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# Theoretical and experimental investigations of vibration waveforms excited by an electro-hydraulic type exciter for fatigue with a two-dimensional rotary valve

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ABSTRACT

Conventional electro-hydraulic excitation is usually controlled by a servo valve employing a sliding spool type construction. However, the comparatively slow response of the servo valve will greatly limit the system's high-frequency performance. Therefore, a rotary valve with the rotary motion of the spool as a new excitation mechanism is proposed to obtain the desired excitation, especially a high-frequency excitation wave for fatigue. An electro-hydraulic exciter using a combination of a three-way two-dimensional rotary valve (2D rotary valve) and an unequal area piston is taken as an example. Analysis of the vibration output to a typical wave input yields an analytic solution of the vibration waveform excited by this electro-hydraulic system. The mathematical formulation of the harmonics is also derived. Additionally, an electro-hydraulic excitation test-bed is built to acquire an experimental excitation wave. Consequently, the analysis of the excitation waveform in an approximate analytical and experimental method is used to verify access to high-frequency excitation, even resonance excitation.



## 1. Introduction

Electro-hydraulic excitation created by hydraulic power focuses attention on stability, bandwidth and wave distortion [1–5]. Therefore, it is always necessary to analyze and improve the high-frequency properties of an electro-hydraulic excitation system. Because the electro-hydraulic servo valve as an important element has a direct influence on the performance and characteristics of the whole system, to obtain satisfactory performance, a basic method is to develop a fast-response servo valve. For this reason, many corporations collaborate with universities or research institutes to design the structure of the control valve and its key components [6–8]. A hydraulically driven shaking table for shock absorbers has been developed by the Beijing Institute of Technology [9]. In this system, a type of D761 series servo valve of America Moog Co. is adopted to control the cylinder. This servo valve frequency at a −90° phase shift is larger than twice the natural frequency of the system, so it is also found in some vibrating test beds for waveform replication [10]. A special three-stage servo valve with all stages being of the spool type is produced by the Nanjing Engineering Research Center [11], and has a better dynamic response than a flapper valve used as the first stage in a two-stage electro-hydraulic servo valve. However, the bandwidth is just 200 Hz, resulting from the flow characteristics of the servo valve. It has been used by the Harbin Institute of Technology in a servo control system of a 6-DOF electro-hydraulic vibration table to simulate the vibration environment of flight space. MTS Co. has designed a two-stage servo valve with a voice coil pilot stage for use as a first-stage valve instead of a flapper-nozzle, while the second-stage valve is universally of the spool type [12]. This servo valve has the characteristics of a pressure feedback servo valve at higher frequencies. Therefore, it is used to control a high-frequency electro-hydraulic excitation system specifically for an elastomeric material test or crack growth test at 1000 Hz or above, which has been integrated at Michigan Technological University.

Electro-hydraulic excitation controlled by the linear motion of a spool is most common. This linear displacement of a spool is created by supplying a cyclically varied current to the electromechanical converter of the servo valve, which in turn varies the alternative flow rate into or out of the cylinder or motor, and subsequently, the piston or rotor is excited with respect to the desired vibrating wave form [13]. This results in the system performance largely depending on the comparatively slow response of the servo

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valve, so it is not possible in practice to obtain a high-frequency excitation wave or complicated excitation wave shapes. This factor limits the high performance improvement of the electro-hydraulic excitation type.

Compared to a servo valve with a multi-stage structure or an improved electrical-to-mechanical transformer, the new structure of the other control valve is always developed instead of the servo valve in an electro-hydraulic excitation system. There have been some studies on the design feature and work performance of the rotary valve with the rotary motion of the spool [14–16]. A novel fluid-driven PWM on/off valve design is presented, which is based on a unidirectional rotary spool [17,18]. Such a design reduces the valve actuation power from a cubic dependence on the PWM frequency to a square dependence by eliminating motion reversals during transition. In addition, a rotary valve has replaced the servo valve to be used in a special excitation mechanism, in which the hydraulic excitation source is caused by water hammer fluctuation. The vibration frequency of this system can be adjusted to 90 Hz [19], which is usually found in vibrating separation or compaction engineering applications characterized by large output forces. A similar scheme is also adopted by Southeast University and applied to marble block vibratory compaction [20]. However, this rotary valve is not only for improving the working frequency of the system, but also for supplying a sinusoidal pressure drop across the load by designing the valve orifice geometry.

Although the rotary valve also connects the electronic and hydraulic mechanical portions of a system that is similar to a servo valve, the rotary valve as a new excitation mechanism is more attractive in a high-frequency electro-hydraulic excitation system [21]. Therefore, a rotary valve-controlled electro-hydraulic exciter is designed and developed in this paper. A new type of two-dimensional rotary valve (called 2D valve [22,23]) is redesigned as a power element of a three-way valve-piston combination in an electro-hydraulic exciter. The single spool of this 2D valve has both rotary and linear motions. The two-dimensional motions are controlled independently which in turn produce cyclic flow or pressure to a hydraulic actuator to output a vibration. Because the rotary spool is lubricated with oil, it is very easy to achieve high vibrational frequencies by increasing the rotary speed of the spool.

In this paper, a 2D rotary valve-controlled high-frequency electro-hydraulic exciter is developed for the fatigue test of materials, and approximate analytical and experimental methods of the excitation waveform are proposed. It remains particularly significant for the theory, design, and application of a medium- or high-frequency electro-hydraulic excitation system, especially in the area of excitation type controlled by the rotary motion of the spool. The rest of the paper is organized as follows. Section 2 describes the system principle, including a load analysis, the basic equations and a math model. In Section 3, the excited waveform to typical wave input is derived using an analytical method, and its characteristics are analyzed. In Section 4, a set of relational models for the orifice area, the geometry of the 2D rotary and the vibration wave form are established with mathematical formulations. The expression of the harmonics is also derived. Section 5 presents the experimental results of the 2D rotary valve controlled electro-hydraulic exciter. Lastly, conclusions are made to summarize the main work of this paper.

## 2. Hydraulic power element of the electro-hydraulic exciter

### 2.1. Valve controlled piston

The vibration waveform of the electro-hydraulic exciter displays more or less distortion because of hydraulic resonance, and the distortion becomes more significant at some special frequencies [24,25]. As the input frequency approaches the hydraulic resonant frequency, the output essentially becomes the hydraulic resonance. Therefore, the combination of the three-way valve and unequal piston areas applies to the electro-hydraulic exciter as a power element. Such a combination contributes an adjustive hydraulic natural frequency so that the working frequency of the exciter or its harmonics could avoid the resonant frequency. Consequently the influence of the hydraulic resonance for output is to be attenuated.

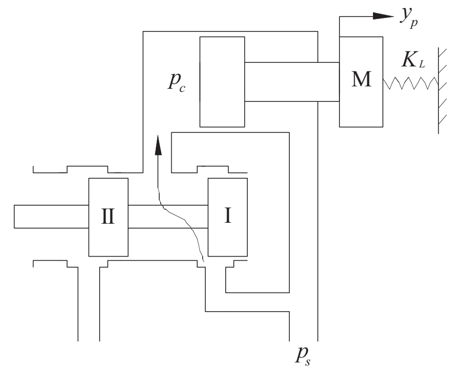


**Fig. 1.** Hydraulic power elements of the electro-hydraulic exciter.

Spring loads are much more common with piston type actuators because of their limited travel [26]. This case is applicable to the electro-hydraulic exciter system in the fatigue test of materials or a structure strength test. At frequencies far below the natural frequency, the mass and viscous forces are small and may be neglected entirely. The load of electro-hydraulic exciter is determined just by the spring constant. A mode of this case is illustrated in Fig. 1.

However, actual loads including mass or viscous force are so complex that mathematical expressions that describe this system are quite involved. Although the development of digital computer programs could solve the governing equations [27], they only apply to a particular system because of the given parameters. Hence, a describing function based on the dominance of the spring load force would be in order.

With no spring load on the piston, a steady-state control pressure acting on the head area is

$$p_{c0} = \frac{p_s}{2} \tag{1}$$

where $p_{c0}$ is the steady-state control pressure, and $p_s$ is system pressure.

This design relation allows the control pressure to rise or fall and to provide equal acceleration and deceleration capability. Eq. (1) is satisfied by making the head area twice the rod area, that is,

$$A_h = 2A_r \tag{2}$$

where $A_h$ is the head side area of piston, and $A_r$ is rod side area of piston.

Assuming a constant supply pressure, the supply and return ports connect to the head chamber. When the supply port (valve orifice 1) is open or active, pressurized oil produces a hydrostatic force to drive the piston moving in the right-hand direction. Assuming that the load acts as a tensile spring at the initial position, the restoring force is in the same direction as the hydrostatic force. Thus, in this process, the spring load is essentially an overrunning load. Under the action of this applied load, the piston will be accelerated, as illustrated in Fig. 2(a). As the piston is continuously moving, the direction of the hydrostatic remains the same as the proceeding process, but the elastic load turns into a compressed spring, which consumes part of the hydrostatic force. Therefore, the spring load becomes a normal load, as shown in Fig. 2(b).