System Analysis and Optimization of a Pressure Control for a Hydraulic-Clutch-Brake-Combination actuated with a Lineardrive

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Abstract—This paper deals with the analysis of an electro-hydraulic system and conception of a pressure control. The controlled system consists of a permanent magnet linear actuator, a hydraulic pressure line and a Clutch-Break-Combination (CBC). The linear actuator generates a force which causes the pressure in the hydraulic line. Depending on the system-pressure, the Clutch-Break-Combination brakes or clutches with a different torque. Because of the integrated prestressed springs of the CBC, the controlled system is very nonlinear. In addition there are different disturbances, which makes it hardly controllable. The linear actuator is driven with a standard inverter.

Index Terms—Control, Pressure, Clutch-Brake-Combination, Permanent Magnet Linear Drive, Nonlinear System, PMSM

I. MOTIVATION

Hydraulic actuators require a continuously available high oil pressure which is usually generated by a hydraulic pump. The rated power of the pump is considerably high, consuming a lot of energy even when no work is done. If a system additionally requires the availability of multiple possible pressure levels, the system gets very complex. In order to reduce energy consumption and complicity drastically, the standard hydraulic pump is exchanged by a permanent magnet linear synchronous motor which exerts a force to a hydraulic cylinder. This generates superior control possibilities over a standard hydraulic system. In order to control the generated system pressure, a sophisticated control algorithm is required. This control algorithm needs to be implemented into a standard inverter in order to achieve a competitive pricing of the complete system. This concept allows a great reduction of the energy consumption, better control possibilities for the complete system, as well as new methods of fault diagnosis. Paper [1] outlines the main development of the control. Within the time, more specification are required. These are eg. more than four pressure set points, a faster and optimized position feed forward and a logic which considered the wear of the CBC.

II. GENERAL DESCRIPTION OF THE CONTROLLED SYSTEM

Figure 1 shows the simplified structure of the whole system. The PMSM linear drive generates a force $F_{Mot}$ onto the hydraulic cylinder A which excites a pressure $p$ into the oil line. Over cylinder B this pressure is dissipated into a force $F_{Res}$ which is mounted onto springs. When no hydraulic pressure is applied, cylinder B has its minimum volume, while the springs have a certain prestress. When applying the maximum allowed pressure, the volume of the cylinder has its maximum.

A. Linearactuator

Linear electromagnetic machines become more and more important for industrial applications because in some cases they have numerous advantages compared to rotating counterparts. They are providing high thrust force directly to the load without the need of mechanical gears. This results in a high efficiency, a high dynamic performance and a more simple construction of the system. The force density of tubular linear machines is excellent [9]. They are the best choice for industrial applications in which high thrust forces are required. The used tubular linear drive concept consists of three main parts, a stator, which is not slotted in the conventional way and build with discrete coils mounted on poles in radial direction, a permanent magnet armature and an internal position sensor based on the hall-effect. The specialty of the motor is an economic production. Every ferromagnetic part is made out of steel without any laminations. Low priced steel components are used to manufacture the linear drive on standard production machines (e.g. turning lathe). The internal positioning sensor is also a low priced component. The prototype of the linear drive

![Fig. 1. Structure of the Controlled System](image-url)
Fig. 2. Construction of the Linear actuator produces an average force of 3090 N. It can move a maximum distance of 210 mm. To pass the force to the hydraulic system, a plunger cylinder is implemented in the inner part of the armature. Figure 2 shows a sectional drawing of the linear drive. When running at rated current and without any load, the actuator reaches a maximum speed of 2 m/s. The maximum possible speed 2.6 m/s is limited by the design parameters of the linear drive concept (EMK, Demand of voltage, power converter...).

B. Clutch-Brake-Combination

A powertrain is often connected to a drive unit, e.g. an asynchronous drive, at the primary part, which has a continuous speed. The secondary part consists of a working unit, which is the load. In case of no continuous usage of speed and torque, it is possible to engage a clutch between these two parts to connect or disconnect them. By engaging the clutch the two parts will be connected and the secondary part accelerates till synchronous speed. By disconnecting the clutch, the secondary part coasts to a standstill. In applications like excenter-presses, this behavior is not requested. This would be dangerous for human and machine, because the movement of the excenter is undefined. To avoid this, it is possible to install a brake to stop the excenter. These brake and clutch has to be synchronized, to engage them at the same time. By connecting break and clutch together in one mechanical device, it is possible to break automatically every time the clutch is not activated.

The principle of the functionality of the CBC is shown in fig. 3. The actuator engages the clutch by moving forward, which causes rising of the oil pressure and therefore pressing the springs, and moving the clutch-plates together (fig. 3 b). If the actuator moves backwards, the springs press the oil out of the cylinder and moves the brake-plates together (fig. 3 a).

In case of an failure, the CBC brakes automatically by the springs, although if there is no pressure in the system.

Fig. 4. Construction of the CBC

Fig. 4 shows the mechanical construction of the CBC. The inner part (1 and 2) is connected to the driveshaft. The outer part (3) on the left side is connected to the fixed flange of the body. The right outer part (4) of the body is connected to the rotating drive unit. The inner part (5) is the plunger cylinder in which the oil is filled to engage the clutch by pressing the plates (6).

III. System Analysis

The force which is produced with the actuator, generates a pressure depending on the area of the plunger.

\[ p = \frac{F}{A} \]  

(1)

Because of the compressibility of the oil, the volume of the oil is not constant over a range of pressure. The compressibility is qualified as compression-module \( K_{oil} \).

\[ \Delta V_{oil} = \frac{V_{oil} \cdot \Delta p}{K_{oil}} \]  

(2)

A position change \( \Delta x_{act} \) of the actuator generates a volume difference of the oil.

\[ \Delta V_{oil} = \Delta x_{act} \cdot A_{cyl} \]  

(3)

For \( p < F_{spring-begin} \cdot A_{cyl} \) and \( p > F_{spring-end} \cdot A_{cyl} \) the position of the actuator can be described as

\[ \Delta x_{act} = \frac{V_{oil} \cdot \Delta p}{K_{oil} \cdot A_{cyl}} \]  

(4)

For \( F_{spring-begin} \cdot A_{cyl} < p < F_{spring-end} \cdot A_{cyl} \) the position of the actuator can be describe as

\[ \Delta x_{act} = \frac{\Delta F}{R_{spring}} = \frac{\Delta p \cdot A_{cbc}}{R_{spring} \cdot i_{act-cbc}} \]  

(5)
A. Air in the Oil

In every hydraulic system there are cavities, e.g. because of the screw connections of the hydraulic line, sensors or other construction conditions of the mechanical elements. By filling the system with oil, in some cavities air is still present. Additionally there is air dissolved in the oil. Figure 5 shows this effect schematically.

![Fig. 5. Dissolved Air in the System](image)

The position difference of the actuator depending on the volume of the air is summarized in formula 6.

$$\Delta x_{act} = \frac{V_{air} \cdot p_0}{\Delta p \cdot A_{cyl}}$$

(6)

Measurements with different volumes of air in the oil is shown in figure 6. The less air volume of the oil, the steeply rises the pressure.

![Fig. 6. Dissolved Air in the System](image)

B. Elastics of the System

With a rising pressure the pipes and flexible tubes expands themselves. This effect is characterized with the factor $K'$, a modified compression module.

$$\Delta x_{act} = \frac{V_{oil} \cdot \Delta p}{K' \cdot A_{cyl}}$$

(7)

C. Temperature

Temperature change of the system will change the position of the armature in relation to the plunger in the CBC. This is caused by the volume change of the oil in the system depending on the temperature (fig. 7).

![Fig. 7. Temperature](image)

D. Leakage

The effect of leakage in the system is the same as the temperature effect. The position will change proportional on the quantity of the lost oil.

E. Mechanical Wear of the plates

With the wear of the plates they become slender. This causes a longer distance for the plunger to move from the brake plates to the clutch plates. Proportional to this distance the motor has to move a longer distance like shown in figure 8.

![Fig. 8. Wear of Plates](image)

F. Complete System

The simulation of the complete hydraulic and mechanical system is shown in figure 9. Summarized the first yellow part is dependent on the solved air in the system, the second green part primarily depends on the spring constant and the third blue part depends on the elastics of the mechanical elements like the pipes and tubes.
IV. EXISTING PRESSURE CONTROL

The existing pressure control consists of a standard position control cascade with an additional PI-controller to control the pressure. Based on the former hydraulic open loop control there are only four set points which are selectable with two digital inputs at the converter. For a higher performance, the pressure controller is provided with a position feed forward table. To include for position difference caused by temperature or leakage, the feed-forward table updates the actual position whenever the linear drive enforces a constant pressure. Because of the highly nonlinear behavior of the hydraulic system, the controller disposes of an i-logic which disables the integral part of the pressure controller for an adjustable time. Due to the small amount of set points it is possible to keep independent controller parameters for each of the four set points. This ensures an optimized performance of the controller at every point although the hydraulic system is very nonlinear.

V. CONCEPT FOR THE NEW PRESSURE CONTROL

The torque of the CBC is linear to the system pressure. With the possibility of a full controlled system, it is possible to control the clutch and brake torque very accurate. To meet this demand the control has to be adaptable.

A. Pressure Controller

The pressure characteristic is nearly linear in three parts of the characteristic curve. The first part is between zero and the spring curve, the second part is the whole spring characteristic and the third part is up to the maximum pressure. With the pressure in the second part, the plunger of the CBC is between the break and the clutch brakes. The secondary part of the powertrain is uncontrolled. Pressure set points in this part will always be avoid. The other two parts can be controlled with a standard PI-controller to keep the control as simple as possible.

B. Calculation of Position

By starting the control the first time, an initialization routine is done. During the initialization the pressure controller gets the starting position with a set point of 1 bar. After reaching this point, the position is written into an array. The control increments the pressure set point with one while 100 bar is reached and saves the position of every set point after reaching it. At the end, the array consists of a pressure-position-relation from one to 100 bar. This table is used as a feed forward table for the position.

With changing temperature or leakage, the pressure-position-relation changes as shown in figure 7. To keep the initialization table constant, a temperature coefficient $x_1$ is established. This coefficient only contains the deviation between the expected feed forward value $x_{fft}$ and the actual position at the set point. This coefficient is constant over the whole table. The adapted feed forward position is for every new set point is $x_{ff} = x_{fft} + x_1$.

Another coefficient observes the position difference $x_2$ caused by the wear of the plates. This is a very moderate progress which is only observable by moving from the brake state to the clutch state or reverse. In this case the distance between clutch and brake increases as shown in figure 8. In this case the feed forward position is $x_{ff} = x_{fft} + x_1 + x_2$.

C. Pressure Controller Logic

To obviate an overflow of the pressure PI-controller, the whole controller is deactivated while reaching a position, depending on the control deviation and on the time, if the actual pressure value has not reached the set point and does not change for a period of time. A deactivation only with the time is not recommendable because the time with worn plates can be up to three times longer. In this case the logic would activate the pressure controller too early and would cause an overshoot of the pressure.

D. Parameter Calculation

To reach a high performance of the control there has to be different parameters for the pressure-, position- and speed-controller depending on the actual pressure $p_{act}$ and the pressure difference $\Delta p$. In case of an higher pressure set point then the actual pressure parameters have to set to a maximum value. In case of a lower set point, the parameters has to be decreased. The dependency of the parameters is shown in fig. 11.

The resulting pressure curves with the calculated parameters are shown in fig. 12.

Figure 13 shows the whole optimized control structure.
VI. MEASUREMENTS

A. Position Calculation

Two times within 230 seconds, the volume of the oil is decreased by opening an valve. The minimized volume of the oil causes a position difference of \(2300\, \mu m\). The calculated position difference is shown in fig. 14.

A measurement of the whole pressure characteristic is shown in fig. 15. The difference of \(2300\, \mu m\) is also the same then calculated.

B. Clutch and Break Cycle

Figure 18 shows a fast engaging of the clutch within \(100\, ms\) and a reference curve of a superimposed process controller for \(240\, ms\). The pressure follows the reference curve immediately.

A measurement of the pressure characteristic shows the same difference than calculated (17).
For the breaking process the pressure has to decrease to less than 28 bars. This lasts within 100 ms. For the next 150 ms there is also a superimposed reference curve impressed.

VII. CONCLUSION

In the paper a specially designed pressure controller based on an industrial inverter is figured out in order to drive an highly nonlinear system. The designed control structure is able to fulfil all the requirements that were made concerning dynamics and accuracy. A comprehensive system analysis has contributed to an optimal behavior of the pressure control.

With the optimized position feed forward, the pressure controller is able to reach every set point in the range between 1 an 100 bar with a very fast response time. An overshooting of the actual pressure is suppressed by temporarily deactivating the pressure controller. The heavily non-linear characteristic of the hydraulics has been taken into account by providing different parameters for the PI based pressure with a function depending on the actual pressure and the reference pressure for each available set point.

VIII. TABLE OF USED SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$x_{ff}$</td>
<td>Feed forward position</td>
</tr>
<tr>
<td>$x_{fft}$</td>
<td>Expected feed forward position</td>
</tr>
<tr>
<td>$p_{act}$</td>
<td>Actual pressure</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure</td>
</tr>
<tr>
<td>$x_1$</td>
<td>Position correction factor 1</td>
</tr>
<tr>
<td>$x_2$</td>
<td>Position correction factor 2</td>
</tr>
<tr>
<td>$x_{act}$</td>
<td>Actual position</td>
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<tr>
<td>$V_{oil}$</td>
<td>Volume of oil</td>
</tr>
<tr>
<td>$V_{air}$</td>
<td>Volume of air</td>
</tr>
<tr>
<td>$K_{oil}$</td>
<td>Compression module of oil</td>
</tr>
<tr>
<td>$K'$</td>
<td>modified compression module</td>
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<tr>
<td>$A_{cyl}$</td>
<td>Area of hydraulic plunger</td>
</tr>
<tr>
<td>$A_{cbc}$</td>
<td>Area of CBC plunger</td>
</tr>
<tr>
<td>$F$</td>
<td>Force</td>
</tr>
<tr>
<td>$R_{spring}$</td>
<td>Spring constant</td>
</tr>
<tr>
<td>$i_{act-cbc}$</td>
<td>Ratio between actuator and CBC</td>
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</table>

REFERENCES