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CONTROL OF HYDRAULIC ACTUATED FATIGUE TESTING MACHINES - A REVIEW

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Abstract.
This paper presents an overview of the literature on hydraulic actuated fatigue testing machines. A complete overview and classification of control schemes for fatigue testing machines is not a simple task, and there are quite a few academic publications on the subject. This paper reviews some contributions from the tribology, modeling and control techniques found in servo-hydraulic control literature. In particular, a set of models and tools for servo-hydraulic fatigue testing machine design and control is provided. To this end, this paper starts by describing dynamic models for testing machines. Then, a discussion of the relevant analysis techniques provided followed by a review of hydraulic control methods reported in the literature.

Keywords: *control hydraulic servo-systems, fatigue testing machines, overview of hydraulic actuated fatigue testing*

1. INTRODUCTION

Industries such as Nuclear, Aerospace, Petroleum, among many others have tirelessly tested materials trying to obtain: (1) quality assurance; (2) assurance of safety; (3) assurance of economic merits; (4) development of innovative products and materials, and so on. The interest in material testing started during the early 1800's when railroads mechanisms and large trains components made of ductile steel started to fail after a relatively short period of usage (Callister, 2001). The material used was prone to develop small cracks which would rapidly propagate due to tension concentration loads. Rankine (1843) suggested that the problem was due to material crystallization and, thus it became fragile as a result of tension fluctuations*. Up until then, when considering the overall machine design, the problem was not treat as a whole and the dynamic loads and their consequences were neglected. It was only in 1858 that August Wöhler, a specialist in material and fatigue published a scientific paper reporting *fatigue failure* in testing axles. Wöhler tested specimens subjected to bending and torsion and thus demonstrating the existence of an endurance limit.

Subsequently, researchers and companies wanted to ensure that fatigue failure would not occur so frequently and also to have means to be able to predict when a failure would appear. In the mean time, there was no testing equipment available that was accurate enough to meet their requirements and did not give the necessary minimum information to perform even the basic fatigue tests (Pelloux and Brooks, 1964). At the time, the American Society for Testing and Materials International (ASTM) developed procedures and standards for fatigue and fracture testing (ASTM International, 2008, E-647), (ASTM International, 2011a, E-647), (ASTM International, 2011b, E-648).

In the 1960's, many manufacturing companies of hydraulic servo-systems and electronic devices were being consolidated. Companies such as Instron Engineering Corporation, MTS Systems Corporation and Shimadzu, which are the leading companies in the sector, started the development of material testing instruments and machines. Since automated testing is possibly, researchers described their techniques to establish conformance to the various ASTM standards to fatigue and fracture testing (Miller *et al.*, 1985; Vecchio *et al.*, 1985; Sooley and Hoeppner, 1985; Kondo and Endo, 1985; der Sluys and Futato, 1985; Joyce and Sutton, 1985; Cullen, 1985). ASTM states that:

> "ASTM's fatigue and fracture standards provide the appropriate procedures for carrying out fatigue, fracture, and other related tests on specified materials. These tests are conducted to examine and evaluate the behavior, susceptibility, and extent of resistance of certain materials to sharp-notch tension, tear, axial fatigue, strain-controlled fatigue, surface crack tension, creep crack, and residual strain."
>
> *ASTM International, 2013.*

Structures may be subjected to all kinds of dynamic and fluctuating stresses, such as axial (tension-compression), flexural (bending) and torsional (twisting). After a lengthy period of repeated stress or string cycling, a failure may occur. In general, when applying a fatigue test, the goal is to determine how many load cycles a specimen can sustain or the failure load level for a given number of cycles (Callister, 2001). The results of fatigue test may vary from one material to

*First academic publication on fatigue fracture of car axles.

another, so the control system must quickly adapt in order to meet standard requirements. For example, the applied force or torque should never surpass the limits stated in the technical standards (no overshoot).

Fatigue and fracture test laboratories are steadily evolving to meet the demands of standards and providing significant improvements for all industries (Miller *et al.*, 1985; Vecchio *et al.*, 1985; Sooley and Hoeppner, 1985). In previously years, hydraulic servo-system test were controlled by analog function generators, mechanical relay counters, digital voltmeters and X-Y recorders. A considerable savings in time and money can be made when standard fatigue and fracture tests are conducted under a well designed conditions combining control and automated systems (Miller *et al.*, 1985; Vecchio *et al.*, 1985). A great effort has been made to develop control techniques and computational tools that could respond to the rapidly requirement specifications.

Hydraulic systems have many advantages over other types of energy transmission (eg. electrical motors, pneumatics motors, etc.) which make them a natural choice for fracture and fatigue testing systems (Jelali and Kroll, 2003, pp. 2). One main reason to employ hydraulic servo-systems, rather than other technologies, is due to components compact size assemblies with long component life spans, capability of producing high force/torque, and high stiffness under heavy load. Additionally, hydraulic actuators have a high transient speed response for fast starts, stops and speed reversals, which is essential for typical fatigue test where both force and torque are applied at great frequencies (Alva, 2008; Jelali and Kroll, 2003). However, the presence of nonlinearities make hydraulic energy transmission limited. Actuation subsystem and and loading subsystem can be considered mainly the two sources of the nonlinearities (Vossoughi and Donath, 1995). Moreover, oil contamination makes hydraulic control to fail and fire and explosion hazards exists if the hydraulic system is close to a source of ignition (Merritt, 1967; Will and Gebhardt, 2011; Jelali and Kroll, 2003).

Uncertainties parameters in hydraulic actuated fatigue testing machine (HTM) reduce drastically controller performance to maintain load accuracy (Sánchez *et al.*, 2012). In material testing applications, simultaneous selection of many parameters is required therefore manual controller tuning is almost invariably (Lee and Srinivasan, 1990). Most usual approaches are based on classical control with compensation, adaptive techniques, variable structure methodologies and self-tuning control (Bessa *et al.*, 2010; Ahn and Truong, 2009; Guan and Pan, 2008; Serrano, 2007; Sohl and Bobrow, 1999; Lee and Srinivasan, 1990; Vossoughi and Donath, 1995).

This paper presents an overview of the available literature regarding HTMs. We start with a review of fatigue testing techniques and requirements. Then we discuss about control theory applied to hydraulic control. Emphasis is given to the review of control techniques that attempted to overcome common nonlinearities, such as pressure/flow nonlinear characteristics, asymmetrical actuator and transmission.

2. FATIGUE TESTS AND HYDRAULICS MACHINES

Materials tend to fail after being exposed to a repetitive or intermittent intense stress. Fatigue is a condition whereby the material cracks or fails as result of repeated (cyclic) stress. This is an undesirable event that may cause financial losses and, above all, put human lives in danger. The causes can be due to inadequate component design, or no prior knowledge of crack growth rate propagation of a material(Callister, 2001). Materials can be tested to determine fatigue crack growth rates through experimental investigation. Several different test procedures can be implemented on fatigue testing machines. Thus, fatigue testing gives better understanding of in-service life of material. This type of mechanical testing can be performed by different kinds of machines. The loads/stress can be produced by: (1) mechanical deflection; (2) centrifugal forces; (3) electromagnetic forces; (4) hydraulic forces; (5) pneumatic forces and so on (Weibull, 1961) Different types of load/stress profiles can be applied to perform a fatigue test starting with simple sinusoidal load cycles to complex service life load.

The use of computers to control testing machines allows the complete automation of the process. As a result, the time required to perform standard fatigue tests and fatigue crack propagation methods to obtain material characteristics is well reduced (ASTM International, 2008; Vecchio *et al.*, 1985). The data acquired by the computer can be processed in accordance with ASTM International (2011b, E-648) (ASTM International, 2008, E-647) and ISO (2002, ISO 12108).

General purposed machines can be classified according to dynamical and fluctuating stresses. The classes can be categorized as (1) axial loading (tension-compression); (2) repeated bending; (3) rotating bending or (4) torsional (twisting) (Weibull, 1961). Each type of stress requires a specific machine in order to perform the tests (Hosford, 2010; Callister, 2001). The testing machines for special purposes are a modification to general purpose to perform test such as: (1) high frequency; (2) elevated or low temperature; (3) cyclic thermal stresses; (4) corroding environments; (5) fretting corrosion; (6) multi-stress level tests; (7) contact stresses; (8) repeated impact; and (9) combined creep and fatigue tests (Weibull, 1961). Before the test is performed, the choice of the corrected machine and parameter is necessary. An axial HTM is the object of our study. Denoting the maximum and the minimum forces[†] applied in a cycle by F_{max} and F_{min}, respectively, in general the parameters used to specify how a test should be carried out are the following (ASTM International, 2011a, ASTM E-647), (ISO, 2002, ISO 12108): (1) **force range**, $\Delta F = F_{max} - F_{min}$, (2) **force ratio**, $R = (F_{min}/F_{max})$, (3) **waveform profile**, (4) **frequency**, (in Hz), f and (5) **test mode**: constant load (ΔP) or constant stress-intensity factor

[†]The word force, in some definition, can be replaced by stress (Weibull, 1961).

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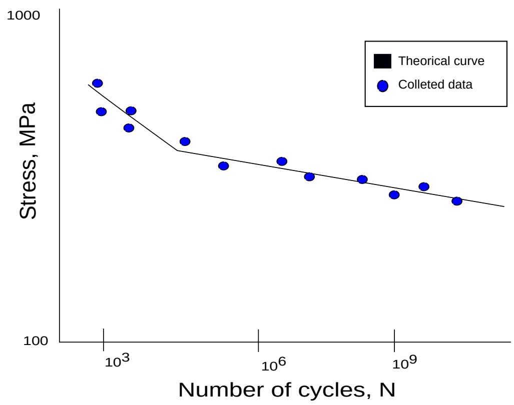
range (ΔK) or decreasing ΔK.

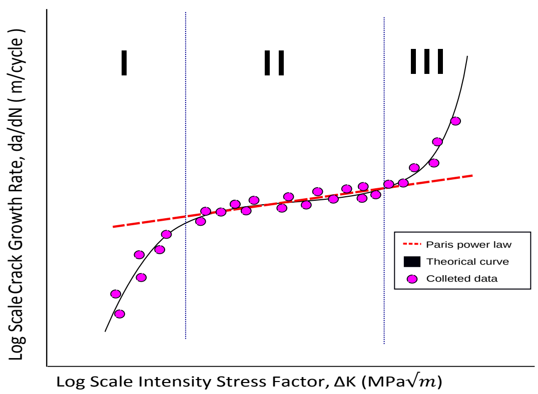
According to Miller *et al.* (1985), an automated testing machine must fulfill some additional requirements : (1) implement a user-friendly interfaces (fatigue testing Human-Machine Interface (HMI)) which allows the programming of the entire procedure; (2) control the independent variables in a given test; (3) monitor and register all test variables for out-of-range or out-of-specification conditions; (4) acquire and store all required raw test data; (5) detect the end-of-test conditions and stop the test; (6) limit raw data register during the test; (7) outcome reduced data in a comprehensible format (graphs, tables, etc.); (8) store test result in a database for future reference; and (9) provide database management tools to manipulate the stored results. Note that, the control design plays an important role in the complete system, mainly due to the fact that each test depends on different specimens and requirements(Lee and Srinivasan, 1990).

The forces applied to the sample during a fatigue test must be precisely controlled. According to ASTM International (2008, ASTM E-647), the maximum deviation tolerated is $\pm 2\%$ of F_{max} and $\Delta F/\Delta K$. Larger deviations invalidate the test since they cause a stress concentration, which may modify the plastic regime for the material and change the testing results.

Figure 1 shows a typical result obtained from a laboratory experiment. In this case, an embedded functionality in the user interface (HMI) provides the mechanism for data interpolation and to render the final graphical result. Figure 1b shows the distinct regimes of crack growth. Hosford (2010) asserts that failure occurs when $K_I = K_{IC}$[‡]. After this, the crack growth rate accelerates rapidly. Generally, fatigue failures can be divided into three distinct stages as illustrated in Fig. 1b: (1) crack initiation(short cracks); (2) crack propagation (long cracks)[§], and (3) fast fracture (final fracture). Therefore, the fatigue testing system must monitor this variable as well. The most common fatigue test is the one referred to as *Constant Amplitude*. Since the stress-intensity factor (K) is a function of specimen's geometry, the stress applied and the crack length. As the test evolves, ΔK increases accordingly to crack length. To illustrate the result in such a test, Figure 1a shows a plot of the stress level (S) against the number of cycles. Usually, these tests are heavily time consuming. Depending on the cycle frequency, they can last for months. A fatigue test conducted to 10^9 cycles at a considerable high frequency 50 Hz, for example, done over a traditional servo-hydraulic testing machine would take over 231 days to finish (Morgan and Milligan, 1997).



(a) $S \times N$ curve or Wohler curve.



(b) Schematic of crack growth rates and the main regions

Figure 1: Schematic graphs. Adapted from Hosford (2010, Fig. 17.6, pp. 278 and Fig. 17.20, pp. 290) and (ASTM International, 2008, ASTM E-647)

In industry, there are three main types of actuation: (1) servo-hydraulic, (2) electromagnetic, and (3) shakers. HTMs are widely used for this type of testing. Their main advantages are the broad load range and high precision for a wide frequency range. The disadvantages are the costs of installation, maintenance and nonlinearities present in each subsystem.

Hydraulic actuated fatigue testing machine

Over the years, more advanced hydraulic servo-systems and computers have been developed and applied to HTM's aiming to improve their performance (ASTM International, 2008; Miller *et al.*, 1985). In (Miller *et al.*, 1985; Vecchio *et al.*, 1985; Topp and Dover, 1983; Sooley and Hoeppner, 1985) the problem of acquiring and processing HTM data in real-time was considered. A wide range of application and measurement techniques using HTM are found in Cullen (1985). Although, other types of testing machines are available, our focus is on hydraulic ones.

Usually, an HTM has the following main components (Sun *et al.*, 2011; Alva, 2008; Serrano, 2007; Pelloux and

[‡] K_I is intensity stress factor in mode I and it is designated and applied to the crack opening mode that means $K_I = \lim_{r \to 0} \sqrt{2\pi r}\sigma_{yy}(r, 0)$ and K_{IC} is the critical intensity stress factor in mode I (Hosford, 2010).

[§] (ASTM International, 2008, E–647) describes test methods for measurement of fatigue growth rates.

Brooks, 1964): (1) main frame; (2) hydraulic actuator; (3) hydraulic power supply ; (4) servo-valve; (5) controller/computer; (6) load cell; (7) grips and (8) specimen to be tested (Fig. 2). All elements of a HTM have their specific importance and influence the system as a whole. For instance, if the hydraulic pump should not deliver a constant supply pressure, the actuator would not be able to deliver the calculated force. The constructive components are assumed to be perfect build and no leakage is found(Will and Gebhardt, 2011; Jelali and Kroll, 2003; Valdiero, 2004; Plummer, 2007).

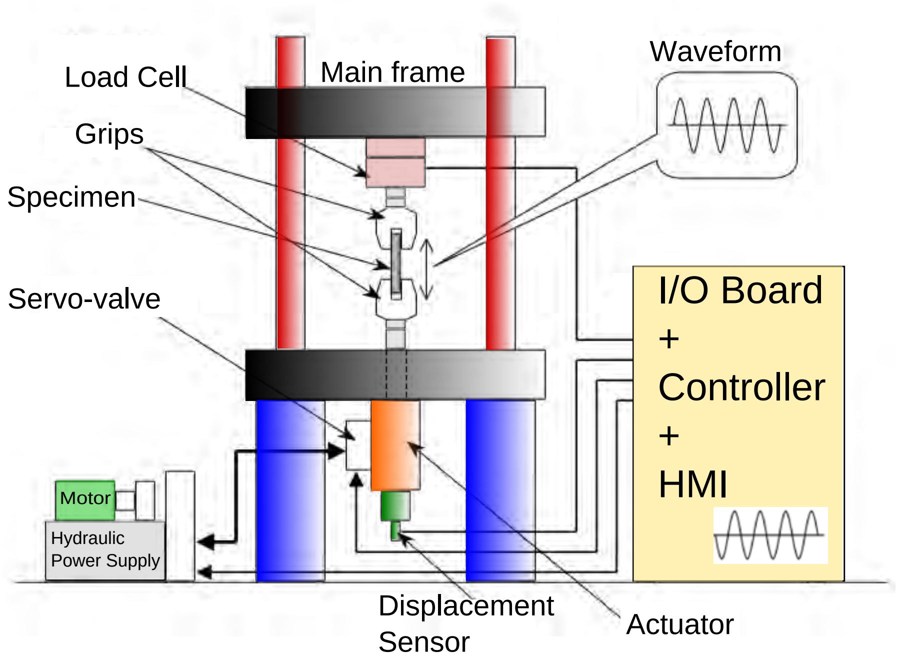


Figure 2: Constructive aspects of a HTM: components proposed by Sun *et al.* (2011); Alva (2008); Serrano (2007); Pelloux and Brooks (1964). Adapted from (Kasprzyczak and Macha, 2008).

The framework is responsible for the control and measurement of testing and non-testing variables, in order to achieve test specifications. They set and control the actuator operating modes, as well as the testing conditions such as force amplitudes and mean values. Some variables measured are displacement, distortion, speed, acceleration, pressure inside the cylinder, and so on. Machines where the hydraulic actuator is operated by a servo-valve electrically controlled is called electro-hydraulic testing machines (EHTMs) (Ahn and Truong, 2009; Rabie, 2009; Sohl and Bobrow, 1999).

The EHTM's commercially available vary from small capacities ($\sim 10\,\mathrm{N}$) up to large capacities ($\sim 1000\,\mathrm{kN}$). They can produce small displacements ($\sim 1\,\mu\mathrm{m}$) or large ones ($\sim 0.1\,\mathrm{m}$). A large variety of waveforms, speeds, and frequencies are selectable. Examples of common waveforms are sine-waves, rectangular, triangular, trapezoidal, ramp, hold, complex, and random-waveforms. Speeds can range from $0.5\,\mu\mathrm{m/min}$ to $20.0\,\mathrm{m/sec}$ and frequencies starting at $10^{-4}\,\mathrm{Hz}$ to $300.0\,\mathrm{Hz}$.

Armstrong-Hélouvry *et al.* (1994) is considered an essential reference in friction modeling. Friction has its importance and dominant disturbance as it can cause typical steady error over the positioning control and cause instabilities also (Canudas-de-Wit *et al.*, 1995; Canudas-de-Wit and Ge, 1997). Controller's performance is diminished due to friction's effect and has a considerable effect on the controller's can be modeled as a function of velocity Sohl and Bobrow (1999). Armstrong-Hélouvry *et al.* (1994) have shown a performance reduction due to the non-linearity of the friction. Some effects due to friction include stick-slip, hunting, standstill and quadrature glitch (Valdiero, 2004). Adaptive controllers that require little prior information regarding the plant parameters while assuring closed loop stability can be found in Stoten (1992); Guan and Pan (2008); Clarke and Hinton (1997); Bessa *et al.* (2010).

In order to satisfy all the requirements listed above, a computer system operating under real-time conditions may be satisfactory (Miller *et al.*, 1985). The control system monitors and corrects continuously any deviation of load behavior to maintain the reference signal (Jelali and Kroll, 2003). In the next section, we try to cover in a nutshell some important aspects when considering the control design of a HTM.

3. CONTROL THEORY FOR HYDRAULIC SERVO-SYSTEMS

The development of hydraulic servo-systems (HSSs) have attracted the attention of the control community and generates several industry applications. Although some technologies such as electrical, pneumatics, mechanical, solar, and others, are nowadays in the spotlight, it is well known that hydraulic technology plays an important role (Will and Gebhardt, 2011; Jelali and Kroll, 2003; Alleyne and Liu, 2000; Merritt, 1967). Indeed, hydraulic system is attractive mainly

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due to its good power/mass ratio, its high mechanical stiffness, its fast response during the startup/stop and its fast response with respect to control direction reversion (Jelali and Kroll, 2003).

The hydraulic actuator can be assembled in order to produce linear or rotational displacements (Will and Gebhardt, 2011; Linsingen, 2001), allowing a wide range of applications, such as: excavator, mining vehicles, flight simulator platforms, Stewart platforms and testing machines. Hydraulic systems have become a natural solution for actuation in testing machines due to their reliable components and unique characteristics which surpass the overall performance of other technologies (Will and Gebhardt, 2011; Jelali and Kroll, 2003; Merritt, 1967).

As mentioned before, the advantages of the hydraulic servo-systems when compared to electrical servo-systems (for example) are the ability to: generate higher force/torque, higher load stiffness and faster transient response during speed reversal (Jelali and Kroll, 2003). On such conditions, electro-motors tend to overheat (Vossoughi and Donath, 1995). In addition, it must be highlighted that HSSs are able to operate continually under different required operation conditions (Will and Gebhardt, 2011; Bessa *et al.*, 2010; Jelali and Kroll, 2003; Six *et al.*, 2001; Alleyne and Liu, 2000; Sirouspour and Salcudean, 2000; Sohl and Bobrow, 1999; Lu and Lin, 1993; Merritt, 1967).

Hydraulic systems are also attractive for developing high performing applications and high power applications, however, according to Bessa *et al.* (2010); Jelali and Kroll (2003); Sirouspour and Salcudean (2000); Sohl and Bobrow (1999); Merritt (1967) there some issues that must be considered, such as: (1) hydraulic components have a significant cost when specified to achieve small tolerances; (2) hydraulic system has complex nonlinear characteristic which is hard to model and/or control; (3) in all hydraulic systems, fluid contamination is an important issue and almost impossible to avoid completely; (4) near ignitions sources, explosion is not precluded *a priori*; (5) an undesirable working environment can arise due to fluid leakage. Therefore, hydraulic servo-systems implementations may become inadvisable when considering low power applications. It is straightforward to mention that, in spite of the challenges/difficulties concerning hydraulic systems, several results can be found in the literature which seek for improvements in the modeling and/or control of HSSs (Will and Gebhardt, 2011; Bessa *et al.*, 2010; Jelali and Kroll, 2003; Sirouspour and Salcudean, 2000; Sohl and Bobrow, 1999).

Pelloux and Brooks (1964); Miller *et al.* (1985); Vecchio *et al.* (1985) present a fatigue machine without feedback, i.e., under open-loop control, with a hydraulic actuator. The control system is composed by a hydraulic pressure generator and an electronic rack containing an oscilloscope, a program command unit and a function generator. Conversely, several feedback control strategies were proposed for hydraulic systems Will and Gebhardt (2011); Jelali and Kroll (2003); Merritt (1967), since the early 1960's. Applications using HSSs normally require position control. However, in the context of fatigue analysis, force control is also needed.

In particular, results for position control can be found in Bessa *et al.* (2010); Karpenko and Sepehri (2010); Kim *et al.* (2010); Guan and Pan (2008); Kim and Lee (2006); Sirouspour and Salcudean (2000); Merritt (1967); Vossoughi and Donath (1995); Stoten (1992). Moreover, results regarding speed and force control can be found in Ho and Ahn (2012); Kim and Lee (1996); Guo and Schwarz (1989) and in Choi (2012); Sánchez *et al.* (2012); Truong and Ahn (2011); Will and Gebhardt (2011); Ahn and Truong (2009); Alva (2008); Serrano (2007); Niksefat and Sepehri (1999); Kasprzyczak and Macha (2008); Jelali and Kroll (2003); Clarke and Hinton (1997), respectively.

In the following section, we describe a single basic mathematical model for representing the servo-valve, actuator and specimen. No comprehensive control strategy dealing with all nonlinearities in HTM is found. In general, the nonlinearities due to deviation from the operating conditions are considered as plant uncertainties (Vossoughi and Donath, 1995).

3.1 Modeling

The HTM is equipped with an actuator (cylinder), control elements (valve, sensors, etc.) and a hydraulic power supply (Fig. 2). Each subsystem components of the HSS are described, including the dynamic model, in Will and Gebhardt (2011), Jelali and Kroll (2003) and Merritt (1967). The hydraulic pumps convert mechanical energy into hydraulic energy, when the fluid is pumped through pipes and connectors. Spool valves are common used for flow control. As a result, the flow in the cylinder's chamber is modified resulting in mechanical movement (Fig. 3).

In general, a mathematical model for representing the servo-hydraulic dynamics is required to achieve acceptable fatigue test results (Bessa *et al.*, 2010; Vossoughi and Donath, 1995; Lee and Srinivasan, 1990). Denoting the piston displacement by y, the y-dynamics can be described by the following mass-damper-spring system (Fig. 3):

$$M_t\ddot{y} + B_t\dot{y} + K_t y = F_H - F_f - F_L - F_G\,, \tag{1}$$

where

$$F_H = A_1P_1 - A_2P_2\,,$$

is the force generated by the piston, also referred as the hydraulic force, $F_f = g(\dot{y})$ is the force of Coulomb friction and

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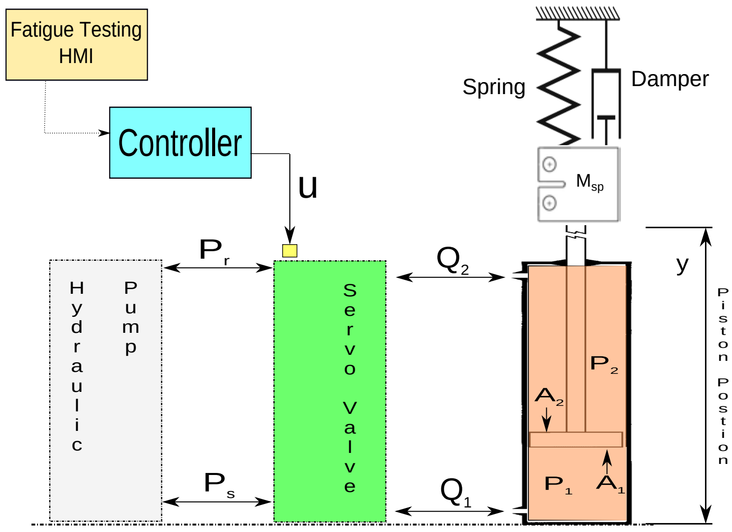

Figure 3: Piston movement with specimen behavior modeled as second order system . Adapted from (Sohl and Bobrow, 1999).

stiction[¶], F_L is the external force, F_G is the gravitational force and P_1 and P_2 are the pressure in each individual cylinder chambers, A_1 and A_2 are the cross-section areas of the chambers and M_t, B_t and K_t are the total load mass, viscous damping coefficient and spring constant, respectively (Fig. 3). Here, the total load includes: all the grips, the load cell, the specimen and piston (Bessa *et al.*, 2010; Sohl and Bobrow, 1999; Vossoughi and Donath, 1995; Lee and Srinivasan, 1990). Note that, due to the specimen pre-crack, the force (F) applied to the specimen can be slightly different from the hydraulic force (F_H) generated by the actuator. Moreover, the force F applied over the specimen is measured by a load cell (force transducer).

Figure 4 illustrates all the components related to HTMs, where the HMI assists the user to create, monitor the test parameters and generate force waveforms. As presented in Jelali and Kroll (2003), Pereira (2006), Cunha (2001), Sohl and

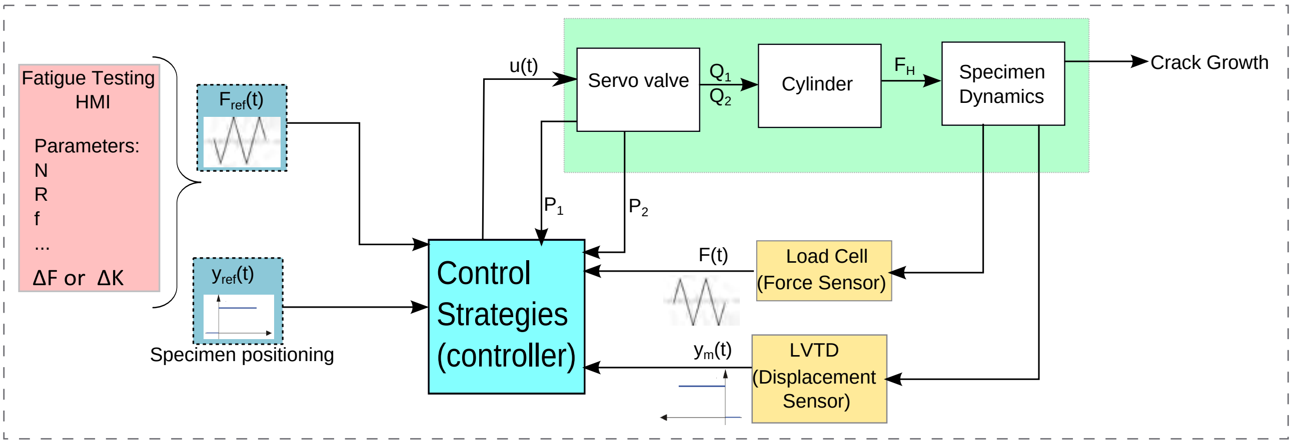

Figure 4: Schematic of a HTM.

Bobrow (1999) and Merritt (1967), the hydraulic fluid compressibility (bulk modulus) has a significant influence on the HSSs dynamics. When the fluid flow is null then one has that the fluid bulk modulus is described by $\beta = -V(dP/dV)$, where V and P are the chamber volume and pressure, respectively. Moreover, Jelali and Kroll (2003) asserts that, in fact, the bulk modulus depends on pressure, entrained air and mechanical compliance. An empirical formulation for the effective bulk modulus, named β_e, is proposed in the form $\beta_e = 1/[\kappa_1 + \kappa_2(P/P_0)^{-\lambda}]$, where κ_1, κ_2 and λ are appropriate constants and P_0 is a nominal value for the chamber pressure P (Jelali and Kroll, 2003, p. 35). Now, considering the presence of the fluid flow Q_1 and Q_2 in each chambers of the cylinder, the pressure dynamics can be modeled by

$$\dot{P}_1 = \frac{\beta_e}{V_1}\left(-\dot{V}_1 + Q_1\right) , \tag{2}$$

[¶]Stiction is the static friction that needs to be overcome to enable relative motion.

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$$\dot{P}_2 = \frac{\beta_e}{V_2}\left(-\dot{V}_2 + Q_2\right) , \tag{3}$$

where V_1 and V_2 are the total fluid volumes inside each chamber.

The servo-valve's mechanism is controlled by an electrical signal u, which can be considered directly proportional to the spool position (Fig. 3). A full description of the physical variables relevant to model the servo-valve dynamics can be found in Rabie (2009); Vossoughi and Donath (1995); Lee and Srinivasan (1990). As in Sohl and Bobrow (1999), the time constant of the servo valve dynamics can be neglected in comparison with the hydraulic dynamics time constant. In addition, according to Jelali and Kroll (2003), valve leakage can also be neglected or considered as a disturbance for the mechanical subsystem (1). Hence, the fluid flows Q_1 and Q_2 are statically related to the electrical command u by:

$$Q_1 = \begin{cases} c_1 u\,\sqrt{p_s - P_1}, & \text{for} \quad u \geq 0\,, \\ c_2 u\,\sqrt{P_1 - p_r}, & \text{for} \quad u < 0\,, \end{cases} \tag{4}$$

$$Q_2 = \begin{cases} -c_3 u\,\sqrt{P_2 - p_r}, & \text{for} \quad u \geq 0\,, \\ -c_4 u\,\sqrt{p_s - P_2}, & \text{for} \quad u < 0\,, \end{cases} \tag{5}$$

where c_i $(i = 1, 2, 3, 4)$ are the valve orifice coefficients, p_s is the supply pressure and p_r is the reservoir pressure. For more details, see (Will and Gebhardt, 2011; Jelali and Kroll, 2003; Sohl and Bobrow, 1999; Boulet *et al.*, 1992). Differentiating the hydraulic force generated by the piston yields

$$\dot{F}_H = A_1\dot{P}_1 - A_2\dot{P}_2\,. \tag{6}$$

Thus, noting that $V_1 = V_{10} + yA_1$ and $V_2 = V_{20} + (L - y)A_2$, where V_{10} and V_{20} are the initial volume values, L is the maximum piston stroke, one can write $\dot{V}_1 = A_1\dot{y}$ and $\dot{V}_2 = -A_2\dot{y}$. Therefore, the dynamics between u and the hydraulic force F_H can be written as:

$$\dot{F}_H = -y\beta_e\left(\frac{A_2^2}{V_2} + \frac{A_1^2}{V_1}\right) + z(y, P_1, P_2)u\,, \tag{7}$$

where

$$z = \begin{cases} \beta_e(\dfrac{A_2c_3}{V_2}\sqrt{P_2 - p_r} + \dfrac{A_1c_1}{V_1}\sqrt{p_s - P_1}), & \text{for} \quad u \geq 0\,, \\ \beta_e(\dfrac{A_2c_4}{V_2}\sqrt{p_s - P_2} + \dfrac{A_1c_2}{V_1}\sqrt{P_1 - p_r}), & \text{for} \quad u < 0\,. \end{cases} \tag{8}$$

Equations (1) to (8) represent the non-linear model. For control design, a linear model for the hydraulic actuator is considered in Pereira (2006); Cunha (2001); Serrano *et al.* (2008):

$$G_a(s) = \frac{2\beta_e/V_tM_t\,(K_{qu1}A_1 + K_{qu2}A_2)}{s\,[s^2 + (B_t/M_t + (2\beta_eK_t)/V_t) + \beta_e(B_tK_t + A_1^2 + A_2^2)/V_tM_t)]} \tag{9}$$

where K_{qu1} and K_{qu2} are the flow-pressure coefficients. The complete linear analysis can be found in Pereira (2006). Using the state vector $\mathbf{x} = \left[\begin{array}{cccc} y & \dot{y} & P_1 & P_2 \end{array}\right]^T$, the system can be represented by (Alleyne and Liu, 2000):

$$\begin{aligned} \dot{x}_1 &= x_2, \\ \dot{x}_2 &= x_3A_1 - x_4A_2 - F_f/M_t, \\ \dot{x}_3 &= \beta_e/(V_{10} + x_1A_1)(-x_2A_1 + Q_1(x_3, u)), \\ \dot{x}_4 &= \beta_e/(V_{20} + (L - x_1)A_2)(-x_2A_2 + Q_2(x_4, u))\,. \end{aligned} \tag{10}$$

Figure 5 illustrates the EHTM subsystems. As mentioned before, a precise model of the servo-valve is difficult to obtain since many non-linear dynamic effects are apparent, such as: dead-band, saturation, hysteresis, and so on. When the control objective is to increase flow rate and response speed to obtain large deflection at higher frequencies , pipeline dynamics should also be taken into account (Fig. 5) (Morgan and Milligan, 1997). In particular, for high frequency testing, pipeline vibration may appear. In this case, in order to attenuate vibration, an accumulator is usually located closely to the main hydraulic machine. Otherwise, pressure fluctuation should be considered in the control design (Will and Gebhardt, 2011; Jelali and Kroll, 2003).

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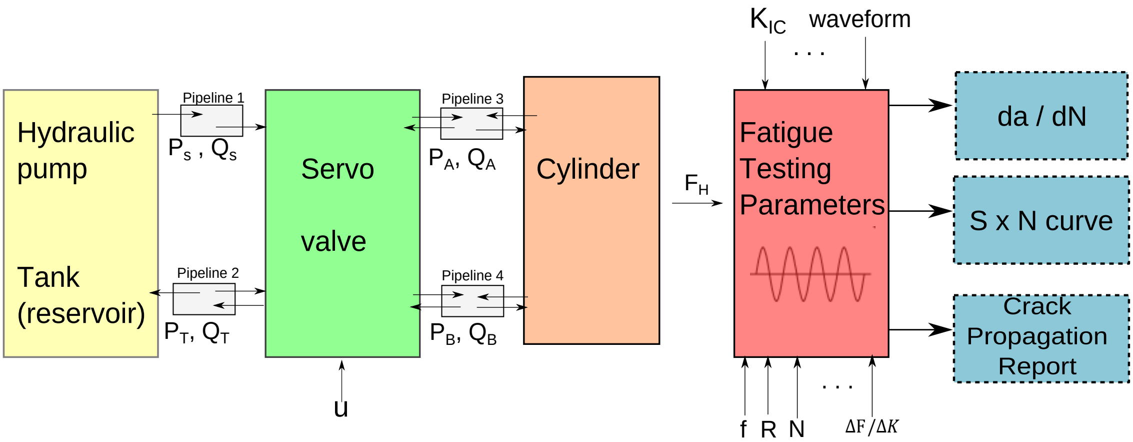

Figure 5: Subsystems of servo-hydraulic fatigue testing machine and expected results. Adapted from (Jelali and Kroll, 2003).

3.2 Control Strategies

Hydraulic servo-system control strategies via output feedback date from the early 67's (Merritt, 1967; Anderson, 1989). Advanced control techniques have being developed to overcome some difficulties, in particular, regarding force control (Truong and Ahn, 2011; Sun *et al.*, 2011; Ahn and Truong, 2009; Alleyne and Liu, 2000; Niksefat and Sepehri, 1999; Sohl and Bobrow, 1999). For example, in Sohl and Bobrow (1999) it is shown that the local linear approximation model may become unstable, for some initial conditions (or operation point). The practical importance of a proper estimation of the dead-band for the valve is highlighted in Sohl and Bobrow (1999), where it is verified that force tracking performance degradation may become significant, for large values of the estimation error. Moreover, the friction force must be considered in order to obtain acceptable tracking accuracy. In Sohl and Bobrow (1999), the proposed control law

$$u = \frac{1}{z}\left(\dot{F}_{hd} - \kappa_F(F_H - F_{hd}) + \dot{y}\beta_e\left(\frac{A_2^2}{V_2} + \frac{A_1^2}{V_1}\right)\right) \tag{11}$$

assures that the tracking error $\varepsilon = F_H - F_{hd}$ converges exponentially to zero according to

$$\frac{d\varepsilon}{dt} = -\kappa_F\varepsilon\,,$$

by using the Lyapunov function $V = (1/2)\varepsilon^2$, where F_{hd} is the desired force trajectory for the hydraulic force F_H, $\kappa_F > 0$ is a design error gain and z is defined in (8).

According to Alleyne and Liu (2000), classical PID controllers can not overcome some problems related to tracking force (or pressure) in hydraulic systems, mainly due to friction. In order to increase flow rate and response speed, in general, traditional servo-controller requires a high accurate system model. In this sense, a robust backstepping control scheme was developed in Alleyne and Liu (2000) for an electro-hydraulic actuator (EHA), but the price paid is the increasing computational effort.

In this context, nonlinear quantitative feedback theory (QFT) was considered in Niksefat and Sepehri (1999), providing a closed loop control strategy robust with respect to parametric uncertainties. The nonlinear plant is equivalently replaced by a family of linear time invariant (LTI) plants. The key idea is to consider a family of second-order linear transfer functions of the form

$$G_eq(s) = \frac{K_p}{(1 + s/\alpha_1)(1 + s/\alpha_2)}\,, \tag{12}$$

obtained via Golubev's method, to approximate the dynamics between the actuator (piston) displacement (which is assumed proportional to the electrical control signal) and the generated hydraulic force, where the uncertain parameters K_p, α_1 and α_2 belong to some known interval. Three scenarios were considered: (1) the effect of variation environmental stiffness, (2) pump pressure, and (3) loading conditions. These three situation are expected on a fatigue test, due to (1) crack growth, (2) continuously and long period operations and (3) dynamically force operating point (changing ΔK). The corresponding linear controller is given by

$$C(s) = \frac{k_1(1 + s/k_2)(1 + s/k_3)}{s(1 + k_4 s + k_5 s^2)}\,, \tag{13}$$

where k_i $(i = 1, \ldots, 5)$ are appropriate constants. The main problem here is that overshoot can appear in the applied force. Therefore, since force overshoot can modify the stress intensity factor (K), the fatigue test result may become questionable Hosford (2010).

Later on, in Kasprzyczak and Macha (2008) by assuming that both servo-valve and actuator can be modeled by second order systems, a PID controller was designed and tuned. However, after several tuning process, it was verified that the force transient always contains overshoot. As mentioned before, the overshoot must avoided for an acceptable fatigue test. On the other hand, some practical hints on how to tune PID controllers for fatigue testing machines was provided. A previous solution was made using self-tuning control to closed loop material testing (Lee and Srinivasan, 1990). Differently from Kasprzyczak and Macha (2008), the nonlinearities appeas due to : (1) the servo-valve was not critically centered, but had a small overlap and leakage should be taken in account; (2) servo-valve mechanical is null offset, that is zero electrical voltage input to the servo-amplifier; (3) actuator leakage effect was significant and hence necessary to model (Lee and Srinivasan, 1990). In other to achieve higher frequency, greater demands on online identification is considered.

Adaptive cascade control of hydraulic actuator (HA) is made to compensate unknown dead-zone can be seen in Cunha (2001). The assumption that the valve has a dead-zone with unknown parameters. The strategy is to see the HA as two interconnected subsystems: a hydraulic and mechanical one. An adaptive controller is proposed, but they shown that the convergence of each parameter is not straightforward.

Later on, an adaptive fuzzy sliding mode controller for an EHA system with unknown dead-zone was designed in Bessa *et al.* (2010) and the closed loop system stability was demonstrated. The system was slight different that other found for HTM, a four-way proportional valve, a hydraulic cylinder and variable load force. Some assumptions were made (1) the dead-zone is not available to be measured, (2) it is limited, (3) the modeling function is bounded with positive valued derivatives . The strategy is interesting for fatigue testing, because the system has parametric uncertainties, unmodeled dynamics and an unknown dead-zone. Only numerical simulations was conducted, but the results encourage practical control implementations.

In a similar approach, Ahn and Truong (2009) considered self-tuning of quantitative feedback theory (QFT) for control for HTM. The first step was to find a family of uncertainties of the plant transfer function. Based on previous work found in (Niksefat and Sepehri, 1999), they performed a identification procedure and reduce the model in Eq. (9) to:

$$P(s) = \frac{k}{(1+p_1 s)(1+p_2 s)} = \frac{k\omega_p^2}{s^2 + 2\xi_p\omega_p + \omega_p^2}\,, (N/mV) \tag{14}$$

where $k \in [3.97,\, 6.81](N/mV)$, $p_1 \in [0.85,\, 1.16]$, $p_2 \in [0.85,\, 1.16]$, $\omega_p = 1/\sqrt{p_1 p_2}(rad/s)$ and $\xi_p = (p_1 + p_2)/(2\sqrt{p_1 p_2}) \leq 1$. The self-tuning controller is considered to be: $C(s) = k(s+a)/[(s+b)(s+c)] = (ks + ka)/(s + m^2 + \omega^2)$The controller time varying parameters $(a,\, b,\, c)$: $(STQTF)_{t+1} =$ (new set)if the satisfying conditions were met, otherwise would load previous setting, *i.e.*, $(STQTF)_{t+1} = (STQTF)_t$, where each element comes from the QFT algorithm. They compared the results between: (1) conventional PID controller, (2) QFT controller and (3) self-tuning QFT controller. For different environment conditions, the suggested self-tuning QFT controller had better tracking performance, following control criteria of maximum percentage of overshoot $\leq 2\%$.

In order to increase the performance of an excavator, Choi (2012) argues that a classical PID controller is not able to perform the task and an advance technique is required. Choi (2012) presents a robust sliding mode controller for hydraulic actuators designed via backstepping. Three pressure closed loop step responses are obtained with the proposed controller and compared with the conventional PID controller. The conventional control performance presents larger overshoot for a low pressure test and larger settling time for a high pressure test.

All these (hybrid) control schemes try to estimate the system uncertainties. When dealing with friction and hydraulic uncertainties, learning control and neuro-fuzzy learning control (Sánchez *et al.*, 2012), (Alva, 2008), and feedback linearization with fuzzy compensation (Tanaka *et al.*, 2012) were developed to overcome uncertainties in order to increase the frequency of the applied load cycles in fatigue tests. In (Alva, 2008) a comparison was made between PI, PID controller and learning controller. A forth order linear model was considered, and a learning method control was applied on a HTM. The results show that no prior knowledge is necessary to improve the learning gain. Alternatively, a comparison between learning control and neuro-fuzzy system is found in (Sánchez *et al.*, 2012). For the same simulation and experiment condition, the neuro-fuzzy learning control converges faster than conventional learning control and the difference between then is that neuro-fuzzy learning control requires less memory to store the tuning parameters.

The problem with uncertainty estimation, such as friction and compressibility of hydraulic fluid (bulk), must be carefully considered. Indeed, in the mentioned references were friction is disregarded the fatigue test degradation is apparent and high frequency tests are precluded.

4. CONCLUSIONS

Hydraulic control applied to fatigue testing machines was reviewed. The main issues around hydraulic fatigue testing machines were identified and explained. We noticed that several attempts were made to describe how to design and implement a computer system for usage in fatigue testing, but fewer results are available regarding a control perspective. Most of them, deal with the computational issues concerning data acquisition.

The fatigue testing was briefly presented with the main variables involved in the test. While servo-hydraulic actuation and computers have improved the fatigue and fracture tests, the main question that still arises is the development of an standard testing environment to conform the mechanical properties of materials.

A historical review of some of the techniques applied to force control were presented, providing an important insight on how to develop and design servo-hydraulic fatigue testing machines in order to achieve standard practical requirements. Moreover, the main nonlinearities of servo-hydraulic system were pointed out and some ideas on how to deal with these nonlinearities were presented. Future works include: performance tests for different compensation algorithms, simulations and experimental evaluation.

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