Aachen 2010

Efficiency Optimisation of Tracked Vehicles Using Secondary Control in a Single-Circuit Load Sensing System

Dipl.-Ing. Peter Dengler

Institute for Mobile Machines, Karlsruhe Institute of Technology, Germany

Dipl.-Ing. Lars Völker

Institute for Mobile Machines, Karlsruhe Institute of Technology, Germany

Dr.-Ing. Wolfgang Kauß

Bosch Rexroth DSI

Prof. Dr.-Ing. Marcus Geimer

Institute for Mobile Machines, Karlsruhe Institute of Technology, Germany

ABSTRACT

The paper describes a concept of a secondary controlled traction drive integrated in a common Load Sensing system. When used in a single circuit system, a secondary controlled drive for tracked vehicles shows a big efficiency optimisation potential compared to other drives. The high losses at turns can be avoided if two secondary controlled units are connected to a single Load Sensing valve. The traction drive is still supplied in a traditional way using Load Sensing. For steering, the secondary controlled units are integrated in a control loop where the steering ratio is used as command value.

SUBSCRIPTS

i	inner
o	outer
act	actual
W	weight
r	roll

NOMENCLATURE

ε	steering ratio	-
v	rotary speed	rpm
F	force	N
c	distance of instantaneous center	m
l	length of track contact surface	m
w	width of track contact surface	m
b	distance of contact surfaces	m
r	turn radius	m
$\dot{\varphi}$	angular speed	rad/s
μ	friction coefficient	-
k	geometric factor	-

1 INTRODUCTION

A single circuit Load Sensing (LS) system in mobile machines allows a power transmission to all actuators in an easy, load independent and cost efficient way. The velocity of the working hydraulics as well as the traction drive hydraulics can be controlled independently of their loads with only one pump (**Figure 1**). By adapting the system pressure to the highest load pressure the system can generally been driven efficiently. However, high power losses occur if an actuator needs a high flow at a pressure far below the system pressure. This appears at turn motions of skid steering tracked vehicles.

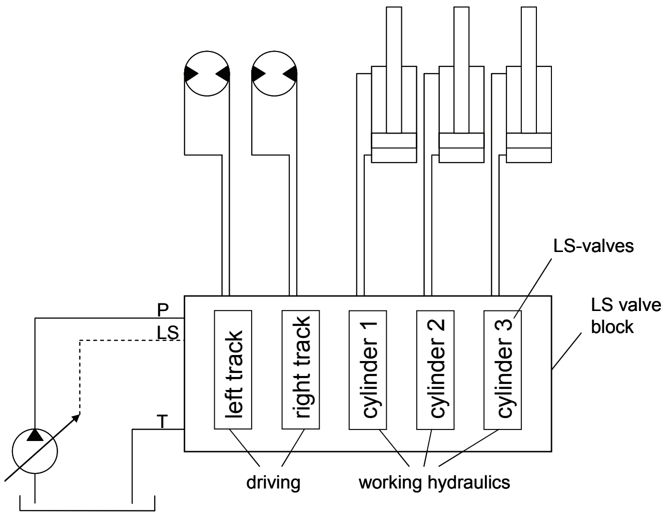

Figure 1: Single circuit LS-system of a mobile machine

In order to turn the vehicle, the driving motors of left and right track must turn with different rotary speeds as shown in **Figure 2**. The difference of velocities of the tracks makes the centre of gravity CG of the machine turning around a centre of turn CT with the radius *r* so that the tracks can be distinguished in inner and outer track. Furthermore, an on-the-spot turn is possible if both tracks are driven in reverse direction. In this case both tracks are outer tracks.

Beside of the turn radius a turn can be described by the steering ratio which is defined here as the ratio of the rotary speeds of inner and outer track (**Equation 1**).

$$\varepsilon = \frac{v_i}{v_o} \qquad (1)$$

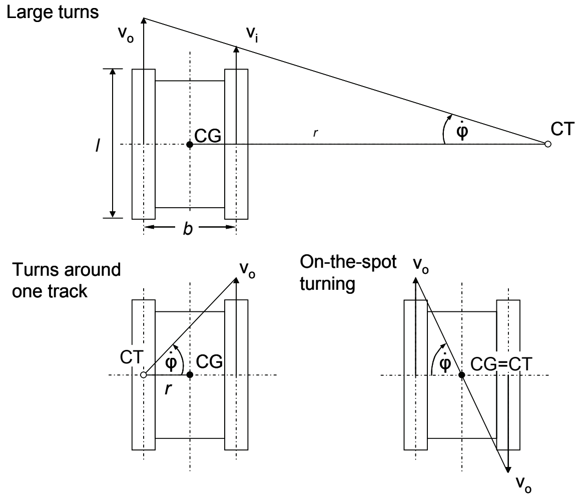

Figure 2: states of motions at turns of tracked vehicles

The disadvantage of this kind of steering is the high slip between track and ground. It causes a high load pressure on the driving motor of the outer track and a very low load pressure on the motor of the inner track. By running in a single circuit LS-system, this pressure difference may create a high power loss in the hydraulic system, especially when a high flow is demanded on both drives.

In order to reduce the power losses in the hydraulic system and therefore to increase the efficiency of the traction drive at turns, the driving motors with fixed displacement can be replaced by motors with variable displacement. These motors with variable displacement are integrated in a speed control loop and change their displacements according to the principle of secondary control. Showing the potential of a secondary controlled traction drive of tracked vehicles at turns and its functioning in a LS-system with a single pump is the main goal of this paper.

2 POTENTIAL OF EFFICIENCY OPTIMISATION OF TRACKED VEHICLES AT TURNS

2.1 Influence of friction forces of tracked vehicles at turns

For the calculation of the friction forces at turn movements many publications have already been done. Exemplarily /Fla66/, /Ehl91/ and /Kit76/ should be named. The estimation of loads used in this paper is based on their work.

Figure 3 shows motions and frictional forces of the track at turns. Supposing an instantaneous centre IC which is created at each turn beside of the contact area of each track at the distance *c* the track slips around this instantaneous centre with the same angular velocity as the vehicle turns around the centre of turn CT. Considering an area element of the contact surface the slip velocity of the track at this point corresponds to the velocity v_{slip} with its components in *x* – and *y* – direction.

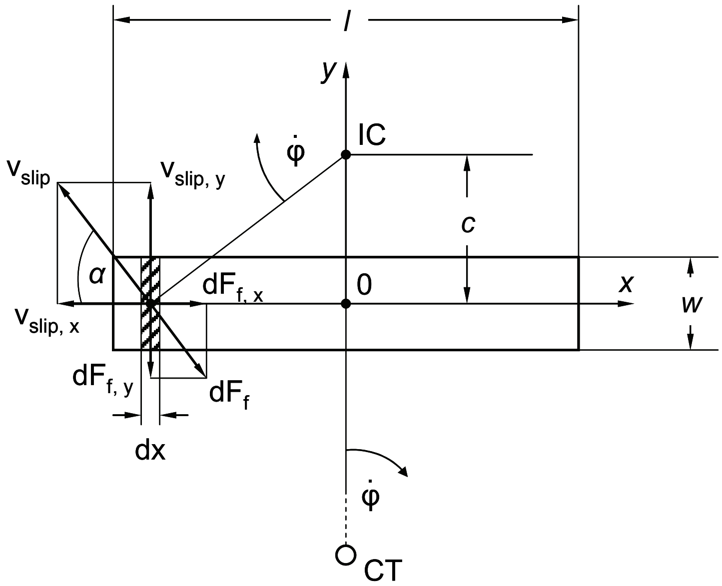

Figure 3: Resulting forces on a contact area element at turns

Supposing an equally distributed weight force F_W with a friction coefficient for turns μ_{Turn} the additional tension of the track can be determined.

$$dF_{f,x} = dF_f \cos\alpha \; ; \; \cos\alpha = \frac{c}{\sqrt{x^2 + c^2}} \; ; \; dF_f = \frac{F_W \mu_{Turn}}{wl} w\,dx \tag{2}$$

$$F_{f,x} = 2\int_0^{l/2} \cos\alpha \frac{F_W \mu_{Turn}}{wl} w\,dx = F_W\, \mu_{Turn} \left(\zeta \ln \frac{1+\sqrt{1+\zeta^2}}{\zeta} \right) = F_W \cdot \mu_{Turn} \cdot f\left(\zeta\right) \tag{3}$$

with $\zeta = 2c/l$.

In turn motion the outer track of the turn slips against the direction of travelling so that more power must be delivered to the outside travel drive. In the same time the inner track slips in travelling direction so it must be braked in order to create a necessary momentum to turn the vehicle. Supposing a rolling resistance μ_r between track and ground the traction force for the inner and outer track can be determined according to /Fla66/ to

$$F_i = F_W \left[\mu_r - \mu_{Turn} f\left(\zeta\right)\right] \qquad (4)$$

for the inner track and

$$F_o = F_W \left[\mu_r + \mu_{Turn} f\left(\zeta\right)\right] \qquad (5)$$

for the outer track.

With the geometric dimensions of the hydraulic motor the necessary pressure for the inner and the outer drive can be derived from the traction forces calculated above.

Because of the logarithmic increase of $f(\zeta)$ the pressure difference between inner and outer drive can already reach important values at steering ratios close to 1. Considering the efficiency of the hydraulic system as the ratio of consumed power at the motors to the produced power by the pump a simulation run has been done with the load model developed above. For the simulation model a linear relationship between steering ratio ε and distance of the instantaneous centre of turn c is supposed with $\zeta = k\ (1-\varepsilon)$. This leads to equal loads at straight travelling ($\varepsilon = 1$) and maximal load difference while turning around one track ($\varepsilon = 0$), depending on the geometry of the vehicle (factor k). A turn-on-place has to overcome an even higher friction but not considered here because of the equal distribution of loads at this movement. **Figure 4** shows the simulation results and the test results of a tracked excavator. A characteristic drop down of efficiency can be observed with a minimum at approximately $\varepsilon = 0{,}95$ due to the high pressure difference and the high flow for the inner motor (which is still at 95% of the outer motor). At this point the pressure of the inner motor has its minimal possible value, defined by the braking device which protects the motor from cavitation. With steering ratios $\varepsilon < 0{,}95$ the growth of pressure at the outer motor slows down and also the flow to the inner motor becomes less important. This leads to a continuous increase of efficiency for smaller turns. It is evident that the highest losses are not at tight turns where the highest friction must be overcome but rather at large turns or even little drive adjustments at straight travelling.

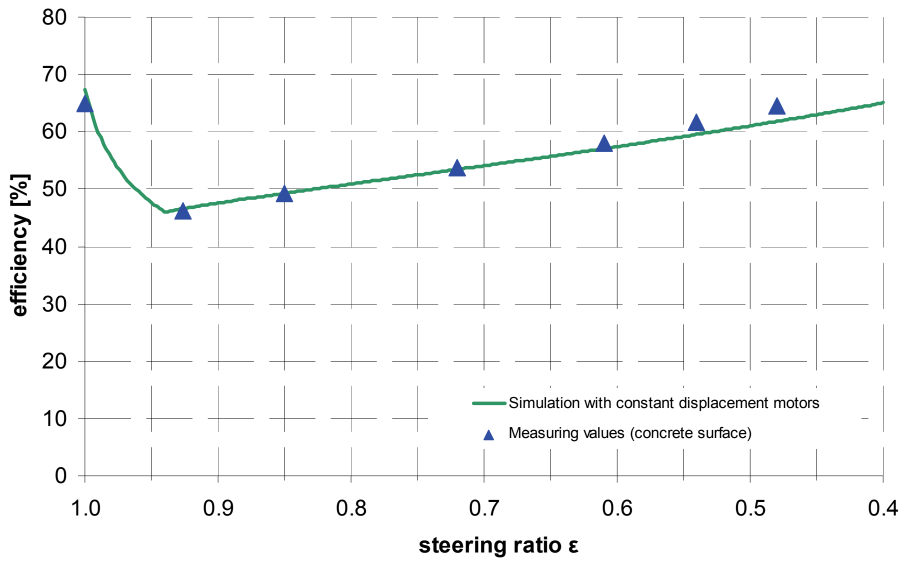

Figure 4: Efficiency evolution of tracked vehicles at turns

2.2 Integration of secondary controlled drives in Load Sensing systems

The principle of secondary control is a power control which is not effectuated by the pump or valves but by the hydrostatic drive itself. In order to create power control by secondary control the system needs to have a pressure controlled pump, a hydraulic accumulator and a hydraulic motor which can operate in four quadrants /Kor96/, /Haa89/. Integrated into a speed control loop, the hydrostatic drives adapt their power consumption individually by an adaptation of displacement. Therefore, no power controlling valves are needed so no significant power losses can occur between pump and motor. According to the high load difference at steering of tracked vehicles described in **2.1** the efficiency at turns can be optimised if both driving motors share a pressure line and adapt their power by secondary control.

Common secondary controlled drives work at a high constant pressure so that the hydraulic system of the traction drive needed to be separated from the circuit of the working hydraulics in order to avoid high power losses at the hydraulic cylinders.

Another possibility is to integrate hydraulic transformers which transform the high pressure of the main system to a lower working pressure needed at the hydraulic cylinders /Vae09/.

In order to keep the power losses as low as possible and not to change the entire hydraulic system the secondary controlled drive needs to work at variable pressure /Zäh93/. A subsystem is created where both drives are connected to a common line which is itself connected to the LS-system by an inlet LS-valve as shown in **Figure 5**.

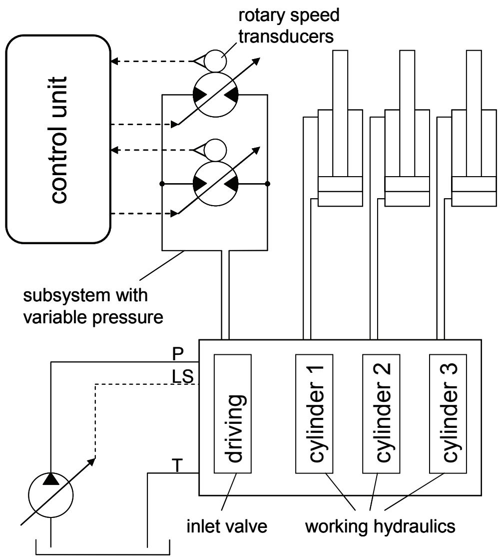

Figure 5: Mobile machine using a secondary controlled drive in a LS-system

The pressure of this subsystem is “automatically” controlled by the LS-pump. The velocity of the vehicle and the direction of driving are enabled by simple actuation of the inlet valve. Turns are effectuated by an adaptation of displacement of the drives which creates different driving torques and enables therefore a steering of the vehicle.

With steering by an adaptation of displacement the pressure does not need to be throttled so high power losses which appear at the conventional system are eliminated. To turn the machine, a high torque difference between inner and outer motor at steering ratios $1 > \varepsilon > 0{,}95$ and a braking torque on the inner drive for $\varepsilon < 0{,}95$ is needed. This leads quickly to very low or even negative displacements of the inner motor. Therefore, the flow demand of the inner motor becomes very low, for steering ratios $\varepsilon < 0{,}95$, a recuperation of flow at the inner drive takes place. In consequence, the flow demanded by the subsystem is considerably reduced which decreases in the same extent power losses in the valve block. **Figure 6** shows that the elimination of power losses combined with a strong reduction of flow enables even an increase of efficiency at turns. The mechanics of tracked vehicles at turns and the advantages of secondary control leads in this case to a higher efficiency at turns than at straight travelling.

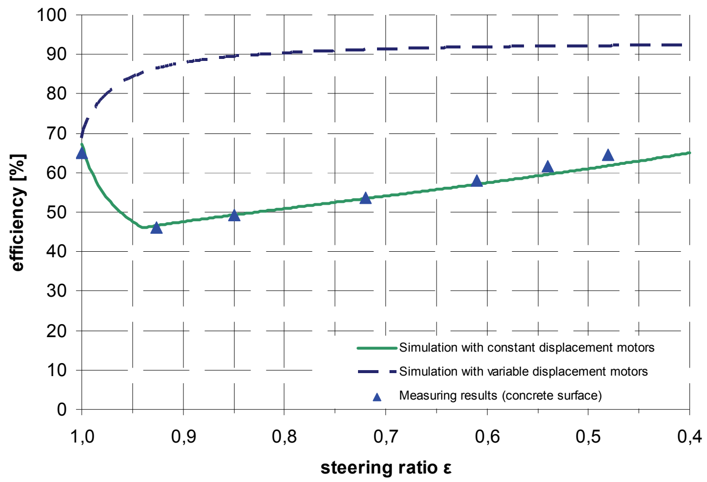

Figure 6: Efficiency evolution of a secondary controlled drive in a LS system

3 CONTROL METHOD

3.1 Basic function

As described in **2.2**, both drives are connected together to a line and create a subsystem with variable pressure. Contrarily to a "classical" secondary control the rotary speeds of the driving hydraulic motors can not be controlled independently because of the impressed flow of the LS-system. This would lead to a superposition of the speed control by LS and the speed control by secondary control. Basically, a direct speed control of the motors is not necessary because it is already created by the LS system. By shifting the spool of the inlet valve straight and backward travelling is possible (see Figure 5). In order to enable steering, the steering ratio ε is introduced as a new command variable. The advantage is that the range of the setpoint is always between -1 and 1 which simplifies the control task. The definition which drive is the inner and which one the outer drive can change and depends only on the drivers request (see **3.2**).
At straight travelling with equal loads at both sides, both motors are swivelled to maximum displacement. The flow is divided into the same amount to the drives. In this

mode the subsystem can be described from two different points of view: as a common flow control with fixed displacement motors or as secondary control at given pressure with fully swivelled variable displacement motors. When different torques are demanded the motor with the higher load keeps maximum displacement and defines the pressure for the other motor which swivels back and reduces its driving torque according to the principle of secondary control.

3.2 Control method under various maneuvering tasks

To enable the three major motion states: straight travelling, steering and turn-on-place around the z-axis, an explicit signal for each task needs to be defined which is compared to an actual value. Basically, the control unit controls the displacement of the drives in a way that at least one motor is always swivelled at maximum. In a first step the rotary speeds of both drives are measured and transformed into the actual steering ratio ε_{act}. The way how ε_{act} is determined depends on the driver's request: if the driver wants to turn left the control unit determines the actual ratio $\varepsilon_{act} = (v_{left} / v_{right})_{act}$, in case of a right turn the actual steering ratio is created as $\varepsilon_{act} = (v_{right} / v_{left})_{act}$ independently of the real motion state of the machine. The determined actual value can therefore reach values >1. This allows to create a deviation $\Delta\varepsilon = \varepsilon_{set} - \varepsilon_{act}$ even if a change of the steering direction is initiated with the same steering ratio as at the precedent turn. According to the deviation to the actual steering ratio $\Delta\varepsilon = \varepsilon_{set} - \varepsilon_{act}$ the control unit realises which drive is „too fast“ and adapts its displacement until the set steering factor ε_{set} is reached. In the case of a left turn, the control unit identifies the left unit as too fast, if the deviation $\Delta\varepsilon = \varepsilon_{set} - \varepsilon_{act}$ is negative and the right motor if the deviation is positive. After having identified the faster motor the control unit will at first try to swivel the slower motor out in order to accelerate it. If this is not sufficient or if the drive is already at maximum displacement the control device will actuate the faster motor and swivel it back in order to brake it. This method always allows to keep at least one drive at maximum displacement so that the pressure in the subsystem is kept at a minimum level.

A special feature of tracked skid steering vehicles is a turn-on-place around the z-axis by driving the motors in counter ways direction. The control method works also in this case if the signs of the rotary speeds are included into the calculation. This should be explained in an example: the machine is driving straight forward with fully swivelled motors (e.g. with equal loads on both units) at *X rpm* for each motor, the actual steering ratio is therefore $\varepsilon_{act} = v_{left} / v_{right} = [+X\ rpm] / [+X\ rpm] = 1$. The driver wants to turn left around the z-axis with a rotary speed on the right drive two times higher than the left drive. This leads to a set ratio of $\varepsilon_{set} = v_{left} / v_{right} = [-X\ rpm] / [+2X\ rpm] = -0{,}5$ and the deviation is $\Delta\varepsilon = \varepsilon_{set} - \varepsilon_{act} = -0{,}5 - 1 = -1{,}5$ and therefore negative. As described above, the control unit recognizes the left motor as too fast and will start the regulation just as at normal turns until the commanded steering factor is reached. The set steering ratio can only be reached if the left motor rotates in reverse direction.

For backward movements the flow is reversed by a shift of the inlet LS-valve so that the signs of the velocities for left and right motor turn both into negative. Turns can be therefore effectuated by the same method as for turns at forward movements.

CONCLUSION

The paper describes the high power losses of tracked vehicles at turns when equipped with a single circuit load sensing system. Replacing the traction drive motors with fixed displacement by secondary controlled drives with variable displacement allows to eliminate the losses. A further advantage is the energy recuperation of the inner motor when a braking torque is demanded. With the described control method which uses the steering ratio as command variable the secondary control of the hydraulic motors is decoupled from the actual rotary speed control of the drives, effectuated by the LS-valve. Therefore, the LS-system does not need to be changed which allows an easy integration so that no further costs for additional pumps and hoses come up.

The outer drive consumes the most power and is always swivelled at maximum displacement. Due to the higher efficiency of motors when driven with higher swivelling angles the outer drive has a high efficiency /Zäh93/. This makes a motor at variable pressure more efficient than a motor in a “classical” secondary control with constant pressure where the motor is swivelled at maximum only if maximum power is demanded. An accumulator which is usually used in secondary controlled systems is not needed here which reduces costs.

REFERENCES

/Fla66/ **Flach, W.**, *Das Kurvenfahren von Raupenfahrzeugen mit zwei Raupen*, Baubetriebstechnik, Heft 1 p.16-20, Krausskopf Verlag, Mainz, Germany, 1966

/Ehl91/ **Ehlert, W.**, *Prüfstandssimulation der Fahrwiderstände von Gleisketten-fahrzeugen, insbesondere bei Kurvenfahrt*, Dissertation, Universität der Bundeswehr, Hamburg, Germany, 1991

/Kit76/ **Kitano, M., Jyozaki, H.**, *A theoretical analysis of steerability of tracked vehicles*, Journal of terramechanics p.241-258, Elsevier Science, Amsterdam, Netherlands, 1976

/Kor96/ **Kordak, R.**, *Hydrostatic drives with control of the secondary unit*, The Hydraulic Trainer Vol. 6, Bosch Rexroth, Lohr am Main, Germany, 1996

/Haa89/ **Haas, H.-J.**, *Sekundärgeregelte hydrostatische Antriebe im Drehzahl- und Drehwinkelregelkreis*, Dissertation, RWTH Aachen, Aachen, Germany, 1989

/Vae09/ **Vael, G.et al.**, *Hydrid-Antriebe für Gabelstapler*, 2. Fachtagung Hybridantriebe für mobile Arbeitsmaschinen p.157 - 168, Karlsruhe, Germany, 2009

/Zäh93/ **Zähe, B.**, *Energiesparende Schaltungen hydraulischer Antriebe mit veränderlichem Versorgungsdruck und ihre Regelung*, Dissertation, RWTH Aachen, Aachen, Germany, 1993