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FORCE CONTROL BASED ON PARAMETER ESTIMATION APPLIED TO FATIGUE TESTING MACHINES

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Abstract. *This paper addresses the force control of fatigue testing machines applied to material characterization under corrosive and pressurized environment. It is verified via numerical simulations that the specimen stiffness variation due to the crack propagation should be compensated via force feedback. Since the specimen stiffness varies slowly, the system has two main time scales allowing us to use a normalized least squares identification algorithm to estimate the specimen stiffness which is subsequently used in the control design. The aim is to keep the force error within the standard limits while the specimen stiffness varies along the fatigue test. Simulation results are presented to demonstrate the effectiveness and feasibility of the proposed solution.*

Keywords: *fatigue testing machine, force control, parameter estimation*

1. INTRODUCTION

The Brazilian pre-salt oil and gas discoveries delivered new challenges to material characterization. During oil extraction the equipment are exposed to corrosive and pressured environment. Although corrosion and mechanical properties of the used equipment are well established, the combination of cyclic loading and corrosive environment is a major concern for subsea equipment design (nac, 2015).

Mechanical fatigue and fracture testing on a corrosive medium is researched and developed at the Non-destructive Testing, Corrosion and Welding Laboratory (LNDC/COPPE/UFRJ) of the Federal University of Rio de Janeiro (UFRJ). The corrosive and pressurized environment requires suitable machine that can carry out tests in accordance with international standards. The corrosion mechanism is time dependent, thus the loading frequency for corrosion-fatigue test is recommended to be set between $0.01\,\mathrm{Hz}$ to $2\,\mathrm{Hz}$ (Knop *et al.*, 2010). However, at the present time no consensus about corrosion-fatigue is found in the literature and international standards provide a guideline for determining fatigue crack growth rate at atmospheric conditions (ISO-12108, 2002; ASTM-E1820-08, 2008; ASTM-E647-08, 2008).

The fatigue tests consist in evaluating a metallic materials under mainly linear-elastic stress condition and with force applied only to the crack plane of test specimen. According to (ISO-12108, 2002), the testing machine should initialize smoothly and maintain within $\pm 2\%$ of the desired force range during the test. Different actuators can be used in fatigue testing machines, a conceptual scheme depicted in Fig. 1a and Fig. 1b. The actuation systems for fatigue testing machines can be classified into four categories: (1) servo-hydraulic, (2) electromagnetic, (3) electromechanical and (4) shakers (Cullen, 1985). Since the specimen stiffness varies as the crack propagates, parametric uncertainty is one of the problems found on the fatigue test machine development so that the control system should adapt quickly to fulfill the requirements. Hydraulic actuation system have been used in fatigue testing machines due to attractive characteristics (Pereira-Dias *et al.*, 2013; Jelali and Kroll, 2003). Recently, an alternative force control was proposed where amplitude and mean value of the force are estimated by modulation/demodulation and reducing the tracking problem as two regulation problems (Pereira-Dias, 2014). However, hydraulic servo-systems have intrinsic non-linearities that introduce difficulties for the control design.

Electromechanical actuated fatigue testing machines (EMTMs) are found to be an alternative cost reduction solution. The motor is connected to a gearbox which converts the rotatory movement into linear, thus applying load to the test

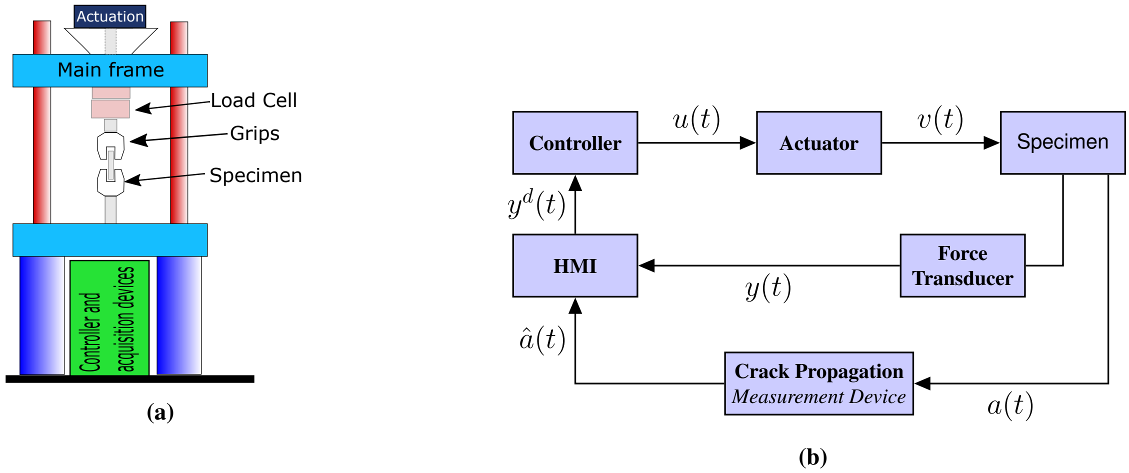

Figure 1: Main components of a fatigue testing machines and block diagram of components' interaction.

specimen. Even though EMTM has an easier setup, the backslash (in the gearbox) introduces a non-smooth non-linearity in the system. A overview of hydraulic actuated fatigue testing machine (HTM), EMTM and convenient instrumentation is found in (nac, 2015; Pereira-Dias *et al.*, 2014, 2013; de Lima, 2014; Marsch *et al.*, 1991).

In this paper we use a normalized least squares identification algorithm to estimate the specimen stiffness which is subsequently used in a conventional proportional-derivative control scheme with feedforwad action. The main contributions of this paper lies in three aspects: (i) a general problem formulation is presented covering hydraulic actuated machines and electromechanical machines; (ii) a suitable formulation is proposed allowing to obtain an estimate for the specimen stiffness via a normalized least squares scheme and (iii) simulation results illustrates the applicability of the proposed scheme, even when the equivalent stiffness is subjected to variation.

2. PROBLEM FORMULATION

Consider the following general formulation for modeling fatigue testing machines. Let the single-input single-output (SISO) linear time varying system composed by the following two subsystems in cascade,

$$\mu(t) = G_a(s)u(t)\,, \qquad b_2\ddot{z}(t) + b_1\dot{z}(t) + b_0K_{sp}z(t) = \mu(t) + d\,, \qquad y(t) = K_{sp}z(t)\,, \tag{1}$$

where $u \in \mathbb{R}$ is the control input of the actuator subsystem, $\mu \in \mathbb{R}$ is the force generated by the actuator (assumed to measured), $G_a(s)$ is the actuator transfer function*, $z \in \mathbb{R}$ is the unmeasured position of mechanical subsystem, $y \in \mathbb{R}$ is the measured force of the mechanical subsystem (the specimen is assumed to satisfy the Hook's law), d is a disturbance (usually due to the gravitational force) and K_{sp} is the **specimen stiffness**. The parameters b_0, b_1 and b_2 depend on: the specimen mass M_{sp} (moving parts), the total viscous friction coefficient B_{sp}, the ratio of the reduction system and the gravitational acceleration g. The following assumption (Callister, 2010; Hosford, 2010; Anderson, 2005) is considered:

(A0) The uncertain specimen stiffness satisfies $\dot{K}_{sp} = 0$ and $K_{sp} > 0$.

Note that **(A0)** is not restrictive, since based on reliable technical and experimental data, it is reasonable to consider that the specimen stiffness K_{sp} varies slowly with time as the crack propagates (non-linearities caused by crack closure is disregarded) (Pereira-Dias, 2014). Moreover, it allows us to cover different materials with the same control scheme proposed in this paper. In fact, this formulation is also general in the sense that it **covers fatigue testing machines** including: hydraulic actuated and electromechanical machines (with actuation provided by rotational and linear motors). The assumption that the force generated by the actuator is available is not restrictive since in the electromechanical actuator the armature current i_a is usually measured. Thus, the induced torque τ_m can be obtained as $\tau_m = k_i i_a$, where k_i is the toque constant obtained from the manufacture datasheet of the dc motor. Moreover, when hydraulic actuation is considered, the pressures in the chambers (P_a and P_b) are usually measured. Hence, the hydraulic force F_H can be directly obtained by $F_H = (P_a - P_b)A$, where A is the effective area of the piston obtained from the manufacture datasheet. Finally, when linear motor is used for actuation, the force generated can be obtained by considering the Park and Clarke's transformation (Instruments, 1997).

* The symbol "s" represents either the Laplace variable or the differential operator "d/dt", according to the context. The output y of a linear time invariant (LTI) system with transfer function $H(s)$ and input u is denoted by $y = H(s)u$.

Mechanical Part

An **electromechanical machine actuated by a DC motor** with a reduction system having a gear box with $1:N$ ratio, can be modeled by

$$M_{sp}\ddot{z} + B_{sp}\dot{z} + K_{sp}z = F_{sp} - M_{sp}g\,, \tag{2}$$

where $F_{sp} = -N\tau_L$ and $-\tau_L$ is the torque reaction acting at the motor after the gear box (with gear ratio N). By noting that $\theta = Nz$, one can write

$$M_{sp}\frac{\ddot{\theta}}{N} + \frac{\dot{\theta}}{N}B_{sp} + K_{sp}\frac{\theta}{N} = -N\tau_L - M_{sp}g\,. \tag{3}$$

Now, by applying the Newton's law to the motor one has that $J_m\ddot{\theta} + B_m\dot{\theta} - \tau_L = \tau_m$, where τ_m is the induced torque. Thus, $-\tau_L = \tau_m - J_m\ddot{\theta} - B_m\dot{\theta}$ and one can further write

$$\left(\frac{M_{sp}}{N^2} + J_m\right)\ddot{\theta} + \left(\frac{B_{sp}}{N^2} + B_m\right)\dot{\theta} + \frac{K_{sp}}{N^2}\theta = \tau_m - \frac{M_{sp}g}{N}\,. \tag{4}$$

Finally, the equivalent specimen stiffness constant observed by the motor is given by $K_{sp}^m := K_{sp}/N^2$ and the dynamic equation can be written as

$$J\ddot{\theta} + B\dot{\theta} + K_{sp}^m\theta = \tau_m - \frac{M_{sp}g}{N}\,, \tag{5}$$

where $B := \left(\frac{B_{sp}}{N^2} + B_m\right) \approx B_m$ and $J = \left(\frac{M_{sp}}{N^2} + J_m\right) \approx J_m$. In addition, since $\theta = Nz$ one can further write $JN\ddot{z} + BN\dot{z} + K_{sp}^m Nz = \tau_m - \frac{M_{sp}g}{N}$. Therefore, regarding the general formulation Eq.(1), one has: $b_2 = JN$, $b_1 = BN$, $b_0 = 1/N$, $\mu = \tau_m$ and $d = -M_{sp}g/N$. Now, considering an **hydraulic actuator**, one can write that

$$M_{sp}\ddot{z} + B_{sp}\dot{z} + K_{sp}z = F_H - M_{sp}\,g\,, \tag{6}$$

where the hydraulic force F_H satisfies $F_H = (P_a - P_b)A$, with A being the effective area of the piston and P_a and P_b being the pressures in the chambers. Hence, in view of the general formulation Eq.(1), one has: $b_2 = M_{sp}$, $b_1 = B_{sp}$, $b_0 = 1$, $\mu = F_H$ and $d = -M_{sp}g$. Finally, when **linear motor** is used for actuation a similar relationship can be obtained, i.e.,

$$M_{sp}\ddot{z} + B_{sp}\dot{z} + K_{sp}z = F_L - M_{sp}\,g\,, \tag{7}$$

where F_L is the force generate by the linear motor which comes from the Park and Clarke's transformation.

Actuator and Sensor Parts

One can argue that the linear approximation for the actuator $G_a(s)$ is quite conservative. However, it must be highlighted that in all the three fatigue machines considered here, one can find an internal control loop in the actuator so that this linear approximation holds for a significant range of operating frequencies.

For the electromechanical machine with conventional rotational DC motor the power driver incorporates an internal PI-controller for the angular velocity. In this case, by using the conventional model for the DC motor (Mohan *et al.*, 2002), one can verify that a static relationship approximates the transfer function from the control input (volts) u to the velocity $\dot{z}$. Moreover, by considering the internal filter for the force measurements, one can obtain a transfer function from the control input (volts) u to the force y in the form $K/(\tau s^2 + s)$,for some positive constants K and τ. Similarly, for the electromechanical machine with linear motor, the three phase driver imposes an internal control loop and for the hydraulic machine the control valve add a first order linear dynamics from the control input and the piston velocity.

2.1 Control Objective (Output Tracking)

The control objective consists in finding a control signal $u(t)$ such that the **output tracking error** $e_0(t) := y^d(t) - y(t)$ tends to a small neighborhood of zero as $t \to \infty$ for arbitrary initial conditions, where $y^d \in \mathbb{R}$ is the desired force output for the fatigue testing. In most cases of fatigue testing (constant load), the desired trajectory is defined as

$$y^d(t) = A^d\sin(\omega t) + B^d\,, \tag{8}$$

where ω is the (known) fatigue testing operating frequency, A^d is the desired amplitude for the force oscillations and B^d is the corresponding mean value.

3. CONTROL DESIGN

For the control design we must consider the internal control loop located in the actuator subsystem. As mentioned before, the first order linear approximation for the actuator is reasonable since an angular velocity PI controller is commonly employed in the power driver unit for electromechanical machines while the servo valve provides the PI action for the hydraulic actuated machines. Hence, the relationship between the control input u and the force y is given by a second order dynamics, so that one can write

$$\tau\ddot{y} + \dot{y} = K_m u\,, \qquad K_m = KK_{sp}\,, \tag{9}$$

where τ is the time constant of the power unit internal resulting closed loop PI control, K depends on the reduction/transmission system, u is the power unit control input and y is the force applied to the specimen.

Therefore, a simple PD controller with *feedforward* action is enough to assure a satisfactory tracking closed loop performance. Indeed, as we present next, such term is time dependent of the first and second time derivative of the desired trajectory.

Consider the tracking error as $e_0 := y^d - y$, such as y^d is the desired force reference for the load constant fatigue testing and the time derivative of e_0 given by $\dot{e}_0 = \dot{y}^d - \dot{y}$ and $\ddot{e}_0 = \ddot{y}^d - \ddot{y}$. Thus, by adding and subtracting $\tau\ddot{y}^d + \dot{y}^d$, the system described in Eq.(9) can be rewritten as:

$$\tau\ddot{e}_0 + \dot{e}_0 - \tau\ddot{y}^d - \dot{y}^d = -K_m u\,. \tag{10}$$

Now, considering the dynamic control law

$$u(t) = \frac{\kappa_p e_0 + \kappa_d \dot{e}_0}{K_m} + u_{ff}\,, \qquad \text{with (\textit{feedforward}) term} \quad u_{ff}(t) = \frac{\tau\ddot{y}^d + \dot{y}^d}{K_m}\,, \tag{11}$$

then the error equation is given by

$$\tau\ddot{e}_0 + (1+\kappa_d)\dot{e}_0 + \kappa_p e_0 = 0\,, \tag{12}$$

where κ_p and κ_d are design constants. Clearly, a simple proportional feedback ($\kappa_d = 0$) is sufficient to make the error to tend asymptotically to zero, according to the dynamic of a second order system Eq.(12). Note that, κ_p can be tuned by conventional method to ensure adequate bandwidth and transient response. One can observe that for low frequencies of y^d, the time derivatives can be disregard, thus the feedforward term Eq.(11) becomes unnecessary.

Several results can be found in the literature to achieve the desired tracking accuracy the anticipatory control requires the knowledge of system parameters. In this case, the uncertain parameter is K_{sp}. Small errors in the identification of K_{sp} can lead to errors greater than the allowable limits of fatigue standards. Furthermore, the value of K_{sp} is a function of the specimen characteristics and the crack size (increasing over a fatigue test). Therefore, the parameter K_{sp} varies during a fatigue test or different tests.

The fatigue testing machine is designed to use the attainable full range frequency, so it is appropriate to introduce a form of control that could adapt to conform to uncertainty of K_{sp}. For the case when K_{sp} is unknown and varies slowly we replace K_{sp} by its estimate $\hat{K}_{sp}$, given in what follows, so that the control law (11) is redefined as

$$u(t) = \frac{\kappa_p e_0 + \kappa_d \dot{e}_0}{\hat{K}_m} + u_{ff}\,, \qquad \text{with} \quad u_{ff}(t) = \frac{\tau\ddot{y}^d + \dot{y}^d}{\hat{K}_m} \quad \text{and} \quad \hat{K}_m := K\hat{K}_{sp}\,. \tag{13}$$

4. SPECIMEN STIFFNESS ESTIMATION

For the estimation scheme we must consider that τ_m (the induced torque in the rotational DC motor actuator), F_L (the induced force in the linear motor actuator) and F_H (the hydraulic force in the hydraulic actuator) are available for feedback. Note that this is not restrictive since the armature current in the rotational DC motor, the currents in the linear motor driver and the pressures in the hydraulic actuator are all usually available for feedback.

Now, the plant output y (1) can be written as

$$y = \frac{a_0}{s^2 + a_1 s + a_0} v\,, \quad G(s) := \frac{a_0}{s^2 + a_1 s + a_0}\,, \tag{14}$$

where a_0 and a_1 are the constant system parameters, v is an available signal and a_0 is the only unknown parameter which depends on the specimen stiffness K_{sp}. Indeed, for each individual actuator one has:

$$v = N\tau_m - M_{sp}g\,,\ a_0 = K_{sp}/(JN^2)\,,\ a_1 = B/J\,, \qquad \text{rotacional dc motor actuator}\,, \tag{15}$$

$$v = F_H - M_{sp}g\,,\ a_0 = K_{sp}/M_{sp}\,,\ a_1 = B_{sp}/M_{sp}\,, \qquad \text{hydraulic actuator}\,, \tag{16}$$

$$v = F_L - M_{sp}g\,,\ a_0 = K_{sp}/M_{sp}\,,\ a_1 = B_{sp}/M_{sp}\,, \qquad \text{linear motor actuator}\,. \tag{17}$$

In addition, the transfer function $G(s)$ can be implemented as the result of a negative feedback (unitary feedback gain) connection of the open loop transfer function $a_0 H(s)$, i.e,

$$\frac{Y(\mathrm{s})}{V(\mathrm{s})} = \frac{a_0 H(s)}{1 + a_0 H(s)}\,, \quad H(s) = \frac{1}{s^2 + a_1 s}\,. \tag{18}$$

This will be useful for developing the estimator described in the sequel.

Normalized Least Squares Estimator

Let $e := v - y$ and define the auxiliary error

$$\varepsilon := y - \hat{a}_0\, H(\mathrm{s})\, e\,, \tag{19}$$

where $\hat{a}_0$ is an estimate of a_0. Note that the error $\varepsilon = y - \hat{a}_0 H(\mathrm{s})\,(v - y)$ is an available signal. Moreover, by noting that $y = a_0 H(\mathrm{s}) e$, one can rewrite Eq.(19) as

$$\varepsilon := \tilde{a}_0 H(\mathrm{s}) e\,, \quad \tilde{a}_0 := a_0 - \hat{a}_0\,, \tag{20}$$

which is in an appropriate form to obtain the estimate $\hat{a}_0$ by a Normalized Least Square (NLS) Estimator, which is a commonly used parameter estimation algorithm for updating $\hat{a}_0$ in the error model Eq.(20) (Tao, 2003).

Before implementing the NLS estimator and in order to keep the standard notation, we rewrite the output as

$$y := \theta^* \phi(t)\,, \quad \theta^* := a_0\,,\ \phi := H(s) e\,, \tag{21}$$

and the error Eq.(20) as

$$\varepsilon := -\tilde{\theta}\phi(t)\,, \quad \tilde{\theta} := \theta^* - \theta\,, \tag{22}$$

where $\theta := \hat{a}_0$ is the NLS estimate given by (Tao, 2003):

$$\dot{\theta} = -\frac{P\phi\varepsilon}{m^2}\,, \qquad \theta(0) = \theta_0 \tag{23}$$

$$\dot{P} = -\frac{P^2\phi^2}{m^2}\,, \qquad P(0) = P_0 > 0 \tag{24}$$

$$m^2 = 1 + \kappa\,\phi^2 P\,, \qquad \kappa > 0\,. \tag{25}$$

Reminding that $a_0 = K_{sp}/(JN^2)$ (for rotational DC motor actuator), $a_0 = K_{sp}/M_{sp}$ (for linear motor actuator), $a_0 = K_{sp}/M_{sp}$ (for hydraulic actuator) and $\theta := \hat{a}_0$ the specimen stiffness estimate is given by

$$\hat{K}_{sp}(t) = JN^2\theta(t)\,, \qquad \text{rotacional dc motor actuator}\,, \tag{26}$$

$$\hat{K}_{sp}(t) = M_{sp}\theta(t)\,, \qquad \text{hydraulic actuator}\,, \tag{27}$$

$$\hat{K}_{sp}(t) = M_{sp}\theta(t)\,, \qquad \text{linear motor actuator}\,. \tag{28}$$

5. SIMULATION RESULTS

The aim of the following numerical simulation is to evaluate the control performance under slow time variations in K_{sp}, similar to what happens during real fatigue tests. The values of K_{sp} used in the simulation were based on real experimental data and calculated from the compliance (inverse of stiffness) relationship found in (ASTM-E1820-08, 2008, A12.2, p.32). However, in order to speed up the simulation, the time variation in K_{sp} was increased. Fig. 2 illustrates the K_{sp} time variation. The specimen stiffness K_{sp} ranges from 3.5×10^6 ($\mathrm{N\,m^{-1}}$) to 3.5×10^7 ($\mathrm{N\,m^{-1}}$). The tracking error violates the fatigue testing standards without using the NLS estimator (curves not shown). From Fig. 3 we can notice the main features of the system response obtained with the proposed control and the NLS estimation algorithm. The plant Eq.(9) is considered with $K = 1/5000$ and the equivalent specimen stiffness constant is initialized with the value $K_{sp}(0) = K_{sp}^{nom} = 0.9{\times}3.5 \times 10^7\mathrm{N\,m^{-1}}$. It was observed that the initial values of the NLS estimate and the covariance, θ_0 and P_0, respectively, are the key parameters to improve the estimation to the specimen stiffness K_{sp}. The reference signal y^d is a sinusoidal wave with amplitude $A^d = 3 \times 10^3\mathrm{N}$, mean value $B^d = 5 \times 10^3\mathrm{N}$ and frequency equals to $1\,\mathrm{Hz}$. The plant initial condition is $y(0) = B^d$ and $\dot{y}(0) = 0$, this condition can be achieved with some pre-loading procedure. In Fig. 3, the tracking error vanish as a consequence of the corrected estimation of the specimen equivalent stiffness.

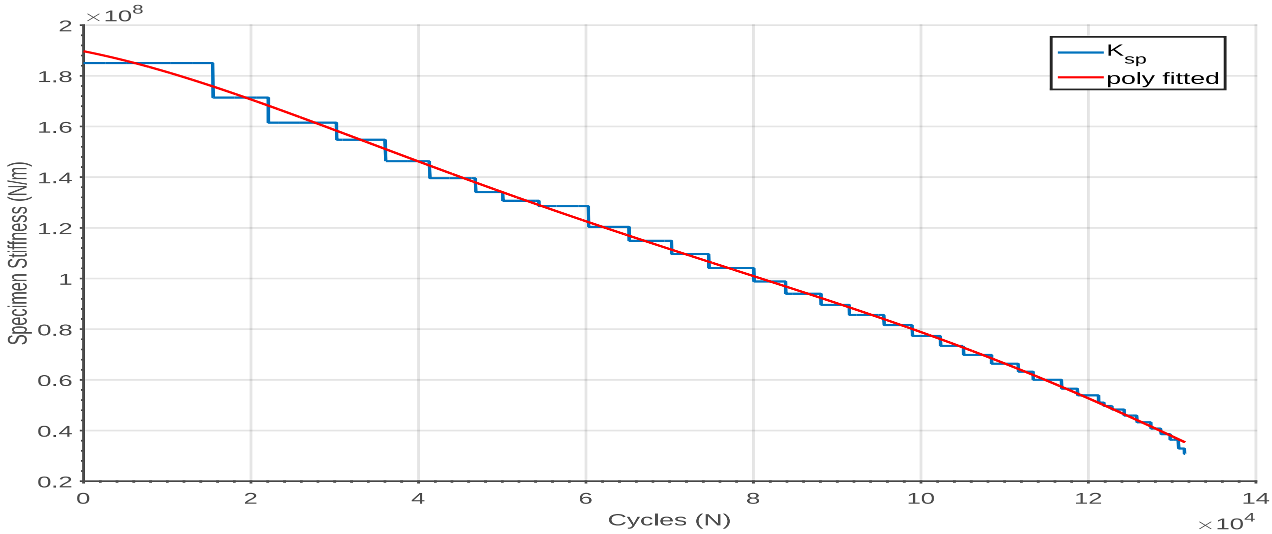

Figure 2: Crack propagation measurements and specimen stiffness interpolation: (blue dashed line) crack propagation and (green solid line) K_{sp} time varying behavior. The time axes is presented in number of cycles (N) divided by 1×10^4 cycles. *Source*: adapted from (Pereira-Dias, 2014).

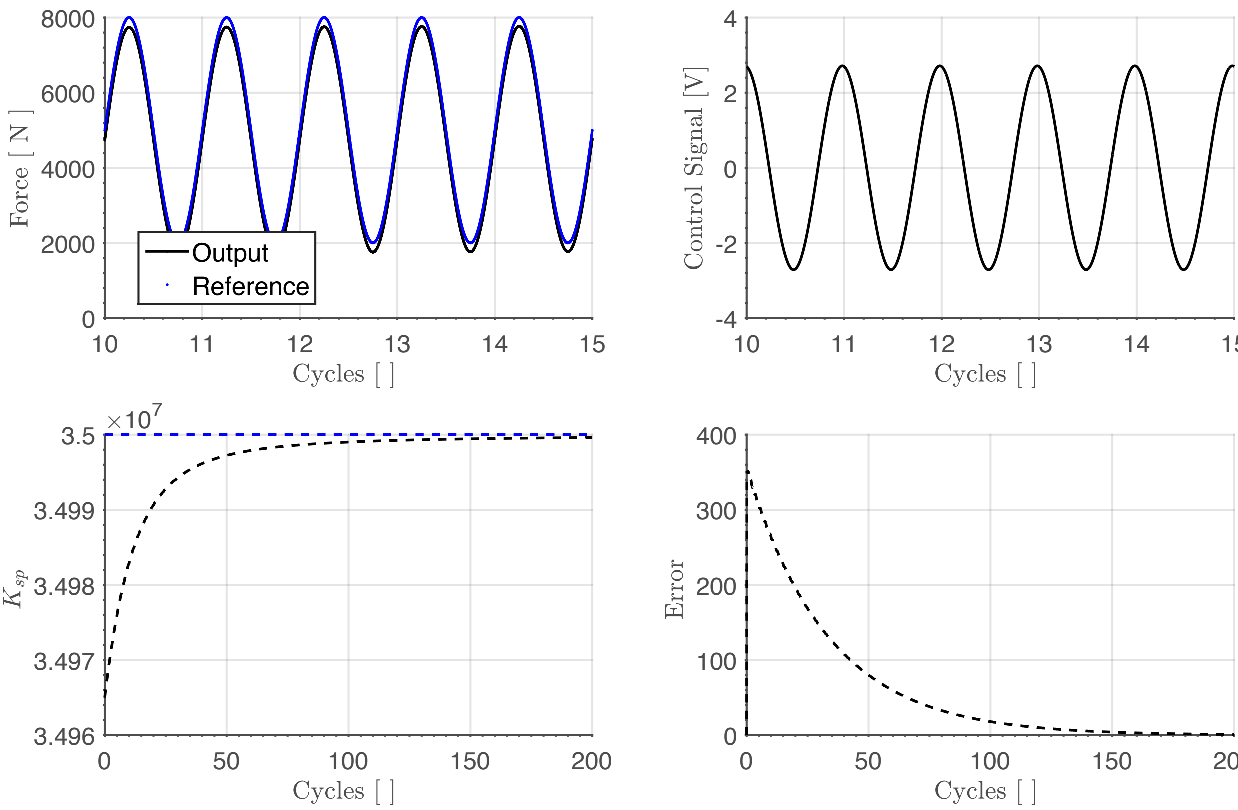

Figure 3: Simulation results using the NLS estimation algorithm: (a) force output, (b) control effort, (c) specimen stiffness estimate $\hat{K}_{sp}$ and (d) tracking error.

6. CONCLUSIONS

The motivation for this work was to develop a force control for fatigue testing machines applied to material characterization under corrosive and pressurized environment. A general testing machine was considered so that the proposed method can be applied to hydraulically actuated testing machine as well as to electromechanical actuator with linear or rotational electrical dc motor. It was verified via numerical simulations that the specimen stiffness variation due to the crack propagation should be compensated via force feedback in order to keep the force error within the standard limits. A normalized least squares identification algorithm was employed to estimate the specimen stiffness. This estimate was used to obtain the feedforward term of a conventional proportional-derivative controller. It was verified that the initial conditions of the estimator play an important role to improve the estimation. The overall closed loop stability analysis including the normalized least squares estimator was left for future work. However, due to the slow time variation of the specimen stiffness (while the crack propagates), the system has two significant different time scales so that one can verify that the closed loop system remains stable. Numeral simulation demonstrates the feasibility of the proposed solution with the specimen stiffness being updated every 1000 cycles which is acceptable since a carbon steel fatigue test endure for 130 thousand cycles.

7. ACKNOWLEDGEMENTS

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