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# Off-line displacement error correction method for servo-hydraulic testers

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**Analysis and modelling**

## ABSTRACT

**Purpose:** This paper presents an approach towards improving the test rig performance in reproduction of random excitation to which a prototype hydraulic damper is subjected. The method is intended to be implemented as a software solution without modifications either in hardware or the settings of the servo-hydraulic tester. These are the conditions of fatigue tests when a specific signal sequence (block) is repeated until failure of the sample.

**Design/methodology/approach:** Experimental validation of the proposed correction method was conducted using a servo-hydraulic test rig and a transfer function inverse model which was identified based on the operational data.

**Findings:** The proposed method, both in the frequency and time domain, improves the tracking of the test signal and allows an accuracy of more than 95% to be gained using the best fit measure in the case of reproduction of coloured noise signals.

**Research limitations/implications:** It is possible to consider more advanced model-based methods for performing off-line error correction, e.g. state-space models.

**Practical implications:** The proposed method was validated and implemented in the LabVIEW® software to automatically perform the correction of the test signal before the test. The method was validated for the test rig with and without the tested object.

**Originality/value:** The paper proposes an off-line control strategy that improves the reproduction of the load signal in case of repeatable test sequences achieving more than 99% agreement between the applied and measured load.

**Keywords:** Off-line error cancellation; Signal profile correction; Hydraulic actuator performance

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## 1. Introduction

A crucial stage of hydraulic damper manufacturing is the validation process [1-2]. Hydraulic dampers are evaluated regarding comfort, noise and lifetime, using synthesized or road excitation data [3-4]. A segment excitation signal, repeating in-the-loop has to be accurately tracked by test rigs, which the most frequently are found application in the validation process of hydraulic damper prototypes. While performing either durability or comfort evaluation tests, obtaining repeatable and reproducible results is one of the most important steps in automotive hydraulic damper development [5]. In this case, the challenge is to apply the realization of the loading sequence to a hydraulic damper, minimizing the error between the reference and measured signals. Moreover, it is important in the automotive industry to standardize tests which are typically performed in a few locations using servo-hydraulic tests rigs from different manufacturers. The test results are affected by numerous contributors, such as calibration accuracy, properties of the actuator and oil, ambient and oil temperature, and finally by different control strategies applied for particular test rigs. In such a case, designing a robust control process and obtaining the same accuracy in reproducing test signals is extremely unlikely and might be problematic using conventional on-line control algorithms. In the literature, several papers [6] discuss off-line feedback or feed-forward controllers to tackle this problem. The controllers perform an iterative off-line correction process with the use of an inverse model of the test rig [7]. The model is identified using system identification techniques. However, this paper proposes a basic iterative method of off-line error correction which in each iteration obtains the difference between the reference (command) and measured (test rig response) signals. The error decreases gradually in a few iterations, achieving excellent tracking of random signals on a servo-hydraulic tester. When iterations are performed on the test rig, the standard PID-FF controller is set to the optimal setting that minimizes the error in the on-line mode. The method was validated with the use of the servo-hydraulic test rig and its model using different types of signals. The model allowed numerical tests to be performed without the influence of disturbances which are normally present during experimental tests, e.g. oil supply system. The paper presents the architecture of the servo-hydraulic test rig and briefly describes each component of its model. The significance of each physical parameter, as well as the degree to which each parameter influences the behaviour of the model, is discussed in Section 2. The problem of modelling control systems is also addressed in the paper. The simulations described in this paper were aimed primarily at obtaining a qualitative insight into the modelled vibration phenomena, and allowed the irrelevant effects to be disregarded in later stages of the model development process.

The remaining content of the paper is divided into 5 sections. Section 2 discusses a servo-hydraulic test rig and its control model, while Section 3 described the proposed method and its evaluation criteria. Section 4 illustrates and discusses the experimental and simulation results and Section 5 includes final thoughts and a summary of what is presented in the paper.

## 2. Servo-hydraulic test rig

Experimental tests were performed on a MTS-850 servo-hydraulic test-rig, equipped with an MTS-407 electronic controller (Fig. 1).

The test rig was used to excite a hydraulic damper and capture its dynamic characteristics, i.e. displacement vs. force. Data acquisition was performed with an 8-channel, 24-bits Analog Input card manufactured by National Instruments. The test rig is equipped with an oil supply system (the so-called power-pack) that provides a pressure of 20.5 MPa at a nominal flow-rate of 300 l/min. The actuator provides 50 kN force at the rod, while the maximum stroke is 250 mm at the maximum achievable velocity of 4.5 m/s. The actuator rod is coupled to the adapter, which transfers the movement to a hydraulic damper mounted on a test rig.

Fig. 1. A servo-hydraulic test-rig used in experimental investigations

The main components of the servo-hydraulic system with the closed-loop control are: MTS-407 controller, the hydraulic actuator with integrated displacement transducer (LVDT) in a piston-rod assembly and the three-stage servo-valve system (Fig. 2).

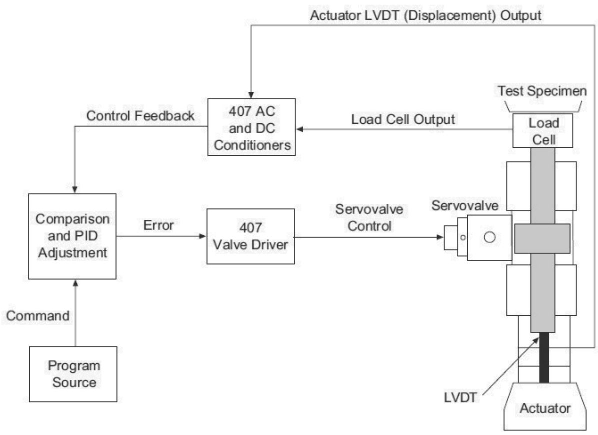


Fig. 2. MTS 850 servo-hydraulic system diagram [15]

The electronic controller uses a program command and sensor feedback to control the servo-valve. A test rig controller uses a group of gain controls, i.e. proportional (P), integral (I), derivative (D) and feed-forward (FF) gains (Fig. 3).

The MTS-850 machine is equipped with Moog G761 series servo-valve (Figs. 4-5). This industrial servo-valve consists of a polarized electrical torque motor and two stages of hydraulic power amplification. This valve is modelled by a transfer function whose input is a voltage and output is the spool displacement.

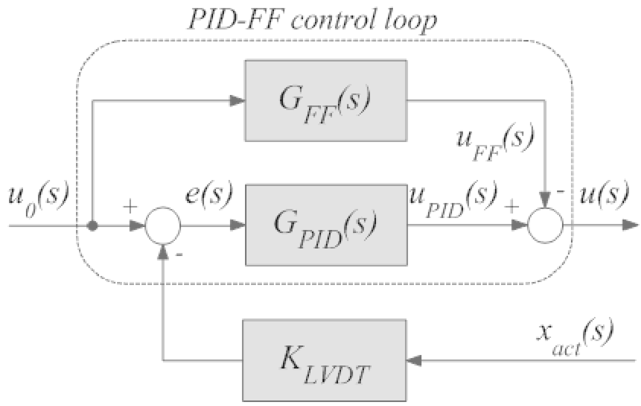


Fig. 3. PID control with feed-forward loop

The combined output of the PID-FF controller is determined as follow:

$$u = u_{PID} + u_{FF}\,, \tag{1}$$

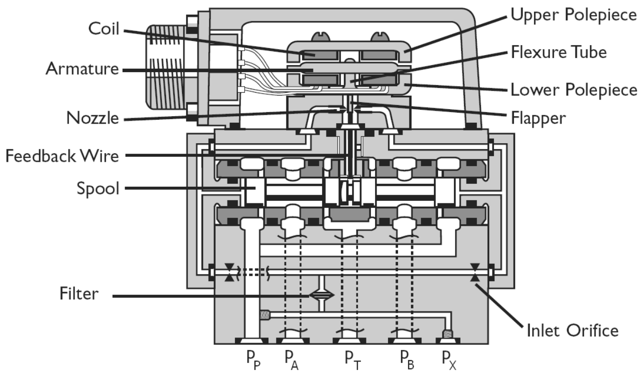


Fig. 4. The MOOG G761 servo-valve assembly [15]

The electromagnetic torque motor, driving the flapper, is controlled by an electrical current inputted from a current amplifier. A valve control input u (voltage) is converted into a current, with a current amplifier gain $K_e$. The hydrodynamic forces acting on the spool are neglected. Therefore, the spool position only depends on the input voltage

$$G_{servo} = \frac{x_{spool}(s)}{u(s)} = \frac{K_e \cdot K_{spool}}{1 + \left(\frac{2\zeta}{\omega_{n,servo}}\right)s + \left(\frac{s}{\omega_{n,servo}}\right)^2} \tag{2}$$

where

$$\omega_{n,servo} = \sqrt{\frac{k_{servo}}{m_{servo}}}; \quad \zeta = \frac{b_{servo}}{2}\sqrt{\frac{1}{k_{servo} \cdot m_{servo}}} \tag{3}$$

Spool movements are limited by the values of stiffness

$$k_{servo} = \begin{cases} k_{servo,\inf} & if \quad x_{spool} > x_{spool,\max} \\ k_{servo,nom} & if \quad x_{spool,\min} \le x_{spool} \le x_{spool,\max} \\ k_{servo,\inf} & if \quad x_{spool} < x_{spool,\min} \end{cases} \tag{4}$$

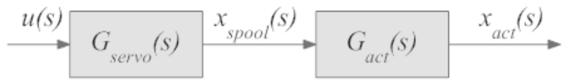


Fig. 5. Controlled elements loop

Finally, the displacement of the actuator $x_{act}$ which moves a mass M and have to overcome a damping force is determined as follow:

$$G_{act} = \frac{x_{act}(s)}{x_{spool}(s)} = \frac{1}{\left(\frac{M}{Ak_v}\right)s^2 + \left(\frac{k_d}{Ak_v}\right)s} \tag{5}$$

## 3. Evaluation of the proposed method

The proposed method considers no interference into the control system. All operations are performed only on an input command $u_0$. Signal correction can be performed off-line, so without using test-rig. A modified input command can be verified before using it, which is safer and less exploitable for testers. A chosen correction method is a model-based algorithm, which uses an identified test-rig model. The identification can be conducted by a single stimulus and machine response.

The correction algorithm used in research consists of 4 stages:

- multi-sine waveform stimulation,
- frequency response estimation,
- system identification,
- iterative learning control (ILC) algorithm application.

Multi-sine waveform is a kind of stimulation with flat PSD over some range of frequencies. In other words, it is a waveform that possesses harmonics of all frequencies from some range (harmonics within a frequency bin - discrete-time signal).

Frequency response function (FRF) presents system amplification and phase shift capabilities. The identification is conducted according to a standard procedure that contains the following stages:

- choosing model structure,
- estimation of model parameters,
- model validation,
- model approval (or disapproval and repeating the whole procedure with different model structure).

As a model structure, the ARX parametric model was chosen. It is described as follows:

$$x_{act}(z^{-1}) = z^{-k}\frac{B(z^{-1})}{A(z^{-1})}u_0(z^{-1}) + \frac{1}{A(z^{-1})}e(z^{-1}) \tag{6}$$

where

$$A(z^{-1}) = 1 + a_1 z^{-1} + \ldots + a_{dA} z^{-dA}$$
$$B(z^{-1}) = b_0 + b_1 z^{-1} + \ldots + b_{dB} z^{-dB} \quad (7)$$

According to [14] ILC is "a technique for improving tracking response in systems that repeat a given task over and over again. The control objective is to find the control input $u_0(t)$ so the corresponding output $x_{act}(t)$ precisely tracks some reference signal $u(t)$". ILC requires predefined input $u(t)$ and the algorithm must be repeated several times. For test-rig both conditions are met. The control law for ILC is given as follows:

$$u_{k+1}(t) = u_k(t) + L \cdot e_k(t), \quad (8)$$

# 4. Simulation and experimental results

The correction method was tested in terms of three aspects:
- correction effectiveness,
- influence of hydraulic damper on test,
- real-world test.

For test, three different excitation signals were used (Fig. 6):
- multi-sine stimulus,
- pink noise stimulus,
- stimulus, which was acquired from the car, driving on the road.

All results were evaluated by a signal fit parameter given by the formula (9). Results are presented in both: time and frequency domain.

$$R^2 = 1 - \frac{\frac{1}{N}\sum_{i=1}^{N} \left| x_{act}(i) - u_0(i) \right|^2}{\frac{1}{N}\sum_{i=1}^{N} \left| u_0(i) \right|^2}, \quad (9)$$

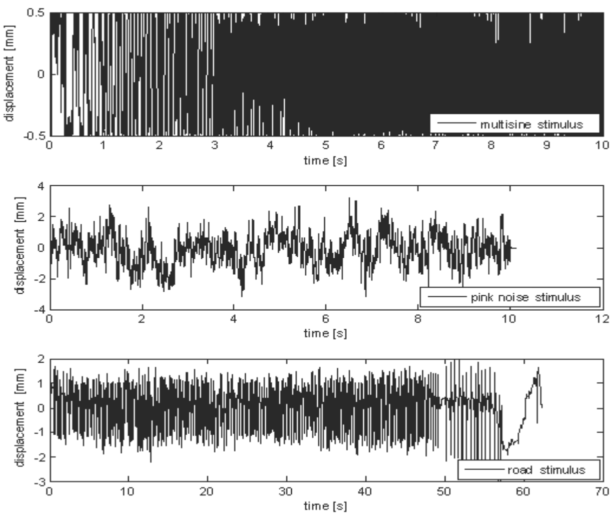


Fig. 6. Excitation signals used during the test

## 4.1. Multi-sine effectiveness

Effectiveness means how well a test-rig response is fitted to reference signal in a desirable frequency band. For the test 0-100 Hz multi-sine stimulus was used. Using an excitation with flattened PSD in specific frequency band assures a response with the same frequency band without an influence of other frequencies on test results. Graphical results of correction for multi-sine stimulus are presented in the Figs. 7-10. The iterative signal fit results are presented in Tables 1-2.

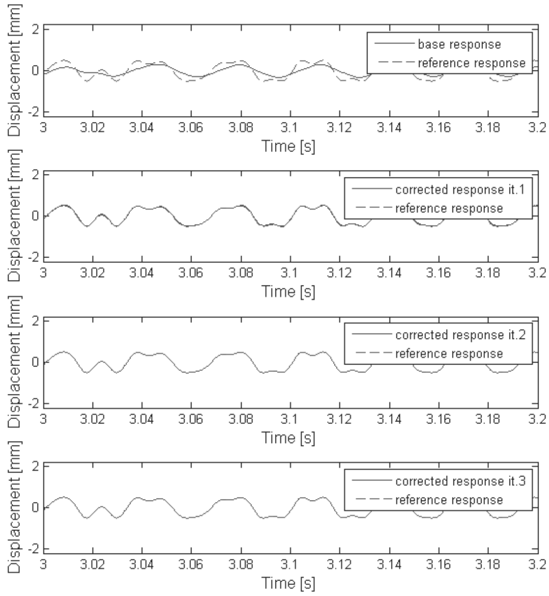


Fig. 7. Model response correction, multi-sine stimulus - time domain

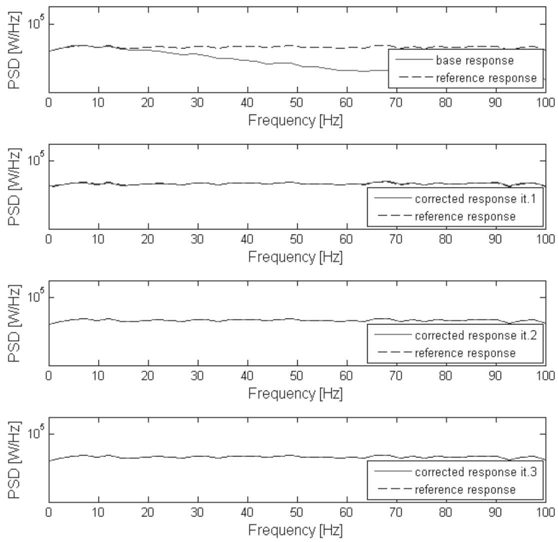


Fig. 8. Model response correction, multi-sine stimulus - frequency domain

Table 1.
Model response to reference response correlation ratio

| | ARX model response fit [%] |
|---|---|
| Initial | 33.95 |
| 1st iteration | 94.27 |
| 2nd iteration | 100.00 |
| 3rd iteration | 100.00 |

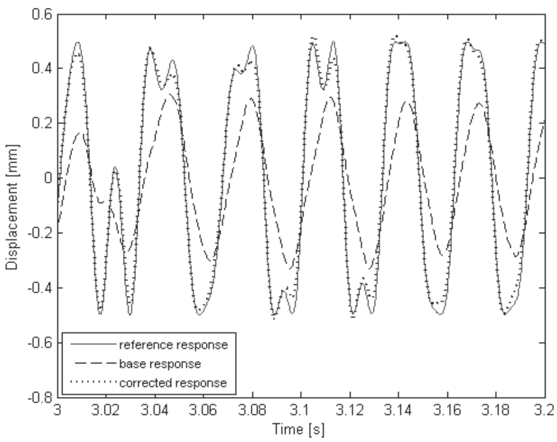


Fig. 9. Tester responses, multi-sine stimulus - time domain

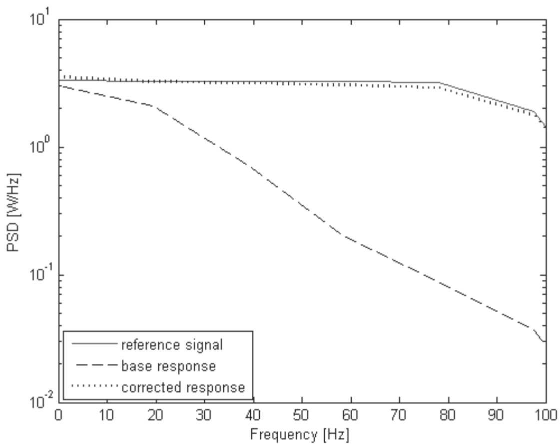


Fig. 10. Tester responses, multi-sine stimulus - frequency domain

Table 2.
Tester response to reference response correlation ratio

| | Test rig fit [%] |
|---|---|
| Initial | 33.95 |
| Corrected | 93.07 |

## 4.2. Colored noise stimulus

Graphical results of correction for pink noise stimulus are presented in the Figs. 11-14. The iterative signal fit results are presented in Tables 3-4.

Table 3.
Model response to reference response correlation ratio

| | ARX model response fit [%] |
|---|---|
| Initial | 72.99 |
| 1st iteration | 69.96 |
| 2nd iteration | 100 |
| 3rd iteration | 100 |

Table 4.
Tester response to reference response correlation ratio

| | Test rig fit [%] |
|---|---|
| Initial | 72.99 |
| Corrected | 88.21 |

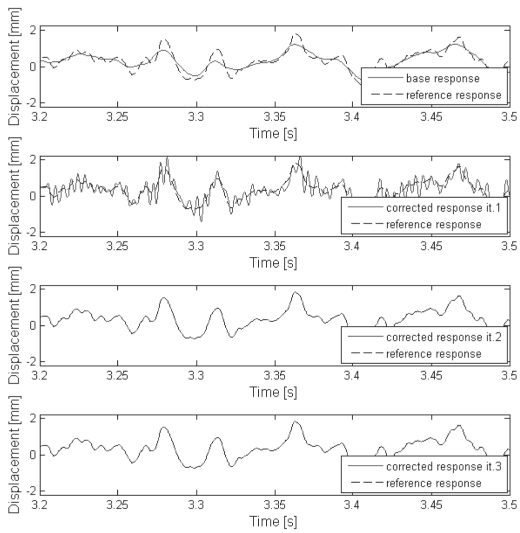


Fig. 11. Model response correction, pink noise stimulus - time domain

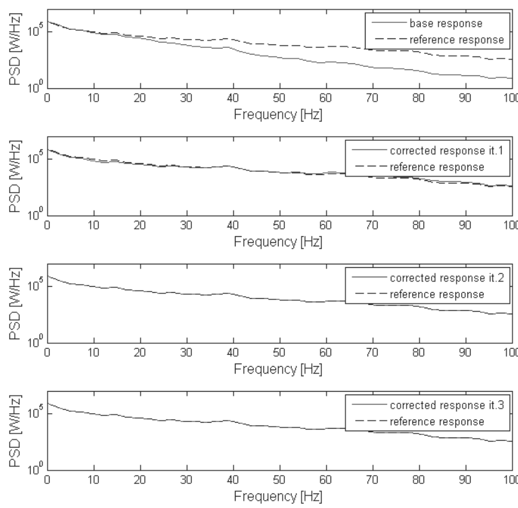


Fig. 12. Model response correction, pink noise stimulus - frequency domain

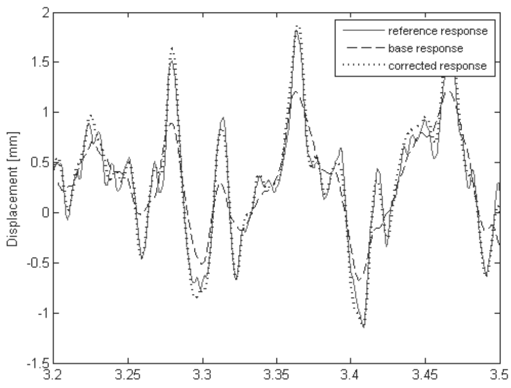


Fig. 13. Tester response correction, colored noise stimulus - time domain

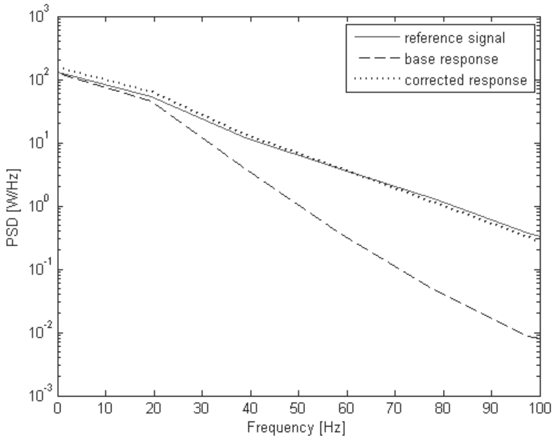


Fig. 14. Tester response correction, pink noise stimulus - frequency domain

## 4.3. Road stimulus

Graphical results of correction for road stimulus are presented in the Figs. 15-18. The iterative signal fit results are presented in Tables. 5-6.

Table 5.
Model response to reference response correlation ratio

| | ARX model response fit [%] |
|---|---|
| Initial | 94.65 |
| 1st iteration | 80.47 |
| 2nd iteration | 100 |
| 3rd iteration | 100 |

Table 6.
Tester response to reference response correlation ratio

| | Test rig fit [%] |
|---|---|
| Initial | 94.65 |
| Corrected | 96.41 |

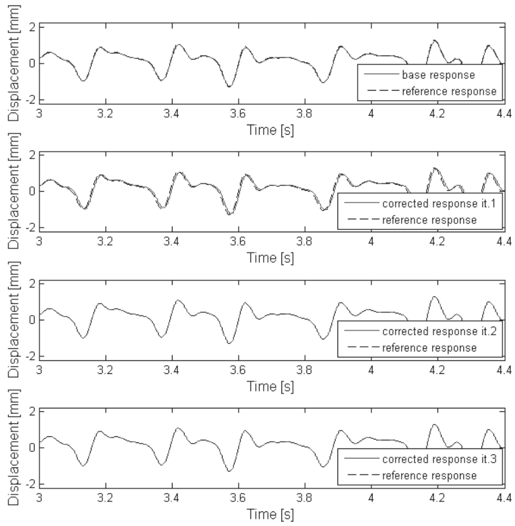


Fig. 15. Model response correction, road stimulus - time domain

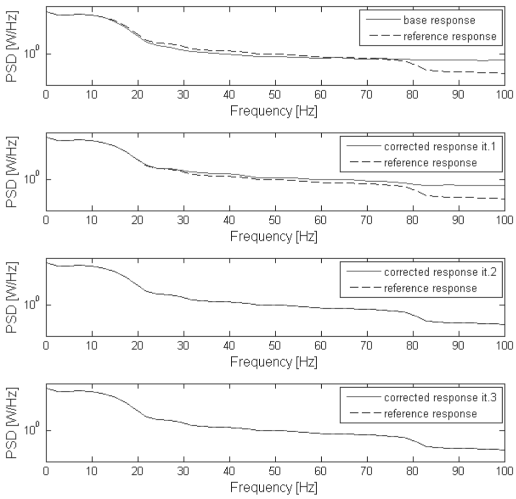


Fig. 16. Model response correction, road stimulus - frequency domain

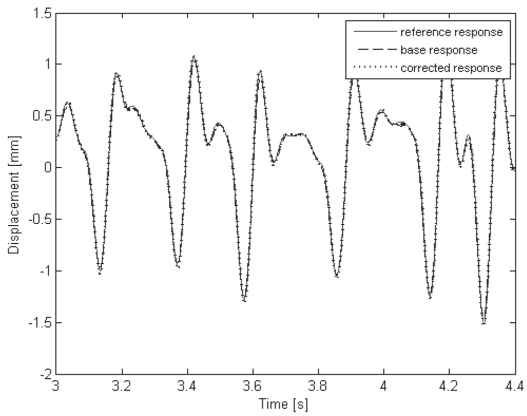


Fig. 17. Tester response correction, road stimulus - time domain

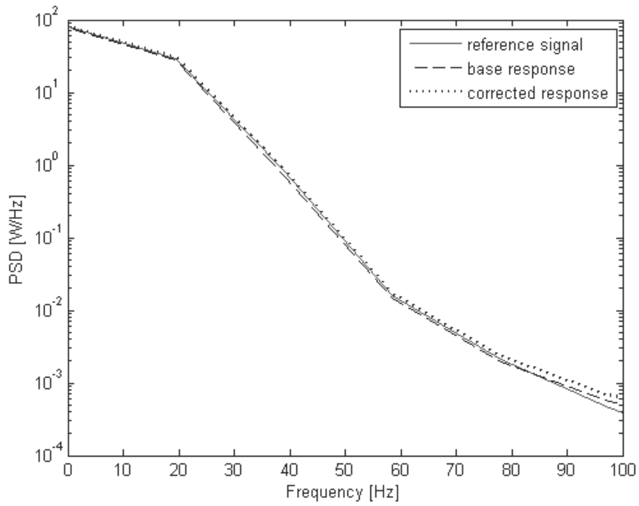


Fig. 18. Tester response correction, road stimulus - frequency domain

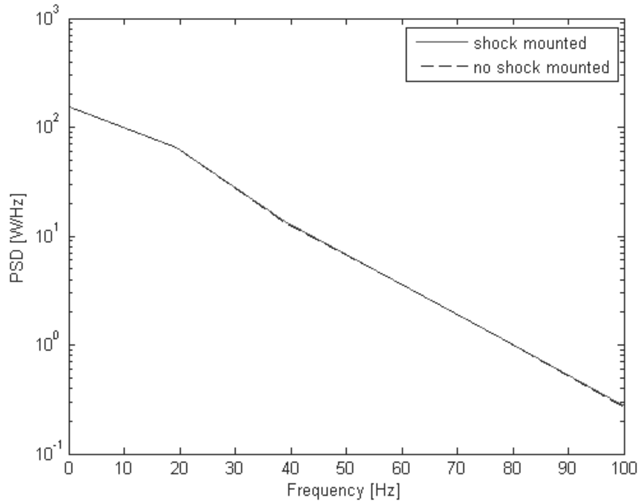


Fig. 20. Comparison of tester corrected response with and without mounted hydraulic damper, pink noise stimulus - frequency domain

## 4.4. Hydraulic damper influence

Graphical results of correction for pink noise stimulus with and without mounted hydraulic damper are presented in the Figs. 19-20. The iterative signal fit results are presented in Table 7.

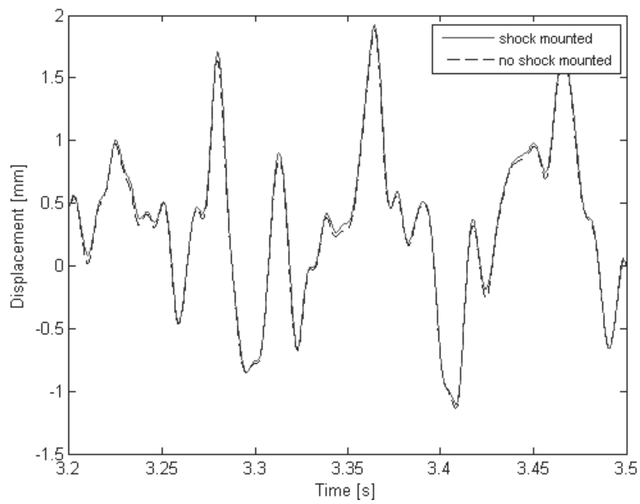


Fig. 19. Comparison of tester corrected response with and without mounted hydraulic damper, pink noise stimulus - time domain

Table 7.
Tester responses correlation ratio

| | Test rig fit [%] |
|---|---|
| Without hydraulic damper | 88.21 |
| With hydraulic damper | 87.87 |

## 5. Summary

This paper presents an approach towards improving the test rig performance for random testing signals. The goal is to develop a method for correction of the test signal profile in the off-line mode. The method is suitable in case of avoidance of hardware parameters changes (e.g. PID settings). The authors performed the feasibility study applying the non-parametric approach to improve tracking of the test signal on the servo-hydraulic test rig MTS-850 equipped with the electronic controller MTS-407 [15].

Tests demonstrate response improvement in relation to the reference response. The improvement is noticeable in time and frequency domain. According to performed tests results a correction up to 95% (relating to reference response) can be achieved in full desired frequency range.

Test proved that influence of hydraulic damper on dynamical parameters of test-rig is negligible. This allows identifying tester model and its further effective usage without shock presence. The ILC algorithm is fast-convergent. Three iterations are absolutely enough to reach fine results. More iterations did not improve signal. All correction procedure lasts at most few seconds what illustrates quick ILC performance.

The greatest influence on correction effects has model identification. A parametric ARX model is easy to implement, but is not the best to describe the tester model. Still, other and more advanced methods can be used to achieve better accuracy of model fit, e.g. using another parametric model, using neural networks for system identification, etc.

Another challenge of system identification is inverse model stability. Although the base model fits the test-rig very well, an inverse model is unstable what makes ILC performance senseless. To solve the problem, a method finding stable and minimum-phase models should be used (e.g. Wiener filter). A great improvement would be finding a method, which is capable of identifying model automatically, without manual setting of any parameters. In compliance with a method suggested in [13] all model zeros and poles are set manually. The identification method proposed in the master thesis needs always two parameters to set. These parameters are polynomial degrees - dA and dB.

Because of hydraulic actuator non-linearity a correction can be used successfully only around the operating point. This problem, however, is not essential now, because test-rigs are not able to drive signals with high frequency and high amplitude at once (velocity limit).

## Nomenclature

| | |
|---|---|
| $a$ | polynomial coefficient, |
| $A(z^{-1})$ | model denominator, |
| $b_{servo}$ | polynomial coefficient, |
| $B(z^{-1})$ | model nominator, |
| $dA$, $dB$ | polynomial degree, |
| $e(z^{-1})$ | discrete error, |
| $e_k(t)$ | error correction, |
| $k_{servo}$ | equivalent stiffness of spool spring, |
| $k_v$ | Amplification, |
| $L$ | inverse plant model, |
| $m_{servo}$ | spool mass, |
| $t$ | continues time domain, |
| $u(t)$ | spool voltage [V], |
| $u_0(t)$ | expected displacement of actuator [m], |
| $u_k$ | last corrected stimulus, |
| $u_{k+1}$ | new corrected stimulus, |
| $u_{PID}$ | output voltage from PID controller [V], |
| $u_{FF}$ | output voltage from feed forward gain [V], |
| $x_{spool}$ | displacement of servo valve spool [m], |
| $x_{spool,min}$ | lower position of servo valve spool [m], |
| $x_{spool,max}$ | upper position of servo valve spool [m], |
| $K_e$ | voltage-current amplifier proportional gain [A/V], |
| $K_{spool}$ | current-displacement proportional gain [m/A], |
| $x_{act}$ | displacement of actuator [m], |
| $M$ | mass of moving part of actuator (actuator piston, actuator rod, adapter) [m]. |

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