MODELLING AND EXPERIMENTAL VERIFICATION OF A SECONDARY CONTROLLED SIX-WHEEL PENDULUM ARM FORWARDER

Alessandro Dell'Amico\textsuperscript{a}, Liselott Ericson\textsuperscript{a}, Fredrik Henriksen\textsuperscript{b} and Petter Krus\textsuperscript{a}

\textsuperscript{a} Division of Fluid and Mechatronic Systems, Department of Management and Engineering, Linköping University, Sweden
\textsuperscript{b} Skogforsk, the Forestry Research Institute of Sweden, Sweden

Abstract

One of the major concerns in the forest industry is the impact on the soil caused by the forest machines during harvesting, where damages can have a negative impact on e.g. further growth. One of the main reasons is wheel slip. Another concern is the working environment of the operator due to the harsh ground in the forest. Both these issues have a negative impact on productivity. An attempt to overcome these challenges is made within a collaborative research project, which among others also includes Linköping University, where a new six-wheel pendulum arm forwarder is being developed. The new forwarder aims at reducing the soil damage by an even pressure distribution and smooth torque control, as well as increased damping of the complete chassis, and thereby improving the working environment. This is possible since each wheel, driven by its own hydraulic motor, is attached to a pendulum arm allowing to control the height of each wheel independently of each other. The forwarder has a total maximum weight of 31 tonnes, including 14 tonnes maximum load. It consists of two steerable joints and is driven by a 360 bhp diesel engine. The transmission consists of two hydraulic pumps and six hydraulic motors.

This paper deals with the development of the driveline and presents the first experimental tests of the implemented control strategies, where a secondary control approach is chosen for its ability to individually control the torque on each wheel. The control strategies, presented in the paper, include pressure control, velocity control of the vehicle and an anti-slip controller. To support the development of the control strategies, models of the vehicle and hydraulic subsystems are derived. The aim with this paper is to verify the concepts on the actual vehicle. The initial results are promising, indicating that the suggested concept is feasible.

Keywords: Secondary control, forest machinery, forwarder, modelling

1 Introduction

Compared to a traditional forwarder, a forwarder with pendulum arms and individual control of each wheel will have clear benefits when it comes to the impact on the soil caused by the machines during harvesting. Within a collaborative research project between Skogforsk (the Forestry Research Institute of Sweden), Bosch Rexroth, Extractor AB (machine owner), Linköping University and The Royal Institute of Technology (KTH) in Stockholm, such a forwarder with six wheels is being developed, where the overall objective is to minimise soil damage and improve the operator’s working environment. The project initially started as a student project, followed by a Master thesis. More information on these can be found in Larsson et al. (2013) and Henriksen (2014). One of the main reasons behind soil damage is wheel slip. Studies have been conducted to register the impact different forwarders have on the soil and can be found in Edlund et al. (2012) and Edlund et al. (2013).

While the traditional forwarder uses one pump, one motor and a mechanical linkage to drive the forwarder through sequential control of the hydraulic machines, this new forwarder comprises two hydraulic pumps and six hydraulic motors. Each hydraulic motor is attached to each wheel, allowing individual control of the motion of each wheel. Since each wheel is also attached to a pendulum arm, the height of each wheel can be freely adjusted with respect to the chassis, and each other, allowing for a levelling of the chassis and an even pressure distribution between the wheels and the ground. These two properties should help to minimise damage to the soil. The complete chassis can also be utilized to reduce movements of the cab, thereby improving the operator’s working environment. A comparable machine is the el-forest, forest (2015). El-forest has some similarities to the forwarder presented in this paper, but has an electric driveline instead of a hydraulic driveline.
This paper deals with the development of the control strategy for the driveline and is a continuation of the work in Ericson et al. (2015). The main idea is to use secondary control due to its ability to control the torque or velocity on each wheel independently, without any throttling losses, and can cope with wheel slip more easily compared to a sequential controlled driveline. The objective with the paper is to implement the secondary control strategy in the forwarder and evaluate the performance on flat ground by comparing the drivability to the sequential control strategy. A first version of the torque based anti-slip strategy developed in Ericson et al. (2015) is also implemented and tested. The experiments will give an indication of the suitability of the concept in this machine. The modelling procedure, as well as the control strategies, are also outlined.

2 System Description

The forwarder being developed has a total maximum weight of 31 tonnes, including a maximum load of 14 tonnes and is shown in figure 1(a). It consists of three bodies with two actuated steering joints and six wheels, each attached to a pendulum arm, as illustrated in figures 2(a) and 2(b). The transmission consists of two closed hydraulic circuits with one Bosch Rexroth A4 variable pump and three Bosch Rexroth A6 variable motors for each circuit, as shown in figure 1(b). The two pumps are powered by the same 6 cylinder, 360 bhp diesel engine. System parameters are stated in table 1. The pumps can go over centre but not the motors. A gear is installed between each hydraulic motor and wheel to increase the output torque. Rotational velocity sensors are attached to each wheel and four pressure sensors measure the high and low pressure of each hydraulic circuit. In addition, the exact position of each pendulum arm can be calculated from the pendulum arms’ position sensors.

![Fig. 1: The forwarder and a schematic of the hydraulic system.](image1)

![Fig. 2: Kinematics of the forwarder’s three turning axes and the pendulum arm.](image2)
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel motor power</td>
<td>$P = 268 \text{ kW}$</td>
</tr>
<tr>
<td>Pump rotational speed</td>
<td>$n_p = 2000 \text{ rpm}$</td>
</tr>
<tr>
<td>Pump displacement</td>
<td>$D_p = 140 \text{ cm}^3/\text{rev}$</td>
</tr>
<tr>
<td>Secondary machine displacement</td>
<td>$D_m = 107 \text{ cm}^3/\text{rev}$</td>
</tr>
<tr>
<td>Mass of the vehicle</td>
<td>$m = 17 \text{ 000 kg}$</td>
</tr>
<tr>
<td>System pressure, high pressure side</td>
<td>$p = 32 \text{ MPa}$</td>
</tr>
<tr>
<td>System pressure, low pressure side</td>
<td>$p = 3 \text{ MPa}$</td>
</tr>
<tr>
<td>Gear ratio</td>
<td>$U = 48.3$</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>$r = 0.74 \text{ m}$</td>
</tr>
</tbody>
</table>

### 3 Control

Secondary control is a technique known for high dynamics and the possibility to recuperate energy and avoiding throttling losses. It was first developed in 1977 Nikolaus (1977) and was investigated in Palmgren and Rydberg (1987) for example. Historically, one of the drawbacks has been the necessity for advanced electronic controllers, where a purely hydromechanical solution is not possible. However, this can not be seen as a drawback with the computational technology available today. A control strategy for a secondary controlled hydrostatic transmission was developed in Berg and Ivantysynova (1999).

The basic principle of secondary control is shown if figure 3(a). The primary and secondary machines are controlled separately in the sense that the pump is controlled to keep the system pressure constant, while the secondary machine directly controls the output, hence the name secondary control. The basic schematic figure in figure 3(a) shows an open circuit. The closed circuit architecture used in this paper is shown in figure 3(b). The systems work according to the same principle. However, with the closed circuit, the secondary machine does not need to swivel over zero to brake the rotational speed as in open circuit architecture. At braking the high pressure side is changed and the secondary machine begins to work as a pump, thus braking the vehicle.

![Fig. 3: The secondary control in an open circuit and a closed circuit respectively. The pump controller controls the system pressure and the controller for the secondary machine controls the transmission output.](image)

To recover energy in the open circuit, the secondary machine needs to swivel over zero. In a closed circuit, however, energy can be recuperated even if the secondary machine does not swivel over zero through an arrangement of valves, similar to Sprengel and Ivantysynova (2013) for example. With the secondary control architecture, several drives can be added to the same pressure line with no load interference. This is the most important property for this work, while energy recuperation is not within the scope of this paper.

The secondary control concept should be compared to the common sequential control, which in its simpler form is open loop control. Here is the reference velocity set by the throttle pedal’s position. The displacement settings of the hydraulic machines are mapped to the pedal position, where a certain setting would give a certain velocity.

The simulated transmission has two separate hydraulic systems, each with one pump and three motors. Two different approaches for secondary control were presented in Ericson et al. (2015), but only one of them is implemented in this paper. The velocity controller adjusts the output torque on each wheel in order to maintain a certain vehicle speed. The wheel’s rotational speed is defined by the torque equilibrium of each wheel. The differential speed of the inner and outer wheel during cornering is therefore solved automatically. Wheel sleep is handled by an additional controller.

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Addition to the secondary controller, the source pump needs another control strategy to maintain a certain pressure level. The controllers are kept simple for easy implementation at this stage.

3.1 Pressure control
Pressure controlled pumps were investigated by Palmberg et al. Palmberg and Kangzhi (1987), where the pump with its hydro-mechanical controller was described by an inductance and a break frequency for the pump, together with the break frequency of the system volume, made up the complete control system. In this work, however, the pump is not of the pressure control type, but has an integrated electro-hydraulic displacement controller. A first order lag is assumed to represent the displacement control, with a time constant of 20 rad/s. The pressure control loop is then implemented in the software based on the measurement of the pressure. A schematic of the pump controller is shown in figure 4. The low pass filter represents the internal pump controller. A PI-controller is used to control the pressure.

![Fig. 4: Schematic of the pump controller.](image)

3.2 Velocity control
Figure 5 shows a schematic of the implemented velocity controller. The outer loop and the controller, C_D, can be either a software implemented controller maintaining a certain vehicle speed or a driver controlling the vehicle speed. The velocity control error, equation 1, is to be minimised by controller C_D.

$$e(v_i) = v_{ref} - v_{current}$$  \hspace{1cm} (1)

In this work, the outer controller is a PI-controller that calculates the drive torque needed to follow the reference velocity, v_{ref}. Initially, the total torque is divided equally over all six motors. The corresponding motor displacement setting for each motor is calculated in block G_T according to equation 2.
\[
e_i = \frac{1}{6} \frac{2\pi T_{tot}}{D_m \Delta p} \quad \text{where} \quad i = 1 \ldots 6
\]

where \(D_m\) is the motor displacement and \(\Delta p\) the system pressure. The system pressure, controlled by the pump controller, fluctuates in the simulation by a maximum of \(\approx 1\%\), which means that \(e_i\) is approximately proportional to \(T_{tot}\). In the actual vehicle, however, the pressure fluctuations were larger and the reference pressure was used instead to avoid introducing these oscillations.

The slip is controlled by an inner controller by comparing the actual rotational speed of each wheel with the calculated reference rotational speed. In the case of one wheel slipping, the rotational speed of that wheel will increase. If the difference against the model calculated speed is too large, the displacement setting of the corresponding motor will be decreased. The control rule for the motors’ displacement setting \(e_{m,i}\) from the inner control loop is stated as:

\[
e_{m,i} = e_i - C_i (\omega_i - \omega_{ref}) \quad \text{where} \quad i = 1 \ldots 6
\]

where \(\omega_i\) is the measured rotational velocity of the hydraulic motor and \(\omega_{ref}\) the reference rotational speed. The calculation of the desired rotational speed of each wheel is based on a kinematic model of the vehicle and the reference vehicle velocity, see section 4.1. The velocity can be derived in different ways. In Ringdahl et al. (2012), the wheel slip is estimated using GPS signals and a similar estimation is needed for this vehicle. The estimation of the vehicle speed is not within the scope of this paper and the vehicle speed is therefore derived from a vehicle model. \(C_i\) is a tuneable parameter, dependent on the amount of slip that is allowed. The choice of parameter value is also subject to expected model error. In the actual vehicle the measured wheel rotational velocity was low-passed filtered.

As with the pump controller, a first order lag filter is implemented to represent the displacement control. The time constant is set to 20 Hz.

4 Modelling

Developing models of the forwarder and subsystems under study facilitates the development of control strategies as well as giving a first indication of the performance of the vehicle and the suitability of the control concepts. During the initial phase of the project, a fairly simple vehicle model is used, though it covers the important aspects when studying the driveline. The driveline itself consists of the hydraulic machines and volumes. At this point, the diesel engine is not modelled and delivers a constant rotational speed. A slip model is implemented to investigate how the controllers behave and how they handle loss of traction of a wheel.

4.1 Vehicle model

For this paper’s analysis, the kinematic model is simplified. The vehicle has two articulated steering joints, see figure 2(a). The first body turns with angle \(\alpha\) and the last body’s steering angle \(\beta\) will follow the first body in such a way that no lateral slip occurs on each wheel. This means that the radii \(R_P, R_M\) and \(R_B\), coincide at the same centre point during turning. The wheels perceive different speeds during turning. The vehicle has different distances between the steering joints and the wheel axes, which makes the steering radius individual for each wagon to prevent slip. Note, this is only one possible steering condition. The driving transmission is the main focus in the paper and this simplified driving mode is satisfactory.

The resistive forces considered in the vehicle model are rolling resistance, \(F_r\), slope resistance, \(F_g\) and a disturbance force from a simulated bump, \(F_b\), see Ericson et al. (2015). The rolling resistance is calculated as in equation 4,

\[
F_r = C_r F_N
\]

where \(F_N\) is the normal force. The resistance is independent of the torque and it can be justified due to the low torque. The slope resistance \(F_g\) is calculated as:

\[
F_g = mg \sin(\gamma)
\]

The vehicle acceleration can now be calculated as:

\[
m \ddot{v} = \sum_{i=1}^{6} F_{m,i} - \sum_{i=1}^{6} F_{r,i} - F_g - F_b
\]

where \(F_{m,i}\) is the driving force for each hydraulic motor.

The model only considers longitudinal acceleration since only low speed manoeuvring is considered. The model is used both for the vehicle itself and the reference model. The vehicle model uses the torque from the hydraulic motors to calculate the driving force and with the kinematic constraints it calculates the velocity of the vehicle, \(v_r\).
4.2 Hydraulic model

The hydraulic pumps and motors are modelled according to the standard equations for pump and motor flow and torque, Eq. 7 to 10.

\[
q_p = \varepsilon_p D_p n_p \\
q_m = \varepsilon_m D_m n_m \\
T_p = \frac{\varepsilon_p D_p}{2\pi} \Delta P \\
T_m = \frac{\varepsilon_m D_m}{2\pi} \Delta P
\]

where \(D\) is the machine displacement, \(\varepsilon\) the displacement setting, \(n\) the rotational velocity and \(\Delta P\) the pressure difference across the machine. Indexes \(p\) and \(m\) denote pump and motor respectively. The losses in the machines are for this work simplified with a leakage and a viscous friction. The inertia of the machines is modelled with newton’s second law, while the dynamics of the displacement setting is assumed as a first order lag. The hydraulic machines are connected with hoses constituting a volume. This is modelled with the continuity equation for a volume as in Eq. 11,

\[
q_p - q_m = \frac{V}{\beta_e} \frac{dp}{dt}
\]

where \(V\) is the volume, \(\beta\) the resulting bulkmodulus and \(p\) the pressure in that volume.

5 Experimental performance tests

Four different tests are designed to verify the performance of the different controllers, the secondary control as concept and the anti-slip function. All results are presented in section 6.

Response of pressure controller

In this test the forwarder is standing still and only the pressure controllers for the two pumps are active. The proportional and integral gains are tuned until a satisfactory trade-off between speed and overshoot is achieved.

Response of velocity controller

In this test the controller gains of the pressure controller are set according to previous test. Various steps in reference velocity is applied while the controller gains of the velocity controller are tuned. To get an idea of the performance required without interfering with the driver’s experience of the vehicle, the sequential controller is also verified for different velocity steps. The reference velocity is in this case set with the pedal and a perfect step is not possible to apply. Since the sequential controller has no feedback, only the performance of the machines will decide the vehicle performance. It is assumed that this case represents a sufficient behaviour of the forwarder.

Driving in an eight

In this test the forwarder drives around an eight with purpose to verify the secondary control as a concept, alternating inner and outer wheel during cornering. Both pressure and velocity controllers are set according to previous tests. The pressure controller is set to hold 150 bar for good performance.

Performance of anti-slip controller

For simplicity the anti-slip controller is only implemented for the rear left wheel. The front wheels serve as a reference velocity assuming they always grip. The forwarder only drives forward and compensation for inner and outer wheel speed difference is not necessary and therefore not implemented at this stage. To simulate a loss of grip the rear left wheel is being lifted while driving forward with the help of the pendulum arm cylinder.

6 Results

6.1 Response of pressure controller

Figure 6 shows the results of the pressure steps at different levels and amplitude. Some overshoot is allowed to increase the performance. The settling time is quite long but the rise time is instead quite fast. This should be advantageous when smoother inputs are applied.
Fig. 6: Response of the pressure controller at different pressure levels. Dotted line is the reference pressure. Both high pressure sides are plotted on top of each other, as well are both low pressure sides.

6.2 Response of velocity controller

Figure 7(a) shows the response of the velocity controller compared to the sequential controller in Fig. 7(b). From what it was possible to measure it is not straight forward to compare the two plots. The secondary controller, however, tends to be somewhat slower and oscillate more than the sequential controller at this point.

Fig. 7: Response of the velocity controller for secondary control compared to the response of the sequential controller. Dotted line is the velocity reference.

6.3 Driving in an eight

Figure 8 shows the results of driving in an eight. Figure 8(b) shows that the control signal to hydraulic motors are equal. The plot shows the signal of the two front motors. Figure 8(c) clearly shows the velocity difference between the inner and outer wheel. Figure 8(d) also shows that the pressure stays around the preset value of 150 bar but tends to oscillate.

6.4 Performance of anti-slip controller

Figure 9 shows the performance of the anti-slip controller. When the rear left wheel is being lifted the cylinder pressure of the corresponding pendulum arm is increased, as seen in Fig. 9(a) as the red dotted line. The control current at that point is increased, which corresponds to a decrease in displacement setting and a lower torque on that wheel, shown in Fig. 9(b). The idea is that all wheels should keep the same velocity regardless of the grip level. At this point the controller was not able to keep the lifted wheel at a constant velocity but it was greatly reduced allowing the forwarder to keep its velocity. This is shown in Fig. 9(c). There is also a slight increase in displacement setting to the rest of the motors to compensate for the loss of torque of the lifted wheel.
Fig. 8: Performance test while driving around in an eight. The plots show the turning angle of the joints, hydraulic motor signals and speed, and the performance of the pressure controller.

Fig. 9: Performance of the anti-slip controller.
7 Discussion and future work

The pressure controller was set to allow a certain overshoot in order to increase the performance, Fig. 6. A step response with high amplitude pushes the controller to its limits and is not representative for actual driving. The low pressure side also dips in pressure as the pressure increases on the high pressure side, resulting in an lower overshoot for the pressure difference, which is the torque generating pressure.

The velocity controller also shows some oscillations, Fig. 7. Since the sequential controller was set from the pedal and is open loop, it was not straight forward to reproduce a sharp step at exact levels as for the secondary controller. At this point, however, the sequential control shows somewhat better performance.

There are several measures that can be undertaken to improve the performance of the controllers. In this paper simple PI-controllers were used, since the main purpose is concept verification. Future work will see a proper system analysis, which will help developing more sophisticated controllers. There is most likely a lot of phase shift coming from the hydraulic machines, limiting the performance of the controllers. A proper analysis will indentify this. The control unit also has built in PID controllers at each output port in order to control the current to each hydraulic machine. For the experiments standard settings were used. This is perhaps a further possibility to improve the machine performance. The underlying idea with secondary control is to keep the pressure constant and the control of the velocity would only be dependent on the dynamics of the hydraulic motors. This is in contrast to sequential control where also the pressure has to be built up before the forwarder can move. It is therefore justified to believe that with better controllers the secondary control concept should reach at least the same performance as the sequential control.

In the velocity controller, the reference pressure was used to calculate the desired torque instead of using the actual pressure. The reason was to avoid introducing further oscillations in the system since the pressure was oscillating somewhat. However, when the pressure oscillates the output torque oscillates as well, having negative impact on the velocity. If the actual pressure is used in the velocity controller, the controller could compensate for this and results would be a smoother velocity. This will only work if the hydraulic machines are fast enough and the verification of this is subject to future work. A smoother pressure would of course also help. The common assumption when analysing secondary controlled systems is that the pressure is constant, which is far from the reality in this low capacitance system. In a future system analysis it would be interesting to see the interference of the pressure on the velocity. An increase in system capacitance, by e.g. using accumulators, would probably also help.

Figure 8 clearly shows how the basic idea behind the secondary control concept works. While negotiation the eight the control signal is equal to all machines, hence the drive torque is equal on all wheels. Figure 8(c) shows the speed difference between the inner and outer wheel.

A simple implementation of the anti-slip control concept allowed to verify its functionality. As the rear left wheel was lifted it lost its grip. Without the controller the wheel would spin and consume all oil delivered from the pumps and the vehicle would stop. With the anti-slip controller active the displacement setting was decreased, showed as an increase in current in Fig. 9(b), reducing the output torque and hence the wheel speed. The controller was not able to keep the lifted wheel at the same speed as the other wheels and is something that will be investigated in future work. While the rear wheel is lifted the total drive torque is reduced and the displacement setting on the other machines is increased to compensate for this.

8 Conclusions

The results show that the main ideas with secondary control of the forwarder and the anti-slip controller works, which is also according to simulation results. This concludes that it is worth continuing the work and improve the controllers and thereby the performance of the forwarder.

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References


