Linear Motion Control with a Low Power Hydraulic Switching Converter - Part I: Concept, Test Rig, Simulations*

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Abstract

Switching control can also be employed for hydraulic motion control. Among the many hydraulic switching methods investigated so far, the so called hydraulic buck converter convinces by its simplicity. It consists of two switching valves, an inertance pipe, and a hydraulic accumulator to flatten the pulsation resulting from switching. This accumulator entails significant softness and nonlinearity which, in combination with sealing friction of the hydraulic cylinder, may lead to inferior performance with simple control concepts. In this paper a sub kilowatts hydraulic buck converter, its components, design, dimensioning, and steady state performance characteristics are presented first. Measurements on a testrig for linear motion employing a simple P-controller show the necessity of a model based control to achieve the desired control performance. Therefore, a dynamic model of the converter and the linear drive is derived and studied by simulations. It is shown, that the model incorporates the relevant physical effects and that it is qualified for a flatness based control, which is derived in Part II of the publication.

Keywords: hydraulics, buck converter, switching control, energy efficiency

1 Introduction

1.1 Switching Control

Switching control is an established technique in modern power electronics. With the upcoming of powerful semiconductors it replaced successively older control concepts of electric drives, power supply units, or of power amplifiers for bass speakers. Switching control enables much higher efficiency than resistance control and a fast response if high switching frequencies are used. The analogy between electric and hydraulic systems suggests trying out switching control also in hydraulic systems to gain similar efficiency improvements like in electrical systems. In fact, historically hydraulic switching control was developed and used long before electrical drives were known at all. Montgolfier's hydraulic ram of 1796 (see, e.g. Wikipedia [1]) is a step up converter and was invented in a basic version already 1772 by John Whitehurst.

In modern fluid power drive technology, however, published work appears not before the late eighties of the last century. See Brown [2], Brown et al. [3], Gall and Senn [4], Scheidl et al. [5, 6], Scheidl and Riha [7] for publications till 2000 on switching control employing some inertance elements, a prerequisite for achieving efficiency improvements over resistance control, if hydraulic power is supplied at constant pressure. Nowadays, hydraulic switching control is an established area of fluid power research and is considered as a subset of digital fluid power (Linjama [8]).

1.2 The Hydraulic Buck Converter

The simplest energy saving hydraulic switching control concept is the so called hydraulic buck converter (HBC), which is basically analogue to the electric buck converter (EBC). The circuit diagrams of both converter types are shown in Fig. 1. EBCs operate at constant switching frequencies in pulse-width mode and they are typically used in small switch-mode power supply units. The output voltage is the controlled state and energy only flows to the consumer; there is no recuperation. The HBC

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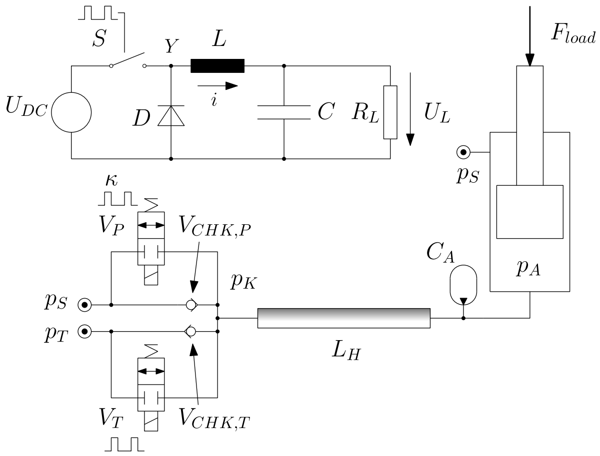


Fig. 1: Electric and hydraulic buck converter schematics

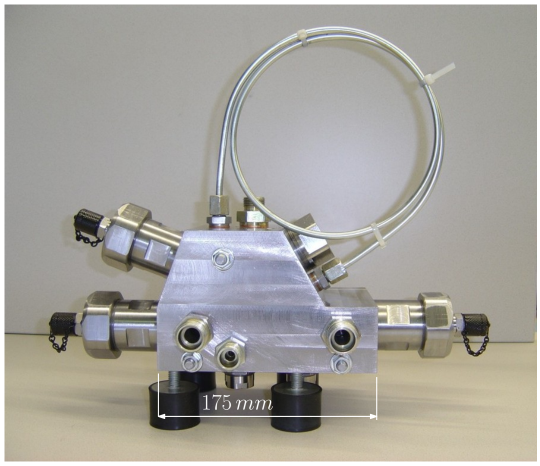


Fig. 2: The prototype *HBC030*

is serving motion control purposes, as indicated in Fig. 1 by the hydraulic cylinder and the load force F_{load}. In load lifting mode, the converter feeds hydraulic power from the supply lines to the cylinder, in load lowering mode energy is transferred back to the constant pressure supply system. In lifting – or forward mode – the HBC corresponds fully to the EBC, in lowering – or backward mode – the HBC is a step up converter and corresponds widely to the hydraulic ram. To operate in forward mode, the pressure sided switching valve (V_P) is actuated for a certain time. Fluid in the inertance pipe (hydraulic inductance L_H) is accelerated. When V_P is switched off the fluid momentum in the pipe makes the flow going on a while. The pressure at the pipe entrance falls below the tank pressure p_T and fluid is drawn from the tank line via the check valve $V_{CHK,T}$ to the system. This portion of energetically "cheap" fluid gives higher efficiency than resistance control, which just throttles the pressure difference $p_S - p_A$ causing a corresponding energetic loss. For load lowering V_T is switched, allowing fluid to flow to the tank line. When the valve is closed rapidly the momentum of the fluid makes it flowing to the pressure line via $V_{CHK,P}$; energy is recuperated. In hydraulics this switching process takes place at frequencies in the order of $100\,Hz$. The accumulator (C_A) flattens flow and pressure pulsation. Brown et al. [3] presents and studies the HBC theoretically, also in a somewhat modified version, namely with two inductance pipes which allows a symmetric actuation of an equal stroke cylinder. Scheidl et al. [5] is a theoretical and experimental study of a modified version of the HBC, namely without check valves and a pulsation flattening accumulator C_A. Scheidl et al. [6] deals with a switching converter employing a special arrangement of several inductance pipes of different lengths which in conjunction with a properly tuned switching frequency, filters pulsation at the output port without an accumulator. Johnston [9], Wang et al. [10], Pan et al. [11] are theoretical and experimental studies on switched inertance systems addressing various aspects, for instance, wave effects or pulsation suppression by a proper phase shift of two consecutive pulses. The work of the authors on the HBC is presented in Scheidl et al. [12] concerning a first prototypal realisation, mathematical modelling, and experimental results of the steady state performance characteristics. Kogler and Scheidl [13] is a theoretical study of a multiple HBC system driving one cylinder drive. If these HBCs are operated in a phase shifted mode, the accumulator (C_A) can be most likely avoided or at least drastically diminished. The control is eased without the accumulator, since this device makes the system soft and has a nonlinear characteristics.

This paper deals with the control of a sub-kilowatt HBC. Its development was motivated by a cooperation with the Italian Institute of Technology on the development of energy efficient control of hydraulically actuated legs of mobile robots. The design, use, and control of a corresponding HBC have been studied in papers Kogler et al. [14], Guglielmino et al. [15], Peng et al. [16]. Furthermore, small hydraulic systems and components typically have relatively bad efficiencies due to the scaling of loss driving effects. Thus, if a small converter can have good efficiencies, then larger converters can be expected to be even better. This HBC prototype represents the third design generation since the very first concept tests. It has the internal type designation *HBC030* and is depicted in Fig. 2. All details of its design can be found in Ehrentraut [17], some in Guglielmino et al. [15].

The paper is organized as follows: Chapter 2 de-

Fig. 3: Check valves for energy saving and recuperation

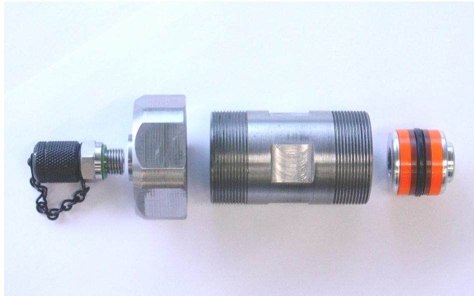


Fig. 4: Piston attenuator

scribes the *HBC030* with its main design data and experimental results of its performance characteristics obtained from experiments with a resistive hydraulic load. In Chapter 3 the testrig for linear drive experiments is explained and first measurements are presented. Chapter 4 deals with the modelling of the HBC for the simulation of the physical dynamic effects. Conclusions and an outlook to Part II of the publication are given in Chapter 5.

2 Design and Steady State Performance Characteristics

2.1 Design of the HBC030

The *HBC030* prototype's main design parameters are listed in Tab. 1. The most critical functional elements turned out to be the two check valves ($V_{CHK,P}$ and $V_{CHK,T}$ of Fig. 1), which are identical plate valves. Photos of the components are shown in Fig. 3. In particular, the wave spring had limited lifetime far below the vendor specifications. Fast and reliable check valves for switching applications are an ongoing research subject. Some results for a smaller check valve of similar principle design are given in Leati et al. [19]. The check valve seems to have insufficient dynamics resulting from oil stiction problems of the valve plate. Moreover, the valves must switch very fast in order to save energy; at lower dynamics even additional energy is lost in some operating points compared to resistance control. The inductance pipe is formed to a coil with two windings for compactness reasons. Curved pipes exhibit two vortexes in the cross section due to centrifugal forces which cause additional losses and reduce the converter efficiency. Yamamoto et al. [20] analyse this phenomenon and Kogler [21] investigates its influence on the inductance pipe's performance. The switching valves are prototypal valves - type FSVi - of the Linz Center of Mechatronics. Some aspects of their development and experimental performance results are reported in Plöckinger et al. [18]. The components are integrated in one block. Care was taken to get short internal flow channels and, in particular, to keep the channel volume between the metering edges of all valves and the beginning of the inductance pipe small. This volume in combination with fluid compressibility forms a hydraulic capacitance, which needs to be loaded every time when the system pressure valve V_P is switched on. Loading a linear capacitance from a constant pressure source causes a loss which equals the compression energy of that capacitance. At high switching frequencies this loss might be substantial. Simulation studies on this loss can be found in Kogler and Scheidl [22].

The considered prototype uses 3 gas loaded accumulators. Besides the already mentioned pressure attenuator C_A at the output of the converter, also 2 more accumulators must be spent in order to decouple the system from the supply line and tank line, respectively. All of them are realised as piston accumulators, which are depicted in Fig. 4. The main advantage of this component is its robustness against a high operating- to pre-pressure ratio and a low in- and outflow impedance. On the other hand the gas spring is not hermetically sealed due to the slide sealings in the piston. However, the gas tightness was satisfying at least during all test phases so far. Also when the test stand was out of operation for several weeks, the gas pre-pressure did not fall off noticeably.

2.2 Steady State Performance Characteristics of the HBC030

The prototype was extensively tested with respect to its steady state performance. For this purpose a purely resistive load realised by a remotely adjustable throttle valve was used in order to replace the hydraulic cylinder of Fig. 1. Flow rates and pressures were measured at different settings of pulse width and throttle valve. The main results of the basic measurements are illustrated in Figures 5 and 6. A maximum efficiency improvement of more than 30% over a hydraulic proportional drive in the forward flow direction was found. In Fig. 6 the measured recuperation efficiency characteristics is illus-

Parameter	Value	Remark
supply pressure	$p_S = 100\,bar$	supply power $P_S = 1.5\,kW$
tank pressure	$p_T = 8\,bar$	pressure relief valve
oil viscosity	$\nu \approx 15\,cSt$	Shell T 15
accumulators	$V_A = 0.04\,l$	piston type, Fig. (4)
maximum load flow rate	$\overline{q}_{A_{max}} = 5^{l}/_{min}$	
inductance pipe length	$l_p = 1.15\,m$	coil with 2 windings
inner diameter of pipe	$d_p = 3\,mm$	
switching valves	$Q_{N_{SV}} = 10^{l}/_{min}@5bar$	Plöckinger et al. [18]
check valves	$Q_{N_{CV}} = 20^{l}/_{min}@5bar$	Fig. (3)
switching frequency	$f_S = 100\,Hz$	

Tab. 1: Functional specifications

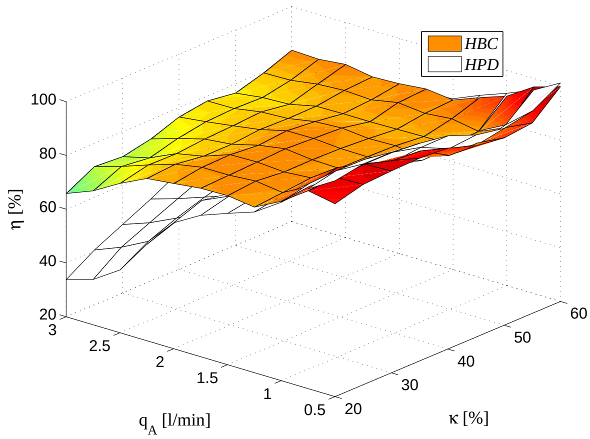


Fig. 5: Efficiency measurements

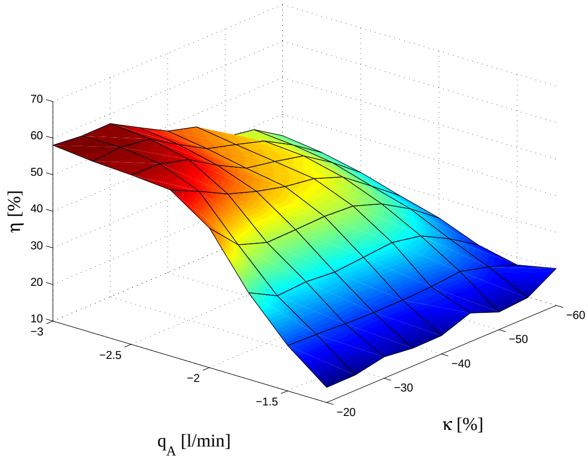


Fig. 6: Efficiency in recuperation mode

trated. A certain loss of efficiency at the converter characteristics at low flow rates is depicted in Fig. 5. There, the efficiency of the HBC is even below the performance of resistance control. As suspected in the previous subsection, this may be attributed to dynamic response limitations of the check valves, which cause a certain cross flow from the pressure side to the tank.

3 Test Rig

In the following first experiments on the closed loop performance of the HBC are compared with a conventional proportional drive (HPD). In Fig. 7 the test rig for the intended measurements is depicted. The corresponding hydraulic scheme of the test configuration is illustrated in Fig. 8. The intention is to control the position of the dead load either via the proportional valve (HPD) or with the HBC. The tank pressure is pre-loaded to a pressure of $8\,bar$ by the use of a pressure relief valve. The necessary amount of fluid, which is drawn from the tank in the extending movement of the cylinder is stored in the accumulator A_T. In fact, the tank pressure will be lowered when the HBC draws fluid from the tank, but in the retracting direction of the cylinder, respectively, in recuperation mode the tank will be refilled. The smaller accumulators D_S and D_T are installed to dynamically decouple the converter from the surrounding pipe system. The accumulator V_A is designed for pressure attenuation at the output of the converter. In order to compare the energy consumption of both, the HBC and the HPD, the hydraulic supply power is measured by the supply pressure p_S and the supply sided flow rate Q_S. This corresponding hydraulic power consumption is integrated with respect to time in order to calculate the total energy consumption. For a correct measurement of the individual energy consumptions the corresponding ball valves $(B_1, \ldots, B_5)$ must be switched properly.

In Fig. 9 the closed loop performance of the HBC with an empirically adjusted P-controller is opposed to the HPD. Figures 9a and 9b show an extending and a retracting movement of the piston at a

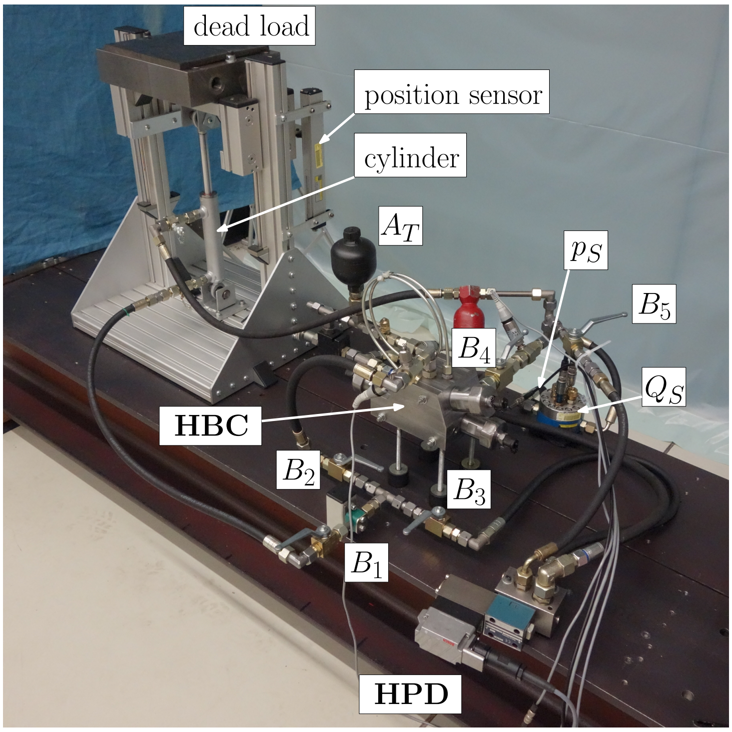


Fig. 7: Test stand

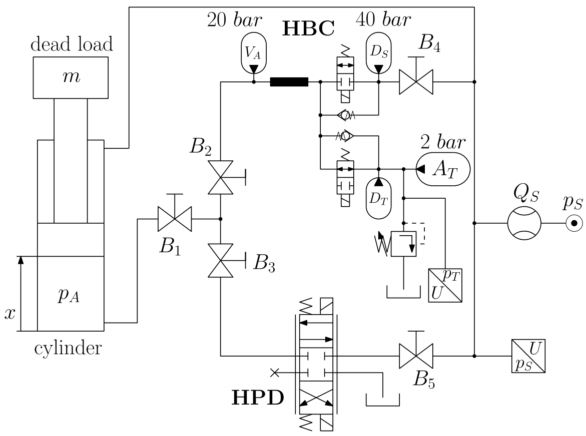
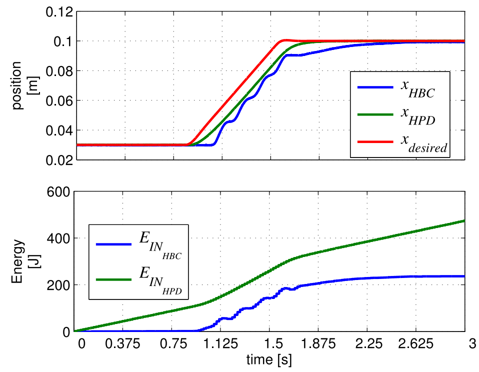
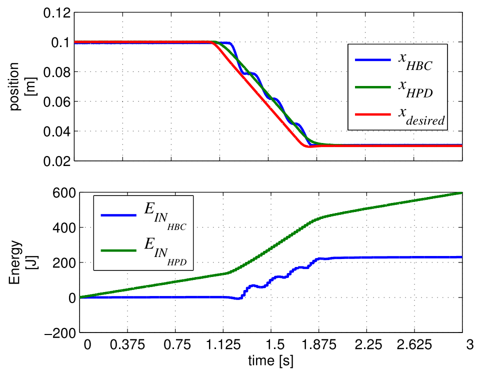


Fig. 8: Hydraulic scheme



(a) Extending piston movement



(b) Retracting piston movement

Fig. 9: Measurements of the HBC with P-controller compared with HPD

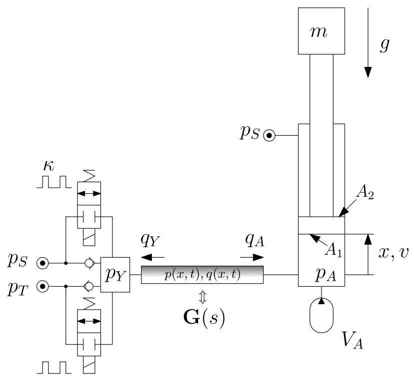


Fig. 10: Advanced model for simulation

constant velocity of $70\frac{mm}{s}$. The HBC configuration shows poor trajectory tracking and unwanted oscillations due to the softness of the gas spring in the attenuator at the output of the converter. But, the energy consumption is already much better than the energy consumption of the HPD. Since the used proportional valve has nearly no valve overlap - which is in turn important for good control performance - the leakage over the valve worsens the energy consumption of the HPD even when the piston is not moving. The valves of the HBC are just on/off valves with no significant leakage in the closed position, which prevents any energy consumption in rest positions. However, the illustrated results show, that for a satisfying control performance a qualified control strategy is necessary. This could be, for instance, a model-free control according to [23]. But since the major physical effects were identified from the measurements a dynamic model for control can be derived easily, which is shown in the following section.

4 Modelling

4.1 Simulation Model of the HBC

The dynamics of the HBC is modelled in time domain according to Fig. 10. In contrast to the test rig according to Fig. 7 the pressure attenuator at the output of the converter is directly attached to the piston sided cylinder chamber. This is valid, because the fluid inertia in the connecting hoses L_p in combination with the cross-section area of the piston A_1 can be scaled to a transformed dead load $\widetilde{m} = m + L_p A_1^2$. The considered physical effects of the converter are the square-root characteristics of all valves, linear wave propagation in the inductance pipe, and a parasitic capacitance of a certain dead volume between the valve connections and the entrance of the inductance pipe. The two switching valves are controlled by the actual duty ratio κ at a constant switching frequency of $100\,Hz$. The duty ratio remains constant for one switching period T_P and is updated after each period to the actual set value of the controller. The switching dynamics of the valves is modelled by a ramp having the same effective switching time as the real valves. Furthermore, a valve overlap of 50% of the full spool stroke is considered. The dynamics of the check valves is neglected, because the used plate valves are supposed to switch in some tenth of a millisecond and, thus, much faster than all other dynamics in the system. The effect of wave propagation in the converter inductance is modelled by a linear method of characteristics (MOC).

Using the momentum balance of the mechanical part of the system and applying a polytropic change of state model for the pressure build up in the cylinder including the accumulator results to the following nonlinear autonomous dynamic system

$$\begin{bmatrix}\dot{x}\\ \dot{v}\\ \dot{p}_A\end{bmatrix} = \begin{bmatrix} v \\ \frac{1}{m}\left(p_A A_1 - p_S A_2 - mg - F_F\right) \\ \frac{E_{oil}}{V_A\left(1+\left(\frac{E_{oil}}{p_A \varkappa}-1\right)\left(\frac{p_{0_G}}{p_A}\right)^{\frac{1}{\varkappa}}\right)}\left(-A_1 v + q_A\right)\end{bmatrix}, \tag{1}$$

with the state vector

$$\mathbf{x} = \begin{bmatrix} x \\ v \\ p_A \end{bmatrix} \tag{2}$$

and the control input q_A. The indicator E_{oil} represents the compressibility modulus of the hydraulic oil, p_{0_G} stands for the gas pre-pressure of the gas spring in the accumulator and $\varkappa$ is the polytropic exponent. Furthermore, the friction force F_F is accounted by a static friction model (see e.g. Armstrong-Hélouvry [24]), which reads

$$F_F = d_v v + \mathrm{sign}(v)\left(d_c + \left(r_H - d_c\right) e^{-\left(\frac{v}{v_0}\right)^2}\right), \tag{3}$$

where d_v and d_c denote the viscous and the Coulomb friction, respectively, and r_H the stiction coefficient. The parameter v_0 represents the reference velocity of the Stribeck-effect. Additionally, the stick-slip effect of the piston movement is considered, which is represented by a change of the mechanical degree of freedom in the stiction case. The main simulation parameters of the HBC are listed in Tab. 1. The simulation parameters of the differential cylinder, the dead load, and the friction model are summed up in Tab. 2. All simulations were carried out in *Matlab/Simulink*$^{\mathrm{TM}}$.

Parameter	Value
dead load	$m = 20\,kg$
diameter of the piston	$d_P = 32\,mm$
diameter of the rod	$d_R = 20\,mm$
viscous friction	$d_v = 200\,\frac{Ns}{m}$
coulomb friction	$d_c = 100\,N$
stiction force	$F_H = 400\,N$
gravity	$g = 9.81\,\frac{m}{s^2}$

Tab. 2: System parameters for simulations

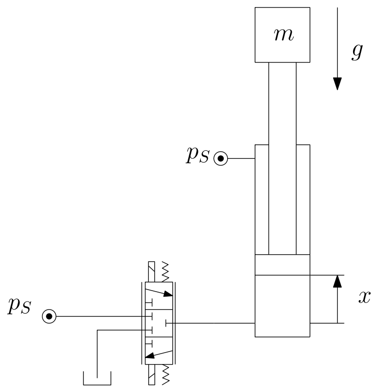


Fig. 11: Hydraulic proportional drive

4.2 Hydraulic Proportional Drive Model for Comparison

The considered HBC configuration acts basically as a 3/3-proportional valve, but at higher efficiency. For this reason the performance of the HBC will be compared to a hydraulic proportional drive (HPD) according to Fig. 11. The dynamics of the cylinder is identical to the simulation model described in the previous Subsection 4.1. Also the same friction model of the piston is taken in to account. Furthermore, the dynamics of the proportional valve is considered by a *PT-2* behaviour with a cut-off frequency of about $f_c \approx 30\,Hz$. This dynamic valve response is common for lower-cost proportional valves, but of course not for modern advanced servo valves. The annulus chamber of the cylinder is also connected to supply pressure permanently. Since only the pressure in the piston sided chamber is proportionally controlled, i.e., by metering of the valve edge, the size of the valve does not play any role in the simulations, at least if no saturation effects occur, what is assumed in this analysis. In all simulations and, furthermore, in all experiments the closed loop system of the HPD is realised with a simple P-controller.

4.3 Control Simulations

The proportional controller according to Fig. 12 is the simplest closed loop controller for the investigated system. It does not use any information of the dynamics of the plant. The design of the controller is carried out empirically. The controller gain was increased until the stability border of the closed loop was crossed, i.e., when the system started to oscillate. In Fig. 13 the simulation result of the tracking performance of the P-controller at a $0.33\,Hz$ sinusoidal trajectory is illustrated. The poor performance of this configuration can be examined especially from the velocity signal in the lower diagram of Fig. 13. The considerable oscillations in the piston velocity occur due to the softness of the gas spring mass oscillator, which in combination with sealing friction leads to stick slip effects. If the controller gain is lowered to avoid the oscillations, the tracking performance is worsened.

Compared to the measurements according to Fig. 7, the natural frequency of the hydromechanical oscillator can be identified by adjusting the parameters of the dead load and the polytropic gas spring in the accumulator.

5 Conclusion and Outlook

Compared to electric switching control, today only relatively low switching frequencies are feasible in hydraulics, mainly because of the limitations of valves response dynamics. In turn, accumulators for pulsation attenuation must have quite high hydraulic capacity which gives the system high softness and significant nonlinear behaviour. Consequently, the natural frequency of such a system is very low. Simple proportional control of such a converter may lead to bad performance, for instance, if the desired response dynamics are in the range of the lowest natural frequency of the system. However, since a proper dynamic model of such a hydraulic switching application was found, a model based control can be employed, which compensates the softness of the system. From the viewpoint of control theory it is noticeable, that the mathematical system of the cylinder drive presented in this paper is a so called "flat" system, which suggests to use a flatness based approach for motion control. The detailed formulation of the flatness based controller, corresponding simulation results and the related experiments are presented in Part II of the publication.

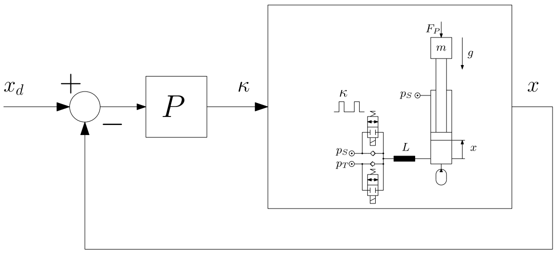


Fig. 12: Control scheme with P-controller

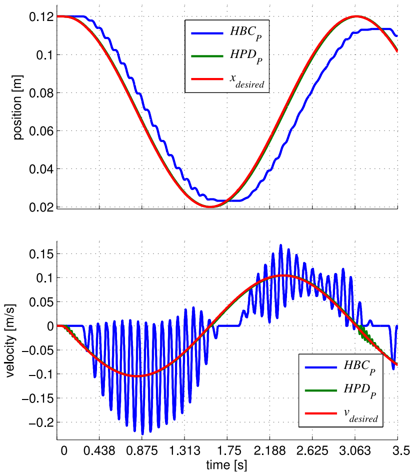


Fig. 13: Simulation of sinusoidal trajectory with P-controller

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Nomenclature

A_1 cross-section area of piston .... $[m^2]$

A_2 cross-section area of annulus chamber ... $[m^2]$

d_c coulomb friction coefficient .... $[N]$

d_v viscous friction .... $[\frac{Ns}{m}]$

E_{oil} compressibility modulus .... $[Pa]$

f_c cut off frequency .... $[s^{-1}]$

F_F friction force .... $[N]$

g gravity .... $[m/s^2]$

m dead load .... $[kg]$

p_{0_G} gas pre-pressure .... $[Pa]$

p_A load pressure .... $[Pa]$

p_S supply pressure .... $[Pa]$

p_T tank pressure .... $[Pa]$

q_A flow rate at output of the converter .... $[m^3/s]$

r_H stiction coefficient .... $[N]$

v velocity .... $[m/s]$

v_0 reference velocity of Stribeck effect .... $[\frac{m}{s}]$

V_A accumulator volume .... $[m^3]$

x position .... $[m]$

$\varkappa$ polytropic exponent .... $[1]$