred to as a wafer-stepper, shown in Fig. 1, a pneumatic suspension system is used to achieve the following objectives:

- Support the payload (a complex lens system weights a few thousand kilograms).
- Isolate the payload from floor vibrations on six Degrees-Of-Freedom (DOF) in a broad frequency band.
- Reject the force disturbances directly applied to the payload.

Shown in Fig. 2, the pneumatic suspension system has three pneumatic isolators. They have the advantage of low stiffness and high capacity of payload gravity compensation [9]. However, the high frequency dynamics of the pneumatic isolators is difficult to be accurately modeled or measured, which limits the control performance. Besides, the pneumatic isolators are difficult to be implemented in vacuum environment.

Therefore, an Electro-Magnetic Isolator (EMI) is being designed as an alternative. Three of the EMIs can be used to replace the three pneumatic isolators in Fig. 2 to form a 6-DOF contactless suspension system. A single EMI has the following properties:

- No mechanical contact between the translator and the stator.
- Capability of gravity compensation of hundreds of kilograms based on passive permanent magnetic (PM) force.
- Inherent instability, proved in [8].

A 3-DOF model for a candidate EMI design is described in [7]. The nonlinear force-position relation is time-invariant so that it is reasonable to assume that the nonlinear force-position relation can be accurately measured. Feedback linearization can be used not only to compensate the nonlinearity but also to decouple the 3-DOF model into three exactly the same 1-DOF models, each of which is a double integrator disturbed by external forces. Vibration isolation methods are studied for this 1-DOF model.

Control strategies to simultaneously achieve stabilization and vibration isolation of a contactless electromagnetic system were proposed by some researchers. Stabilization and vibration
isolation is achieved in [5] by absolute position feedback, which is not feasible in this application. A 6-DOF suspension system composed of four contactless electromagnetic actuators is built [4] for a payload of maximal 200 [kg]. A PI controller is used to stabilize the electromagnetic actuator based on relative position feedback. An \( H_m \) controller is used to isolate the floor vibration based on absolute acceleration feedback and to deal with the changing parameters and uncertainties simultaneously. In our previous study [7], a combination of feedback linearization and \( H_m \) control is applied and robust performance is achieved. Theoretically, the \( H_m \) control would achieve comparable performance to any linear controller if the weights are properly designed. However, the \( H_m \) controller is complicated to design. Alternative solutions with simpler control schemes are studied.

In a suspension system, vibration isolation can be achieved by varying the system parameters, including stiffness, damping ratio, and payload mass. The ideas of improving vibration isolation by turning on and off the spring force or damping force has been studied by many researchers. Karnopp [1] invented the on-off skyhook control, in which, the payload deceleration is achieved by turning on the damper only when the absolute velocity of the payload is of the same sign with the relative velocity. S. Rakheja [6] introduces another on-off damping control, in which, the damper is turned on only when the absolute velocity and the relative displacement had the opposite signs. Mehdi Ahmadian [2] proposes the no-jerk on-off skyhook control which removes the discontinuity of the damping force. Yanqing Liu [3] proposes the on-off stiffness control, in which, the spring stiffness should be as small as possible when the relative displacement had an opposite sign with the absolute velocity. All of these methods need absolute velocity sensors. However, seismic acceleration sensors were intended because of their comparably low cost and low noise at low frequencies. Most of previous studies on active suspensions were for car suspensions. The vibration isolation performance is evaluated by comparison of both active and passive time responses, or frequency response up to a few dozen Hz. However, vibration isolation performance at higher frequencies (up to hundreds of Hz) is also concerned for this application.

In this paper, two vibration isolation methods by virtually varying the payload mass based on absolute acceleration feedback are proposed. Each of them is applied in parallel with a stabilizing controller to a feedback-linearized model for the vertical DOF of a candidate EMI design. Before the actuator is manufactured, the performances of vibration isolation up to 1000 [Hz], seeking performance, step response to constant force disturbance are evaluated by simulation. The effects of the unit delay and sensor resolutions are considered.

II. PROBLEM DESCRIPTION

Fig. 3 shows the overall control diagram for the nominal model, which is described in subsection A. The nominal model is still unstable. It is stabilized by a Stabilizing Controller based on relative displacement feedback. The performance requirements of the closed-loop system are described in subsection B. The Vibration Isolation Controller is designed to improve the vibration isolation performance based on absolute acceleration feedback.

A. Nominal Model

The 3-DOF model of the candidate EMI design is described in [7]. Its two main components, the translator and the stator, do not have any mechanical contact. The relation between the PM force and the translator-stator relative position is highly coupled and nonlinear. However, since the PM force is time-invariant, the PM force-position relation is feasible to be accurately measured. Subsequently, the nonlinear and coupled PM forces can be compensated by feedback linearization. As a result, the 3-DOF model is decoupled to three exactly the same 1-DOF models, each of which is a double integrator disturbed by the force \( f \), as shown in Fig. 4. This 1-DOF model is defined as the Nominal Model. It is also illustrated in the dashed square in Fig. 3. Since all the three 1-DOF models are exactly the same, only the model for the vertical translational DOF is studied. The working range is defined as \( d_A \in [-1, 1] [\text{mm}] \). Beyond this range, mechanical hard-stops are installed between the translator and the stator for safety.
One important property for the PM force-position relation is that there exist a unique position in which the vertical PM force \( F_z \) reaches the peak. At this position, the stiffness \( k_z \) defined by

\[
k_z = -\frac{dF_z}{dc}
\]

is zero. However, the stiffness \( k_z \) is in the order of \( 10^4 \) [N/m] at the working range boundary.

In this study, exact knowledge of the PM force-position relation is assumed to be known. However, the passive PM force compensation still has error because of the time delay and limited position sensor resolution. The error of the PM force compensation, denoted by \( f \) in Fig. 4, depends on the stiffness, the absolute velocity, and the sampling rate. Under the same condition of floor vibration and sampling rate, higher stiffness induces higher force compensation error due to the unit delay. Many seismic acceleration sensors have massive noise at low frequencies and gain rise at high frequencies. The low frequency noise is filtered by a high-pass filter with a low cutoff frequency. The gain rise should be reduced by a low-pass filter with a high cutoff frequency. The block Band-pass Filter is a combination of the two filters.

**B. Control Objective**

There are three performance requirements in the final application. The vibration isolation performance is of the most important. It is evaluated by the transmissibility, defined by the transfer function from the floor displacement to the absolute displacement of the translator. The transmissibility magnitude should be 0 [dB] for low frequencies up to a cut-off frequency and decreases at higher frequencies. The cut-off frequency is required to be lower than 1 [Hz] and the decreasing rate at high frequencies is at least -40 [dB/dec]. In the final application, the translator is possibly required to be positioned at certain relative positions. Therefore, the desired controller is required to follow the reference and seek to any relative positions within the working range. The overshoot is preferably zero but the settling time can be as long as hundreds of seconds. The reference signal can be re-shaped by a pre-filter before it is sent to the control loop. Therefore, the seeking performance of the stabilized control loop is easy to achieve. Since feedback linearization is involved, the steady-state error of the closed-loop system response to a stepwise constant force disturbance is required to be zero in case of PM force compensation error. Besides, the peak of this response should be as small as possible. A large peak raises the danger of collision between the translator and the hard-stop. The frequency response to the force disturbances \( F_d \) in Fig. 3, defined as the compliance, should have low magnitude.

The control structure is a stabilizing controller and a vibration isolation controller connected in parallel. The stabilizing controller is designed for stabilization and acceptable compliance based on relative displacement feedback. The vibration isolation controller is designed for the stabilized closed-loop based on an additional feedback signal (absolute acceleration) to improve the transmissibility without compromising the compliance.

**III. VIRTUAL VARYING MASS CONTROL**

Previous studies on semi-active suspension system proposed many vibration isolation algorithms based on varying damping ratio and varying stiffness. However, they are all based on absolute velocity feedback. For a suspension system with fixed stiffness and damping ratio, larger payload mass results in better vibration isolation performance and better force-disturbance rejection performance. Based on this principle, two vibration isolation methods by virtually varying the payload mass based on absolute acceleration feedback are proposed. The On-Off Mass (OOM) control has the form of

\[
F_m = -\Delta_m a_A, \quad \text{where} \quad \Delta_m = \begin{cases} m & \text{if } a_{AVR} \geq 0, \\ 0 & \text{if } a_{AVR} < 0, \end{cases}
\]

where \( a_A \) and \( v_R \) are the absolute acceleration and the relative velocity, respectively. The relative velocity can be obtained as the derivative of the measured relative displacement. Based on observation of response to sinusoidal vibrations, the absolute velocity is of the same sign of the absolute acceleration whenever the relative velocity and the absolute acceleration have the same sign. The control effort of the on-off mass control is against the absolute velocity and the payload vibration is therefore reduced. The VIRTually Increasing Mass (VIM) control is proposed as

\[
F_m = -\Delta_m a_A, \quad \text{where} \quad \Delta_m = m.
\]

The control diagram of both control methods are shown in Fig. 5(a). \( M \) is the payload mass and \( m \) is a constant parameter to be designed. Both virtually varying mass control methods are equivalent to increase the payload mass from \( M \) to \( M + \Delta_m \). The equivalent control diagram is shown in Fig. 5(b).

**IV. PERFORMANCE EVALUATION**

The purpose of this simulation is to evaluate the effects of the two proposed vibration isolation methods to the three control objectives. A Matlab model for the the vertical DOF of the nonlinear EMI is built for real-time simulation. The mass of the payload is \( M = 827.55 \) [kg] and the gravitational acceleration constant is \( 9.81 \) [m/s²]. The relation between vertical PM force \( F_z \) and vertical relative displacement \( d_R \) is taken from [7],

\[
\text{Fig. 5. Virtually varying mass control diagram.} \quad r: \text{the reference, } d_c: \text{floor displacement, } F_c: \text{force disturbance.}
\]
shown in Fig. 6. The controller is a combination of a stabilizing controller and one of the two proposed virtually varying mass controllers with different parameters \( m \in \{0, 1000, 2000\} \) [kg] as defined in (2) and (3). Note that \( m = 0 \) indicates that only the stabilizing controller is functioning. The stabilizing controller used in simulation is also taken from [7].

\[
C_s = \frac{40000(s + 0.3474)(s + 0.05756)}{s(s + 4)}. \tag{4}
\]

The high-pass filter is determined by the frequency response of the acceleration sensor. It is given by

\[
\frac{(s + 0.0001)^2}{(s + 1)^2}. \tag{5}
\]

The low-pass filter for the two vibration isolation methods are different. The nonlinear behavior of the OOM controller would damage the response to high frequency sine vibrations. The low-pass filter for OOM control has a lower cutoff frequency to reduce the nonlinear behavior at high frequencies. It is designed as

\[
\frac{100^2}{(s + 100)^2}. \tag{6}
\]

The low-pass filter for VIM control is used to remove the resonance peak of the acceleration sensor. It is designed as

\[
\frac{3000^2}{(s + 3000)^2}. \tag{7}
\]

The position sensor resolution is assumed to be 0.5 \([\mu \text{m}]\). The acceleration sensor resolution is \(m0.1\) \([\text{mm/s}^2]\) and the noise is simulated by normally distributed random numbers. The sampling frequency is 10 \([\text{kHz}]\). There are two types of vibration signal used in simulation. The white noise vibration is a set of normally distributed random numbers generated by Simulink and most of the numbers are limited in the range \([-0.1, 0.1] \) [mm]. The sweep sine vibration is a sine signal with an amplitude of 0.1 [mm] and its frequency varies from 0.1 to 1100 [Hz] linearly with the time from 0 to 50 [s].

A. Theoretical Compliance

The payload weighs a few thousand kilograms, which provides a passive filter against the force disturbances. Therefore, the compliance would not be tested in the final application. Therefore, only the theoretical compliance of VIM control is plotted in Fig. 7. It shows that the compliance magnitude is reduced by increasing parameter \( m \). Since both virtually varying mass control methods have the same working principle, the effects of OOM control on the compliance is expected similar to that of VIM control except the high frequency nonlinear behaviors.

B. Time Domain Performance

Under white noise floor vibration, the seeking performance is simulated by tracking a low-pass filtered step signal from the origin to the boundary of the working range (1 [mm]). The simulation results for VIM control and OOM control are shown in Fig. 8(a) and Fig. 8(b), respectively. For both control methods, it can be concluded that the seeking performance have approximately zero steady-state error with varying \( m \). With increasing \( m \), the response tends to oscillate, which may subsequently result in overshoot. This problem is very likely to be solved by changing the pre-filter of the reference signal. Therefore, the proposed control methods with proper \( m \) would not damage the seeking performance.

Under the same white noise floor vibration, the closed-loop system responses to stepwise 1 [N] disturbance force for both control methods are plotted in Fig. 9(a) and Fig. 9(b). For both plots, all of the three response curves are almost coincide with each other. A conclusion can be drawn that the proposed control methods with proper \( m \) would not compromise the performance requirement on step response to constant disturbance force.

C. Transmissibility – White Noise Vibration

Under the same white noise floor vibration, the vibration isolation performances of the two control methods are tested
with different control parameters $m$. The reference is kept zero. 50 [s] of both the floor displacement and the absolute payload displacement were collected and they were used to calculated the transmissibility magnitude. The transmissibility magnitude is calculated by the Discrete Fourier Transform (DFT) of the absolute payload displacement over the DFT of the floor displacement. Since the floor displacement is simulated by the white noise signal generated by Matlab, its DFT amplitude has increasing deviations with frequencies, shown by the blue curves in Fig. 10. This deviation would lead to large noise at high frequencies in the magnitude plot of transmissibility. A smoothing algorithm, which averages the neighbor frequency amplitudes for chosen frequencies, is applied to both the DFT amplitude of floor displacement and the DFT amplitude of the absolute payload displacement. The smoothed DFT amplitude of the floor displacement is plotted as the curve marked by red stars in Fig. 10. Based on the smoothed DFT amplitude, the smoothed transmissibility magnitude with different control parameter $m$ for both control methods were calculated and plotted in Fig. 11 and Fig. 12. It shows that the transmissibility magnitude peak is shifted to lower frequency by the two virtually varying mass controllers and the larger $m$ leads to lower peak frequency of the transmissibility magnitude.

**D. Transmissibility – Sweep Sine Vibration**

The transmissibility is also evaluated by sweep sine floor vibration. The transmissibility magnitude is calculated by the DFT of the absolute payload displacement over the DFT of the floor displacement. Note that no smooth algorithm is applied. The transmissibility magnitude curves for both vibration isolation methods with different parameters $m$ are plotted in Fig. 13 and Fig. 14.
With the same parameter \( m \), VIM control and OOM control have almost the same transmissibility peak frequency. The magnitude peak in transmissibility for VIM control rises with larger \( m \) but it does not change much in OOM control. For both control methods, larger parameter \( m \) leads to lower frequency of the transmissibility magnitude peak, which is consistent with the transmissibility simulated with white noise vibration.

At high frequencies, the transmissibility for \( m \neq 0 \) shows nonlinear behaviors: magnitude oscillation in VIM control and magnitude flattening in OOM control. Magnitude oscillation means that the blue and green magnitude curves in Fig. 14 go up and down with continuously rising frequencies so that these curves looks thicker. This oscillation is more significant for larger \( m \) and higher frequency because the sensor noises and the nonlinear on-off actions become dominating the response when the excitation frequency is close to the sampling frequency. Magnitude flatting is the phenomenon that the slope of magnitude over frequency in Fig. 13 is different from the theoretical slope and the magnitude tends to be flattening or even forming a undesired peak. It is probably excited by the force compensation error and worsened by the VIM control action at excitation frequency which is close to the sampling frequency.

V. CONCLUSIONS

Two virtually varying mass control were proposed for an active suspension platform against the floor vibrations. They were separately applied to a stabilized feedback-linearized model for the vertical DOF of a candidate EMI design. Simulation shows that

- Seeking performance: The oscillation rises with \( m \) but it is still acceptable.
- Vibration isolation: The transmissibility tested with both white noise and sweep sine vibration is improved at low and mid frequency. The high-frequency nonlinear behavior is acceptable.
- Force disturbance rejection: The theoretical compliance is improved by VIM control. Both VIM and OOM control do not affect the response to constant force disturbance.

A conclusion can be drawn that the proposed vibration isolation methods are feasible to the presented application.

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