Optimization of the efficiency of hydrostatic drives

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Abstract

Heavy duty mobile machines such as road construction vehicles, agricultural and forestry machines are frequently driven by hydrostatic transmissions. The compact hydraulic components in these transmissions allow to provide continuous speed control while very large torques can be produced. Because of the increasing fuel price and stringent emission regulation, manufacturers of hydrostatic driven applications are pushed to improve the efficiency in order to stay competitive. Different strategies can be chosen to realize this objective: the hydraulic pump and motor can be replaced by components with higher efficiencies or the control of the different drive components can be improved by taking into account their interaction. While in many cases efficient pumps and motors are already being used, the control of these components is suboptimal. In this abstract an optimization procedure for the speed control of a hydrostatic drive train is described and validated on a 110 kW test bench.

KEYWORDS: hydrostatic drive, efficiency, test bench

1. Hydrostatic drives

1.1. Introduction

A hydrostatic transmission exists in an open and closed circuit variant /1/. In mobile traction applications mainly the closed loop topology is implemented /2/. The system

consists of an internal combustion engine (ICE) and a hydraulic pump and motor. A gear box can be installed between the ICE and the pump to increase the input speed of the pump with respect to the ICE output speed. This allows to use both the ICE and the pump in their typical speed range. The hydraulic motor drives the wheels mostly over a mechanical reduction. **Figure 1** shows a schematic diagram of the power components in the traction of a vehicle,. Next to these power components, subsystems should be installed in a closed loop hydrostatic drivetrain respectively to compensate the volumetric losses (refill pump, check valves, filter, ...) and to avoid an overload in the hydraulic motor are adjustable in volumetric displacement and are controlled by an electrical or hydraulical unit.



Figure 1: Power components in hydrostatic drive

In a moving hydrostatically driven machine, a certain output speed desired by the driver has to be realized. To fulfil this speed demand, a certain oil flow through the motor is required (equation (1)). As a result the rotational input speed of the hydraulic pump has to be set according to equation (2) such that this flow demand is actually delivered.

When a machine is moving, the hydraulic motor has to deliver a torque, depending on the load condition, the machine dynamics and the desired speed command of the driver. To overcome the load torque at the output shaft, a pressure difference, as calculated in equation (3), has to be realized across the motor. Since the hydraulic pump has to build up this pressure difference, the ICE should drive the pump with a torque as calculated in equation (4).

MOTOR		PUMP	
$Q_m = \frac{V_{g,m} \cdot n_{m,out}}{\eta_{m,vol}}$	(1)	$n_{p,in} = \frac{Q_p}{V_{g,p} \cdot \eta_{p,vol}}$	(2)
$\Delta p_m = \frac{2\pi \cdot T_{load}}{V_{g,m} \cdot \eta_{m,hm}}$	(3)	$T_{p,in} = \frac{\Delta p_p. V_{g,p}}{2\pi. \eta_{p,hm}}$	(4)

By using these equations it is possible to calculate the settings: motor displacement $V_{g,m}$, pump displacement $V_{g,p}$ and pump speed n_p , for a certain load condition at the output shaft of the hydraulic motor /3/. In equation (1) and (2) oil leakage is taken into account using the volumetric efficiency parameter in equation (1) and (2), while mechanical friction and hydro mechanical losses are taken into account using the hydro mechanical losses are taken into account using the hydro mechanical efficiency parameter in equation (3) and (4). It is important to stress that both the volumetric and hydro mechanical efficiency from the pump and motor depend on the rotational speed, the pressure difference Δp over the component and the volumetric displacement Vg of the component. The relation between these parameters can be found in iso-efficiency charts, delivered by the manufacturer. As an example, **Figure 2**shows the overall efficiency of a hydraulic motor as the multiplication of the volumetric and hydro mechanical efficiency, in relation to the rotational speed and pressure difference. In the left plot the efficiency is plotted for Vg equal to 30% and in the right plot for Vg at maximal level /4/.



Figure 2: Iso-efficiency plots hydraulic motor

The input power at the pump shaft is delivered by the ICE. The efficiency of the engine can be calculated from a consumption chart, **Figure 8**.

1.2. Standard speed control

Traditionally, in many hydrostatic driven applications a standard control approach is used to determine the driveline settings (the motor displacement $V_{g,m}$, pump displacement $V_{g,p}$ and pump speed n_p) when a certain vehicle speed variation is requested /5/. In this approach, where energy efficiency is not targeted, three different stages can be distinguished, as illustrated in **Figure 3**.

First, when the vehicle is driving at low speed, the ICE is set at a nominal speed level and the displacement of the hydraulic motor is set at the maximum level. By doing this the motor has the maximum torque to pressure ratio. Consequently the motor can deliver his maximum torque and strongly accelerate, if desired. If an increase of speed is required, the displacement of the pump is increased.

When the maximum swash plate angle of the pump is reached, the second stage begins. In this stage an increase of vehicle speed is possible by decreasing the displacement of the hydraulic motor.

In the third and last stage, the maximum vehicle speed is reached by increasing the ICE speed to the maximum level. Although the complete speed range can be reached in this way, this standard control approach is suboptimal from an energy point of view.



Figure 3: Standard speed control

1.3. Optimized speed control

From Eqs. 1 and 2, it is clear that in a hydrostatic transmission, a speed setpoint can be achieved by changing the speed of the ICE (n_{ICE}), the swash plate of the hydraulic pump ($V_{g,p}$) or the motor displacement ($V_{g,m}$). Because three parameters can be adjusted, the same speed can be realized with different parameter settings, each resulting in a different energy consumption. As a consequence, by selecting the correct parameter settings, it is possible operate at the optimum energy efficiency. In many applications though, the above mentioned, standard speed control method is applied, where energy consumption is not taken into account. At the left side of **Figure 4**, the workflow of a two-steps optimization routine is presented, which minimizes the overall driveline's energy consumption by selecting the optimal driveline settings.



Figure 4: Optimization workflow and measurements

Goal of the optimization routine is to select the driveline parameter settings such that the overall drive train efficiency, as calculated by the multiplication of the hydrostatic efficiency and the ICE efficiency, **Figure 4**, is minimized . Because the hydrostatic efficiency and the ICE efficiency depend on the operating point of their drive components, hydraulic motor/pump and engine respectively, the routine uses the following digitalized 3D efficiency maps to characterize these components:

- Hydraulic pump and motor: volumetric and hydro mechanical efficiency as a function of work pressure, displacement and rotational speed
- ICE: efficiency in relation to speed and torque of the engine

In a first step, the routine calculates, for a given setpoint of hydraulic motor speed and torque, whether a certain combination of ICE speed (n_{ICE}), pump displacement ($V_{g,p}$) and the motor displacement ($V_{g,m}$) can fulfil the demands taking into account the limitations of the different components:

•	1100 rpm	<	ICE speed	< 2100 rpm
•	0 cc	<	Pump displacement	< 165 cc
•	45 cc	<	Motor displacement	< 160 cc
•	0 bar	<	Max system pressure	< 420 bar

If so, the total efficiency of the system is calculated in a second step. This procedure is repeated for all possible system settings. In a final step, the combination with the maximum efficiency is selected as the optimal setting.

The output of the optimization routine are the driveline parameter settings to reach the highest total efficiency that can be obtained for a certain speed and torque delivered by

the hydraulic motor: the optimal engine speed, motor displacement $V_{g,m}$ and pump displacement $V_{g,p}$, **Figure 5**.



Figure 5: Output optimization routine: upper left total efficiency, upper right engine speed [rpm], bottom left swash plate pump [%], bottom right motor displacement [%]. All in relation to output speed [rpm] and torque [Nm]

2. Measurements

A test bench is developed to validate the optimization routine. By performing measurements on this test setup, a practical comparison between the standard speed control approach and the optimized speed routine has been made.

2.1. Test bench

The test bench, conceptually shown in **Figure 6**, is built to validate the optimization routine. Because all measurements are performed in laboratory conditions, the ICE is emulated by an electric motor (110 kW) directly driving the pump. A second electric motor allows to apply a load torque at a certain speed demand to the hydraulic motor. **Table 1** shows the manufacturer, type and most important pump and motor specifications.



Figure 6: Schematic view of the test bench

The input power P_{in} and output power P_{out} can be measured and are used to calculate the hydrostatic efficiency η_{hydr} , **Figure 6**. Since the torque T_{ICE} and actual speed n_{ICE} of the driving electric motor are also measured, the motor efficiency η_{ICE} . can be estimated by using a fuel consumption chart of the emulated diesel combustion engine, Multiplying η_{hydr} and η_{ICE} leads to the efficiency of the total drive train η_{TOT} .

PUMP	MOTOR
Sauer-Danfoss	Sauer-Danfoss
H1	Serie 51
165 cc/rev	160 cc/rev

Table 1: Pump and motor specifications

2.2. Measurements

Main goal of the measurements is to compare the efficiency of the standard speed control approach with the optimized settings. The total efficiency is experimentally determined for both speed control methods. Because measuring the efficiency of a hydrostatic unit is a difficult task with several potential misinterpretations /6/, the obtained results are comparatively examined. A validation is performed for different speed and load conditions at the output of the hydraulic motor.

In the standard speed control method, the nominal ICE speed has a great impact on the efficiency of the combustion engine. To examine the impact of this parameter, the standard speed control is applied with three different nominal engine speeds: 1500 rpm, 1750 rpm and 2000 rpm.

The optimization routine is allowed to vary the ICE speed in the range 1100 rpm to 2200 rpm.

2.2.1. Results

Figure 7 shows the overall drive train efficiency η_{tot} for the three different output speeds of the hydraulic motor: 500 rpm, 1000 rpm and 1500 rpm. In each chart, four trends are displayed. The upper trend represents η_{tot} if the optimized control is implemented. The other three trends are representing the standard speed control, respectively with nominal engine speed of 1500 rpm, 1750 rpm and 2000 rpm.



Figure 7: Overall drive train efficiency depending on the load for three motor speeds: 500 rpm, 1000 rpm and 1500 rpm

Figure 7 confirms that the optimized speed control always results in the highest total efficiency. The difference between the standard control decreases if the nominal engine speed is set to a lower level. Because the maximum ICE efficiency is 40 %, the overall efficiency hardly reaches 30 %.

The ICE speed has an important impact on the total efficiency. By comparing two set points in the engines consumptions charts (see **Figure 8**) the positive effect can be explained. Both points are representing the same load condition of the hydrostatic drive: output speed is 1000 rpm and load torque is 400 Nm. If the standard speed control is used with the engine speed at 1750 rpm, the torque is 374 Nm, the specific consumption of the engine will be 217 g/kWh and the engine has to deliver 69 kW of power.

The optimization routine decreases the engine speed to 1196 rpm. At this point, the engines load torque is 490 Nm, the specific fuel consumption is 207 g/kWh and the

engine power is only 60 kW. Two interesting effects explain the improvements in total drive train efficiency:

- 1. The reduction in input engine power proves that the application now works at an improved hydrostatic efficiency, caused by the more optical settings of pump and motor displacement. Less input power can fulfil the applied load condition.
- 2. The optimization routine moves the engines working point to a zone, where the specific fuel consumption is decreased to 207 g/kWh. As a result, the engines efficiency is improved in respect to the standard situation.



Figure 8: ICE specific fuel consumption chart

For end users of mobile machinery not the total efficiency, but the fuel consumption is the most important factor. Assuming that a machine operates 1000 hour per year, the engines consumption can be calculated for each load situation. **Figure 9** shows the fuel consumption of the standard settings related to the optimized settings in terms of percentage, for different speed demands (500 rpm, 1000 rpm, 1500 rpm) and load torques.



Figure 9: Fuel consumption in terms of percentage, related to optimized level

It is clear that by using an optimized control strategy significant reductions in fuel consumption can be obtained, in the entire working range. The difference can even run up to several tens of percentages. The difference is decreasing while speed and load torque is increasing.

3. Conclusions

The ICE is an essential drive train unit in mobile machinery. Comparison of the efficiency of the different drive train components (engine, pump, hydraulic motor) shows that the engines are the least efficient. Therefore it has also a major impact on the overall drive train efficiency. Operating the engine as long as possible in his minimum specific consumption zone, will drastically improve the overall efficiency.

The developed optimization routine is maximizing the total efficiency by selecting the optimal combination of engine speed and displacement of the pump and motor to meet a certain load and speed demand on the output axis. A test bench is built to implement different speed control algorithms. Measurements on this test bench validated the improvement in efficiency using the optimal settings, calculated by the optimization routine.

The speed and load set point of the application affect the profit in efficiency that can be obtained. Machines with engines operating at high speeds while low power performance is needed, have a significant margin to optimize the fuel consumption.

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Symbols

Q_m	Input flow motor	m³/s
$Q_{ ho}$	Output flow pump	m³/s
V _{g,m}	Motor displacement	m ³ /rotation
$V_{g,p}$	Pump displacement	m ³ /rotation
n _{m,out}	Rotational speed motor out	rotation/sec
n _{p,in}	Rotational speed pump in	rotation/sec
$\eta_{m,vol}$	Volumetric efficiency motor	%
$\eta_{m,hm}$	Hydro mechanical efficiency motor	%
$\eta_{p,vol}$	Volumetric efficiency pump	%
$\eta_{{\scriptscriptstyle p},{\scriptscriptstyle hm}}$	Hydro mechanical efficiency pump	%
$\eta_{\scriptscriptstyle hydro}$	Total hydrostatic efficiency pump & motor	%
$\eta_{\scriptscriptstyle ICE}$	ICE efficiency	%
η_{tot}	Overall drive train efficiency: ICE & hydrostat	%
$\Delta p_{ ho}$	Pressure difference pump	Ра
Δp_m	Pressure difference motor	Pa