Generator Speed Control Utilizing Hydraulic Displacement Units in a Constant Pressure Grid for Mobile Electrical Systems

Thomas Dötschel, Michael Deeken, Dr.-Ing. Klaus Schneider
Liebherr-Werk-Nenzing GmbH, Dr.-Hans-Liebherr-Straße 1, 6710 Nenzing (Österreich), E-mail: Thomas.Doetschel@liebherr.com

Abstract

Liebherr mobile harbor cranes use electrical generators to provide electrical power for load attachment devices such as container spreaders or magnets. Upcoming exhaust and noise emission standards and energy saving considerations lead to a broad diesel engine speed range. The challenging design aspect is to ensure a constant speed of the asynchronous generator by the hydraulic drive system. In addition, electrical load profiles of inductive consumers usually have DT1 system characteristics with very small time constants. They evoke fast torque variations interfacing the hydraulic transmission.

Liebherr mobile harbor cranes, see Figure 1, usually have a closed hydraulic circuit containing a hydraulic pump with a high displacement volume that is adjusted electronically in accordance to the current diesel engine speed. Regarding the energy saving aspects, a further minimization of the diesel engine speed leads to a larger pump size with increasing torque losses.

Depending on the pressure setting, the volume flows can be reduced in constant pressure grids. Especially in part-load operation this results in better efficiency compared to closed hydraulic circuits by minimizing the displacement volume of hydraulic components. To obtain a stable generator speed, it is essential to adjust the displacement volume of the hydraulic unit for equalizing its input torque with the
generator load torque. In interaction with the software-based control architecture, the stability of the electrical frequency depends on the mass inertia of the generator drive and time constants of the embedded hydraulic actuators.

The system model, represented by ODEs is established and verified with a hydraulic simulation software. On that basis, the design approach of a PI-state-controller is presented. Corresponding controller gains and state feedback parameters are determined by pole placement techniques.

To conclude this investigation a comparison between the hydraulically closed circuit and the constant pressure grid is shown by simulation and measurement data.

KEYWORDS: Mobile hydraulics, displacement control, control design

1. Introduction
In harbor mobile cranes with diesel engines as a primary power source, an electrical power supply is provided by an asynchronous generator. In /4/, the increasing importance of energy efficiency is pointed out. Beside from power recuperation, an diesel engine power adjustment by the engine speed according to the power demand of the whole system is state-of-the-art. A criterion for the quality of an electrical power grid is its frequency, which directly depends on the generator speed. It is needed to be constant. In this work, a comparison between an open hydraulic drive in a constant pressure grid and a closed hydraulic circuit with volume flow controlled pump serving as drive units for an asynchronous generator is shown. The theory for speed controlled hydraulic displacement machines according to Figure 3 is provided by /2/.

![Figure 2: Closed hydraulic circuit with a volume flow controlled pump and a hydraulic motor with a constant displacement volume.](image-url)
In Figure 2 and in Figure 4, it becomes obvious that both drives are controlled by regulators for stabilizing the angular velocity of the hydraulic machines. The goal of the controller is to reject disturbances on the system as well as to account for a variable speed of the diesel engine.

**Figure 3:** Control of the angular velocity of a hydraulic machine with a mechanical and an electronic controller, proposed by /2/.

The advantage of the constant pressure grid system in Figure 4 is the independence of the controller to the variable diesel engine speed. A system architecture with huge closed circuit hydraulic pumps can be avoided by smaller pressure controlled pumps with a higher pressure level.

**Figure 4:** Open hydraulic circuit in a constant pressure grid for driving an asynchronous generator with a control of its angular velocity.
Moreover, the implementation to a multiple-hydraulic-consumer-system is possible. An autonomously controlled volume flow in the constant pressure grid is enabled by pressure controlled pumps. As a consequence, a suitable control law for stabilizing the generator frequency with a high robustness towards disturbances is presented. The comparison of the shown drive concepts is concluded by the stability of the generator frequency.

2. Dynamic behavior of hydraulic machines for generator drives

In order to obtain a robust control law, a model-based control design approach is derived. Therefore, the next sections show an appropriate system model as well as its linearization in an operational set point.

2.1. Modelling approach of a hydraulic machine

The nonlinear state space system is described by the set of nonlinear equations with

\[ \dot{x} = f(\theta, x, u, z) \quad \text{and} \quad y = g(x). \]  

(1)

It contains the parameters vector \( \theta \), the state vector \( x \), the scalar system input value \( u \), and \( z \) as the summarizing vector for all disturbance values. In this work, disturbance values are described with physical impacts that are uncertain and difficult to detect in experimental measurements and tests. Therefore, they will be defined in a later part. Previously, the state vector shall be defined by \( x = [V_G, \omega_{HM}]^T \). The displacement volume \( V_G \) of the hydraulic machine and the angular shaft velocity \( \omega_{HM} \) are chosen as state variables, because they show the direct correspondence of the torque and the shaft velocity. The system is considered as a drive shaft with a high stiffness and a high rotary mass \( J \). The hydraulic machine is connected to the generator without any transmission ratio. Therefore, the lumped inertia is calculated with \( J = \sum J_i \) as a sum of all inertias of the available mechanical parts. While operating the hydraulic machine, it will be controlled by the normed input value \( \zeta_{V_G} \) in order to adjust the angular shaft velocity \( \omega_{HM} \) to a constant value. As a result, the drive system input is

\[ u = \zeta_{V_G}. \]  

(2)

The output equation is described by

\[ y = \omega_{HM}. \]  

(3)

In a stationary process with a constant displacement volume \( \dot{V}_G = 0 \) with respect to the considered timeline, the input value leads to a unique displacement volume in a proportional manner like
Note, that the considered hydraulic machine has a direct proportionality between the input current $I_{VG}$ and the displacement volume $V_G$, where $\zeta_{VG} \equiv \frac{l_{HM}}{i_{HM,max}}$ is a normalized relation regarding the maximum current $I_{HM,max}$ of the electro-proportional pressure reducing valve. In the following the first order delay dynamics for the hydraulic machine displacement is treated as a simple approach to the real displacement dynamics, refer to /5/. In connection to the shaft drive dynamics, for the nonlinear state space dynamics it can be written

$$
\begin{bmatrix}
\dot{V}_G \\
\dot{\omega}_{HM}
\end{bmatrix} = \begin{bmatrix}
\frac{1}{T_{HM}} V_G + \frac{v_{G,max}}{T_{HM}} \cdot \zeta_{VG} \\
\frac{p_{CPG} - p_{BFL}}{2\pi J} \cdot V_G - \frac{1}{J} \cdot T_{El} - f_{Loss}(\omega_{HM}, \dot{\omega}_{HM}, V_G, \dot{V}_G, \theta)
\end{bmatrix}.
$$

(5)

The presented set of equations shows the relationship between the displacement of the hydraulic machine and the dedicated drive torque. The effective drive torque $T_{HM}$ of the hydraulic machine is summarized with

$$
T_{HM} = \left(\frac{p_{CPG} - p_{BFL}}{2\pi J}\right) \cdot V_G - f_{Tj}(\omega_{HM}, \dot{\omega}_{HM}, V_G, \dot{V}_G, \theta).
$$

(6)

It contains the pressure of the constant pressure grid $p_{CPG}$, the backflow pressure $p_{BFL}$ of the hydraulic machine as well as the underlying displacement volume $V_G$. From a theoretical point of view, these values lead to the ideal drive torque. In real applications, the effective drive torque is diminished by friction and oil leakage impacts.

\[\begin{array}{c}
\zeta_{VG} \\
\frac{V_{G,max}}{T_{HM}} \\
V_G \\
p_{CPG} \\
p_{BFL} \\
J \\
\omega_{HM} \\
T_{EL} \\
T_{HM}
\end{array}\]

\[\begin{array}{c}
Q_{HM} \\
G
\end{array}\]

**Figure 5:** System model structure of the generator drive.

The presented hydraulic generator drive configurations have to adjust a constant angular velocity characterized by the frequency of the electrical power grid. In the closed hydraulic circuit, a volume flow controlled pump forces a hydraulic motor to a fixed angular velocity. Subsequently, the system reacts with a hydraulic pressure that caused in dependency of the electrical load torque. In a constant pressure system, the connected hydraulic displacement machine provides a drive torque. This drive torque
results in an angular velocity as well as a consumption of an oil volume flow. Thus, the pressure-dependent oil leakage of the hydraulic machine has a neglectable impact in the presented system model. That means, if a rising oil leakage appears, the amount of the oil volume flow of the hydraulic machine in the constant pressure grid increases. The volume flow demand of the hydraulic displacement machine is calculated with

\[
Q_{HM} = \frac{\omega_{HM}}{2\pi} \cdot V_G + f_{Q_L}(p_{CPG}, p_{CPG}, \omega_{HM}, \dot{\omega}_{HM}, V_G, \dot{V}_G, \theta).
\] (7)

For the hydraulic oil leakage \(Q_{HM,V} = f_{Q_L}\), a nonlinear relation depending on the pressure \(p_{CPG}\), the angular velocity \(\omega_{HM}\) and the displacement volume \(V_G\) can be employed.

An oil supply in constant pressure grid can be guaranteed by hydraulic pumps and storages. The maximum provided amount of oil volume flow has to satisfy at least the demand of the oil volume flow of the simultaneously operated consumers including its oil leakage volume flows in the constant pressure grid. The constant pressure \(p_{CPG}\) is subject to disturbances which can be evoked by displacement time constants of constant pressure pumps reacting on changing oil volume flow demands of the connected hydraulic consumers. Such disturbing impacts can be reduced by applying hydraulic storages. From a reasonable economic point of view, small variations of the constant pressure \(p_{CPG}\) can be tolerated, which simplifies the system architecture. Nevertheless, for the control design that aims for the keeping the generator angular velocity constant, the constant pressure \(p_{CPG}\) is determined to become part of the disturbance vector \(z\).

Another element of the introduced disturbance vector is provided by the electrical load torque \(T_{EL}\) of the generator. The electrical load can be obtained with \(T_{EL} = \sum T_{EL,i}\) as a sum of all generator torques acting on the rotating system with the nomenclature \(i = \{fric, cool, exc, curr\}\). The variable \(T_{EL,fric}\) accounts for the friction in the bearings of the generator and \(T_{EL,cool}\) describes the impact of the air cooler which is under all circumstances capable of keeping the generator operating temperature constant. Furthermore, the asynchronous generator consists of an exciting machine for the rotor unit which leads to a much higher reactive power compared to synchronous electrical machines for maintaining their rotary electrical field leading to the torque \(T_{EL,exc}\). The exciting current of the rotary electrical field \(I_{exc}\) is regulated in accordance to a constant voltage on the three phases of the generator ports. In case of appearing electrical loads of inductive consumers by rising electrical currents \(i_{curr}\) in the three phases, the voltage at the generator ports is maintained by increasing the drive torque.
In general, a direct proportionality to the electrical load torque at the generator shaft is given with

\[ T_{curr} = f_{T_{curr}}(i_{curr}). \] \hspace{1cm} (8)

To keep simplicity, the unknown function \( f_{T_{curr}} \) describes the interaction between the main magnetic flux as well as the magnetic exciter field in dependence of upcoming electrical currents \( i_{curr} \) in the three phases. It shows the resulting torque of the electrical machine \( T_{curr} \) in order to maintain frequency and voltage. For the employment of dynamic effects and disturbances in the mobile electrical power grids the dynamic change of the torque is

\[ \dot{T}_{EL} = f_{EL}(\omega_{HM}, T_{EL}). \] \hspace{1cm} (9)

With respect to this, the disturbance vector is written by

\[ z = [ T_{EL}, p_{BFL}, p_{CPG} ]^T. \] \hspace{1cm} (10)

Due to the variability of the backflow pressure \( p_{BFL} \), it is declared as a disturbance variable. For a stable control of the frequency of the angular shaft velocity, the influence of these values is of high importance. Although a control design method for linear systems is chosen, the impact of the disturbance values is available in the system matrix by the known set points for the disturbances with \( z_0 = [ T_{EL,0}, p_{BFL,0}, p_{CPG,0} ]^T \). Mainly, they will depend on the hydraulic system architecture.

3. Generator frequency control

Controlling the displacement of an axial piston machine in a constant pressure grid directly influences the torque on the mechanical shaft. Alternating load torques on that shaft have to be compensated by adjusting the displacement in order to keep the generator speed constant. Small deviations in the stationary generator speed lead to frequencies in the electrical grid that are acceptable for a short time range.

3.1. Model extension by a disturbance dynamics

The nonlinear dynamic system with the hydraulic machine and generator dynamics as well as the displacement behavior is extended by the disturbance equation. As a result the new dynamic system can be written like
Consequently, the new state vector \( x = [V_G, \omega_{HM}, T_{EL}]^T \) is actualized by the load torque \( T_{EL} \) caused by the asynchronous generator. The additional dynamics \( f_x(\omega_{HM}, T_{EL}) \) consists of parameters showing the impact of \( \omega_{HM} \) and \( T_{EL} \), which has to be identified.

### 3.2. Disturbance identification of the nonlinear system behavior

System parameters of the mechanical and hydraulic system part are well known as a result of the design procedure. In contrast to that, the parameterization for the mobile electrical power grid underlies multiple caused by impacts of the system architecture and its consumers that are in operation. Thus, the availability of measurements obtained by tests on different generator drives enables the parameterization of the disturbance dynamics. These models can be taken in a following design stage for simulative robustness investigations.

![Figure 6](image.png)

**Figure 6:** Comparison of measurement (black) and simulation (grey) as a result of the parameter and disturbance identification.

In **Figure 6**, the result of the parameter identification with an appropriate system model and measurement data is depicted. Therefore, the parameter vector \( \theta = [\theta; \theta_z] \) is separated by known parameters \( \Theta \) and unknown disturbance parameters \( \Theta_z \), which has to be detected by an identification method. In order to do this, it is necessary to minimize the quality criterion

\[
\Gamma = \min_{\theta_z}(\omega_{HM,meas} - \omega_{HM}) .
\] (12)
The obtained disturbance dynamics, depicted in Figure 7, is employed for the robustness proof of the following PI-state controller that is parameterized with pole placement techniques.

Figure 7: Estimation of the load torque (left) and the constant pressure (right) as normalized disturbance variables.

3.3. State space controller design

For the control of systems with a robust disturbance rejection a state controller in combination with a linear PI-controller is proposed, compare to Figure 8.

Figure 8: PI-state controller layout for the generator frequency control with a open hydraulic circuit in a constant pressure grid.

The advantages of a state controller design technique can be found in the possibility to define eigenvalues for the controlled system in order to fulfill the desired dynamic behavior. Furthermore, such methods imply the system dynamics directly in their feedback gain calculation, in which the chosen state variables will be stabilized. Therefore, the controller error is defined with
\[ \dot{e} = w - y. \] (13)

Here, the desired angular velocity \( w = \omega_{HM,d} \) as a representative of the generator frequency is compared to the underlying angular velocity \( y = \omega_{HM} \) in order to obtain stationary accordance \( y = w \) for \( \dot{e} = 0 \) between the actual and the desired generator frequency, see (11). The control equation is described by

\[
u = -k_x^T x + k_i e + k_p \dot{e}. \] (14)

With reference on Figure 8, the parameter \( k_x^T \) describes the state feedback gain for stabilizing oscillations in the system with low damping characteristics. The proportional part of the PI-controller is parameterized with \( k_p \), whereas \( k_i \) weights the integrated controller deviation. Due to the possibility of a destabilizing controller parameterization, the controller gains are obtained by pole placement using the Ackermann method. In this case, the calculation of the controller matrix becomes necessary, leading to the statement of the controllability of the underlying system. The controllability statement characterizes if a system can be transformed from one desired state into another desired state.

### 3.4. Simulative robustness proof

As shown in Figure 6, a simulative robustness proof is shown simultaneously with the identification of the disturbance dynamics \( \dot{T}_{EL} = f_z(\omega_{HM}, T_{EL}) \). This model is utilized as an unknown disturbance to the dynamic generator drive model (11) stabilized by the PI-state controller. Although, the controlled system has poles that are placed by the user, there exists a dynamic disturbance behavior which is capable of destabilizing the controlled system in theory. In practical applications, the disturbance characteristics has a unique shape with reference to its cause, which allows statements of the structure and a parameter region of it.

### 4. Experiment and testing

After applying theoretical methods, the sustainability of the system design in combination with the controller implementation in the software has to be validated. Different electrical loads are switched to the power grid in order to disturb the frequency control of the hydraulic machine. Moreover, a practical parameter study is performed by the implementation of a high as well as a low backflow pressure stage for the open circuit hydraulic machine. In Figure 10, the normed control current as an approximation of the hydraulic displacement volume is shown for the closed circuit hydraulic system (black) as well as the open circuit hydraulic system in the constant
pressure grid (grey). In Figure 11, the pressure in the closed hydraulic circuit shows large magnitudes which decrease over time. Obviously, the elasticity of the hydraulic system evokes oscillations in the frequency. Due to the fixed pressure in the constant pressure system (grey), a certain stiffness of the hydraulic system is achieved by a hydraulic decoupling of the pressure controlled pumps. In case of increasing load torques, it is only necessary to adjust the displacement of the hydraulic machine, which leads to small deviations from the reference frequency of the nominal frequency. In this case, the frequency stability strongly depends on the time constant of the hydraulic displacement dynamics. The deviation of the frequency can be diminished because the higher pressure differential over the displacement machine increases in comparison to the lower backflow pressure configuration.

**Figure 9:** Comparison of the control current of the hydraulic displacement machine in the constant pressure system (grey) to the closed hydraulic circuit (black).

**Figure 10:** Comparison of the frequency stability of the hydraulic displacement machine in the constant pressure system (grey) to the closed hydraulic circuit (black).
5. Conclusion
For a hydraulic powered asynchronous generator, a robust software control strategy implying structural known disturbances for hydraulic displacement machines in a constant pressure grid is presented. For electrical power grids a constant frequency is essential. Therefore, a comparison of the performance of the closed hydraulic circuit to the constant pressure architecture is done. The higher stability of the constant pressure system is pointed out by lower deviations from the target frequency of the asynchronous generator.

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