COLEÇÃO

DESAFIOS DAS ENGENHARIAS:

ENGENHARIA MECÂNICA 2

HENRIQUE AJUZ HOLZMANN

JOÃO DALLAMUTA

(ORGANIZADORES)

Atena Editora

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APRESENTAÇÃO

A Engenharia Mecânica pode ser definida como o ramo da engenharia que aplica os princípios de física e ciência dos materiais para a concepção, análise, fabricação e manutenção de sistemas mecânicos. O aumento no interesse por essa área se dá principalmente pela escassez de matérias primas, a necessidade de novos materiais que possuam melhores características físicas e químicas e a necessidade de reaproveitamento dos resíduos em geral. Além disso a busca pela otimização no desenvolvimento de projetos, leva cada vez mais a simulação de processos, buscando uma redução de custos e de tempo.

Neste livro são apresentados trabalho teóricos e práticos, relacionados a área de mecânica, materiais e automação, dando um panorama dos assuntos em pesquisa atualmente. A caracterização dos materiais é de extrema importância, visto que afeta diretamente aos projetos e sua execução dentro de premissas técnicas e econômicas. Pode-se ainda estabelecer que estas características levam a alterações quase que imediatas no projeto, sendo uma modificação constante na busca por melhores respostas e resultados.

De abordagem objetiva, a obra se mostra de grande relevância para graduandos, alunos de pós-graduação, docentes e profissionais, apresentando temáticas e metodologias diversificadas, em situações reais. Sendo hoje que utilizar dos conhecimentos científicos de uma maneira eficaz e eficiente é um dos desafios dos novos engenheiros.

Boa leitura.

Henrique Ajuz Holzmann
João Dallamuta

SUMÁRIO

CAPÍTULO 1 1

EFICIÊNCIA E CONSUMO ENERGÉTICO: UMA ANÁLISE COMPARATIVA NO SETOR AUTOMOTIVO BRASILEIRO

Rafael Guimarães Oliveira dos Santos

Aloísio Santos Nascimento Filho

https://doi.org/10.22533/at.ed.8292128101

CAPÍTULO 2 8

ANÁLISE COMPARATIVA DOS TIPOS DE ÁGUA E ELETRÓLITOS DE UM GERADOR DE HIDROGÊNIO PARA UM VEÍCULO CONVENCIONAL

Thiago Gonçalves de Oliveira

Fábio Luís Figueiredo Fernandes

https://doi.org/10.22533/at.ed.8292128102

CAPÍTULO 3 19

A REVIEW ON ITERATIVE AND SERIES SOLUTIONS FOR KEPLER’S EQUATION

Paula Cristiane Pinto Mesquita Pardal

Mariana Pereira de Melo

João Francisco Nunes de Oliveira

Leonardo de Oliveira Ferreira

Pedro Novak Nishimoto

Roberta Veloso Garcia

https://doi.org/10.22533/at.ed.8292128103

CAPÍTULO 4 36

A-CAES (ADIABATIC COMPRESSOR AIR ENERGY STORAGE):APARATO EXPERIMENTAL EM ESCALA LABORATORIAL

Roberto Sihnel

Thiago Antonini Alves

https://doi.org/10.22533/at.ed.8292128104

CAPÍTULO 5 51

LA IMPORTANCIA DE LA FACTORIZACIÓN EN LA INGENIERÍA MECÁNICA PARA DETERMINAR LA ALTURA MÁXIMA DE PRESIÓN EN LAS TURBOBOMBAS

Juan Antonio Tena Verdejo

Francisco Santiago Gabino

Sandra Zulema Tena Galván

Víctor Francisco Cortes Ávila

José Salvador

Oropeza Ramírez

https://doi.org/10.22533/at.ed.8292128105

CAPÍTULO 6 59

MODELO MATEMÁTICO DEL COMPORTAMIENTO DE LA ENERGÍA CINÉTICA DEL

ÁLABE DE UN IMPULSOR CERRADO DE LAS TURBO BOMBAS
Juan Antonio Tena Verdejo
Francisco Santiago Gabino
Sandra Zulema Tena Galván
Víctor Francisco Cortes Ávila
https://doi.org/10.22533/at.ed.8292128106

CAPÍTULO 7..........68

PROJETO DE TÚNEL DE VENTO DE BAIXA VELOCIDADE
Arthur de Lima Queiroga
Rhander Viana
Olexiy Shynkarenko
https://doi.org/10.22533/at.ed.8292128107

CAPÍTULO 8..........79

ANÁLISIS Y SIMULACIÓN DE UN MECANISMO DE SCOTCH-YOKE A TRAVÉS DE LA INTERFAZ SOLIDWORKS-MATLAB
Javier Guevara Rivera
Adolfo Manuel Morales Tassinari
María Esperanza Velasco Ordóñez
Carlos Efrén Jiménez Acosta
https://doi.org/10.22533/at.ed.8292128108

CAPÍTULO 9..........88

COMPARATIVO ENTRE UM OSSO COM PROPRIEDADES MECÂNICAS ISOTRÓPICAS E ORTOTRÓPICAS PELO MÉTODO DOS ELEMENTOS FINITOS EM UMA FRATURA
Igor Emanoel Espindola Loureiro
Celso Júnio Aguiar Mendonça
Ivan Moura Belo
https://doi.org/10.22533/at.ed.8292128109

CAPÍTULO 10..........100

ANALYSIS OF DELAY COMPENSATION METHODS IN HARDWARE-IN-THE-LOOP TESTS
Eduardo Moraes Coraça
Janito Vaqueiro Ferreira
https://doi.org/10.22533/at.ed.82921281010

CAPÍTULO 11..........113

METODOLOGIA SEIS SIGMA: APLICAÇÃO NUMA EMPRESA DE ROLHAS METÁLICAS DO POLO INDUSTRIAL DE MANAUS
Edry Antonio Garcia Cisneros
Daniel Guzmán del Río
Israel Gondres Torné
Vitor Hugo Machioly
https://doi.org/10.22533/at.ed.82921281011

CAPÍTULO 12..127

PROPOSAL OF AN ELECTRONIC BRACELET DEVICE FOR THE MEASUREMENT OF VITAL SIGNS

Eliel Eduardo Montijo-Valenzuela
Elvis Osiel Covarrubias-Burgos
Darío Soto-Patrón
Esthela Fernanda Torres-Amavizca

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SOBRE OS ORGANIZADORES 137

CAPÍTULO 10

ANALYSIS OF DELAY COMPENSATION METHODS IN HARDWARE-IN-THE-LOOP TESTS

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ABSTRACT: Computer simulations are performed in order to analyze systems dynamic behavior. However, some subcomponents may be too complex to be modeled, which occurs due to nonlinearities and coupling of subsystems. In this scenario, a hardware-in-the-loop (HiL) test can be done, which consists of separating the system into two subsystems: a physical one, containing the complex component, and a numerical one, to be simulated. The numerical subsystem calculates the variable to be imposed on the hardware, such as a displacement. The hardware responds to the input and its response is measured and sent back to the numerical subsystem. Thus, the HiL test is a real-time closed-loop system, which has been used in the automotive area in applications such as anti-lock brake systems and semi-active suspensions. In order to close the loop, a transfer system composed of actuators and sensors is required. Since it is a dynamic system, the transfer system cannot respond instantaneously to a command, causing a time lag that deteriorates the closed-loop response and may lead to an unstable condition. In this paper, a HiL test is performed on a mass-spring-damper system using the spring as the hardware. Methods for delay compensation are presented, simulated and compared experimentally.

KEYWORDS: hardware-in-the-loop, control, dynamics, delay

1 I INTRODUCTION

In the development stage of new technologies, system simulation before its conception is indispensable, leading to lower cost with prototypes and safer product validation. However, a mathematical model of the system of interest may not be easily obtained, since non-linearities and complexities may be present. In this context, a Hardware-in-the-Loop (HiL) test may be employed, which consists of dividing the overall systems into two subsystems, one that is a numerical model (the software) and the other that is the physical component of interest (the hardware). With actuators and sensors, the two subsystems are coupled and a real-time simulation can be employed, thus analyzing the overall system dynamics. Figure 1 shows a generic HiL simulation block diagram. Signal is the external situation imposed in the system, like a force or displacement, and signal is the physical subsystem's measured response. Signal is the numerical subsystem input, which is calculated based on signals and . The software output is converted to a continuous signal through a D/A converter which is then sent to an actuator, who will impose it to the physical component through signal . The component's response is measured by a sensor, yielding , which is converted through an A/D converter to . From this analysis it is clear that the HiL test is a closed-loop real-time system.

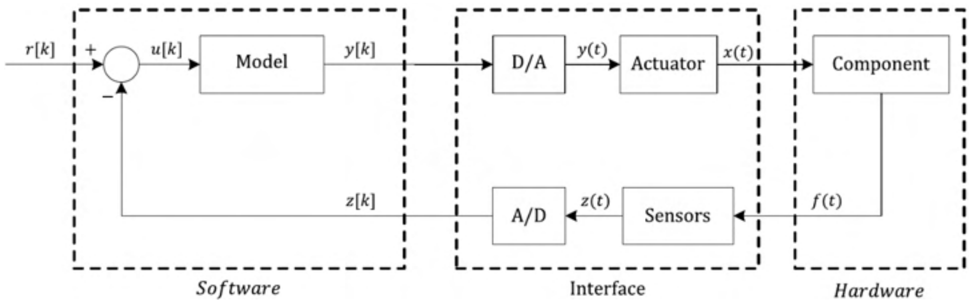

Figure 1: Typical HiL test block diagram.

HiL testing begun in aerospace industry, through flight simulators and missile location systems (BACIC, 2005). In the past 20 years, the technique has been used mainly in automotive applications. Svenson *et al.* (2009) developed a HiL simulator in order to test heavy trucks stability control systems, where the braking system and its controllers were the hardware and, as software, the truck's model was employed. One of the main applications of HiL is within suspension systems testing. Hong *et al.* (2002) used HiL test to evaluate a Skyhook control method, developed by the authors, for McPherson semi-active suspensions. The hardware in this situation was the variable damper, which is a very non-linear component, while the controller module and the other suspension components were the modeled software. The main problem that arise from HiL simulations is a time delay inherent to the interface between software and hardware. Since the actuator cannot respond instantaneously to a command, there is an inevitable time delay which is inside the closed-loop system. Hence, the response presents an error and, depending on the amount of delay and in dynamic characteristics of the subsystems, it may also lead to instability. Delay compensation methods were proposed and many works on this area were performed in order to improve HiL results, but no method is perfect. The Forward Prediction Method (WALLACE, 2005) adjusts a polynomial to the past calculated output values of the software and extrapolates this curve, forward predicting the signal's value in the future, in a time equal to the time delay. Thus, the closed-loop system is compensated.

Another method was proposed by Smith (1959), but not for the HiL context. Known as the Smith Predictor, the technique uses a model of the system under test in order to predict its future response, thus compensating the time delay. However, a model of the plant is not available when HiL test is employed. By employing an approximation of the plant, errors arise, thus this is not the most indicated method. A similar idea was proposed by Gawthrop *et al.* (2008) for real-time simulations, where a model is needed but, again, this is not the best solution for this application. A different approach where an analysis of the discrete operations performed by the acquisition boards was employed in order to characterize the time delay (CARRILLO, 2012). By choosing the simulation's sample time as larger than the time delay, the author noticed that the equivalent delay becomes equal to one time step, which allows for a different kind of correction based on the board's tasking order alteration. However, a high sampling period may not be suitable for some applications, leading even to stability issues (CARRILLO, 2014).

In this paper a Hardware-in-the-Loop test is applied to a mass-spring-damper system. In order to investigate how the time delay affects the system's dynamics, the hardware is a spring, because it can be easily identified and modeled. Delay compensation methods are presented, applied and, because the hardware is parametrized, it is possible to compare the results to a reference response, thus validating the experiment. A new compensation method is then proposed and compared to the previous ones. Numerical simulations are run by using a virtual spring and, then, experiments are conducted with the real spring by using MATLAB/Simulink software, dSPACE acquisition board and SCHENCK hydraulic actuator.

2 | EXPERIMENTAL SETUP

The following equipment was used:

- MATLAB/SIMULINK software, to model the numerical subsystem;
- dSPACE DS-1102 board, which contains A/D and D/A converters and the ControlDesk software to monitor the signals;
- Instron/SCHENCK actuator system, used for displacement control, equipped with a LVDT sensor for displacement sensing and load cell for force sensing, shown in Figure 2.

Figure 2: Instron/SCHENCK actuator.

The dSPACE board works as a zero-order holder system, which means that the digital input signal in the D/A board is held constant during one sampling interval.

3 I SYSTEM MODELING

A mass-spring-damper system is analyzed, with the spring being the physical component.

3.1 Full mass-spring-damper model

The mass-spring-damper system is indicated in Figure 3a, where is the weight force in N, which refers to mass in kg, is the stiffness of the spring in N/m and is the viscous damping parameter in Ns/m. Figure 3b shows a free-body diagram for this problem, where and are the forces due to the spring and damper, respectively, in N.

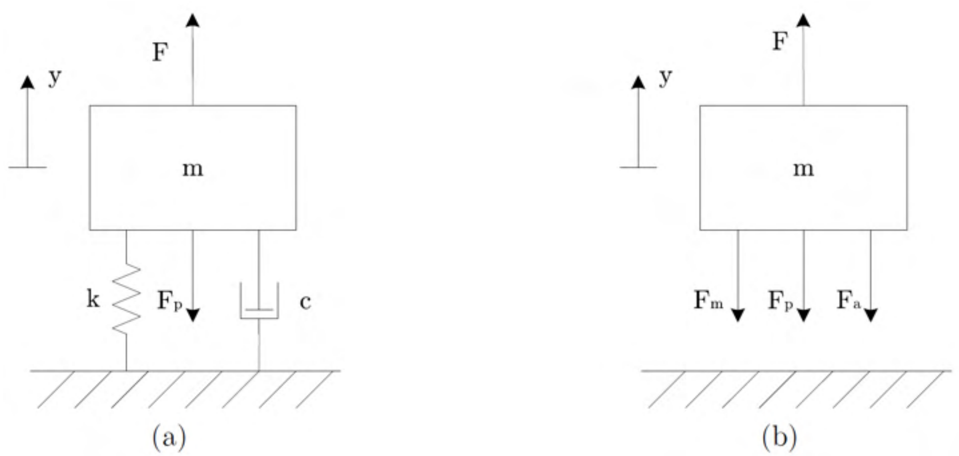

Figure 3: Model of the system under analysis. (a) Mass-spring-damper, (b) free-body diagram.

The equation of motion is then given by:

$$m\ddot{y} + c\dot{y} + ky = F - F_p. \tag{1}$$

In order to represent the system in state-space form, these equations need to be re-written in terms of states as indicated by Equations 2 and 3, where is the state vector, contains the system's outputs and represents the inputs.

$$\dot{\boldsymbol{x}} = \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}\boldsymbol{u}, \tag{2}$$

$$y = \boldsymbol{C}\boldsymbol{x} + \boldsymbol{D}\boldsymbol{u}. \tag{3}$$

The following states are adopted: $x_1 = y$ and $x_2 = \dot{y}$. The system's input is $\boldsymbol{u} = F - F_p$

while the output is displacement, $y = x_1$. This leads to:

$$\dot{x} = \begin{bmatrix} 0 & 1 \\ -k/m & -c/m \end{bmatrix} x + \begin{bmatrix} 0 \\ 1/m \end{bmatrix} u, \tag{4}$$

$$y = [1 \quad 0]x + [0]u. \tag{5}$$

3.2 HiL model with physical spring

In this case, the term $\mathcal{F}_m$, corresponding to the spring restoring force, is modeled as an input, which implies that $u = F - F_p - F_m$. Then, Equation 1 can be re-written in state-space form, adopting the same state vector, as:

$$\dot{x} = \begin{bmatrix} 0 & 1 \\ 0 & -c/m \end{bmatrix} x + \begin{bmatrix} 0 \\ 1/m \end{bmatrix} u, \tag{6}$$

$$y = [1 \quad 0]x + [0]u. \tag{7}$$

3.3 Hardware identification

Knowledge about the spring, sensor and delay characteristics is needed in order to validate the HiL simulation results. A displacement sine wave was imposed to the spring and the force/displacement behavior is shown in Figure 4(a). Since non-linearity was noted, a third order polynomial was adjusted to the curve, which is described by Equation 8, where the unit of is kN and the unit of is mm. Note that for the spring is under compression.

$$F_m = -1.28x10^{-7}y^3 - 3.441x10^{-5}y^2 + 0.01238y - 0.01173. \tag{8}$$

The system delay is defined as the time needed by the actuator to execute a commanded displacement and it was modeled as a transport delay, as is commonly done in the literature, although other methods such as Padé approximation could be used. By inputting a sine wave and measuring the response, it was possible to quantify the time delay as $\mathcal{T}'_{\mathcal{D}} = 6$s. Figure 4(b) shows the actuator's behavior due to the input sine wave with a zoom applied in order to visualize it.

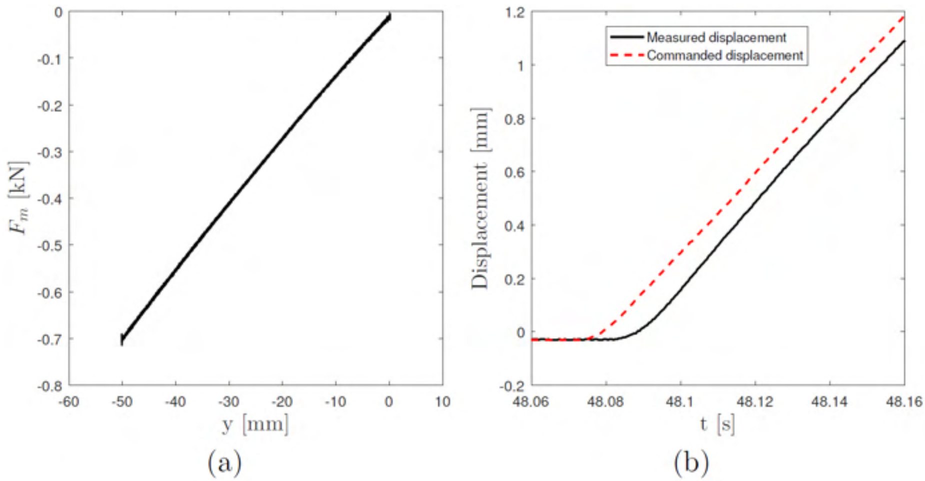

Figure 4: System characteristics. (a) Force and displacement behavior of the spring under test, (b) Delay between commanded and executed displacement.

Finally, in order to determine the gains involved in the voltage to displacement conversion and vice-versa, a voltage sine wave was generated in SIMULINK, sent to the D/A board with the help of ControlDesk software and then sent to the actuator, which led to a 40 mm/V gain. Then, by measuring force rather than displacement, a 10 kN/V gain was determined for the force to voltage conversion.

4 | DELAY COMPENSATION METHODS AND SIMULATIONS

Three different methods are presented and simulated in this work. The following subsections describe their functionalities.

4.1 Tasking order sequence alteration method

The response delay is exactly T_s' when $T_s' > T_D'$, where T_s' is the sampling time. The discrete state equations are given below, where the subscript in the matrices represent their discrete equivalent (CARRILLO, 2014).

$$\boldsymbol{x}[k+1] = \boldsymbol{A}_d\boldsymbol{x}[k] + \boldsymbol{B}_d\boldsymbol{u}[k], \tag{9}$$

$$y[k] = \boldsymbol{C}_d\boldsymbol{x}[k] + \boldsymbol{D}_d\boldsymbol{u}[k]. \tag{10}$$

By analyzing the output equation in time step $\mathcal{K}$, it can be seen that the calculation of $y[\mathcal{K}]$ depends on $\boldsymbol{x}[k]$ and $\boldsymbol{u}[k]$. For this implementation, the tasking order states that vector $\boldsymbol{y}[k] = f(\boldsymbol{x}[k], \boldsymbol{u}[k])$ is calculated before the state update equation $\boldsymbol{x}[k+1] = f(\boldsymbol{x}[k], \boldsymbol{u}[k])$.

However, as there is one time step delay when measuring the physical subsystem's reaction force, the state equations input is $\boldsymbol{u}[k] = F[k] - F_P - F_m[k-1]$, which leads to an incorrect value for **u** [k]. The correction lies on the tasking order alteration where the state vector is updated before the output equation, which means that $\boldsymbol{x}[k] = f(\boldsymbol{x}[k-1], \boldsymbol{u}[k-1])$ is calculated and, then, $\boldsymbol{y}[k]$ is computed. Note that, in order to achieve $\boldsymbol{u}[k-1]$, the input force must also be delayed. This method is valid as long as the output $\boldsymbol{y}[k]$ does not depend on $\boldsymbol{u}[k]$, so matrix $\boldsymbol{D_d}$ must be zero, and it is given by:

$$\boldsymbol{x}[k] = \boldsymbol{A_d}\boldsymbol{x}[k-1] + \boldsymbol{B_d}\boldsymbol{u}[k-1], \tag{11}$$

$$y[k] = \boldsymbol{C_d}\boldsymbol{x}[k]. \tag{12}$$

4.2 Polynomial extrapolation method

This method consists of using the previous values of a signal $y(t)$ to predict its future value (WALLACE, 2005). In order to do so, an $\mathcal{N}$-th order polynomial is calculated each time step by using and its previous values $y[k-1], y[k-2], \ldots, y[k-(n-1)]$, where is the number of data points and it is related to the polynomial order according to $n = \mathcal{N} + 1$. The value of $\boldsymbol{y}$ predicted time steps ahead is denoted $y' = y[k+P]$ and is given by:

$$y'[k] = [a_0 \quad \cdots \quad a_N]\begin{bmatrix} y[k] \\ \cdots \\ y[k-N] \end{bmatrix}, \tag{13}$$

where each a_i term is assembled into vector $\boldsymbol{a}$, which is given by:

$$\boldsymbol{a} = [1 \quad PT_S \quad \ldots \quad P^N T_S^N]\begin{bmatrix} 1 & 0 & \cdots & 0 \\ 1 & -T_S & \ddots & (-T_S)^N \\ \vdots & \vdots & & \vdots \\ 1 & -(n-1)T_S & \cdots & (-(n-1)T_S)^N \end{bmatrix}^{-1} \tag{14}$$

4.3 Taylor series method

A novel method is proposed here based on the Taylor series, which approximates a function $f(x)$ near a point a based on values of its derivatives evaluated at this point, as indicated below, where is the -th derivative of $f(x)$.

$$f(x) \approx \sum_{i=0}^{n} \frac{f^{(n)}(a)}{n!}(x-a)^n. \tag{15}$$

When applied to a real-time simulation, the objective is to predict the numerical subsystem output at a future time $y\ (t + \mathcal{T})$, where t is the current time. By adopting $a = t$ and $x = t + \mathcal{T}$ on Equation 15, the prediction is given by:

$$y(t+T) \approx y(t) + y^{(1)}(t)T + \cdots + \frac{y^{(n)}(t)T^n}{n!}. \tag{16}$$

By modeling the numerical subsystem on state-space form, one has access to $y(t)$, $y^{(1)}\ (t)$ and $y^{(2)}\ (t)$ through the states and inputs of the system. It is known that $y(t) = x_1(t)$ and $y^{(1)}\ (t) = x_2\ (t)$. The second derivative can be obtained from the input u and the states. Through Newton's Second Law, it is known that $y^{(2)} = \dot{x}_2 = u/m - cx_2/m$. By substituting the previous expressions into Equation 16, the predictor's expression is given by:

$$y(t+T) = x_1(t) + x_2(t)T + \left(\frac{u(t)}{m} - \frac{cx_2(t)}{m}\right)\frac{T^2}{2} + \epsilon(t), \tag{17}$$

where $\in(t)$ represents the truncation error. The discrete version of the predictor is given below:

$$y'[k] = x_1[k] + x_2[k](T_S P) + \left(\frac{u[k]}{m} - \frac{cx_2[k]}{m}\right)\frac{T^2}{2} + \epsilon[k]. \tag{18}$$

Adopting $\boldsymbol{C}' = [1 \quad T_S P - c(T_S P)^2/(2m)]$ and $D' = (T_S P)^2/(2m)$, Equation 18 can be written in matrix form as:

$$y'[k] = \boldsymbol{C}'\boldsymbol{x}[k] + D'u[k] + \epsilon[k]. \tag{18}$$

4.4 Numerical simulations

In order to compare the methods listed before, numerical HiL simulations were conducted by using a model of the spring as the virtual hardware. The effect of the sampling time was analyzed through two simulation setups: Setup 1 with $\mathcal{T}_S > \mathcal{T}_D$ and Setup 2 with $\mathcal{T}_S < \mathcal{T}_D$. Block diagrams were created in SIMULINK in order to simulate the HiL test with delay. The one used for $\mathcal{T}_S > \mathcal{T}_D$ is shown in Figure 5a, while the one used for $\mathcal{T}_S < \mathcal{T}_D$ is shown in Figure 5b. When adopting $\mathcal{T}_S > \mathcal{T}_D$, the effective delay becomes one time step, which is modeled by the 1/z block. For $\mathcal{T}_S < \mathcal{T}_D$ the delay was modeled as being proportional to $\mathcal{T}_S$ by a factor of d. The $\mathcal{K}_o$ term refers to the value of $\mathcal{F}_m(x = 0)$ from Equation 8. An uniformly distributed random wave signal with amplitude $\mathcal{A}_\mathcal{F}$ is the external force acting on the mass. Physical parameters chosen for the simulations are m = 50kg, $\mathcal{F}_\mathcal{P}$ = 490.5 N, $\mathcal{A}_\mathcal{F}$ = 100 N, $\omega_\mathcal{F}$ = rad/s. For the damping characteristic, it was used c = 600 Ns/m for Setup 1 and c = 200 Ns/m for Setup 2.

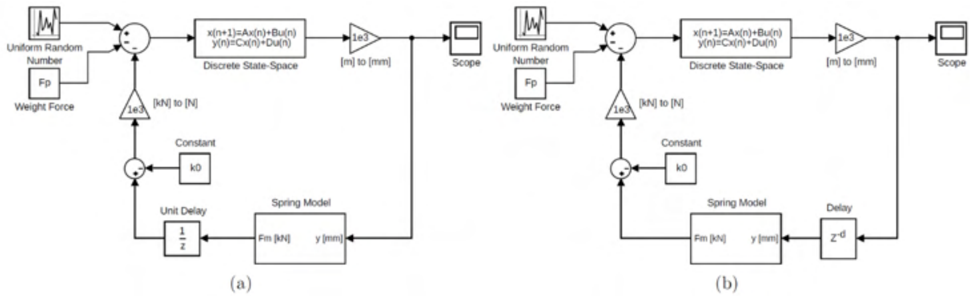

Figure 5: Block diagrams used for numerical simulations for (a) Setup 1 and (b) Setup 2.

The time delay used for the simulations was equal to the measured one. $\mathcal{T}'_S$ = 0.05 s was chosen for Setup 1 and $\mathcal{T}'_S$ = 2 ms for Setup 2. Figures 6a and 6b show the results for Setups 1 and 2, respectively, compared to the non-delayed system.

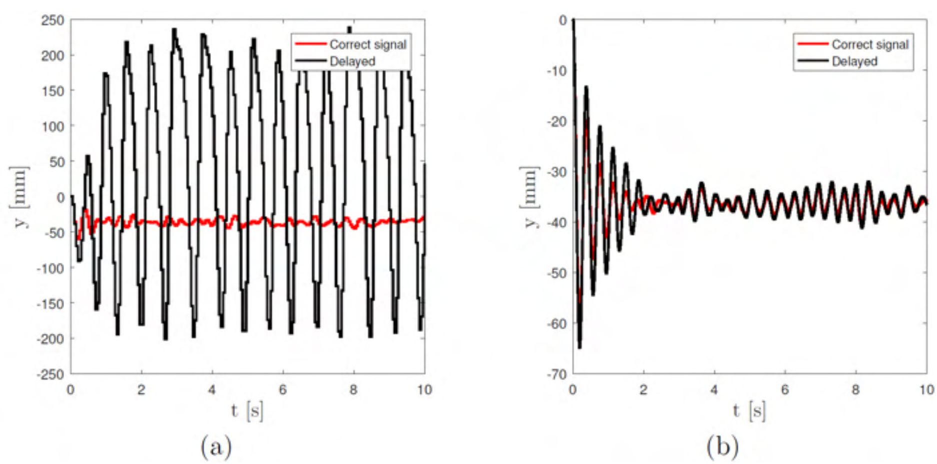

Figure 6: Simulation results for (a) Setup 1 and (b) Setup 2.

The delay compensation methods were implemented on SIMULINK. A one time step delay was used because the method was derived considering $\mathcal{T}'_S > \mathcal{T}'_D$. In order to simulate Setup 2, the block ‘Unit Delay’ is replaced by a general ‘Delay’ block. Figures 7a, 7b and 7c show the simulation results for Setup 1 using the Tasking Order Sequence Alteration, Polynomial Extrapolation and Taylor Expansion Methods, respectively, and Figure 7d shows the difference between all the previous compensated signals compared and the correct one. It can be seen that the Tasking Order Sequence Alteration Method provides, in simulation level, a perfect delay compensation. But it must be noted that it is valid when $\mathcal{T}'_S > \mathcal{T}'_D$ and **D** = 0. Thus, its use may not be suitable due to sampling period limitation, since it cannot be used for HiL testing of rapidly varying systems. For the Polynomial Extrapolation and Taylor Expansion Methods, it is possible to notice that the second one achieved a good result, while the first did not. This fact indicates a limitation of the polynomial method related to sampling time, while the Taylor method provides better results. Finally, the results for Setup 2 are given in terms of the difference between correct and compensated signals, as shown in Figure 8. It can be seen that the Tasking Order method presented a bad performance compared to the other methods, which are more suitable for situations where $\mathcal{T}'_{S} > \mathcal{T}'_{\mathcal{D}}$.

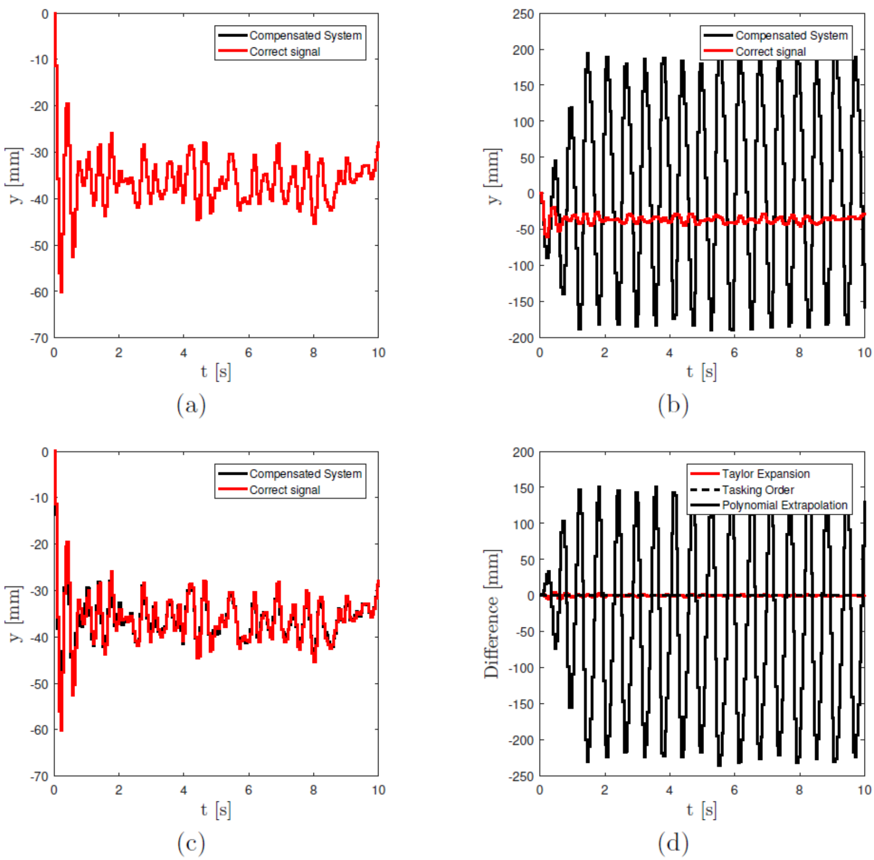

Figure 7: Simulation results for Setup 1. (a) Tasking Order Sequence Alteration Method, (b) Polynomial Extrapolation Method, (c) Taylor Expansion Method, (d) Difference between correct and compensated signal.

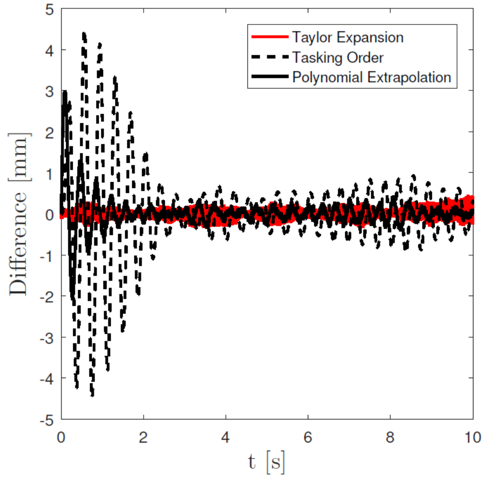

Figure 8: Difference between correct and compensated signal for all methods for Setup 2.

5 I EXPERIMENTAL RESULTS

The methods presented in the previous section were implemented on the real hardware. By using the equipment listed in Experimental Setup section, the HiL test was conducted. A damping of c = 300 Ns/m and time steps of 0.1 s and 2 ms were chosen. For $\mathcal{T}_s'$ = 0.1 s, Figure 9a shows a comparison between compensated, uncompensated and correct signals. Due to noise in the signals, the compensation method is not able to provide a perfect test, but it can be seen that the response was improved by employing the method. Figure 9b shows the results for the Polynomial Extrapolation Method with $\mathcal{T}_s'$ = 2 ms. The measured signal tracked the correct one almost perfectly when compared to the uncompensated response. For this case, a second-order polynomial was used. Finally, Figure 9c shows the test result when using the Taylor Expansion Method, which clearly differs from the correct result.

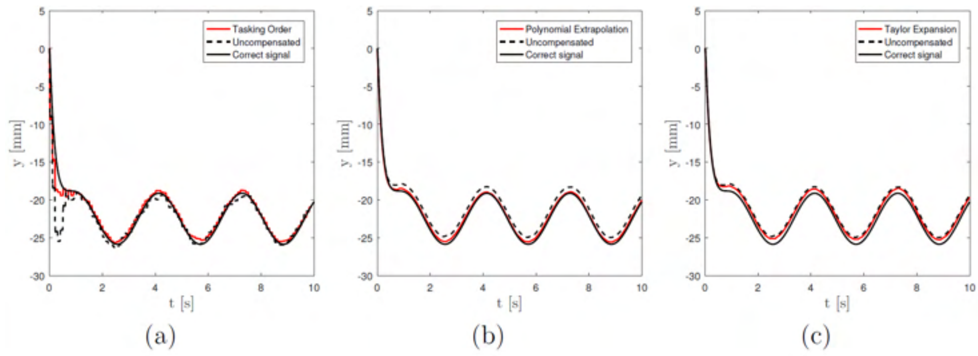

Figure 9: Experimental results with delay compensation methods. (a) Tasking Order Sequence Alteration Method, (b) Polynomial Extrapolation Method, (c) Taylor Expansion Method.

The compensated systems did not follow the simulation because, in a real scenario, there is measurement noise and, also, time delay may not hold constant. The Tasking Order Sequence Alteration Method presented good result. However, it can only be used when large time steps are chosen, which may not be possible in certain applications since it influences the stability of discrete systems. In addition, large time steps may lead to abrupt changes, as can be seen in the results for $t < 1$ s from Figure 9a. The Polynomial Extrapolation Method was the most efficient, due to the proximity between correct and compensated signals. A small steady state error can still be perceived, which may have occurred due noise and variable time delay. Although it was not investigated in this paper, a limit for the $\mathcal{T}_S/\mathcal{T}_D$ relationship may exist in order to maintain stability and accuracy, and it should be investigated in future works. Finally, the Taylor Expansion Method presented a very low performance compared to its excellent simulation behavior. This may have happened mainly due to the error estimation $\epsilon\,[\mathcal{K}]$, which is affected by noise in the force measurement, and also due to variable time delay. This method should be further improved in future works. Although not analyzed in this work, it is believed that the methods can be used for more complex systems including non-linearities and more degrees of freedom, since no approximation was done to derive such compensation techniques. Different test setups can also be used, such as different actuator and sensors, where the only difference would be regarding the amount of time delay and the dynamic characteristics of the sensors, which would have to be identified.

6 | CONCLUSIONS

In this paper, the Hardware-in-the-Loop simulation technique was presented, studied and implemented in a mass-spring-damper system, where the hardware chosen was an already parametrized spring, in order to validate the results. A time delay was identified and its effects on the closed-loop behavior were analyzed. It was shown the sampling rate of the discrete HiL system influences stability and the relation between delay and sampling time must be analyzed prior to the test. Two delay compensation methods, Polynomial Extrapolation Method and Tasking Order Sequence Alteration Method, were studied and implemented. A new technique, based on Taylor series, was presented and compared. In terms of numerical simulations of the HiL test, it was seen that the proposed method overcame the other two. In terms of experimental tests, the Polynomial Extrapolation Method showed better performance. New ways for estimating the truncation error and also studies of the measurement noise influence on the methods should be considered for future works. Since no approximation regarding the system's dynamics was made, the techniques can be employed for more complex systems and this should be tested in future works.

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ÍNDICE REMISSIVO

A

A-CAES 5, 36, 37, 38, 39, 40, 41, 42, 43, 44, 47, 48

Aerodinâmica 68, 69

Ar comprimido 36, 37, 38, 39, 40, 41, 42, 48, 50

Armazenamento 36, 38, 39, 40, 41, 44, 48, 49

B

Biomecânica 88

C

Cavernas 36, 39, 40, 41

Células de inovação 127

Ciência de dados 1

Confiabilidade 113, 115, 121

Consumo de combustível 1, 8, 10

D

Desenvolvimento sustentável 1, 2

E

Eletrólise 8, 9, 10, 11, 14, 18

Eletrólitos 5, 8, 9, 14, 17, 18

Energia 3, 8, 9, 11, 17, 36, 37, 38, 39, 40, 41, 42, 43, 45, 47, 48, 49, 50, 68, 70, 90

Energia limpa 8, 9

F

Fluidodinâmica computacional 68

Fluido incompressível 68

Fratura coronal 88, 89, 99

G

Geração de hidrogênio 8, 9, 10, 11, 17

H

Huntorf 36, 37, 39, 40, 41, 49

M

Medição de sinais vitais 127

Método dos elementos finitos 6, 88, 90, 98

Metodologia seis sigma 6, 113

O

Ortotrópico 88, 89, 92, 96, 97, 98

P

Paradas não planejadas 113, 114, 118, 119, 120, 122, 124, 125

Perdas metálicas 113

Pulseira eletrônica 127

R

Renovável 36, 38, 42, 48

Router CNC 68, 69, 70

S

Setor automotivo 5, 1, 2, 3, 7

Sistema remoto 127

T

Tipos de água 5, 8, 10, 11, 12, 17

Túnel de vento 6, 68, 70, 71, 74, 78

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