Process-driven component adjustment on variable speed pump drives – development of a strategy to increase the overall energy efficiency

Tobias Speicher*, Tobias Thies*, Oliver Kettenhofen** and Jochen Gessat*

htwsaar – Hochschule für Technik und Wirtschaft des Saarlandes, Forschungsgruppe Fluidtechnik, Goebenstr. 40, D-66117 Saarbrücken, Germany*
InnoMa System GmbH, Heinrich-Hertz-Straße 11, D-66773 Schwalbach-Hülzweiler, Germany**
E-Mail: tobias.speicher@htwsaar.de

Regarding the trend of optimizing energy efficiency and meeting upcoming regulations of energy consumption there are many ways to refine existing hydraulic drive systems. To gain more knowledge about components, combinations of those components and their interaction with the overall process, a combination of measurement, simulation and calculation of energy consumption is used to build the foundation for finding optimization approaches regarding the efficiency of electro-hydraulic pump drives. This is a three-step process focusing on the following topics: increased component efficiency, matching pump drive components and adjusted process layouts. By utilizing this strategy, a manufacturer- and customer-dedicated optimization of pump drive systems can be realized.

Keywords: Variable Speed Pump Drives, Energy Efficiency, Design Strategy, Gear Pumps
Target audience: Design Process, Displacement Units, Systems

1 Introduction

Ever-expanding commodity prices and energy costs fan the flames of the trend towards energy efficient process solutions and increased efficiencies of drive systems. The emphatic demand for CO_2 savings and the optimal utilization of energy in technical applications are already part of legally required guidelines. For manufacturing companies, energy costs are a significant part of their operating costs. Customers of, e.g. machine tools, therefore show intensified interest in power-saving solutions for their machines. The manufacturers of hydraulic drive technologies are hereby obliged to provide drive technologies for energy efficient realization of functions like feed motion, clamping operation et cetera.

In this context, novel system architectures for hydraulic drives gain increasing significance. Valve-controlled systems, as a matter of principle lossy due to their energy losses at the cross sections of the valves, are replaced by variable speed pump drives to allow a needs-based use of the system /1/. Possible savings in energy consumption, compared to conventional hydraulic drives, are estimated to about 70 to 80%, what generates significant cost advantages for the user and simultaneously reduces the ecological damage /2/, /3/.

First systems were introduced in applications such as forming, cutting and pressing processes /4/ as well as in forging or injection molding machines /5/, /6/. In the meantime, almost every manufacturer of hydraulic systems offers variable speed pump drives.

Rapid progress in the electrical drive technology have therefore promoted the fast market presence of electro-hydraulic motor-pump-units (MPU). However, there is no continuous development but an assembly of existing components more or less suitable for the individual application. Usually, this "Bottom-Up" approach leads to oversized systems and therefore misses the target of an energetic and economic optimum.

The presented work addresses this issue by providing a process-driven „Top-Down"-approach with all tools necessary. It allows more efficient process solutions that are also quantifiable and therefore even permit economic and eco-friendly decisions.

There are two possible objectives in the optimization of the energy efficiency of variable speed pump drives. For one thing, especially for the developer of a pump drive system, an increased efficiency of that drive in general is needed. On the other hand, an individual optimization regarding a specific application can be the goal. In both cases, the optimization strategy, as shown in *Figure 1*, is based on the three complementary approaches:

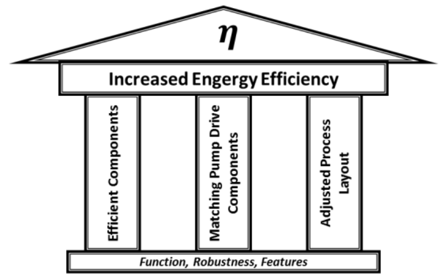


Figure 1: The three steps of optimization

In general, the system efficiency can be optimized by using components with fewer losses. In addition, there is also interest in the optimization of existing components especially for their use in applications with variable speed pumps – in particular to further reduce their losses. All these measures are combined under the topic "Efficient components".

Beyond this fundamental requirement to use components as efficient as possible, also the best possible interaction of all pump drive components has to be ensured. An unpropitious combination can shift the point of operation of one component to an unfavorable range, causing an unnecessary decrease of the system efficiency. "Matching pump drive components" therefore is the second approach of optimization.

Finally, no optimization is complete without the inclusion of the process – the systems requirements to the MPU define its points of operation. Vice versa, it is also wise to fit the systems requirements to the strengths and weaknesses of the pump drive. This third approach is the "Adjusted process layout".

2 Gear Pumps

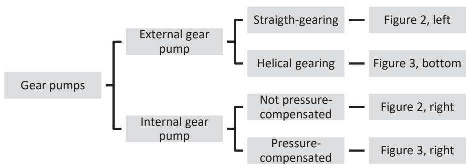
2.1 Design Types

Geared machines transform energy with losses, which, as stated by the DIN ISO 4391 /7/, divide into two main types of losses. On the one hand, there are volumetric losses, which are caused by spacing, sealing gaps and pressure- or temperature induced deformation of material. On the other hand, there are hydro-mechanical losses that are caused by friction of moving parts and viscous friction as well as compressibility of the fluid.

Type of losses	Definition	Sum of losses for a hydraulic pump
Volumetric losses	$\eta_{vol} = \frac{Q_{eff}}{Q_{theo}} = \frac{Q_{eff}}{n \cdot V_{geom}}$	$\eta_{pump} = \eta_{vol} \cdot \eta_{hm} = \frac{Q_{eff}}{n \cdot V_{geom}} \cdot \frac{\Delta p \cdot V_{geom}}{2 \cdot \pi \cdot M_{eff}} = \frac{Q_{eff} \cdot \Delta p}{M_{eff} \cdot \omega}$
Hydro-mechanical losses	$\eta_{hm} = \frac{M_{theo}}{M_{eff}} = \frac{\Delta p \cdot V_{geom}}{2 \cdot \pi \cdot M_{eff}}$	

Table 1: Losses of a hydraulic pump as defined by DIN ISO 4391

Gear pumps, within the group of fixed-displacement pumps, are preferably used mainly due to their cost advantage over other types and are divided by their design into external gear pumps and internal gear pumps.



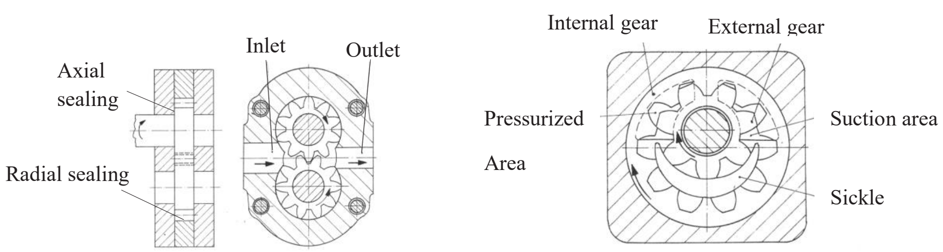
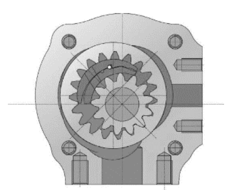


Figure 2: Unifilar drawing of an external and an internal gear pump

In case of this study, further pump designs are taken into consideration. On the one hand a pressure-compensated internal gear pump. Its design is based on a divided sickle, which reacts to pressure by reducing the spacing between the internal- and external gear and thus minimizing volumetric losses. On the other hand, a helical geared pump that is constructed to reduce flow pulsation and noise emission.



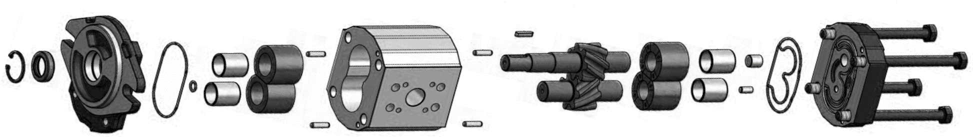


Figure 3: Sectional drawing of a pressure compensated IGP /8/ and exploded view of a helical geared EGP /9/

2.2 Temperature influence

Type and temperature of the oil used must be taken into consideration whilst measuring hydraulic pumps, because of its influence on the pump efficiency. The change of the pump efficiency for the usage of HLP 46 and a reduction of the oil temperature by 10°C from T_1=45°C to T_1=35°C is used as an example and depicted in *Figure 4.*

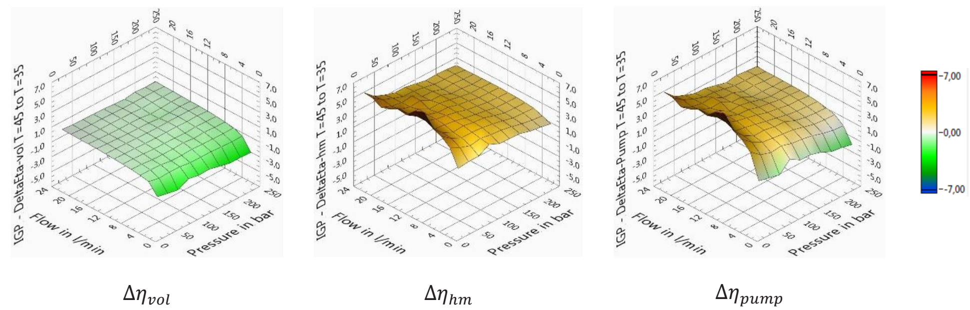


Figure 4: Changing of pump efficiency $\Delta\eta = \eta(T_1) - \eta(T_2)$ for a decreasing fluid temperature (T_1=45°C to T_2=35°C)

Whilst the oil temperature is reduced, its viscosity rises. Thus, the internal leakage of the pump is reduced whereby its volumetric efficiency improves. However, simultaneously the friction of the pump increases whereby its hydro-mechanical efficiency drops. In consequence, the overall efficiency of the pump increases as well as decreases depending on its point of operation.

Dynamic viscosity	Density
$\mu = \mu_0 \cdot e^{-\lambda_1} \cdot (T - T_0)$	$\Delta\rho = -\beta_T \cdot \rho_1 \cdot (T_2 - T_1)$

Table 2: Dynamic viscosity and change of fluid density in dependence of temperature change

The reason for temperature dependence of the efficiency can be examined by considering the definitions of density and viscosity of hydraulic fluids as shown in *Table 2*. A change in the density of a fluid depends on a change in oil temperature. While the viscosity is dependent on an operating- and reference temperature.

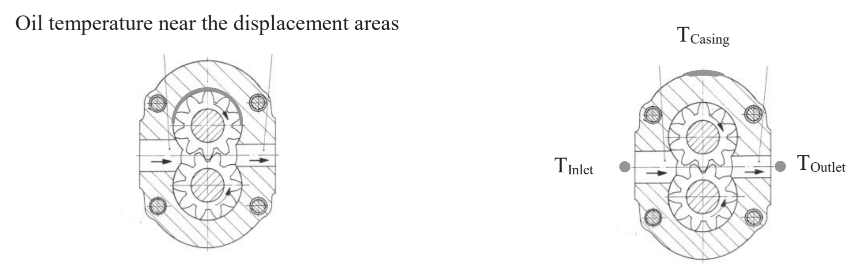


Figure 5a: Ideal sensor placement *Figure 5b: Actual sensor placement*

To implement temperature monitoring into the measurement of pumps, temperature sensors are needed. *Figure 5a* shows an ideal sensor placement, as near to the displacement chambers of the pump as possible. Due to this problematical execution, a more applicable solution of a configuration for temperature sensors is used as shown in *Figure 5b.*

Further details and usage of the temperature sensors for the work as described in this paper follow in the affiliated discussion.

3 Experiments

The pump studies are based on the standards defined by the ISO 4409 /11/ and /12/. Exemplary, those are used as guidance to establish the actual displacement of a pump and measure pump- and system efficiency, as well as regulations on how to depict such measurement results.

3.1 Test layout and properties

On the one hand, test pumps are examined regarding their efficiency and on the other hand regarding their leakage behavior. Measurements of the efficiency produce data for the degree of energy conversion with regard to the overall operating range of the pump and the measuring unit. Boundaries of the operating range are defined by either the pump or the measurement unit as maximum pressure, volume flow, torque as well as temperature values are exceeded. The measurement of the leakage behavior serves as mean to establish the internal leakage whilst building up pressure against a closed tube end. Here internal leakage is the flow volume of oil pressed through sealing gaps from the pressurized area to the suction area.

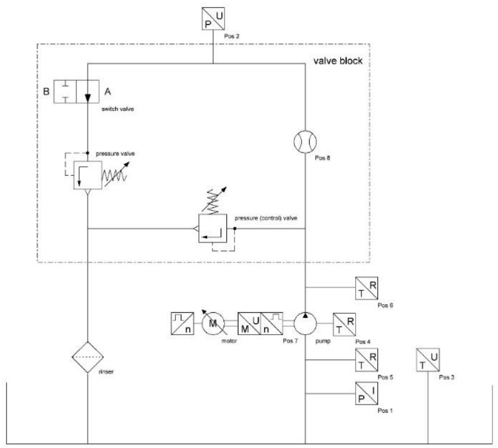


Figure 6: Schematics of pump measurements

Type of sensor	Pos.
Suction pressure	1
System pressure	2
Temperature (oil tank)	3
Temperature (casing)	4
Oil temperature (suction area)	5
Oil temperature (pressurized area)	6
Torque Rotational speed	7
Volume flow	8

Table 3: Overview of Sensors

Figure 6 depicts the schematics of pump measurements and the positioning of the used data sensors (cf. *Table 3*) within the system layout.

The switch valve is used to choose between leakage measurement and efficiency measurement. Switch setting A (*open*) enables the circular flow needed for performing the efficiency measurement while switch setting B (*closed*) closes the circuit to provide the setup needed to run the leakage measurement. The pressure relief valve protects the measurement unit against excess pressure. *Figure 7* depicts the configuration of the temperature sensors applied to the test pumps. Here the Temperature of the casing, of the oil in the suction area and in the pressurized area as well as the temperature of the oil tank are monitored.

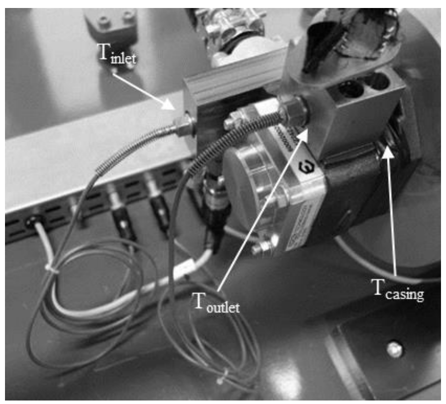


Figure 7: Arrangement of temperature measurement

3.2 Test process and analysis

In order to consider the influence of the system temperature on the efficiency of a pump, a concatenation of efficiency- and leakage behavior measurements, starting at room temperature (approx. 25°C) of the oil- and casing temperature and ending at typical operating temperature of pump drives (approx. 40-45°C), is used to collect data at gradually increasing temperature levels.

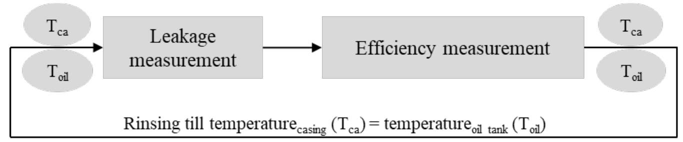
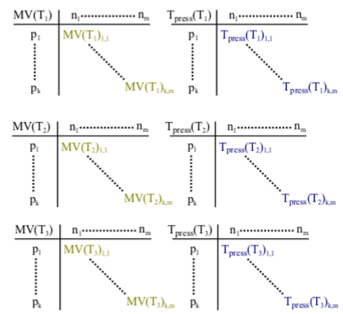


Figure 8: Cycle of handle whilst measurement

Measurement data is interpolated over a minimum of three sets of data for different temperature levels. For each measurement value at a given point of operation (a specific rotational speed and pressure), this interpolation is executed to calculate the value at the desired temperature resulting in a new data set for the desired temperature.



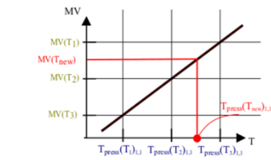


Figure 9b: Interpolation of measurement data

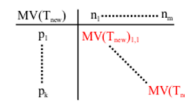
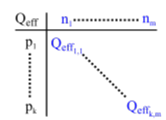


Figure 9a: Measurement data for temperature T_1 to T_3 *Figure 9c: Measurement data for temperature T_{new}*

The measurement system operates speed-controlled thus delivering speed-depending measurement data (cf. *Figure 10b*). Concerning energy flow studies volume flow-depending data is more sensible (This way measurement data is shaped correspondingly to the hydraulic power used as a reference value for energy flow studies) (cf. *Figure 10d*). To process data accordingly, actual volume flow values (cf. *Figure 10a*) are transformed into actual rotational speed values (cf. *Figure 10c*), which in turn are used to convert speed-depending into volume flow-depending data.



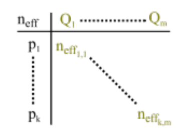
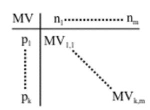


Figure 10a: Measurement data effective volume flow *Figure 10c: Calculated data effective rotational speed*



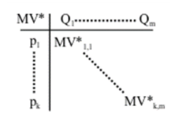


Figure 10b: Measurement data in regard to rotational speed *Figure 10d: Measurement data in regard to volume flow*

All data used in the following is generated and converted (in regard to temperature and volume flow) like this and serves as foundation for the subsequent researches.

4 Optimization Strategy and Results

4.1 Energy – Flow - Analysis

The introduced optimization strategy for variable speed pump drives focusses on the maximization of energy efficiency by minimizing the losses and by best possible adjustment of system and process.

The center of this strategy is the measurement and in-depth analysis of the energy flow of motor-pump-units from its electrical supply to the hydraulic process – from source to function (cf. *Figure 11*).

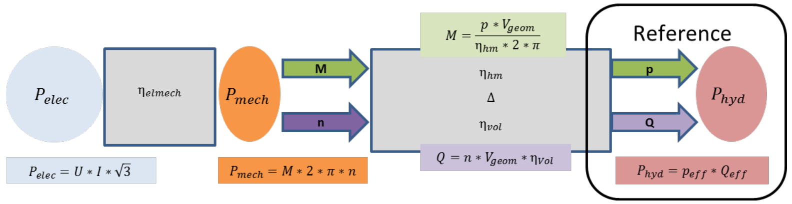


Figure 11: Flow of energy in an electro-hydraulic pump drive

The energy flow helps identifying the strengths and weaknesses of the components, analyzing interdependencies between the MPU's components and the interaction with the process.

This analysis allows the direct comparison of pump drives, enables the search for preferred combinations and, together with detailed modelling of the process driven, also its optimization.

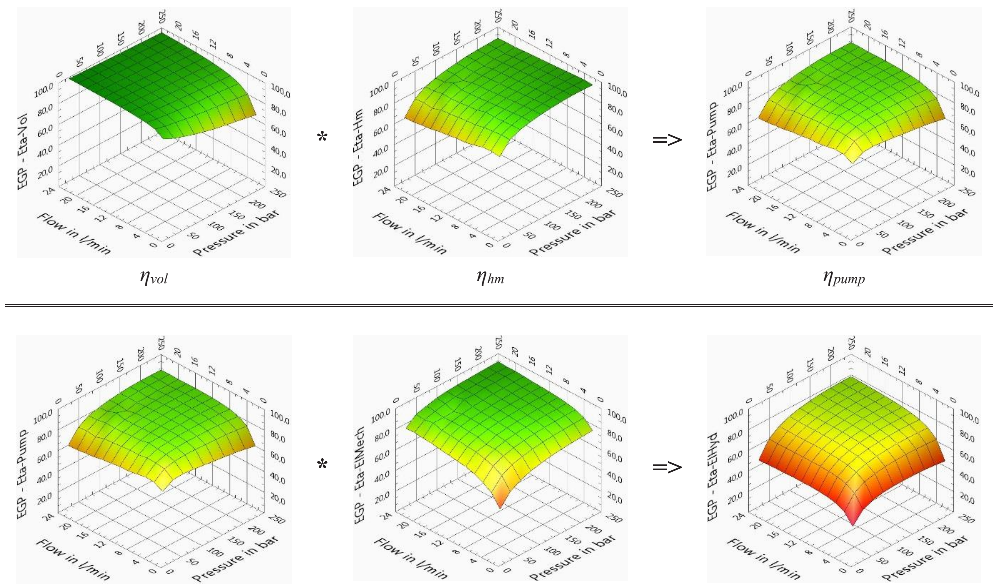


Figure 12: Overview of the partial efficiencies of a variable speed pump drive with electrical motor

Here, the hydraulic power available for the process always represents the reference point and is depicted in relation to its defining values, effective pressure and effective flow. Additional parameter with influence to the flow of energy, such as type of fluid or fluid temperature, are also considered – they are integrated in the efficiencies of the components (*Figure 12*).

4.2 Optimization Approaches

4.2.1 Efficient components

Starting point of each optimization is the selection of components as lossless as possible. By direct comparison, the efficiency of competing components can be analyzed and the best option can be chosen. Basis for the comparability is that the reference is fixed for all comparative figures and for all components and systems. Therefore, each operation point is set by the effective pressure and the effective flow, values that are defined by the process and not pump drive specific. Such a comparison for two different types of internal gear pumps is shown in *Figure 14*.

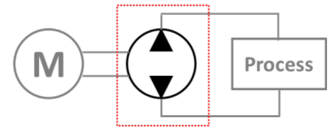


Figure 13: Best pump efficiency

IGP-A is an internal gear pump with fixed sickle, IGP-B a gap compensated internal gear pump with divided sickle and pierced internal gear (principle by Eckerle). For the volumetric (a), hydro-mechanical (b) and overall pump efficiency (c), *Figure 14* displays the differences between the two pump types. As expected, the volumetric efficiency clearly shows the influence of the gap compensation. Especially when there is only minimal flow and a high load is applied, the volumetric efficiency (cf. *Figure 14a*) of IGP-B is up to 23% better than the efficiency of IGP-A.

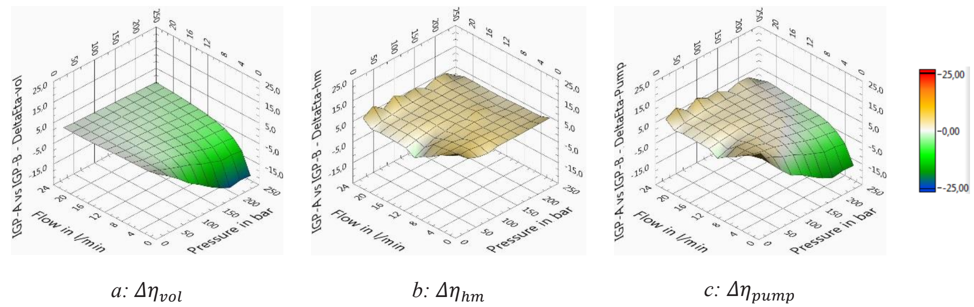


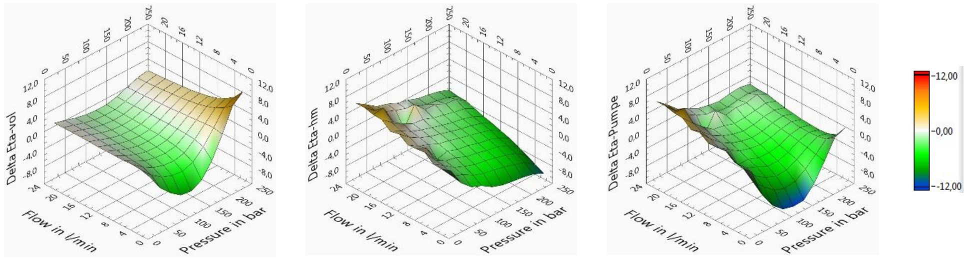
Figure 14: Comparison of pump efficiencies for two different types of internal gear pumps (IGP-A: pump with fixed sickle; IGP-B: pump with compensation by divided sickle – principle by Eckerle) @ T_{Fluid} = 40°C

For the hydro-mechanical efficiency (*Figure 14b*), this situation is completely inverted. Here, for the whole range of performance, IGP-A is more efficient than IGP-B due to more power losses that are caused by the increased internal friction that is produced by the gap compensation.

Finally, the overall pump efficiency (*Figure 14c*), the product of volumetric and hydro-mechanical efficiency, shows divided results. Pump IGP-B has, because of less internal leakage, the better efficiency for operating points with high loads and rather low flow. Because of the increased friction, for all the other operating points, IGP-B is less efficient than IGP-A.

In addition to their help with the search for the least lossy component, the comparisons provide the basis for detailed analysis of the energy loss mechanisms or the advantages and disadvantages of certain pump or motor types.

A similar comparison, this time for two external gear pumps with different functional principle, is shown in *Figure 15*. One pump has straight gearing (EGP-straight) and the other has helical gearing (EGP-helical). Again, all three characterizing efficiencies are displayed. Regarding the volumetric efficiency (*Figure 15a)*, for the most part, the pump with helical gearing shows less leakage than the pump with straight gearing and therefore better efficiency. Nevertheless, it stands out that the helical geared design has, in the range of moderate pressures of about 100bar to 125bar, increased leakage. Up toward maximum pressure, the leakage decreases again and the efficiency is again good, up to 10% better than with the straight gearing. Unlike with the internal gear pumps, the hydro-mechanical efficiency is not completely in favor for one specific pump type. At high flow and with minimal load, the helical gearing provides less friction and a better efficiency. For all other operating points, the efficiency of the pump with straight gearing is superior (*Figure 15b).*



a: $\Delta\eta_{vol}$ b: $\Delta\eta_{hm}$ c: $\Delta\eta_{pump}$

Figure 15: Comparison of pump efficiencies for two different types of external gear pumps (EGP-Helical: pump with helical gearing; EGP-Straight: pump with straight gearing) @ T_{Fluid} = 40°C

Regarding the overall pump efficiency (*Figure 15c)*, the EGP-straight shows up to 12% better values, especially for low flow. Only for high flow level and very low load, the helical system provides an up to 10% better efficiency.

Exceeding the analysis of available components, there is an interest in further optimizing these components especially regarding their use in applications with variable speed pump drives – which in the first place means reducing their losses. For this purpose, this analysis can also provide initial hints by what measures such an optimized construction can be realized.

4.2.2 Matching pump drive components

After the optimization of single components, the next step is to do the same thing for the whole pump drive unit. Therefore, also the electrical system and its losses is taken into consideration to enable the comparison of systems to each other, as it is shown in *Figure 17.* The energy flow oriented approach again allows the analysis of the strengths and weaknesses of specific combinations and helps to find the optimal system.

In *Figure 17*, the comparison of the two external gear pumps is revived (*Figure 17a)* and correspondingly supplemented. The influence of the electrical components, motor and servo amplifier, to the system efficiency is shown in *Figure 17b.*

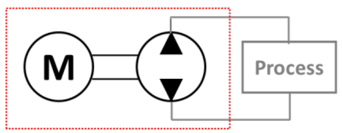
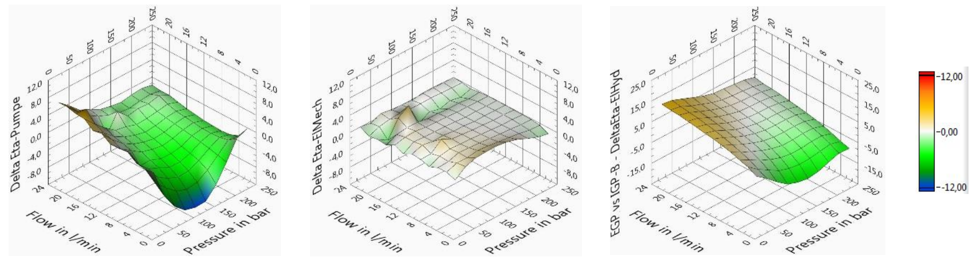


Figure 16: Matching components

As might be expected, this influence is only minimal. The electro-mechanical efficiency is depending on the rotational speed of the drive shaft and the torque applied. If both pumps were ideal machines without losses, there would be the same rotational speed and torque for both combinations for a specific operating point. This would lead to identical electro-mechanical efficiencies. Caused by the volumetric and hydro-mechanical losses of the pumps, the rotational speed and torque needed to provide flow and pressure are increased. This shift is defined by the pump efficiencies and therefore different for each pump. These, in most cases small, differences effect a slight shift of the motors operating point, thus also changed efficiencies as shown in *Figure 17b.*

In this case, the result is a slightly higher electro-mechanical efficiency for the system with the helical gearing. This indicates that the motor operating point, which is moved to a higher rotational speed and torque due to the less efficient pump (EGP-helical), is a more efficient one.

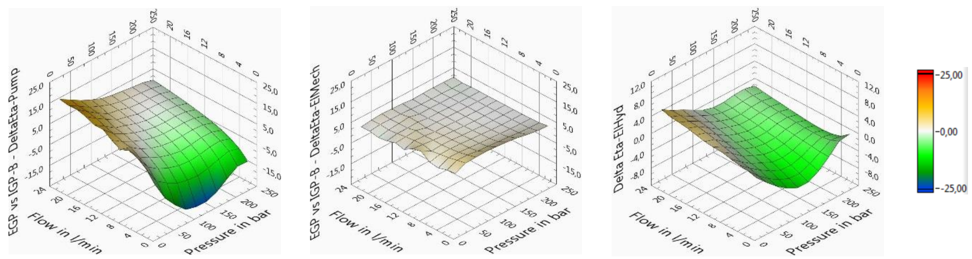
The overall efficiency of the system, consisting of pump, motor and servo amplifier, is shown in *Figure 17c.*



a: $\Delta\eta_{pump}$ b: $\Delta\eta_{elmech}$ c: $\Delta\eta_{elhyd}$

Figure 17: Comparison of efficiencies for two different types of pump drives (EGP-Helical: external gear pump with helical gearing; EGP-Straight: external gear pump with straight gearing)

Another, similar, comparison is shown in *Figure 18*. The system with the helical external gear pump is compared to a system with a gap compensated internal gear pump, both applied to the same electrical motor and amplifier. Again, both pumps and systems show significant differences in their efficiencies.



a: $\Delta\eta_{pump}$ b: $\Delta\eta_{elmech}$ c: $\Delta\eta_{elhyd}$

Figure 18: Comparison of efficiencies for two different types of pump drives (EGP-Helical: external gear pump with helical gearing; IGP-B: pump with compensation by divided sickle – principle by Eckerle)

The external gear pump with helical gearing has a better efficiency than the internal gear pump when high flow is requested but is less efficient when high loads are applied (*Figure 18a*). The influence of motor and amplifier, again, is only small (*Figure 18b*), there is an advantage in combination with the external gear pump of about 3%. Finally, the overall efficiency of the pump drive shows two clearly distinguishable ranges. The left corner of *Figure 18c)* shows better efficiencies for the EGP-system, for all other operating points, the system with the internal gear pump is preferable.

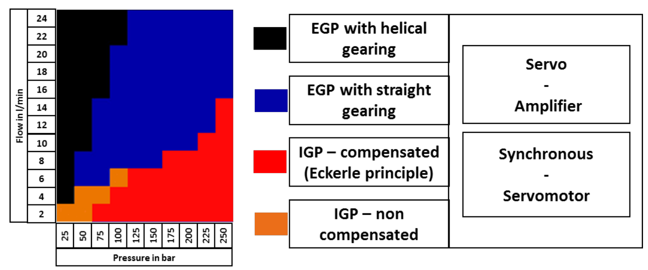


Figure 19: System with highest efficiency – comparison of four combinations

Besides these comparisons of two systems, which aim at a detailed analysis, it is also possible and of interest to compare a whole bunch of systems with different pump or motor principles or from several manufacturers. *Figure 19* shows the results of such a comparison for a pump unit that provides up to 24 l/min at a pressure of up to 250bar.

The system is electrically driven by a servo-drive and four different gear pumps are applied. In *Figure 19*, for every point of operation, represented by pressure and flow, the system with the best efficiency is displayed.

As one can see, for each of the four combinations, there are operating points with best efficiency. The systems with internal gear pumps seem to prefer higher loads at only small flow whilst those with external gear pumps show better results especially at high flow. In addition to the information about the best system, it is also important to quantify the potential increase of the efficiency. This potential is also very dependent on the point of operation, as can be seen in *Figure 20.* It shows the differences between the overall efficiency of the best compared to the worst system, in absolute percentages. Especially for small power (little flow and low pressure) and in the range around the nominal power (maximum pressure and high flow), this potential increase in efficiency is only about a few percent.

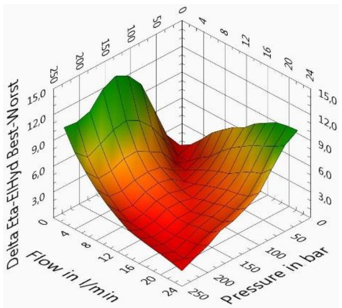


Figure 20: System with highest efficiency – comparison of four combinations

This can be expected because all existing systems are optimized for use at their nominal power and therefore provide high efficiencies. Vice versa, because of physical laws, the percentages of the losses at low power are high which causes less efficiency. However, it also shows that with partial load, one can achieve up to 15% more overall efficiency by selection of the best possible combination.

4.2.3 Adjusted process layout

So far, all conducted measures targeted a general increase of the efficiency of a single component or pump drive. However, for the user or system manufacturer, the efficiency of a pump drive for a certain process is important. As the results of *Figure 19 and Figure 20* show, this massively depends on the operating points and its shares to the process.

In order to take this into account, for the final step of optimization, the process is taken into consideration by specific load cycles. Therefore, the interaction of the pump drive with the driven process is included in the energy flow analysis and as a result, the energy consumption for this process can be calculated.

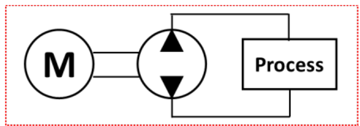


Figure 21: Adjusted process layout

The load cycle, as displayed in *Figure 22a)*, is specified in form of pressure and effective flow as information over the cycle time. The mechanical power needed to perform this hydraulic cycle is then calculated with the pump efficiencies. These include additional influences like the type of the fluid or its temperature. The mechanical power then serves as the reference for the further calculation of the electrical input power needed. An integration of these powers over the load cycle then leads to the energy, which the system consumes to perform one process, as shown in *Figure 22b)*.

The exemplarily load cycle in Figure 22a) describes a positioning and clamping process. Via a cylinder, a work piece is moved to the position where it is clamped for a machining step. Then, the cylinder returns to its initial position and the process can be repeated.

For this process, Figure 22b) describes the energy balance, standardized to the hydraulic energy, which is defined to 100%. The necessary energy at the pump drive shaft (E_{mech}) as well as the electrical energy consumption (E_{elec}) are shown. The differences between the four compared pump drives are significant – at best (*Figure 22b, black*), the system efficiency is 50%, so one has to provide two times the energy that can be used for the process. The worst system even consumes 220% of the hydraulic energy, which means only 45% efficiency (*Figure 22b, yellow*). Thus, the combination using the external gear pump with helical gearing, (*Figure 22b, black*), closely followed by the system using the EGP with straight gearing, are the most efficient options for this specific process.

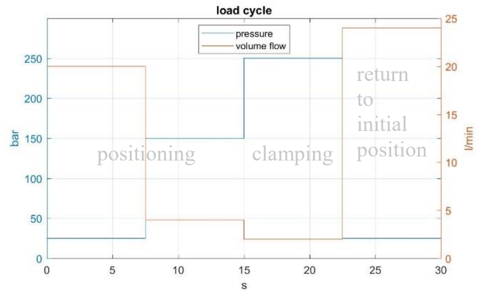


Figure 22a: Exemplary load cycle

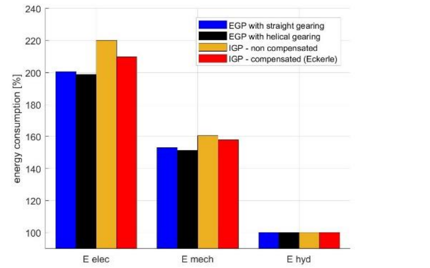


Figure 22b: Electric and mechanic energy to be deployed in relation to the extractable hydraulic energy

Furthermore, comparing the mechanical and ele

äctrical energy absorption, it shows that in this specific application, the losses split up in equal parts to the pump and the electric drive.

An even higher efficiency than only by choosing the best suitable pump unit can be achieved by also adapting and optimizing the process itself. Such an adjustment is applied to the load cycle in *Figure 23*. This takes place by a variation of the positioning (0s to 15s in *Figure 22a and Figure 23a*).

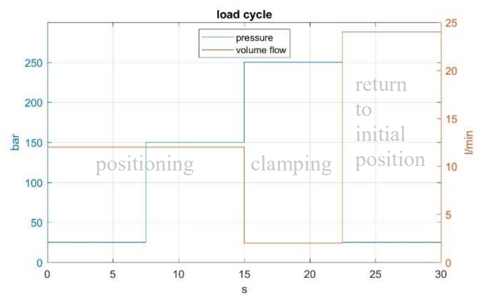


Figure 23a: Adapted load cycle

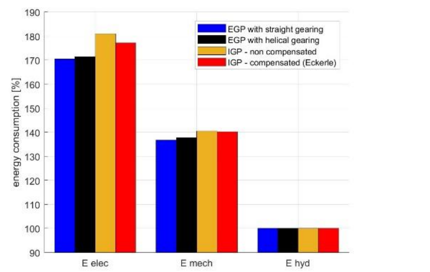


Figure 23b: Electric and mechanic energy to be deployed in relation to the extractable hydraulic energy - Optimized

Instead of a quick extension of the cylinder until it reaches the work part and then a slow approach to the clamping position, both parts are carried out with constant speed and therefore, also constant flow. The time needed to perform, as well as the hydraulic energy for one cycle, do not change due to this measure, but the system efficiency can massively be increased. As it is shown in *Figure 23b*, the electric energy consumption now is merely between 170% (best) and 182% (worst) of the energy available in the hydraulic circuit, meaning an increase in the efficiency up to 59% for the best combination. Furthermore, the best combination has now changed to the system using the EGP with straight gearing (*Figure 23b, blue*).

5 Summary, Conclusion and Outlook

This paper explains how energy analysis, based on detailed measurements, can be used to increase the energy efficiency of hydraulic variable speed drive systems. The optimization strategy therefore follows three approaches simultaneously – efficient components, matching pump drive components and adjusted process layout – each having a potential to enhance the energy efficiency. Depending on the optimization goal, the strategy is fitted accordingly, resulting in the usage of a single or a combination of the above stated approaches. However, the greatest possible impact is achieved by taking single components, the drive system as well as the overall process into consideration.

For the examples given in this paper, the overall system efficiency is increased by up to 10 %.

Current studies are restricted to gear pumps but will be expanded and include piston pumps and variable displacement pumps in order to generate an overall market overview and to exploit the maximum gain of efficiency.

Volumetric modelling of gear pumps is used to investigate possible changes of the pump design beneficial to raising the component efficiency. Though the development effort has to be rated as high compared to the increase in efficiency that can possibly be achieved. Therefore, the focus of the current investigations is placed on system optimization by best possible adjustment of pump drive and process.

Apart from a mere optimization toward the best possible efficiency, topics like robustness of the system, its lifetime and controllability, etc. must be taken into consideration when constructing or refining drive systems as well as the whole process.

Finally, acquisition- and maintenance costs will also influence the calculation of the total cost of ownership. The question if such a system with an increased efficiency justifies potential additional costs has also to be taken into consideration.

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GEFÖRDERT VOM
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für Bildung
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Nomenclature

Variable	*Description*	*Unit*	*Variable*	*Description*	*Unit*
I	Current	[A]	T_1	Temperature (status one)	[°C]
M	Torque	[Nm]	T_2	Temperature (status two∂)	[°C]
M_{eff}	Actual torque	[Nm]	T_{press}	Oil temperature (press. area)	[°C]
M_{theo}	Theoretical torque	[Nm]	U	Voltage	[V]
MV	Measured value	[-]	V_{geom}	Theoretical displacement	[cm³]
n	Rotational speed	[rpm]	β_T	Coefficient of thermal expansion	[K⁻¹]
n_{eff}	Actual rotational speed	[rpm]	η_{elmech}	Electromechanical efficiency	[%]
p	pressure	[bar]	η_{elhyd}	Overall efficiency	[%]
Δp	Pressure difference	[bar]	η_{hm}	Hydro mechanical efficiency	[%]
p_{eff}	Actual pressure	[bar]	η_{pump}	Overall pump efficiency	[%]
P_{elec}	Electrical power	[W]	η_{vol}	Volumetric efficiency	[%]
P_{hyd}	Hydraulic power	[W]	μ	Dynamic viscosity	[Nsm⁻²
P_{mech}	Mechanical power	[W]	μ_0	Dynamic viscosity at reference temperature T_0	[Nsm⁻²
Q	Volume flow	[l/min]	λ_1	Viscosity-temperature-coefficient	[K⁻¹]
Q_{eff}	Actual volume flow	[l/min]	$\Delta\rho$	Change in density	[kg/m³
Q_{theo}	Theoretical volume flow	[l/min]	ρ_1	Density at reference temperature one	[kg/m³
T	Temperature	[°C]	ω	Angular velocity	[s⁻¹]
T_0	Reference temperature	[°C]			

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