Simulation-Based Investigation of the Energy Efficiency of Hydraulic Deep Drawing Presses

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Abstract: Hydraulic deep drawing presses are widely used for industrial sheet metal forming today. Small manufacturers of drawn parts and suppliers of the automotive industry especially appreciate these machines because of their high flexibility in process design. But the energy efficiency of modern hydraulic presses is nearly unknown due to a lack of experimental investigations and suited simulation models. The authors' objective is to reduce this gap by analyzing the energy efficiency and to generate the basic knowledge that is necessary for being able to improve the press hydraulics systematically. This article focuses especially on the simulation works.

Keywords: hydraulic press, energy efficiency

1 Introduction

Hydraulic deep-drawing presses are very flexible machines that can be used for a wide range of different tasks in sheet metal forming. Especially when small part series and drawn parts with complex geometrical shape are produced, hydraulic presses are the first choice against their electro-mechanical opposites. Presses are normally customized to the user's demands. Thus, there is a big variety of machine and drive structures. *Figure 2* shows a typical example structure. The slide drive, which is mounted in the head piece, moves the upper part of the forming tool and performs the forming motion. For production presses the slide drive is mostly pump controlled. Single-action machines have also a die cushion, which is mounted below the table. It is necessary for deep drawing and allows controlling the material flow by applying an adjustable clamping force on the flange of the drawn part. The die cushion drive is normally valve controlled. Additionally, there are further hydraulic systems for auxiliary functions, such as cooling.

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Though forming processes have a low specific energy demand compared with other production technologies, the machines and their drives show a certain energy saving potential [1]. During the last decades electric energy was cheap and there were no regulations concerning the energy efficiency of machine tools. This is changing dramatically now. Costs for electricity and their influence on the overall production costs are constantly rising. Additionally, there are plans to supervise the energy efficiency of machine tools by law and technical standards. Examples for this are the directive 2009/125/EC of the European parliament and of the council for "establishing a framework for the setting of ecodesign requirements for energy-related products" and the new ISO 14955 "Machine tools – Environmental evaluation of machine tools".

Several years ago, there were already research activities focusing on the energy efficiency of hydraulic presses [2, 3, 4]. An important conclusion was that there is a strong dependence on the machine settings and on the forming process. But in the meantime, the design of press hydraulics has changed because of the availability of new electro-hydraulic components, digital controllers and software. So, the applicability of the results on modern machines is limited.

In the literature different proposals for new, energy saving hydraulic structures for presses can be found. Some examples shall be given here: In [5] a secondary control of the slide drive and a displacement control for the die cushion were discussed. The topic of [6] was the experimental investigation of three different energy saving hydraulic clamping concepts for the blank holder in single action deep drawing presses. [7] proposes an automated switching concept for

Figure 1. *6300 kN production machine (left), 1600 kN research machine (middle) and 2500 kN research machine (right)*

the slide cylinders as well as a parallel leveling system for the slide, which is force neutral, and an energy saving drive concept for the die cushion. [8] contains different approaches for energy saving hydraulic press drives: For valve controlled machines, the usage of accumulators with different pressure levels is advantageous. Directly driven presses can be improved through reconverting the kinetic energy of the slide, the mechanical work of the die cushion and the hydraulic compression energy into electric energy by displacement control. In [9] the energy recovery from the die cushion by using variable displacement units was discussed. [10] describes a new modular slide drive, which contains a differential cylinder and two speed variable motor-pump-units, each for one cylinder chamber. This reduces significantly the number of valves and the corresponding throttling losses. In [11] the slide drive and the die cushion drive were equipped with speed variable motor-pump-units. Positive effects are improved energy efficiency, higher dynamics, more flexibility and better controllability.

Unfortunately, those papers do not contain sufficient information about the relations between drive structure, machine settings and forming process. Consequently, the general knowledge about the energy efficiency of hydraulic deep drawing presses is only poor. To satisfy the growing demand for experimental and theoretical analyzing methods and results, the machine manufacturer Schuler SMG & Co. KG, the hydraulics specialist Moog GmbH, the automotive supplier MA Automotive Deutschland GmbH and the Institute of Fluid Power at TU Dresden work together in a research project. This is the prerequisite for being able to improve systematically the energy efficiency of hydraulic press drives.

In the first step, machines were experimentally analyzed. Selected results can be found in [12]. The topic of this article is mainly about the simulation based investigations, which have been done in order to analyze the energy efficiency of the press hydraulics in detail and to evaluate the potential of alternative hydraulic structures.

2 Machines and forming processes

2.1 Demonstration machines

Three machines have been analyzed, see *Figure 1*. The first one is part of a line with a total of six presses and produces sheet metal parts for the automotive industry. The other two machines are used within research at the TU Dresden. They are smaller and have a maximum slide force of 1600 kN and 2500 kN. They are not utilized in industrial production making their availability for experiments and parameter studies very good.

Table 1 compares the machines and gives an overview of the most important properties.

Table 1. *Machine properties*

Property	6300 kN production machine	1600 kN research machine	2500 kN research machine
Type	Single action	Single action	Single action
Year of manufacture	2004	1992	2012
Slide (force, stroke)	6300 kN, 1300 mm	1600 kN, 200 mm	2500 kN, 600 mm
Pressing speed	139 mm/s	36 mm/s	41 mm/s
Fast forward speed	700 mm/s	150 mm/s	380 mm/s
Table dimensions	3540 x 2210 mm	800 x 780 mm	1600 x 1300 mm
Die cushion	1500 kN, 250 mm	400 kN, 100 mm	1000 kN, 200 mm
Slide cushion	1000 kN, 200 mm	-	-

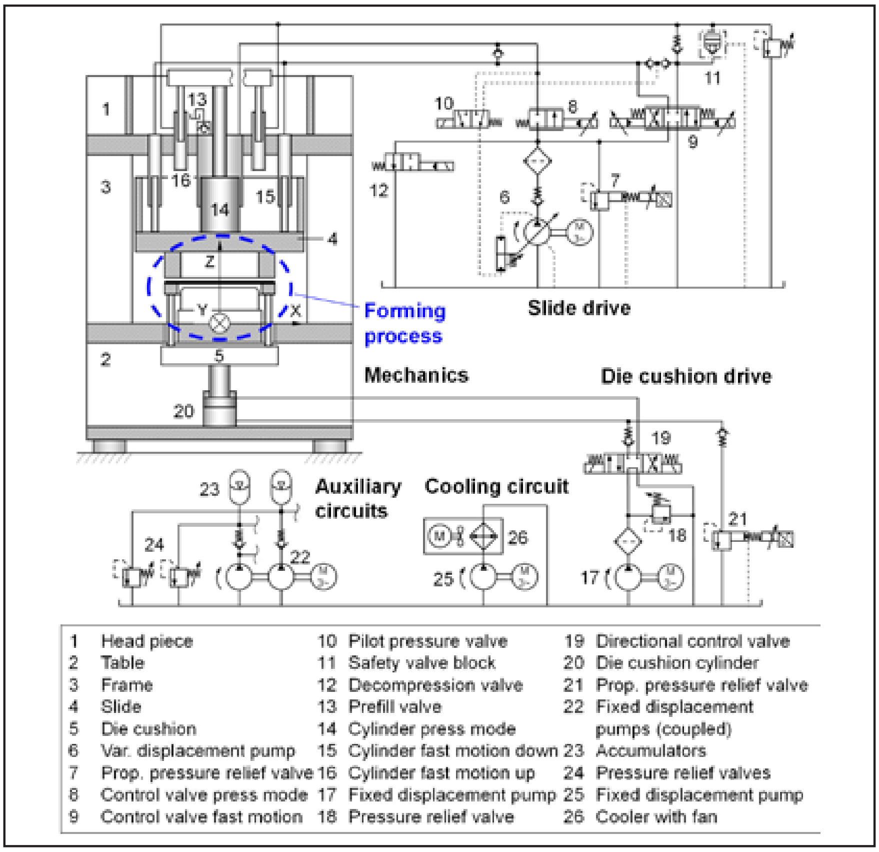

Figure 2. *Simplified structure of the 1600 kN research machine*

The simulation works, which are presented in this article, focus especially on the 1600 kN research machine. So its structure, which is shown in *Figure 2*, shall be explained more in detail.

The head piece (1), table (2), which carries the forming tool, and frame (3) are the non-moving mechanical parts. The slide (4) holds the upper tool and generates the vertical forming motion. The die cushion (5) generates a variable clamping force, which is put on the flange of the sheet metal blank and controls the material flow.

The slide drive is actuated by a pump (6) with a load sensing system. The pilot pressure valve (10) decides if valve (8) or valve (9) will work as a measuring throttle. The proportional pressure relief valve (7) limits the maximum slide pressure. The safety valve block (11) ensures that the slide cannot fall down because of its own weight. The decompression valve (12) inhibits the rise of the pump pressure when the machine is in idle. The prefill valve (13) connects the pressing cylinder (14) with the tank when inactive. Fast motion down is realized with two plunger cylinders (15) while two other cylinders (16) are responsible for the fast motion up.

The die cushion is equipped with a fixed displacement pump (17). The pressure relief valve (18) limits the pump pressure to about 20 bar. Using the directional control valve (19), the die cushion can be moved independently from the slide. The cylinder (20) is the interface between hydraulics and mechanics. A proportional pressure relief valve (21) is responsible for pressure control.

Two fixed displacement pumps for high pressure and low pressure (22) supply the auxiliary circuits with oil. These circuits are necessary for functions like the adjustable end stop of the slide, the cutting damper system and the pilot actuation of valves. The accumulators (23) store energy and satisfy peaks in demand. The pressure relief valves (24) limit the maximum pressure level.

The cooling circuit has a fixed displacement pump (25) and a cooler with an electrical fan (26). All pumps and the fan are driven by asynchronous motors, which are supplied from the three-phase electricity network with 400 V and 50 Hz.

2.2 Forming tools

Typical processes that run on the regarded machines are deep drawing, stretch drawing and cutting. They have completely varying force-stroke-characteristics. In industrial production a forming part is normally produced in steps. The initial drawing process is followed by several cutting and calibration operations. Each one is done with a

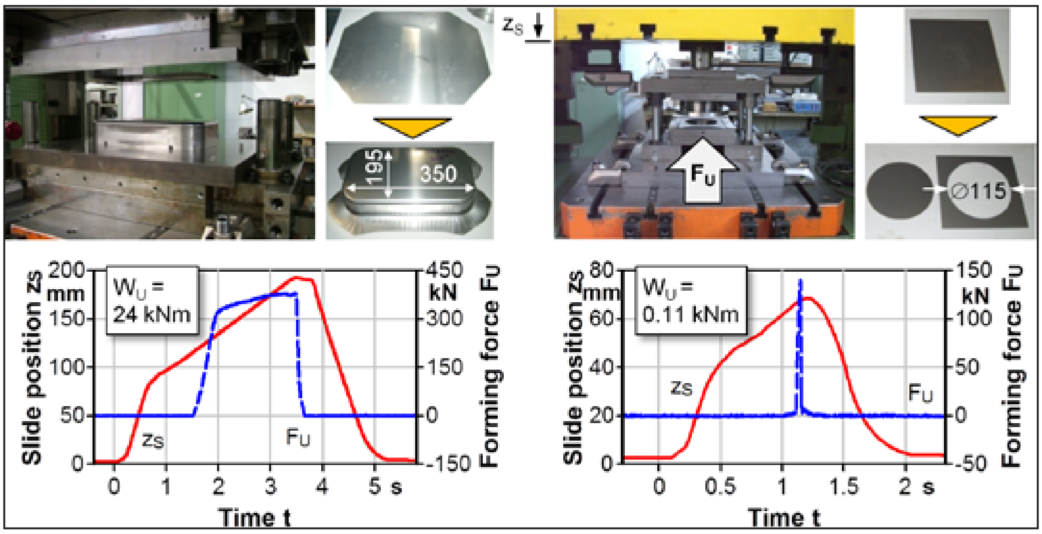

Figure 3. *Drawing tool (left) and cutting tool (right), mounted in 1600 kN research machine*

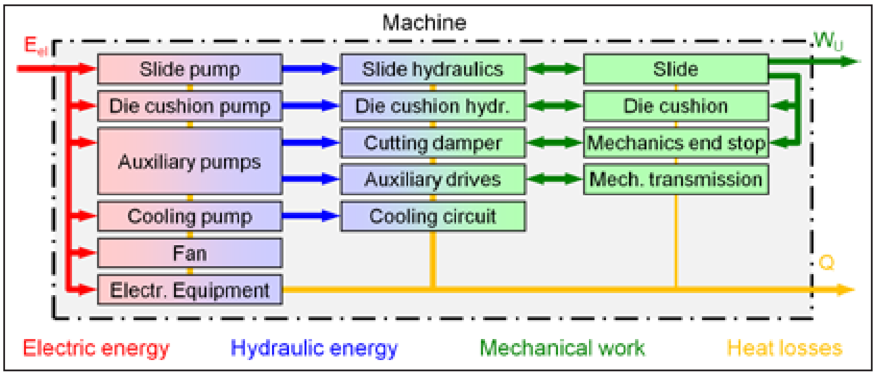

Figure 4. *Energy flow chart for the 1600 kN research machine*

single tool in one press. Typically, a tool set consists of up to six tools.

Both research machines were tested with two different tools, see *Figure 3*. The drawing tool produces a rectangular tub with a maximum height of 100 mm. The cutting tool performs a circular cut. Both are fully symmetrical. For drawing, the forming work W_u is much higher than for cutting. The blank material was the dual phase steel HCT500X (1.0939) with a thickness of 1 mm.

3 Methodology of the investigation

In order to be able to systematically analyze the energy efficiency of a press, the definition of appropriate system boundaries is necessary. *Figure 4* shows the energy flow chart for the 1600 kN research machine.

Motor-pump-units are responsible for the conversion of electric into hydraulic energy. Additional components with electrical supply are the cooling fan and the electrical equipment. The hydraulic energy is conducted through valves, pipes and blocks. During that process, it is partially dissipated into heat. The cylinders are responsible for the conversion of hydraulic energy into mechanical work. Between mechanical parts, which are in relative motion to each other, friction occurs and causes heat losses. To sum this up, it can be said that the electric energy is converted by the machine into forming work W_U and heat Q. Stored energy occurs temporarily, but can be neglected when regarding complete working cycles.

In general, power is the mathematical product of potential variable and flow variable. Hence, it is not possible to directly measure electric, hydraulic and mechanical power. The electric power of polyphase systems can be determined according to DIN 40110-2. The active power $P\Sigma(t)$ in a system with n wires is calculated using phase currents i_μ and virtual star voltages $u_{\mu 0}$ [13]:

$$P_\Sigma(t) = \sum_{\mu=1}^{n} u_{\mu 0} \cdot i_\mu \tag{1}$$

Instead of the latter, it is possible to use the voltages measured against any neutral point. Electric power *Pel*, which is equivalent to the collective active power *P*Σ, is defined as the arithmetic average:

$$P_{el} = \overline{P_\Sigma(t)} \tag{2}$$

The hydraulic power *Phyd* is calculated using pressure p and flow rate Q:

$$P_{hyd} = p \cdot Q \tag{3}$$

The mechanical power *Pmech* can be determined with velocity v and force F:

$$P_{mech} = v \cdot F \tag{4}$$

The energy E respectively the work W is the time-based integral of power P:

$$E = \int_{t_1}^{t_2} P\,dt \tag{6}$$

The efficiency η, which is the quotient of outgoing power *Pout* and incoming power *Pin*, is used to evaluate components and systems in stationary operation:

$$\eta = \frac{P_{out}}{P_{in}} \tag{7}$$

During a press cycle the machine passes different operating points. This fact causes the efficiency to fluctuate permanently. Because of this reason, it is advantageous to analyze the energy efficiency ε, which is the quotient of outgoing energy *Eout* and incoming energy *Ein* of a complete working cycle:

$$\varepsilon = \frac{E_{out}}{E_{in}} \tag{8}$$

The analysis of the energy efficiency of presses should consider all relevant operation modes: standby, idle, tool changing and production.

4 Simulation works

4.1 Modeling strategy

A system simulation model of the 1600 kN research machine was built in order to be able to analyze the energy efficiency in detail. At first, the model should be able to calculate the static and dynamic properties of relevance. The second step was the integration of mathematical descriptions of electric, hydraulic and mechanical losses.

Besides mechanics, hydraulic drive systems and PLC, the model also contains the process in a simplified way. The auxiliary circuits influence significantly the energy efficiency of the plant. Therefore, they have to be incorporated in the simulation. The software tool, which is used, provides predefined model objects for all physical domains of relevance [14].

The mechanical parts of presses show elastic deformations under load and store energy. For being able to decide, which mechanical structures are important for the system model, a preliminary investigation was done. Based on a simplified CAD model, the mechanical parts of press frame, slide and die cushion were statically and dynamically analyzed with an FE tool. A linear-elastic material description was used for the welded steel construction. In the static analysis the maximum load forces were applied. *Figure 5* shows the resulting displacement in *Z*-direction. For better visibility, the deformation is displayed magnified.

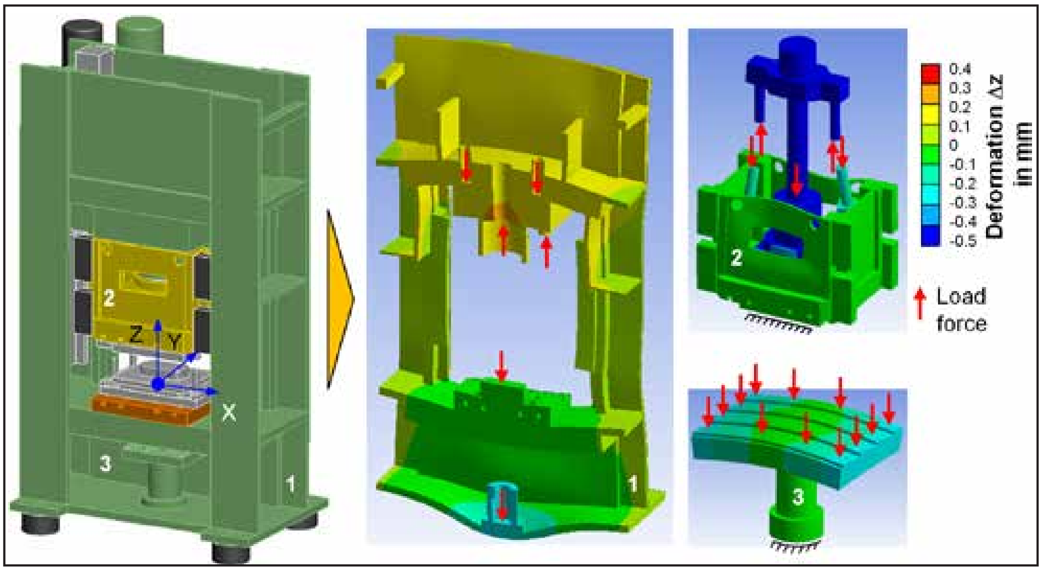

Figure 5. *CAD-model (left) and calculated displacement in Z-direction of frame (middle, cross section), slide (upper right) and die cushion (lower right)*

The stiffness of the mechanics, which can be estimated with the load forces and resulting displacement, is at least four times higher than that of the hydraulic cylinders. The latter depends on the spring rate c_{Oil} of the oil in the cylinder chambers. For one chamber the formula is:

$$c_{Oil}(z) = \frac{K' \cdot A^2}{V(z)} \quad (9)$$

The parameters are the equivalent bulk modulus *K'*, the piston area *A* and the volume *V*.

The dynamic properties of the press mechanics were analyzed with an FE modal analysis. The lowest natural frequencies are at least three times higher in comparison with that of the hydraulics f_h:

$$f_h = \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{c_{Oil}}{m}} \quad (10)$$

The parameter *m* is the overall mass that is connected with the cylinder rod. Based on that result, it is supposed that the mechanical parts play only a minor role. The press frame is modeled rigidly within the holistic machine model. Slide and die cushion are represented by discrete mass elements.

Mostly, the data sheets of electrical and hydraulic components such as motors, pumps and cylinders provide only poor information about the energy efficiency and losses.

Typically, for standard asynchronous motors the efficiencies are given for 50 %, 75 % and 100 % of the nominal load. So a simple loss model is needed. [15] delivers a suitable approach that describes the dependency between power loss in stationary operation *PV* and mechanical power at the motor shaft *P*:

$$P_V = P_b \cdot \left(k_1 + k_2 \cdot \left(\frac{P}{P_b} \right)^2 \right) \quad (11)$$

The parameter *Pb* is the nominal power. The unknown constants *k*1 und *k*2 can be calculated with two values of power loss and mechanical power. The following formula converts the given efficiencies from the data sheet into the power loss:

$$P_V = \frac{1-\eta}{\eta} \cdot P \quad (11)$$

Normally, the losses of hydraulic displacement units are given by the manufacturer in form of characteristic lines. But for the machine simulation it is advantageous to approximate the data with compact and smooth mathematical functions. For the leakage *Q1L* a linear dependency from the system pressure *p*1 was assumed here:

$$Q_{1L} = m_L \cdot p_1 + n_L \quad (12)$$

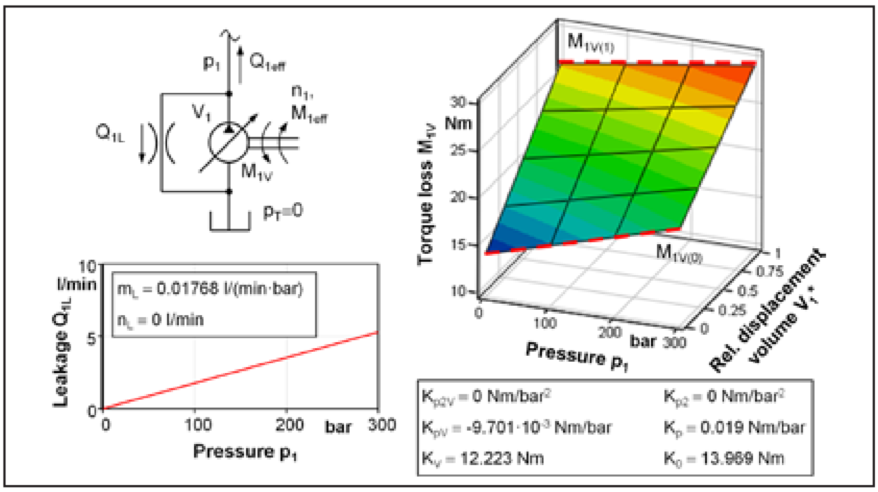

Figure 6. *Leakage (left) and torque loss (right) of the slide pump*

The parameters are the gain *mL* and the offset n_L. The dependency from the rotational speed is negligible due to the nearly constant value. The friction torque M_{1V} was modeled as a function of the pressure *p*1 and of the relative displacement volume *V*1*:

$$M_{1V} = K_{p2V} \cdot p_1^{\,2} \cdot V_1 * + K_{pV} \cdot p_1 \cdot V_1 * + K_V \cdot V_1 * + K_{p2} \cdot p_1^{\,2} + K_p \cdot p_1 + K_0 \qquad (13)$$

The model parameters are named with *K*. For fixed displacement pumps the first three terms on the right hand side can be omitted. *Figure 6* shows exemplarily the losses for the slide pump. The model parameters were determined with the data sheet.

4.2 Selected results

Figure 7 compares exemplarily measurement and simulation for the 1600 kN research machine working in semiautomatic mode with the built-in drawing tool. The oil temperature of the machine is in the range for normal operation.

On the left side, the position, force and power of slide and die cushion drive are displayed. During forming, the mechanical power of the die cushion reaches negative values because the die cushion is displaced, generating a counterforce that acts against the slide motion. The slide has to overcome the forming force as well as the die cushion force.

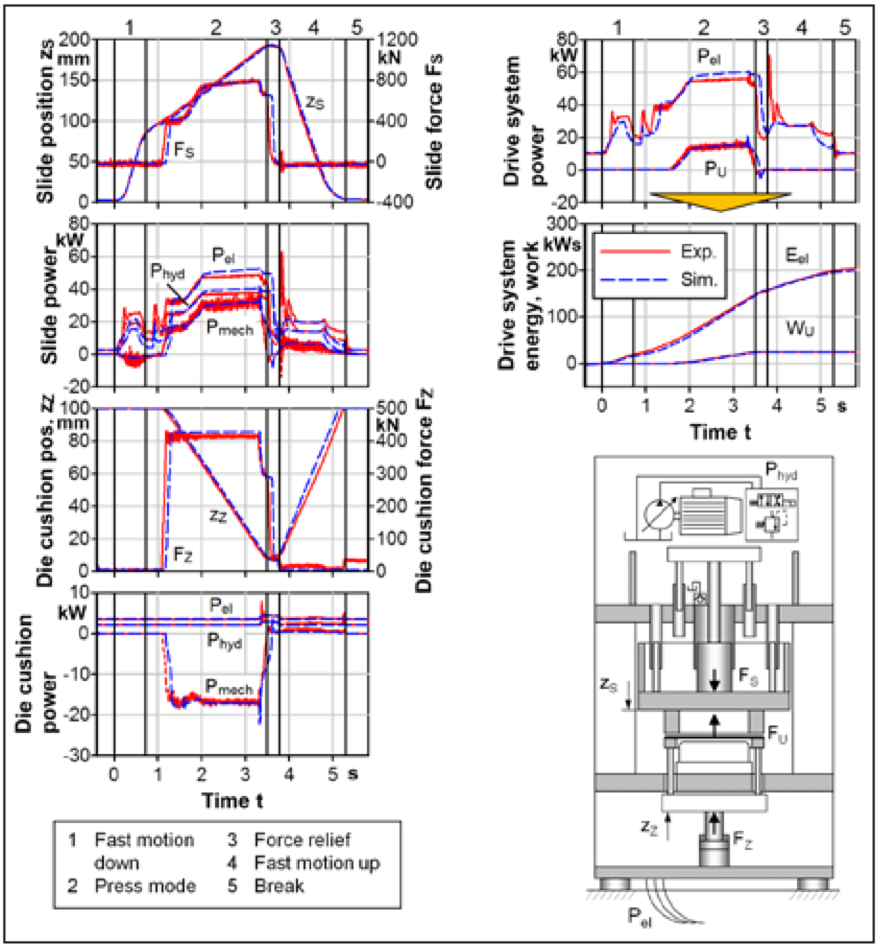

Figure 7. *Model check with time-based data (drawing process, semiautomatic mode)*

On the right side, the power and the energy as well as the mechanical work are shown for the whole machine, beginning with the electrical supply and ending at the forming process. While the process has a power demand during press mode only, electric energy is consumed at all times. When regarding the whole press cycle, which consists of fast motion down, pressing, force relief and fast motion up, the energy efficiency is here ε = 11,8 %.

Besides this example, further simulations were carried out and compared with measurements.

Generally, the simulation is in sufficient accordance with reality.

Figure 8 compares experimental data, based on ten single measurements, and the simulation for the whole press cycle. The machine settings were the same as for Figure 7. The model allows predicting the consumption of electric energy as well as the mechanical work of slide, die cushion and forming process. In

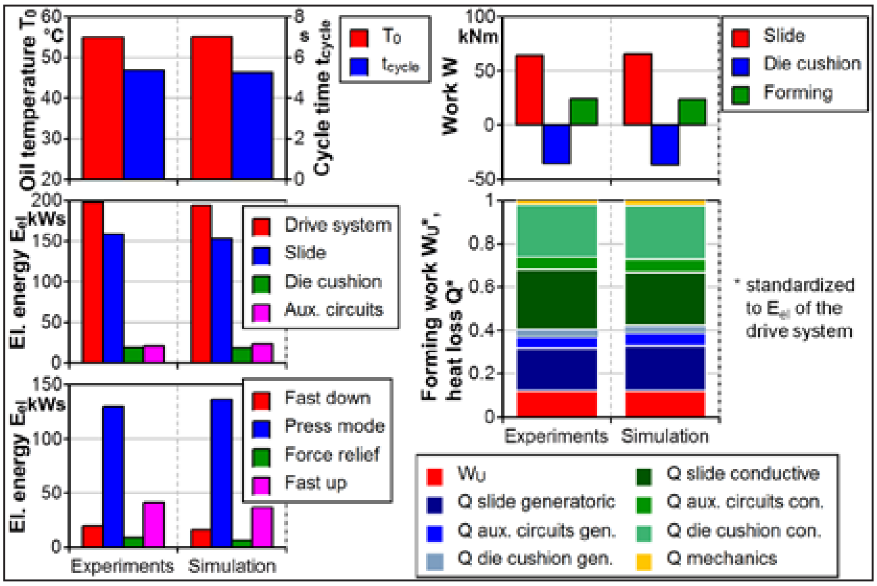

Figure 8. *Model check for whole press cycles (drawing process, semiautomatic mode)*

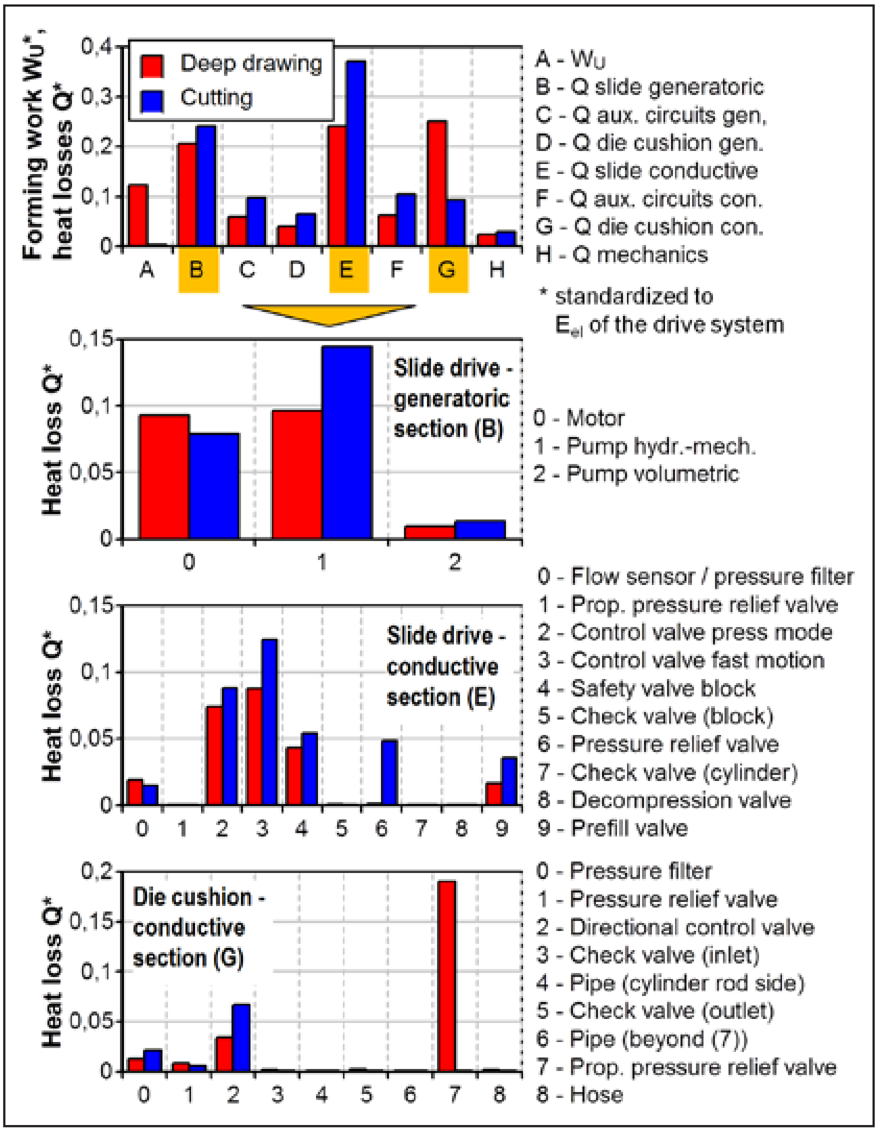

Figure 9. *Results of an extended model based study of the energy efficiency*

comparison with the other drives, the slide drive consumes the largest amount of energy. As expected, the highest energy consumption is reached during press mode. But the fast upwards motion has also certain influence. For the energetic evaluation of the machine the local distribution of the losses is of great interest. With 45...50 % the generatoric and the conductive section of the slide drive are dominant. The losses of the die cushion hydraulics reach a share of about 25 %.

With the simulation model the machine can be analyzed in all operation modes of relevance without further time-consuming experimental investigations. Another advantage of this tool is that it allows a more detailed analysis, since it is not possible to use sensors at every place of interest in the real machine.

Figure 9 contains the results of an extended, model based study of the energy efficiency. The machine is in semiautomatic mode with the velocity set to the maximum value. The slide stroke is 190 mm for the drawing tool and 70 mm for the cutting tool. The first diagram shows the forming work and assigns the heat losses to the substructures of the hydraulic drive system. The charts below visualize exemplarily the component losses inside the most significant subsystems.

The losses in the generatoric section of the slide drive are mainly caused by the asynchronous motor (B0) as well as the hydraulic-mechanical efficiency of the variable displacement pump (B1). An improvement could be reached by using more efficient components. Unfortunately, there are tight physical restrictions for new components with higher efficiencies. The largest losses in the conductive section of the slide drive are caused by the load sensing system, which needs a constant pressure drop at the control valves for press mode (E2) respectively the valve for fast motion (E3). The balancing system uses pressure relief valves (E4, E6), which cause significant losses, to hold the slide in upper position against the gravity force. Through replacing the load sensing control by an electronically controlled pump and recovering the potential energy of the slide, savings of electric energy of about 25 % seem to be realistic.

In the conductive section of the die cushion drive, the directional control valve (G2) causes significant losses as soon as the die cushion pump is switched on. The proportional pressure relief valve (G7), which is responsible for the closed loop control of the die cushion pressure, leads to big losses only for deep drawing processes. With a completely pump controlled die cushion energy savings of about 20 % may be achieved.

Based on these results modified hydraulic structures were implemented and tested within the holistic machine model. For the slide drive an electronically controlled pump, a speed variable motor and a slide balancing system with hydraulic accumulator were tested. The original cylinder configuration of the machine was maintained. The die cushion was virtually equipped with a variable displacement pump for closed loop pressure and position control. Further modifications were a speed variable motor and a low pressure accumulator instead of the tank. The energy efficiency with these modifications depends on machine settings and forming process. But in general, the simulation results support the estimated savings.

5 Conclusion

Although the importance of the energy efficiency of modern hydraulic deep drawing presses is growing, only little information is

available from past research. Hence, systematic technical improvement was previously very difficult. This is the motivation for the presented research activities.
The experimental analysis of machines under the conditions of industrial production was the first step. Based on this, simulation works have been carried out to look more into detail and to test different hydraulic drive concepts. The modeling methodology has been outlined in this paper. Additionally, selected results have been presented.

The next steps will be the further evaluation of hydraulic concepts for presses and the experimental test of some new solutions.

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Nomenclature

A	piston area	mm²
c_{Oil}	spring rate of oil	kN/mm
E	energy	kWs
F	force	kN
f_h	natural frequency of hydraulics	Hz
i_μ	phase current	A
K'	equivalent bulk modulus	bar
K	parameter of pump friction torque	

k	parameter of motor power loss	-
m	mass	kg
m_L	parameter of pump leakage	l/(min·bar)
M_{1V}	friction torque	Nm
n	number of phases	-
n_L	parameter of pump leakage	l/min
P	power	kW
p	pressure	bar
Q	heat, volume flow	kJ, l/min
t	time	s

T_0	oil temperature	°C
$u_{\mu 0}$	virtual star voltage	V
V	volume	l
v	velocity	mm/s
W_U	forming work	kNm
z	position	mm
Δz	deformation	mm
ε	energy efficiency	-
η	efficiency	-

Raziskave energetske učinkovitosti hidravlične stiskalnice za globoki vlek na osnovi numeričnih simulacij

Razširjeni povzetek

Hidravlične stiskalnice se široko uporabljajo za izvedbo globokega vleka pri industrijskem preoblikovanju pločevin. Manjši izdelovalci globoko vlečenih izdelkov iz pločevine ter dobavitelji za avtomobilsko industrijo cenijo uporabo hidravličnih stiskalnic zaradi njihove visoke fleksibilnosti. Energetska učinkovitost sodobnih hidravličnih stiskalnic pa je zaradi pomanjkanja raziskav na tem področju skoraj nepoznana. Glavni cilj tega prispevka je zapolniti vrzel na tem področju z analizo stanja z vidika energetske učinkovitosti in generiranja novih znanj. To pa v nadaljevanju lahko vodi k izboljšavam izkoristkov hidravlične stiskalnice za globoki vlek pločevine. Prispevek se osredotoča predvsem na numerične simulacije delovnih parametrov hidravlične stiskalnice.

Slika 1 prikazuje tri hidravlične stiskalnice za globoki vlek, prva je 6300 kN iz proizvodne linije, druga 1600 kN in tretja 2500 kN sta eksperimentalni stiskalnici z inštituta avtorjev prispevka. Na drugi in tretji stiskalnici so bile izvedene meritve in kasneje izdelan numerični preračun delovanja. V tabeli 1 so prikazani tehnični podatki za omenjene tri hidravlične stiskalnice. *Slika 2* predstavlja poenostavljeno hidravlično shemo in legendo za eksperimentalno stiskalnico z največjo možno silo 1600 kN. Na *sliki 3* sta prikazana fotografija orodja in izdelka (levo zgoraj) ter rezultat meritev med globokim vlekom (levo spodaj). Na desnem delu slike 3 so prikazani orodje in izdelek (desno zgoraj) ter rezultat meritev (desno spodaj) med postopkom rezanja pločevine. *Slika 4* prikazuje diagram energijskega toka za eksperimentalno hidravlično stiskalnico 1600 kN. Tridimenzionalni model in numerično izračunane deformacije ogrodja in pomičnega dela stiskalnice so prikazani na *sliki 5*. Volumetrične in mehansko-hidravlične izgube hidravlične črpalke s spremenljivo iztisnino za pomik orodja stiskalnice so prikazane na *sliki 6*. *Sliki 7* in *8* prikazujeta primerjavo med rezultati meritev in numeričnih simulacij na hidravlični stiskalnici. Upoštevane so vse glavne operacije v procesu preoblikovanja pločevine. Rezultati numeričnih preračunov pri podrobni študiji delovanja hidravlične stiskalnice so prikazani na *sliki 9*.

Avtorja prispevka ugotavljata, da je mogoče na podlagi predstavljene numerične analize napovedati, kakšna bo energetska učinkovitost hidravlične stiskalnice in s tem posredno vplivati na izboljšave.

Ključne besede: hidravlična stiskalnica, energetska učinkovitost

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