Research on Load Estimation for a PMLM Actuated Vibration Stage

Fan Liangzhi¹,², Cheng Qingsheng¹, Kang Huazhou¹, Yang Xiaofeng¹,³

¹School of Microelectronics, Fudan University, Shanghai, 200433, China
²School of Mechanical Science and Engineering, Wuhan Textile University, Wuhan Hubei, 430073, China
³Academy for Engineering & Technology, Fudan University, Shanghai, 200433, China

Email: 406023831@qq.com

Abstract. This paper puts forward a new load estimation method in time domain for PMLM servo loop tuning. This method takes advantage of the symmetry of cogging force distribution and identifies the friction force in constant velocity phase. Experiments have been carried out on a PMLM actuated vibration stage to validate the method. It’s discovered that it’s better to take an average of two estimations as the true value with movement range by 0.25 pitch’s difference so that cogging effect could be eliminated. Moreover, movement range is better to be greater than 1 pitch. Motor force constant should be calibrated according to its actual output.

1. Introduction

Most conventional vibration stages are hydraulically actuated in simulating vehicle road test. As state of the art vehicle performance goes higher, vibration stages are expected to provide better frequency response and broader bandwidth by a low cost. PMLM (Permanent Magnet Linear Motor) is a good choice for this purpose. However, expected high performance cannot be fulfilled unless the servo-loop has been correctly tuned, because PMLM has more obvious nonlinear features, such as cogging effect, friction by unbalanced attraction, etc.

As a preliminary step, load estimation should be done before latter tuning steps, e.g. the velocity loop, the position loop, etc. When rotary motor driving system is considered, frequency method is quite mature. Shats et al. [1] identified the parameters of an electro-mechanical system by frequency response. LI et al. [2] calibrated the mass of a nano precision motion stage by the use of power spectrum analysis. Nonlinear factors were eliminated from the model by constant movement assumption so that they could be modelled as certain frequency. However, such assumption isn’t true for PMLM because no continuous rotation exists. Nonlinear parameter identification receives intensive attention [3-8] in academic society. But such methods mainly model control systems in state space or employs complex mathematical treatment that is not appropriate for shop-floor processing.

2. Preliminaries

2.1. Fundamentals of Conventional Load Estimation

Conventional load estimation method is based on Fourier analysis. With the help of Bode figure, the slope of -40dB magnitude-frequency response line exactly represents the coefficient $m$. Static
Coulomb friction could be eliminated by certain constant velocity with a noise signal or a sweep frequency signal injected. At the same time, the cogging effect could be modelled as certain frequency according to certain speed. Such frequency could be easily found in Bode curve as some sort of pin point which should be smoothed by notch filter. Thus, the -40dB slope will not be affected.

But in studied PMLM vibration stage, the travel length is limited inside about 3 magnetic pitches, and as PMLM has an iron core, both cogging force and friction force are quite big. There’s no enough motion space for a continuous constant velocity, only back and forth movement is available in a limited short distance. As a result, conventional method is not appropriate for studied PMLM actuated vibration stage load estimation.

2.2. Closed loop servo control
It’s necessary to have a closed loop control. Bode figure represents the transmission function

\[ G(s) = \frac{x(s)}{F(s)} = \frac{1}{ms^2 + cs + k} \],

wherein \( m \) is the slope of -40dB line in Bode figure.

3. The Proposed Load Estimation Method
Conventional load estimation fails because nonlinear factors could not be modelled as a periodical signal. Alternative method could be employed in time domain, which utilize step response time series according to certain measurement compensation criteria, e.g. velocity, acceleration, jerk, etc.

3.1. The Principle of the New method
Back and forth movement based method is employed for workload estimation in the PMLM actuated vibration stage. During the forward movement, acceleration feed forward coefficients \( K_{af} \) will be set to zero. Regarding acc/dec along desired trajectory, which could be examined by optical encoder in real time, the workload could be represented by:

\[ a = f_K I_{friction} - I_{cogging} \]  

, where \( f_K \) is the motor force constant. Back and forth movement trajectory is shown in figure 1.

![Figure 1. Back and forth movement trajectory](image)

Feedback acc/dec(acceleration/deceleration) trajectory is compared with actual driving current output by power amplifier, thus

\[ m = \frac{K_f}{K_f} \int_0^t d(I_{friction} - I_{cogging}) + \int_0^t d(I_{friction} - I_{cogging}) \]

could be evaluated which represents the relationship between driving current and resulted acc/dec. This value will be substituted into acceleration feed forward block for back-move segment verification purpose, during which acc/dec impact trajectory should be minimized or eliminated as much as possible.

3.2. Friction force identification.
Friction is usually nonlinear and difficult to identify. However, by certain length of constant velocity movement, friction force could be modelled as corresponding current magnitude curve. Although the
friction force is nonlinear, by time average treatment \( I_{\text{friction}} = \frac{\int_{t_2}^{t_1} dI_{\text{const, velocity}}}{t_2 - t_1} \), it could be simplified as a constant to agree with the dynamic Coulomb model.

3.3. Cogging force elimination.

Cogging effect is a periodically alternative force along many magnetic pitches, and it is an un-linear variable inside one magnetic pitch. When the peak force just happens at the acc/dec position, the acc/dec curve would be distorted. As a result, corresponding driving current will be distorted too. It means that the selection of back and forth movement range affects the estimation precision.

The cogging force could be represented by Fourier series with base period 0.5pitch based on a single slot model \([9-10]\). Fortunately, as it is symmetric in sign inside one pitch, if two points are selected by 0.25pitch’s difference, the average of corresponding two forces should be zero in theory.

![Figure 2. Cogging force of a single slot model](image)

By such experiment arrangement, the cogging effect could be theoretically eliminated and the workload should be:

\[
m = \frac{K_f}{2} \left[ \frac{\int_{t_2}^{t_1} d(I - I_{\text{friction}})}{\int_{t_2}^{t_1} da} + \frac{\int_{t_2}^{t_1} d(I - I_{\text{friction}})}{\int_{t_2}^{t_1} da} \right]_{\text{Start}}^{\text{Start+0.25pitch}}
\]

(2)

The back movement mainly serves the purpose of verification, during which feed forward coefficient is set so that acc/dec curve could be minimized or eliminated. Workload estimation could be worked out by the average of two estimations with 0.25pitch difference in movement range according to formula (2). If the friction force is small enough compared with acc/dec force, it is possible just neglect it. The higher the acc/dec force, the less the friction force affects.

![Figure 3. Actual driving current vs position/velocity trajectory](image)
4. Experiment on Load Estimation

4.1. Experimental Setup

Experiment setup is illustrated in figure 4. Although there are two power amplifiers (Akribis ASD240-0418S1J1) in studied PMLM actuated vibration stage, only one actuator is able to work during load estimation process because of synchronization difficulties. Each actuator has 2 PMLMs (ACM4-W-S2, Akribis Ltd.Co.) sandwiched back to back so that it may achieve dynamic performance like 10G (about 100m/s²) maximum acc/dec. Sliders on both sides are connected rigidly. Each ACM4 is rated to 2510N, and overall output is 10kN. The moving set is designed to be 90kG, including 4 sliders, holding framework, connection rod, and locking nuts, etc.

![Dual PMLMs actuated vibration stage configuration](image)

The left actuator is driven by one amplifier, and the right one is driven by another. Overall system is a classic redundant driven system, especially from mechanical viewpoint. Moreover, although the rated value of each PMLM is known, the actual electric driving parameters, e.g. the force constant, would be different from its rated values because of the redundancy configuration.

Unfortunately, it should be addressed that some experiment setup defects exist: un-calibrated force constant, non-rigidly placed or un-fixed vibration stage base, etc. However, such defects do not affect the effectiveness of proposed principle in load estimation experiments below, but the estimated load value would be severely affected by such defects, which should be noticed in future applications.

4.2. Result

Restricted by power amplifier synchronization, the load estimations have to be done by only one actuator on either side in each experiment. Altogether, 40 rounds of estimation have been made with respect to the movement range that varies from 0.2pitch to 3pitches by 0.1pitch interval. Each side has 10 tests from the same start point, and another 10 tests started by approximately 90 mm offset.

![Actuator on the left side (1.2Pitch Actual Movement Range)](image)

Figure 5. Actuator on the left side (1.2Pitch Actual Movement Range)
4.2.1. **Left side actuator.** Figure 5 includes all 20 tests on the left actuator. Actual movement range is about 1.2 pitches. Average estimation curve has fallings by approximately 0.5 pitch interval with respect to movement range both in 5(a) and 5(b). Moreover, it is discovered that different start point also affects the load estimation results. It is interesting that experiment time also affects the estimated loads. Data in midnight shows much more consistency than those obtained in the morning or afternoon. Another obvious fact is that all the estimated loads are quite different from the true value 90kg.

4.2.2. **Right side actuator.** Figure 6 includes another 20 tests on the right actuator. Corresponding actual movement range is about 2.5 pitches. Estimated load curves in figure 6 display similar oscillation features, such as the fallings by 0.5 pitch interval in range, the difference of estimated load from true value 90kg, the sensitivity of movement start position, etc. However, as the movement range is larger, the estimated load curve tends to be smoother, especially in 6(b).

5. **Result Analysis**

5.1. **Rigidity of the Vibration stage base**

As described above, there is a confusing phenomenon that is about the big difference between 4 rounds of experiments. The average estimation varies from a minimum 80kg to a maximum 230kg. Regarding section 3.3, motor force constant is only able to enlarge or compress estimated load by a constant coefficient. No data floating would be introduced by the force constant $K_F$. Since 4 sets of data curves all display randomness dealing with experiment time, the driving current or measured acc/dec must have been affected by something dealing with time. The only explanation is that the support of the vibration stage is not rigidly connected to the ground. As the ground vibrates more during daytime than it does in the middle night, such difference should be reasonable, shown in figure 5(a) and 5(b). Besides, see figure 6, the base framework support is made of wood. During the back and forth movement, the framework would have been ejected backward and measured acc/dec would have been reduced although driving current was quite large. As a result, estimated loads would be greater and volatile with respect to different experiment time. The wooden support of the vibration stage is not able to maintain a constant physical Mass-Damping-Spring model.

5.2. **Movement range selection**

Experiment data shows that cogging effect has relationship with estimated load by about 0.5 pitch. Nominally, cogging force is a half pitch periodical oscillating force, and the experiment data curves do show an oscillation regarding approximately 0.5 pitch movement difference.

Back and forth movement range is better set to be more than 1 pitch so that the movement trajectory is sure to have a constant velocity part. What’s more, it is not able to utilize the cogging effect symmetry unless the movement range is less than 1 pitch.
5.3. Start point selection
The end part of rail is usually worse in parallel or friction condition. It is better to locate the start point in the middle of the rail guidance so that smooth and consistent movement could be achieved.

5.4. Force Constant
$K_f$ is proportional to the estimated load. In the experiments, the force constant was set according to operation manual. Larger current is needed for the same acc/dec, which leads to a bigger estimation according to formula (2). Force constant calibration should be carried out in future.

6. Conclusion
For the purpose of a high performance PMLM driven vibration stage, this paper puts forward a load estimation method in time domain. Factors that matter include movement range selection, movement start point selection, PMLM motor force constant, and the rigidity of the vibration stage placement on the ground in addition to its own rigidity. Movement range selection is better to be more than 1 pitch, and it’s better to take an average of two estimations as the true value by movement range in 0.25 pitch’s difference so that cogging effect could be eliminated. PMLM force constant should be calibrated; otherwise, resulted estimation would deviate a lot. The rigidity is also important that may lead to confusing phenomenon like time dependent volatile results. For accurate load estimation, the vibration stage should be rigid in structure and fixed rigidly to the ground. Future research will follow this guideline for much more accurate and consistent workload estimations.

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