Accurate Modeling and Identification of Servo-Hydraulic Cylinder Systems in Multi-Axial Test Applications

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Abstract: Servo-hydraulic control is regarded as a key technology in most mechanical testing systems for the investigation of strength and fatigue properties of materials, mechanical components and structures. High performance but robust control strategies are necessary as part of the test control system to ensure high test quality and reproducible test results which are of special importance when considering multi-axial component tests using high channel count test rigs with several active and synchronously operating servo-hydraulic test cylinders. Such high performance multidimensional control algorithms demand sufficient information about the dynamic characteristics of the used test cylinders, the test rig and – at least as a good estimate – of the device under test. In this paper we introduce a special kind of identification strategy for deriving accurate mathematical models for a class of commonly used servo-hydraulic test cylinders: low-cost symmetric test cylinders with significant friction. In order to accurately describe the dynamics of the servo-hydraulic cylinder system, we first build a mathematical model of the system which includes some still unknown parameters. These unknown parameters are determined by utilizing a special identification scheme based on the black box identification method to estimate initial values of the unknown parameters, followed by the grey box identification method for accurately estimating the complete cylinder system model. The developed black-grey box identification method delivers accurate information on the cylinder dynamics. The comparison between simulation results and measured data from real world experiments demonstrates the high accuracy of the derived servo-hydraulic cylinder system model, and thus validates the effectiveness of the developed system identification method.

Keywords: Electro-hydraulic servo system, servo-hydraulic test cylinder, fatigue test, friction, system modeling, system identification, simulation

1 Introduction

Multi-axial testing systems with servo-hydraulic test cylinders and dedicated control systems are widely used in experimental mechanics and related test applications because of their advantages, such as high energy density, good linearity, little dead zone, high sensitivity, fine dynamic performance, fast response, high precision and so on. With further development of the electro-hydraulic servo control technology due to the higher demands on the industrial testing system performance, their application fields and ranges are expanding constantly. Consequently, the electro-hydraulic transmission and control systems become more and more complex and the requirements on their control algorithms become higher and higher. For instance, the duration and/or the costs of mechanical tests should remain low, but with increased test precision and reproducibility it is necessary to ensure high suppression of external disturbances and the ability to cope with non-linearities and time dependency in the actuator dynamics and the overall plant. Furthermore, regarding multi-axis electro-hydraulic servo control systems, severe coupling of the used control axes can appear in combination with non-ideal mechanical link elements. These effects have to be considered in the development and implementation of such test control systems and control strategies. It is further essential for any test engineer to understand the dynamic characteristics of the used test equipment and the plant in order to be able to setup such a test system with optimum dynamic response, control precision and working reliability.

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Throughout this paper, we describe a certain kind of system identification for building mathematical models for a commonly used class of servo-hydraulic test cylinder systems with significant friction. These test cylinder systems are composed of the cylinder itself, with a mechanically guided and sealed cylinder rod, the servo-valve and the distance and/or force measurement. Our proposed black-grey-box modeling process, based on system identification experiments, can build and parametrize an accurate model of the dynamic properties of these cylinder systems to enable sufficient actuator knowledge as demanded by high performance but robust test control algorithms in dynamic multi-axial test rigs.

2 Related Work

Based on a brief discussion of the modeling and identification processes for servo-hydraulic systems, the work relevant to the content of the paper in hand will be addressed in this section.

2.1 System Modeling

In general, to obtain a mathematical model of a certain dynamic system, three kinds of modeling methods are possible for deriving a proper mathematical system description [1,2]:

If we know all the relevant information describing our system, such as its physical structure, the operation mechanism, important material parameters and so on, we can use the white box modeling method based on a purely analytical system description.

The second method is the so-called black box modeling method, applied when we have little or even no knowledge about the system and we can only use experiments with known input signals to estimate a mathematical system description from the measured system response.

The third method is the grey box modeling method: we have good information about the system dynamics, such as its physical structure and operation mechanism, however some system information, such as the values of some physical and/or material parameters, is unknown.

In [3], we have built the mathematical model of a 250kN type hydraulic cylinder system with a hydrostatic bearing and thus with negligible friction within the cylinder sub system. Based on a grey-box modeling approach, we have derived an accurate system model which we reduced to a low order model without significant loss in the quality of the system dynamics description for most of our applications. The results obtained from both simulation studies and real world experiments show that this method of modeling and simplification is effective for test cylinders with negligible friction. This modeling process fails when significant friction phenomena are appearing in a cylinder system, such as from mechanical guiding elements and rubber seals.

2.2 System Identification

System identification is the method for estimating the structure and parameters of a dynamic system based on the information from the input and output signals of the related systems. After 50 years of development and research in the field of system identification of electro-hydraulic servo systems, several powerful identification methods have been developed and successfully implemented [4,5,6].

In [7], Jang has applied adaptive network algorithms to realize system identification. His algorithm has a self-learning function, nevertheless, the complexity of the obtained overall control system grows exponentially when increasing the number of model dimensions of the system. Zhang has proposed a Fuzzy-Tree algorithm [8,9] which has some advantages, such as fast calculation and high precision. Many research works on identification algorithms based on neural networks prefer BP feed forward neural networks, however, these algorithms have several disadvantages, such as slow convergence speed. In [10], an adaptive algorithm to adjust the learning rate has been applied to speed up convergence. In [11], a new parameter identification method based on neural network has been proposed by using the upper and lower bounds of the parameters of an electro-hydraulic load simulator. In recent years, several researchers have used neural networks in combination with genetic algorithms. They have realized an optimal design of the topological structure and the weighting parameters of the neural networks [12]. A new kind of genetic algorithm has been applied in a parameter identification scheme of the digital electro-hydraulic governing system for a steam turbine [13].

All these identification algorithms can be used mainly in single-input single-output (SISO) systems. With the increase of the number of control axes and thus increased control dimensions of the system, suitable identification algorithms are much more complex. Hence it is difficult to apply identification algorithms in multi-input multi-output (MIMO) systems. Consequently, it is desirable and in some cases necessary to develop simple and effective identification algorithms which are suitable for MIMO systems. Several essential principles for the MIMO system identification have been introduced in [14]. Furthermore, the principles and methods of the system identification are also important for technicians and they can be found in [15].

3 Modeling and Identification of Servo-Hydraulic Cylinder Systems

An electro-hydraulic servo system is a fully integrated system which includes a digital control unit, a hydraulic actuator and the assisting components, such as cabling, mechanical connections and so on. When considering the control unit, we can in general distinguish between valve controlled systems and

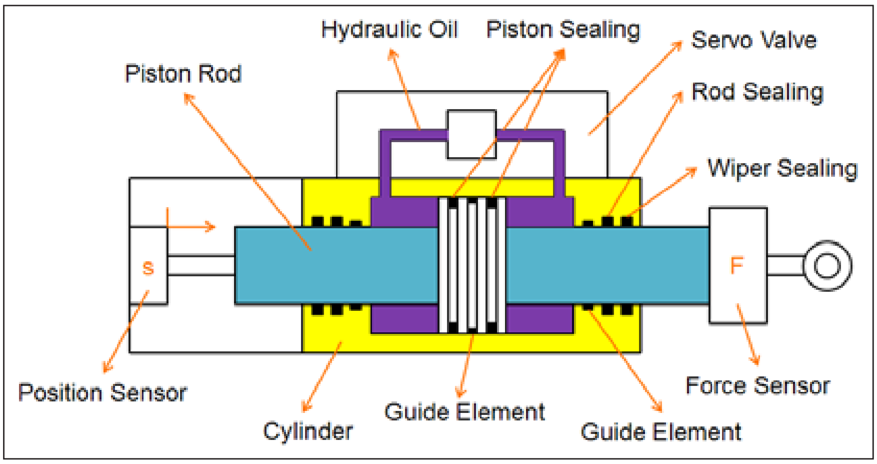

Figure 1. *Principal structure of a valve controlled symmetrically operating cylinder system with attached position and force sensor, sealing rings and mechanical guidance elements.*

pump controlled systems. Throughout this paper, we only consider a *valve controlled servo hydraulic cylinder system operated in position control*, composed of the cylinder structure, the servo valve and the sensor unit. These key components can be treated as individual sub-systems of the overall cylinder system, and consequently, we derive the corresponding mathematical submodels and then combine them into the overall cylinder system model during the proposed system modeling process. *Figure 1* shows the principal structure of a typical symmetrically operating servo-hydraulic cylinder system.

3.1 Modeling of Test Cylinder Systems

The first step to sufficiently describe the dynamic characteristics of an electro-hydraulic servo control system is to obtain an accurate mathematical model of the system dynamics. Without a proper mathematical model it is difficult or sometimes even impossible to achieve an acceptable control performance for the overall rig control for ensuring high test quality–especially for dynamic test applications.

As the first step in the derivation of the overall model of a position-controlled servo-hydraulic cylinder system, the analytical models of the test cylinder, the servo valve and the involved sensors are derived separately. These sub-system descriptions are then combined to obtain the full system model as the final step in our modeling process.

3.1.1 Modeling of the Cylinder

The cylinder model is built from three basic mathematical relationships: the pressure-flow equation of the control valve, the continuity equation of the fluid and the pressure-load equation.

Pressure-flow equation of the control valve

The flow through the valve orifices is usually described by the orifice equation with the relationship between the valve spool position x_v and the load pressure p_L [16]:

$$Q_L = c_v x_v \sqrt{p_s - p_L} \tag{1}$$

where c_v is the flow coefficient and p_s is the supply pressure. We assume that p_s is constant.

Q_L has a nonlinear relationship with x_v and p_L, and consequently, we have to linearize it by introducing the first order Taylor series expansion to Eq. 1 to obtain

$$Q_L = Q_{L0} + \Delta Q_L = Q_{L0} + \frac{\partial Q_L}{\partial x_v}\Big|_{x_{v0},p_{L0}} \Delta x_v + \frac{\partial Q_L}{\partial p_l}\Big|_{x_{v0},p_{L0}} \Delta p_l \tag{2}$$

where Q_{L0} is the flow of the working point (defined by x_{v0} and p_{L0}). Eq. 2 is the linearized equation, only valid in the vicinity of the working point. When simplifying the modeling process by focusing only on the so-called zero working point ($Q_{L0} = 0$, $x_{v0} = 0$, $p_{L0} = 0$), one can calculate the linearized pressure-flow equation of the control valve [17]:

$$Q_L = K_q x_v - K_c p_L \tag{3}$$

with K_q being the flow gain and K_c being the flow-pressure coefficient.

Continuity equation of the fluid

According to [16], the continuity equation of the fluid is given by

$$Q_L = A\frac{dy}{dt} + C_t p_L + \frac{V_t}{4K}\frac{dp_L}{dt} \tag{4}$$

where A is the piston working area of the cylinder, y is the displacement of the piston rod, C_t is the leakage coefficient of the cylinder, V_t is the whole volume of cylinder chambers, and K is the bulk modulus of the used oil. Note that we do not consider the oil volume within the pipelines and the servo-valve since with very short pipelines this oil volume can be neglected when compared to the oil volume inside the cylinder chambers.

Pressure-load equation

The pressure-load equation can be written as:

$$Ap_L = m\frac{d^2y}{dt^2} + B\frac{dy}{dt} + K_m y + F \tag{5}$$

where m is the load mass (including piston rod), B is the kinetic damping coefficient, K_m is the spring stiffness of the elastic load and F is the load force.

Applying the Laplace transform to Eq. 3, Eq. 4 and Eq. 5 leads to the following equations

$$Q_L = K_q X_v - K_c P_L \tag{6}$$

$$Q_L = AsY + (C_t + \frac{V_t}{4K}s)P_L \tag{7}$$

$$AP_L = ms^2Y + BsY + K_mY + F \tag{8}$$

We further assume that we can consider a cylinder rod without externally applied forces, i.e. $F = 0$. In this case, the open loop transfer function of the cylinder can be derived by combining Eq. 6, Eq. 7 and Eq. 8 into:

$$\frac{Y(s)}{Q_0(s)} = \frac{\frac{1}{A}}{\frac{V_t m}{4KA^2}s^3 + [\frac{m(K_c + C_t)}{A^2} + \frac{BV_t}{4KA^2}]s^2 + [1 + \frac{B(K_c + C_t)}{A^2} + \frac{K_m V_t}{4KA^2}]s + \frac{K_m(K_c + C_t)}{A^2}} \tag{9}$$

where $Y(s)$ is the Laplace transform of the displacement signal of the piston rod and $Q_0(s)$ is the Laplace transform of the no-load flow signal of the servo valve.

Throughout our experiments, we use a 40kN type cylinder with significant friction, where friction arises mainly from the type of the used mechanical rod guiding elements (bearing strips).

Consequently, we build a simple friction model for this kind of a test cylinder, using the relation

$$F_f = B_f v \tag{10}$$

where F_f is the friction force, B_f is the friction coefficient and v is the velocity of the piston rod.

In our model, the friction can be treated as part of the damping, hence Eq. 9 remains vaild as the open loop transfer function of the used test cylinder.

The open loop transfer function of the cylinder (Eq. 9) can be simplified to

$$\frac{Y(s)}{Q_0(s)} = \frac{\frac{1}{A}}{s(\frac{s^2}{\omega_h^2} + \frac{2\xi_h}{\omega_h}s + 1)} \tag{11}$$

where ω_h is the natural frequency and ξ_h is the damping ratio.

The most important known physical parameters of the used test cylinder are given as follows: the piston working area of the cylinder $A = 1300.6\,\mathrm{mm}^2$, the volume of the oil in both the inlet chamber and the outlet chamber of the cylinder $V_t = 167523\,\mathrm{mm}^3$ and the mass of the piston rod $m = 6.07\,\mathrm{kg}$.

Furthermore, friction is one of the most important factors that influence the dynamic characteristics of the 40kN type cylinder. Nevertheless, it is in general difficult – and in our case impossible – to directly calculate the occurring friction parameters, emphasizing the need for proper system identification as part of the system modeling process.

The conducted modeling process can be interpreted as a *classic grey box modeling for the cylinder structure* – an analytical system description with some unknown system parameters.

3.1.2 Modeling of the Servo Valve

The servo valve is itself a complex mechatronic system and many parameters can only be determined as a wide range or are completely unknown. Nevertheless, manufacturers' datasheet information usually provides the step response and the frequency response for such valves [18]. Consequently, we use this information in a simple 1st order model approximation for the used valve units (valve types G761-3003 and G761-3004 of MOOG). Due to the high quality and the high operational bandwidth of the used valves, the chosen 1st order model remains sufficiently accurate for our test application. The model for the given servo valve unit is then determined by

$$\frac{Q_0(s)}{I(s)} = \frac{K_{sv}}{\frac{s}{\omega_{sv}} + 1} \tag{12}$$

where K_{sv} is the flow gain and ω_{sv} is the cut-off frequency. These valve-specific parameters have been obtained from the datasheet of the used servo-valve [17,18].

3.1.3 Modeling of the Sensor Part

For the measurement of the cylinder rod position, a high quality MTS Temposonics R-Series position sensor has been attached inside the cylinder. This sensor possesses a negligible linear position measurement uncertainty of less than 0.01 %. Furthermore, the dynamic non-idealities for this sensor type are significantly above the bandwidth of the used combination of servo valve and test cylinder. Hence we can validly assume that the sensor-amplifier system can be considered ideal and its transfer function reduces to $F(s) = 1$.

3.1.4 Overall Model of the Test Cylinder system

The overall mathematical model of the valve controlled servo cylinder system is obtained by combining the previously derived sub-models of the cylinder, the servo valve and the sensor, and can be written as:

$$\frac{Y(s)}{I(s)} = \frac{\frac{K_{sv}}{A}}{s(\frac{s^2}{\omega_h^2} + \frac{2\xi_h}{\omega_h}s + 1)(\frac{s}{\omega_{sv} + 1})} \tag{13}$$

but the included system parameters ω_h, ξ_h, ω_{sv} and K_{sv} are unknown and thus have to be estimated by the following system identification process.

3.2. System Identification

In our real world experiment, the cylinder system described above is connected to a digital control system; in our application, a state-of the-art PID-controller is implemen-

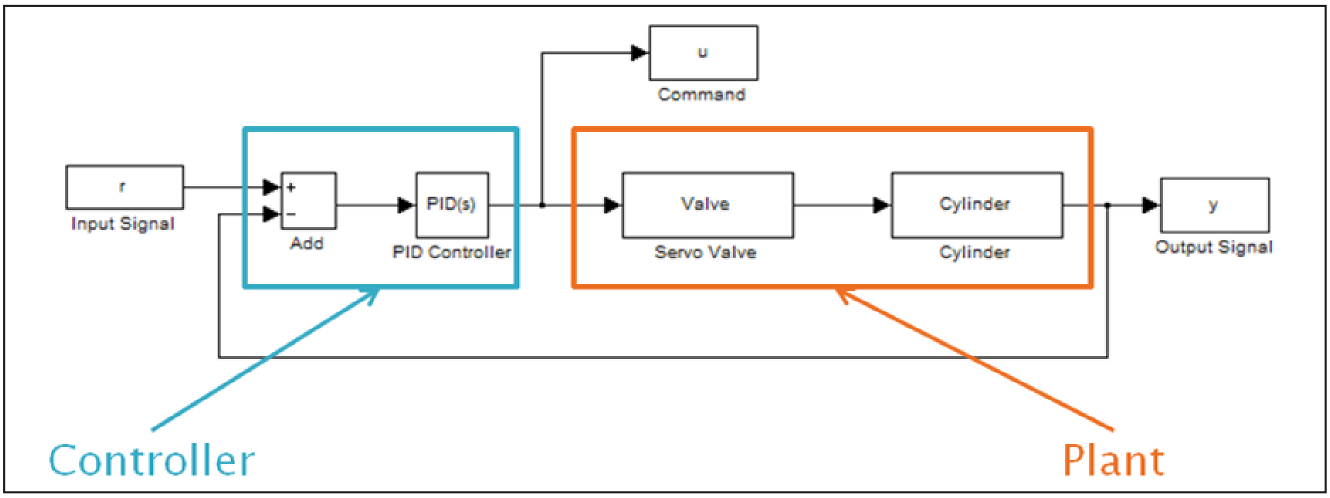

Figure 2. *Schematic system diagram: The cylinder system (plant) consists of the servo valve and the test cylinder structure (including its hydraulic-mechanic components).*

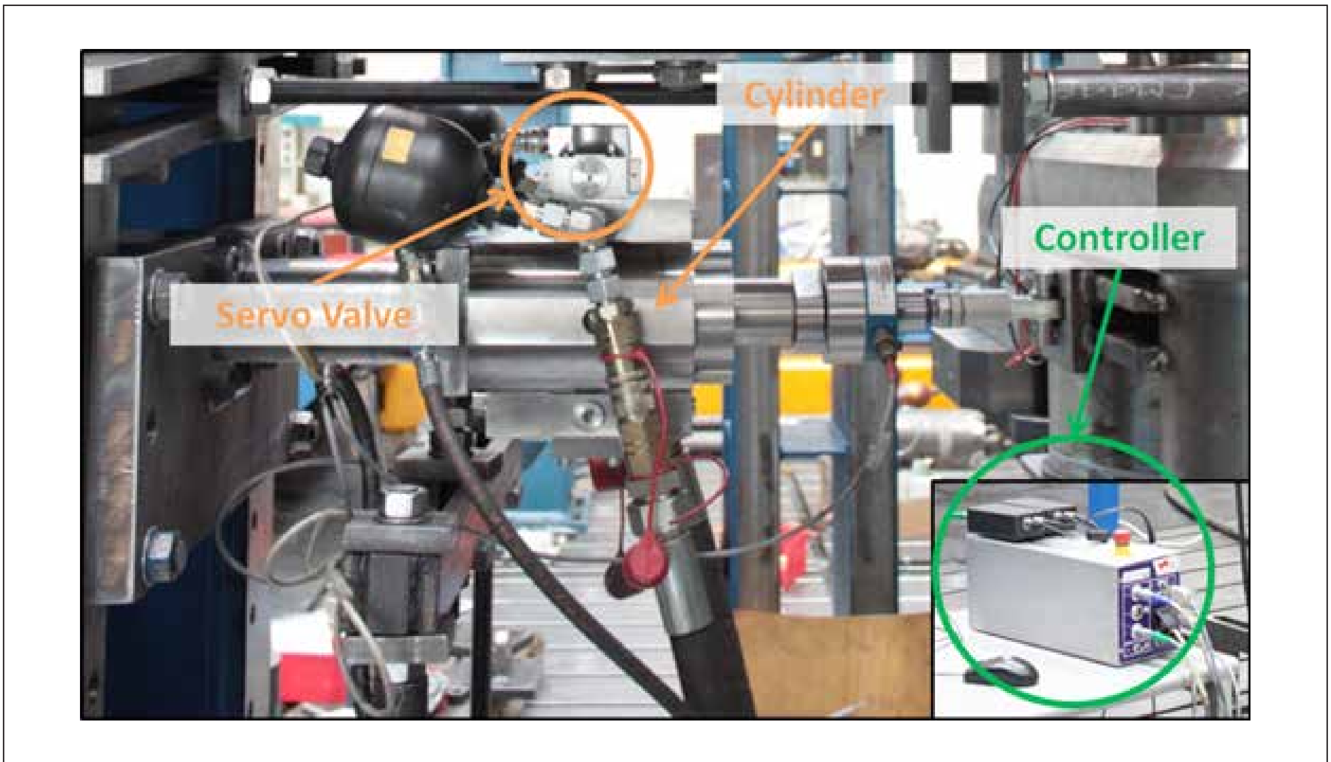

Figure 3. *Test cylinder system, including the cylinder unit with the included position sensor, the directly attached servo valve and the digital cylinder controller unit.*

ted as part of the real time cylinder control system (SPH SmartControl). The overall test system is thus composed of the PID controller, the servo valve and the 40kN test cylinder (with significant friction), as shown in *Figure 2* for our single-channel control system (the sensor unit is considered ideal, as discussed previously). *Figure 3* provides a photograph of the used real world test system – the test cylinder (as part of a multi-axis test rig) with the attached servo valve and the connected digital cylinder controller.

3.2.1 Input Signal Design for Identification

In the system identification procedures, two types of suitable colored noise signals are used as reference setpoint signals: a) zero mean white Gaussian noise, 1st order low-pass filtered with a cutoff frequency of f_c=10Hz, and b) zero mean white Gaussian noise, 6th order bandpass filtered with low and high cutoff frequencies of f_l=150Hz and f_h=450Hz.

We assume that the cylinder system can be considered linear, thus we superimpose the low frequency signal on the high frequency signal for identification and verification.

3.2.2 Black-Grey Box Model Identification

The grey box model identification method is not directly applicable to our cylinder system composed of the 40kN type cylinder and the G761-3003 servo valve because the value of friction is still unknown. We cannot estimate the initial values of the unknown system parameters from Eq. 13 accurately, but precise values are needed for the grey box model identification (as demanded by identification routines, such as the System Identification Toolbox in Matlab [19]).

Consequently, we apply a *combined black-grey box model identification method*. First, we build a white box model of the whole system, i.e. we roughly estimate the values of the unknown parameters in Eq. 13 according to our experience. Then, we apply the black box model identification procedure which improves the white box model by adjusting some model parameter estimates. Using the obtained improved model as a new initial model, we apply the grey box model identification so as to derive an overall system model with satisfying accuracy. These steps are explained in more detail below.

Black Box Model Identification

Based on the measured system input signal (position setpoint signal) and the measured output signal (cylinder rod position), we can calculate the power spectrum density (PSD) of the plant. This procedure is a kind of a nonparametric black box model identification method since we use a non-parametric PSD estimation (periodogram averaging, Welch method). The result of this black box model identification allows us to improve the white box model with better estimates of the values of the unknown parameters, such as ω_h and ξ_h .This step is essential for the following grey box model identification in order to obtain accurate identification results.

Grey Box Model Identification

For the grey box model identification, we utilize the well-known prediction error method (PEM) of the System Identification Toolbox of Matlab [19].

To obtain an accurate mathematical description of our test cylinder system, we combine the black box model identification method and the grey box model identification method into the so-called *black-grey box model identification*.

Black-Grey Box Model Identification

The principle of the proposed black-grey box model identification method is shown as a process chart in Figure 4 and is described as follows: firstly, based on the hydraulic theory, we build the grey box model of the whole system. According to our practical expertise, we include pre-estimates for the values of the unknown parameters as rough estimates (some of the values may have big errors) so as to achieve an inaccurate white box model.
Moreover, after designing the setpoint signal r, the signal is applied to the controlled servo-hydraulic cylinder system, delivering the output cylinder rod position signal y as the system response to our input signal. Additionally, we also acquire the applied control variable u, i.e. the output signal of the PID controller. Then, the black box model identification method (PSD estimation from measured system signals) is used to obtain the frequency response of the system in a non-parametric representation: we calculate the cross power spectral densities P_{ry} and P_{ru} with Welch's method in Matlab, which enables us to derive the frequency response of the plant (the cylinder system) by evaluating the ratio of the obtained cross spectral densities $P_{ry} = P_{ry} / P_{ru}$.

When comparing the frequency response of the inaccurate white box model estimate of the whole system with the experimentally obtained frequency response estimation from the above mentioned process, we can tune the values of the uncertain parameters ω_h and ξ_h from the white box model until the two frequency response curves roughly fit together on an amplitude basis. This enables us to obtain a proper initial theoretical model, including valid estimates for a) the initial values of some unknown parameters for the subsequent grey box model identification of the system and b) the initial values of the still unkown system parameters in the open loop transfer function (Eq. 13). Then, the standard grey box model identification method called prediction error method (PEM) is used to derive the final mathematical model of the overall electro-hydraulic servo system.

4 Results

When utilizing the proposed black-grey box modeling for identifying the dynamic model of our cylinder system with friction, we obtain very accurate results, at least for the relevant operational bandwidth of such cylinder types ranging from 0 to 100 Hz.

In order to verify the identification results, a suitable Simulink system

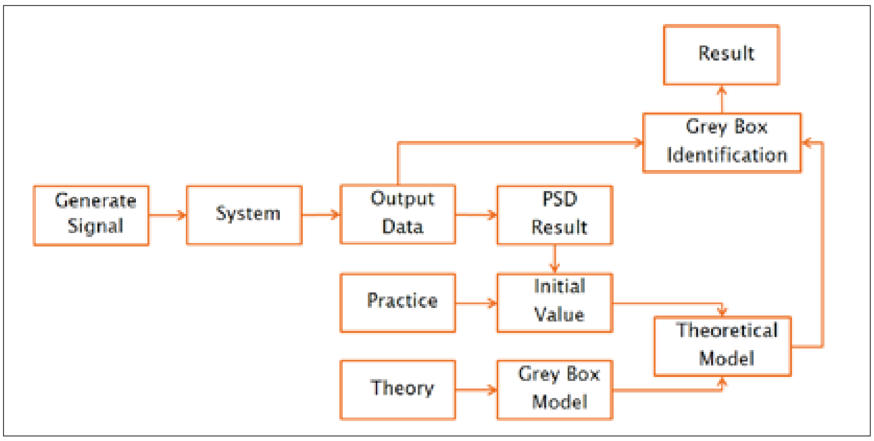

Figure 4. *Schematic diagram of a black-grey box model identification.*

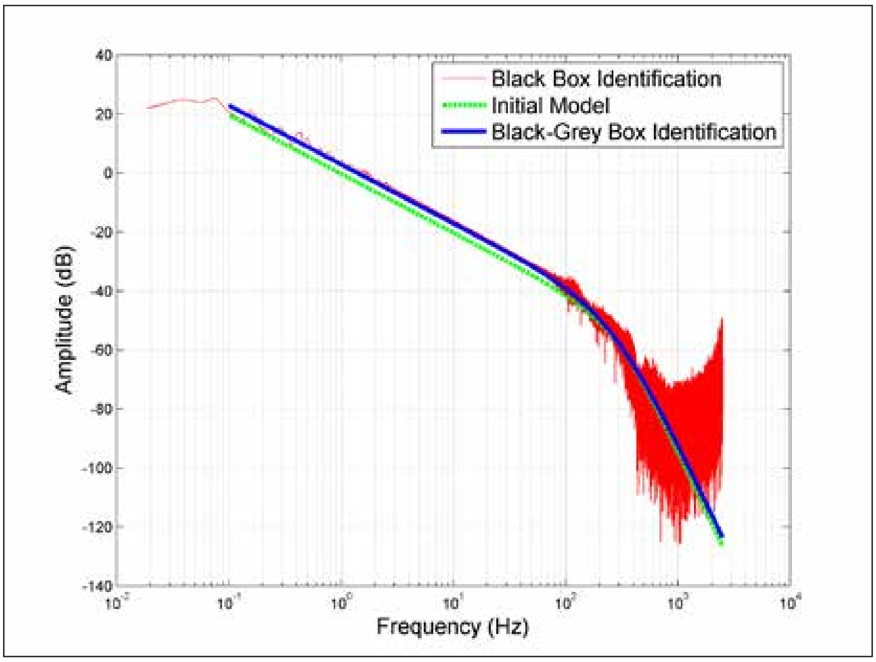

Figure 5. *Comparison of the frequency response from the simulated cylinder system (green line: analytical grey box model, blue line: black-grey box model approach) with the real world measurement data (red line: frequency response from PSD estimation).*

simulation with the identical input signal as the setpoint sequence in the real world system has been performed. *Figure 5* provides a comparison of the identification results with the real world data from our experiments: the red line shows the plot of the real system frequency response obtained from measurement data. The frequency response function (FRF) is obtained, as detailed above, by a non-parametric PSD estimate which can be regarded as a standard black box identification process. The green line shows the FRF obtained from the simulation, using only the grey box model where the unknown parameters have been improved based on prior PSD result. Using this model as the new initial model for the consequent grey box identification, we obtain an accurate model for our cylinder system. The corresponding final FRF estimate is shown with the blue line. By comparing the three results, we can draw the following conclusion: the black box identification result (PSD) can improve our theoretical model which is now feasible to be considered as the initial theoretical model for finally obtaining an accurate system model with the grey box identification method.

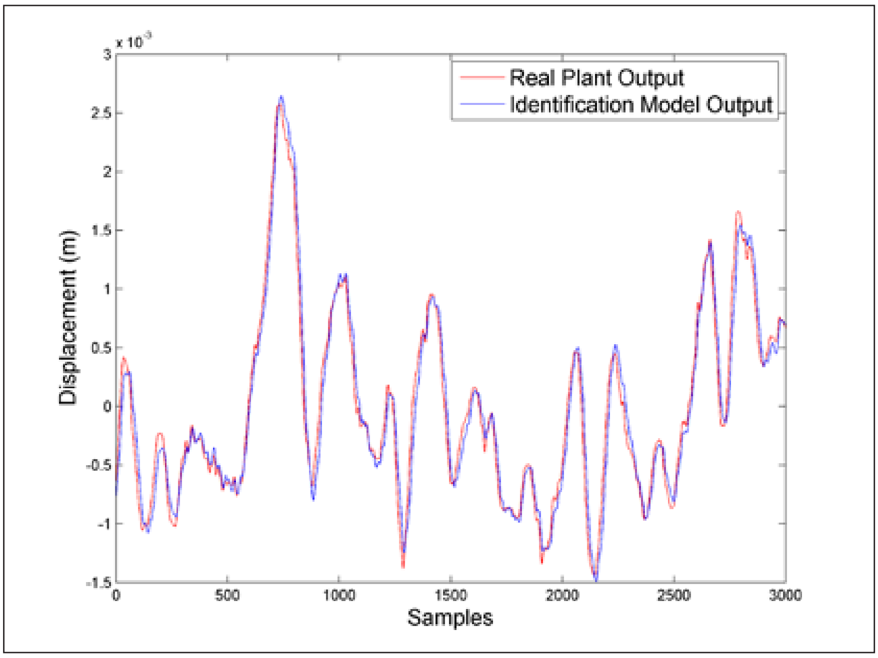

Figure 6. *Comparison of the system output signal (the cylinder rod position) for the real world test cylinder obtained with measurement (red signal), and simulation data obtained using the previously identified cylinder model (blue signal).*

Figure 6 shows the output signal measured in the real world cylinder system compared to the output signal from the simulation model. As can be seen, the waveform of the simulated system output signal is very close to the measured output signal waveform (the measured cylinder rod position), showing a nearly perfect agreement between simulation and reality also for the comparison on a time series basis.

The obtained results prove that a) our simple friction model is reasonable and b) the developed black-grey box model identification method is reliable and delivers accurate mathematical descriptions of our test cylinder system dynamics.

To verify the universality of this black-grey box model identification method, we use a changed servo-hydraulic cylinder system with the same 40kN type cylinder but a different servo valve G761-3004 of MOOG with double specified flow rate. Taking into account these systematic changes in our cylinder system, we would expect the obtained mathematical model to differ from the previously obtained model only by the servo-valve system gain K_s, while delivering identical results for the cylinder-based parameters. We conducted the same experiments for the changed cylinder system and then again performed our proposed system identification method. Comparing the identification

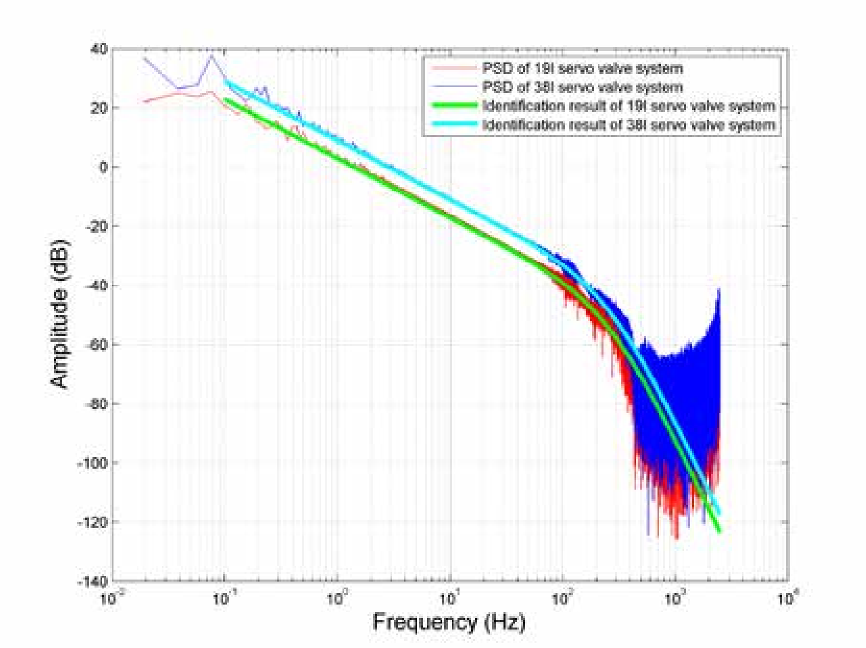

Figure 7. *Comparison of the identification results (frequency response) of the two different electro-hydraulic servo systems.*

Table 1. *Comparison of the determined values of certain parameters of the cylinder system which are obtained by the developed system identification process*

Systems	$\omega_h\,(\mathrm{Hz})$	ξ_h	$\omega_{sv}\,(\mathrm{Hz})$	$K_{sv}\,(\mathrm{m^3/(s \cdot A)})$
Cylinder system with 19 l/min servo valve (type Moog G761-3003)	343	0.9766	160	0.0114
Cylinder system with 38 l/min servo valve (type Moog G761-3004)	343	1.0012	160	0.0230

results for both cylinder systems, we observe that only the servo-valve gain parameter has been increased by a factor of 2, as we have expected.

5 Conclusion

In this paper, we derive the mathematical model of an electro-hydraulic servo system utilizing a novel black-grey box model identification method for the servo-hydraulic test cylinder systems with significant friction. The identification method performs well for cylinder systems with some unknown physical parameters that cannot be estimated accurately prior to the identification process. Simulation results and their comparison with measurement data from real world experiments show that the developed identification strategy is satisfying in terms of delivering an accurate model of the cylinder system dynamics. Further experiments and identification results with a different system validate the universality of our proposed method.

Future research will focus on the development of identification and control strategies for multi-axial servo-hydraulic test applications.

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Natančno modeliranje in identifikacija sistemov s servohidravličnimi valji pri večosnih testnih aplikacijah

Razširjeni povzetek

Uporaba servohidravličnih sistemov je ključnega pomena pri napravah, namenjenih preizkušanju trdnostnih lastnosti in utrujenosti materiala tako strojnih delov kot konstrukcij. Da lahko zagotovimo visoko kakovost testiranja in dobro ponovljivost rezultatov, je potrebno uporabljati robustne strategije vodenja takšnega sistema. To je še posebej nujno v primeru večosnih testnih sistemov, kjer uporabljamo večje število sinhrono delujočih servohidravličnih valjev. Za učinkovito uporabo primernih regulacijskih algoritmov so potrebne natančne informacije o dinamičnih lastnostih aktuatorjev in celotne testne naprave.

V prispevku je obravnavana posebna metoda identifikacijskega sistema za pridobitev natančnega matematičnega modela, primernega za skupino najpogosteje uporabljenih, cenovno ugodnih simetričnih servohidravličnih valjev z večjim vplivom trenja. Da lahko natančno opišemo dinamiko takšnega servohidravličnega valja, je bil v prvem koraku zasnovan matematični model hidravličnega valja, voden s servoventilom. Model je zasnovan na klasičnem pristopu, na podlagi tokovne enačbe, ki opisuje tok tekočine skozi odprtine ventila spremenljivega prereza na podlagi enačb, ki podajajo spremembe tlaka v komorah valja, ter na podlagi zakona o ravnotežju sil na batu valja. Dinamika servoventila je podana kot PT1-člen, medtem ko dinamika merilnega člena ni upoštevana. Dobljena skupna prenosna funkcija sistema, ki povezuje položaj valja kot izhodno veličino sistema in krmilni signal na ventil kot vhodno veličino, je četrte stopnje z integralnim delovanjem. V tako dobljenem modelu so lastna frekvenca in stopnja dušenja valja ter lastna frekvenca in faktor ojačenja servoventila neznani parametri. Ti parametri se določijo kasneje na podlagi posebne identifikacijske sheme, ki temelji na metodi črne škatle (»black-box«) za oceno začetne vrednosti neznanih parametrov, kasneje pa se za natančnejšo oceno celotnega modela sistema uporabi metoda sive škatle (»grey-box«). Na ta način lahko predlagana metoda črne škatle zagotavlja natančnejše informacije o dinamiki sistema.

To dokazujeta primerjavi frekvenčne karakteristike zasnovanega modela z eksperimentalno posneto v širokem frekvenčnem območju vhodnega signala kot tudi primerjava dinamičnih odzivov, dobljenih na podlagi simulacije in eksperimentalnih rezultatov, pri čemer so bili v obeh primerih za vzbujanje uporabljeni naključni signali. Univerzalnost predlagane metode je testirana na podlagi opisanega aktuatorja, vodenega s servoventilom z enkrat večjim nazivnim pretokom. Primerjava rezultatov simulacije in eksperimenta kaže visoko natančnost predstavljenega modela uporabljenega servohidravličnega sistema, kar potrjuje učinkovitost razvite metode identifikacije.

Prihodnje raziskave bodo usmerjene v razvoj metode za identifikacijo in regulacijske strategije, primerne za večosne servohidravlične sisteme.

Ključne besede: elektrohidravlični servosistem, servohidravlični valj, test utrujanja materiala, trenje, modeliranje, identifikacija, simulacija