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Minimal-model for robust control design of large-scale hydraulic machines

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***Abstract*—Hydraulic machines are in use where the large forces, at relatively low velocities, are required by varying loads and often hazardous and hard-to-reach environments, like e.g. offshore, mining, forestry, cargo logistics, and others industries. Cranes and excavators equipped with multiple hydraulic cylinders are typical examples for that. For design of the robust feedback controls of hydraulic cylinders, already installed into large-scale machines, there is a general lack of reliable dynamic models. Also the suitable and feasible identification techniques, especially in frequency domain, yield limited. This paper proposes a minimal-modeling approach for determining the most relevant open-loop characteristics of hydraulic cylinders installed on a bulk loader crane. The resulted model allows for robust control design without knowledge of the overall complex system behavior. The total system gain, non-negligible input dead-zone, and aggregated phase lag are identified from the simple open-loop experiments. An aggregated phase lag is captured for the assumed bandwidth of the system, and that without knowledge of the higher-order residual system dynamics. Based thereupon, the robust feedback position regulator is designed and extended by the velocity feed-forwarding. The proposed modeling approach, together with the designed control system, are evaluated on the third axis of a hydraulic loader crane.**

I. Introduction

Large-scale hydraulic machines are in use wherever heavy payloads and, correspondingly, considerable forces are required in combination with often hazardous, harsh, correspondingly outdoor, environments. Examples for such machines, demanding appropriate control of the kinematic members, are the cranes [1], [2] or manipulators [3], [4], excavators [5], [6], forestry machines [7], [8], and others. A recent survey on the control of hydraulic manipulators can be found in [9]. For basics of the hydraulic control systems we also refer to [10]. Regardless of a targeted hydraulic machine, it is mostly the linear hydraulic cylinders that constitute the basic actuators, incorporated in a variety of kinematic structures and converting mechanisms, like for example rack-and-pinion slew joints.

Several approaches and design methodologies have been pursued for developing robust feedback controllers, with specification on either motion or force regulation or a combination of both. Only some of them to be mentioned are briefly referred below. For more detailed survey see for instance in [9]. Experimental evaluation of various control strategies for hydraulic servo manipulators can be found in [11], [12]. A coordinated motion control of two hydraulic actuators handling a common object has been reported in [13]. This configuration can be also seen as a possible benchmark for hydraulic actuators counteracting with environment, while some impedance constraints should be met. Adaptive motion control and force control of hydraulic cylinders have been presented in [14] and [3] correspondingly. LPV control of an electro-hydraulic servo system has been reported in [15], while attempting to deal with inherent system nonlinearities through a parameter-varying linear modeling. In [16], a Lyapunov-based control design has been proposed for a stable haptic manipulation with hydraulic actuators. A second-order sliding mode control of a mobile hydraulic crane has been reported in [17]. An optimal tuning of PID control, combined with a dead-zone compensator and denoted as ‘nonlinear PID’, has been proposed in [18] for hydraulic systems. Also the design and experimental evaluation of a controller of electro-hydraulic actuator based on the quantitative feedback theory [19] has been proposed in [20], with an objective to increase the robustness of the force controller.

The above mentioned control design strategies require mostly a certain system modeling, with the necessary assumptions about the system dynamics structure and related identification of the free system parameters. A herewith associated issue is that for a ready-assembled large-scale machine, there is a general lack of reliable dynamic models. Also the suitable and, above all, feasible identification techniques, especially in frequency domain, reveal mostly limited. The practice-related challenges are evoked also due to a necessary system instrumentation. An installation of auxiliary and accurate sensors, also of the internal system states, is often costly and can be also inherently restricted by the environmental conditions.

The present paper aims to propose a minimum required model which should be, however, sufficient for designing a robust motion controller for the large-scale hydraulic manipulator (crane). Each single-channel hydraulic cylinder actuator is supposed to be considered solely from the input-output point of view, while making the most obvious assumptions of a free integrator behavior and aggregated phase lag due to unknown residual dynamics. The proposed minimal-model provides a direct and straightforward procedure for designing and tuning the simple combined feed-forward and feedback control.

The rest of the paper is organized as follows. In Section II, a minimal-model is proposed and discussed along with the open-loop analysis. The hydraulic loader crane, which is the target experimental system, is briefly described in Section III. The required minimal-model parameters estimation are shown in Section IV. The control design and evaluation are reported in Section V. Finally, conclusions are drawn in Section VI.

II. Minimal-model and open-loop analysis

Accurate modeling of the single joints within a complex multi-body dynamics of the large-scale machines actuated by hydraulic cylinders constitutes a challenging task, which may reveal barely feasible in practical applications. Even if a detailed model structure is known and all components' data sheets are available, an accurate control-oriented identification of the relevant system parameters, which essentially influence the system behavior, can be hardly accomplished for a ready-assembled equipment. Examples of a detailed modeling of stand-alone hydraulic cylinders can be found in [14], [21], [13], [16], [22], while of those incorporated into the overall machine dynamics in [1], [12], [23], [4].

The proposed minimal-model approach bases on the assumption of a low-frequency dynamics, i.e. of bulky moving masses in the hydraulic machine, comparing to the higher frequency dynamics of the directional control valves and hydraulic circuits of single cylinders. This assumption is justified as the mechanical time constants associated with the permanent weights and payloads are larger, by at least one order of magnitude or even more, than the time constants associated with the low-level-regulated directional control valves and flow-pressure dynamics, cf. e.g. [15], [22].

Since the relative joint position, which is in a direct kinematic relationship to the linear stroke of the driving hydraulic cylinder, is the measurable output of primary interest and use for feedback control, the plant model assumes one free integrator additionally to the total plant gain K. Furthermore, the hydraulic cylinder plant is assumed as sufficiently damped so that no additional derivative feedback control action is generally required. Recall that a hydro-mechanical cylinder system is inherently damped through the cylinder o-rings and seals and through the total hydraulic volume, see e.g. [22]. Therefore, the proportional control gain can be seen as a single feedback design parameter, for which computation the knowledge of the total plant gain is particularly required. Recall that both multiplied gains determine the crossover frequency of open-loop, which is characteristical for stability analysis based on the phase and gain margins, see e.g. in [24].

For a harmonic system excitation, the impact of unaccounted residual system dynamics is similar to that of the dead-time transfer element $\exp(-T_d s)$, while $T_d = \mathrm{const}$ can be assumed for a particular angular frequency ω_b. Since the phase lag of open-loop at the characteristic angular frequency, i.e. crossover frequency, is crucial for stability analysis, an adequately accurate estimate of T_d can provide sufficient information about the control open-loop phase response in proximity to the critical frequency range. Note that this is without explicit knowledge of the overall system transfer function. Such configuration of the amplitude and phase responses is schematically shown in Fig. 1, that for the gained integrator with time-delay element connected in series.

Apart from the unknown residual system dynamics, which phase lag is aggregated into a time-delay element as described above, an additional impact on the output behavior is caused by the dead-zone nonlinearity associated with the directional control valve at each hydraulic cylinder. For the given plant input u, this can be expressed as

$$u^* = \Phi(u) = \begin{cases} u - 0.5W\mathrm{sign}(u), & \text{if } |u| > 0.5W, \\ 0, & \text{else.} \end{cases} \quad (1)$$

The total dead-zone width is W, within which no control action u^* occurs, independently of the recent input value u.

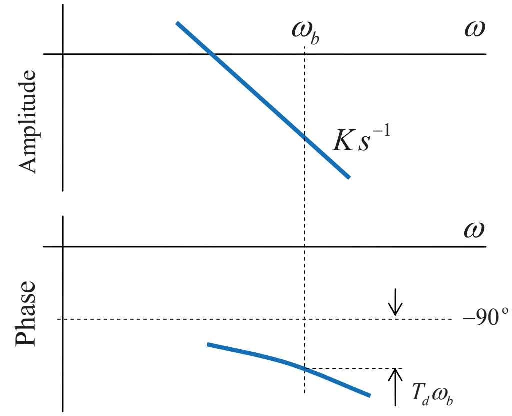

Fig. 1. Fragment of amplitude and phase response of the gained integrator with dead-time in vicinity to the angular frequency of interest.

Based on the above considerations, the minimal-model, yet suitable for robust identification and sufficient for a subsequent control design, is given by

$$\hat{x} = \frac{K}{s} \exp(-T_d s) \Phi(u), \quad (2)$$

where $\hat{x}$ is the (model) predicted cylinder stroke. Three system parameters K, T_s, and W need to be identified from the plant response. This will be shown by means of the simple time domain experiments described further in Section IV. Note that in case of a largely-varying external load, the total gain K is subject to variations os that the identification and control tuning should be repeated. It is also worth noting that due to a rather slow dynamics of operating the crane, no cross-couplings between the joints are considered and each one is, hence, assumed as a single-input-single-output (SISO) system.

III. Hydraulic loader crane

The standard hydraulic loader crane HMF 2020-K4, see image of the laboratory setup in Fig. 2, has been used for experimental evaluation. The crane is of the knuckle-boom-type, possessing three rotary and one telescopic joints. Due to the workspace limitations and, at the same time, similarities in the dynamic behavior of both rotary joints in the vertical plane, only the 3rd one has been considered for evaluation. Note that the 2nd and 3rd joints disclose certain structural similarity in mechanical arrangement and actuation principles. Both are driven by the linear hydraulic cylinders (see in figure) and are subject to the impact of gravity. On the contrary, the 1st rotary joint, which turns the column of the crane, is equipped by a rack-and-pinion type actuator which differs from the directly cylinder-driven 2nd and 3rd one. For more details on the hydraulic crane system the reader is refereed to [1], [2].

At this point, it is important to emphasize that the UDP (user datagram protocol) based external data interface to the integrated control cabinet of the crane allows for up to 1 kHz sampling rate for the input/output channels. However, the accessible encoder values, which are reflecting the linear stroke of hydraulic cylinders and therefore used for feedback control, are updated in the digital processing of control cabinet at a lower sampling rate. Several experimental evaluations reveal

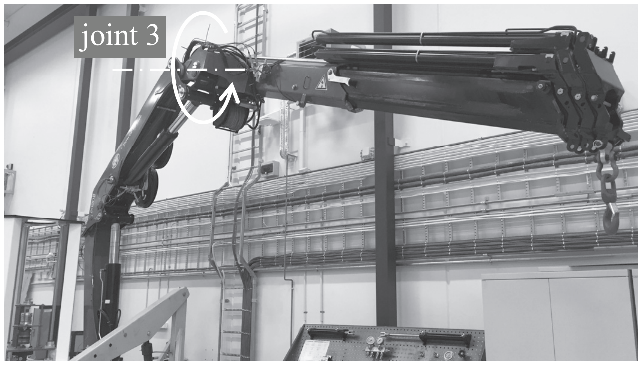

Fig. 2. Hydraulic loader crane with considered 3rd rotary joint.

the minimal update period of 6 msec and maximal one of 12 msec. Thus, only an uncertain average sampling rate of about 100 Hz can be assumed. Furthermore, one should emphasize that measurement of the linear cylinder stroke is done by encoders connected to the moving links via the prestressed steel wires of a relatively high length. This results in the additional spurious disturbances and vibrations, occurring as an unavoidable measurement noise.

IV. Parameter estimation

For reliable estimation of the model parameters in (2), the system is excited at the characteristic frequency ω_b, here assumed to be $0.2 \cdot 2\pi$ rad/sec. For that purpose, the control valve input $u \in [-1, 1]$ is subject to the sinusoidal reference. Recall that the assumed maximal operation frequency ω_b can be seen as the control bandwidth when subsequently designing the feedback loop. Further we note that due to the heavy moving masses and large hydraulic cylinders, the assumed maximal frequency is already sufficiently high. Using the measured (t, u, x) data, the simple curve fitting technique is applied based on the iterative search least-square algorithm (Levenberg-Marquardt). The measured stroke of the 3rd joint cylinder and the fitted model response are shown over each other in Fig. 3. The convergence of numerical parameters as

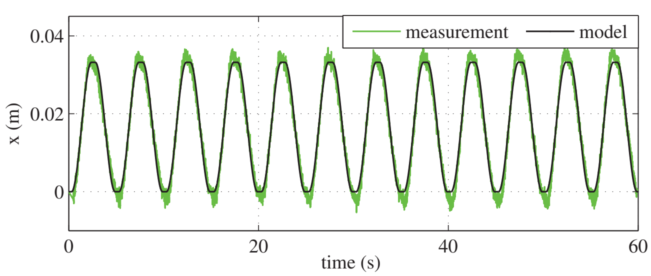

Fig. 3. Measured and model fitted stroke response of the 3rd joint cylinder.

function of the algorithm iterations is shown in Fig. 4. The residual identification errors spread is further shown by the histogram in Fig. 5. It can be seen that the residual identification errors are Gauss-shaped and well-distributed around zero mean-value.

In order to justify reliability of the modeling and, correspondingly, applied identification approach, another experiment with input excitation at $0.1 \cdot 2\pi$ rad/sec (half frequency

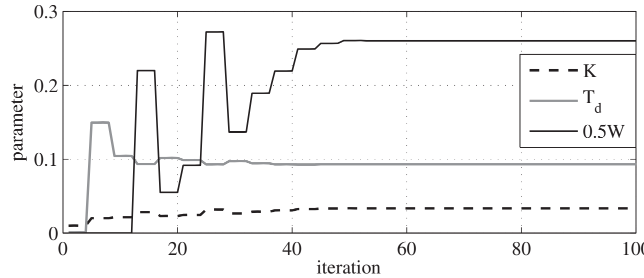

Fig. 4. Parameters convergence during identification.

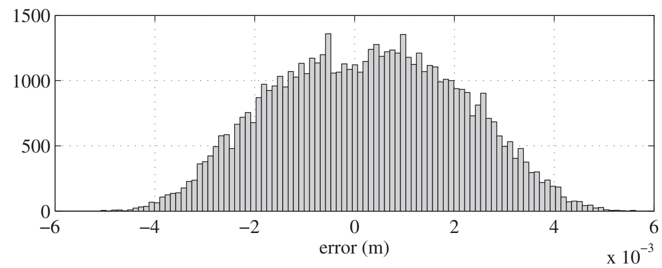

Fig. 5. Residual identification errors spread.

of the previous one) has been performed. The new data set has been equally used for determining parameters by means of the same identification routine. The estimated parameters for both cases are compared to each other in Table I. Both sets

TABLE I. Identified model parameters.

Parameter	$0.2 \cdot 2\pi$ rad/sec	$0.1 \cdot 2\pi$ rad/sec
K	0.033	0.032
T_d	0.093	0.112
W	0.52	0.50

of the identified model parameters lie in a close vicinity to each other. All parameter values are well comparable for both different frequencies, that argues for generality of the derived minimal model and pursued identification strategy.

V. Control design and evaluation

The control design bases on the identified parameters of the minimal-model, cf. Sections II and IV, which proves to be sufficient for deriving the two-degrees-of-freedom control structure, with the feedforward and feedback control parts as shown in Fig. 6. Note that once the operational conditions change, or an application-related control tuning should be repeated, the above identification procedure – in open-loop manner with feed-forward actuation only – can be executed again, and that without requiring any detailed system analysis or specific (additional) measurement equipment.

For the considered maximal angular frequency $\omega_b = 0.2{\cdot}2\pi$ rad/sec the assignment of the feedback control gain by $K^{-1} \approx 30$ results in $\omega_b = \omega_c$, where ω_c is the crossover frequency of the open-loop. Therefore the phase margin can be directly evaluated, as phase response of the gained integrator and time-delay element of the minimal-model, by

$$\phi(\omega_c) = \pi - \pi/2 - \omega_c \cdot T_d = 1.47 \ \text{rad} \ \approx 84.5 \,\text{deg}. \quad (3)$$

The open-loop phase margin (3) is unnecessarily high so that the feedback control gain is further enhanced to $P = 50$ which slightly shift the cross-over frequency to the right and slightly reduces the phase margin to 82.3 deg. Since the feedforward control gain is the inverse of the identified plant gain its value is directly used so that $F = 30$.

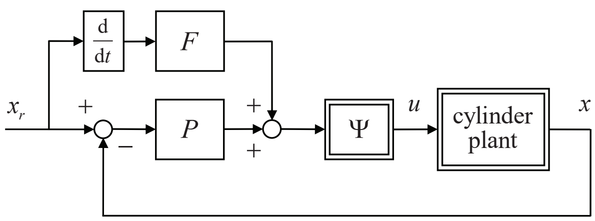

Fig. 6. Block diagram of combined feedforward and feedback controller with an additional inverse dead-zone compensator.

The pre-filter-type inverse dead-zone compensator is designed with respect to the estimated dead-zone width W. Further it is worth noting that this-way estimated size of the dead-zone coincides well with other off-line identification approaches for dead-zone identification, cf. e.g. with [2]. For the sake of clarity, the impact of the input dead-zone is demonstrated below by means of the time response of cylinder stoke to the applied low-rate input ramp. Both measured signals, control input and stroke output, are shown opposite to each other in Fig. 7. Note that for the sake of better visualization, the depicted input ramp is scaled down by the factor 0.1. It is evident that below certain input amplitude, that reveals a dead-zone behavior, no changes of the output stroke first occur. Only with a continuously increasing input magnitude a relative motion of the hydraulic cylinder sets on. One should note that this, at first look, straightforward experiment is less suitable for a direct dead-zone estimation, since the onset of relative motion at input ramp can be directly coupled with considerable and uncertain static friction and therefore initial system sticking [25]. A proper decomposition of the input dead-zone and cylinder friction appears as unfeasible at these driving conditions. On the contrary, a harmonic excitation with multiple periods allows for significantly reducing the system stiction, so that more accurate dead-zone estimation is provided by the identification procedure as described in Section IV.

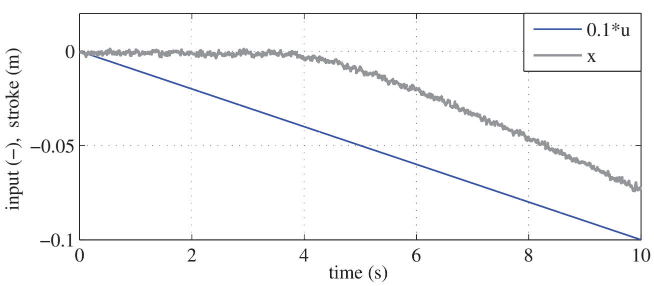

Fig. 7. Measured output stroke response to the ramp input.

After determining the dead-zone width, the inverse dead-zone map Ψ is applied to the input channel as in Fig. 6. In order to avoid the step-wise discontinuity in the inverse of dead-zone, cf. eq. (1), the sigmoid-type function is assumed for Ψ, instead of the sign-function. The resulted dead-zone compensator, with the input u^*, yields

$$\Psi(u^*) = u^* + 0.5\, W \, \frac{1 - \exp(-\alpha u^*)}{1 + \exp(-\alpha u^*)}, \quad (4)$$

where α is the scaling factor around zero-crossing. The assumed $\alpha = 400$ depends once on the u^*-domain and once on the required sharpness of the sigmoid transitions. Note that larger α-values yield the sigmoid function closer to the sign one. The latter can be seen as a boundary case, i.e. $\alpha \to \infty$, with a discontinuity at zero crossing.

In order to compare the control performance with and without use of the feedforward part, the sinusoidal reference trajectory (further denoted as Trajectory A) with 0.02 Hz frequency and 0.12 m amplitude is evaluated for both control structures. The control errors are shown in Fig. 8 opposite to each other. It is obvious that the combined (feedforward + feedback) controller performs as superior while providing about 50 % peak error reduction comparing to the single feedback control. The reference and measured response of cylinder stroke of the 3rd joint under evaluation are shown in Fig. 9 for the combined (feedforward + feedback) controller. The same combined controller is also evaluated with another sinusoidal reference trajectory (further denoted as Trajectory B) which has the double-increased frequency 0.04 Hz. Recall that the maximal bandwidth, correspondingly cross-over frequency for the designed feedback control, lies at 0.2 Hz. The amplitude on Trajectory B is decreased to 0.04 m in order to comply with the actuator saturations. The corresponding reference and measured response are shown in Fig. 10 over each other.

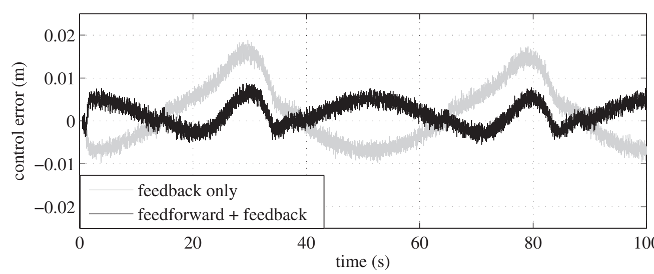

Fig. 8. Evaluated control error (trajectory A) for the single feedback and combined feedforward + feedback controllers.

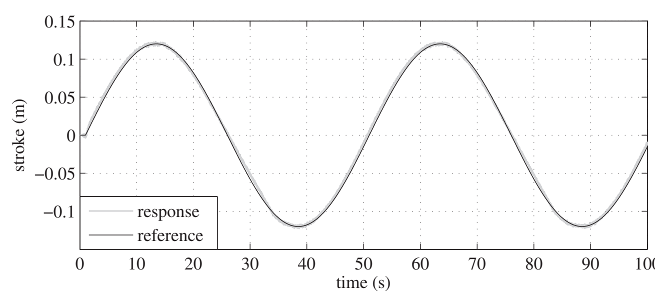

Fig. 9. Reference and measured stroke response, trajectory A.

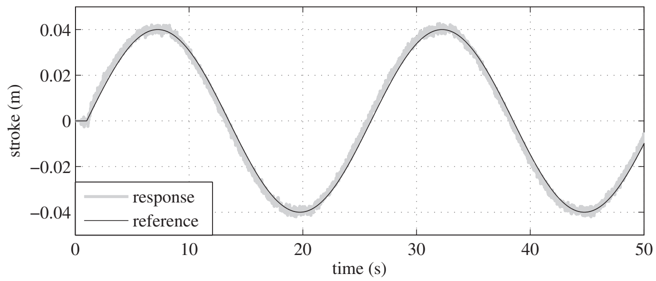

Fig. 10. Reference and measured stoke response, trajectory B.

VI. Conclusions

In this paper, a minimal-model of the controlled cylinders actuating the large-scale hydraulic machines is proposed, as being sufficient for a straightforward parameters estimation and thereupon based robust control design. The minimal-model assumes solely one free gained integrator and dead-time element, which should capture the overall phase lag occurring due to residual non-modeled dynamics. Furthermore the model incorporates the input dead-zone associated with the directional control valves. For a lower frequency range, to which the operation of heavy cylinders is limited, the modeling assumptions reveal as sufficient and provide direct conclusions about the overall system gain and phase margin required for the relative stability analysis. The identified parameters render a quite accurate system description in vicinity to the set characteristic frequency that limits the control bandwidth. The robust parameters convergence is shown for the simple to realize open-loop experiments, without prior knowledge of the overall complex system dynamics. The identified minimal-model parameters allow for a direct parametrization of the combined feed-forward and feedback control loop. The designed controller is experimentally evaluated on two reference trajectories of different angular frequencies, while showing an accurate position tracking despite relatively high level of the process and measurement noise, and relatively low sampling rate of the underlying control system.

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