# Advanced Control Concepts Suitable for Energy Efficient Hydraulic Systems

Tadej Tašner[1,2], Vito Tič[1] and Darko Lovrec[2]
[1] *HAWE Hidravlika d.o.o., Petrovče, Slovenia*
[2] *Faculty of Mechanical Engineering, University of Maribor, Maribor, Slovenia*

Keywords: Hydraulics, Simulation, Control, Asynchronous Motor, Variable-frequency Drive Controller, Constant Pump, Variable Displacement Pump, Hydraulic Tubing.

Abstract: Today there are ever-increasing demands for more efficient hydraulic drive technology in terms of reducing energy consumption, increasing reliability plus robustness, and for minimising the maintenance interventions on the drive. In addition, the requirements and directives on the reduction of the noise, and development tendencies in the direction of environmentally user-friendly drives, are leading to an ever-increasing usage of the electro-hydraulic drive technology. There are two main concepts for converting electrical into hydraulic energy: constant speed motor coupled with variable displacement pump and variable speed motor coupled with constant pump. This article presents completed PhD tasks including modelling and simulation of both of the two concepts along with a new concept of variable speed motor coupled with constant pump. All the concepts are compared in dynamics and efficiency based on the simulation results. The expected results of the PhD are a newly-synthesized SIMO controller for 'bi-variable' control, that will be able to control hydraulic systems in order to operate within the areas of maximum efficiency, highest dynamics or a compromise between these two. A further contribution to the hydraulic system developers' community will be an experimentally proven mathematical model for the simulations of different hydraulic drive concepts. Such a model may be used for optimising the energy-efficiencies of existing and new hydraulic machinery, as well as efficiency prediction when building new hydraulic systems. Moreover it may also be used for determining the most suitable drive concept for a hydraulic system with a predefined operating cycle.

## 1 RESEARCH PROBLEM

Today there are ever-increasing demands for more efficient hydraulic drive technology in terms of reducing energy consumption, increasing reliability plus robustness, and for minimising the maintenance interventions on the drive. In addition, the requirements and directives on the reduction of the noise, and development tendencies in the direction of environmentally user-friendly drives, are leading to an ever-increasing usage of the electro-hydraulic drive technology. This has come to light especially in areas where higher energy density is required, for example, on modern milling and forming machines as well as in the segment of machinery and equipment that operates continuously, autonomously, at remote sites and without the presence or supervision of the maintenance staff. Such applications include:

- highly loaded machining centres
- wind-farms (especially those installed in such as waterside bay areas)
- mobile machines (excavators, cranes, etc.)

Hydraulics are mainly used in systems where big forces and power are required for normal operations. High-power requirements reflects in high-energy consumption. Even the slightest (at the percentage level) optimisations regarding the energy efficiency of such machines could result in significant savings over long operating periods. Therefore, improved efficiency and reduced energy consumption are two of the main goals during modern electrohydraulic drive systems designing.

## 2 STATE OF THE ART

Most of the efficiency increases could be achieved



Tašner T., Tič V. and Lovrec D. (2013).
Advanced Control Concepts Suitable for Energy Efficient Hydraulic Systems.
In *Doctoral Consortium*, pages 3-11
DOI: 10.5220/0004637000030011

during the electrical to hydraulic energy conversion stage. In general hydraulic energy can be controlled in two main ways:

- the throttling principle
  by throttling on the directional valve
- the volumetric principle
  by adjusting the pump displacement volume

The throttling principle displays good dynamic behaviour, but its energy losses are substantial. The volumetric principle is more energy-friendlier but has even worse dynamic response (Majumdar, 2000). The volumetric principle is mostly used due to its better efficiency.

There are two more commonly used drive concepts for the volumetric adjustment of hydraulic energy:

- the direct concept
  by adjusting the displacement of a variable displacement pump
- the indirect concept
  by adjusting the rotational speed of a constant displacement pump

Most machines within the field of hydraulic drive technology are still using the classic drive concept—variable displacement pump driven at constant speeds. Desiring greater robustness and lowering the price of hydraulic drives over recent years, as well as lower prices for variable-frequency drive controllers, has led to the more and more popularity for using speed-controlled constant pumps. However, such a concept can not meet the dynamic requirements of the classic drive concept (Lovrec and Ulaga, 2007; Lovrec et al., 2005). Therefore the question arises as to 'Which drive concept would be more efficient for a particular hydraulic application?'.

Table 1 shows a comparison between both drive concepts: variable pump and constant motor (C1), as well as constant pump and variable speed drive (C2). C2 has asserted itself mainly due to good efficiency and a wide operating range (Xu et al., 2010). It also excels at lower operating costs and consecutively smaller affects on the environment. Lower operating costs and higher efficiency have come mainly from a more efficient connection of the motor to the electrical grid—via a variable-frequency drive controller (Ferreira et al., 2011). However, C2 also has one major drawback a slow system response that occurs due to rotating parts' moment of inertia (motor rotor and rotary parts of the pump). The response of C1 is up to 5 times faster than the response of C2 (Lovrec et al., 2005), but both responses are good enough for the most hydraulic applications (Lovrec et al., 2009).

Table 1: Comparisons between both the more-commonly used hydraulic drive concepts.

| | **Concept 1 (C1):** Asynchronous Motor Variable Axial Piston Pump | **Concept 2 (C2):** Frequency Inverter Asynchronous Motor Constant Gear Pump |
|---|---|---|
| **efficiency** | lower | higher |
| **reliability** | high | high |
| **operating costs** | higher | lower |
| **system dynamics** | higher 4.4 | lower 1 |
| **purchase price** | higher | lower |

If both concepts (C1 and C2) were to be combined, then we would obtain increased dynamics due to the variable displacement pump (Song et al., 2008) and increased efficiency due to the variable-frequency drive controller (Ferreira et al., 2011). Such a combined drive concept (Figure 1), as well as the effect of different motor speeds and different pump displacements on the efficiency and dynamics of the system, yet this is rarely mentioned in literature.

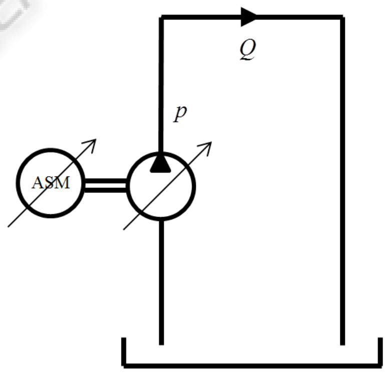



Figure 1: Combined drive concept (C3) - variable speed asynchronous motor coupled with a variable displacement pump.

## 3 OUTLINE OF THE OBJECTIVES

Based on the experience gained during the initial study of the dynamic behaviour and implementation of appropriate control strategies of both drive concepts, this new drive concept (a combination of variable speed drive and variable displacement hydraulic pump) needs to be explored in more detail. In order to achieve this objective both existing concepts will have to be studied on the basis of simulation and experimental analysis. The presented study will focus on both the selection and implementation of an appropriate control concept for this new drive concept, as well as the comparisons between the efficiencies for all three drive concepts.

The results obtained on the basis of these studies will be transferred to the proposed drive concept. Firstly, a new solution for the control concept will have to be found, because multiple physical quantities have to be monitored and changed at the same time:

- swashplate angle of the variable axial piston pump
- rotational speed of the motor

As both quantities directly effect oil pressure within the hydraulic circuit, an appropriate control strategy for such Single Input Multiple Output (SIMO) system will have to be synthesised. Such a control strategy then raises some further questions. Which quantity should be changed and how much, in order to maximise the efficiency and/or dynamics of the drive? What are the efficiency benefits of the new drive concept for the particular operating cycles of different machines?

# 4 METHODOLOGY

These research activities have an interdisciplinary nature, since they combine the knowledge of technical apparatus (specifications, principle of operations and performances of the individual components) as well as classic and modern methods for controlling systems. The starting point of scientific activities is a good knowledge of the individual components, their specific structures and modes of operation that is obtained from the manufacturer or provider of components. All of those properties are needed when constructing a mathematical model for simulating all three drive concepts. It is necessary to evaluate all the existing drive concepts and compare them using simulation as well as experimentally.

Appropriate and powerful simulation tools need to be used for simulation. The existing test site and equipment to be used for experimental verification of the simulation results will have to be redesigned and updated in terms of performance, flexibility and versatility. A new system for controlling and monitoring the test site will have to be designed that includes automatic set-point and disturbance generation, as well as data acquisition and archiving.

# 5 EXPECTED OUTCOME

The expected outcome of this research is to make some original contributions to the development of this scientific discipline. The first scientific contribution of the PhD work is a mathematical model for evaluation of the dynamics and efficiency of different hydraulic drive concepts that can be verified on a test-site. A further contribution is the design of an appropriate control strategy for 'bi-variable' (variable displacement, variable speed) hydraulic drive units that will be based on modern control and decision-making concepts.

# 6 STAGE OF THE RESEARCH

Most of the theoretical part of the work has been already done, meaning that the simulation model already consists of the following mathematical models:

- variable-frequency drive controller,
- asynchronous motor,
- variable displacement axial piston pump,
- fixed displacement internal gear pump,
- and hydraulic tubing

All the simulations regarding the efficiencies of different drive concepts have been run. Some results have already been experimentally verified on a test-site.

Therefore the used simulation model, including the simulation results, are presented in this article.

# 7 SIMULATION MODEL

The simulation model of C1 consists of an asynchronous motor directly connected to grid, a variable displacement axial piston pump, and hydraulic tubing. The simulation model of C2 consists of a variable-frequency drive controller with an asynchronous motor connected to it, a fixed displacement internal gear pump, and hydraulic tubing. The simulation model of C3 consists of a variable-frequency drive controller with an asynchronous motor connected to it, a variable displacement axial piston pump, and hydraulic tubing.

The asynchronous motor is coupled to the pump that pumps the hydraulic fluid from a hydraulic tank through a long length of hydraulic tubing, and back to the tank. The flow through the tubing causes a pressure drop that is measured directly after the pump using a pressure sensor. The following subsection presents the models of all the components used in simulation.

## 7.1 Asynchronous Motor Model

The more simplified model of the three-phase asynchronous motor consists of two pairs of magnetically-coupled symmetrical three-phase windings. Both of the three-phase windings (stator winding and rotor winding) are identical. The same model also applies for a squirrel-cage rotor, where currents start flowing due to electro-magnetic induction. The more commonly used model for the dynamic simulation of asynchronous motors is based on the so-called 'T' equivalent circuit (Figure 2). Such a model is used for static and dynamic simulations, although it neglects core losses (saturation and eddy current losses). (Diaz et al., 2009) The following equations can be obtained

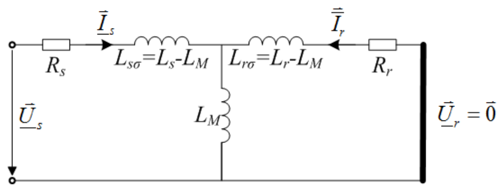



Figure 2: Electrical representation of the 'T' equivalent circuit of an asynchronous motor.

from the 'T' equivalent circuit for a squirrel-cage rotor:

$$\begin{aligned} \underline{\vec{U}}_s &= R_s \cdot \underline{\vec{I}}_s + \mathbf{L}_s \cdot \dot{\underline{\vec{I}}}_s + \mathbf{L}_M \cdot \dot{\underline{\vec{I}}}_r \\ \vec{0} = \underline{\vec{U}}_r &= R_r \cdot \underline{\vec{I}}_r + \mathbf{L}_r \cdot \dot{\underline{\vec{I}}}_r + \mathbf{L}_M \cdot \dot{\underline{\vec{I}}}_s \end{aligned} \tag{1}$$

where $\underline{\vec{U}}/\underline{\vec{I}}$ are vectors of voltage/current phasors for each phase, $R$ winding resistances and $\mathbf{L}$ inductances, where subscript 'r' denotes rotor, 's' stator and 'M' mutual. The electromechanical torque can be written as:

$$\underline{\vec{I}}_s^T \mathbf{L}_M \cdot \dot{\underline{\vec{I}}}_r = T_{em} = J\frac{\mathrm{d}\Omega}{\mathrm{d}t} + b \cdot \Omega + T_L \tag{2}$$

where $\Omega$ is the mechanical rotational frequency of the rotor, $b$ the viscous friction coefficient and $T_L$ the load torque. (Delaleau et al., 2001)

All the necessary parameters of the asynchronous motor needed for the simulation model were calculated from the measurements' results from the locked rotor and no-load tests.

## 7.2 Variable-frequency Drive Controller Model

The variable-frequency drive controller is a device that can change the rotational frequency or torque of an electric motor by modifying the frequency and amplitude of the motor's supply voltage. A typical variable-frequency drive controller consists of (Figure 3):

- a rectifier,
  rectifies input AC voltage
- DC link,
  energy storage
- an inverter,
  converts DC to AC
- and a control circuit.
  measures motor currents and controls the inverter

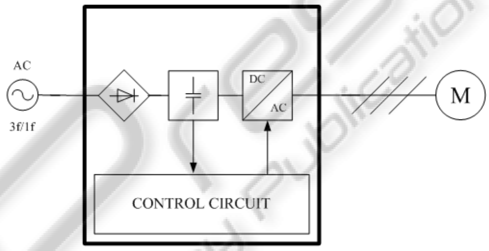



Figure 3: Block diagram of a variable-frequency drive controller.

### 7.2.1 Rectifier Model

The more commonly used rectifier type in variable-frequency drive controllers is a full-wave rectifier with an LC filter. Such a rectifier consist of three pairs of rectifier diodes and a filter inductor, which charge the DC link capacitor. Electrical circuit of such rectifier is shown in Figure 4. Assuming that the LC filter is ideal and that the voltage drop on the diodes is constant, the rectifier losses can be expressed as (3), where $U_F$ is the diode forward voltage. The factor 2 in the equation represents current flows through one lower and one upper diode.

$$P_{RECT} = 2 \cdot I_{DC} \cdot U_F \tag{3}$$

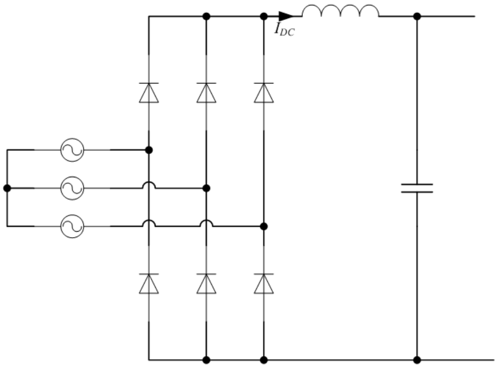



Figure 4: Three phase full wave rectifier with LC filter.

#### 7.2.2 DC Link Model

An ideal DC link capacitor is used in our variable-frequency drive controller.

#### 7.2.3 Inverter Model

Most of the inverters are controlled based on the principle of space vector modulation. A three-phase vector Pulse Width Modulation (PWM) inverter requires three transistor bridges. Each bridge consists of two transistors operating in the switching region. Such an inverter can drive any three-phase load, with or without the neutral line. Figure 5 shows such an inverter, where the transistors are represented as switches. The transistors used in the inverters are either Isolated Gate Bipolar Transistors (IGBTs) or Metal Oxide Semiconductior Field Effect Transistors (MOSFETs) and are controlled by a microcontroller. A simplified loss model for switching and conductive losses of MOSFETs will be used (Shen et al., 2006), because our inverter uses MOSFETs. The energy loss in one switch of both transistors within a transistor bridge, can be expressed as (4). Where all the data are characteristics of the MOSFET except $U$ and $I$, are dependent on the inverter load, and represent the voltage on the MOSFET and the current flowing through the MOSFET, respectively.

$$\begin{aligned} W_{SW} = & \frac{1}{2} I_D U_D \left( t_{ON} + t_{OFF} \right) \\ & + \frac{1}{2} \left( C_{GD} + C_{DS} \right) U_D^2 \end{aligned} \tag{4}$$

The conducting losses for the transistors during their on time are (4)

$$P_{ON} = R_{DS(on)} \cdot \left( i_a^2 + i_b^2 + i_c^2 \right) \tag{5}$$

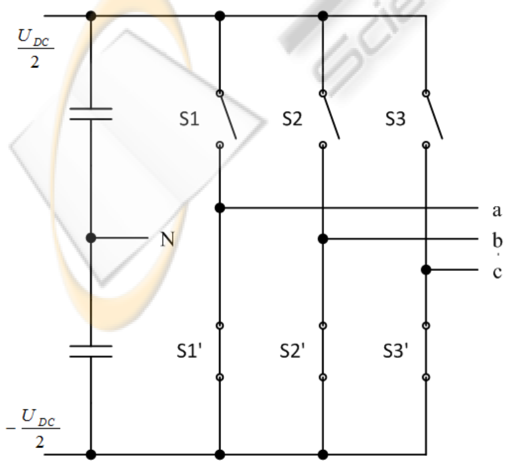



Figure 5: Three phase bridge inverter.

and the total losses are (6)

$$P_{INV} = P_{ON} + \frac{W_{SW}}{f_{SW}} \tag{6}$$

where $f_{SW}$ is the inverter switching frequency.

#### 7.2.4 Control Circuit Model

Most variable-frequency drive controllers control the drives using the Sensorless Vector Control (SVC) principle. SVC needs an inverse dynamic model of the asynchronous motor in dq coordinates for it to function. Synthesis of such a model is beyond the scope of this article. The basic parts of its synthesis are presented in (Stekl, 2007). A simplified block diagram of the control circuit is shown in Figure 6, its main blocks are:

- rotor flux estimator,

  rotor flux is needed for decoupling
- decoupling,

  decouples d and q voltage components so each can be changed without effecting the other
- two current controllers,

  control magnetising (d) and torque (q) current components by changing their voltages
- speed controller,

  controls motor rotational speed by changing the torque current component
- and load compensation.

  detects the load and increases the required torque current accordingly, in order to maintain the desired rotational speed

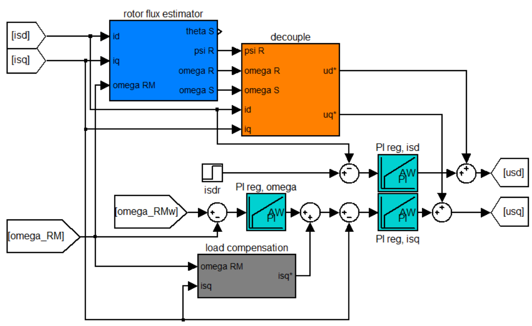



Figure 6: Control circuit block diagram.

### 7.3 Hydraulic Pump Models

Gear and an axial piston pump were used for the conversion of mechanical energy into hydraulic energy. Gear pumps have constant displacement volume, whereas axial piston pumps may also be adjustable. The flow through them can be changed by adjusting the swash-plate angle.

Wilson (Wilson, 1949) started modelling hydraulic pumps at the end of the first half of the 20th century. He designed a static model that takes into account volumetric and frictional losses of pumps. His model was later improved by some authors, their work was summed-up by Rydberg (Rydberg, 2009). However, these models are only useful for the already-made and tested pumps, because their coefficients are obtained from measurements and experiments on the stations. (Jeong and Kim, 2007) Our models are based on Wilson's model and use certain modifications mentioned by Rydberg.

#### 7.3.1 Fixed Displacement Internal Gear Pump Model

The flow $Q$ of the hydraulic fluid through the gear pump is the proportional to the rotational speed $n$ of the pump. Both quantities are connected by the pump's displacement $V_g$, which is a characteristic property of each pump. At higher pressures the pump's flow slowly starts to drop, due to the compressibility of the fluid and the greater leakage losses. These losses are covered under the concept of the volumetric efficiency of the pumps, which depends on the pressure difference $\Delta p$ that the pump must create. Such losses can easily be presented as a parallel hydraulic resistance $R_p$ of the ideal pump (an analogy with Norton's theorem in electrical engineering). (7)

$$Q = n \cdot V_g - \frac{\Delta p}{R_p} \tag{7}$$

The pump's operating torque is proportional to the pressure difference that the pump creates, due to the law of energy conservation. The torque $T$ must, in addition to creating pressure difference, overcome those losses that occur due to lubrication of the pump and friction between the moving parts of the pump $b$. (8)

$$T = \Delta p \cdot V_g + n \cdot b \tag{8}$$

#### 7.3.2 Variable Displacement Axial Pump Model

Similarly as for the constant gear pump, an equation can also be written for the variable displacement axial piston pump. The flow of the pump is affected by the pump's displacement setting $\alpha$, therefore we obtain (9) from (7).

$$Q = n \cdot \alpha \cdot V_g - \frac{\Delta p}{R_p} \tag{9}$$

The pump's displacement setting has a similar effect on the required operating torque. The losses appear due to the lubrication and friction of the rotating parts $b$, but additional losses of the friction between piston and walls $C$ must also be taken into account. (10)

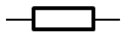
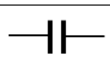
$$T = \Delta p \cdot \alpha \cdot V_g + \omega \cdot \alpha \cdot b + \omega \cdot \frac{\tan\alpha}{\tan\alpha_{\max}} \cdot C \tag{10}$$

### 7.4 Hydraulic Tubing Model

The dynamic behaviour of the fluid within the pipeline can be modelled in several different ways. The most exact model is based on the Navier-Stokes equations and the law of mass conservation, which results in a system of partial differential equations that are too much time consuming for such simulations.

Such an exact model of a hydraulic pipeline was unnecessary, therefore a more appropriate—discrete model was chosen, also known as a model with concentrated parameters. The discrete model is similar to electrical circuits used by electrical engineers, where the properties of a circuit are represented by resistance, capacitance and inductance. In hydraulics the properties of a pipeline system are hydraulic resistance $R_H$ (pressure drop in a tube due to flow), hydraulic capacitance $C_H$ (pressure drop in a tube due to tube volume increase/decrease) and hydraulic inductance $L_H$ (pressure drop due to fluid acceleration/deceleration). The analogy between electronics and hydraulics is presented in Table 2.(Tašner and Lovrec, 2011)

Table 2: Electrical-hydraulic analogy.

| **Electrical Symbol** | **Electrical Equation** | **Hydraulic Equation** |
|---|---|---|
|  | $U = R \cdot I$ | $p = R_H \cdot Q$ |
|  | $p = L \cdot \frac{dI}{dt}$ | $p = L_H \cdot \frac{dQ}{dt}$ |
|  | $U = \frac{1}{C} \int I dt$ | $p = \frac{1}{C_H} \int Q dt$ |

The hydraulic pipeline system split into $n$ segments can be represented using the electrical symbols as shown in Figure 7. Each segment represents part of a pipeline with a length of $l/n$, where $l$ is the total length of the pipeline. The number of segments also equals the number of possible pressure measurement points (For example, if the tube is modelled as one segment, the pressure in the middle of the tube cannot be calculated.)

The transfer function of such a segment can be expressed as a second-order ordinary differential equa-

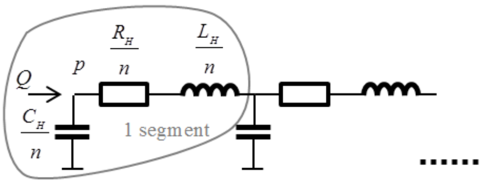



Figure 7: Hydraulic tubing represented using electrical symbols.

tion within the Laplace frequency domain (11).

$$\frac{p}{Q} = \frac{L_H \cdot s + R_H}{C_H L_H \cdot s^2 + C_H R_H \cdot s + 1} \tag{11}$$

## 8 SIMULATION RESULTS

Simulations were performed on different combinations of simulation models, as described in the previous section, using MATLAB/Simulink software. The same hydraulic tubing was used as a load when simulating all three concepts. Pressure control was realised using a PID controller using a clamping anti-windup method for the integral part.

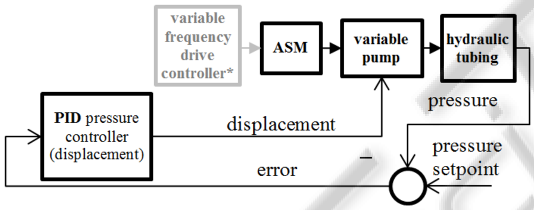



Figure 8: Control block diagram of C1 and C1*.

- In C1 the pressure is controlled by changing the axial piston pump's displacement, as shown in Figure 8. The asynchronous motor (ASM) was connected directly to power-grid (C1), and via the variable frequency drive controller, set to a constant rotational speed of 1500 min$^{-1}$ (C1*). C1* was just added to include the variable-frequency drive controller's losses to C1.
- In C2 the pressure is controlled by changing motor's rotational speed that drives the constant gear pump, as shown in Figure 9.

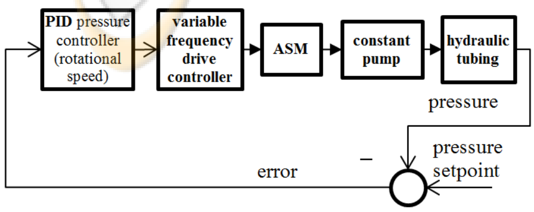



Figure 9: Control block diagram of C2.

- In C3 the pressure is controlled by two parallel PID controllers that change the motor's rotational speed and the axial piston pump's displacement, as shown in Figure 10.

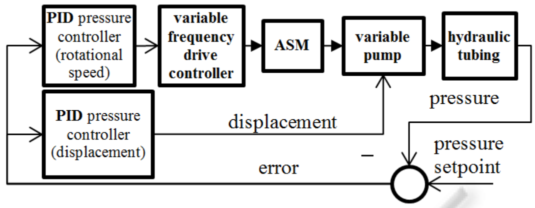



Figure 10: Control strategy of C3.

The pressure set-point was changed according to a combined cycle (sine, ramp and step). The pressure responses of the different drive concepts were observed and compared according to dynamics and efficiency. The simulation results are presented in Figures 11- 12, and in Table 3. The upper part of the Table 3 shows the efficiency comparisons. The average efficiency is calculated as a quotient of the produced hydraulic energy and the required electrical energy. A first look at the efficiencies reveals that C1 had the highest efficiency, due to the fact that a variable frequency drive controller was not used in C1. The new combined concept (C3) was the most efficient from amongst all the concepts that use variable-frequency drive controllers. If we had wanted to compare operating cycle costs, apparent power would have to have been compared, as companies pay for both the real power [W] and the reactive power [VAr]. For most of the new variable-frequency drive controllers, real power is almost equal to apparent power, due to built-in power-factor corrections (their cos$\phi \approx 1$, as reactive power flows only between the motor and the controller).

The bottom part of the Table 3 compares all the concepts according to control performances. Settling-times in response to step from 50-150 bar were calculated - a close-up of the step responses is shown in Figure 12. Settiling-time is the time when the pressure within the pipeline settled within 1 % of the pressure set-point. Moreover, the Root Mean Square Error (RMSE) was calculated for the whole cycle and for the tracking part (ramp and sine). It can be seen that the C1 had the best dynamics and lowest overall RMSE. C2 had the worst RMSE as well as the lowest dynamics with settling-time 6 times slower than C1. This was mostly due to the moment of inertia of the motor rotor and the rotating parts of the pump, as the motor must accelerate and decelerate according to the pressure set-point changes. Although C1 had the best dynamics, C3 tracked the pressure set-point more accurately.

Table 3: A comparison between different drive concepts' simulation results.

| | C1 | C1* | C2 | C3 |
|---|---|---|---|---|
| **average efficiency** | 72 % | 60 % | 65 % | 67 % |
| **energy required [kJ]** | 31.6 | 33.8 | 30.9 | 32.2 |
| **energy produced [kJ]** | 22.7 | 20.1 | 20.2 | 21.7 |
| **RMSE [bar]** | 4.92 | 6.40 | 12.9 | 7.16 |
| **tracking RMSE [bar]** | 0.231 | 0.227 | 5.95 | 0.178 |
| **settling time [ms]** | 92 | 95 | 570 | 176 |

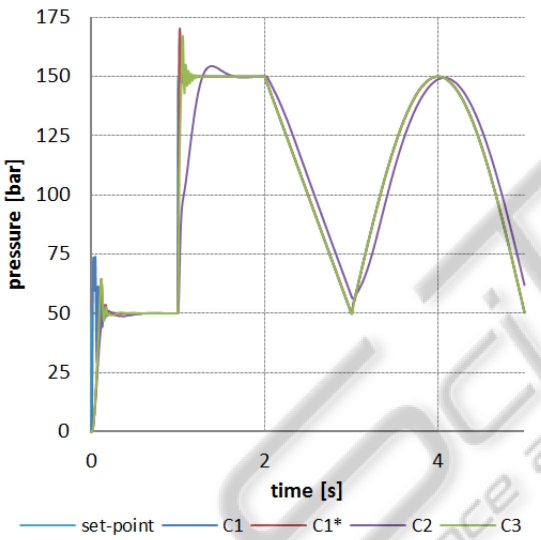



Figure 11: Responses of the different drive concepts to the combined cycle.

## 9 CONCLUSIONS

The simulation results showed that all three concepts are useful in the field of hydraulic drives. Although C2 had relatively slow dynamics, it can be used for processes where high dynamics is not critical. While C1 is the oldest concept, it still has the best efficiency - but we must also consider the reactive power of the asynchronous motor when it is powered directly from the grid. The newly-proposed concept (C3) had a very good tracking performance of slower changes of the pressure set-point, moreover it also had the best efficiency of all the variable-frequency drive controlled

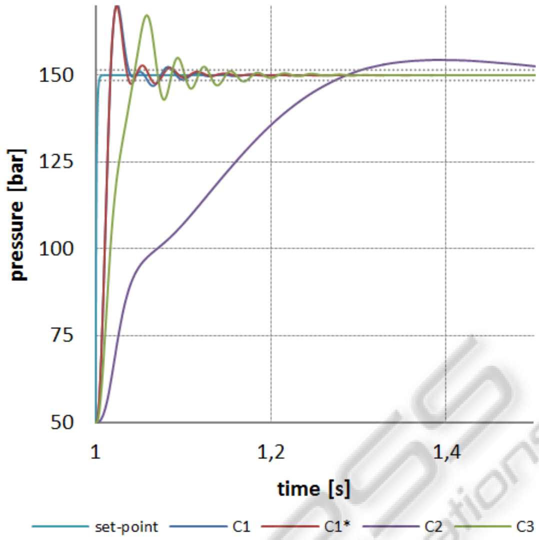



Figure 12: Step response close-up.

concepts. Its dynamics could be improved by implementing a better controller for the 'bi-variable' pressure control. The controller could use a look-up table for efficiencies and set the swashplate angle and motor rotational speed to such value that maximum efficiency is achieved.

## 10 FURTHER WORK

Further steps in pursuing the PhD are:

- experimental verification of efficiencies for each modelled component on the test site,
- synthesis of a new SIMO controller for 'bi-variable' pressure control (max dynamics or max efficiency control),
- tuning and testing of the new controller within a simulation environment,
- experimental evaluation of the tuned SIMO controller and
- experimental evaluation of the efficiencies of all three concepts.

## 11 EXPECTED OUTCOME

The expected results of the PhD are a newly-synthesized SIMO controller for 'bi-variable' control, that will be able to control hydraulic systems in order to operate within the areas of maximum efficiency, highest dynamics or a compromise between these two. A further contribution to the hydraulic system developers' community will be an experimentally proven mathematical model for the simulations of different hydraulic drive concepts. Such a model may be used for optimising the energy-efficiencies of existing and new hydraulic machinery, as well as efficiency prediction when building new hydraulic systems. Moreover it may also be used for determining the most suitable drive concept for a hydraulic system with a predefined operating cycle.

## ACKNOWLEDGEMENTS

HYDRAULIK

Investing in your future
OPERATION PART FINANCED BY THE EUROPEAN UNION
European Social Fund

Operation part financed by the European Union, European Social Fund. Operation implemented in the framework of the Operational Programme for Human Resources Development for the Period 2007-2013, Priority axis 1: Promoting entrepreneurship and adaptability, Main type of activity 1.1.: Experts and researchers for competitive enterprises.

## REFERENCES

Delaleau, E., Louis, J.-P., and Ortega, R. (2001). Modelling and control of induction motors. *International Journal of Applied Mathematics and Computer Science*, 11(1):105–129.

Diaz, A., Saltares, R., Rodriguez, C., Nunez, R., Ortiz-Rivera, E., and Gonzalez-Llorente, J. (2009). Induction motor equivalent circuit for dynamic simulation. In *Electric Machines and Drives Conference, 2009. IEMDC '09. IEEE International*, pages 858–863.

Ferreira, F. J. T. E., Fong, J., and De Almeida, A. (2011). Ecoanalysis of variable-speed drives for flow regulation in pumping systems. *Industrial Electronics, IEEE Transactions on*, 58(6):2117–2125.

Jeong, H.-S. and Kim, H.-E. (2007). A novel performance model given by the physical dimensions of hydraulic axial piston motors: Experimental analysis. *Journal of Mechanical Science and Technology*, 21(4):630–641.

Lovrec, D., Kastrevc, M., and Hribernik, D. (2005). Comparison of variable electrohydraulic supply systems by example of pressure control. *Ventil*, 11(3):153–160.

Lovrec, D., Kastrevc, M., and Ulaga, S. (2009). Electro-hydraulic load sensing with a speed-controlled hydraulic supply system on forming-machines. *The International Journal of Advanced Manufacturing Technology*, 41(11-12):1066–1075.

Lovrec, D. and Ulaga, S. (2007). Pressure control in hydraulic systems with variable or constant pumps? *Experimental Techniques*, 31(2):33–41.

Majumdar, S. R. (2000). *Oil Hydraulic Systems: Principles and Maintenance*. McGraw-Hill Education (India) Pvt Limited.

Rydberg, K.-E. (2009). Efficiencies for variable hydraulic pumps and motors  mathematical models and operation conditions.

Shen, Z., Xiong, Y., Cheng, X., Fu, Y., and Kumar, P. (2006). Power mosfet switching loss analysis: A new insight. In *Industry Applications Conference, 2006. 41st IAS Annual Meeting. Conference Record of the 2006 IEEE*, volume 3, pages 1438–1442.

Song, X., Yao, X., Li, Z., and Wang, J. (2008). Equipping variable-speed pumping source to reduce useless power of missile-loaded hydraulic actuator. In *Systems and Control in Aerospace and Astronautics, 2008. ISSCAA 2008. 2nd International Symposium on*, pages 1–5.

Stekl, P. (2007). *3-phase AC Induction Vector Control Drive with Single Shunt Current Sensing*.

Tašner, T. and Lovrec, D. (2011). Comparison of modern electrohydraulic systems. In *34th International Conference on Production Engineering*.

Wilson, E. W. (1949). Performance criteria for positive displacement pumps and fluid motors. In *ASME Semi-annual Meeting*, volume 71.

Xu, M., Chen, G., and Ni, J. (2010). Hierarchical control of energy regulation based variable-speed electrohydraulic drive. In *Information and Automation (ICIA), 2010 IEEE International Conference on*, pages 2401–2405.