Int J Adv Manuf Technol (2009) 41:1066–1075
DOI 10.1007/s00170-008-1553-y

ORIGINAL ARTICLE

# Electro-hydraulic load sensing with a speed-controlled hydraulic supply system on forming-machines

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Received: 29 September 2007 / Accepted: 29 April 2008 / Published online: 5 June 2008


**Abstract** High-power transmission capacity at low-energy losses, low noise emissions, high reliability, and ease of maintenance, as well as convenient cost-effectiveness, are required for industrial applications regarding variable electro-hydraulic supply systems. Only up-to-date principles regarding volumetrically controlled hydraulic energy using variable supply systems and appropriate control strategy have proven satisfactory for these requirements. The aim of the present work was to study the applicability of a low-priced drive concept using a speed-controlled induction motor in combination with a constant-displacement pump applied in a load-sensing control strategy. The suggested approach of the drive concept has been experimentally verified on a prototype of the drive. A hydraulic press-brake used for the machining of casting products in the automotive industry was taken into consideration.

**Keywords** Metal-forming machines · Electro-hydraulic system · Load sensing · Speed-controlled constant pump · Experiment

## 1 Introduction

Reductions in energy consumption and noise reduction regarding drive systems, as well as cost-effectiveness, are increasingly important factors in modern machinery design. All of the above requirements are especially important for machines equipped with hydraulic supplies and drive systems.

High-power transmission capacity at low-energy losses and cost-effectiveness are required for industrial applications using hydraulic supply systems. Only the principle of volumetrically controlled hydraulic energy (variable supply systems e.g., with the use of variable pumps), has proven satisfactory for the above requirements. In this regard, electro-hydraulic solutions, in particular, make it possible to use all the advantages of modern electrical signal transmission and controller designs.

Such variable supply systems can be, in principle, controlled hydraulically or electrically. Constant-speed induction motors in combination with a variable displacement volume pump is a commonly used solution in order to control flow, or, consequently, the pressure of the medium. The second concept, for the same purpose, representing the application of a constant displacement volume pump in combination with variable rotational speed motors, has recently gained significance in practice due to its attractive price. Such systems provide several advantages such as economical use of energy, user-friendly process control, and ease of maintenance, when compared to hydraulic-mechanical solutions.

In particular, electro-hydraulic variable-speed constant pumps have recently been the object of some fundamental research work (see e.g., [1–3]) and work describing early applications of novel control principles (e.g., for use in plastic injection molding machines [4, 5]). The speed-controlled pump drive concept is still an attractive subject for research [6–8].

However, in practice, state-of-the-art hydraulic power stations are often equipped with pumps controlled by

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hydraulic-mechanically controlled devices. One of the reasons for such a situation is the custom of using traditional drive concepts in spite of the known advantages of a fixed displacement pump vs. variable pumps (simple design, reliability, low noise, cost effectiveness, etc.). Commonly mentioned prejudices for the application of speed-controlled constant pumps are the additional cost of frequency converters and digital control equipment. This reproach is incompetent because of the decreasing prices of the necessary electrical equipment, its reliability, and user-friendliness. In addition, it provides important features such as the integration of the controller into the converter, ease of controller parameterization, ease of monitoring, etc. Those digital devices and electrical sensors necessary for operating the frequency converter can also be used to significantly improve system dynamics—intelligent electro-hydraulic systems. Digital control devices enable tuning of the control dynamics to the motor's limits and thus help to maximize system performance and allow adaptation of the real load conditions of the machine's work cycles—load sensing control strategy.

In order to meet the high standards regarding the steady-state and dynamic performance of machinery, designers can (if they are aware of all the above-mentioned advantages) redesign and update existing control solutions. While redesigning the drive system from a hydraulic-mechanical system to a system using speed-controlled constant pump, certain questions of the designer often arise, including: "What are the characteristic properties and comparative advantages or disadvantages of the speed-controlled constant pump concept, particularly in regard to dynamic behavior?" and "Is it possible to replace an existing hydraulic supply system powered by a hydraulic-mechanically controlled variable pump, with a speed-controlled constant pump? The main concern is the high inertia of electro motors' rotational parts, compared to the swash-plate of a variable pump.

In order to answer the above questions, the case of an industrial press-brake was considered, as used for cast product finishing in the automotive industry. Part of the hydraulic scheme of the discussed hydraulic press brake with its existing hydraulic-mechanically controlled axial piston pump needs to be replaced, as shown in Fig. 1.

The aim of this paper is to study the applicability of a speed-controlled induction motor in combination with a constant displacement pump driven within the energy-saving load-sensing drive concept, where the pressure is adapted to the actual loads in the system. The suggested drive concept was experimentally verified on a prototype of a machine drive.

In order to achieve this goal, it was first necessary to clarify the current state, working characteristics, and machine parameters, and to define the working phases and operational profile of the machine. The operational process of the machine was then examined by an equivalent prototype of the drive using a speed-controlled induction motor and constant pump applied within energy saving load-sensing control strategy.

## 2 Working profile of hydraulic press brake

A specially designed test rig was built as described in Sect. 3 in order to examine the applicability of the speed-controlled induction motor in combination with a constant displacement pump in the field of forming machines or presses. In order to carry out the intended task, it was necessary to establish a real working-profile of the machine and its pressure characteristics over the whole working cycle. Measured pressure profiles describe the actual dynamics of the existing hydraulic supply system and actuator. The pump-pressure profile of the most dynamically demanding machining was chosen. The associated pressure profiles and corresponding machining phases are presented in Fig. 2.

There are two pressure values presented in Fig. 2, cylinder pressure ($p_{cyl}$) and pump pressure ($p_1$). Cylinder pressure provides information regarding the progress of work and corresponding forces on the cylinder. The pump-pressure profile is decisive for providing the required system's dynamics. The pump has to supply all subsystems of the machine and not just the work tool driven by the cylinder. It has to react to all load changes such as the activities of turning, fixing, and ejection devices. The acquired pump-pressure profile presented in Fig. 2 results from supplying all the above-mentioned subsystem activities. The presented pump-pressure profile $p_1$ was used as a starting point for further studies.

The acquired pump-pressure profile has been simplified in the sense that only dynamically have the most demanding pressure changes been reproduced in the working profile. All the important time parameters and pressure amplitudes have been preserved: start times and end of pressure change (e.g., switching ON/OFF of the additional load), duration of pressure change, pressure amplitude, etc. Only the dynamically unimportant phases have been neglected. In addition, a step-change in pressure amplitude was applied representing the dynamics of the most demanding case for testing the control concept's dynamics.

A dynamically comparable pressure profile was generated that was achieved by using a speed-controlled constant pump in load-sensing control concept. The simplified profile is presented in Fig. 3.

A simplified pressure profile was generated using a proportional pressure valve (Fig. 4), controlled using Visual Designer software. All other control and regulation func-

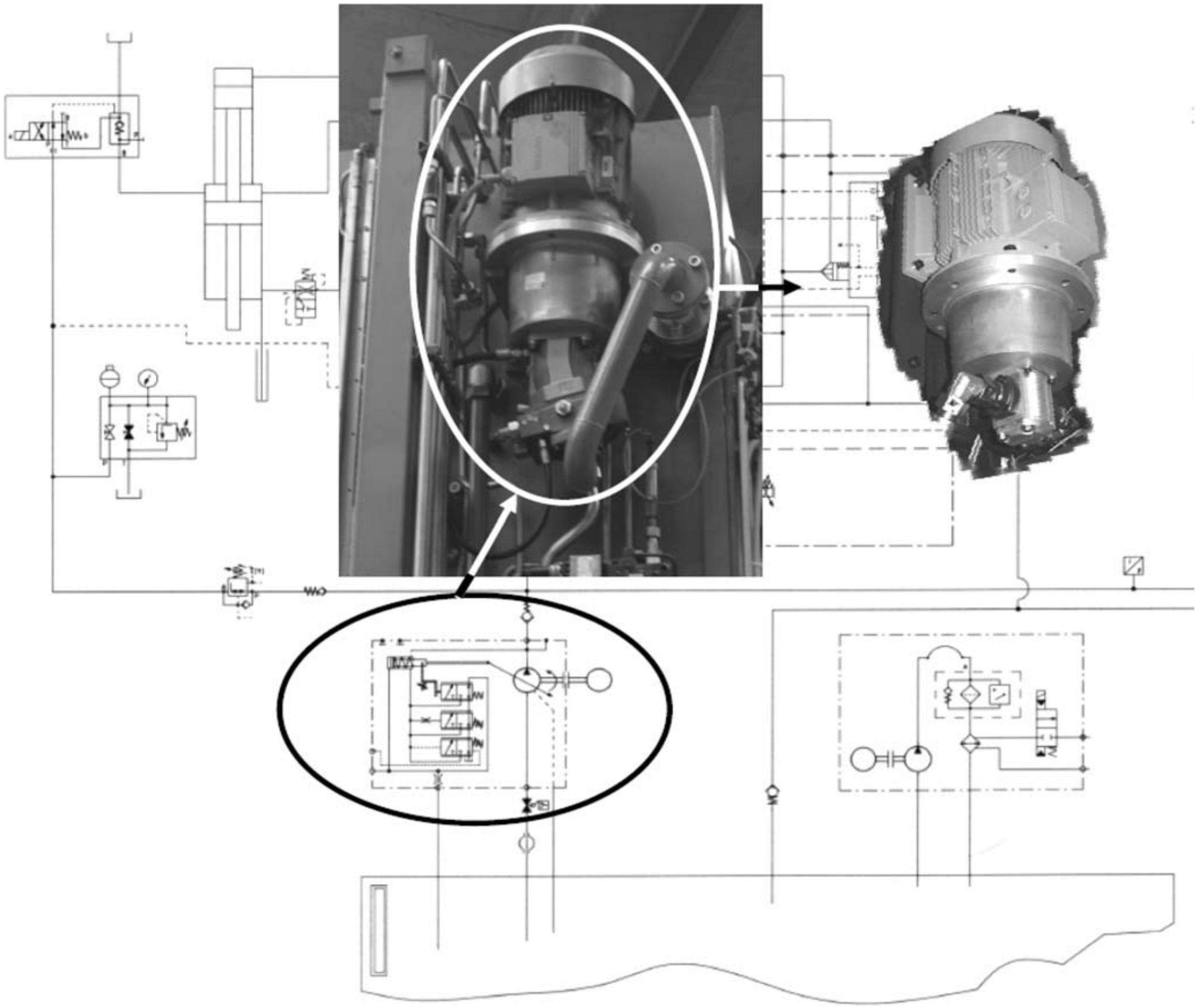


Fig. 1 Existing hydraulic-mechanically controlled pump

tions of the process were also generated within the same environment. Visual Designer is a software tool that serves as a programming tool in the field of data acquisition, control and process management, and testing and condition monitoring. Visual Designer has been used to generate a reference pressure profile, pump pressure, and EM speed data acquisition. Volumetric flow of the pump and consumed EM power was further calculated from the acquired data. Visual Designer has also been used for visualizing results.

## 3 Test-rig design

A special test-rig, as presented in Fig. 4, was designed for realistic simulation of a real situation on the machine. The drive capacity of the existing power unit was considered. A selection of those components to be used on the suggested drive concept was introduced for the test-rig. Low-cost industrial components available on the market were used intentionally to build the rig.

The existing hydraulic-mechanically controlled axial piston pump was replaced by a constant internal gear pump (PGF3 31/032RE07VE4, Bosch-Rexroth) driven by a 15-kW speed-controlled induction motor, which was to be used for the new drive concept. The electric motor is driven by an appropriate frequency converter (Midimaster vector 6SE32, Siemens) [12]. The pump supplies a hydraulic pipeline system of equivalent length and dimensions (comparable hydraulic capacity and inductivity [9]), and corresponds to the existing pipeline system of the press-brake.

Overloading of the system is prevented by an additional pressure-relief valve. The remaining components of the

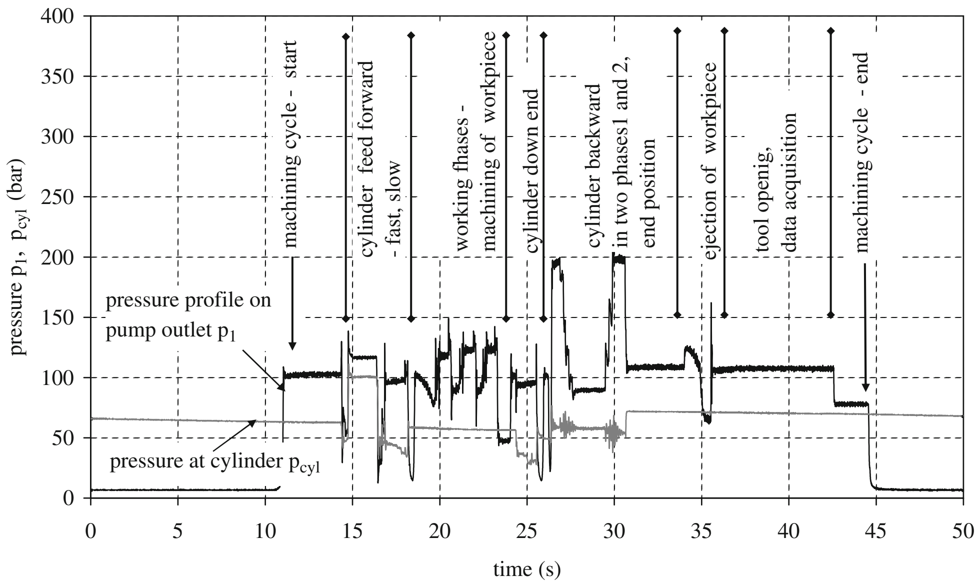



**Fig. 2** Pressure profile over the working-cycle, with working phases

test-rig are the loading unit, pressure sensors, and the control and signal-acquisition system. The loading unit consists of two throttle valves $TV_1$ and $TV_2$ which, used together with a proportional pressure valve, make up the load, hydraulic cylinder of the press. The components of the loading units enable realistic simulation of the real system's workloads. Disturbances of the control, such as switching on/off the additional actuators, was simulated by a directional-valve mounted in front of the throttle valve $TV_2$, enabling an almost instant change in hydraulic flow. Two additional pressure sensors were applied at the pump outlet port ($p_1$), and at the end of the pipeline system ($p_2$), for pressure variation acquisition. All the devices needed for control of the constant pump and variable-speed electric motor (setting of control structure, controllers, etc.) were integrated into the frequency converter. The setting of reference values, data acquisition, monitoring, and graphical interfacing were performed using a personal computer.

## 4 Load-sensing control concept

There are two main principles of hydraulic energy control: by throttling the fluid flow between the generator part (pump) and the motor part (actuator) using a valve - throttling principle, or by using a hydraulic pump with adjustable swept-volume - principle of volumetric control. Advantageous high dynamics, substantial energy loss due to throttling and, consequently, poor running efficiency, are characteristic of the first principle. An inferior dynamic

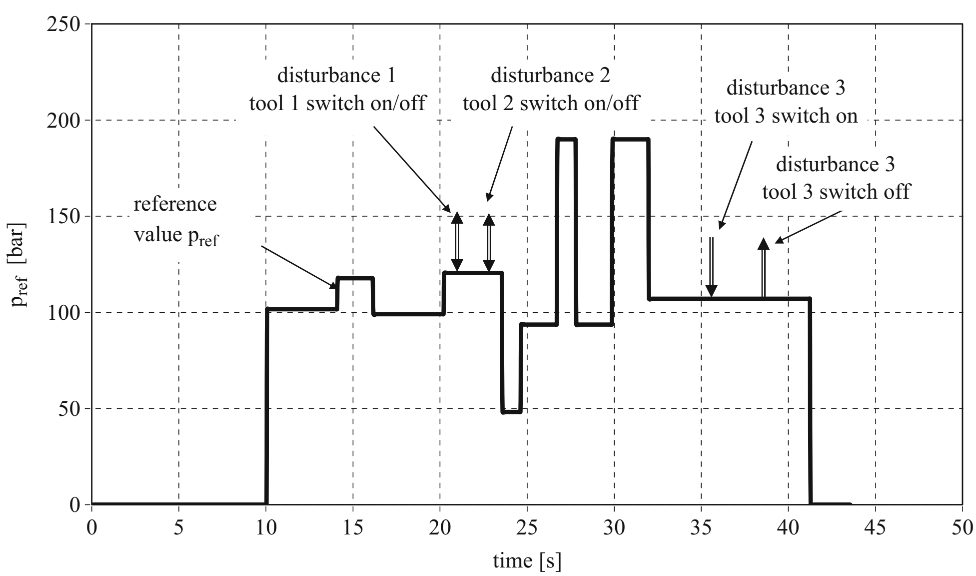
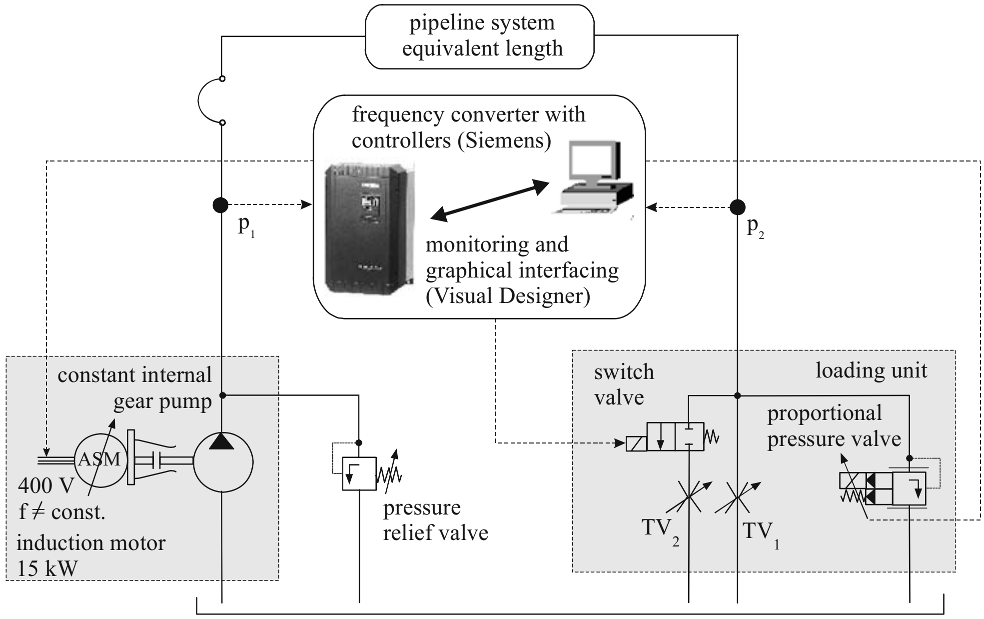



**Fig. 3** Simplified pressure profile

**Fig. 4** Structure of the test-rig



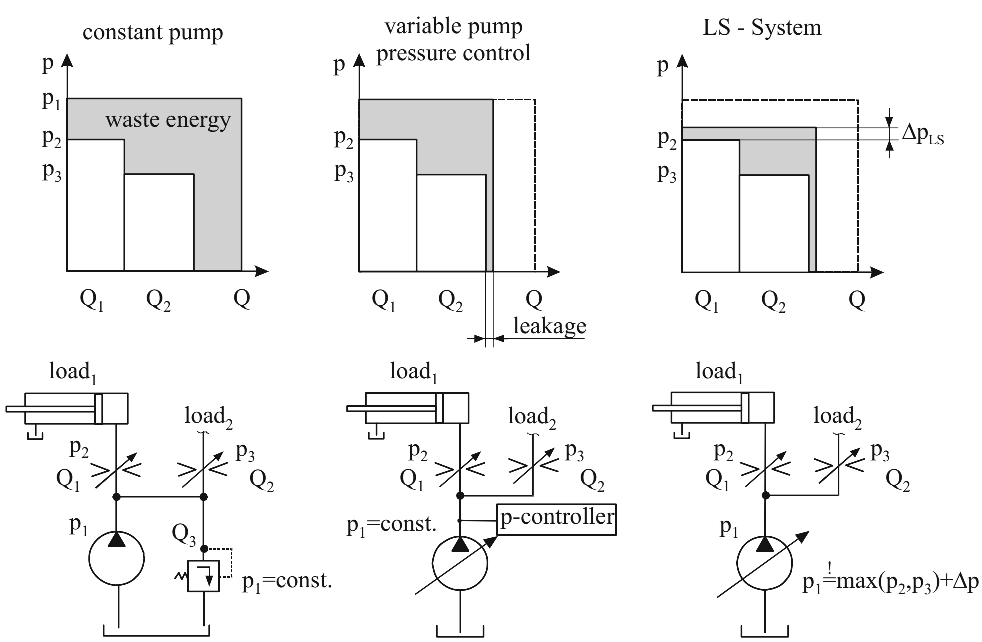

behavior but substantially better power efficiency rate can be expected for the principle of volumetric control.

The load-sensing (LS) concept represents a combination of both the above principles advantageous properties. When supplying a number of independent valve-controlled actuators with different loads with using a single pump, within the load-sensing concept, the supply pressure is adjusted to suit the actuator's highest pressure value. A comparison between the used and wasted powers for different supply principles is shown in Fig. 5.

When using a constant pump (Fig. 5, left), the pressure $p_1$ set on the pressure relief valve must always be set higher than the highest required pressure regarding applied load, likewise the volumetric flow of the pump must be great enough to supply all actuators. When pressure at the control valves, due to throttling, is reduced from $p_1$ to $p_2$ or $p_3$ respectively, it leads to energy losses - pressure dependent losses. Additional losses are conditioned by the fact that pumps usually supply more than is needed for the actuators. Superfluous fluid is drained through the pressure relief valve. Some energy is due to the throttling transformed into heat - flow dependent energy losses. It not only leads to higher energy consumption but, amongst other things, to the necessity of installing larger pumps and more elaborate cooling equipment.

If a variable pump is used, for example a hydraulic-mechanically-controlled axial pistons pump with swash

**Fig. 5** Comparison of used and wasted powers for different supply principles



plate, as in the case of the discussed forming machine, it is possible to adjust the flow and, consequently, reduce the energy losses of (Fig. 5, middle). In its function as a pressure control pump, this variable pump delivers only the amount available to the desired constant supply pressure.

In a load-sensing system, the supply pressure is variable. It changes to accommodate the highest load pressure. It lies at a value $\Delta p_{LS}$ above the highest load pressure. When the LS- system with variable pump is used, the pressure and flow-dependent losses are reduced (Fig. 5, right). Pressure is measured at the most loaded actuator $p_2$, while the pump is controlled according to the pressure increased by a certain $\Delta p_{LS}$ (actual pump control pressure is: $p2 + \Delta p_{LS}$), needed to cover the losses in a hydraulic pipeline system. The use of a variable displacement pump during a load-sensing operation presents the most economical energy supply type for several valve-controlled actuators.

The hydraulic-mechanical design of LS (HMLS) has been well known for some time within the field of mobile hydraulics as a driving concept with high-running efficiency. The HMLS design has certain drawbacks due to its mainly hydraulic-mechanical signal-flow, as applied in such systems. The following are restrictions in the design of the hydraulic system, unfavorable dynamic behavior, oscillations and unstable operation, and a limited choice of controllers that cannot easily be avoided by the introduction of load sensing.

This problem can be solved by using an electro-hydraulic design with electronically performed signal transmission. Electro-hydraulic load sensing (EHLS) offers possibilities for improving the dynamics of a valve-controlled actuator in load sensing, by using control-engineering measures that enable high dynamics of signal transmission and more freedom regarding system design and weight reduction. An EHLS system was realized using a speed-controlled electric motor, and a constant pump, as a variable supply system, as shown in Fig. 6.

Electronic pressure sensors, electrically controlled valves and process electronics are needed to create an electro-hydraulic load-sensing system. Actuators are usually controlled by proportional valves; so a pressure-sensor needs to be added to each actuator. Flow-control, and consequently pressure is carried out electro-hydraulically. Electro-hydraulically controlled (relatively expensive) variable pumps are often used for this purpose. The suggested solution with variable speed-constant pumps enables lower price and robust design. Sensor-signal processing, control of actuators, and pressure-control are carried out by process electronics or computer.

In the case of HMLS with more valve-controlled actuators, the highest pressure can be determined by selection valves. Pressure compensators need to be used for each actuator. They provide a constant pressure level via a control valve. In this way, the volume flow to the actuator with lower load is only dependent on valve spool displacement and not on the present load pressure. It enables preservation of constant flow and constant-speed [10].

In the case of EHLS, the entire control is realized electronically, and consequently, some conventional hydraulic components are redundant because their function can be performed by existing valve or signal correction.

Pressure signals are collected by pressure sensors and lead to control electronics. It executes suitable control strategy and sends, in our case, control signals to a frequency converter and the valves of different actuators. As mentioned before, it is necessary in the case of simultaneously operation, to distinguish between the most loaded and other less loaded actuators. In the case of LS concept, control of the pump always precedes the pressure of the most loaded actuator. Pressure drop over the valve that controls the most loaded actuator is controlled with a variable rotational speed motor and constant pump to a certain value of $\Delta p_{LS}$ (according to practical experience and literature [4] typical value of $\Delta p_{LS}$ = 20 bar, which is enough to cover losses in the circuit and on applied proportional valves).

A simplified hydraulic scheme of a control system for the most loaded actuator and associated signal flow is presented in Fig. 7. The required values are system-pressure (pump pressure) $p_1$, actuator-pressure at the inlet side $p_{load}$ and control-signal $J$, which is forwarded (as required by the speed of the actuator) to the corresponding control valve. The control values of the process are the control voltage of the frequency converter $U_P$, and the voltage of the control valve of actuator $U_V$. The pressure difference between

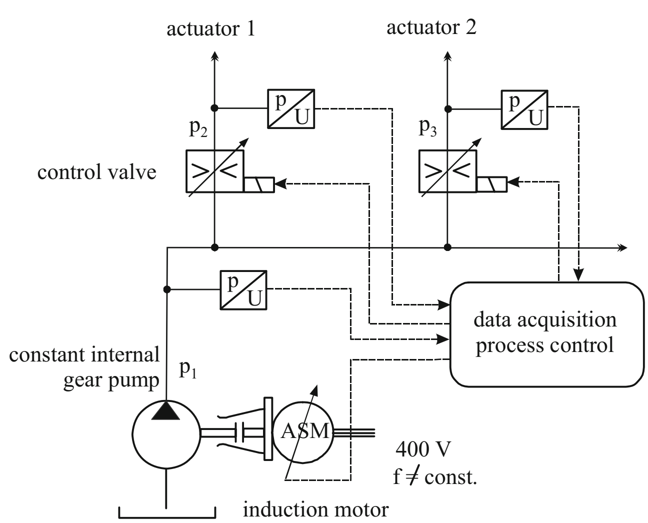



**Fig. 6** Electro-hydraulic LS-system (EHLS-system)

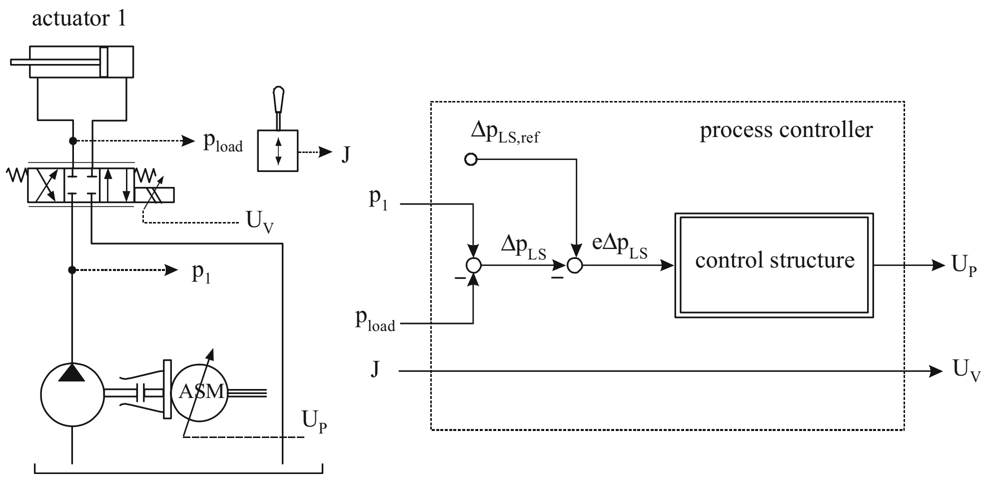



**Fig. 7** Hydraulic scheme of EHLS control system and signal flow

system pressure $p_1$ and actuator pressure $p_{load}$ represents the present pressure difference due to the load $\Delta p_{LS}$:

$$p_1 - p_{load} = \Delta p_{LS} \stackrel{!}{=} konst. \tag{1}$$

Control error $e\Delta p_{LS}$ as an input value of the pressure controller is defined by given reference value $\Delta p_{LS,\ ref}$. In our case, a controller integrated into the frequency converter was used for controlling the pressure, the whole control structure being designed according to the applied variable speed motor and constant-pump control concept.

## 5 Features of the constant-pump control concept

When constant-pump and speed-controlled induction motors are used for pressure (or for flow as well as power) control, there are two active control loops: the speed-control loop of the induction motor as a secondary (internal) control loop, and the pressure control as a primary control loop. Such a solution is known as "cascade control" [11].

In the case of cascade control, two variables, pressure $p$ and electric motor turning speed $n$ are monitored. Both values are measured and controlled. A principal block diagram of pressure control in the case of a constant pump is shown in Fig. 8.

In the case of cascade control, special attention should be paid to choosing an appropriate type of controller (see e.g., [11]). It is important to ensure that the selected secondary controller provides sufficient control-loop dynamics, which must be as fast and stable as possible over the whole expected operating range. Consequently, a P-controller was used, providing proportional dynamics without delays. The primary controller must allow for optimal pressure-control behavior. In the case under consideration, a P-controller was used in the secondary loop, and a PID-controller in the primary control loop. When both secondary and primary controllers are applied, more appropriate dynamic behavior can be expected. The application of a secondary controller allows for faster disturbance handling in the secondary loop. The LS-control concept is, in fact, a pressure control approach, where the controlled value is not 'given from outside' but generated inside the controlled system, depending on the actual load. The required pressure level corresponds to the current load level increased by the value of the pressure difference due to load $\Delta p_{LS}$. To reduce power losses, $\Delta p_{LS}$ should be as small as possible, yet big enough to ensure the expected dynamic. In the following

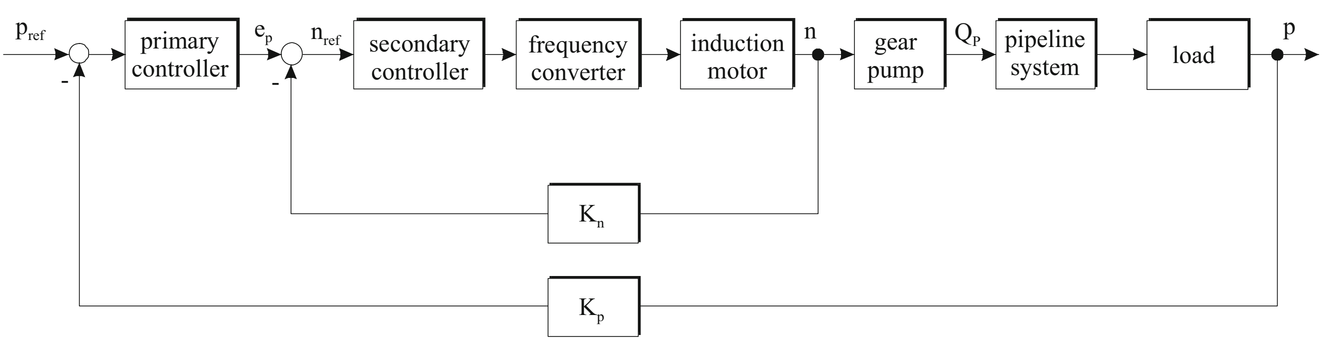



**Fig. 8** Principal block diagram of discussed cascade control

section, the LS concept using a variable-speed motor and constant pump was tested experimentally for a previously presented operational profile of a metal-forming machine.

## 6 Experimental results

Only a LS strategy with a hydraulic cylinder as the most loaded actuator is presented in the continuation of this paper, which was chosen due to the fact that the hydraulic cylinder of the press, for most of the working-cycle, operates as the most loaded actuator with variable load (except in the initial and final phases during product manipulation; see Fig. 2). Above all, the main purpose of this work was to check the feasibility of the LS control concept with a variable-speed motor and constant pump in the case of a real operational profile for a metal-forming machine. Application of such a concept was motivated by skepticism regarding the dynamic properties of control systems using variable pumps with higher inertia of rotational parts. It could even be slower in combination with the LS control concept, where the controlled value is generated inside the controlled system.

The original controllers, installed in the existing frequency converter, were used for realization of cascade-control. It was necessary to configure the control structure and set appropriate controller parameters to meet the requirements of the described cascade control. Definition and optimization of both controllers was carried out in accordance with the principles of the control technique: always in a stable and optimally steady-state, and with dynamic behavior. The above-mentioned necessary tasks were also done in compliance with the manufacturer's instructions for the applied frequency converter [12].

There are two PID-type controllers integrated into on applied frequency controller, which do not allow for completely switching off the integral part. Consequently, the integral parameter was set as low as possible to achieve behavior similar to the suggested simple P-controller. The differential part was set at zero. A PID controller was used as primary controller and to control the pressure, whereas a higher integral parameter was set to achieve pressure control without steady-state error.

An actual profile of the achieved control values using the suggested LS control concept is shown in Fig. 9. The actual profile is in accordance with a simplified pressure profile as shown in Fig. 3. The variable flow of the constant pump and power is presented, in addition to the pump's pressure profile. The latter two profiles were calculated from the actual turning speed and geometrical volume of the pump regarding the influence of compressibility and the temperature of the media.

The pressure, rotational speed, and calculated power of the driven electrical motor are achieved using load-sensing reference value $\Delta p_{LS,ref}$ = 10bar. As mentioned before, it must be sufficient to cover all pressure losses in the system and to enable expected dynamics. As can be seen from Fig. 9, $\Delta p_{LS,ref}$ = 10 bar enables suitable dynamics of the actuator $p_{load} = p_2$ - press cylinder to follow pressure changes during the working cycle.

Choosing a suitable $\Delta p_{LS,ref}$ is very important for achieving adequate dynamics. This system will provide very good control dynamics if too high a value is selected, but will operate at higher pressure than needed. Consequently, energy consumption and power losses will be increased. In the case where $\Delta p_{LS,ref}$ is set too low, power losses will decrease, but it will be impossible to achieve the required control dynamics. This can be observed in the case of Fig. 10, where $\Delta p_{LS,ref}$ was set too low ($\Delta p_{LS,ref}$ = 6

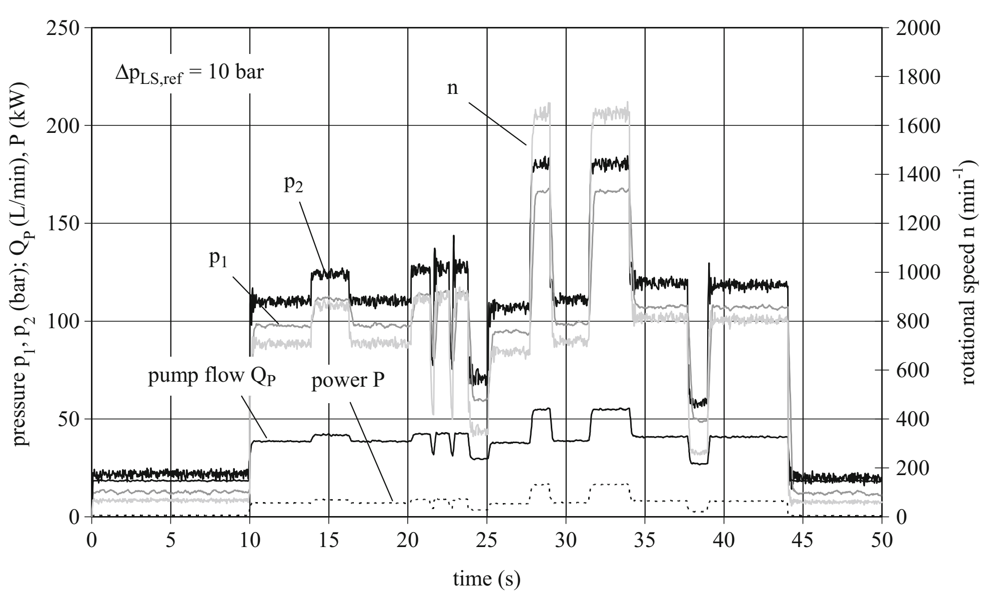



**Fig. 9** Experiment: pressure, rotational speed, and power distribution in the case of a speed-controlled motor and internal gear pump $\Delta p_{LS,ref}$=10 bar

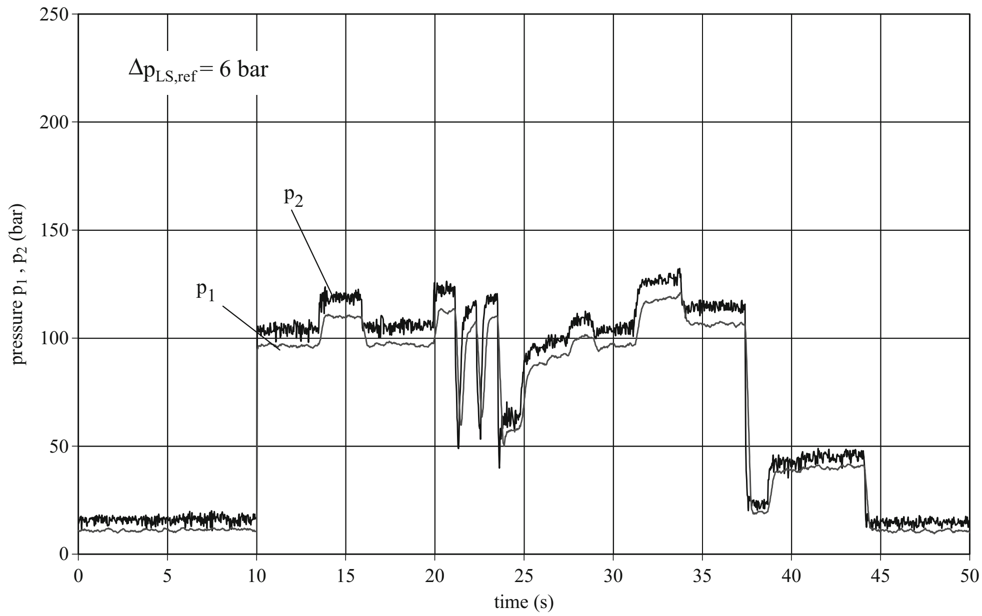



**Fig. 10** Experimental results: pressure and rotational speed of load-sensing concept with $\Delta p_{LS,ref}$=6 bar

bar). In order to satisfy general power-loss reduction efforts, it must certainly be taken into account!

The same control concept (cascade control) is presented in both the presented examples using the same type of controller.

As can be seen from the experimental results, the suggested LS-control concept using a variable-speed motor and constant pump enables low energy losses and reasonably priced solutions. It can be used for metal-forming machines, despite some doubt regarding dynamics.

Experimental results prove that the selection of components for the supply systems fulfils those expectations set for a hydraulic press-brake, and encourages further activities in the sense of supply-system implementation using a speed-controlled induction motor on other machines. The proposed control concept was also applied, in practice, to control the hydraulic supply system of a metal-forming machine.

## 7 Conclusions

The presented work deals with a modern controlled-hydraulic supply system, where the control of flow or pressure (also control of power) is achieved using a constant pump in combination with a speed-controlled induction motor. The suggested drive concept was applied in a load-sensing control system. Load sensing is known as an energy-saving system for supplying valve-controlled hydraulic actuators with the highest efficiency. Variable pumps are usually used for these purposes. In the case of a

**Table 1** Comparison of both drive concepts

| Variable hydraulic-mechanically controlled axial pump | Constant internal gear pump |
|---|---|
| Traditional approach | New drive concept |
| Mechanically complex | Mechanically simple therefore robust |
| Relatively expensive pump | Cheaper pump |
| Induction standard motor | Induction standard motor (or other) |
| No additional electric equipment needed | Frequency converter with controllers needed |
| Control inflexible because of “hardware engineering” | Control in electric side, very flexible |
| No pressure sensor needed | Additional pressure sensor needed |
| Constant e-motor speed | Variable e-motor speed; e-motor isn’t usually design for high speed >3,000 rpm and low speed (heating) |
| Higher, nearly constant noise level (controllable with simple measures) | Low noise level, frequency dependent |
| Small inertia (pump swash plate) | Higher rotational inertia (e-motor rotor) |
| Wear mechanisms at classical pump parts | Bigger tribologic problems at low speeds |
| Significant efficiency drop in partial load | Higher efficiency over the whole range |
| Complex maintenance | Ease of maintenance |

speed-controlled constant-pump's application, the high inertia mass of the electric motor is obstructive to achieving the desired dynamic of the process. The advantages and disadvantages of the suggested approach in comparison to the variable mechanically controlled axial-pump are presented in Table 1.

Experimental results prove that the suggested drive concept fulfils, on the whole, the goals and expectations aimed at in this work: low energy losses, low noise emission, improved control dynamics, reduced steady-state error, and convenient cost-effectiveness. These goals were achieved by using only those components available on the market. The presented work can serve as a reference study when implementing supply systems using speed-controlled induction motors in a load-sensing system in the wide field of metal-forming and other machines.

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