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Best of Both Worlds – Load-Dependent and Load-Independent Hydrostatic Driving Switchable On Demand with Innovative Hybrid Electro-Proportional Control

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Abstract.

When selecting a hydrostatic transmission, there is a choice between load-dependent and load-independent pump control. With certain limitations, the load stiffness characteristics can be tuned using software, although this requires sensors. In addition, a system controlled by an ECU can cause a small but noticeable delay for the machine operator. Danfoss now introduces the Hybrid Load Control (HLC) that combines load-dependent and load-independent control characteristics in one control valve without additional actuators and sensors. This paper explains the functionality and design of this innovative control valve, as well as its switching mechanism.

Keywords. Pump control, servo system, hydrostatic propel system, mobile hydraulics

1. INTRODUCTION

When designing hydrostatic propel systems, mobile machine manufacturers must select the machine's fundamental driving behaviour. State-of-the-art systems are categorized into two types.

The first is a load-independent behaviour, also known as speed mode. This is used for constant-speed applications or for scenarios where the system load should not affect the driving behaviour.

The second type is a load-dependent behaviour, also referred to as torque mode. This approach utilizes the power of the prime mover by creating a ratio between flow and pressure. In this mode, the flow is dependent on the system pressure p_{L}, equation 1.1.

$$q(p_{\mathrm{L}}) = \frac{V_{\mathrm{g,\,max}} \cdot n \cdot \tan\alpha(p_{\mathrm{L}}) \cdot \eta_{vol}}{\tan\alpha_{\mathrm{max}}} \tag{1.1}$$

A new control valve developed by Danfoss combines both control characteristics, allowing the driver to switch between load-dependent and load-independent control behaviour [1]. The control valve has the same interface as an EDC valve. Additionally, the essential control loops are working hydromechanical. In detail, this means:

- no additional actuator
- no additional sensors
- fast control behaviour

The basic idea of this 'hybrid control' was introduced in [1,4,5]. The current paper presents the design realization of this new type of control, compares it to existing controls and elaborates how each mode of the control offers specific advantages in certain applications. Chapter 2 and 3 provide the context by describing the designs of the load-independent and load-dependent controls. Chapter 4 explains the design and function of the hybrid control. Chapter 5 presents simulation results of drive scenarios on a vehicle-level in a realistic 3D environment. These simulations include the propel hardware and software, which is also a crucial component for the overall performance.

2. EDC FUNCTION (LOAD-INDEPENDENT / SPEED MODE)

The Electric Proportional Displacement Control (EDC) is a type of Displacement Control (DC) for closed-loop pumps. In an EDC system, swashplate angle of the pump, and therefore its displacement, is proportional to the input current. The flow output is ideally proportional to the pump displacement; however, volumetric efficiency deteriorates the precise relationship between the input signal and the resulting flow. Overall, when a variable pump with an EDC is used in a propel application, the vehicle speed is primarily dependent on the input signal of the operator, and the speed of the prime mover.

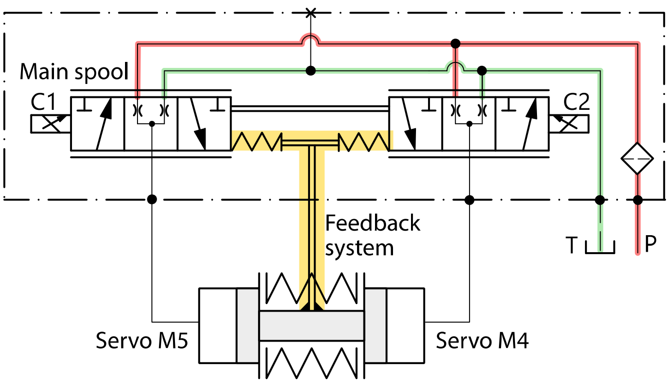

Figure 2.1: Schematic of EDC Control and servo system

The schematic of a Danfoss EDC control is depicted in Figure 2.1. The EDC incorporates a pair of proportional solenoids on either side of a common control spool, which is linked to the swashplate of the axial piston unit via a mechanical feedback system.

The solenoids convert the electrical input signal into force, moving the three-position, four-way control spool from its neutral position. This movement increases pressure on one side of a double-acting servo piston, which rotates the swashplate until the forces from the mechanical feedback and the electrical input are balanced.

This hydro-mechanical feedback system is also referred to as a servo system. Figure 2.2 shows a schematic diagram of a servo system. Only one solenoid and one piston surface is depicted, sufficient to control displacement of the swashplate in one direction. The feedback spring is a soft spring intended to produce a force on the main spool equivalent of the swashplate displacement. In steady-state condition the main spool is closed and the solenoid force in perfect balance with the force produced by the feedback spring. The servo spring is a much stiffer spring, to make sure the swashplate angle returns to neutral at zero current condition. The axial piston unit produces reaction moments on the servo system which are dominantly pressure and speed-dependent but also vary with displacement and temperature. The force on the servo piston caused by swashplate reaction moments is shown as F_{d}.

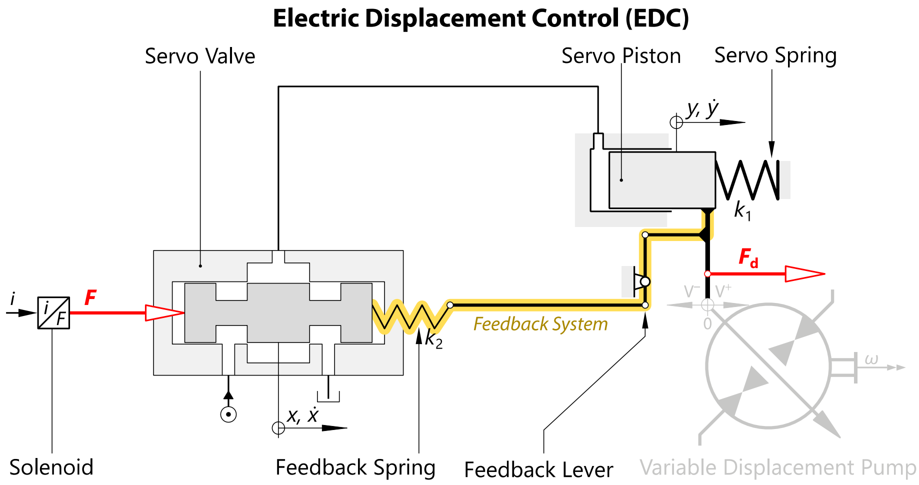

Figure 2.2: EDC with hydro-mechanical feedback system. The swashplate reaction force F_{d} is a disturbance.

3. NFPE FUNCTION (LOAD-DEPENDENT / TORQUE MODE)

The Non-Feedback Proportional Electric Control (NFPE) is a type of non-feedback control for closed-circuit pumps. It consists of two pressure control valves to control the pressure in each servo piston, see Figure 3.1. A simplified schematic is shown in Figure 3.2 with only one solenoid and a single servo cylinder chamber, which is sufficient to explain the action in one direction. It is clearly seen that there is no mechanical feedback between the servo cylinder and the control valve. The reaction force of the axial piston unit and the force of the servo springs result in a force F_{d} acting on the servo piston [2]. The preloaded servo springs produce a position-dependent force, and they are important for a fail-safe design to bring the pump back to neutral position. As a result, the servo system displacement is proportional to the valve current. However, because there is no mechanical feedback, the swashplate reaction moments can disturb the swashplate angle and render the pump load-dependent.

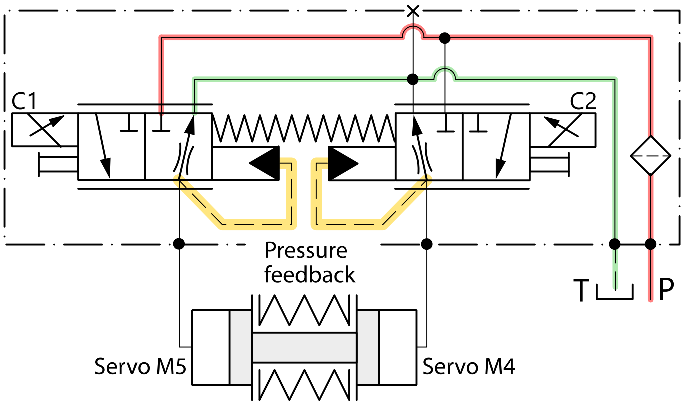

Figure 3.1: Schematic of NFPE Control and servo system

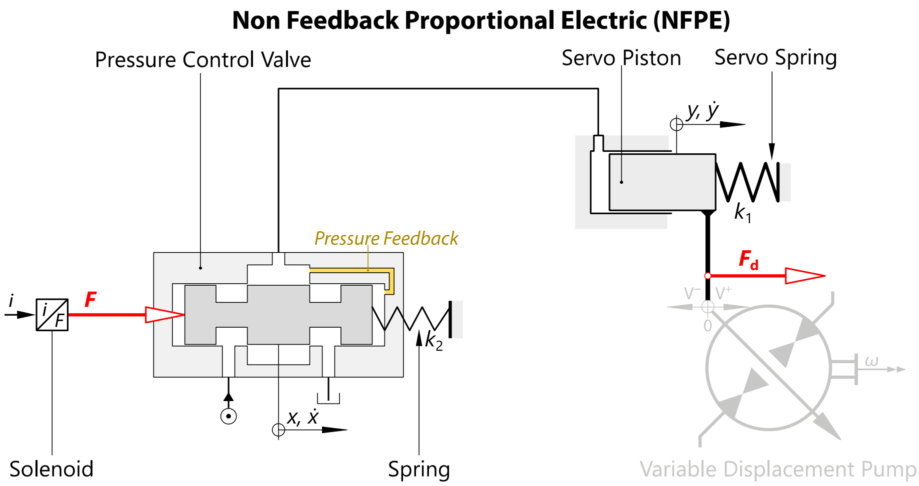

Figure 3.2: Simplified schematics of half of a non-feedback control system (NFPE). The valve realizes a pressure control. The servo piston finds an equilibrium position in relation to the external forces

If a variable pump with an NFPE control is used in a propel application, the oil flow and vehicle speed depend not only on the inputs of the operator. It is also influenced by the system pressure which affects the swashplate moments. The combination of system pressure and pump speed creates the load-dependent characteristic.

A pump with an NFPE control optimizes the use of available engine power by converting it into an ideal ratio of flow and pressure. This maximizes vehicle speed and tractive force without stalling the engine, making NFPE the preferred solution for applications with low engine power relative to the required corner power.

4. HYBRID LOAD CONTROL (HLC) FUNCTION

The patented Hybrid Load Control (HLC) is fundamentally an EDC with the added capability to create load-dependent behaviour, see Figure 4.1 [4,5]. It achieves this by allowing the servo pressure to superpose the mechanical feedback force. The primary difference between the HLC and a standard EDC is the control spool. It contains two nested inner spools. In their default positions, these inner spools open a passage for flow through an orifice cascade from the servo chambers M4/M5 to case. This orifice cascade acts as a pressure divider and lets the servo pressure produce a force on the control spool. On the side of the main spool which is pushed by the active solenoid the inner spool is pressed against the main spool and the orifice cascade is shut closed. It is the opposite side of the main spool, where pressure feedback can take place.

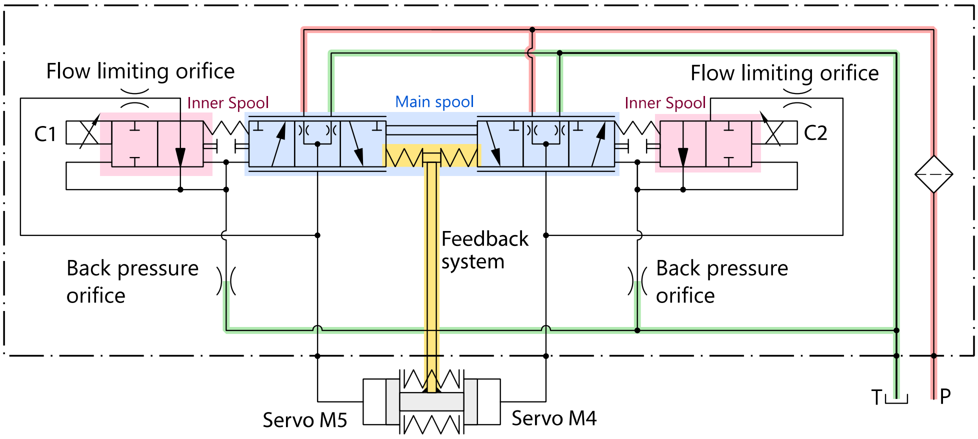

Figure 4.1: Schematic of HLC valve and system.

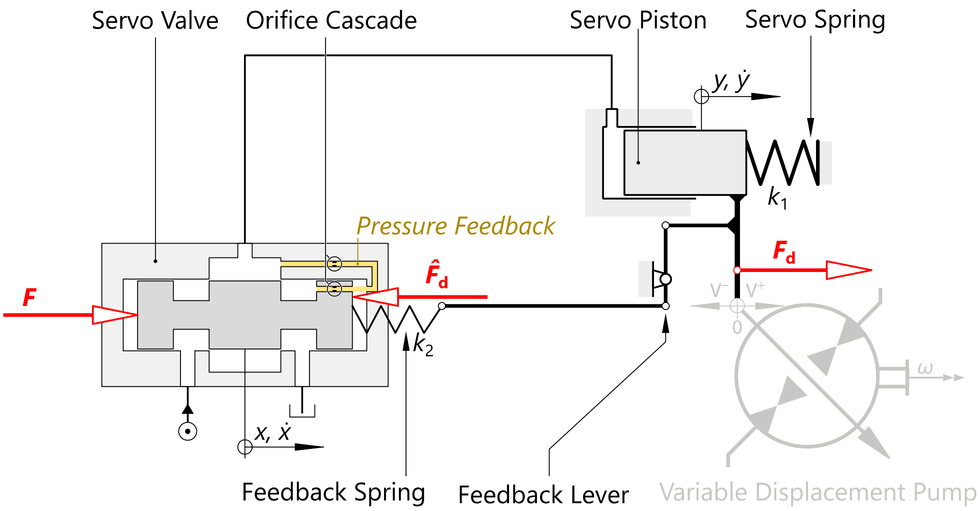

Figure 4.2: Simplified system schematic of half of HLC. In comparison with schematics of EDC and NFPE it includes both, pressure and position feedback. Pressure feedback can be switchted on or off.

The working principle can be explained by a simplified schematic in Figure 4.2. Here we consider the left side of the servo valve being pushed by the active solenoid. On the opposite side the orifice cascade (‘pressure feedback’) is seen, which transmits the servo pressure to the right side of the spool. It divides the servo pressure with a predefined ratio related to the ratio of orifice areas. The reduced servo pressure offsets the force produced by the solenoid and thereby reduces the set point of the servo control. Essential to the HLC is that the pressure feedback can be switched off. The orifice cascade is shut close, when the inner spool is slightly pushed against the main spool by the opposite (inactive) solenoid. It creates a disturbance for the active spool, but the force for closing the inner spool is small and can be easily compensated.

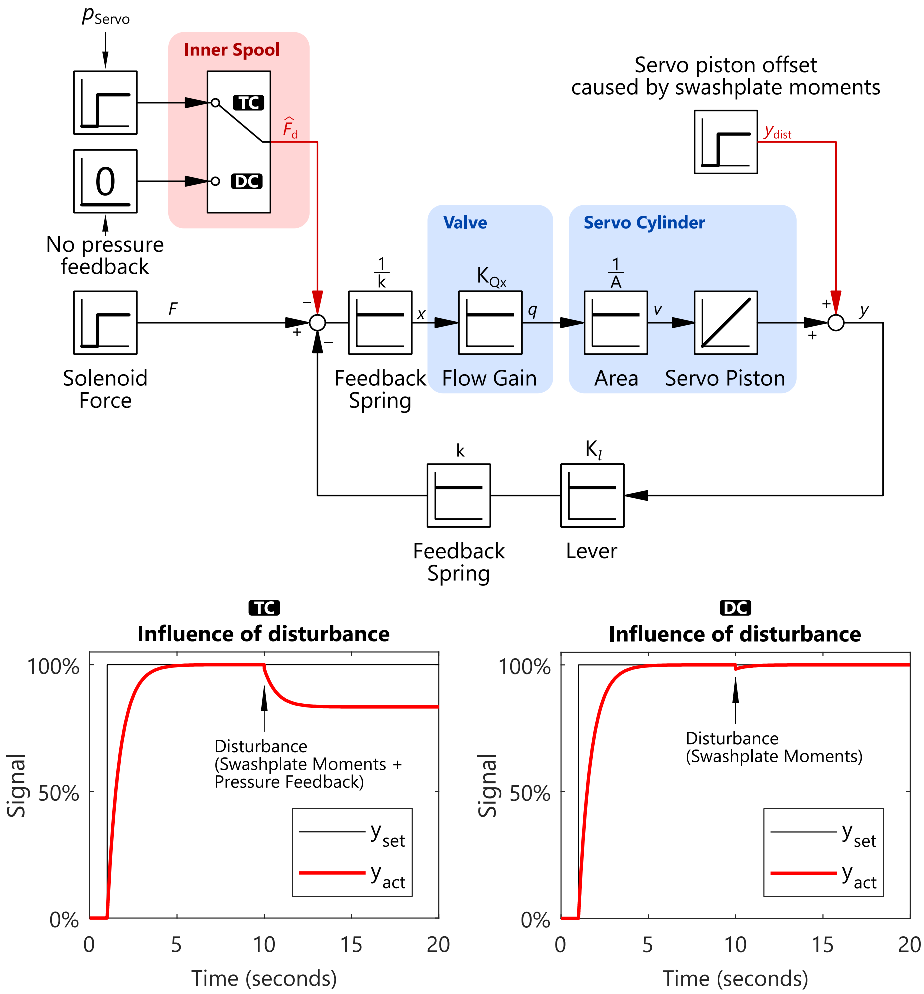

Figure 4.3: Block diagram of servo system with switchable pressure feedback and swash moment disturbance (top) and impact of disturbance (bottom).

Figure 4.3 shows a signal diagram of the control in the top. It can be used to derive the load stiffness with and without pressure feedback, compared in the bottom. For this analysis we assume that the valve dynamics are much faster than the dynamics of the servo piston. Therefore, the valve is represented simply by a flow gain K_{Qx} that is proportional to the sum of the force components from solenoid, and mechanical and hydraulic feedback. The force component from the hydraulic feedback is indicated by the top left signal branch in red. Its switchable nature is indicated by the 'Inner Spool' switch. It is open in the default position.

In load-dependent mode, the reduced servo pressure is applied to the valve spool as a disturbance force $\hat{F}_{\mathrm{d}}$. It is a function of A_{Spool} spool area, and α ratio of the pressure divider:

$$\hat{F}_{\mathrm{d}} = A_{\mathrm{Spool}} \cdot \alpha \cdot p_{\mathrm{Servo}} \tag{4.1}$$

This disturbance force offsets the solenoid force and reduces the setpoint. As the servo pressure increases, the offset causes a de-stroking of the servo piston.

The closed-loop set-point transfer function can be written as

$$\frac{Y(s)}{F(s)} = \frac{\frac{K_{\mathrm{Qx}}}{kA}\frac{1}{s}}{1 + \frac{K_{\mathrm{l}}K_{\mathrm{Qx}}}{A}\frac{1}{s}} = \frac{\frac{1}{K_{\mathrm{l}}k}}{\frac{A}{K_{\mathrm{l}}K_{\mathrm{Qx}}}s + 1} \tag{4.2}$$

Where $F(s)$, $Y(s)$ are the Laplace transforms of spool force and servo position. The factor $\frac{A}{K_{\mathrm{l}}K_{\mathrm{Qx}}}$ represents the time constant of this first order transfer function. The steady-state relationship between force F and servo position y is given by

$$\lim_{s \to 0} \frac{Y(s)}{F(s)} = \frac{1}{K_{\mathrm{l}}k} \tag{4.3}$$

Therefore, the steady state relationship between servo pressure and servo displacement can be expressed as

$$\lim_{s \to 0} \frac{Y(s)}{p_{\mathrm{Servo}}(s)} = \frac{A_{\mathrm{Spool}}\alpha}{K_{\mathrm{l}}k} \tag{4.4}$$

And the load stiffness can be given as the inverse.

In load-independent mode, there is no disturbance by the servo pressure on the control spool. Or, in other terms, the orifice ratio α is 0. Reaction forces from the rotating unit that act directly on the servo piston do not affect the closed loop control.

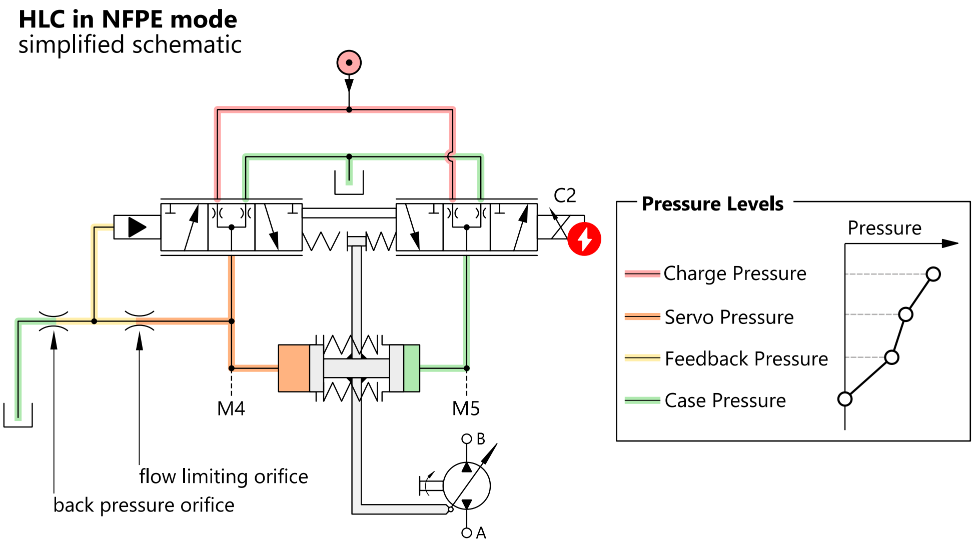

Figure 4.4: Pressure divider to superimpose reduced servo pressure on the valve spool.

Figure 4.4 shows a hydraulic schematic of the HLC, where solenoid C2 is activated. This leaves the orifice cascade on the opposite side open. Different pressure levels are indicated by colour. The servo pressure M4 is divided by the orifice cascade and produces a small force on the main spool. It opposes the force from solenoid C2. The set value is reduced, and as a result, the control behaves load-dependent.

To switch into load-independent (EDC) mode C1 is energized slightly with a current that does not affect the control action of C2 but is enough to close the orifice cascade via the small internal spool. The HLC does not require additional actuators or sensors to switch between modes. This is illustrated in Figure 4.5, which shows the control with both solenoids. The right solenoid is the active one and the left solenoid determines whether operation is in NFPE or EDC mode. The blue arrows indicate the current ranges when operating in load-dependent mode. The red arrows indicate the current ranges when operating in load-independent mode. The passive solenoid is given a small current that is below the dead-band threshold of the EDC control.

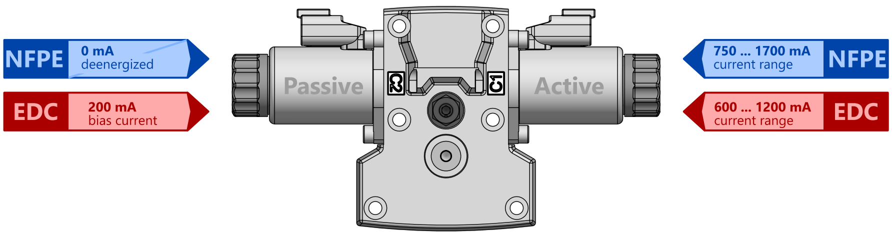

Figure 4.5: Actuation of Danfoss Hybrid Load Control

5. HLC OPERATION

Three driving scenarios are considered to illustrate the functionality of the HLC:

1. Driving a bucket into a pile of sand
2. Driving precisely over an obstacle
3. Downhill braking

For the first scenario, a load-dependent control characteristic is advantageous as this type of control prevents the diesel engine from stalling by de-stroking the pump. For the second scenario, a load-independent control characteristic is preferable because precise manoeuvring over an obstacle is required. A load-independent control characteristic regulates the displacement of the pump in a closed control loop, as mentioned above. The vehicle can thus maintain a nearly constant speed despite leaks. This enables precise manoeuvrability. In the third scenario, the braking behaviour between load-dependent control (NFPE) and HLC in load-dependent control mode is shown. It should be noted that load dependency is inactive during braking. During braking, the swashplate moment switches direction and now acts on-stroking. The control maintains the swashplate angle by pressurizing the opposite side. As can be seen from Figure 4.1 this opposite side cannot cause an offset to the servo spool because the orifice cascade is shut off by the active solenoid. To demonstrate the advantages of HLC in the scenarios mentioned above, a simulation was used that includes models of the hydraulic hardware, drive train components, a 3-D multi-body dynamic simulation of a vehicle and of the electrical control and software. Some parts of this real-time driver-in-the loop simulation setup are described also in [3]. The key data of the wheel loader and the drivetrain used in the simulation can be found in Table 1.

Table 1: Vehicle data.

Wheel rolling radius	0.6 m
Mass	8400 kg
Differential Gear ratio	16.51
Final differential gear ratio	15.24
Pump	H1P 89 cm³
Motor	H1B 160 cm³
Diesel engine max. power	103 kW

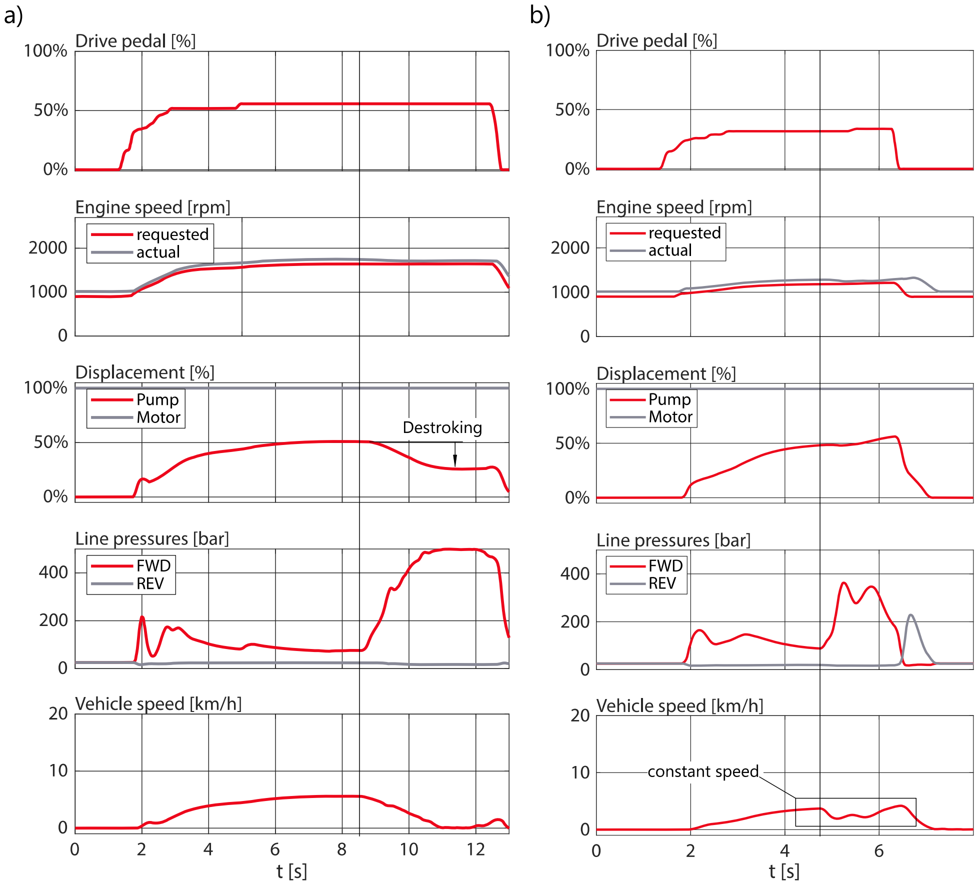

Figure 5.1: Simulation results showing two different driving scenarios with HLC operating in different modes. Left: Driving of bucket into pile (load dependent). Right: Precise maneouvre over obstacle (load independent)

Figure 5.1 shows simulation results for these driving scenarios. "Driving of bucket into a pile" (scenario 1) is shown on the left and "precise manoeuvre over obstacle" (Scenario 2) on the right. Considering scenario 1, the wheel loader approaches the pile of sand at almost constant speed. The drive pedal remains in approximately the same position until the bucket is full. As soon as the bucket reaches the pile of sand, the driving resistance increases, indicated by a steep rise in line pressure. Because of the servo pressure feedback, the pump de-strokes automatically and the diesel engine is protected from stalling.

In the second scenario, load-independent speed mode is selected because the driver wants to drive over an obstacle precisely and at almost constant speed. The corresponding signals are shown in Figure 5.1 on the right. As soon as the wheel loader reaches the obstacle, the load pressure increases. Due to the selected speed mode, the servo pressure cannot influence the control spool, and the vehicle speed remains approximately constant. The vehicle manoeuvres precisely over the obstacle.

The HLC provides a better braking performance, even in load-dependent mode. As stated before, the load-dependency is not active during braking. During braking the swashplate moments are on-stroking. Therefore, load-dependency, if it was active, would cause the pump to increase displacement during braking. This lowers the engine RPM and therefore reduces the available engine braking power. It can be demonstrated that with HLC, even in the load-dependent mode, braking performance is just as if the pump control was load-independent: fast and precise. Software can be used effectively to optimize the overrun operation of the diesel engine. Figure 5.2 shows a comparison of simulation results for downhill breaking between a vehicle that uses a standard NFPE pump control (left) and one that uses a HLC in load-dependent mode (right). The simulated vehicle is equipped with a 103 kW engine. Its braking power is the limitation for the achievable deceleration.

The driving scenario can be divided into three phases:

A. Acceleration on flat ground while approaching the incline,
B. Driving downhill a 10% slope,
C. Hydrostatic braking.

Braking starts at the same speed. As the upper plot in Figure 5.2 shows, with the HLC pump control, the software can better utilize the available engine braking power by controlling displacements to achieve maximum engine speed in the overrun operation. The vehicle with standard NFPE pump control stops after 9 s and 23.6 m, whereas the vehicle with HLC pump control stops after 7s and 20.8 m. It should be noted that the braking performance with NFPE control has also been optimized using software.

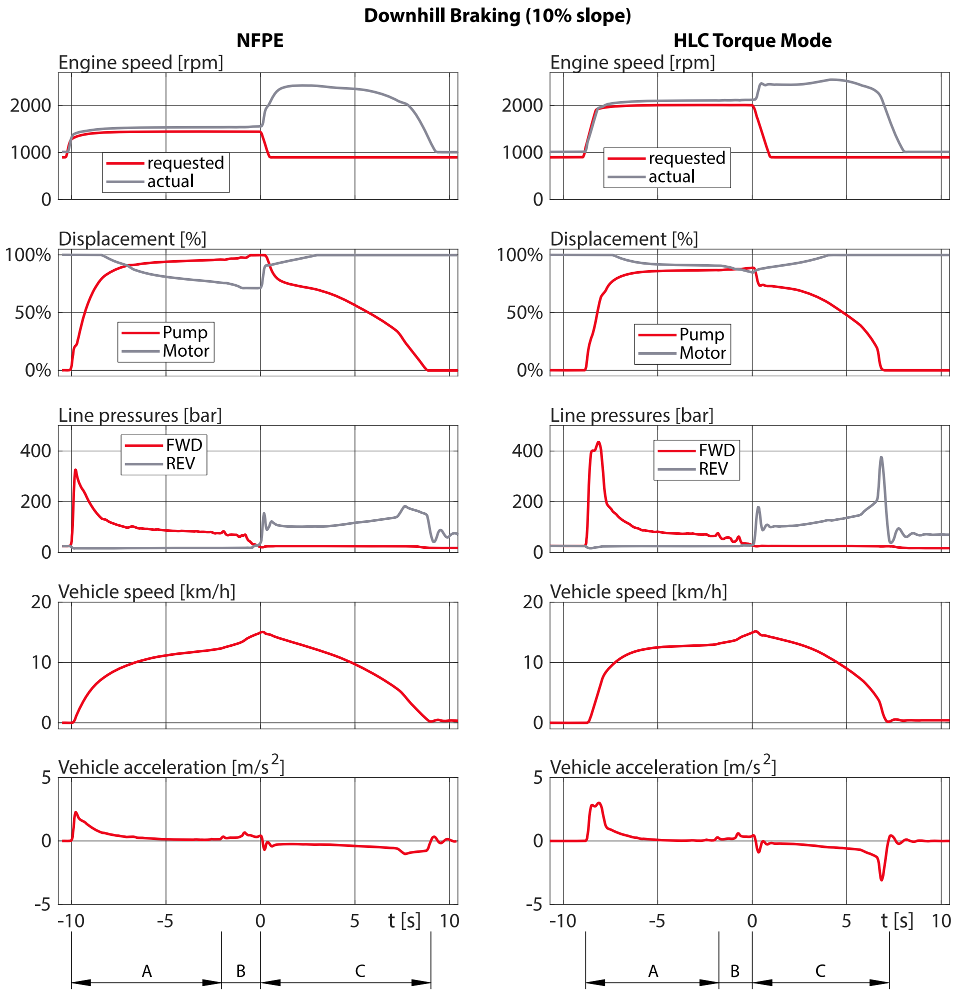

Figure 5.2: Simulation results of downhill braking. Left: Vehicle with standard NFPE pump control. Right: Vehicle with HLC pump control in load-dependent mode. A: acceleration on flat ground. B: driving downhill. C: hydrostatic braking.

Summary and conclusion

The new Hybrid Load Control combines the best of both worlds, load-independent and load-dependent hydrostatic driving. By specifically using the dead band for a new function and making use of unused space, it was possible to combine two control valves into one. No additional actuators or sensors were required, nor was any additional space needed. The components that determine the characteristics of the control loops were adjusted so that the control characteristics of the HLC in the torque control mode are very similar to those of the current NFPE pump control. Since the control loops are still hydromechanical, the responsiveness of the established control valves is achieved and has been demonstrated on Danfoss Application Development Center. These validation results will be shown in a future publication.

NOMENCLATURE

Abbreviation	***Description***
DC	Displacement Control
EDC	Electric Displacement Control
FWD	Forward
HLC	Hybrid Load Control
NFPE	Non-Feedback Proportional Electric
REV	Reverse
TC	Torque Control

Variable	***Description***	***Unit***
A	Area	m^2
F_{d}	Disturbance force (from swashplate moments)	N
$\hat{F}_{\mathrm{d}}$	Force from pressure feedback	N
k	Spring constant	N/m
K_{l}	Feedback link ratio	-
K_{Qx}	Flow gain	m^2/s
p_{Servo}	Servo pressure	Pa
s	Laplace variable	s^{-1}
V	Volume	m^3
$V_{\mathrm{g,max}}$	Maximum displacement volume	$\mathrm{m}^3/\mathrm{rev}$
x	Set point	m
x_{dist}	Disturbance displacement	m
y_{act}	Actual servo piston position	m
y_{dist}	Servo piston offset due to disturbance	m
y_{set}	Desired servo piston position	m

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Biographies

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Dr. Matthias Liermann currently is Senior Manager Dynamic Simulation for Hydrostatics at Danfoss Power Solutions based in Neumünster, Germany. He studied mechanical engineering at RWTH Aachen University and received a Ph.D. in mechanical engineering under supervision of Professor Hubertus Murrenhoff at the Institute for Fluid Power Drives and Controls at RWTH Aachen in 2008. From 2009 to 2017 Dr. Liermann was an Assistant and Associate Professor at the Maaroun Seemaan Faculty of Engineering and Architecture at the American University of Beirut, Lebanon.

Dr. Robert Rahmfeld is the Senior Director Engineering for Hydrostatics at Danfoss Power Solutions, being technically responsible for hydrostatic axial piston units. He is with Danfoss over 20 years, holds several patents, and is author or co-author of more than 60 scientific journal and conference papers. Dr. Rahmfeld completed his PhD on new displacement controlled hydraulic linear actuators at the Technical University of Hamburg-Harburg in 2002 under supervision of Professor Monika Ivantysynova, and his MSc in mechanical engineering at Duisburg University in 1996. He was also holding a postdoc position in fluid power for 2 years at the Technical University of Hamburg-Harburg and Purdue University, Indiana, USA. Dr. Rahmfeld represents Danfoss within the VDMA fluid power board and is also the chairman of the corresponding fluid power research fund with more than 60 member companies. He was awarded in 2024 by Purdue University with the *Monika Ivantysynova Medal* for technology advancements of hydrostatic units and their controls, and for leadership in fostering industry - academic initiatives to promote fluid power.