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A NOVEL APPROACH TO DYNAMIC FORCE CONTROL

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This paper presents a new approach to dynamic force control of mechanical systems, in particular frame structures, over frequency ranges spanning their resonant frequencies. This is achieved using series elasticity and displacement compensation. Most mechanical actuation systems such as hydraulic and electromagnetic actuators are inherently velocity sources, that is, an electrical signal regulates their velocity response. Such systems are therefore by nature high-impedance (mechanically stiff) systems. In contrast for force control, a force source is required. Such a system logically would have to be a low-impedance (mechanically compliant) system. This is achieved by intentionally introducing a flexible mechanism between the actuator and the structure to be excited. In addition, to obtain force control over frequencies spanning the structure's resonant frequency, a displacement compensation feedback loop is needed. The actuator itself operates in closed-loop displacement control. The theoretical motivation as well as the laboratory implementation of the above approach is discussed along with experimental results. Having achieved a means of dynamic force control, it can be applied to various experimental seismic simulation techniques such as the Effective Force Method and the Real-time Dynamic Hybrid Testing Method. These methods are discussed in a companion paper.

Keywords: *Force control, Dynamic hybrid testing*

1 Background

The implementation of advances seismic testing techniques such as the effective force method [1] and real-time dynamic hybrid testing [2] require the implementation of dynamic force control in the hydraulic actuators. This control is sensitive to the acceleration and force measurements, to the modeling of the compressibility of fluid, to the nonlinearities of the servo control system (servovalves) and other stiffness issues as indicated by [3] and [1].

2 Issues in dynamic force control

By its physical nature, a hydraulic actuator is a rate-type device or velocity source, i.e., a given controlled flow rate into the actuator results in a certain velocity. Moreover, hydraulic actuators are typically designed for good position control, i.e., to move heavy loads quickly and accurately. They are therefore by construction high impedance (mechanically stiff) systems ([4]). In contrast for force control, a force source is required. Such a system logically would have to be a low-impedance (mechanically compliant) system. Thus force control using hydraulic actuators is an inherently difficult problem. It was recognized by [5] that closed-loop control with force feedback is ineffective without velocity feedforward or full state feedback. This is further supported by [3] and [1] in their work on the effective force method. **Figure 1** shows the force control with velocity compensation proposed by [3]. Rearranging terms by multiplying by A and dividing by s, the control loop in **Figure 2** is obtained. It can be seen that the oil behaves as a spring providing the flexibility required for force control. Relative deformation occurs across this spring and force is applied through it.

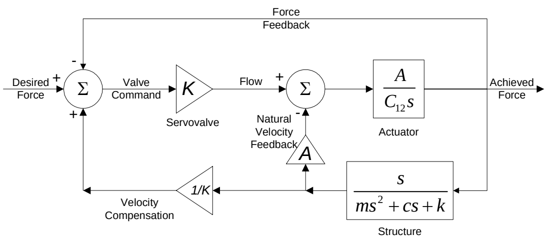

Figure 1: Force control of [3].

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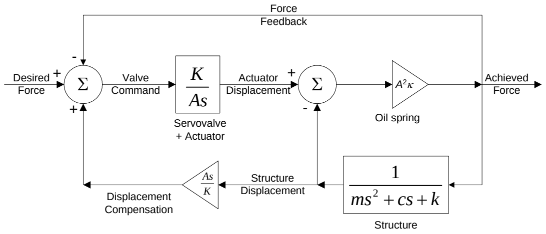


Figure 2: Force control of [3] rearranged.

Actuators designed for position control have stiff oil columns, making force control very sensitive to control parameters and often leading to instabilities. Moreover friction, stick-slip, breakaway forces on seals, backlash etc. cause force noise, making force a difficult variable to control. Several strategies have been introduced to work around this problem. For instance the dual compensation scheme of [6] uses a primary displacement feedback loop with force as a secondary tracking variable. It also supports features such as acceleration compensation to compensate for some of the effects that distort the control force. In robotics, and impedance control strategy has been employed wherein the force-displacement relationship is controlled at the actuator interface ([7] and [8]). [9] has used the idea of series elastic actuators where a flexible mechanism is intentionally introduced between the actuator and the point of application of force, along with force feedback.

3 Proposed solution

Motivated by these observations and by the fact that causality requires a flexible component in order to apply a force ([10]), in the force control scheme described here, a spring is introduced between the actuator and the structure as shown in **Figure 3**. Notice that the scheme is similar to that of **Figure 2** except that (1) the intentionally introduced series spring, K_{LC}, assumes the role of the oil spring and (2) there is no force feedback loop.

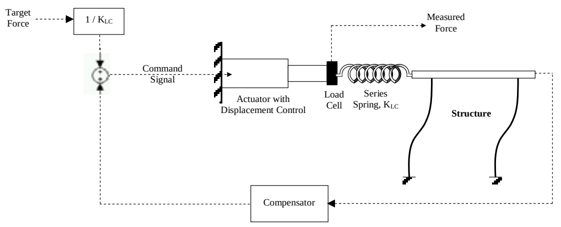


Figure 3: Proposed force control scheme

In the schemes of both **Figure 2** and **Figure 3**, the actuator behaves as a displacement device. The tuning of the hydraulic actuator in displacement control can be done relatively easily, very precisely and independent of the structure and leads to a well behaved closed-loop transfer function with good force disturbance rejection. Hence the actuator in the control scheme of **Figure 3** is operated in closed-loop displacement control with a PIDF controller. The resulting block diagram is shown in **Figure 4**.

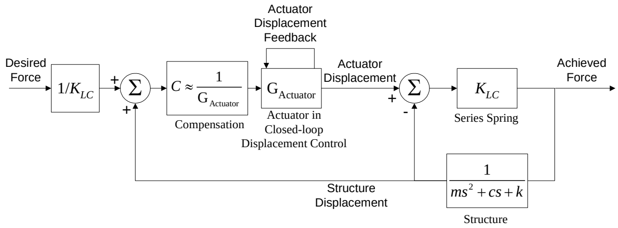


Figure 4: Block diagram of proposed force control

Although the system as a whole controls force input, *internally the actuator operates in closed-loop displacement control*. Hence, there is no need for an additional force feedback loop to ensure stability. Besides, this would prevent the actuator responding to spurious force errors caused by the sources listed above. The force transfer function is given by:

$$\frac{\text{Achieved Force}}{\text{Desired Force}} = \text{CG}\frac{ms^2 + cs + k}{ms^2 + cs + k + K_{LC}\left(1-\text{CG}\right)} \qquad (1)$$

In the ideal case where C = 1/G, the transfer function has the value of one. The advantages of using the series spring are also now apparent: (1) the actuator can be well tuned and operated in displacement control, (2) it provides for one more parameter than can be altered in the control design (the oil stiffness cannot be) and (3) the term K_{LC}(1-CG) in the transfer function indicates that the smaller the value of KLC the less sensitive is the transfer function to deviations of C from 1/G. The following paragraphs present results from applying the above control procedure.

4 Experimental results

Experiments were performed using a small-scale test setup and are described below.The test setup is shown in **Figure 6**. An MTS 252.22 two-stage servo-valve and an MTS actuator were used. The controller used was the MTS FlexTest GT. The structure was designed to have a resonant frequency of 3 Hz. The force control system consisting of the compensator C and the displacement feedback was implemented as a cascade controller around the MTS controller using *Simulink* and *xPC Target* [11].

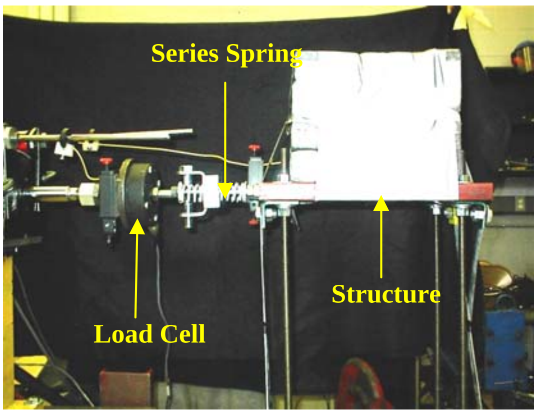


Figure 5: (a) Small scale force control setup

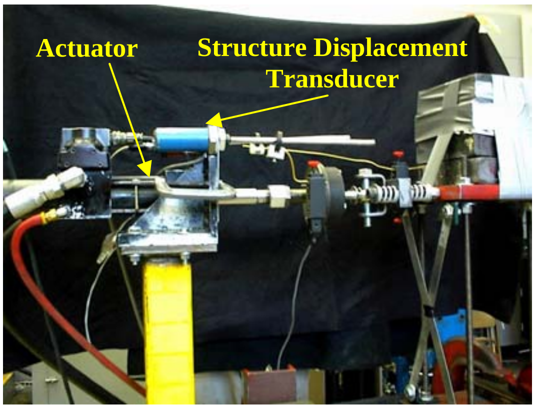
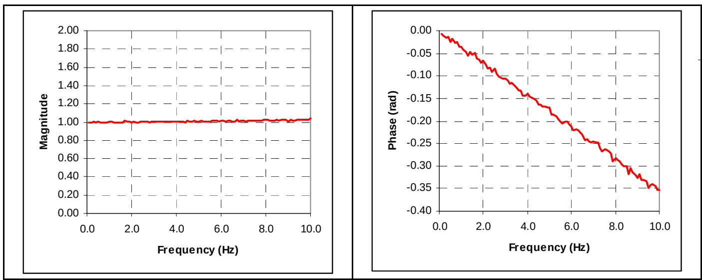


Figure 6: (b) Small scale actuator detail

The actuator was tuned first in displacement control and the closed-loop displacement transfer function was measured. This is shown in **Figure 7**. For the frequency range of interest, it is seen that the actuator can be modeled as a pure time-delay system with a delay of 5.6 ms.



(a) Magnitude response | (b) Phase response

Figure 7: Frequency response function of the actuator in displacement control

The resulting force transfer function magnitude of the system in **Figure 3** is shown in **Figure 8**. There is a significant drop in magnitude at the resonant frequency, 3 Hz, of the structure. Also shown in the figure is the result from a numerical simulation performed by modeling the actuator as a pure time-delay of the measured value. The correspondence between the results demonstrates that time-delay is indeed the cause for the drop in the frequency response magnitude. This is because near this frequency, there is a large gradient in the phase response of the structure and its response is therefore very sensitive in the phase lag caused by time-delay in the feedback. The existence of time-delay implies that any control action can be applied only a certain amount of time after it has been computed.

5 Predictive compensation

To correct for this therefore, the control action needs to be *predicted*. A Smith-type predictor [12] is used here for this purpose as shown in **Figure 9**. Another approach is to use a lead-lag compensator to produce a phase-lead at the resonant frequency of the structure (see for example [1]). The resulting magnitude of the input force-output force frequency response function is shown in **Figure 10**. The deviation from unity is a result of a first order approximation of the exponential term in the predictor during its digital implementation. A better digital redesign of the Smith predictor is the subject of current work.

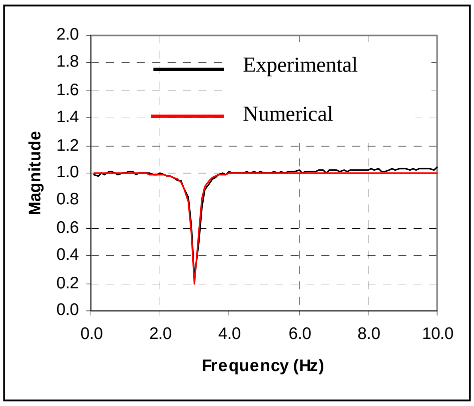


Figure 8. Magnitude of the input force - output force frequency response function

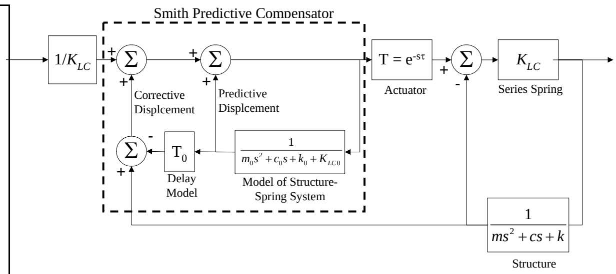


Figure 9. Smith predictive time-delay compensator

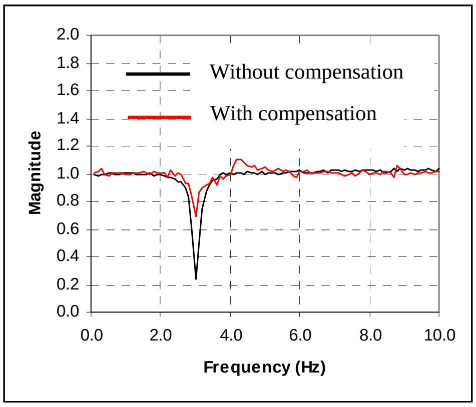


Figure 10. Magnitude of the input force - output force frequency response function

6 Summary and conclusions

A novel approach to the challenging problem of dynamic force control has been presented using series elasticity and displacement feedback compensation. Due to the phase sensitivity of response near resonance, time-delay is found to cause a significant drop in the magnitude of the output force near this frequency. A Smith type predictor is proposed for time-delay compensation and is found to result in significantly better performance. A better digital redesign of the Smith predictor is the subject of current work.

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