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System for Velocity Adjustment and Control of a Closed-Circuit Secondary Adjustment Hydrostatic Transmission

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Abstract: *This article focuses on the design and testing of a close-circuit secondary adjustment hydrostatic transmission. The interface between the acquisition board and the computer is done by LabView programming environment. The process is regulated by a PID controller. The system uses the controller output value to push the process variable toward the setpoint value. Control schematics include a feedforward multiply loop for more precise response.*

Keywords: *Secondary adjustment, hydrostatic transmission, closed-circuit, feedforward, PID controller*

1. Introduction

Over the past few decades, stationary and mobile machinery has become more automated and productive thanks to fluid power and motion control systems. The high power density, quick reaction, good controllability of movements, and affordable, direct rotary actuation using hydraulic motors are some of the main benefits of hydraulic systems. Different well-known and tested system designs are available to hydraulic system designers: power and motion control might be based on proportional valves, servo-valve, direct pump, or secondary control [1].

In traditional hydraulic valve control systems, to vary the flow rate supplied by the pump, needle valves that shrink the flow section are used. A downside of those types of equipment is that they lead to an increase of pressure in the system that causes the relief valve to open, high-pressure flow rate enters the tank and causes futile power use. Reduced use of the relief valve can decrease power consumption and increase system performance. This can be obtained by using a variable displacement pump, motor or both, rather than a relief valve in response to load demands (higher or lower velocity or torque). This is where hydrostatic transmissions intervene, allowing hydraulic power to be transferred easily from one point to another in a controlled manner. With other types of drives, obtaining such high requiring demands is very difficult or even impossible. These performances have been made feasible by the control of hydrostatic drives utilizing analog and digital electronic equipment in closed open-loop control systems [2, 3].

The closed-circuit hydrostatic (HST) drives used in industrial equipment stand out for their high power density, improved controllability, and wide speed-torque range. To create its design principles and components, the HST drive needs a steady-state performance analysis based on operational characteristics. Without creating prototypes, the appropriate HST components (pumps and motors) can be chosen for investigation based on the projected performance of the drives [4].

An HST's performance qualities depend on how it is set up, including whether it has a fixed or variable displacement pump, motor, or both. These setups and their individual performance characteristics are summarized in Figure 1 and described below. Depending on its configuration, the HST can drive a load from full speed in one direction to full speed in the opposite direction, with infinite variation of speed between the two maximums—all with the prime mover operating at its optimum speed. A fixed-displacement pump drives a fixed-displacement motor in the most basic hydrostatic transmission (Fig. 1A). Although this transmission is cheap, it only has a few applications, mainly because more energy-efficient alternatives to it exist. The pump must be sized to operate the motor at a set speed under full load since pump displacement is fixed. Fluid from the pump output flows over the relief valve when running at full torque is not necessary and heat is wasted as a result.

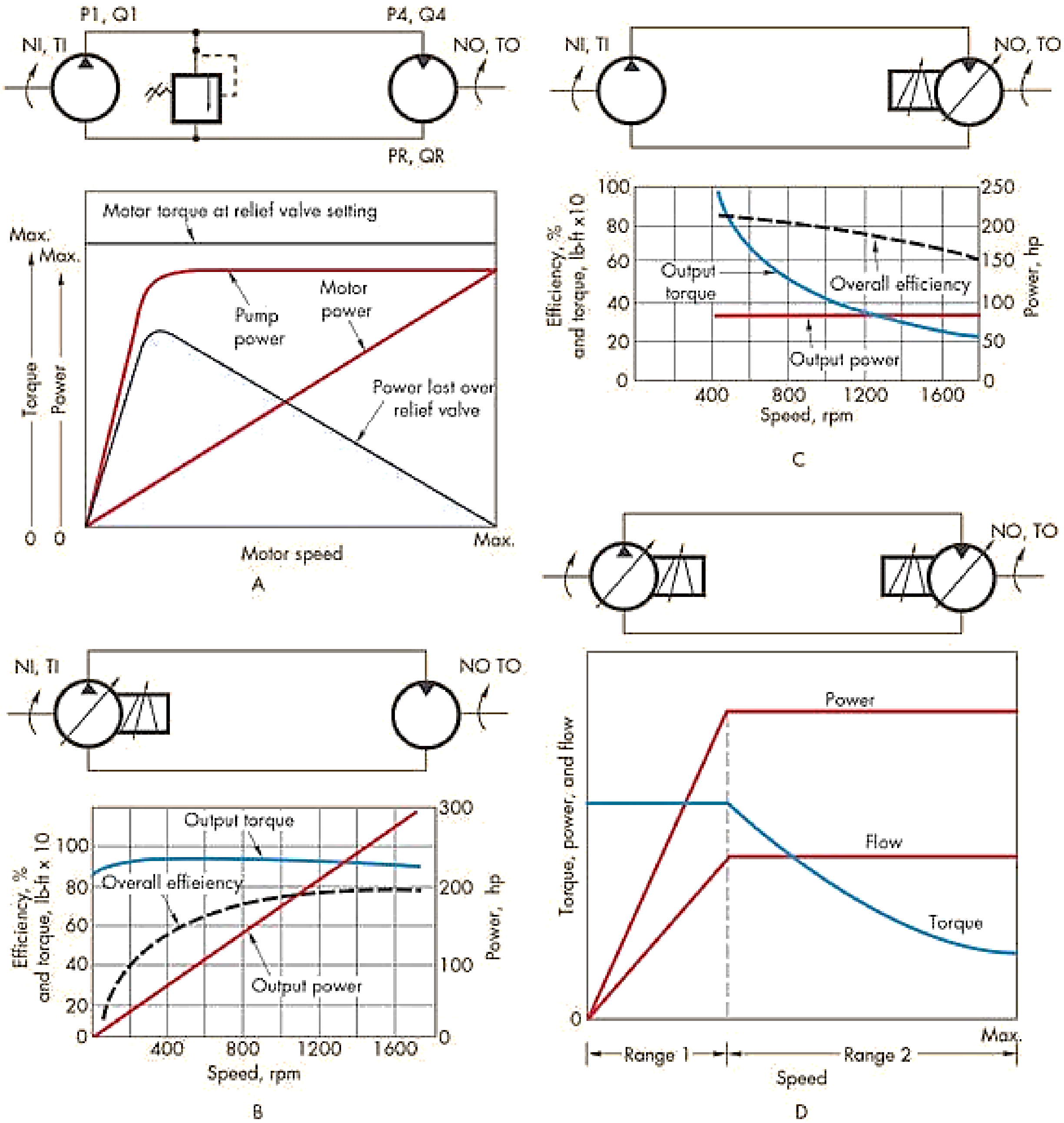

Fig. 1. Types of adjustment in closed hydrostatic transmissions – adapted from [4]

Primary adjustment hydrostatic transmission (PAHT): A constant torque transmission is produced by switching from a fixed displacement pump to one with a variable displacement (Fig. 1B). Because torque is solely dependent on fluid pressure and motor displacement, torque output is constant at any speed. Motor speed rises or falls as pump displacement increases or decreases, although torque is largely constant. As a result, power rises as pump displacement does.
Secondary adjustment hydrostatic transmission (SAHT): Transmissions that give constant power are made by combining a fixed-displacement pump and a variable-displacement motor (Fig. 1 C). Power delivered remains constant if flow rate to the motor is constant and motor displacement is changed to keep the ratio of speed to torque constant. Motor displacement reduction raises motor speed while lowering torque, which together maintains constant power.
Mixt adjustment hydrostatic transmission (MAHT): Figure D depicts a variable-displacement pump and motor [5].
Two ranges of adjustment are shown by the curves in Fig. 1D. Pump displacement reaches from zero to maximum while the motor displacement remains constant at its peak in "Range 1". Pump

displacement growth does not affect torque; however, it does affect power and speed. The second "Range" starts as soon as the pump achieves its maximum displacement and continues as the motor's displacement falls. While speed rises across this range, power stays constant. Torque declines. (Theoretically, the motor speed might be endlessly raised, but, in practice, dynamics would be the limit).

The secondary adjustment principle, which works in a regime of quasi-constant pressure, assumes the existence of a SAHT capability, which, when operating at a nominal pressure, seeks the appropriate volumetric capacity to maintain a constant rotation speed [6]. A fast-acting electro-hydraulic servomechanism is used to vary the displacement of the axial piston motor and is controlled by a command unit. The command unit automatically calculates the torque necessary - in just a few milliseconds - to ensure that the rotation speed is kept to the set point.

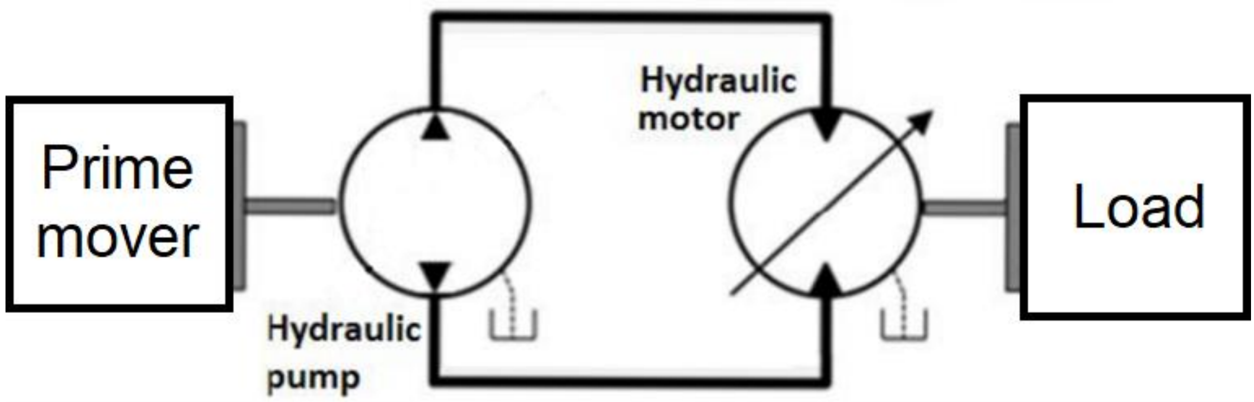

Fig. 2. Simplified schematic of hydrostatic drives with control of the secondary unit – adapted from [7]

Secondary-control drives avoid the need to throttle down the flow rate, as is the case in typical open systems, even when several utilizing units with different loads are connected to the system. By adjusting the machine's volume displacement to the loading situation, the power drain or energy return to the supply network is managed to match needs.

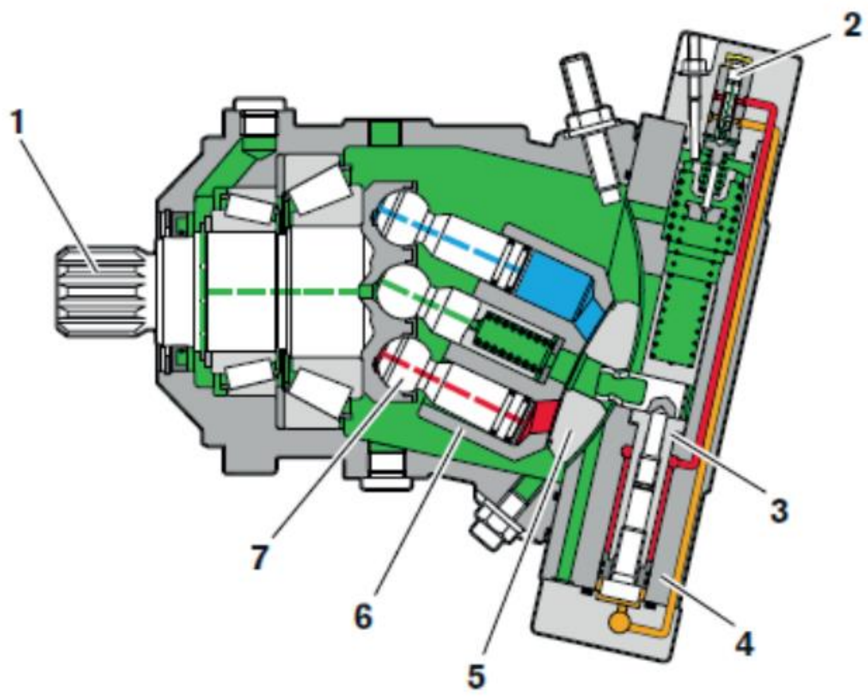

Fig. 3. Cross-section of the A6VM variable displacement axial piston hydraulic motor by Bosch Rexroth
1 – driving shaft, 2 – control piston, 3 –piston stroke control unit, 4 – casing in which the control piston is placed, 5 – lens plate, 6 – cylider block, 7 – piston [9]

This explains why the number of units acting as pumps or motors can be linked in parallel without affecting the system's energy balance, even in open systems.
The performance of secondary-control drives is unaffected by even extensive distances between secondary and primary units that must be traversed utilizing lengthy hydraulic lines [8]. A major advantage of using hydrostatic transmissions is the ability to vary the speed and torque output with a great degree of accuracy, even when load requirements are changing quickly.
The chosen hydrostatic motor for this system is A6VM Bosch Rexroth axial piston motor, depicted in Fig. 7. The hydraulic motor's inflow and outflow spaces are connected to the pistons 7 inside the cylinder block 6, which are rotating and together with the cylinders constitute the working chamber, through holes in the cylinder block 6's face space. The working volume expands as the plunger moves to the left, joining with the inflow (pressure) area to get filled with liquid. When the pistons moves to the right, the working volume shrinks and the liquid is supplied into the outflow area.
While the motor is running, a portion of its chambers is filled with the working fluid, while fluid from the remaining chambers is discharged into the outflow conduit. The spherical lens plate 5, with spaces connected to the channels in the motor casing and the input and outflow holes, was attached to the motor. Leaks from the high-pressure area into the low-pressure area occur in the motor, particularly in the lens plate 5. In this instance, the directional valve 2 spool position that enables the displacement of the motor to vary is determined by a PID controller.
Regarding the largest speed fluctuations per second as a function of power, figure 3 compares hydrostatic machines with secondary adjustment to machines that are electric and electric machines that are adjustable. The comparative standards were derived from the data in manufacturer leaflets. The B/B1 field in the graphic depicts the secondary regulation's present state (double logarithmic scale).

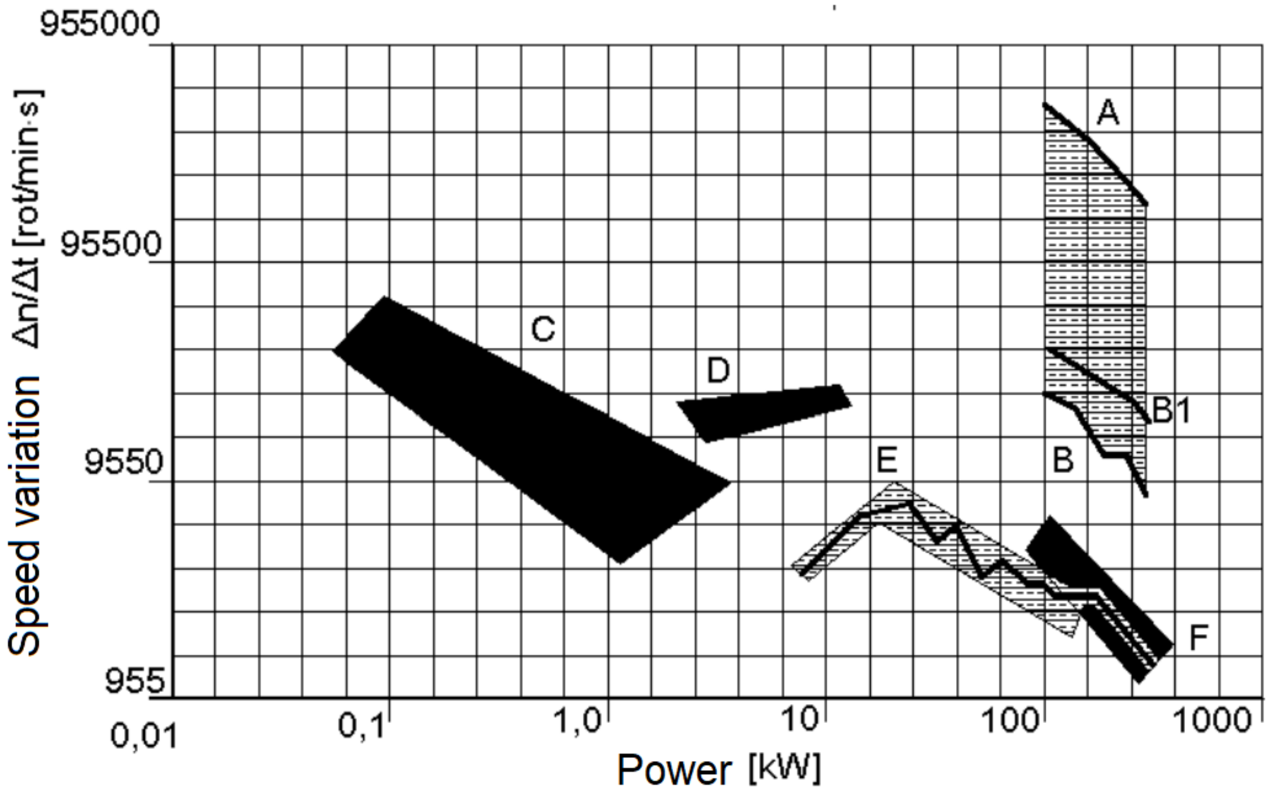

Fig. 4. Speed variations (max.) for various engines as a function of power
A - hydraulic engine (theoretical possible value); B - hydraulic motor (speed adjustment - secondary adjustment A4 VEL); B1 - hydraulic motor (speed adjustment - secondary adjustment A4 VEL); B1 - hydraulic motor (speed adjustment - secondary adjustment A4 VEL). hydraulic motor (speed regulation - secondary regulation A4VS); C - DC servo motor; D - hydraulic motor (speed regulation - secondary regulation A4VS) three-phase servomotors; E - three-phase motor (with frequency regulation); F - three-phase motor (with air venting external ventilation) [3]

2. Material and Method

This chapter is composed of two parts, the mathematical model of the static control characteristics of the hydraulic motor rotary and Hydraulic transmissions with a secondary adjustment experimental system.

2.1 Mathematical model of the static control characteristics of the hydraulic motor rotary

Usually, rotary hydraulic machines can function as a pump as well as a motor. Hydraulic rotary motors are available with fixed or adjustable capacity for a variety of speed and torque ranges, with fast, semi-fast, or slow operation.
Rotary motors, which are used as drive elements, have unique structural and functional characteristics, and some are even special machines. Because a relatively wide range of motors is used in practice, each with its own set of constructional and functional principles, as well as energy and dynamic performances, the most accurate knowledge of the following sizes, as well as the corresponding control methods, is required.
Given its position and role in a hydrostatic transmission, the hydraulic motor can be thought of as an energy converter that accepts energy from a positive displacement pump and converts it into mechanical energy applied to a load, as shown in Figure 5.

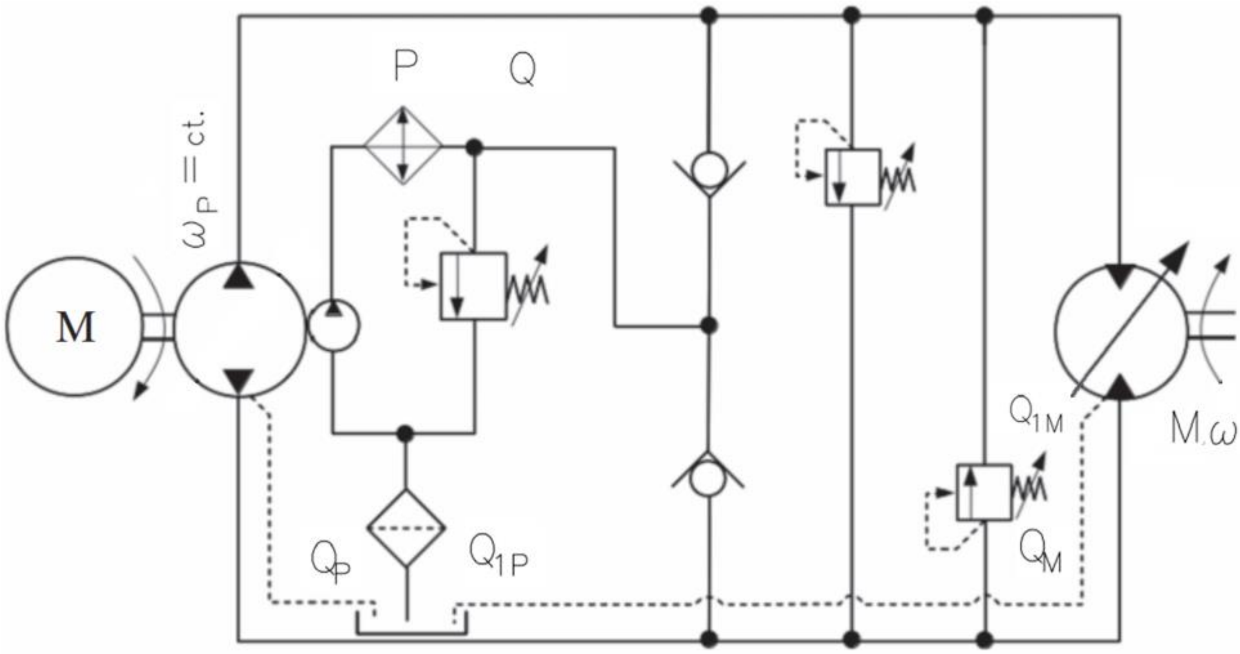

Fig. 5. Schematic diagram of hydraulic transmission with secondary adjustment – adapted from [10]

We will disregard the compressibility of the working fluid and consider flow losses if the system runs only as far as the pressure relief valve in the system may be opened. The equations characterizing stable equilibrium under these circumstances are:

$$Q = \boldsymbol{\omega}_p \cdot \boldsymbol{q}_p - \boldsymbol{\alpha}_p \cdot p = \boldsymbol{\omega}_M \cdot q_M + \boldsymbol{\alpha}_M \cdot p \tag{1}$$

$$M_M = \eta_{mM} \cdot q_M \cdot p \tag{2}$$

These equations lead to the following generic equation for the mechanical characteristic of the engine:

$$\boldsymbol{\omega}_m = \boldsymbol{\omega}_p \cdot \frac{q_p}{q_M} - \frac{\boldsymbol{\alpha}_p + \boldsymbol{\alpha}_M}{q_M^2 \cdot \boldsymbol{\eta}_{mM}} \cdot M_M \tag{3}$$

Where:

$$q_p = q_{p\,max} = constant$$

$$\omega_p = q_{p\,nominal} = constant$$

As the engine flow rate increases drops, hydraulic motor capacity command (q_M) results in mechanical characteristics that are linear but with an increasing slope. The mechanical properties no longer rely on torque and become straight parallel to the torque axis if flow losses in the system are ignored.
By lowering the particular engine flow q_M from $q_M = q_{M\,max}$ to the value $q_M = q_{M\,min}$ corresponding to the maximum speed M_{max}, without going above the maximum mechanical power, the secondary adjustment enables the achievement of greater speeds than the.
The examination of the relationship demonstrates that the dependence of the mechanical feature on the engine specific flow rate is highly non-linear (3), therefore, linearization of the mechanical properties around a static operating point is required for the implementation of a system for controlling the rotational speed of a load utilizing hydraulic motor capacity control.
The linearization of mechanical characteristics is obtained by applying the relation:

$$\boldsymbol{\Delta\omega}_M = \left(\frac{\boldsymbol{\partial\omega}_M}{\boldsymbol{\partial} q_M}\right)_{\mathbf{0}} \cdot \boldsymbol{\Delta} q_M + \left(\frac{\boldsymbol{\partial\omega}_M}{\boldsymbol{\partial} M_M}\right)_{\mathbf{0}} \cdot \boldsymbol{\Delta} M_M \tag{4}$$

After calculating the derivatives from (4) we obtain:

$$\boldsymbol{\Delta\omega}_M = -\left(\frac{\boldsymbol{\omega}_p \cdot q_p}{q_{M0}^2} - \frac{\mathbf{2}\cdot\boldsymbol{\alpha}\cdot M_{M0}}{\boldsymbol{\eta}_{mM}\cdot q_{M0}^3}\right)\cdot \boldsymbol{\Delta} q_M - \frac{\boldsymbol{\alpha}}{\boldsymbol{\eta}_{mM}\cdot q_{M0}^2}\cdot \boldsymbol{\Delta} M_M \tag{5}$$

Where:

$\alpha = \alpha_p + \alpha_M$ –total flow rate loss of the system

Using the notation:

$$K_1 = \frac{\boldsymbol{\omega}_p \cdot q_p}{q_{M0}^2} - \frac{\mathbf{2}\cdot\boldsymbol{\alpha}\cdot M_{M0}}{\boldsymbol{\eta}_{mM}\cdot q_{M0}^3} \tag{6}$$

And

$$K_{\mathbf{2}} = \frac{\boldsymbol{\alpha}}{\boldsymbol{\eta}_{mM}\cdot q_{M0}^{\mathbf{2}}} \tag{7}$$

The linearized mechanical characteristic of the engine can be written in the form:

$$\boldsymbol{\Delta\omega}_M = \frac{\boldsymbol{\omega}_p \cdot q_p}{q_M^2} - K_{\mathbf{2}}\cdot \boldsymbol{\Delta} M_M \tag{8}$$

It is essential to understand the fluctuation of the amplification factor K1 as a function of the variable independent q_M, or the graph of the function, in order to assess the dynamic stability of the speed control system.

$$K_{\mathbf{1}}(q_M) = \frac{\mathbf{Q}_{tp}}{q_M^2} - \frac{\mathbf{2}\cdot\boldsymbol{\alpha}\cdot M_M}{\boldsymbol{\eta}_{mM}\cdot q_M^3} \tag{9}$$

The derivative's equation gives rise to the following mode of variation for this parameter:

$$\frac{dK_1}{dq_M} = -\frac{\mathbf{2}\cdot\mathbf{Q}_{tp}}{q_M^3} + \frac{\mathbf{6}\cdot\boldsymbol{\alpha}\cdot M_M}{\boldsymbol{\eta}_{mM}}\cdot\frac{\mathbf{1}}{q_M^4} \tag{10}$$

Considering the torque in the M_M expressed as a function of pressure, the following results are obtained:

$$\frac{dK_1}{dq_M} = -\frac{\mathbf{2}}{q_M^3}(\mathbf{Q}_{tp} \cdot \mathbf{3} \cdot \boldsymbol{\alpha}\mathbf{p}) \tag{11}$$

Where:

$$\alpha p_{max} \leq 0.02 \cdot Q_{tp}$$

Results:

$$Q_{tp} - 3 \cdot \alpha p > 0$$

Implies:

$\frac{dK_1}{dq_M} < 0$ in any point of the system.

Therefore, the amplification factor K_1 decreases continuously with increasing specific flow rate q_M of the hydraulic motor, as shown in figure 5.

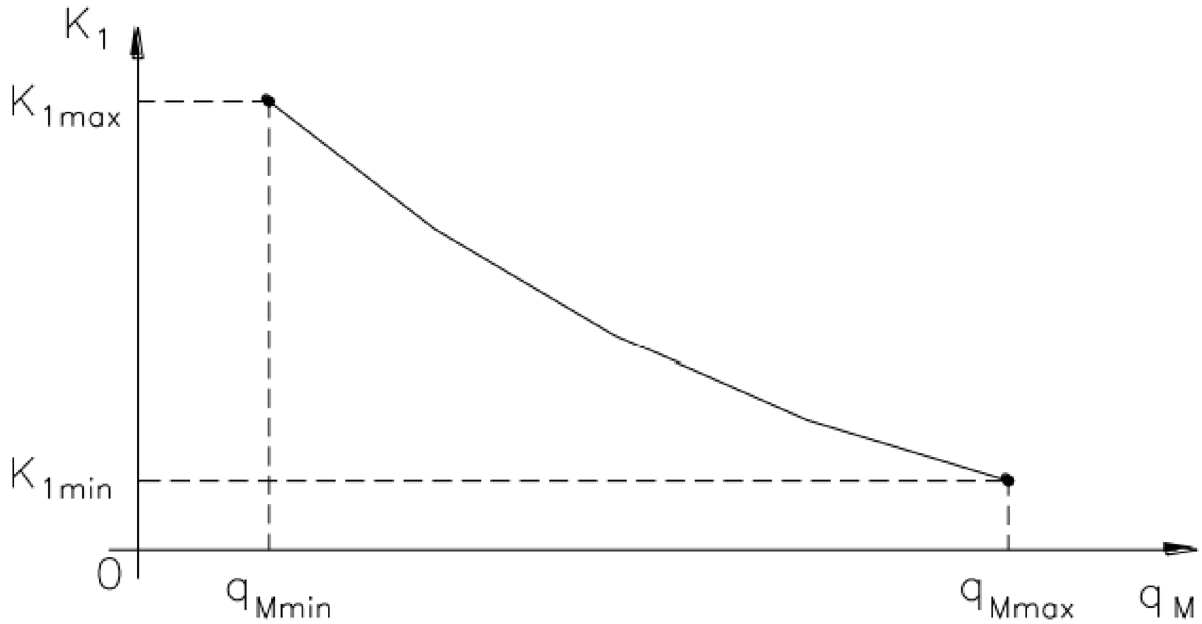

Fig. 6. Variation of amplification factor versus flow rate specific hydraulic motor

The linear displacement x of a moving component, which alters the particular flow rate in accordance with the relation: An electrohydraulic servomechanism for regulating the specific flow to the motor contains this output variable.

$$q_M = q_{M\,max} - K_{qx} \cdot \boldsymbol{x} \tag{12}$$

Where:

K_q – $transfer\ factor\ determined\ by\ kinematic\ elements\ and\ of\ the\ driven\ mechanism$

2.2 Hydraulic transmissions with secondary adjustment experimental system

In figure 7 is presented system on which the experimental results have been produced.
The rotation speed of the motor is controlled such that it is reached at the network pressure, regardless of the load pressure. To that end, the displacement of the motor must be changed until a balance is created between the motor torque and the load, and the desired rotation speed is reached at the same time.

Fig. 7. Experimental stand for the study of the closed-loop transmission with secondary adjustment

The system is composed of two main hydraulic lines that run from the pump to the motor (Branch "A") and back to the pump (Branch "B") as shown in figure 8. Each component either supplies fluid to both hydraulic lines or drain it. Mechanical work is generated by the triphasic electric motor E_M that is transmitted to the hydrostatic primary pump P_P through an elastic coupling, which is further connected to a compensation pump C_P that prevents the apparition of cavitation in the primary pump. Primary pump PP generates a flow rate that is directed to the hydrostatic variable displacement motor H_M. By varying the bent axis angle of the axial pistons of the HM, a larger angle of the pistons produces an increase in velocity, while a smaller angle leads to an increase in the pressure of the system and a drop in power. The displacement of the H_M is accomplished using a hydraulic linear motor, which positions is controlled using a 3/2-way proportional directional valve. The hydraulic linear motor is connected to H_M lens plate, which shifting in position leads to a change in the capacity of the motor.
To prevent pressure bursts that could lead to potential damage in the system, relief safety valves have been used, preventing the main branches to develop pressures bigger than 350 bars, case in which the oil is discharged to the tank.
The hydrostatic variable displacement motor H_M is further linked to the load pump L_P, in between the two is placed a velocity sensor that sends feedback to the acquisition board DAQ NI 6211. The L_P output is connected to a proportional relief valve that shrinks proportionally the flow rate section, which leads to a boost in the pressure and respectively in torque in L_P. The increase of L_P torque is transferred to the H_M, were pressure rise in "A" branch.
For precise measurement, 2 pressure sensor are mounted on both branches of the system and a flow rate sensor is mounted on branch "A". The values registered by the sensor are gathered by the DAQ NI 6211 acquisition board that also communicates with the PC. From the acquisition board, control signal is sent to the REXROTH VT11724 analog amplifier module that actuates the H_M displacement control susbsystem's proportional valve.
The software used to interface the DAQ NI 6211 acquisition board with the PC is LabView. The PID controller used to control the proportional electromagnet of the hydrostatic motor displacement sub-system is emulated by the LabVIEW programming environment, shown in figure 9, and the system then uses the controller output value to push the process variable toward the setpoint value. The PID controller parameters are generated based on an input/output signal. The control schematics include a feedforward multiply loop for a more precise response. In the context of a PID controller, the term "feedforward" refers to a measurement of the projected future error based on the current error and the system's known behaviour. Feed-forward control is a process to minimise the influence of interference in control circuits. The correcting variable is subjected to a disturbance variable which compensates for interference present in the controlled subsystem by estimating the disturbance variable with an observer and connecting it inversely to the feedback controller. The feedforward is multiplied by the proportional, integral, and derivative gains in this instance to determine the output of the controller, which is then used to adjust the system in order to decrease the error and bring it closer to the desired setpoint and provide a more precise and

prompt response to changes in the system, which can help improve the overall performance of the control system.

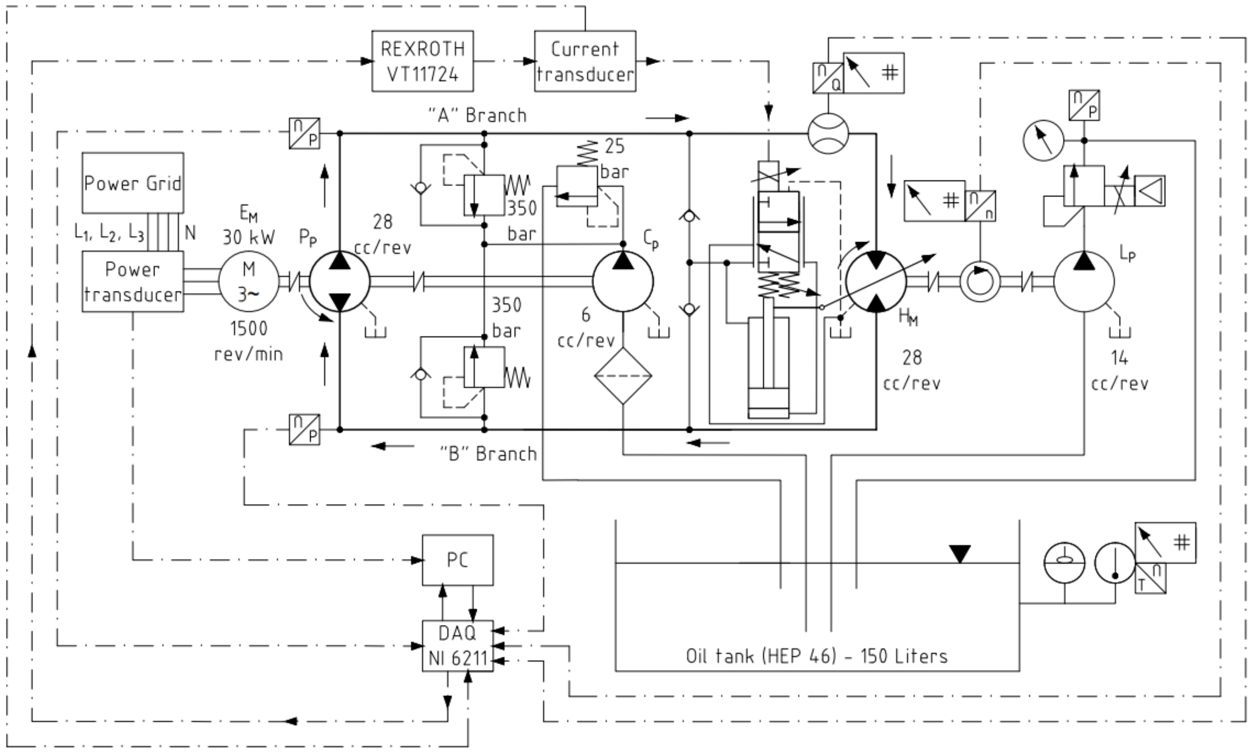

Fig. 8. Hydraulic scheme of the experimental stand for the study of the closed-loop transmission with secondary adjustment and of the servo-mechanism for regulating the motor displacement

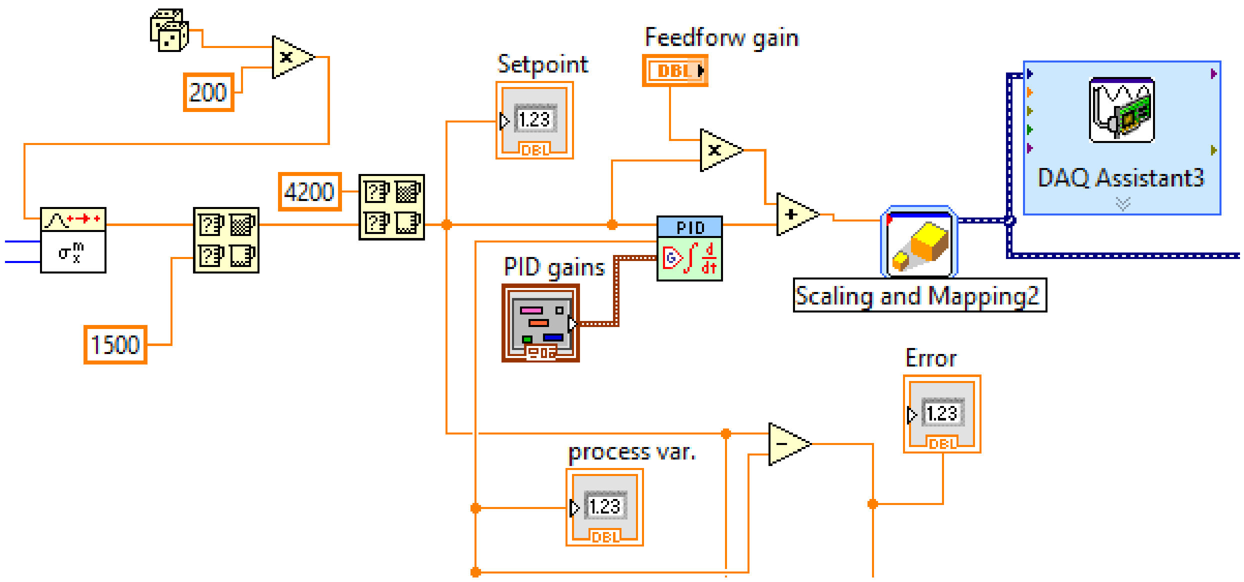

Fig. 9. LabVIEW graphical programming environment network for PID controller with feedforward

3. Experimental results

This chapter presents the results of the experimental testing of the system as described in the chapter 2.2. To vary the load, respectively the torque of the hydrostatic motor H_M, on the load pump L_P circuit, the proportional relief valve vary the flow cross-section. As a result the torque obtained between H_M and L_P is inverse proportional to the opening of the proportional relief valve.

To highlight the performances of the system and the fast response of the control part, the graph presented are over a small time period (25 seconds).
As one can see in the figure 10, the rise time period took 0.75 seconds, after the overshot peak at 1.5 seconds, at 2.5 seconds until the end of the shown period of time, the achieved velocity remained in the interval of the steady-state error within the tolerance band.

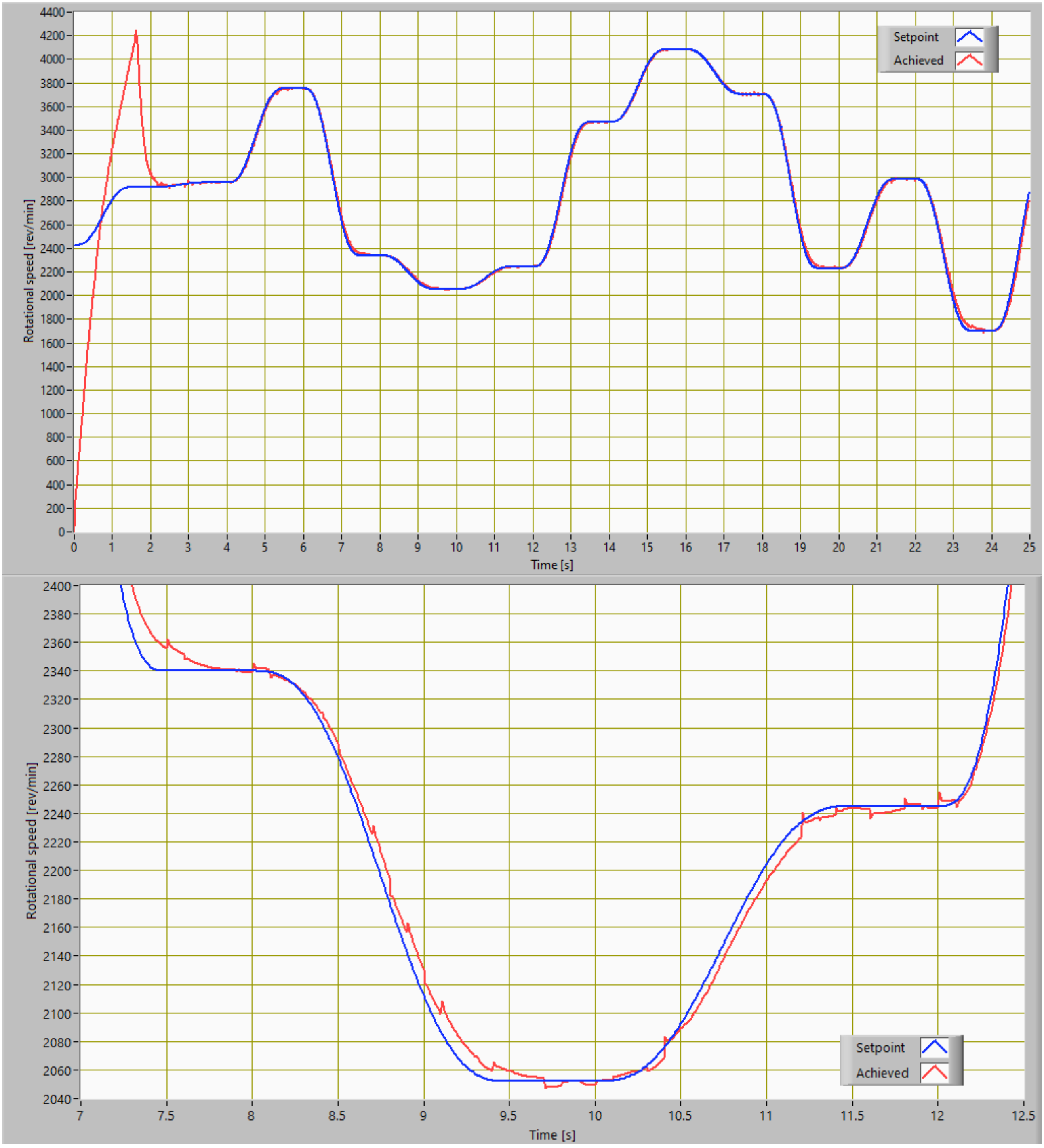

Fig. 10. The setpoint compared to the achieved speed over time

Figure 11 presents the torque produced by the load pump L_P used to mimic the torque variation in an industry application of the SAHT (for example automotive industry). As one can see, the torque varies between 4 and 19 [Nm] values determined by the flow section of the proportional relief valve.

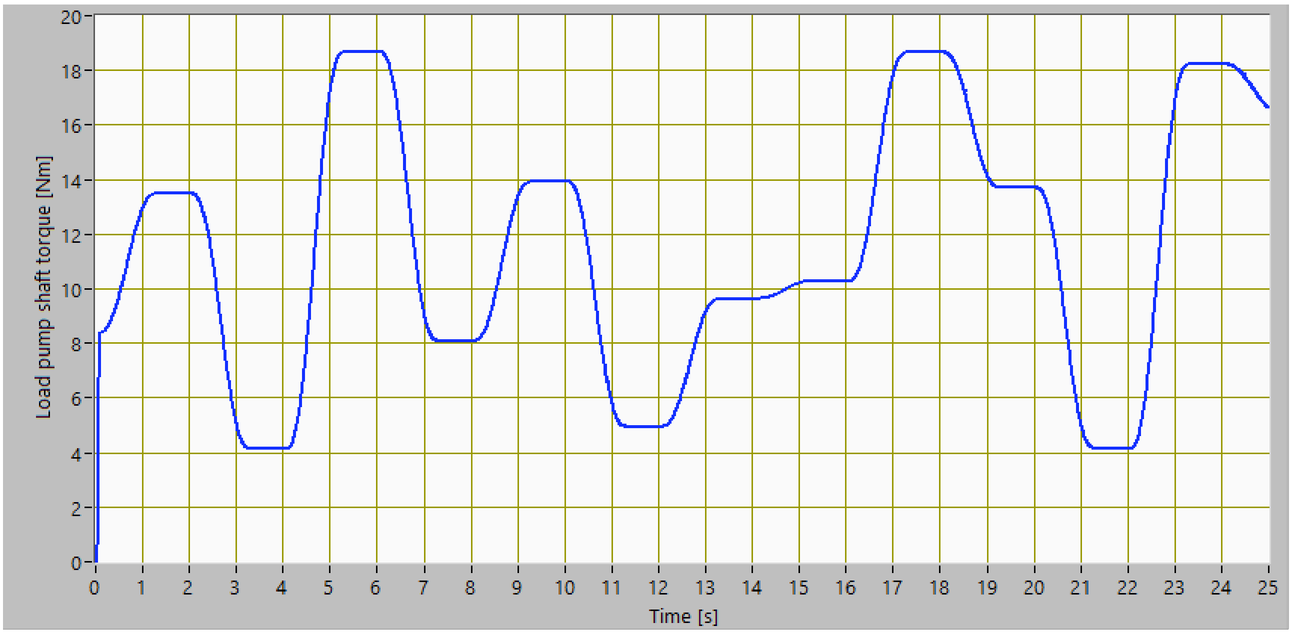

Fig. 11. The load pump shaft torque over time

Figure 12 shows the differential pressure in closed circuit over time (pressure in branch "A" excluding pressure in the branch "B"). This characteristic underlines the direct effect of pressure rises and drops in the load pump L_P to the two main branches of the hydrostatic closed-circuit and shows that smaller pressure in the L_P system (respectively lesser torque) leads to bigger pressure drops in the hydrostatic system. As the pressure in L_P circuit rises, so does the pressure drop in the hydrostatic closed-circuit.

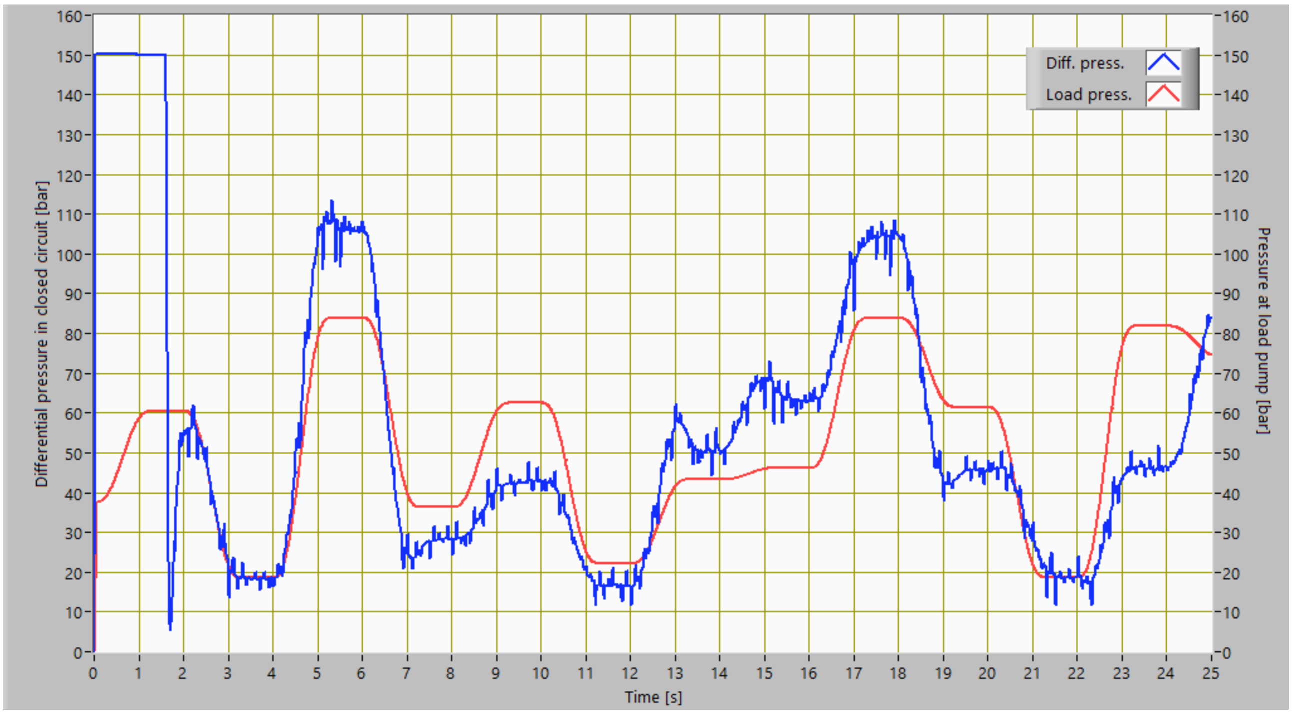

Fig. 12. The load pump shaft torque over time

Figure 13 presents the instantaneous control error and a detail of it. Here the credit of the feedforward multiply loop combined with the PID controller are seen, as the error is almost linear. Compared with the velocity from figure 10, the instantaneous error band is less than ±5%, beside the error generated by the overshot, and also a very fast system response to error.

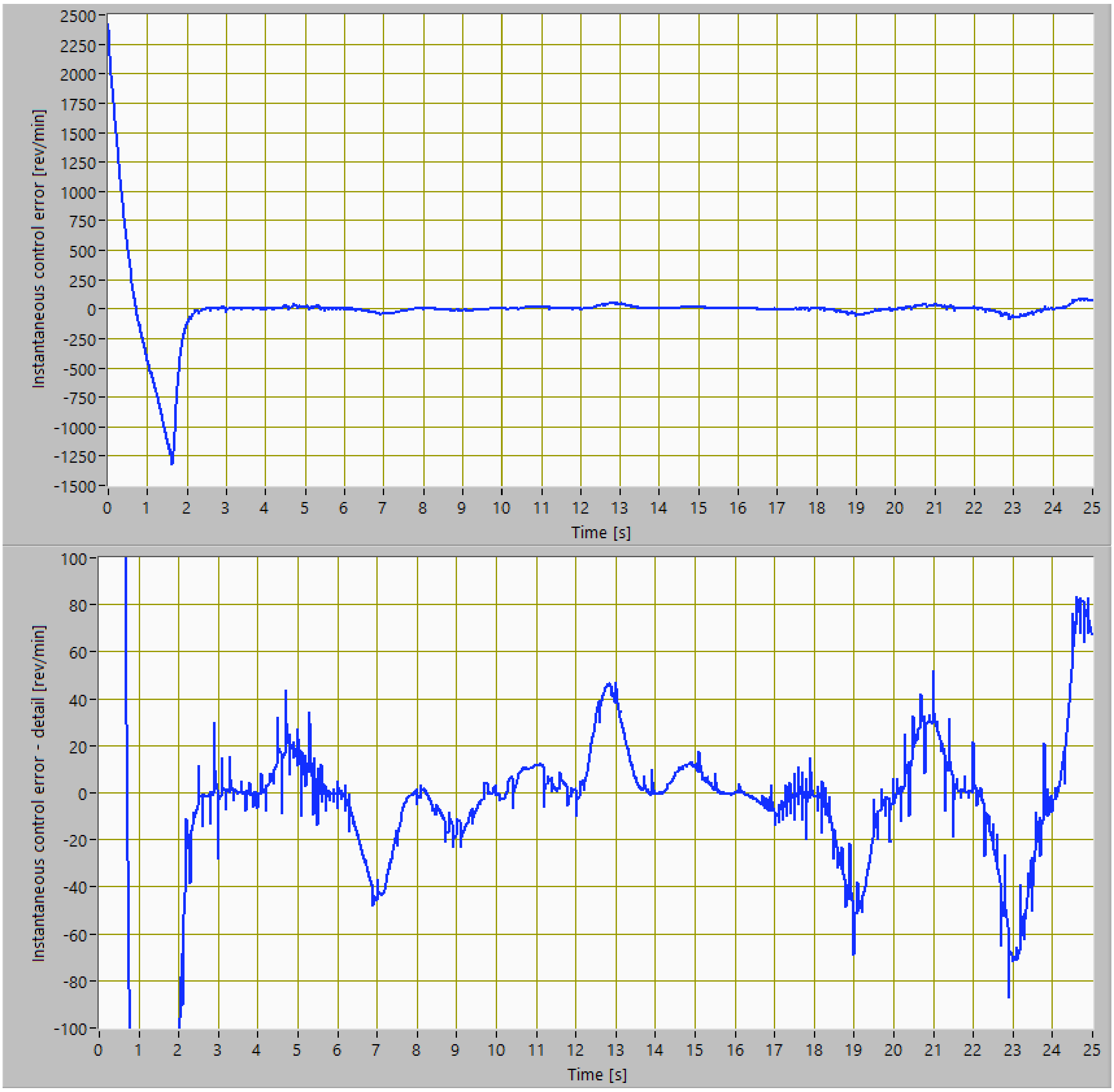

Fig. 13. The instantaneous control error over time

4. Conclusions

This paper developed both a theoretical and experimental system that have been used to emphasise the performances that closed-circuit secondary adjustment hydrostatic transmission can achieve, using the programming environment LabView as an interface between the computer and the acquisition board DAQ NI 6211, that also contain the core components of the control system: the PID controller and the feedforward multiply loop. It has been proven that, using these two together helped lower the control error and the systems response, allowing the velocity of the hydrostatic motor H_M to follow the desired value allows for precise and accurate control of the mechanical device being powered by the motor. This is particularly important in applications where precise and consistent movement is required, such as in manufacturing or other industrial settings. By carefully controlling the velocity of the hydrostatic motor, it is possible to ensure that the mechanical device moves in a consistent and predictable manner, allowing for more efficient and effective operation.

Acknowledgments

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