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ANALYSING LOSSES IN HYDROSTATIC DRIVES

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ABSTRACT

The paper introduces three levels of details in the analysis of losses in hydrostatic drives which are carried out at IFAS of RWTH Aachen University (Germany). The first level of detail investigates the over-all system efficiency, the second level is focused on components while in the third level of detail the tribological contact in hydrostatic displacement machines are analyzed. Due to an increased environmental awareness and rising energy cost energy efficiency becomes a crucial factor in system design, components and single tribological contacts. Therefore the efficient conversion of mechanical into hydraulic energy and vice versa is the main objective when developing e.g. mobile construction machinery. An increase in the energy efficiency of hydraulic pumps, motors, valves and cylinders directly leads to significant reduction in energy consumption of hydrostatic drive lines and working kinematics. From the system point of view the power consumption of the diesel engine, as the power source of most mobile machinery, has to be reduced in order to increase tank-to-wheel efficiency. Modern management strategies for mobile machinery have to include diesel engine, drive line and working hydraulic with focus on an efficient operation of the engine.

KEY WORDS

System efficiency, volumetric losses, hydro-mechanical losses, work of friction, single-piston test stand

NOMENCLATURE

$F_{R,\,ax}$:	axial friction force
$F_{S,\,ax}$:	axial friction force in suction stroke
$F_{P,\,ax}$:	axial friction force in pump stroke
M:	torque at pump shaft
Q_1:	pump inlet flow
Q_2:	pump outlet flow
$Q_{L,\,ext}$:	leakage flow
V_i:	individual displacement
W_f:	work of friction
n:	revolution speed
s:	piston stroke
s_{max}:	max. piston stroke
Δp:	pressure difference
p_1:	pump inlet pressure
p_2:	pump outlet pressure
p_L:	leakage pressure
η_{tot}:	over-all energy efficiency of system
$\eta_{tot,\,C}$:	over-all energy efficiency of component
η_{hm}:	hydro-mechanical efficiency
η_{vol}:	volumetric efficiency
ω:	angular speed
ϑ:	temperature

INTRODUCTION

Due to the widely spread increased environmental awareness and rising energy costs the energy efficiency becomes a crucial factor in systems, components and the single tribological contact. Efficient conversion of mechanical into hydraulic energy and vice versa is one of the most important aspects when developing mobile construction machinery. Because of their high power density and flexible arrangement hydrostatic drives are used in most applications of mobile machinery. An increase in energy efficiency of hydrostatic pumps, motors and cylinders directly leads to significant reduction in energy consumption of hydrostatic drive lines and working kinematics.

In scope of this paper three different levels of detail in the analysis of hydrostatic drives which are carried out at IFAS of RWTH Aachen University are introduced:

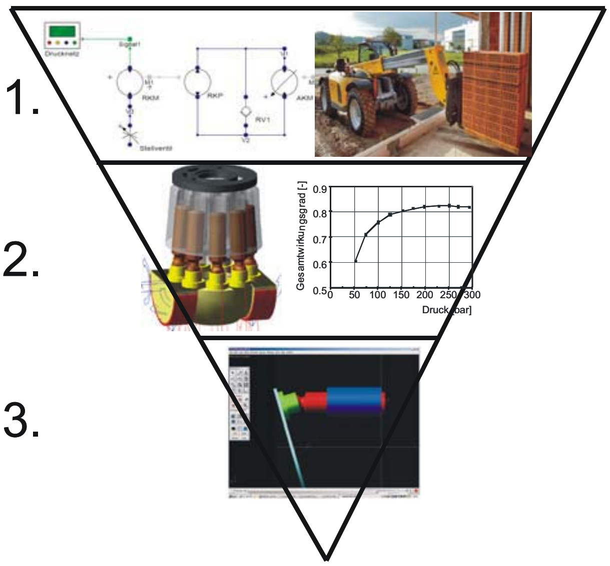

Figure 1 Three levels of detail

1. The first level of detail considers the over-all energy efficiency η_{tot} of the mobile machinery. The optimization of drive line concepts, working hydraulic and control algorithms is supported by dynamic system simulation (DSH*plus*) of the hydraulic circuits implementing all hydrostatic components. The detailed modeling of losses, systems and controls enables valuable information for the further development of hydrostatic drives.

2. The second level of detail is based on measurements of the hydraulic displacement units – pump and motor. Losses and component efficiency are investigated in dependency on speed n, pressure difference Δp, displacement V_i and temperature ϑ. The results of these measurements on the one hand proof the results of the 3rd level and on the other hand serve as an input for the 1st level of detail.

3. The third level of detail aims at the optimization of the hydrostatic displacement machines. Single tribological contact surfaces, e.g. piston-bushing or slipper-swash- plate, are investigated in simulations and measurements. Measured friction forces on the piston result in a work of friction W_f for one revolution of the single piston. This work of friction can be seen as a benchmark of different types of pistons and bushings.

These three levels of detail contribute to the advantageous investigation of hydraulic system solutions. The system depends on components which imply various tribological contacts.

1st LEVEL OF DETAIL
→ Over-all system efficiency ←

Hydrostatic power supply systems for mobile machines are quite complex. Considering a typical hydrostatic transmission, it is a fluid power system with many active and passive components having interfaces to the internal combustion engine (ICE), electronics and mechanics. In the development phase, dynamic simulation of new system solutions allows for optimization in a short cycle time and reduces time for testing significantly. The over-all system efficiency of mobile machines results out of the required output power and losses in every single component, e.g. ICE and transmission (see equation 1). Thereby, a higher over-all efficiency results in lower fuel consumption as well as lower CO_2 and NO_x emissions.

$$\eta_{tot} = \eta_{ICE} \cdot \eta_{trans} \tag{1}$$

Figure 2 introduces a simple design for a driveline containing ICE, hydro-mechanical power split transmission and tire.

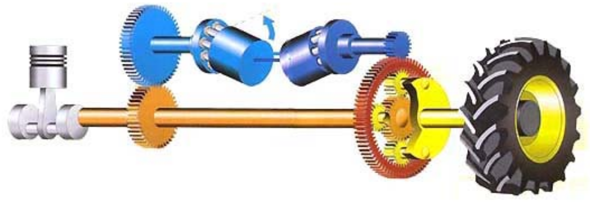

Figure 2 Exemplary drive line including engine, transmission and wheel

In order to achieve near-to- reality simulation results, precise loss modeling for changing operation conditions is required [5]. Modern drive line concepts are controlled by an electronic efficiency management system (EMS) which is regarding best-point-operation of engine and transmission.

Accordingly, the total efficiency of the vehicle can be improved by:

1. implementation of high efficient transmissions, e.g. hydro-mechanical power split transmission (HMT).
2. implementation of high efficient ICE
3. rising the average efficiencies of transmission and ICE due to an advantageous control strategy.

A hydro-mechanical power split transmission (HMT) combines the advantages of mechanical and hydrostatic transmissions. The high efficiency of mechanical gears and the continuously variable transmission ratio of hydrostatic transmission result in a highly efficient continuously variable transmission (CVT). Consequently, vehicle velocity is decoupled from engine speed, which can result in best-point- operation of the ICE. Vehicle velocity can be increased at constant ICE speed or the ICE speed is varied without influence on the vehicle velocity.
In scope of a collaborative research project IFAS is developing a flexible tool for the investigation and further development of drive line concepts for off-highway machines. Based on a loss-based model of drive lines and ICE the fuel consumption and thereby the emissions are evaluated. Recent work shows the huge influence of the ICE operation on the fuel consumption [4]. Three different control concepts for the same drive line concept were investigated for one short and one long loading cycle of a 120 kW wheel loader:

- **Control concept 1 (CC1)** – ICE throttle is held in a constant percentage which results in a nearly constant ICE speed. Vehicle velocity is controlled by the ratio of the implemented HMT.
- **Control concept 2 (CC2)** – In modern wheel loaders the ICE speed governs transmission ratio and thereby vehicle velocity. This concept is adapted to a drive line including a HMT.
- **Control concept 3 (CC3)** – Aims on high torque loads on the ICE shaft and higher ICE efficiency. The current power demand of drive line and work hydraulic is determined and ICE speed is adapted to the required power.

Constantly high ICE speed for **CC1** results in low torque loads on the driving shaft, high fuel consumption and thereby low over-all efficiency. Due to **CC2** the ICE speed varies with vehicle velocity and torque loads as well as efficiency can be increased. Savings of 13% fuel consumption for the short loading cycle and 6% for the long cycle seem possible. The ICE speed adaption according to the determined power requirement of the machine (**CC3**) results in saving of 15% for short and long loading cycle. ICE speed can be reduced for low power requirements and the efficiency is improved. Part of the improvement is also caused by the HMT. Advanced control strategies are therefore capable of increasing the average efficiencies of HMT and ICE.
For all simulations in scope of this paper DSH*plus* is used as the dynamic system simulation tool using an extended component library.
All these loss-based simulations of drive line concepts including the ICE rely on precise loss models of the implemented components. The losses of hydrostatic pumps, motors and transformers are essential for the investigation of partly and fully hydrostatic solutions. Additionally, the implemented control concept bears huge potential for reduced emissions and fuel consumptions. Precise measurements are required for the development of loss models for the flow and torque losses of hydrostatic units. These measurements are described in the 2nd level of detail.

2nd LEVEL OF DETAIL
→ Losses of hydrostatic displacement machines ←

Efficiency measurements of hydrostatic units – pumps and motors – require reliable sensor technology with high accuracy. In a first step, the real displacement of the hydrostatic units is determined. This can be done using different methods, as described in [3]. When running-in is completed, the machine is driven at chosen operating points and measurements are taken. Figure 4 shows the input and output power flows of a pump [6] and thereby the required measurement results.
Measured speed and torque result in the mechanical input power at the pump shaft which generates the hydraulic power in terms of flow and pressure (Figure 3).

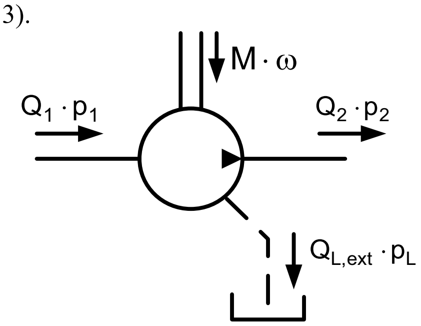

Figure 3 Input and output power of a pump [6]

Losses of the hydrostatic unit have to be separated into flow and torque losses by means of the real displacement for further detailed analysis. Flow and torque losses result in volumetric η_{vol} and hydro-mechanical efficiencies η_{hm}. These efficiencies are calculated according the following relations:

$$\eta_{hm} = \frac{V_i \cdot (p_2 - p_1)}{2 \cdot \pi \cdot M_{eff}} \quad (2)$$

$$\eta_{vol} = \frac{Q_2}{n \cdot V_i} \quad (3)$$

$$\eta_{tot,C} = \eta_{hm} \cdot \eta_{vol} \quad (4)$$

The reliable determination of the real displacement V_i of the machines is essential for a precise separation into volumetric and hydro-mechanical efficiencies. The over-all efficiency is not dependent on V_i. Furthermore, fluid temperature ϑ is of huge influence and therefore has to remain constant during the measurements.
To allow for the testing of hydrostatic units in pumping and motoring mode on one test stand, the electric motor drive has two shaft extensions at IFAS. One is used for the test unit and the other one is connected to a variable displacement axial piston machine, which operates as a recovery and load unit. This provides an energy-efficient operation in pumping and motoring mode during testing. Only the occurring losses have to be overcome by the electric motor drive.

3rd LEVEL OF DETAIL → Individual friction at tribological contacts ←

For the optimization of hydrostatic units the step towards the investigation of single tribological contacts is required. At IFAS this investigation is performed by means of simulation and specialized single-piston test stands.
Within the Collaborative Research Center 442 and the additional transfer project "Axial piston machines with PVD-coated components" three different single-piston test stands for swash-plate unit were developed:

- low speed single-piston test stand
- high speed single-piston test stand
- dry-running single-piston test stand

All these realize an inversed kinematics, compared to axial piston swash-plate units. Accordingly, the bushing is housed by the measuring platform which is mounted to stiff force sensors while the piston is moving in axial direction. This principle is described in detail by Renius [7] and allows for measurements of friction forces between piston and bushing. The axial friction force is one of the most relevant losses for swash-plate units. Figure 4 shows the main assembly of the low speed single-piston test stand.

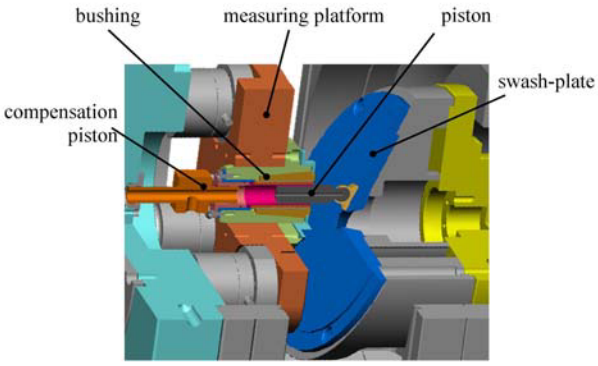

Figure 4 Low speed single-piston test stand

The axial movement of the piston is caused by a rotating wobble-plate and guided by the bushing in the measuring platform. All loads in terms of forces are taken up by four piezo-force sensors supporting the platform. All the lateral forces and the axial friction force are measured by the sensors [1, 2]. A specifically designed compensation piston is required to apply the hydraulic pressure to the back of the measured piston without disturbing the measurements. High and low pressure are varied in dependency of the rotating angle of the swash-plate by a control valve.
The test stand allows for speeds up to 10 rpm and aims at the optimization of the start-up torque of hydraulic motors. Normally, the hydro-mechanical efficiency is poor for low speeds of motors which results in torque ripples. This is mainly caused by the relatively high friction between piston and bushing while there is only little hydrodynamic lubrication in the gap. This low speed test stand is a suitable tool for analysis and optimization in order to rise start-up torque and lower the losses in swash-plate units.

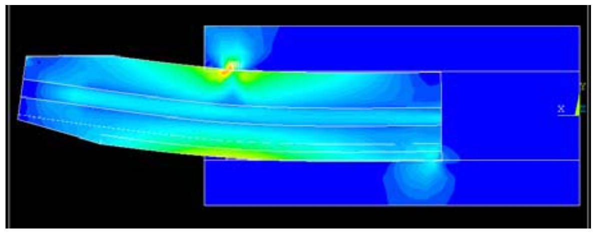

Figure 5 Calculation of stress and deformation

FEM-simulation models are implemented to learn about different influences on the losses and to determine stresses and deformations. Furthermore the design of the tribological contact between piston and bushing can be optimized by means of simulations. Figure 5 shows stresses and deformation of the piston caused by a lateral load force and the guiding bushing.
Especially when there is a hard/hard contact between the materials of piston and bushing the bending of the

piston and the local deformation of piston and bushing has to be regarded. The FEM-simulation help to find an appropriate contouring of the contact surfaces. This means e.g. choosing the radius at the front edge of the bushing and the piston-radius.
Looking at pumps, rotating speeds of more than 500 rpm are required. Therefore a high speed single-piston test stand was recently developed at IFAS. This test stand is capable of running at speeds of 500 rpm to 2,500 rpm. The friction forces are measured according to the same principle which was introduced for the low speed test stand. Certainly, all components were well optimized for this new purpose. Friction forces are much lower at high speeds of the hydrostatic unit than at low speeds. To maintain the high accuracy of the measurements at the low speed test stand, cross-talk of the piezo-force sensors is measured and subsequently compensated for.

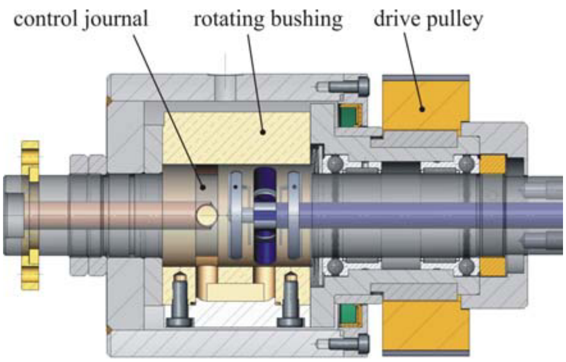

Figure 6 Sectional view of the control valve

Again, the hydraulic connection is realized via a compensation piston. While the commutation of pressure on the back of the piston can be performed by a simple electro-hydraulic valve at low speed, a new control valve was designed for high speeds. Figure 6 shows a cross-sectional view of the control valve.
Using the drive pulley, the rotating bushing is driven at operation speed and synchronized with the wobble-plate. The control journal has two ports: high pressure and low pressure. In the rotating bushing there is an integrated transfer port which realizes the commutation. The described control valve offers a reproducible pressure variation according to Figure 7.

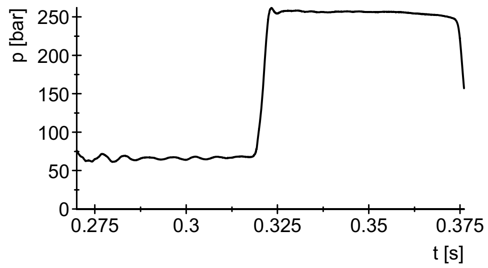

Figure 7 Pressure variation @ 500 rpm

The design and assembly of the third single-piston test stand is not yet completed. It is a simple construction with the purpose to measure dry friction between piston and bushing. Using two hydraulic cylinders a side force can be applied and the piston can be driven in axial direction, as Figure 8 shows.

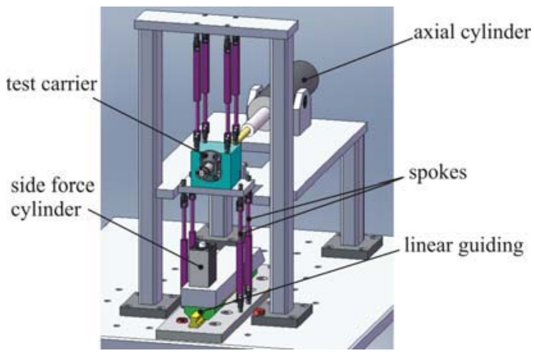

Figure 8 Dry-running single-piston test stand

Friction force is measured by one piezo-force sensor and the test carrier is fastened to the frame by spokes. When the piston is moved by the axial cylinder, the axially guided cylinder for the lateral force moves with the piston. Measuring the friction coefficient using this test stand is supposed to be better than using other principles like a disc/disc test stand for instance.

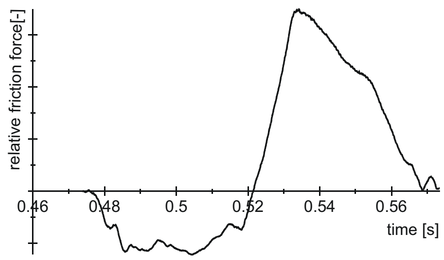

Figure 9 Friction force @ 500 rpm, 300 bar

The most interesting output of all the measurements with the presented single-piston test stands are the resulting friction force diagrams. Usually the friction force is plotted versus time or angle as shown in Figure 9.

With these friction force diagrams it is difficult to compare different designs of pistons and bushings. It is much better to have a close look on the losses which are caused by the friction force for one stroke of the piston. Consequently the friction force is plotted versus the working stroke of the piston (Figure 10).

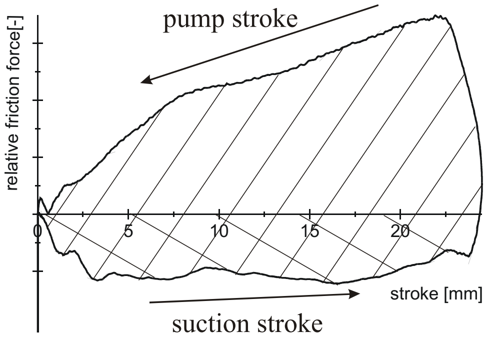

Figure 10 Work of friction @ 500 rpm, 300 bar

The work of friction W_f is calculated by the integration of force in dependency of the working stroke. It is indicated by the hatched area and calculated according to:

$$W_f = \int_0^{s_{max}} F_{P,ax} ds + \left| \int_0^{s_{max}} F_{S,ax} ds \right| \quad (5)$$

Finally, the work of friction W_f is dedicated to be an appropriate benchmark for analysing and comparing axial friction losses between piston and bushing.
In addition to the measurements on single-piston test stands, the performance for most suitable piston-bushing combination can be analyzed on the component test stand at IFAS. Improvements in the single tribological contact will also lead to improvement in the component behaviour. Afterwards, the measured component performance can be integrated into system simulation tools and prove the functionality in the hydraulic system. Small changes on the contact side may result in huge improvements on the unit performance.

SUMMARY

In times of rising energy costs and tightened governmental restrictions regarding the emissions and fuel consumption of vehicles and mobile working machines, losses in hydrostatic units are to be reduced.
At IFAS three different level of detail are utilized to analyze and improve the performance of hydrostatic drives. Tribological contacts are the main loss source in hydrostatic units. FEM-simulation models and advanced measurements result in a good understanding of the origin of volumetric and hydro-mechanical losses in order to derive improvements by geometric changes. Measurements on hydrostatic units, pumps and motors, are a major benchmark for the quality of improved components. Additionally, precise measurements can result in precise simulation models for the over-all system investigation. Beneath the improved efficiency of hydrostatic components the improvement of the over-all system efficiency will be the result of new structures. These structures, e.g. hybrid, motor controlled, will first be analyzed by state-of-the art system simulation tools.
In summary, the introduced tools are capable of improving the efficiency of hydrostatic system by means of simulation, measurement and system know-how.

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