MODEL BASED DESIGN OPTIMIZATION OF OPERATIONAL RELIABILITY IN OFFSHORE BOOM CRANES

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Abstract

This paper presents a model based approach for design of reliable electro-hydraulic motion control systems for offshore material handling cranes. The approach targets the system engineer and is based on steady-state computations, dynamic time domain simulation and numerical optimization.

In general, the modelling takes into account the limited access to component data normally encountered by engineers working with system design. A system model is presented which includes the most important characteristics of both mechanical system and hydraulic components such as the directional control valve and the counterbalance valve.

The model is used to optimize the performance of an initial design by minimizing oscillations, maximizing the load range and maintaining operational reliability.

Keywords: System modelling, system design, counterbalance valve, directional control valve

1 Introduction

Despite the fact that hydraulics, in general, is considered a mature technology, design of hydraulic motion control systems still offers a number of challenges for both component suppliers and manufacturers of hydraulically actuated machines. For the system designer, the main challenge is to meet the functional requirements for the system, a set of design constraints, while satisfying a number of performance criteria such as cost, weight, overall efficiency and response time, which are often conflicting and also subject to constraints.

Design of hydraulic systems has been subjected to extensive research including steady-state based design (Stecki and Garbacik, 2002) as well as component selection using dynamic simulation and numerical optimization (Krus et al., 1991), (Hansen and Andersen, 2001), (Andersson, 2001) and (Papadopoulos and Davliakos, 2004). Automated design through so-called expert systems with the ability to handle both conceptual and detailed design have attracted a considerable amount of interest from researchers (da Silva and Back, 2000), (Hughes et al., 2001), (Liermann and Murrenhoff, 2005) and (Schlemmer and Murrenhoff, 2008). The impact outside academia, however, remains limited. A reason for this may be that design of hydraulic systems is somewhat application dependent. Design criteria and constraints in combination with design traditions differ from one application area to another, making it difficult to set up and maintain design rules for expert systems. Moreover, scepticism and conservatism may contribute to design engineers being reluctant to make use of such systems.

Hydraulic systems are therefore still designed manually and in many cases based on existing systems, reducing the design job to a sizing problem where the system architecture is already given. In these cases the design engineer can certainly make benefit from previously mentioned tools such as dynamic simulation and optimization. However, using these tools still requires a great level of application specific knowledge.

For offshore applications, which are the focus of this paper, reliability and productivity are the most important performance criteria. They are especially important for offshore applications because of remote locations and high cost of down time. Therefore price and efficiency are less important criteria than for other

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applications like agriculture or construction machines.

In the offshore industry, the problem of designing reliable systems is further complicated by limited opportunities to build prototypes to verify new designs. This only promotes the need for model based design approaches where virtual prototypes can be used to evaluate and optimize a design.

In this paper an offshore material handling application is considered which uses a pressure compensated directional control valve (DCV) and a counterbalance valve (CBV) – classically prone to instability and therefore unreliable. A dynamic model and a typical steady-state sizing procedure are presented as seen from a system designer's point of view. Next, an optimization procedure, based on the Complex method, is applied in order to investigate two different design concepts; one that represents a traditional way of choosing design parameters for the considered system and one that represents a new and reliable way of handling the problem of using a pressure compensated DCV together with a CBV. The two methods are compared and their limitations are discussed.

2 Considered System

The considered system is part of a smaller crane used for material handling on an offshore drilling rig. It is put forward as a representative problem within offshore crane design where the mechanical system and operating cycles are determined a priori.

The mechanical system is a crane boom actuated by a hydraulic cylinder, configured as shown in Fig. 1.

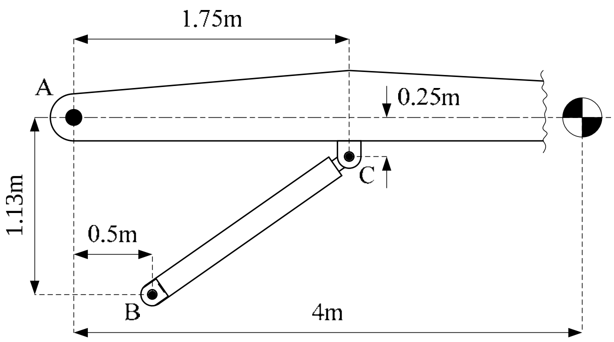

Fig. 1: *Mechanical system*

The total mass of the boom and the load attached to it is 5000 kg with COG located at a distance of 4 m from the pivoting point of the boom.

The cylinder is controlled with an electro-hydraulically actuated DCV with integrated load sensing (LS) circuit and pressure compensator which maintains an approximately constant pressure drop of $p_{comp} - p_{LS} \approx 7 \cdot 10^5$ Pa across the metering edge of the DCV, see Fig. 2. This makes the controlled flow independent of the load pressure and proportional to the position of the DCV main spool.

An externally vented (drained) CBV is used to control the piston pressure, p_2, during load lowering, i.e., when the cylinder is exposed to negative loads. The system is supplied by a hydraulic power unit (HPU) with constant supply pressure, $p_S = 210 \cdot 10^5$ Pa, and return pressure, $p_R = 0$.

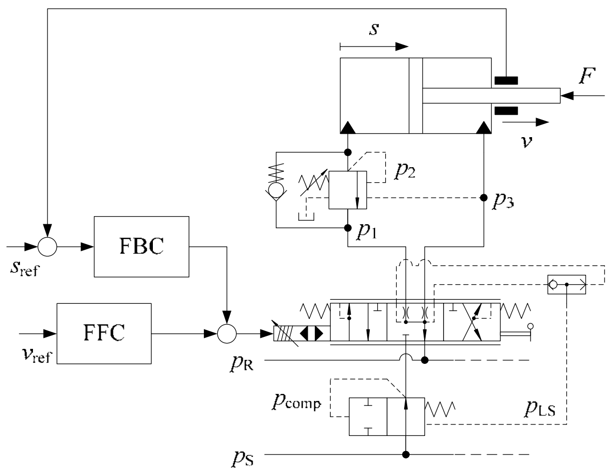

Fig. 2: *Electro-hydraulic motion control system*

The control system consists of a feedforward controller (FFC) which is a scaling of the velocity reference, v_{ref}, and a feedback controller (FBC), which is a PI controller that regulates the actuator position, s, according to the position reference, s_{ref}.

This type of electro-hydraulic motion control system is widely used for offshore material handling equipment where closed loop control of the individual actuators is used to achieve control of a manipulator end point and where several machines are supplied by a common HPU with a ring line system connected to the individual machines.

Design of such a system is an iterative process involving design of both the mechanical system and the motion control system. In practice, detailed design of both systems is carried out separately with constraints imposed by a conceptual design. The conceptual design is then revisited if it later proves to be unsuitable.

As an example placement and size of the cylinder may be determined during the conceptual design phase. To ensure the lifting capacity, the required cylinder piston diameter can be obtained by:

$$D_p \geq \sqrt{\frac{4}{\pi} \cdot \left(\frac{F_{max}}{\eta_{hmc}(p_2 - p_3 \cdot \varphi)} \right)} \tag{1}$$

The force F_{max} includes the steady-state load force and, for offshore equipment, a certain contribution from environmental loads, e.g., wind and waves.

The return pressure, p_3, can often be neglected or alternatively set to e.g., $5 \ldots 10 \cdot 10^5$ Pa to account for the pressure drop through the return path. The inlet pressure, p_2, is typically set to $20 \ldots 30 \cdot 10^5$ Pa below the supply pressure, e.g., $180 \cdot 10^5$ Pa if the supply is $210 \cdot 10^5$ Pa. The hydro-mechanical efficiency, η_{hmc}, depends on the cylinder design and type of sealings. In practice it is typically set to 0.9 for the considered type of cylinder (Rasmussen et al., 1996).

Simultaneously, the minimum required rod diameter must be determined in order to avoid buckling of the cylinder. A simple and common approach is to use the rod diameter as an effective diameter throughout the length of the whole cylinder and apply the formula for Euler buckling of an ideal column:

$$D_{\mathrm{r}} \geq \sqrt[4]{\frac{64 \cdot f_{\mathrm{s}} \cdot (l^2 \cdot F)_{\max}}{\pi^3 \cdot E}} \tag{2}$$

Since the load on the cylinder varies with the position of the piston the maximum product of the squared cylinder length, l, and the force, F, is used and an appropriate safety factor, f_s, is introduced.

The system may have several operating cycles which need to be considered when designing it. For a crane with coordinated control of multiple actuators, inverse kinematics may be applied to obtain the operating cycles for the individual actuators. However, independent operating cycles are usually defined for each actuator and used as design references. The most common is to use a trapezoidal velocity profile to determine the maximum velocity:

$$v_{\max} = \frac{\Delta s}{\Delta t - t_{\mathrm{ramp}}} \tag{3}$$

Cylinders are usually required to use the full stroke, h, i.e., $\Delta s = h$. Δt is the time of the operating cycle, i.e., time for lifting or lowering, and t_{ramp} is the ramping time.

Parameters for the cylinder and the operating cycle, for which the motion control system is to be designed, are given in Table 1.

Table 1: Parameters for cylinder and operating cycle.

Cylinder		
$F_{\max} = 195\text{kN}$	$\eta_{\mathrm{hmc}} = 0.9$	$f_{\mathrm{s}} = 4$
$l_{\min} = 1.536$ m	$h = 1$ m	$E = 2.1 \cdot 10^9$ Pa
$D_{\mathrm{p}} = 0.125$ m	$D_{\mathrm{r}} = 0.08$ m	$\varphi = 0.59$
Operating cycle		
$\Delta t = 10$ s	$t_{\mathrm{ramp}} = 1.5$ s	$v_{\max} = 0.118$ m/s

3 System Modelling

In model based design the actual modelling is closely linked to the design objectives. The main challenge is to minimize the complexity of the system model without ignoring or underestimating important physical phenomena. For systems manufacturers this challenge involves setting up suitable models of a number of sub-supplier components for which the required data may be difficult to acquire or not available. For the considered system this includes, among other parameters, the bandwidth of the DCV, steady-state characteristics of the CBV and cylinder friction.

The model of the hydraulic-mechanical system is developed with MapleSim™ and consists of both predefined library components and custom made components developed via Maple™. This combination facilitates both efficient model development and modelling at a detail level that is not supported by library components.

The design analysis and optimization (section 5) is carried out with MATLAB® and Simulink®. For this purpose an S-function (compiled C-code) is generated from the MapleSim™ model and used to carry out simulations in Simulink®.

The advantage of this use of the two software packages is the speed and efficiency with which models can be developed in MapleSim™ combined with fast simulation of the S-function in Simulink® and predefined functions in MATLAB® for post processing and design optimization.

3.1 Mechanical System

The mechanical system is modelled as a three-dimensional multi-body system with three rigid bodies; boom, cylinder barrel and cylinder piston.

The hinges in points A, B and C (see Fig. 1) are modelled as a revolute joint, a spherical joint and a universal joint, respectively. The translational degree of freedom (DOF) between the cylinder barrel and piston is modelled as a prismatic joint. This gives a system with a single DOF which is actuated by the hydraulic cylinder. Figure 3 shows the chosen model structure as it appears in MapleSim™.

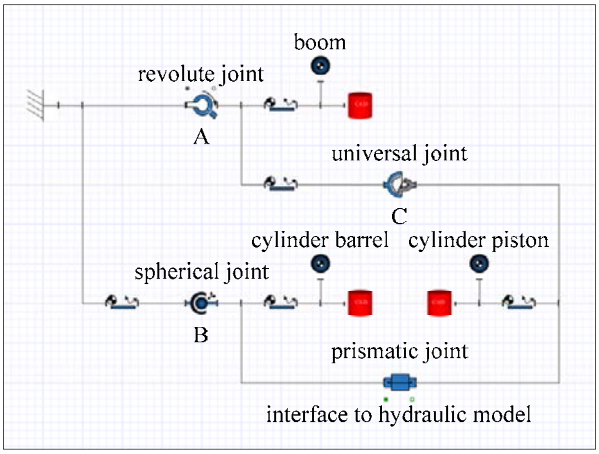

Fig. 3: *Structure of mechanical model*

The flexibility of the boom may have some influence on the system dynamics but is not included here.

3.2 Hydraulic System

For the considered system all hydraulic valves are modelled as variable orifices with linear opening characteristics:

$$Q = \xi \cdot C_{\mathrm{v}} \cdot \sqrt{\Delta p} \tag{4}$$

Here ξ is the relative opening of the valve, i.e., a dimensionless number limited to the interval [0,1]. It can be a function of system pressures or controlled directly via an input signal depending on the considered type of valve.

The flow coefficient in Eq. 4 can be expressed as:

$$C_{\mathrm{v}} = C_{\mathrm{d}} \cdot A_{\mathrm{d}} \cdot \sqrt{\frac{2}{\rho}} \tag{5}$$

The discharge coefficient, C_{d}, and the discharge area, A_{d}, are usually not specified for a valve. Instead C_{v} can be obtained from characteristic flow curves given in the datasheet of the valve. From this, a nominal flow, Q_{nom}, corresponding to a nominal pressure drop, Δp_{nom}, can be identified and used to derive the flow coefficient:

$$C_{\mathrm{v}} = \frac{Q_{\mathrm{nom}}}{\sqrt{\Delta p_{\mathrm{nom}}}} \tag{6}$$

This corresponds to the fully opened state of the valve, so with Eq. 4 it is assumed that the discharge coefficient, C_{d}, is constant and only the discharge area, A_{d}, varies with the relative opening of the valve.

The pipelines between the DCV and the CBV/cylinder are assumed to be short enough to neglect the pressure drop and only the capacitance is modelled:

$$\dot{p} = \frac{\beta}{V} \cdot (Q_{\mathrm{in}} - Q_{\mathrm{out}}) \tag{7}$$

In the following the models of the cylinder, the CBV and the DCV are described.

3.3 Hydraulic Cylinder

The model of the hydraulic cylinder includes the capacitance of the chambers as well as the friction between the piston and the barrel. The cylinder force is:

$$F = \begin{cases} F_{\mathrm{h}} - F_{\mathrm{friction}} & v_0 < v \\ F_{\mathrm{h}} - F_{\mathrm{friction}} \cdot \dfrac{v}{v_0} & -v_0 \le v \le v_0 \\ F_{\mathrm{h}} + F_{\mathrm{friction}} & v < -v_0 \end{cases} \tag{8}$$

The hydraulic force is $F_{\mathrm{h}} = p_2 \cdot A_{\mathrm{p}} - p_3 \cdot \varphi \cdot A_{\mathrm{p}}$. The actual friction in the cylinder is quite complex, especially around zero velocity. As described in Ottestad (2012) it consists of both static and coulomb friction as well as velocity dependent and pressure dependent friction, which may be described with a model of five parameters. Even though the model is not very complex, the number of parameters represents a problem for a system designer because they need to be experimentally determined.

Consequently, an even simpler model must be used:

$$F_{\mathrm{friction}} = F_{\mathrm{S}} + C_{\mathrm{p}} \cdot |F_{\mathrm{h}}| \tag{9}$$

The first term is the static friction which may be set to $F_{\mathrm{S}} \approx A_{\mathrm{p}} \cdot 1 \cdot 10^5$ m^2·Pa. In Ottestad (2012) it was experimentally determined to $F_{\mathrm{S}} = 580$ N for a cylinder of the same size as the one used here. The second term is the pressure dependent friction which may constitute 2…3 % of the hydraulic force, e.g., $C_{\mathrm{p}} = 0.02$. v_0 in Eq. (8) is used to handle the transition around zero velocity in order to avoid computation difficulties and can be set to a small value of, e.g., $v_0 = 0.005$ m/s.

Even though the friction model described by Eq. 9 is quite simple, it is sufficient for the considered system because the operating cycle does not contain any position control around zero velocity.

The pressure gradients in the two chambers are:

$$\dot{p}_2 = \frac{\beta}{V_1} \cdot (Q_1 - v \cdot A_{\mathrm{p}}) \tag{10}$$

$$\dot{p}_3 = \frac{\beta}{V_2} \cdot (v \cdot \varphi \cdot A_{\mathrm{p}} - Q_2) \tag{11}$$

The chamber volumes, V_1 and V_2, are functions of the piston position, s. Q_1 and Q_2 are the flows on the piston-side and the rod-side of the cylinder, respectively.

3.4 Counterbalance Valve

The model of the CBV consists of two components; check valve and pilot assisted relief valve (the CBV itself), both modelled according to Eq. 4.

The relative opening of the check valve is:

$$\xi_{\mathrm{cv}} = \frac{p_1 - p_2 - p_{\mathrm{cr,cv}}}{k_{\mathrm{s,cv}}} \tag{12}$$

The cracking pressure, $p_{\mathrm{cr,cv}}$, of the check valve is usually specified for a CBV and the normalized spring stiffness can be set to a value of, e.g., $k_{\mathrm{s,cv}} = 1 \cdot 10^5$ Pa.

The relative opening for the CBV is:

$$\xi_{\mathrm{cbv}} = \frac{p_3 \cdot \psi + p_2 - p_{\mathrm{cr,cbv}}}{k_{\mathrm{s,cbv}}} \tag{13}$$

The normalized spring stiffness, $k_{\mathrm{s,cbv}}$, is usually not specified for a CBV but may be provided by the supplier on request. With Eq. 13 any non-linearity of the discharge area as function of the spool position is neglected. Also variations in the discharge coefficient and the influence of flow forces are neglected.

To determine the usefulness of the proposed model, experimental work has been carried out with a double CBV often used in offshore cranes. The pilot area ratio of the CBV is $\psi = 5$ and the nominal pressure drop is $\Delta p_{\mathrm{nom}} = 16 \cdot 10^5$ Pa at a nominal flow $Q_{\mathrm{nom}} = 120$ l/min in the fully opened state. This corresponds to a flow coefficient of $C_{\mathrm{v,cbv}} = 1.58 \cdot 10^{-6}$ m^3/s·Pa$^{0.5}$. The cracking pressure is $p_{\mathrm{cr,cbv}} = 180 \cdot 10^5$ Pa.

In the experiment the flow through the valve is increased linearly from 0 to 100 l/min. The flow is recorded along with three pressures, see Fig. 4. This is used to estimate the opening of the CBV using both Eq. 4 and 13 in order to identify the spring stiffness.

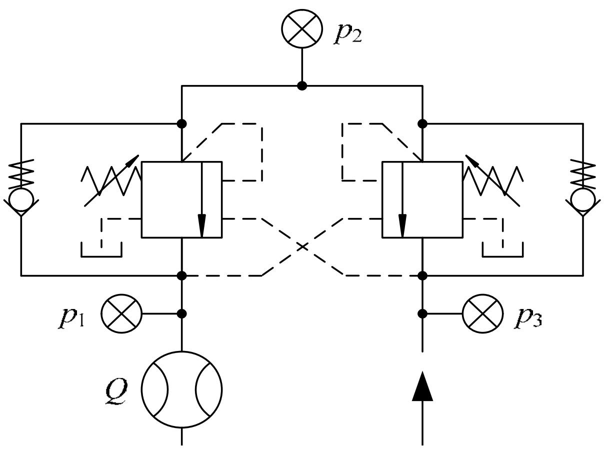

Fig. 4: *Experimental setup for test of CBV*

With the measured flow and pressures, p_1 and p_2, (measured values are indicated by ~) the relative opening can be estimated using Eq. 4:

$$\xi_{\mathrm{est}}^{(1)} = \frac{\tilde{Q}}{C_{\mathrm{v}} \cdot \sqrt{\tilde{p}_2 - \tilde{p}_1}} \tag{14}$$

To investigate the validity of Eq. 13 this is used to estimate the opening with p_2 and p_3 as inputs:

$$\xi_{\mathrm{est}}^{(2)} = \frac{\tilde{p}_2 + \tilde{p}_3 \cdot \psi - p_{\mathrm{cr,cbv}}}{k_{\mathrm{s,cbv}}} \tag{15}$$

The spring stiffness is tuned until the best match between $\xi_{est}^{(1)}$ and $\xi_{est}^{(2)}$ is obtained. The result is shown in Fig. 5 together with the input flow. The relative openings are given in percentage.

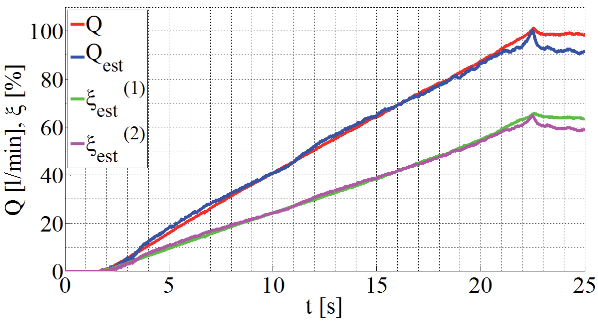

Fig. 5: *Estimated and openings and flow and measured flow*

The best match between the two estimated openings is obtained with $k_{s.cbv}$ = 295·10^5 Pa. Also shown in Fig. 5 is the estimated flow obtained by means of $\xi_{est}^{(2)}$ and p_1 and p_2:

$$Q_{est} = \xi_{est}^{(2)} \cdot C_v \cdot \sqrt{\tilde{p}_2 - \tilde{p}_1} \tag{16}$$

From the correspondence shown in Fig. 5 there seems to be limited influence from flow forces and, although the suggested model is relatively simple, the results show that it is adequate to describe the steady-state characteristics of the considered type of CBV.

3.5 Directional Control Valve

The DCV model consists of several smaller models representing the main components of the valve; main spool, LS circuit and pressure compensator. The model does not include the functionality of the electro-hydraulic actuation, but instead a simplified model of the dynamics between the input signal and the main spool position, represented by a second order system:

$$\frac{u_{spool}}{u_{ref}} = \frac{1}{\frac{s^2}{\omega_v^2} + 2 \cdot \zeta \cdot \frac{s}{\omega_v} + 1} \tag{17}$$

The control signal, u_{ref}, is the sum of the feedforward control signal, u_{FF}, and the feedback control signal, u_{FB}. The spool position, u_{spool}, is a normalized signal which can vary continuously between -1 and 1, with 0 being the centre position of the spool.

For servo valves and high-performance proportional valves the bandwidth, ω_v, can usually be identified from the valves datasheet. For pressure compensated DCVs there is usually no information available about the bandwidth and the only way to identify it may be to carry out a frequency response test of the considered valve. An approach for such a test is described in Bak and Hansen (2012) along with some test results for a Sauer-Danfoss PVG32. The identified bandwidth, ω_v = 30 s^{-1}, and damping ratio, ζ = 0.8, are also used here. This represents the overall dynamics between the input signal and the controlled flow, i.e., it includes the dynamics of the electro-hydraulic actuation, the main spool, the pressure compensator and the LS circuit of the DCV.

The four spool edges are modelled as variable orifices, Eq. 4, of which the relative opening, ξ_{edge}, for each spool edge is a function of the normalized spool position signal as shown in Fig. 6.

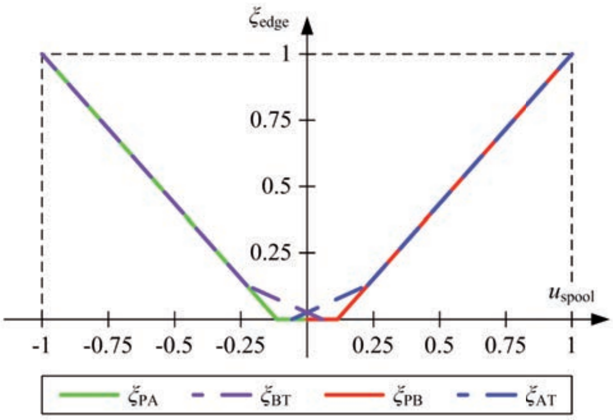

Fig. 6: *Opening functions for spool edges*

The spool edge openings are modelled as piecewise functions to include the overlaps of the metering edges, PB and PA, and the underlaps of the return edges, AT and BT. A more detailed description of the opening functions is given in Bak and Hansen (2012).

Usually very little or no information about the pressure compensator is available from valve datasheets making it difficult to estimate when the pressure saturation will occur. However, if the nominal pressure drop, p_0, is known the compensated pressure, p_{comp}, can be described as:

$$p_{comp} = \begin{cases} p_{LS} + p_0 & p_0 \le p_S - p_{LS} \\ p_S - p_{LS} & 0 < p_S - p_{LS} < p_0 \\ p_{LS} & p_S - p_{LS} \le 0 \end{cases} \tag{18}$$

The first case describes the normal operating condition of the compensator, maintaining the nominal pressure drop. The second case describes the condition where the load pressure is too close to the supply pressure to maintain the nominal pressure drop. The third case describes the build-in check valve function of the compensator that prevents negative flow if the load pressure exceeds the supply pressure.

The LS circuit directing the load pressure to the pressure compensator is modelled as a piecewise function:

$$p_{LS} = \begin{cases} p_3 & u_{spool} < 0 \\ 0 & u_{spool} = 0 \\ p_1 & 0 < u_{spool} \end{cases} \tag{19}$$

3.6 Model Structure

Figure 7 shows the model structure of the hydraulic system as it appears in the graphical environment of MapleSim™. The red lines represent transfer of the two hydraulic power variables, pressure and flow, between the hydraulic components. The blue lines are signal lines.

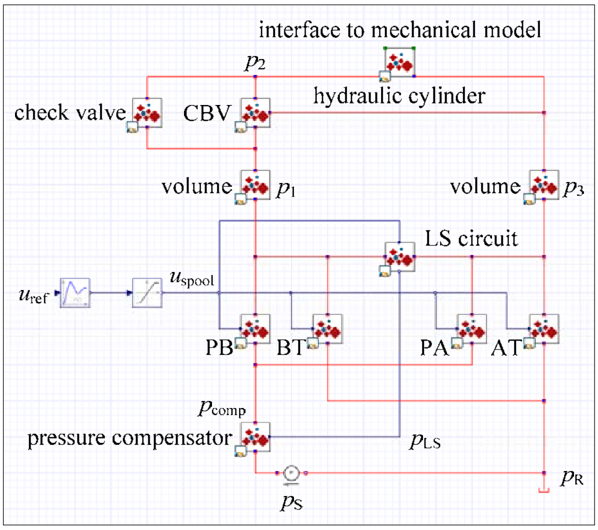

Fig. 7: *Structure of hydraulic model*

3.7 Control System

In the control system model the velocity reference, v_{ref}, is split into two signals, for lifting and lowering, which are scaled by the two feedforward gains, $K_{v,1}$ and $K_{v,2}$ respectively, to make up the feedforward controller (FFC). The velocity reference is integrated to obtain the position reference, s_{ref}, for the feedback controller (FBC).

The structure of the Simulink® model, with the control system and the S-function containing the hydraulic-mechanical model, is shown in Fig. 8.

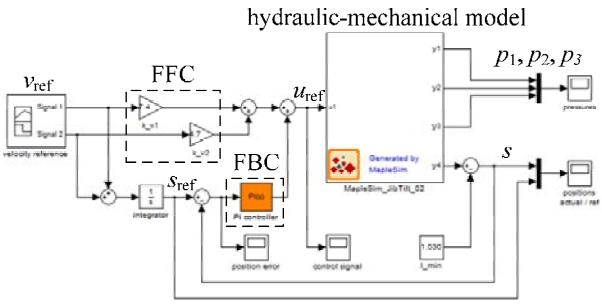

Fig. 8: *Structure of Simulink® model*

The input to the hydraulic-mechanical model is the control signal to the DCV, u_{ref} in Eq. 17. The outputs are the three system pressures, p_1, p_2 and p_3 and the position of the cylinder piston, s. The latter is used to compute the position error, $e_s = s_{ref} - s$, which is the input to the feedback controller (FBC).

4 Steady-State Design

The task of designing a hydraulic system involves two main activities; choice of system architecture and sizing/selection of the system components. The first one is often based on design rules and experience with suitable architectures for the considered application.

Selection and sizing of the system component is an iterative process because the choices of the individual component affect each other. Changes regarding types and sizes of components may need to be made after the first design iteration and even a change of system architecture may need to be included in the following iterations before arriving at a satisfying design.

To reduce the number of iterations the design process can be set up as a systematic and stepwise procedure based on simple steady-state considerations and empirical design rules. An example of a general procedure is described by Stecki and Garbacik (2002).

For the considered system, with the cylinder and the operating cycle given as in section 2, the remaining system may be designed through the following steps:

- Select directional control valve.
- Select counterbalance valve.
- Tune control system.

Furthermore pipelines and any protective components such as shock and anti-cavitation valves also need to be sized. HPU design including selection and sizing of pump(s), sizing of reservoir, design of cooling system and selection of filtering system could represent additional steps in the procedure. However, since the HPU is used to supply several machines, this is designed through a separate procedure based on the requirements and operating cycles of all the machines it is used to supply.

After the actual design procedure the design must be evaluated, first of all to ensure that the choices of system components do not conflict with each other or with any of the design criteria.

4.1 Directional Control Valve

The DCV is selected according to the maximum flow required by the actuator:

$$Q_{max} = v_{max} \cdot A_p \tag{20}$$

To ensure enough flow for all situations, a valve with a higher capacity may be selected. However this choice is often implicitly made since rated flow of a valve is a discrete design variable and the valve with the nearest flow capacity above the required is chosen.

For hydraulic systems with closed loop control, an important property of the DCV is the bandwidth. Manufacturers of servo valves usually recommend choosing a valve with a bandwidth, ω_v, which is at least three times higher than the natural frequency, ω_{sys}, of the system it is used to control (MOOG, 2012):

$$\omega_v \geq 3 \cdot \omega_{sys} \tag{21}$$

This applies if the valve should not affect the overall bandwidth of the series connection of the valve and the hydraulic-mechanical system it is used to control.

For many applications such as material handling cranes, it is the opposite, i.e., the bandwidth of the DCV is lower than the one of the hydraulic-mechanical system. This is also reflected by the fact that pressure compensated DCVs have relatively low bandwidths, up to 5 Hz (Bak and Hansen, 2012).

Figure 9 shows the natural frequency of the considered system as function of the cylinder piston position, s, for three different load cases.

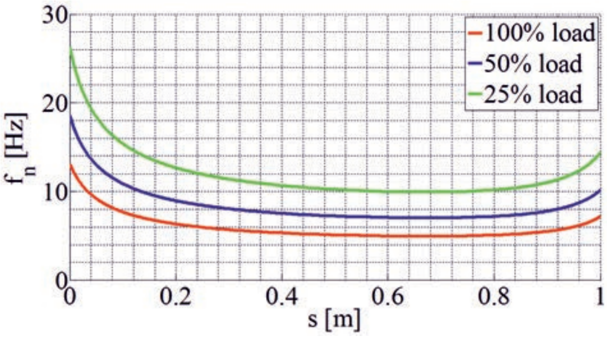

Fig. 9: *Natural frequency of hydraulic-mechanical system*

Choosing a DCV with a bandwidth of 5 Hz is sufficient for this application as long as the ramping times for the velocity reference are not too small.

4.2 Counterbalance Valve

The size of the CBV is chosen according to the maximum flow required or induced by the actuator or it may be chosen to match the rated flow of the DCV.

In order to avoid unintended opening of the CBV the cracking pressure is typically set to a factor of 1.3 above the maximum load induced pressure (Sun Hydraulics, 2012):

$$p_{\mathrm{cr,cbv}} \geq 1.3 \cdot p_{\mathrm{load,max}} \tag{22}$$

One of the most critical design variables of a system containing CBVs is the pilot area ratio, ψ, due to its strong influence on the system stability during load lowering. It is therefore usually chosen based on experience and may need to be changed once the system has been realized and tested.

Some basic steady-state considerations for choosing ψ may be applied to ensure that cavitation does not occur on the metering side of the cylinder during load lowering.

The cylinder force during negative loads:

$$F = \frac{1}{\eta_{\mathrm{hmc}}} \cdot (p_2 \cdot A_{\mathrm{p}} - p_3 \cdot \varphi \cdot A_{\mathrm{p}}) \tag{23}$$

And the opening condition for the CBV:

$$p_2 + \psi \cdot p_3 = p_{\mathrm{cr,cbv}} \tag{24}$$

Can be combined to express the upper limit of ψ:

$$\psi \leq \frac{p_{\mathrm{cr,cbv}} - \frac{F \cdot \eta_{\mathrm{hmc}}}{A_{\mathrm{p}}}}{p_3} - \varphi \tag{25}$$

p_3 is set to the preferred safety margin e.g., $10\ldots20 \cdot 10^5$ Pa in order to avoid cavitation.

4.3 Control System

The feedback controller gains cannot be determined based on steady-state considerations and therefore have to be tuned after the system has been realized. The feedforward gains, on the other hand, can be estimated when using a pressure compensated DCV.

For a DCV with a symmetrical and linear flow characteristic (neglecting the deadband of the valve) the cylinder velocity is:

$$v \approx \frac{u_{\mathrm{ref}} \cdot Q_{\mathrm{max}}}{A_{\mathrm{Q}}} \tag{26}$$

$$A_{\mathrm{Q}} = \begin{cases} A_{\mathrm{p}} & 0 \leq u_{\mathrm{ref}} \\ \varphi \cdot A_{\mathrm{p}} & u_{\mathrm{ref}} < 0 \end{cases} \tag{27}$$

In Eq. 26 Q_{max} is the maximum flow of the valve. With a velocity reference the feedforward control signal can be computed:

$$u_{\mathrm{FF}} = v_{\mathrm{ref}} \cdot K_{\mathrm{v}} \tag{28}$$

Combining Eq. 26 and 28 yields the feed forward gain:

$$K_{\mathrm{v}} = \frac{A_{\mathrm{Q}}}{Q_{\mathrm{max}}} \tag{29}$$

4.4 System Characteristics

Employing the sizing procedure described through Eq. 20 to 29 yields a number of design parameters for the considered system. They are listed in Table 2 together with the most important model parameters.

Table 2: Parameters for hydraulic system.

Directional control valve	
ω_{v} = 30 rad/s	ζ = 0.8
$C_{\mathrm{v}} = 2 \cdot 10^{-6}$ m^3/s·Pa$^{0.5}$	overlap: 10%
$p_0 = 7 \cdot 10^5$ Pa	$k_{\mathrm{s,cbv}} = 0.5 \cdot 10^5$ Pa
Counterbalance valve	
ψ = 3	$C_{\mathrm{v}} = 1.58 \cdot 10^{-6}$ m^3/s·Pa$^{0.5}$
$p_{\mathrm{cr,cbv}} = 235 \cdot 10^5$ Pa	$k_{\mathrm{s,cbv}} = 300 \cdot 10^5$ Pa
$p_{\mathrm{cr,cv}} = 2 \cdot 10^5$ Pa	$k_{\mathrm{s,cbv}} = 1 \cdot 10^5$ Pa
Control system	
$K_{\mathrm{v,1}}$ = 7.4	$K_{\mathrm{v,2}}$ = 4.3

With the velocity reference specified in section 2, see Fig. 10, a simulation is carried out to analyse the response of the system.

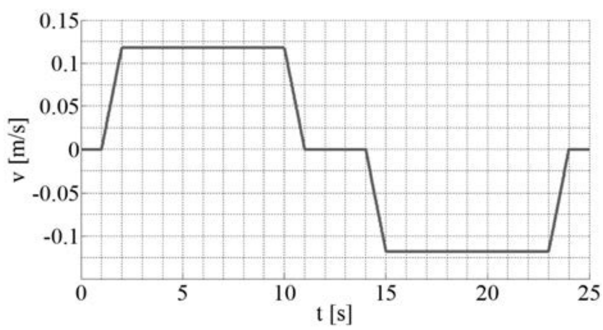

Fig. 10: *Velocity reference, v_{ref}*

With the feedforward gains given in Table 2 and the feedback controller disabled the piston position shown in Fig. 11 is obtained.

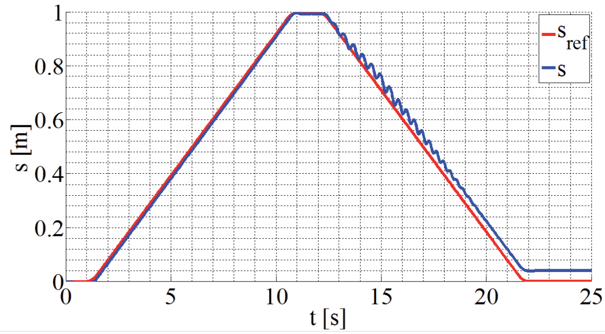

Fig. 11: *Position reference, s_{ref}, versus simulated position, s*

The simulated system pressures are shown in Fig. 12.

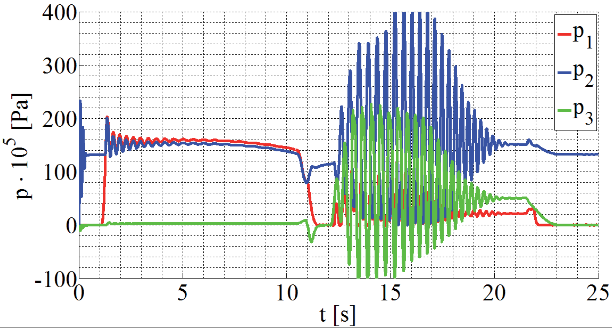

Fig. 12: *Simulated system pressures p_1, p_2 and p_3*

As seen from both Fig. 11 and 12 the system becomes strongly oscillatory during the lowering sequence and it is not able to follow the prescribed position reference with the feedforward controller alone.

As the last step in a design procedure the feedback controller can relatively easy be tuned to remove the accumulated position error seen in Fig. 11.

5 Dynamic Considerations

The system response shown in Fig. 11 and 12 illustrates a classical problem with systems using a CBV in combination with a pressure compensated DCV. If care is not taken the system is likely to become so oscillatory that it is uncontrollable. Furthermore, the negative values of p_3, though they cannot occur in reality, indicate that the oscillations are violent enough to cause cavitation in rod-side chamber of the cylinder.

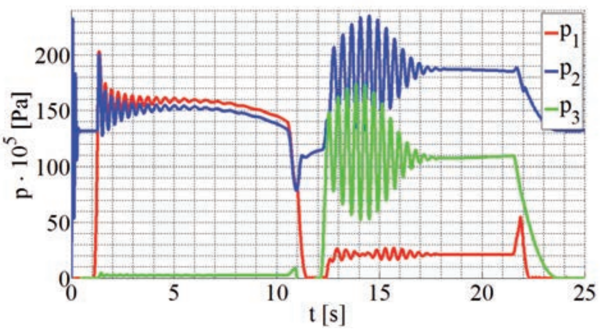

Fig. 13: *Simulated pressures with $\psi = 1$*

The system response is directly influenced by the chosen pilot area ratio, ψ, which is usually set high for energy efficiency purposes and only lowered if the system, as in this case, becomes strongly oscillatory.

From Eq. 25 a pilot area ratio of $\psi = 3$ is obtained. In Fig. 13 the pressure response with $\psi = 1$ is shown.

Lowering the pilot area ratio clearly reduces the oscillations. However, a significant level of oscillation remains and often it is necessary to take extra measures to further reduce the oscillations.

5.1 Reliability and Dynamic Performance

Strong oscillations may lead to excessive noise, wear and fatigue, and as shown in Fig. 12, loss of functionality. Hence, these oscillations are directly related to the reliability of the system.

The oscillatory nature of the considered system has been investigated several times (Miyakawa, 1978), (Overdiek, 1981), (Persson et al., 1989), (Handroos et al., 1993), (Chapple and Tilley, 1994), (Ramli et al., 1995), (Zähe, 1995) and (Andersen et al., 2005) and it is well established that it is caused by having two active throttling control valves in series; the pressure compensator of the DCV and the CBV. The oscillations can be reduced by lowering the pilot area ratio, bleeding of the CBV's pilot line or narrowing the return edge of the DCV. These measures can, however, not remove the oscillatory nature related to the CBV and therefore they do not improve the reliability of the system, they only improve the dynamic performance.

Reliability may be divided into several categories, e.g. operational reliability and safety related reliability, and their importance may vary from one application to another. While both types of reliability are important for offshore applications, the first one is the main concern here. The operational reliability, i.e. how well and reliable the system can be controlled, is directly influenced by its oscillatory nature.

In order to improve the operational reliability, the concept of two throttling valves in series must be abandoned. This can be done either by removing the pressure compensator or by removing the CBV. The first measure is normally avoided in offshore applications because it increases the demands on the feedback controller and therefore introduces another loss of operational reliability. The second measure is only partially possible because the CBV has other functionalities than controlling the load lowering. It can, however, be obtained by forcing the CBV to open completely, as described by Nordhammer et al. (2012), making it work as a fixed orifice. This will, in turn, require that the throttling is handled by the return edge of the DCV.

While this eliminates the oscillations caused by the CBV and therefore improves the operational reliability, the safety related reliability is likely to be reduced due to an increased closing time of the CBV and a possibly increased risk of the CBV getting stuck open. This is critical in case of hose rupture and therefore the method may need to be combined with safety increasing features or monitoring functions. Further safety assessment, however, is not carried out here.

Reducing the pilot area ratio of the CBV or transferring the return throttling to the DCV will reduce the load range that can be handled during lowering. The loss in efficiency, on the other hand, is not a big issue since the considered system is supplied from a ring line with a constant pressure level.

Availability becomes an issue if the return throttling is to be handled by the DCV, since this requires a tailor-made main spool whereas a CBV is usually available with several different pilot area ratios.

The oscillatory behaviour of the considered system can be investigated by means of simulation. The pilot area ratio and the return edge of the DCV are the design parameters that must be chosen with the objective to:

- Minimize oscillations.
- Maximize load range.

Simultaneously, it must be chosen if the operational reliability should be increased by forcing the CBV open and if the availability should be reduced by demanding a tailor-made main spool for the DCV.

5.2 Design Optimization

The most effective way to handle this is by means of numerical optimization using a suitable algorithm based on minimization techniques. In the following, two different design concepts are subjected to optimization and compared with the classical concept of only lowering the pilot area ratio and throttling with the CBV. One of them represents a semi-classical concept; throttling with the CBV and the DCV. The other represents a new concept; forcing the CBV to open completely and throttling with the DCV. Both designs, though, represent less available concepts since they require tailor-made DCV main spools.

For both concepts the design variables are the pilot area ratio of the CBV and the size of the return edge of the DCV. The available range of pilot area ratios is ψ = [1, 1.5, 2, 2.5, 3, 4, 5] and it is assumed that the return edge of the DCV main spool can be machined within the interval $C_{v,BT} = [0.5 \cdot 10^{-6}, 2 \cdot 10^{-6}]$ $m^3/s \cdot Pa^{0.5}$.

5.3 Design Objectives

The first objective is to minimize the oscillations of the load controlling pressure, p_2, during the lowering sequence. To determine the oscillations level, p_2 is first low-pass filtered to determine the steady-state pressure, $p_2^{(ss)}$, around which it oscillates. The oscillations of p_2 can then be found:

$$p_2^{(osc)} = p_2 - p_2^{(ss)} \tag{30}$$

To obtain a single quantity representing the oscillation level, $p_2^{(osc)}$ is squared and integrated over the time of the lowering sequence:

$$O = \int_{t_1}^{t_2} \left(p_2^{(osc)}\right)^2 dt \tag{31}$$

Before starting the optimization procedure a nominal oscillation level, O_{nom}, of the initial design is found. This is used to obtain a normalized oscillation level (an error) during the optimization:

$$e_i^{(1)} = \frac{O_i}{O_{nom}} \tag{32}$$

Here O_i is the oscillation level of the i'th design suggested by the optimization routine.

For the concept of throttling with the CBV and DCV, Eq. 32 is used for the objective function. For the concept of throttling with the DCV, an additional objective is used to penalize the design if the CBV is not fully open:

$$e_i^{(2)} = 0.5 \cdot \frac{O_i}{O_{nom}} + 0.5 \cdot (1 - \bar{\xi}_{cbv_i}) \tag{33}$$

Here $\bar{\xi}_{cbv_i}$ is the mean value of the CBV opening during the lowering sequence.

In both cases two additional objectives must be met; the steady-state level of p_3 cannot exceed $180 \cdot 10^5$ Pa (to ensure the functionality of the DCV) and the steady-state level of p_2 cannot exceed $250 \cdot 10^5$ Pa (rated pressure of cylinder). If any of these two implicit constraints are violated, Eq. 32 and 33 are overruled and the error is set to $e_i = 1$. Similarly, violation of explicit constraints (limits of design variables) is penalized by setting the error $e_i = 1$.

The second objective of maximizing the load range is handled by evaluating Eq. 32 or 33 for two load cases; one with a load of 5000 kg (maximum load) and one with a lower load which is the minimum load the considered design can handle without violating any of the implicit constraints. The objective function used by the optimization algorithm can then be described as:

$$e_i = 0.5 \cdot e_i^{(maxload)} + 0.5 \cdot e_i^{(minload)} \tag{34}$$

Here, both $e_i^{(maxload)}$ and $e_i^{(minload)}$ are represented by either Eq. 32 and 33 depending on the considered design concept.

5.4 Optimization Procedure

The optimization problem, to minimize e_i, is solved with the Complex method (Box, 1965) with the modification that violation of both implicit and explicit constraints is handled by penalization, i.e setting $e_i = 1$ as earlier described.

The method is often used for optimization of hydraulic systems and has the advantage of being fast and easy to implement (Andersson, 2001). The structure of the optimization procedure is shown in Fig. 14.

The initial design, $x_{ini} = [\psi, C_{v,BT}]$ according to Table 2, is evaluated by running a simulation to determine the system pressures, i.e., the response shown in Fig. 12. In the simulation the feedback controller is disabled and only the feedforward signal is used to control the system.

The nominal oscillation level, O_{nom}, is determined by means of Eq. 30 and 31 and passed on for evaluation of new designs. Next, a random design population is generated within the limits of the design variables, X_{lim}. The size of the population is twice the number of design variables, i.e., X_{pop} contains four designs.

Two simulations are then carried out for each design; one with the maximum load to be handled and one with the minimum load the design is able to handle during lowering. The minimum load is identified by running the optimization procedure a number of times, each time reducing the minimum load until it is no longer possible to find a design that works for both maximum and minimum load.

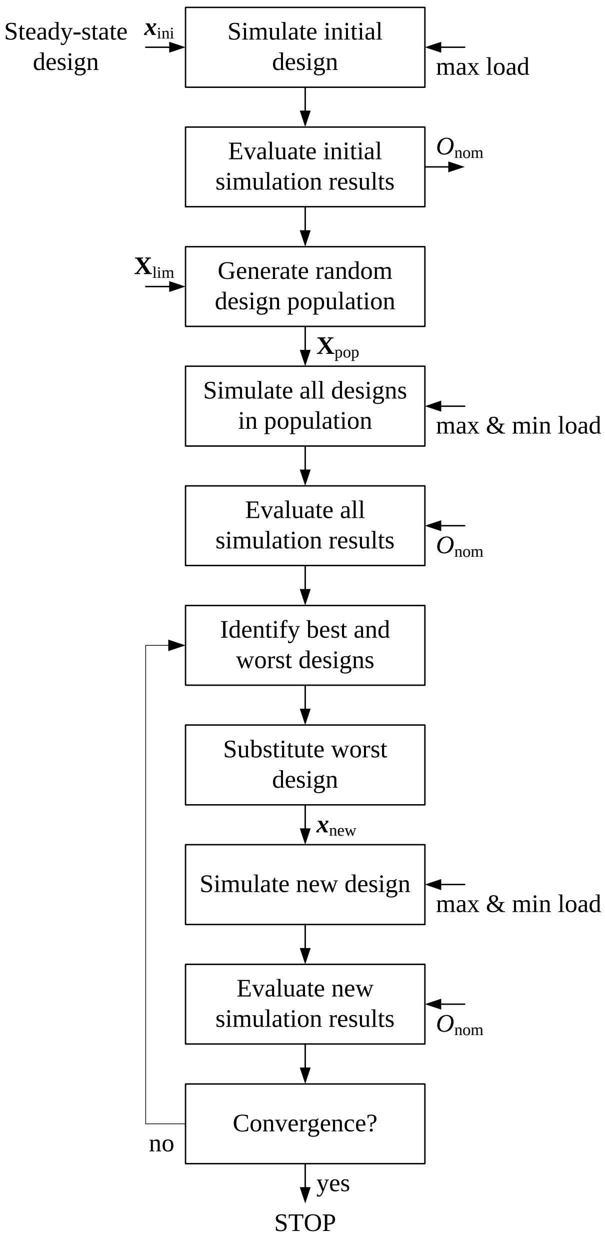

Fig. 14: *Structure of optimization procedure*

Each design in the population is then evaluated by means of either Eq. 32 or 33 together with Eq. 34 depending on the design concept. From the four designs the best and the worst designs are identified. The best design is the one yielding the lowest objective function value and the worst the one yielding the highest. The worst design is then substituted according to the Complex method; i.e., reflecting the point representing the worst design through the centroid of the remaining points in the design space in order to obtain a new design.

A simulation of the new design, x_{new}, is carried out and the design is evaluated. The procedure of substituting the worst design continues until the convergence criterion is met; i.e., the objective function values of the best and the worst design are equal with some tolerance.

6 Results

First, the concept of forcing the CBV to open completely and throttling with the DCV is investigated. This is functional in the load range 3000 - 5000 kg (60 - 100 % load) with ψ = 5 and $C_{v,BT}$ = 0.7·10^{-6} m^3/s·Pa$^{0.5}$ as the optimal design values. The pressures for the maximum and minimum loads are shown in Fig. 15 and 16.

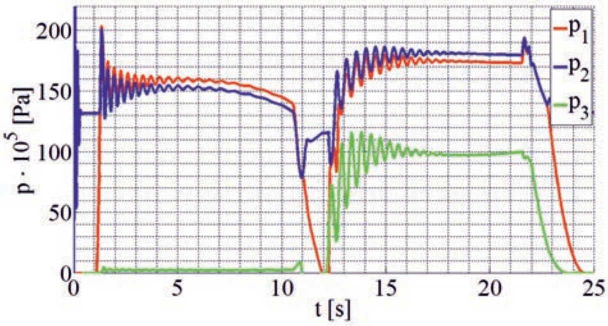

Fig. 15: *100 % load, ψ = 5, $C_{v,BT}$ = 0.7·10^{-6} m^3/s·Pa$^{0.5}$*

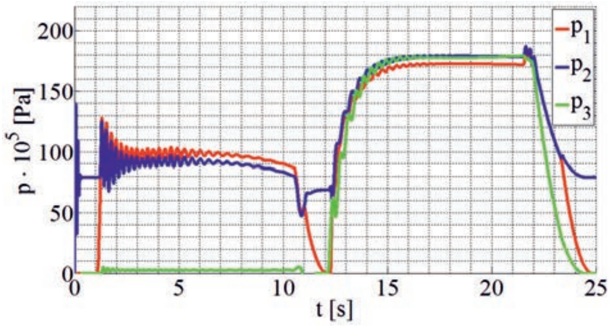

Fig. 16: *60 % load, ψ = 5, $C_{v,BT}$ = 0.7·10^{-6} m^3/s·Pa$^{0.5}$*

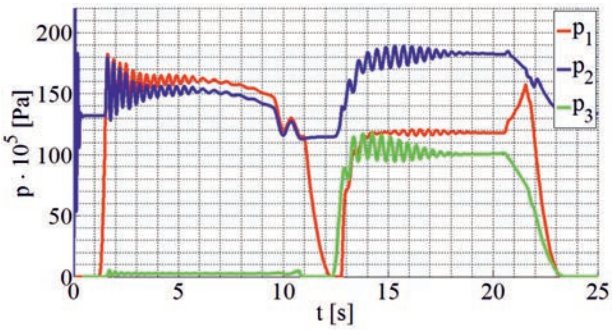

Fig. 17: *100 % load, ψ = 1.5, $C_{v,BT}$ = 0.85·10^{-6} m^3/s·Pa$^{0.5}$*

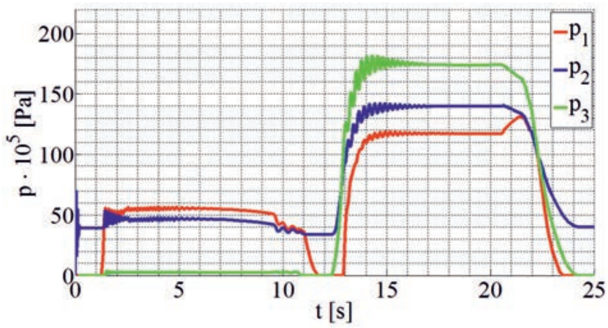

Fig. 18: *30 % load, ψ = 1.5, $C_{v,BT}$ = 0.85·10^{-6} m^3/s·Pa$^{0.5}$*

While the oscillations are nearly removed at 60 % load, some oscillations remain at 100 % load. This is partly due to the limit $p_3^{(ss)}$ < 180·10^5 Pa which prevents further narrowing of the return edge. Also, the nature of the system together with the chosen ramping time will cause some oscillations as also seen from the lifting sequence.

Secondly, the concept of throttling with the CBV and the DCV is investigated. This is functional in the load range1500 – 5000 kg (30 - 100% load) with ψ = 1.5 and $C_{v,BT} = 0.85 \cdot 10^{-6}$ m^3/s·Pa$^{0.5}$ as the optimal design values. The system pressures for the two load cases are shown in Fig. 17 and 18.

In terms of dynamic performance, i.e., oscillation level, the concept of throttling with the CBV and the DCV generally yields slightly better results than the concept of throttling with the DCV. Both concepts are better than the classical concept of throttling with the CBV, in terms of dynamic performance.

When considering the operational reliability, however, the concept of throttling with the DCV is the only fully reliable design because the CBV is active in the other two concepts and, inherently, contribute to less predictable system behaviour. The evaluation of the different concepts can be summarized as in Table 3.

Table 3: Evaluation of return throttling concepts.

Throttling concept	Dynamic performance	Operational reliability
DCV	+	+
CBV + DCV	+	÷
CBV	÷	÷

If only small load variations are to be handled, the concept of forcing the CBV to open completely and throttling with the DCV represents the best design. For larger load ranges, this may not be a functional design, but throttling with the CBV and DCV is a better solution than only throttling with the CBV.

Requiring a tailor-made main spool for the DCV represents an additional cost of the system. For offshore applications operational reliability is the important performance criterion and it is therefore often accepted to acquire components with some degree of customization.

7 Conclusions

In this paper, an approach for model based design of electro-hydraulic motion control systems for offshore material handling cranes is put forward. The design procedure targets the system engineer and therefore one of the main challenges is to establish reliable system models with a suitable level of complexity. Models of the main components of the hydraulic system, which include key parameters such as bandwidth of the directional control valve (DCV) and steady-state characteristics of the counterbalance valve (CBV) is presented.

A typical steady-state design procedure is presented and used to determine the parameters of the main components of the motion control system. The presented system model is then used to demonstrate the problem of the inherently oscillatory behaviour that characterizes the considered system.

In order to improve the dynamic performance and the operational reliability of the system, an optimization procedure, based on the Complex method, is applied to the simulation model in order to optimize design parameters for the DCV and the CBV.

With the objective to reduce the oscillation level during operation of the system, three different design concepts have been investigated:

The classical concept, throttling with the CBV, where the pilot area ratio of the CBV is lowered.

A semi-classical concept, throttling with the CBV and the DCV, where the pilot area ratio of the CBV is lowered and the return edge of the DCV is narrowed.

A new concept, throttling with the DCV, where the CBV is forced to open completely by increasing the pilot area ratio and narrowing the return edge of the DCV.

The main advantage of the classical concept is availability and load range. The main advantage of the new concept is operational reliability. The semi-classical concept and the new concept both have an advantage in terms of dynamic performance.

In an offshore context where operational reliability is the most important criterion, the new concept is ideal as long as its poor load range capability is acceptable, i.e., for applications with small load variations.

Acknowledgements

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Nomenclature

A_d	Discharge area	[m^2]
A_p	Piston area	[m^2]
A_Q	Controlled cylinder area	[m^2]
C_d	Discharge coefficient	[-]
C_v	Flow coefficient	[m^3/s·Pa$^{0.5}$]
C_p	Pressure friction coefficient	[-]
D_p	Piston diameter	[m]
D_r	Rod diameter	[m]
e	Error/objective function value	[-]
e_s	Position error	[m]
E	Young's modulus	Pa
f_s	Safety factor	[-]
F	Load force/cylinder force	[N]
$F_{friction}$	Friction Force	[N]
F_h	Hydraulic force	[N]
F_S	Static friction force	[N]
h	Cylinder stroke	[m]
k_s	Normalized spring stiffness	[Pa]
K_I	Integration constant	[s]
K_P	Proportional gain	[-]
K_v	Velocity feedforward gain	[-]
l	Cylinder length	[m]
O	Oscillation level	[-]
p	System pressure	[Pa]
p_0	Nominal pressure drop	[Pa]
p_{comp}	Compensated pressure	[Pa]

p_{cr}	Cracking pressure	[Pa]
p_{load}	Load induced pressure	[Pa]
p_{LS}	Load sensing pressure	[Pa]
Q	Flow	[m^3/s]
s	Piston position	[m]
t	Cycle time	[s]
t_{ramp}	Ramping time	[s]
u_{FB}	Feedback control signal	[-]
u_{FF}	Feedforward control signal	[-]
u_{spool}	Normalized spool position	[-]
u_{ref}	Control signal	[-]
v	Piston velocity	[m/s]
v_0	Transition velocity	[m/s]
V	Volume	[m^3]
β	Bulk modulus	[Pa]
ζ	Damping ratio	[-]
η_{hmc}	Hydro-mechanical efficiency	[-]
ξ	Relative valve opening	[-]
ρ	Fluid density	[kg/m^3]
φ	Cylinder area ratio	[-]
ψ	Pilot area ratio	[-]
ω_{sys}	System natural frequency	[s^{-1}]
ω_v	Valve bandwidth	[s^{-1}]

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