CONTROLLERS TUNING IN A POWER-SPLIT CONTINUOUSLY VARIABLE TRANSMISSION

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Abstract

The problem considered in this paper is the tuning of the control system of a power-split Continuously Variable Transmission (CVT) used in high-power tractors. Power-Split CVTs are characterized by the combination of a traditional (discontinuous) mechanical transmission and by a continuously-variable transmission. This guarantees, at the same time, smooth variations of the transmission-ratio and high efficiency of the overall transmission system.

1. Problem statement

Power-split CVTs are a very special type of CVTs ([7, 14]); they are used in very demanding applications, where high power transmission, very smooth transmission-ratio variations, and high efficiency of the transmission system are required. High-power agricultural tractors are vehicles where power-split CVTs can be used successfully ([10]).

Power-split CVTs differ from the widely used CVTs for automotive applications ([4, 11, 12, 15]) since they are characterized by the combination of a traditional mechanical transmission, and by a continuously-variable transmission. Among the many kinds of continuously-variable components (pulley-based, toroidal, electrical, hydrodynamic, hydrostatic, etc. – see e.g. [7, 13]), the CVT considered in this work uses an *hydrostatic system* ([5])

Power-split hydrostatic CVTs inherently need a sophisticated electronic control architecture ([6]), since the control and the coordination of the hydraulic and the mechanic part of the CVT cannot be obtained by mechanical elements only.

The control architecture considered in this work can be summarized as follows.

- The final goal of a CVT in an agricultural tractor is to track the required forward speed in a very smooth and regular way. As a matter of fact, due to the huge inertias an loads experienced by a tractor, the speed can suffer of abrupt variations and bumps. Hence, a supervising control unit receives as inputs the actual (v) and desired ($\overline{v}$) forward speeds, and sends a speed requirement $\overline{\omega}_{in}$ to the engine, and a transmission-ratio requirement $\overline{\tau}$ to the CVT (the transmission-ratio of a transmission unit is the ratio between the output and the input rotation speeds, namely $\tau = \omega_{out} / \omega_{in}$).
- The internal structure of the overall power-split CVT system is as follows. The input torque is split and redirected into the hydraulic transmission and into the mechanic transmission. The torques of these two branches are then re-combined at the output of the system. The transmission controller is a synchronizer, which receives as input the desired total transmission-ratio $\overline{\tau}$; its outputs are the desired hydraulic transmission-ratio $\overline{\tau}_{(h)}$, and the switch commands which trigger the mode changes in the mechanic transmission.
- Finally, the hydrostatic transmission is mainly constituted by a motor/pump system, driven by a proportional electro-hydraulic valve. The control architecture typically used for the regulation of this system is a cascade-control system: the inner loop is a servo-loop having the aim of regulating the valve current i_v to a desired value $\overline{i_v}$ over a large bandwidth. The outer loop (using $\overline{i_v}$ as control variable) regulates the hydraulic transmission ratio $\tau_{(h)} = \omega_{(h)out} / \omega_{in}$ to the desired value $\overline{\tau}_{(h)}$ requested by the synchronizer.

This work focuses on the design of the CVT control system. The final goal of the control problem addressed herein hence is to achieve a good tracking of the desired transmission ratio.

The starting point of this work is a simple and empirically-tuned set of controllers, which provide unsatisfactory performance. This is clearly revealed by the simple τ-tracking experiment displayed in Fig.1, where (at fixed engine rotation speed, $\omega_{in} = 1500$ rpm) the CVT is required to track a transmission-ratio ramp, ranging from $\tau = 0$ to $\tau = 0.5$ in about 10s. Note that the measured transmission-ratio is characterized by a large phase lag and by bumps and oscillations. This behavior is mainly due to the narrow bandwidth of the servo-loops of the hydraulic transmission, and to the unsatisfactory synchronization of the hydraulic and the mechanic branches of the CVT. The final goal of this work was to improve these tracking performance.

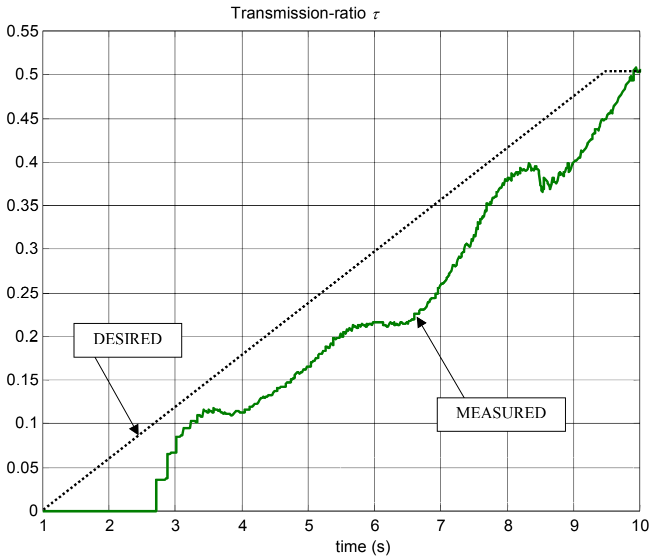

Fig.1. Transmission-ratio tracking experiment (empirically-tuned controllers).

The hydrostatic power-split CVT system considered in this work is constituted by the combination of:

- a *Sauer* hydraulic system driven by a proportional electro-hydraulic valve ([8]):
- a mechanical unit characterized by a set of 10 frictions labelled CF, CR, CL, BLM, CHH, CH, CP, BW, CTR, CVHH. These frictions are driven by on-off electro-hydraulic valves. The activation (or de-activation) of the frictions determines the mode-changes of the mechanic part of the CVT.

The admissible range of the transmission-ratio τ of this CVT is [-2.083,+2.083]. In this range, the mechanical part of the CVT has 6 forward modes and 6 reverse modes. All the controllers used for the regulation of the transmission are implemented on an ECU. The I/O signals of the ECU are:

- measure of the rotation speeds ($\left\{\omega_{in}, \omega_{out}, \omega_{(h)out}\right\}$);
- measure of the current i_v of the electro-hydraulic valve which drives the hydrostatic transmission;
- input d_v (PWM-modulated) of the proportional electro-hydraulic valve;
- on-off triggers to the electro-valves of the frictions of the mechanical part of the transmission.

The ECU is also connected at 250 Kbps to the CAN bus of the vehicle.

2. The current loop

The first control loop which must be considered in the CVT control system design is the servo-loop on the output current of the proportional electro-hydraulic valve which drives the hydraulic transmission. A well-regulated (high-bandwidth) current control system is mandatory for the subsequent design of the hydraulic transmission-ratio control-loop. The control scheme for the current regulation is a classical one-degree-of-freedom feedback architecture, where:

- i_v is the valve current;
- d_v is the Pulse-Width-Modulated (PWM) input signal of the valve;
- $\bar{i}_v$ is the desired value of the valve current;
- $G_i(s)$ is the transfer function which models the I/O dynamics of the valve from d_v to i_v (linear and time-invariant dynamics are assumed);
- $R_i(s)$ is the transfer function of the controller.

The first step in the design of this control system is the development of a mathematical model of the plant. A black-box approach has been chosen. The estimation procedure of the transfer function $G_i(s)$ has been done as follows (see e.g. [3] and references cited therein).

- The plant (the electro-hydraulic valve) has been fed with a set of 8 pure-tone signals ($A\sin(\Omega_n t)$, for 8 different frequencies Ω_n, n=1,2,3,…8). The tested frequencies are $\Omega_n = \alpha_n \cdot 2\pi rad/s$, $\alpha_n \in \left\{0.05, 0.1, 0.2, 0.4, 1, 2, 4, 10\right\}$.
- The corresponding outputs have been measured. Since the plant is linear, the output signals are pure tones at the same input frequency. Using the estimated magnitude and phase of the output signals ([1, 2, 9]), a frequency response $\hat{G}_i(j\Omega_n)$, n=1,2,3,…8 can be estimated.
- A parametric model has been chosen, by direct inspection of the measured frequency response. The selected model is constituted by a pole and a pure time delay, namely $G_i(s) = e^{-sT_i}\left(K_{pi}/1+sT_{pi}\right)$.
- The model parameters have been identified by minimizing a frequency-domain loss-function ([3]):

$$\{\hat{T}_i, \hat{K}_{pi}, \hat{T}_{pi}\} = \\ = \arg\min_{\{T_i, K_{pi}, T_{pi}\}} \left\{ \sum_{n=1}^{8} \left(\hat{G}_i(j\Omega_n) - e^{-j\Omega_n T_i} \left(K_{pi}/1 + j\Omega_n T_{pi}\right)\right)^2 \right\}$$

Using this procedure, the estimated model parameters are: $\{\hat{T}_i, \hat{K}_{pi}, \hat{T}_{pi}\} = \{0.0023, 15.75, 0.023\}$. The quality of the model can be evaluated by Fig.2, where the measured and the estimated frequency responses are plotted. Note that the estimated model, in spite of its simplicity, provides a good fitting of the data (especially on the phase, which is the most critical for control system design).

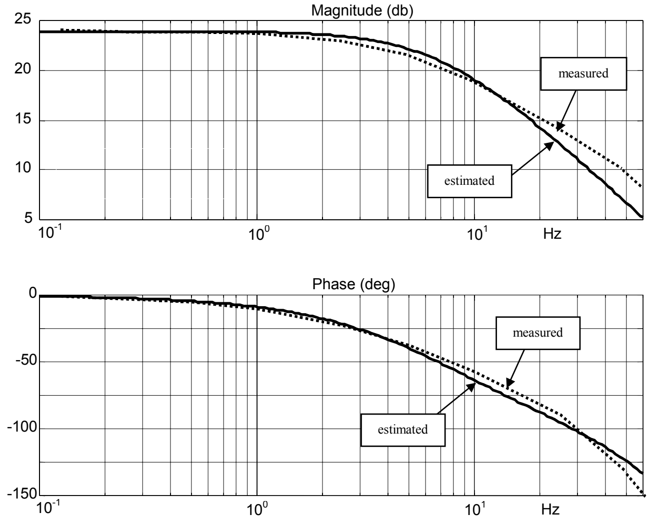

Fig.2. Measured and estimated frequency response of the valve I/O dynamics.

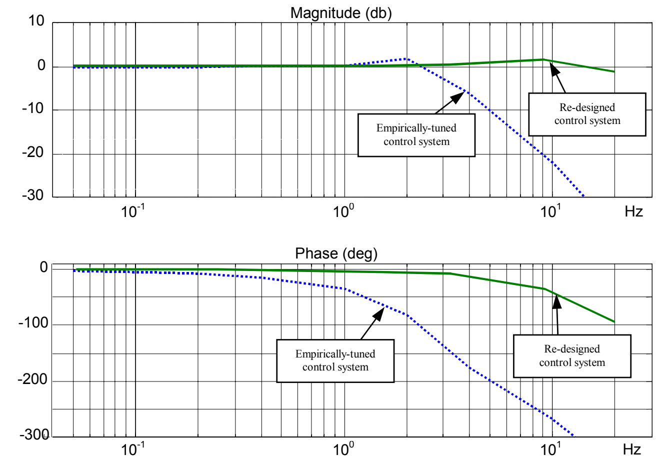

Fig.3. Closed-loop frequency responses (experimentally measured).

On the basis of the estimated model, a PI controller $R_i(s) = K_{ci}\left((1+sT_{ci})/s\right)$ has been designed with the objectives of a large bandwidth and a good phase margin. The parameters of the controller are $\{\hat{K}_{ci}, \hat{T}_{ci}\} = \{5.99, 0.023\}$. Finally, the closed-loop performance have been experimentally tested by measuring the closed-loop frequency response (from $\bar{i}_v$ to i_v). The results are displayed in Fig.3. Note that the closed loop bandwidth is almost 10Hz. This large bandwidth allows a performance improvement to the outer loop on the hydraulic transmission-ratio.

3. The torque loop

The second control loop considered in this work is the servo-loop on the hydrostatic transmission-ratio $\tau_{(h)}$. This is the outer and main controller. This control loop is crucial, since a good synchronization of the hydrostatic and the mechanic transmission branches can be achieved only if the transmission-ratio $\bar{\tau}_{(h)}$ requested by the synchronizer is well-tracked over a large bandwidth. Even in this case, the control scheme is a classical one-degree-of-freedom feedback architecture, where:

- $\tau_{(h)}$ is the actual hydrostatic transmission-ratio;
- $\bar{i}_v$ is the desired value of the valve current, which is used as control input;
- $\bar{\tau}_{(h)}$ is the desired hydrostatic transmission-ratio;
- $G_\tau(s)$ is the transfer function which models the I/O dynamics (from $\bar{i}_v$ to $\tau_{(h)}$) of the hydraulic transmission (linear and time-invariant dynamics are assumed – see discussion below);
- $R_\tau(s)$ is the transfer function of the controller.

The mathematical model of the plant has been developed with the same approach (black-box parametric identification in the frequency-domain) used for the inner loop. For the hydraulic transmission a crucial issue, however, is the possible dependence of its dynamics on the *input rotation speed* ω_i and on the *transmitted torque* (namely the load suffered by the hydraulic transmission). To this end, two sets of 8-point frequency-responses $\hat{G}_\tau(j\Omega_n)$ have been estimated in the following conditions:

- 3 input rotation speeds has been tested: $\omega_i = 1500rpm$, $\omega_i = 2000rpm$, and $\omega_i = 2200rpm$; in all cases the transmission is unloaded (zero torque transmitted).
- 3 load conditions has been tested: null load, medium load, and maximum load. In all cases the input rotation speed was $\omega_i = 1500rpm$.

The analysis of these results has shown that the dependence of the hydraulic transmission dynamics on engine speed and load is negligible. The plant hence can be well-approximated with a unique linear time-invariant transfer function. Even in this case, the selected model is constituted by a pole and a pure time delay, $G_\tau(s) = e^{-sT_\tau}\left(K_{p\tau}/1+sT_{p\tau}\right)$, whose parameters have been identified by minimizing a frequency-domain performance index. The estimated model parameters are: $\{\hat{T}_\tau, \hat{K}_{p\tau}, \hat{T}_{p\tau}\} = \{0.03, 0.0033, 0.094\}$. The frequency-domain fitting of the model to the data is displayed in Fig.4. On the basis of the estimated model a PI controller $R_\tau(s) = K_{c\tau}\left((1+sT_{c\tau})/s\right)$ has been designed, with the objective of a large bandwidth and a good phase margin. The parameters of the controller are $\{\hat{K}_{c\tau}, \hat{T}_{c\tau}\} = \{-2980, 0.094\}$. The closed-loop performance have been experimentally tested by measuring the closed-loop frequency response (from $\bar{\tau}_{(h)}$

to $\tau_{(h)}$). The results are displayed in Fig.5. Note that the closed loop bandwidth is about 1Hz. The fact that the outer-loop is about a decade slower than the inner loop guarantees an almost perfect dynamic decoupling between the two loops

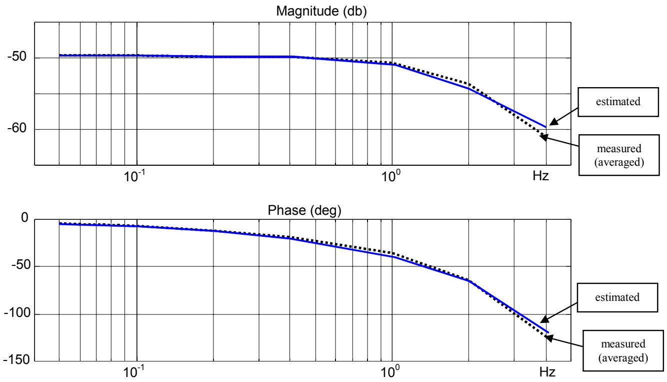

Fig.4. Measured and estimated frequency response of the hydraulic transmission I/O dynamics.

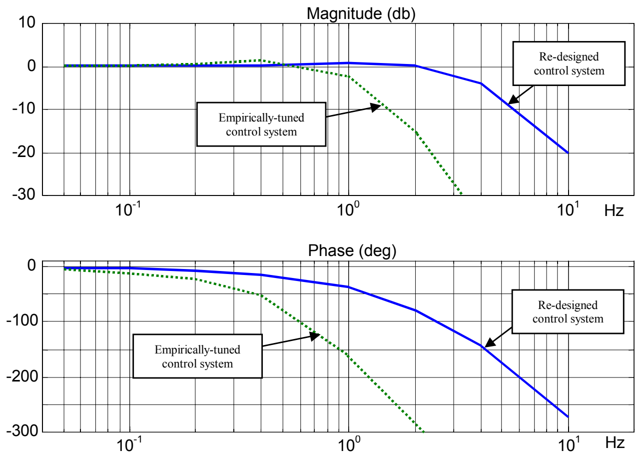

Fig.5. Closed-loop frequency responses (experimentally measured).

4. The synchronizer

The last control problem considered in this work is the design and tuning of the synchronizer. The synchronizer is an hybrid supervising control unit, which mainly performs the following two tasks: generates the signal $\overline{\tau}_{(h)}(t)$, and activates and de-activates the set of frictions, by means of triggering commands to the electro-hydraulic on-off valves.

In an ideal setting, these two tasks are simple and well-defined.

1) Using the algebraic relationship (displayed in Fig.6) between the overall transmission ratio τ and the hydraulic transmission ratio $\tau_{(h)}$, the synchronizer must deliver a ramp-signal $\overline{\tau}_{(h)}(t)$ which joins the actual and the target transmission ratio. It is easy to see that if the attainment of the target transmission-ratio requires a mode change, a sequence of increasing ramps followed by decreasing ramps (or vice-versa) must be generated.

2) The triggering command of the mode change should be activated exactly in correspondence of a peak of the piecewise ramp signal $\overline{\tau}_{(h)}(t)$. In correspondence to the peak, a friction must be switched-off and a friction must be switched-on.

However, due to the non-ideal behavior of the on-off valves, the actual task of the synchronizer is more complicated, and requires some additional insight and tuning.

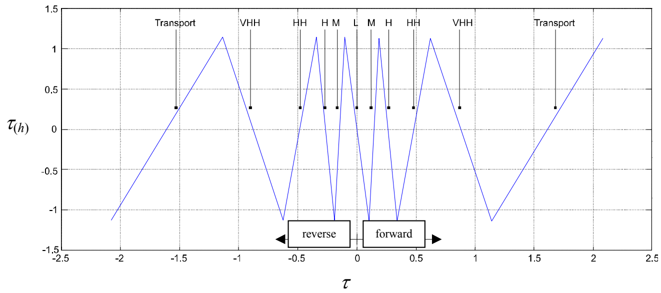

Fig.6. Relationship between the total transmission-ratio τ and the hydraulic transmission-ratio $\tau_{(h)}$.

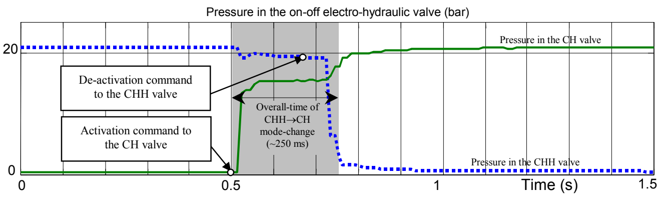

Fig.7. Time-domain behaviour of the internal valve pressure during activation and de-activation.

The actual behavior of a non-ideal on-off valve can be easily understood from Fig.7, where the internal pressures of the electro-hydraulic valves driving the frictions CH and CHH (during the mode-change from HH to H) are displayed in the time-domain. The internal pressure of the valves gives an indirect indication of the valve activation status: the valve is on when the pressure reaches 20 bar (which is the pressure of the hydraulic circuit which drives the valves); the valve is off when the pressure is null. By inspecting Fig.7 the following observations are due.

- After the activation command is sent to CH, the pressure of the valve does not jump instantaneously to the 20bar target pressure; it increases following a well-defined path:first, the pressure is subject to a small pure time-delay of about 20ms; after the time-delay, the pressure rapidly increases, until the pressure of about 15bar is reached (the valve is not activated yet); the pressure remains almost constant for about 130ms, due to process of oil-filling of the valve; finally, the pressure re-start increasing until the target pressure is reached. The overall activation time of CH is about 250ms.
- After the de-activation command is sent to CHH, the

valve pressure follows this simple path (note that the pressure in CHH is slightly influenced by the activation command to CH, since they are driven by the same hydraulic circuit): after the activation command is given, the pressure remains unchanged for about 50ms; the pressure quickly decreases until the null pressure is reached. The overall de-activation time of CHH is about 100 ms.

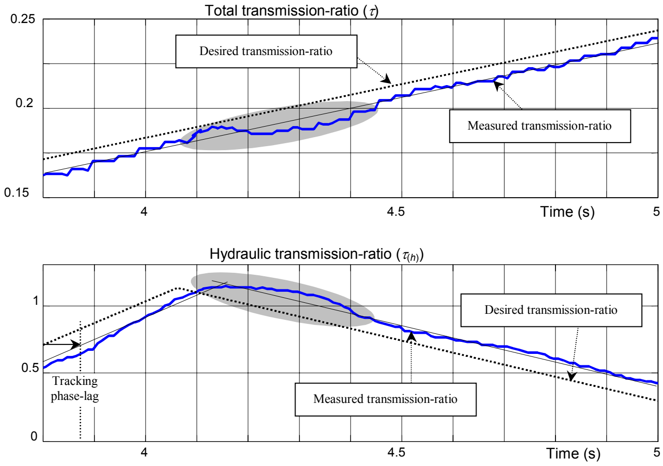

Fig.8. Mode change M→H without compensations.

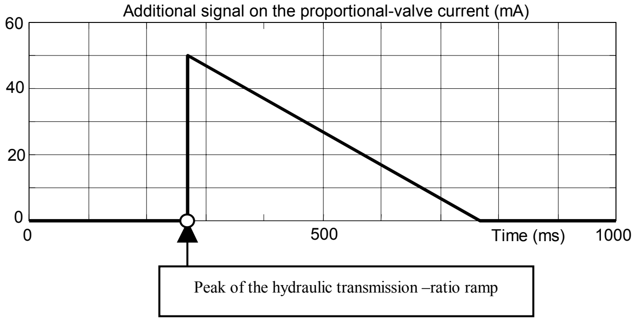

Fig.9. Time-domain profile of the current compensation.

Due to the fact that the activation time is always larger than the de-activation time, the mode-change is performed with an overall time-lag equal to the activation-time of the up-coming friction. The mode-change routine performed by the synchronizer hence is as follows: the up-coming friction is activated; a time-lag equal to the difference between the activation-time of the up-coming friction and the de-activation time of the out-coming friction is waited for; the out-coming friction is de-activated.

The overall mode-change time takes, on average, about 250ms (it only depends on the type of the up-coming friction, and ranges from 170ms to 400ms).

The simplest and most intuitive way of managing a mode-change is to start the routine in correspondence to the peak of the piecewise ramp on $\overline{\tau}_{(h)}(t)$. In Fig.8 an example of a M→H mode change (involving the activation of CH and the de-activation of BLM) performed using this simple rationale is displayed. The accurate inspection of the behavior of the total and of the hydraulic transmission-ratio shows many interesting characteristics of this control system.

- The transmission-ratio $\tau_{(h)}(t)$ is inherently affected by a phase-lag, with respect to the desired reference-signal $\overline{\tau}_{(h)}(t)$. This lag obviously propagates on the total transmission-ratio $\tau(t)$ as well. This is due to the finite bandwidth of the $\overline{\tau}_{(h)}(t)$-tracking servo-loop. In Fig.8 this time-lag is about 70ms. This time-lag can be reduced only by enlarging the bandwidth of the servo-loop on the hydraulic transmission-ratio.
- The hydraulic transmission-ratio $\tau_{(h)}(t)$ exhibits a long overshoot (shaded area in Fig.8) just after the peak of the piecewise ramp on $\overline{\tau}_{(h)}(t)$. This also affects the total transmission ratio (note the two under-tracking and over-tracking bumps in the neighborhood of the mode-change).

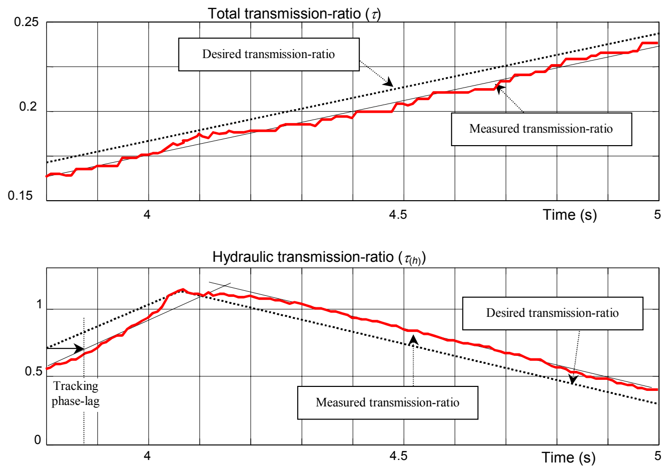

Fig.10. Mode change M→H with open-loop compensations.

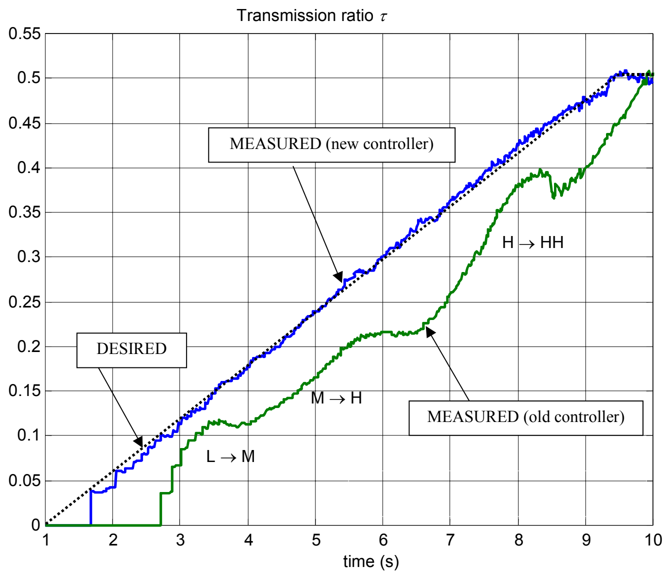

Fig.11. Transmission-ratio tracking experiment (old controllers, and re-designed controllers).

This behavior of the synchronizer has been significantly improved using an open-loop compensation approach. Two simple but effective modifications have been proposed.

- The mode-change routine has been *anticipated* of an amount of time exactly equal to the activation-time of the up-coming friction. If the peak of the piecewise ramp on $\overline{\tau}_{(h)}(t)$ occurs at time T_1, and the activation-time of the up-coming friction is ΔT_u, the mode-change routine is started at time $T_1 - \Delta T_u$. Note that the prediction of the peak-time T_1 can be easily made since the slope of the ramp is known in advance.
- The reference signal $\overline{i_v}$ of the proportional valve current has been *added with an impulse-like correction signal*, just after the peak on $\overline{\tau}_{(h)}(t)$, in order to remove the overshoot on $\tau_{(h)}(t)$. A fine-tuning procedure on the real plant has shown that the best results can be achieved if the impulse-like compensation signal has the triangular shape illustrated in Fig.9. The height of the optimal impulse-like triangular compensation signal slightly depend on the type of mode change.

In Fig.10 the results of the M→H mode change, using the modified synchronizer equipped with open-loop compensations, are displayed. Note that the total transmission-ratio tracking is almost perfect; also the fitting of the actual hydraulic transmission ratio to the reference signal has been strongly improved. Finally, in Fig.11 the results of a τ-tracking experiment over a large range of transmission-ratio (involving 3 mode-changes) is plotted. Note the remarkable improvements achieved by the re-design and re-tuning of the three controllers of the CVT. The results displayed in Fig.11 are very satisfactory, and fulfill the requirements of a modern top-range agricultural tractor.

Acknowledgments

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