Model Based Motion Control for Hydraulic Linear Drives

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1 Introduction

A cylinder drive is an ideal solution to build up an actuator for linear motion with a superior force level. Cylinders have been applied for many years in a lot of technical fields, and also the servo cylinder drive, as the closed loop controlled version, is old established technology. On the other hand, a cylinder drive features some non-linearities and other displeasing characteristics that influence more or less the transfer behaviour. In the last years, model based control concepts have been developed and successfully applied. These concepts considerably enhance the control behaviour. In this paper a methodology is presented how to build up conventional and model based motion control concepts for position and force/ pressure controls using modern computer algebra tools.

2 Conventional vs. model based control concepts

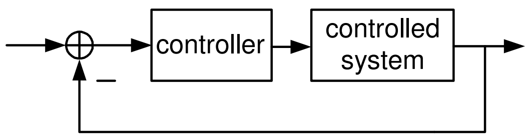


Figure 1: Single loop control

The main soft spots of single looped hydraulic motion controls (**Figure 1**) are two points:

1.) The controller design is based on a linear model, the transfer function G(s). The parameters of that linear model depend more or less on the operating point. Therefore a basic operating point has to be defined, and this point should represent the worst motion case. If the difference of the parameter values between the actual operating point and the basic point becomes significant, the performance of the motion

control will become worse, too.
2.) The actuating signal provoking the desired drive reaction is always based on a control error. But the term ”error“ indicates that the system is at least temporarily not at the very operating point where it should be due to the actuating signal.

As countermeasure to the error evoked by the first point mentioned above the control parameters should be continuously adapted to the model parameter changes due to the actual operation point (“adaptive controls”). In general, this is a quite elaborate solution, but beside this the realisation of adaptive controls for hydraulic cylinder drives often fails because no straight description of the dependence of the model parameter on the operating point is available (q.v. chap. 3 below).

As a measure against the second point you can try to adjust the controller loop gain as high as possible. Thus the needed actuating signal is already generated by small error signals. But the increase of loop gain is quite limited for stability reasons. Hence, the deployment of so-called high level control structures is often proposed. These concepts are motivated by the state space control theory that says that in theory an infinite increase of the system loop gain is possible. As fascinating the prospects of these concepts may appear, the hassle begins when you will implement these solutions (i.e. incomplete state feedback, signal limitations, problems in generating feasible feedback signals via additional sensors, observers or differentiating algorithms, effects due to the dependence on the operation point etc.). These problems are often responsible for the loss of a good portion of the aimed results /5/. Furthermore, there is still no practicable theorem that says from what degree on the theoretically infinite increase of loop gain causes instability of the state controlled system under real conditions. In general, you will detect this limitation just at the built up drive. As long as the difference between the theoretically expected results and the real performance can be judged that badly in an early project design stage, these control concepts will not have the importance and the spread that they might have.

A better and apparently direct way to improve the control performance is to consider the drive non-linearities themselves in the phase of development within the control

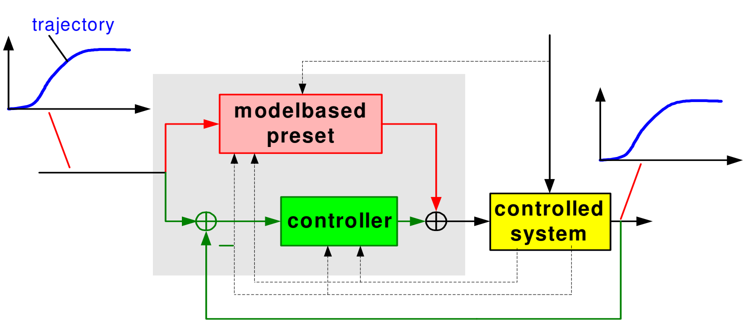


Figure 2: Model based predictive control

project. Some promising conceptions of that kind have been proposed and successfully applied during the last years. One possible approach is the method of adding special functions to the control algorithm, that determine correcting offset signals to compensate more or less the effects of the main nonlinearities /6/. In this paper a model based predictive control is presented as another possible approach aiming in that direction. The basic idea is to determine as much as possible of the actuating signal needed to run the working cycle at any moment in real-time by a model based precontrol circuit (**Figure 2**). If needed, measured process signals can additionally be processed in that model-based circuit. A subordinated closed loop controller is always implemented to provide a possible additional fraction of the actuating signal. The amount of the additional part is that much smaller as the primary part calculated by the precontrol circuit provokes the desired drive motion. In case that the precontrol circuit works fully correct, the control error detected by the closed loop circuit would all the time be equal to zero: that means the controller is inactive. The big advantage of that approach is that open loop control circuits cannot become instable. A need for high controller loop gains is no longer given, because it is not the controller's task any more to generate the entire actuating signal on the base of the control error. In addition, it is no longer the function of integrating controller parts to direct the system (in general quite slowly due to stability reasons) to the desired operating point, because that is done in a model based way by the precontrol circuit. This cooperation of an open loop and a closed loop control circuit creates quite a robust process control structure.

It is important to mention that this control conception presented here stands in a direct relation to the theory of controlling dynamically flat systems by the "two degree of freedom model" /10/. The presented conception can be regarded as a pragmatic straightforward realisation of that fundamental theory, here applied just and only at the given case of valve directed hydraulic cylinder drives. The aim of this approach was to detour the high-levelled mathematical theory. The results obtained by this approach are identical /7/.

As just explained these model based control conceptions do always contain a subordinated conventional closed loop control. This controller does not need very high gains for the mentioned reasons. Hence conventional single loop structures are sufficient. Unfortunately, a straightforward design of such conventional control loops is still a problem for many users, even today. Taking a look in the literature, you will usually find a patchwork of application specific rules and recommendations. Therefore, in the next chapters two simple and universal ways are presented to design and parameterize single loop controls for valve directed hydraulic cylinder drives in a quite systematic und direct way. One solution concerns position controls, and the other concerns pressure or force controls.

3 Problem of drive modelling

The base of any systematic controller design is the existence of a suitable mathematical description of the given case. **Figure 3** shows in 3D view the stationary

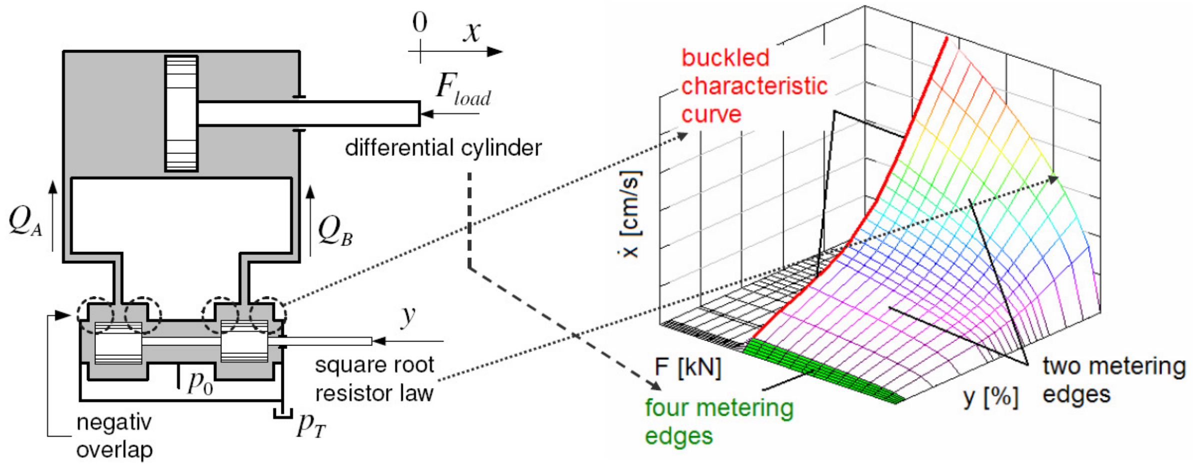


Figure 3: Characteristic curve diagram of the servo drive (differential cylinder with servo valve incl. buckled characteristic curve)

characteristic curve family of an entire valve directed hydraulic cylinder drive (**I**. and **II**. quadrant plotted).

The significant stationary non- linearities of the characteristics are obvious. Another dynamic nonlinearity that can have non-negligible effects on control performance is the dependence of the cylinder eigenfrequency on the piston position (**Figure 4**). And furthermore, cylinder drives are direct drives that means that the magnitude of the mass load attached at the piston does fully influence the drive eigenfrequency. That is a natural disadvantage of linear hydraulic axes that rotatory electric drives mostly do not have. On principle, electric drives have a better starting position in building up controls. So the model of the basic electric DC motor is in first approach linear by itself; the corresponding set of characteristics curves thus form a plane

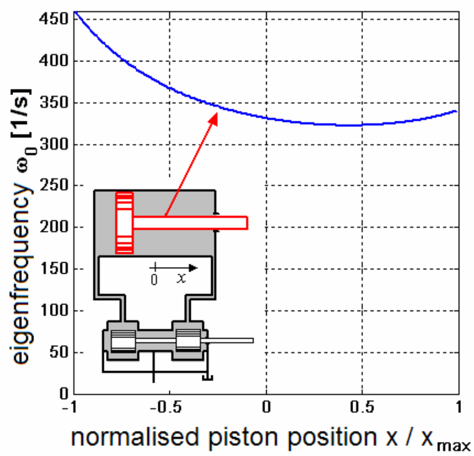


Figure 4: Dependence of the eigenfrequency on the stroke

(**Figure 5**). Additionally the load mass is attached to the electric motor mostly by a gearbox. Hence the magnitude of the load mass generally has almost no influence on the transfer behaviour.

For a systematic controller design you need most the set of characteristic curves as stationary model. The linear model can be derived if this characteristic field is available. But in practice, when doing that step of defining these two basic models, each depending on the operating point, the problems already arise. In literature you will not find a lot of helpful information. In general, all remarks about modelling are restricted to the following special case: It is a assumed to have a synchronizing cylinder (ratio of piston areas α= 1), to have ideal zero overlapped metering edges in the servo valve, to use just piston positions close to the middle of the stroke and to have friction forces that can be neglected at least approximately. In that special case you will quickly obtain quite a simple set of equations. And with a few steps the stationary model (= set of characteristic curves) and the linear dynamic model G(s) can be calculated. But the point is that the models obtained that way have the blemish, that a drive fulfilling these assumptions is quite useless for practice.

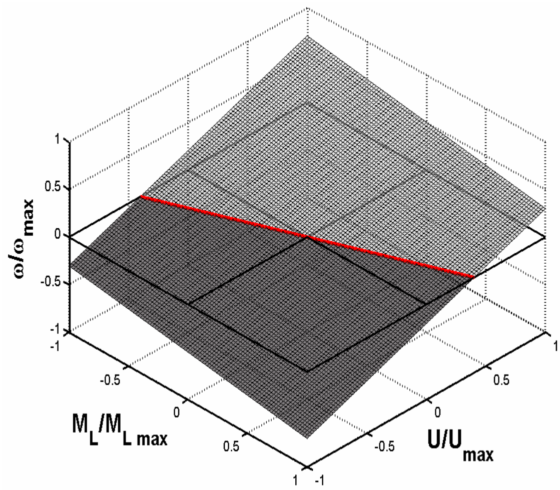


Figure 5: Characteristic diagram of the electric DC-motor

Synchronizing cylinders are unpopular due to the required space, and the piston stroke should be completely usable in control mode. But the worst point concerning these assumptions is, that only negative overlapped metering edges allow to build up a resistor control according to the pressure divider principle. A drive with a zero overlapped valve cannot balance external load forces when the piston is still standing. On the other hand, proven methods are known today to derive usable models, that also consider the dependence on the operating point. These modelling procedures are based only upon data that should anyway be known in a drive project.

Briefly said, there is no need at all to rely on any ivory tower assumptions. Only as far as friction is concerned a characteristic friction curve suitable for servo applications still has to be required /8/. For the numerical calculus of the modelling task very powerful computer algebra tools are disposable nowadays; Figure 3 and 5 and are outputs of results obtained that way.

4 Conventional control conceptions

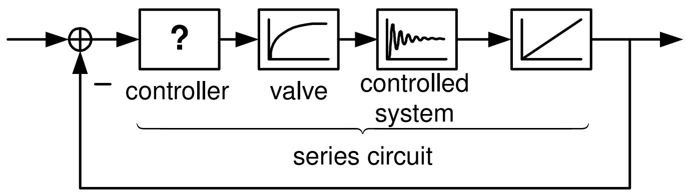


Figure 6: Single loop control circuit

In the following chapter a design method for single loop controllers is presented, one for position control and one for pressure or force control. Both methods stand out by an enormous directness and a universal usability. Unfortunately, they (still) are quite unknown and therefore up to now not often applied in the field of hydraulics.

4.1 Position control design

The open control circuit in the case of a valve driven cylinder drive consists of three elements and the attached integrator that sums up the velocity to the position signal (**Figure 6**). Let us assume that a linear drive model that regards the dependencies on the operation point has been derived as shown above. The free parameters that have to be determined in the design stage are the valve dynamic and the controller parameters. This is in the simplest case a P-controller, but if required it can get enhanced by one or more dynamic correcting elements (PPT1, PDT1 etc.).

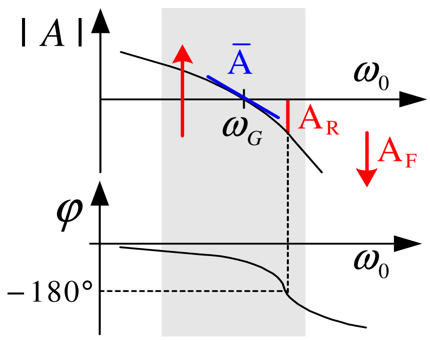


Figure 7: Requirements of the PT2-analogy

The over-all frequency response of the open control circuit is simply obtained by graphical addition of the partial frequency responses connected in series. A very straight and simple approach based upon the frequency domain controller design makes an analogy consideration comparing the open loop and the closed loop frequency response in the case of a PT2-element. When these reflections are transferred to the given case of a cylinder drive, the conclusion is that the aim of a fast step response without overshooting is reached quite well when the following features are given: The over-all frequency response of the open control circuit has a gain drop $\bar{A}$ of about 25 dB/dec at the gain crossover frequency ω_G, the gain margin A_R is about 20 dB, and for frequencies $\omega > \omega_G$ (**Figure 7**), quite a steep amplitude drop A_F (>40 dB/dec) is well advised. Finally, the more the gain crossover frequency is at higher frequency values the higher is the load stiffness and the shorter is the settling time. /2/

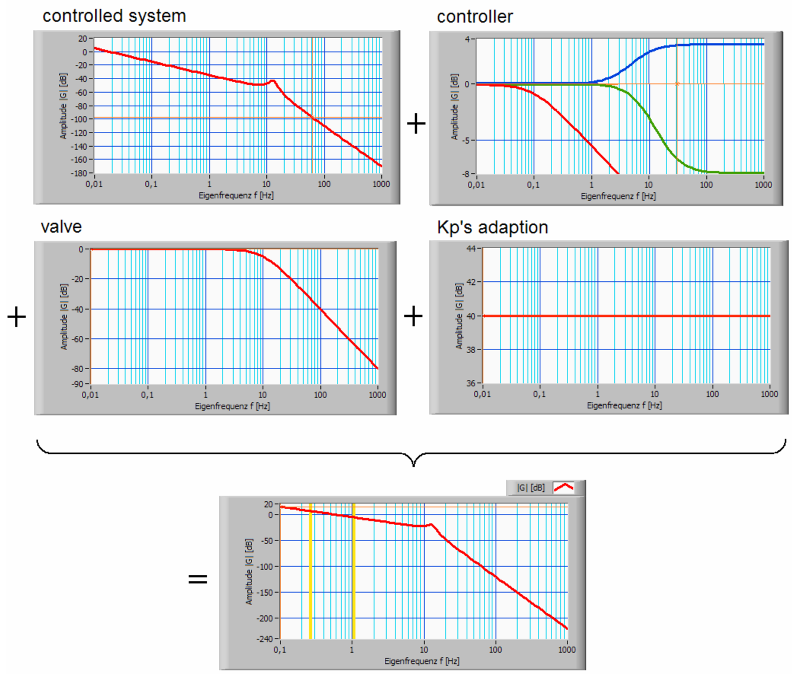


Figure 8: Bode plot for the PT2-analogy method

The entire controller design strategy is now to follow these design aims: The free design parameters in form of the dynamic controller elements, the valve dynamic and the controller gain (→ loop gain) are used in order to make the all over frequency response of the given open control circuit have as good as possible the mentioned features of the PT2 open loop frequency response (**Figure 8**). Thus a simple and universally applicable design and optimisation strategy is defined, that leads step by step very simply and straightforward to the solution. Here a reasonable compromise between effort and effect can already be found in that early stage of design. In many application cases, the performance reached by that design step is fully sufficient, so that a feedback control loop as only controller element will do and a model based control structure is not mandatory.

When no load forces are given, these control concepts produce only quite small stationary control errors due to the friction force (reference response). But for principal reasons they do have a stationary control error of a higher or lower extend if there is a significant load force (disturbance response). Because the controlled system has an integrating behaviour itself, controller concepts with any significant integrating parts are very problematic for stability reasons. Therefore when integrating parts are needed to compensate stationary errors only anti-windup integrating algorithms can be applied. The so-called switching integrator is an example for that type being approved und often applied in hydraulic servo solutions /9/.

4.2 Pressure-/ force control design

Now a corresponding method for the controller design in the case of a pressure or force control is presented, that again can be applied very universally. As far as the controlled signal is concerned, this is not the place to discuss in detail the differences between pressure and force control. It should only be reminded here that the pressure signal can be regarded to be equivalent to the force signal if the friction and the acceleration forces are quite negligible in the given case /1/. But fortunately this is quite often possible in first approach.

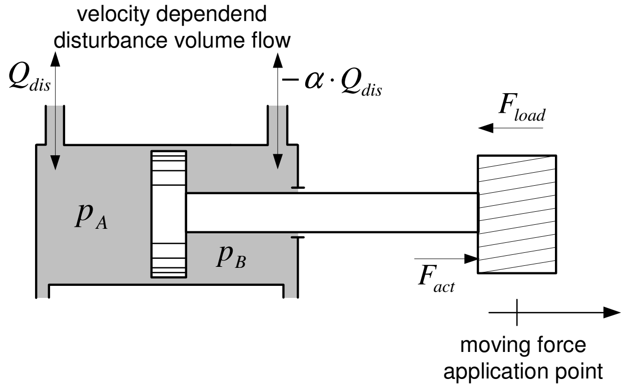


Figure 9: New view of the pressure control

In the context of building up a pressure control a slightly different view of the control task is very helpful; a view that unfortunately is not yet very popular. If the pressure is the output signal of the controlled system, all parts of the signal flow chart being located “after” the pressure signal are now part of the back feeding branch (**Figure 10**). Moreover, the relation between the piston motion and the force reaction F_{load} due to this motion is described in that feedback branch. Separate from the rare case of an ideal spring, this very relation is in general not exactly known, and furthermore an external unknown load force might act upon that feedback branch, too. Hence, it does not really make sense to point out a block indicating a determined signal relation between pressure and motion in the signal flow chart, when the only proven fact is that the occurrence of disturbances will cause disturbing flow rates at the first sum point of the flow chart. It is more consequent to lump together all these relations (Figure 10, dotted block) and to define a disturbing flow rate Q_{dis} correlated with the piston velocity to be the new

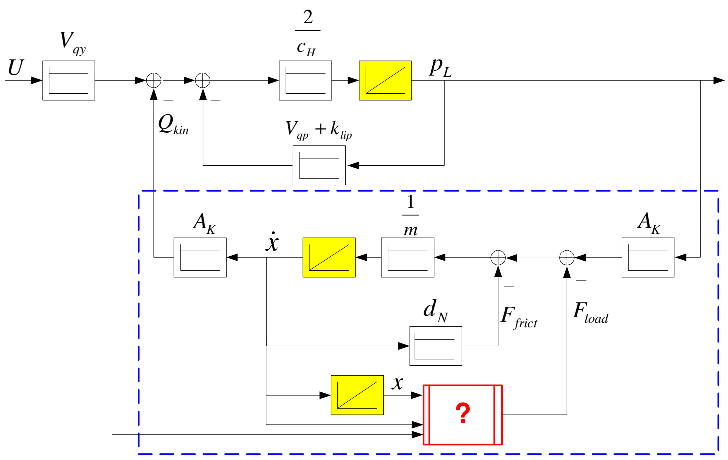


Figure 10: Signal flow chart of the system (I)

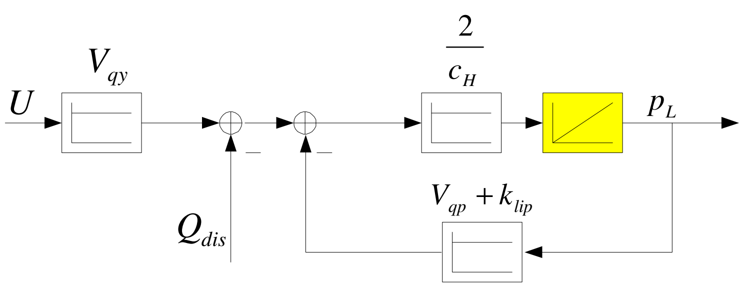


Figure 11: Signal flow chart of the system (II)

disturbance signal (**Figure 11**). The very cause for the piston motion is of no interest for the control task. With that view, the load pressure is the controlled signal, and the disturbance flow provoked by the piston motion is the new disturbance signal. There is no causal link between the magnitude of the disturbance signal and the piston motion. The reference control case now is the pressure rise (or drop) having the piston still standing. Any piston motion is the reason for a disturbance flow as “newly defined” disturbance signal (**Figure 12**).

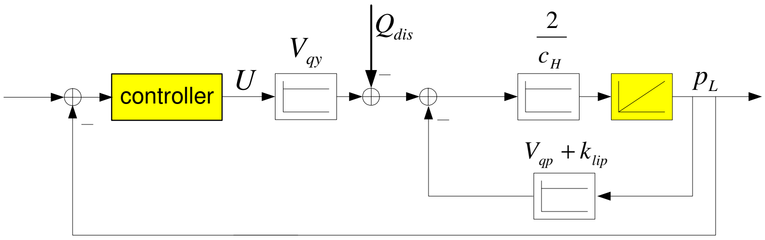


Figure 12: New view of the pressure control loop

For the feedback controller design the cylinder is therefore assumed to be still standing. The design itself rests on the idea of defining a reference response dynamic by compensating the given PT1 dynamic of the controlled system with a PI controller. The reference response should be defined in accordance to the performance data of the servo valve, because the nominal flow rate of the chosen valve is the main influence for the maximum pressure gradient of the given configuration. But when the force application point (**Figure 9**) moves, as often seen in real pressure control, huge control error signals will occur. An integrating part of the controller can compensate these errors only very slowly. It is reasonable to measure the piston velocity as the cause of the disturbance flow and to compensate its effects via a control signal determined in the precontrol circuit; an idea already published 1989 /3/. In /4/ a modified version of this idea is proposed. This version is derived directly by applying the concept of model based pressure control (s. chap 5.2) and above all generally delivers better control results (see Figure 17 and 18).

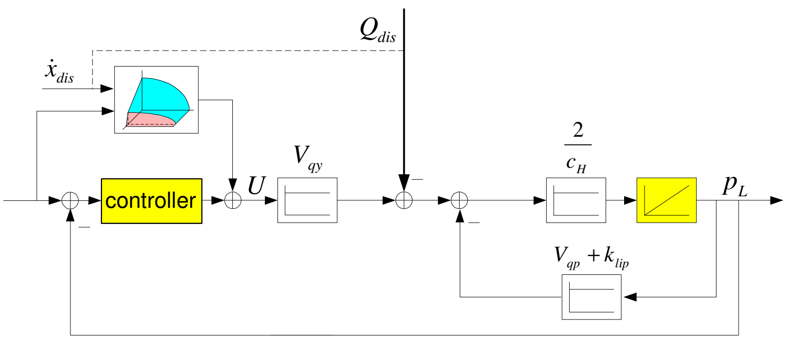


Figure 13: Over-all concept of the controlled system

5 Model based predictive control

In chapter 2 the basic idea of model based predictive control has been explained. Now a design method for the precontrol circuit is presented. This method is based on the set of mathematical model equations describing the dynamic behaviour of a valve driven hydraulic linear actuator. Disregarding the valve dynamics the drive is a

$$m \cdot \ddot{x} = A_A \cdot p_A \quad - \alpha \cdot A_A \cdot p_B \quad - F_{load} \quad - F_{frict} \qquad (1)$$

$$p_L = p_A \quad - \alpha \cdot p_B \qquad (2)$$

$$\dot{p}_L = \dot{p}_A \quad - \alpha \cdot \dot{p}_B \qquad (3)$$

$$Q_{COMP\,A} = Q_{TO\,A} \quad - Q_{OFF\,A} \quad - Q_{KIN\,A} \quad - Q_{L\,A} \qquad (4)$$

$$Q_{COMP\,B} = Q_{TO\,B} \quad - Q_{OFF\,B} \quad - Q_{KIN\,B} \quad - Q_{L\,B} \qquad (5)$$

$$Q_{COMP\,A} = \frac{V_A(x)}{E_K} \cdot \dot{p}_A \qquad (6)$$

$$Q_{COMP\,B} = \frac{V_B(x)}{E_K} \cdot \dot{p}_B \qquad (7)$$

$$Q_{KIN\,A} = A_A \cdot \dot{x} \qquad (8)$$

$$Q_{KIN\,B} = -\alpha \cdot A_A \cdot \dot{x} \qquad (9)$$

$$Q_A = Q_{TO\,A} \quad - Q_{OFF\,A} = f_A(p_A, y) \qquad (10)$$

$$Q_B = Q_{TO\,B} \quad - Q_{OFF\,B} = f_B(p_B, y) \qquad (11)$$

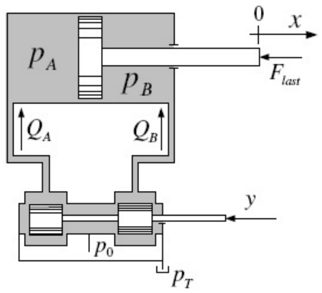
and

$$x = \int \dot{x}\, dt \qquad (12)$$

$$\dot{x} = \int \ddot{x}\, dt \qquad (13)$$

$$p_A = \int \dot{p}_A\, dt \qquad (14)$$

$$p_B = \int \dot{p}_B\, dt \qquad (15)$$



Nomenclature *(Z= compartment Z)*

y	*actuating signal*	*[%]*
x	*position*	*[cm]*
p_Z	*pressure of Z*	*[bar]*
p_L	*load pressure*	*[bar]*
F_{load}	*load force*	*[kN]*
F_{force}	*friction force*	*[kN]*
	volume flow:	
$Q_{COMP\,Z}$	*compression v. f.*	*[l/min]*
$Q_{TO\,Z}$	*v. f. to Z*	*[l/min]*
$Q_{OFF\,Z}$	*v. f. off Z*	*[l/min]*
$Q_{KIN\,Z}$	*kinematical v. f.*	*[l/min]*
$Q_{L\,Z}$	*leakage f.*	*[l/min]*
V_Z	*volume of Z*	*[cm³]*
E_K	*bulk modulus*	*[bar]*
α	*piston area ratio*	*[ ]*
A_Z	*piston area of Z*	*[cm²]*

Figure 14: Model equations of the hydraulic cylinder drive

system of third order in the case of position control, having the position as the controlled signal connected with the velocity signal by one integrator and with the acceleration signal by two integrators. In the case of a pressure or force control you have a system of second order, having the pressure / force as the controlled signal connected with the pressure gradient signal by one integrator. This implies directly that only such curves are suited as a reference signal („trajectory“) that can be generated by single respectively double integration. That is, they have to be differentiable twice in case of position control and once in case of pressure control.

The aim is now to determine the actuating signal y that provokes the required trajectory curves. This value y simultaneously has to fulfil the relevant model equations. With the assumptions $p_T = 0$, $Q_L = 0$ and $p_0 = const.$ you obtain the model equations listed up in **Figure 14** for a valve driven hydraulic cylinder drive. The very task that has now to be solved in real-time is to calculate the actual control input for each control cycle with help of the stationary equations 1-11. In case that the position is the controlled signal, the values $x, \dot{x}, \ddot{x}$ are given by the reference trajectory, and if the pressure (or the force) is the controlled signal, the values $p_L, \dot{p}_L$ are given by the reference trajectory.

6 Model based generation of the precontrol signal for position control

In case of position control the set of equations (eq. 1-11) points out the following constellation: Assuming that the friction force F_{frict} may be negligible in first approach, there are 11 equations with the following 14 unknown variables.

$$p_A, \dot{p}_A, p_B, \dot{p}_B, p_L, \dot{p}_L, F_{load,} Q_A, Q_{KIN\,A}, Q_{KA}, Q_B, Q_{KIN\,B}, Q_{KB}, y$$

In order to solve this set of equations, you therefore need three additional pieces of information about three of the unknown signals. Then the demanded actuating signal

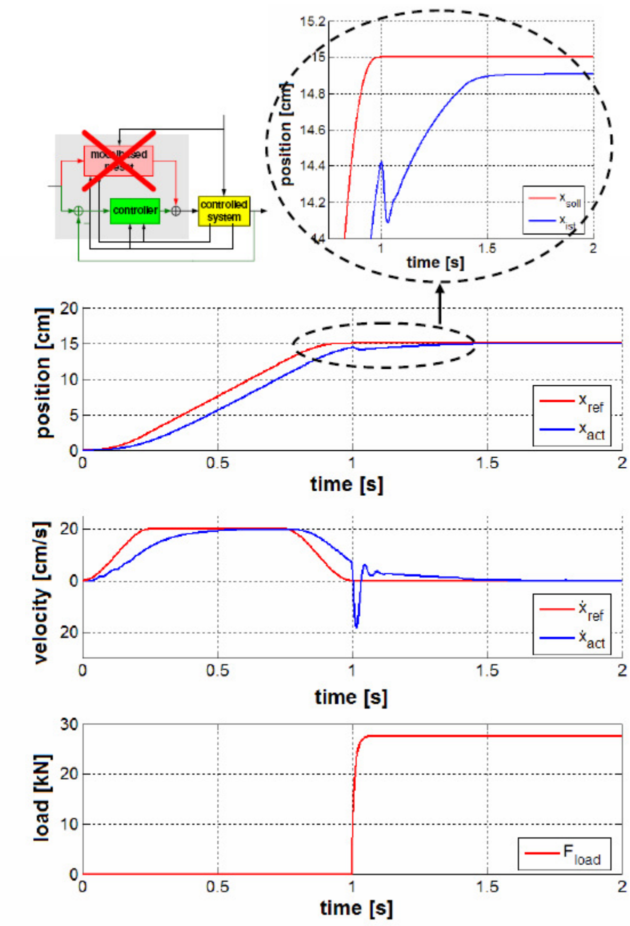


Figure 15: Conventional position control (no integrating parts)

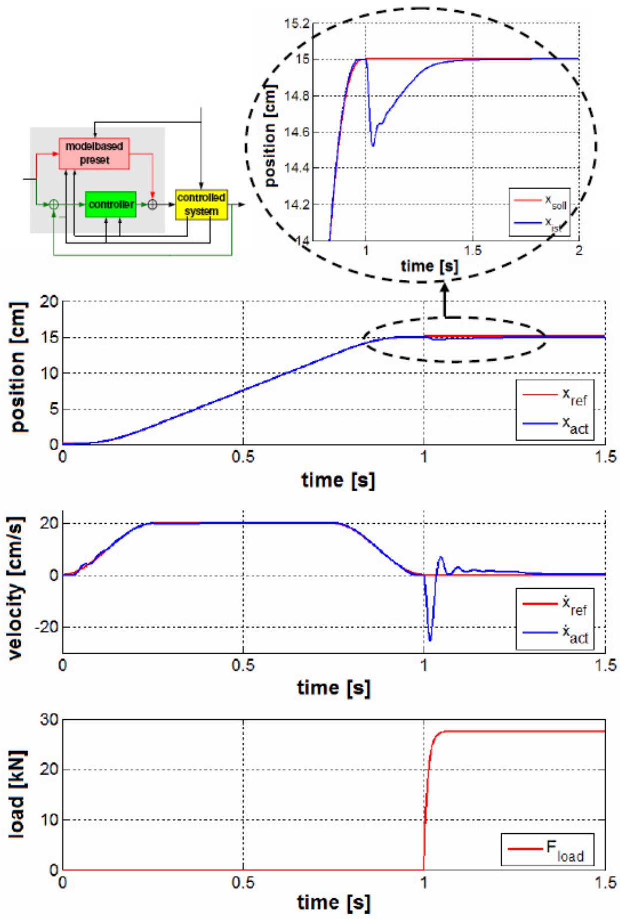


Figure 16: Model based position control

y can be calculated at any point of time by solving this set of equations. So the pre-control circuit at least is just an algorithm solving these equations. In some cases, that solution is possible analytically, but anyway it is possible numerically. The convergence of the numerical solution is generally quite fast because the calculated results of the last time-step are suitable starting values for the next step calculation as long as you can guarantee that there are no discontinuity spots in the occurring signal curves. This can be avoided by a newly developed prefilter (see chap. 5.4). The three needed additional pieces of information that make that set of equations solvable can consist of (reasonable!) assumptions as well as of information obtained by the measurement of process values.

Representative for the multitude of possible variants of additional information just two constellations shall be presented here: In some cases it is justified to neglect the compression flow rates Q_{KA} and Q_{KB} ($Q_{KA} \approx 0$, $Q_{KB} \approx 0$). If in addition one of the three signals p_A, p_B or F_{load} are measured by a suitable sensor configuration then the three needed additional pieces of information are provided. An equivalent situation is given if you measure the compartment pressures p_A and p_B and if it can be assumed that the associated pressure gradients can be derived via the measured pressure values. It is obvious that in genreal the control result will be the better the more the pieces of additional information base on measurements instead of assumptions.

Some results of that model based predictive position control are shown in **Figure 15** and 16 in comparison to the results of a pure single loop control. Model based predictive control avoids contouring errors in the reference response as well as stationary errors in the disturbance response. Due to the model-based concept, such results are obtained also in case of cylinders with small piston area ratio (i.e. $\alpha < 0.5$).

6.1 Model based generation of the preset signal for pressure-/ force control

Similar conclusions can be derived in the case of pressure or force control. Looking again at the mathematical set of equations 1-11 (Figure 14), the following statement is obvious: The reference signals $p_L, \dot{p}_L$ are defined by the trajectory that has to be once differentiable in that case. Again assuming that the friction force may be negligible in first approach you have 11 equations with the following 15 unknown values

$$p_A,\, \dot{p}_A,\, p_B,\, \dot{p}_B,\, F_{load},\, x,\, \dot{x},\, \ddot{x},\, Q_A,\, Q_{KIN\,A},\, Q_{KA},\, Q_B,\, Q_{KIN\,B},\, Q_{KB},\, y$$

For solving that set of equations, you now need four pieces of additional information about any four of the unknown signals. And again, some of the many possible constellations shall be mentioned here, for example:

1.1) the compression flow rates are supposed to be negligible ($Q_{KA} \approx 0$, $Q_{KB} \approx 0$)
1.2) the signals x und $\dot{x}$ are known by measurement **or**

2.1) $x, \dot{x}$ and $\ddot{x}$ are known by measurement
2.2) p_A or p_B are also known by measurement **or**

3.1) x and $\dot{x}$ are known by measurement
3.2) p_A and p_B are known by measurement

In the case of pressure or force control the same statements as mentioned about the position control are still valid. This concerns the aspect that it is better to obtain additional information by measuring than by assuming and that you have to find the optimum between effort and efficiency.

Examples of the results that can be obtained via the model based predictive pressure control are shown in **Figure 17** in comparison to the results obtained by just applying a conventional control circuit (**Figure 18**). The effects of piston motion on pressure control results are largely compensated.

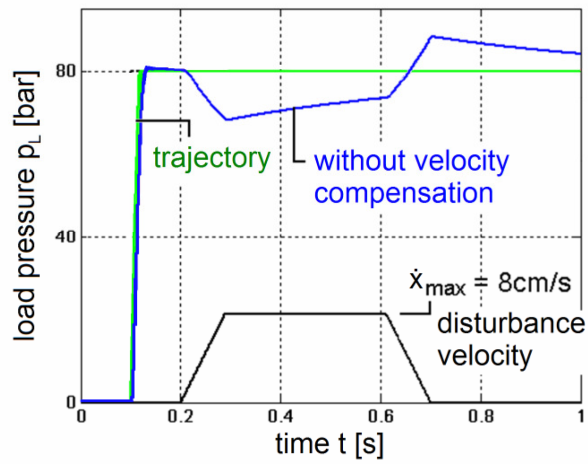


Figure17: Conventional pressure control

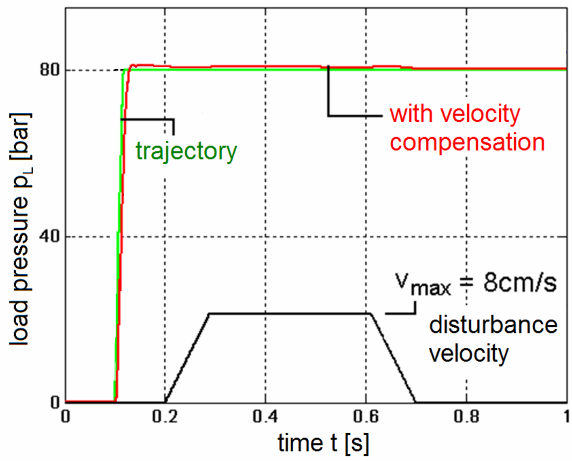


Figure 18: Model based pressure control

6.2 Measurement based online-identification of drive characteristics

The described method of model based predictive control improves the control performance considerably but increases the realisation effort. Fast sensor elements with high resolution are required to generate the needed control signals, and also the dynamic of the servo valves should be quite high. The existence of a correctly parameterised nonlinear model – the set of stationary characteristic curves – is an essential precondition. But the determination of that model will be quite impossible, if

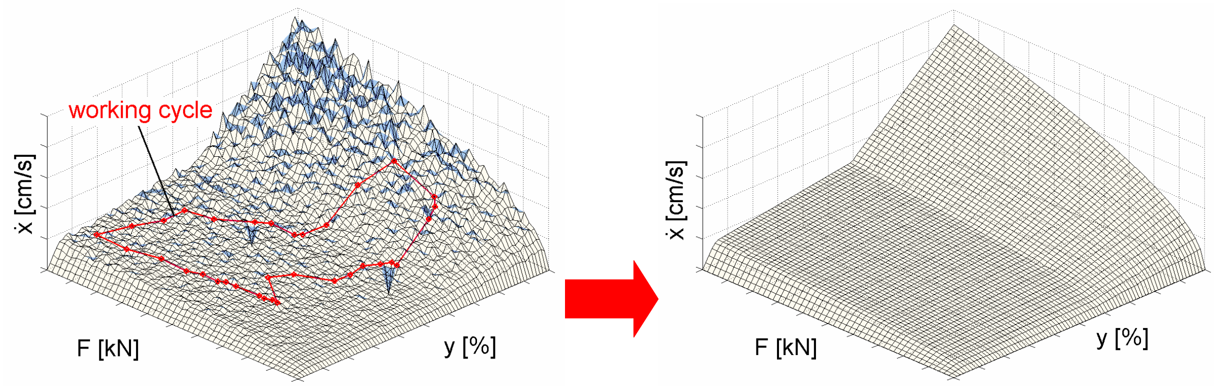


Figure 19: Principle of the online identification of the characteristic diagram

the data of the applied components– especially the data of the valve– are not known with the needed degree of accuracy. But this is quite often the case when valves of huge nominal flow rates are used. In the following a new method is presented that allows to determine the entire characteristic diagram of the drive online without any test bench.

The basic idea is quite simple: Any actual drive state -that is the knowledge of force and force gradient, of the velocity and of the actuating signal- corresponds to one operating point in the needed characteristic diagram. The force and the velocity signals are unequivocally related to the hydraulic process variables load pressure and actual flow rate. If for a given drive these signals are recorded online during a normal working cycle in the background of the controller device, a set of interpolation points of the demanded characteristic diagram can be calculated (**Figure 19**). Even a compensation of superposed dynamical effects (i.e. compression flows) is possible. The big problem now is to transform that over determined set of interpolation points into a directly applicable characteristics diagram and to detect and correct measurement errors that will cause violations of the array monotony. This cannot be achieved by methods of regression calculus or model fitting, because these procedures require the definition of function types for well-defined argument areas. Such areas cannot be defined here in any case. A new method using algorithms of image processing was developed in /7/ and /9/ that solves the given problem in a very unusual but universal way. This identification process is running online at the real machine and does not require any special measurement devices. Besides, it is therefore best suited also to detect the drifting of system parameters or the abrasion of metering edges in the background of the running process.

6.3 Prefilter

The model based predictive control as described above bases upon the idea of generating an actuator signal in a precontrol circuit that provokes the required reaction of the drive. Therefore a drive model is implemented in the precontrol circuit. This model-based preset of course cannot take into account the power limitations that are given at the real drive. Thus it can happen that the calculated reference trajectory would lead to operating points that cannot be reached by the real drive due to the magnitude of the actual disturbance signal. This would make the controller produce futile actuating signals. The trajectory generator algorithm cannot avoid this in advance, because when calculating the position trajectory the actual load force that will occur is not known and so the power reserve is unknown, too. The way out of this problem is based on the fact that the operating point is always known in model based control concepts. To solve this problem you have to determine the actual power reserve in real-time and thereby correct the reference trajectory in a way that the drive is only fed with actuating signals that don't lead it into limitations (**Figure 20**). This problem was solved by the development of a model based prefilter algorithm /4/, /7/.

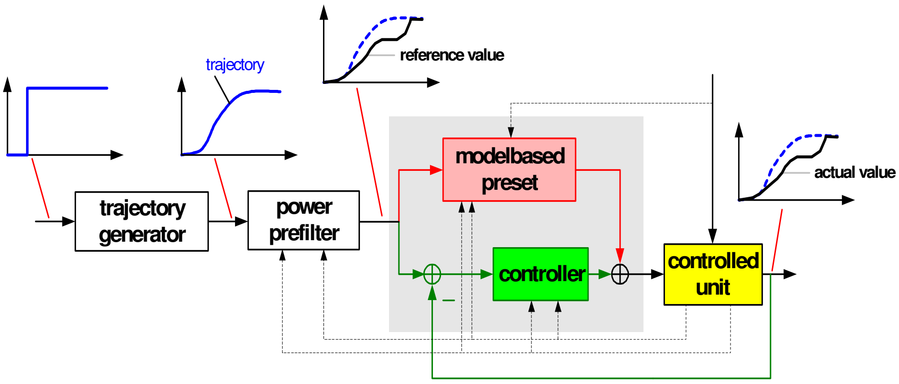


Figure 20: Model based control with prefilter

7 Conclusion and outlook correct

This paper points out a simple, transparent and straight way to design model based predictive position or pressure / force controls for valve directed cylinder drives. All design steps from the determination of the linear and nonlinear system models up to the controller parameterisation can be easily done using the available tools of modern computer algebra systems. Thus a wide scaled set of solutions for a given control project can be found depending on the very requirements. Already in the design stage of a project quite an accurate prediction of the control performance of the drive system can be made. These possibilities are unfortunately not yet consequently used in nowadays servo hydraulics. If servo hydraulic as a drive technology with some very outstanding features wants to safeguard its position in high level drive applications, there is an urgent need for more activity in that field.

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