Control of Power-Shuttle Motion-Inverter

Sergio M. Savaresi, Mara Tanelli, Sergio Bittanti, Alberto Mangili, Francesco Taroni, Fabio Previdi

Abstract - **The topic of this paper is the design of the control system for the automatic motion-inverter in agricultural tractors. The objective of this system is to perform a fully-automated symmetric motion inversion The device (reverser) used for the automatic motion inversion is an electro-hydraulic system, constituted by two clutches (one for the forward, and one for the reverse motion), driven by a Proportional Electro-hydraulic Valve (EVP) and by an on-off Directional Electro-hydraulic Valve (EVD). The measured variables are the input/output rotational speeds of the reverser. All the sub-tasks of the reverser control systems are considered: the design of an inner-loop for the EVP, the open-loop switching strategies, and the design and analysis of the outer control loop which regulates the vehicle speed during the motion-inversion.**

I. INTRODUCTION

THE topic of this paper is the design of the control system for the automatic motion-inverter in agricultural tractors. The objective of this system is to perform a fully-automated symmetric motion inversion (e.g. from a forward speed of 10 Km/h to a reverse speed of -10 Km/h). The device (reverser) used for the automatic motion inversion is an electro-hydraulic system, constituted by two clutches, driven by a Proportional Electro-hydraulic Valve (EVP) and by an on-off Directional Electro-hydraulic Valve (EVD). The ECU of the transmission drives the currents of these valves (which constitute the main control variables); the measured variables are the input/output rotational speeds of the reverser.

The design of the motion-inverter system is a non-trivial task, since it is difficult to find a good compromise between speed (the complete motion-inversion task should be performed in the shortest possible time) and comfort (bumps and oscillations on the longitudinal speed must be minimized). Huge variations in the system (vehicle inertia, temperature, road slope, etc.) also make this control problem particularly challenging.

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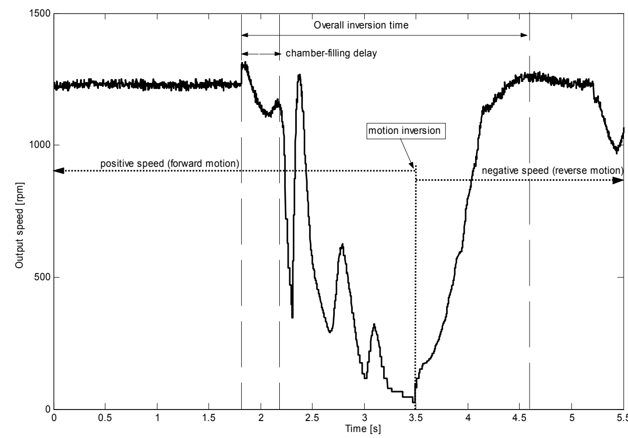

Fig.1. Behavior of the output rotational speed of the reverser (absolute value), during a forward→reverse motion-inversion.

In order to better understand the bulk of this control problem, consider the signal displayed in Fig.1. This signal represents the rotational speed (in [rpm]) of the output shaft of the reverser. The speed of the vehicle is simply proportional to this variable (henceforth the behavior of the signal in Fig.1 is the behavior of the longitudinal speed of the tractor). The inversion starts at a forward speed of 1250rpm and ends (after about 3s) at a reverse speed of 1250rpm. Notice that the signal in Fig.1 is positive both in forward and in reverse speed, since the rotational sensor cannot detect the sign of the rotation; the zero-crossing obviously indicates the change in the speed direction.

In principle, the “ideal” behavior of the speed is constituted by a constant-slope ramp (V-shaped if the absolute value of the speed is displayed). Instead, notice that it strongly differs from this ideal behavior: the first part of the inversion is affected by filling-delays in the oil-chambers of the clutches, the braking and the accelerating behavior are asymmetric and the speed is affected by oscillations. The inversion displayed in Fig.1 contains all these negative aspects; the result is an uncomfortable inversion.

The objective of this paper is to present the whole design procedure of the reverser. All the sub-tasks of the reverser automatic control systems will be considered: the design and analysis of the inner-loop for the EVP current (Section 3), the open-loop switching strategies (Section 4), and the outer-loop which regulates the vehicle speed during the motion-inversion (Section 5).

This work has been developed for research purposes on a Power-Shuttle transmission.

II. SYSTEM DESCRIPTION

The overall scheme of the Power-shuttle transmission used in this work is displayed in Fig.1. Moving from left (engine) to right (wheels), notice that the transmission is mainly constituted by the reverser (shaded box), the three main clutches (Low, Medium, High), and the synchro. The reverser and the three main clutches are controlled by the ECU, which drives the Electro-Hydraulic valves.

Note that the reverser is cascaded with the rest of the transmission. Our work will focus on this part of the transmission only. As already said, the reverser is actuated by two Electro-Hydraulic valves: a proportional valve (EVP) and an on-off directional valve (EVD), which can assume three positions: F (Forward), N (Neutral) and R (Reverse).

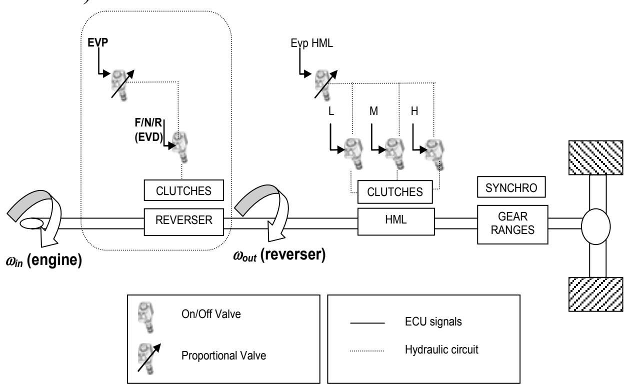

Fig.1. Overall scheme of the power-shuttle transmission.

The main measured variables of the reverser are the input rotational speed of the shaft (ω_{in} , which corresponds to the engine speed) and the output rotational speed of the shaft ω_{out} . In the rest of the paper ω_{out} is assumed to be proportional to the wheel speed, according to the transmission ratio of the rest of the transmission

A schematic diagram of hydraulic circuit which drives the reverser is displayed in Fig.2 (see e.g. [1-3, 5-8] . Note that the hydraulic circuit is mainly constituted by an accumulator, a pump, and many hydraulic "users": the hydro-steer, the Power-Take-Off (PTO), the differential-locking system (BD), the 4-wheels traction (DT), the HML clutches, and the inversion system (INV).

The inversion system (see detail in the zoomed part of Fig.2), it is constituted by two clutches (Forward and Reverse); each clutch is activated if the fluid in the corresponding chamber is pressurized. The bulk of the control problem is to synchronize the activation and the de-activation of these two clutches, in order to provide smooth transitions from one clutch to another. This can be done by means of two Electro-Hydraulic valves: the directional valve (EVD), which is a 3-positions (Forward, Neutral, Reverse) 4-ways on-off valve; this valve is used to activate and de-activate the two-clutches; the second valve (the most critical) is a proportional valve (EVP), which can continuously modulate the pressure in the clutch chamber.

The overall control problem can be split into three sub-problems: the control of the current of the EVP (inner-loop); the switch of the EVD valve, and the open-loop modulation of the EVP during the fist part (about 0.5s) of the inversion; the control of the speed of the output shaft of the reverser, using the controlled EVP (outer-loop).

They will be discussed in the following sections.

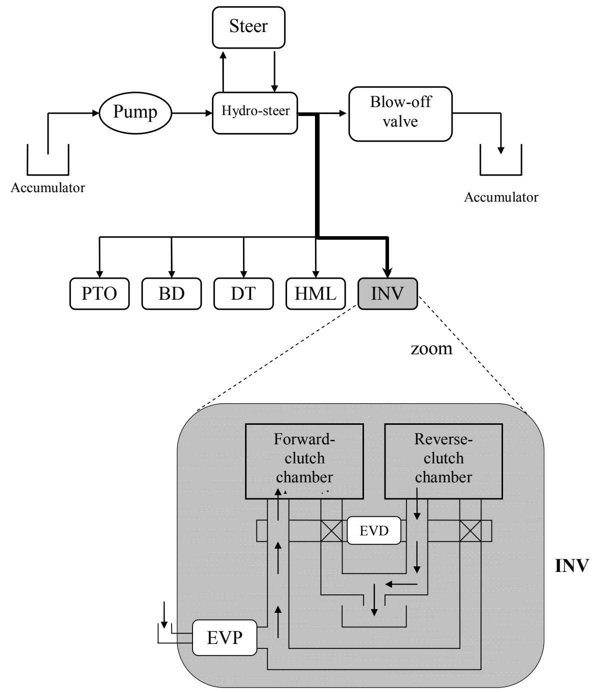

Fig.2. Scheme of the hydraulic circuit.

III. THE INNER-LOOP

The ultimate goal of the Proportional Electro-Hydraulic Valve (EVP) of the reverser is to modulate the pressure of the currently-active clutch chamber. However, the accurate measurement of the output pressure of the EVP in practice is not available in a vehicle, due to cost restrictions; the only measured variable of the EVP is the internal current of the valve electro-mechanical driver. The inner control loop henceforth must be limited to the control of the EVP current.

The structure of the inner-loop is a standard Single-Input-Single-Output (SISO) control scheme, where the controlled variable is the valve current *I*, the control input d_v is the Pulse-Width-Modulated (PWM) input signal of the valve, and the reference signal is the desired value of the valve current I_{ref} . $G_i(s)$ is the transfer function which models the I/O dynamics of the valve from d_v to I (linear and time-invariant dynamics are assumed), and $R_i(s)$ is the transfer function of the controller.

In order to design the current loop, a classical model-based indirect design approach has been used. First, the dynamic behavior of the plant (from the PWM-modulated input to the output current of the EVP) has been identified, using a frequency-domain identification procedure. By feeding the EVP with multi-frequency sinusoidal inputs ([4, 5]), the frequency-response has been experimentally computed. These measured points have been approximated with a simple linear model, constituted by a 1^{st}-order transfer function and a time-delay:

$$\hat{G}_i(s) = \frac{K}{1+sT}e^{-\tau s} \ .$$

The estimated parameters of this transfer function are: $[\hat{K} \quad \hat{T} \quad \hat{\tau}] = [1.54 \quad 10 \quad 1.5]$. The fitting between the modeled and the measured frequency response is very accurate.

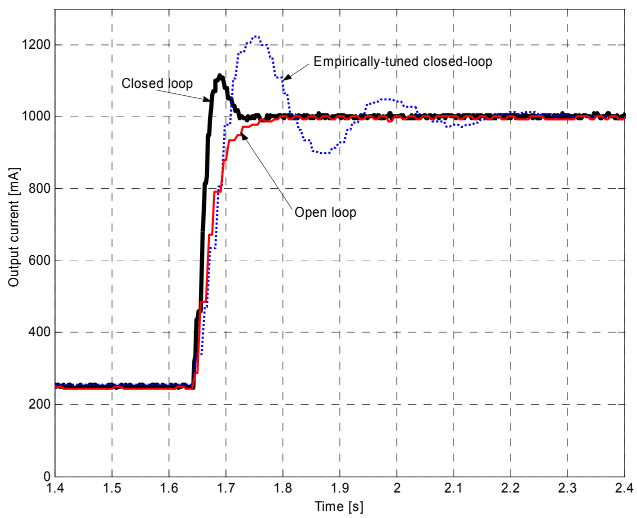

Fig.3. Step responses of the current of the EVP (measured).

On the basis of the above model, a simple PI regulator has been designed.

$$R_i(s) = K_R \frac{1+sT_I}{sT} \qquad [K_R \quad T_I] = [1 \quad 10] \ .$$

The behavior of the closed-loop system is displayed in Fig.3, where the measured step-responses are displayed. Notice that the closed-loop system is faster than the open-loop system (at the price of a little overshoot), and it works much better than an empirically-tuned controller.

IV. Open-loop strategies

When the driver requires a motion inversion, the automatic control system takes full control of the vehicle, until the vehicle speed has reached the same value, with opposite direction. A complete inversion (starting from a maximum speed of 10Km/h) usually takes 3-4s.

When the automatic inversion procedure starts, the first phase of the inversion algorithm is performed in open-loop (this phase takes about 0.5s). During this open-loop phase the following actions are taken (consider, for example, an inversion from Forward to Reverse):

- **EVD:** When the inversion procedure is triggered by the driver, the EVD - which was originally in the Forward position – is immediately switched to the Neutral position (see Fig.4); since the EVP is closed, the pressure in the chamber of the Forward clutch immediately drops; when this pressure goes below (about) 4 Bar, the Forward clutch is completely disconnected (no torque is transmitted through this clutch); when the disconnection of the Forward clutch is completed, the EVD switches to the Reverse position, and the chamber of the Reverse clutch starts being filled. In practice (see Fig.4), since the output pressure of the EVP is not measured, the EVD is switched from Neutral to Reverse not immediately after the pressure in the Forward chamber drops below 4 Bar, but after exactly 150ms. This time window has been empirically estimated, in order to guarantee the complete disconnection of the clutch in every working condition.
- **EVP:** The open-loop strategy on the EVP can be more complicated than the one on EVD, since a more sophisticated modulation (than simple on-off) is possible. When the inversion starts the EVP is immediately fully closed, in order to allow the quickest pressure drop in the Forward clutch. When the EVD is switched to the Reverse position, the EVP is reopened, in order to allow the rise of the pressure in the Reverse clutch chamber. The open-loop control design problem is to find the best shape of the EVP current, in the 300-400ms before the closed-loop control on the forward vehicle speed is activated. This problem (and the solution proposed herein) will be discussed in the rest of this Section.

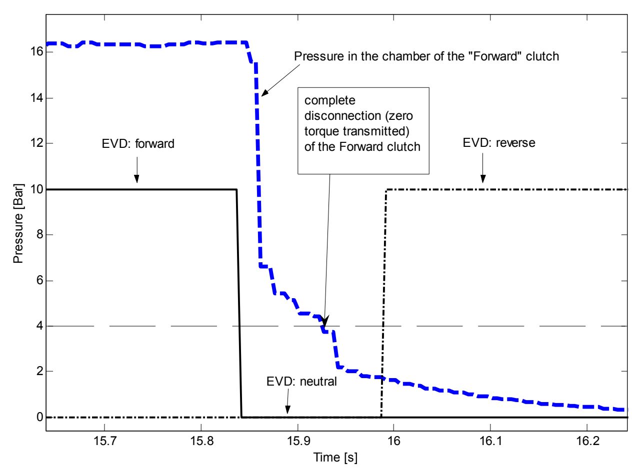

Fig.4. Switching strategy of EVD.

In order to understand the problem of the EVP-current modulation, consider the diagram of Fig.5. Schematically, we can say that the torque transmitted by the clutch increases (from 0% to 100%) as the EVP current increases (from 0% to 100%, where the 100% condition corresponds, in our EVP, to about 1400mA).

The relationship between the transmitted torque and the EVP-current is non-decreasing monotone, but unfortunately it is far from being linear (and, due to the chamber-filling dynamics, it is not even a static function). This relationship is roughly described in Fig.5, where four current ranges are represented:

- In the lowest current range (from $I_{\min} = 0$ to I_C) the corresponding pressure in the clutch chamber is so low that no torque is transmitted by the clutch; this current range is a "dead-zone" for the clutch.
- In the second current range (from I_C to I_M) some torque is transmitted by the clutch; however (if the vehicle starts from a still condition) this torque is not large enough to cause the vehicle movement (this means that it is lower than the overall static friction torques of the vehicle); the current I_C is called "Contact" current (since the clutch begins to establish a contact between its input and its output shaft); the current I_M is called "Movement" current (since the clutch starts to move the vehicle).
- In the third current range (from I_M to I_F) the transmitted torque increases from the minimum "Movement" level to the 100% level (the current I_F is called "Full" transmission current).
- The last current range (from I_F to $I_{\max} = 100\%$) corresponds to full-torque transmission; henceforth, also this current interval is a "dead-zone" from the control system point of view.

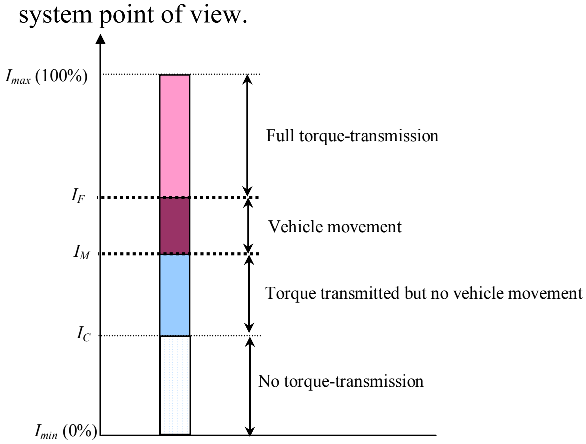

Fig.5. EVP-current ranges.

Apparently, the objective of the about 400ms of the open-loop phase is to smoothly bring the current to the value I_M, and then activate the closed-loop speed control of the vehicle. Along this simple path, the major problem is that the value I_M is spread in a comparatively large range. As a matter of fact – unfortunately – the value of I_M can be affected by differences between different vehicles, due to construction tolerances, different working temperatures, different loads and masses of the vehicle, different slopes of the road where the vehicle is operating, and different gear ratios.

Experimental resuls have shown that the modulation range is small, if compared with the spread of I_M. This makes the overall control problem very challenging. As a matter of fact, since the value of I_M can vary and cannot be estimated on-line, two undesired situations may easily occur:

- the actual value of I_M is lower than the nominal value of I_M: in this case the open-loop procedure would bring the EVP current to a value higher than the "Movement" current: this would cause discomfort since it delivers a step in the transmitted torque;
- the actual value of I_M is higher than the nominal value of I_M: in this case the open-loop procedure would bring the EVP current to a value lower than the "Movement" current: this would cause a delay in the activation of the reverse torque, since – when the closed-loop control is activated – the current is not yet positioned at the Movement condition.

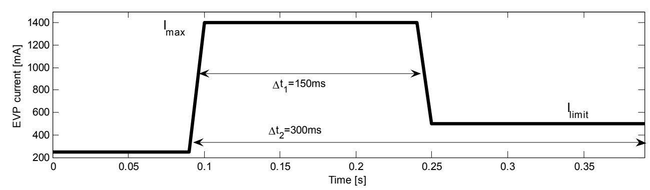

Fig.6. Shape of the EVP-current, during the open-loop phase.

An acceptable compromise between these conditions must be found. The open-loop EVP-current shape proposed herein is displayed in Fig.6.

When the EVD is switched into the Reverse condition (150ms after the inversion is triggered), the EVP current is switched to $I_{\max}$, and kept at this value for 150ms. Notice that – even if the EVP current is much higher than I_M during this time-window - the actual transmitted torque is slower than the Movement torque. As already said, this is due to the fact that the clutch chamber must be filled (when the Reverse clutch is activated, its pressure is very low, and it is partially empty); henceforth, the best strategy is too keep the EVP current at its maximum value, in order to increase the pressure in the camber as quickly as possible. The time- window of 150ms has been computed in order to guarantee that Movement torque is not reached in every working condition.

After 150ms the EVP current is switched from $I_{\max}$ to $I_{\lim}$, where $I_{\lim}$ is the lowest value of I_M, measured in different working conditions (e.g. $I_{\lim} = 44.5\%$, considering the results displayed in Fig.6). This value of the EVP current is kept for another 150ms; then the closed-loop algorithm is activated. Notice that this strategy guarantees that no torque overshoot occurs, at the price of a possible delay in the activation of the movement torque. However, thanks to the previous 150ms of “full-current”, the overall open-loop phase remains very short, and a possible small lag in the Reverse torque activation can be accepted.

V. THE OUTER-LOOP

The third control sub-problem is also the most crucial, since it takes care of the ultimate goal of the inversion: the control of the vehicle movement during the (about) 3-4s of the automatic inversion. The closed-loop controller of the vehicle motion is activated when the open-loop control of the EVP current is concluded. In the open-loop strategy proposed and tested in this work, this happens exactly 450ms after the automatic inversion is triggered.

In order to design the outer-loop, the basic dynamics of the regulated system must be modeled.

If we consider the dynamics of the vehicle in the longitudinal direction (described by the state variable ω_{out}), the following simple dynamic equation can be written:

$$J_V \dot{\omega}_{out} = \tau(I) T_E(\omega_{in}) - T_R \,. \tag{1}$$

In (1), the following variables and parameters are used:

- J_V is the overall inertia of the vehicle
- $\tau(I) = \omega_{out} / \omega_{in}$ is the transmission ratio
- $T_E(\omega_{in})$ is the torque delivered by the engine
- T_R is the resistant torque of the whole vehicle: it is mainly due to friction effects.

Notice that (1) is a 1st-order non-linear dynamic time-invariant model, characterized by three inputs: I, ω_{in} and T_R. In order to linearize the model, consider an equilibrium condition, namely: $\tau(\bar{I}) T_E(\bar{\omega}_{in}) = \bar{T}_R$.

Around this equilibrium (fully defined by $\bar{I}, \bar{\omega}_{in}, \bar{T}_R$), the following variables can be defined: $\delta\omega_{out} = \omega_{out} - \bar{\omega}_{out}$, $\delta I = I - \bar{I}$, $\delta T_R = T_R - \bar{T}_R$, $\delta T_E = T_E(\omega_{in}) - T_E(\bar{\omega}_{in})$.

If we consider the non-linear static functions $\tau(I)$ and $T_E(\omega_{in})$, they can be linearized around the equilibrium condition, so obtaining:

$\delta T_E = K_E \delta\omega_{in}$, where $\delta\omega_{in} = \omega_{in} - \bar{\omega}_{in}$;

$$\delta\tau = \tau_1(\bar{I})\delta I = \left[\frac{1}{\bar{\omega}_{in}}\right]\delta\omega_{out} + \left[-\frac{\bar{\omega}_{out}}{\bar{\omega}_{in}^2}\right]\delta\omega_{in} \,,$$

where

$$\tau_1(\bar{I}) = \left.\frac{\partial\tau(I)}{\partial I}\right|_{I=\bar{I}} , \quad \bar{\tau} = \frac{\bar{\omega}_{out}}{\bar{\omega}_{in}} = \tau(\bar{I}) \text{, and } \delta\tau = \tau - \bar{\tau} \,.$$

Using these relationships, the following linearized model can be easily computed:

$$J_V \delta\dot{\omega}_{out} = [\tau_1(\bar{I})\bar{T}_E]\delta I + [\tau(\bar{I})]\delta T_E - \delta T_R \,. \tag{2}$$

Model (2) is linear and time-invariant; henceforth it can be represented by means of transfer functions; this model is depicted as a block-diagram in Fig.7.

Notice that the system depicted in Fig.7 is characterized by a control input (δI), a non-measurable disturbance (δT_R), and a measurable disturbance ($\delta\omega_{in}$); three controlled outputs can be considered: the transmission ratio $\delta\tau$, the output speed $\delta\omega_{out}$ (which is proportional to the longitudinal speed of the vehicle), and the output acceleration $\delta\dot{\omega}_{out}$ (which is proportional to the longitudinal acceleration of the vehicle). Moreover, among the many parameters which characterize the model, it is worth pointing out that the vehicle inertia J_V can suffer of large changes. Since a single control input is available, it is natural to design a Single-Input-Single-Output (SISO) control loop. A non-trivial issue henceforth is the selection of a wise output (controlled) variable. A concise discussion on each candidate output is now provided.

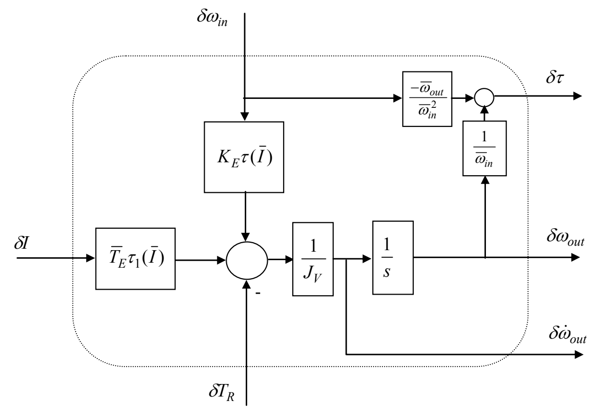

Fig.7. Block-diagram of the model (2).

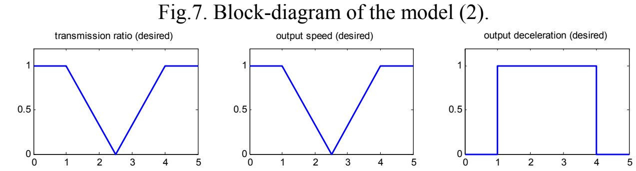

Fig.8. Normalized reference signals for the three candidate output (controlled) variables of the systems.

- The output speed ω_{out} is a natural and intuitive controlled variable; as a matter of fact the inversion procedure can be seen as the smooth transition between $\omega_{out} = 1$, and $\omega_{out} = -1$ (these values are normalized with respect the initial speed); accordingly, the reference trajectory for ω_{out} during a 3s-long inversion (see Fig.8) is a V-shaped function. The major flaw of the choice of ω_{out}, however, is in the fact that it is difficult to synchronize the V-shaped reference signal during the inversion procedure. As a matter of fact, we have seen in the previous section that, due to the uncertainties in the “Motion” current I_M, it is difficult to exactly predict when the vehicle speed will actually start decreasing.

- The transmission ratio τ is another natural controlled variable: the inversion procedure is a smooth transition between $\tau = 1$, and $\tau = -1$; the reference trajectory for τ during a 3s-long inversion is given in Fig.8. Under the assumption that – during the inversion – the engine speed does not change, notice that
$\delta\tau = [1/\overline{\omega}_{in}]\delta\omega_{out} + [-\overline{\omega}_{out}/\overline{\omega}_{in}^2]\delta\omega_{out} = [-\overline{\omega}_{out}/\overline{\omega}_{in}^2]\delta\omega_{out}$;
henceforth, the control of τ is perfectly equivalent to the control of ω_{out}. Henceforth, it also shares the same main pitfall of ω_{out}-control: it must be designed as a tracking control problem, and the synchronization of the reference trajectory is a critical issue. However, the control of τ has another major drawback: the assumption of a constant engine speed ω_{in} during the inversion is far from being fulfilled. Using τ as controlled variable hence could cause a remarkable distortion in the behavior of the vehicle speed.
- The output acceleration $\dot{\omega}_{out}$ does not suffer of the flaws affecting ω_{out} and τ: the controller can be designed as a simple regulation loop (constant set-point); accordingly, no synchronization problems arise.

We conclude this section by presenting the results obtained with the closed-loop control on the acceleration. They are compared with the results obtained with a closed-loop control on τ (the τ-control was previously implemented on a SAME-Deutz-Fahr test prototype). These results are obtained using the newly-designed EVP-current inner-loop, and the new open-loop strategy proposed in the previous Section. In Fig.9a the inversion for a vehicle in a standard configuration (no additional loads; no trailer) are displayed; Fig.9b refers to a vehicle with an additional load of 1000Kg; Fig.9c refers to a vehicle with an additional trailer of 5000Kg. The control strategy used in these three cases is the same (no adaptations are made). It is clear the advantage of using the output acceleration as control variable, instead of τ. Notice that the deceleration and acceleration phases are very similar and symmetric, bumps and oscillations on the vehicle speed are much lower, and the overall inversion is performed in a slightly shorter time.

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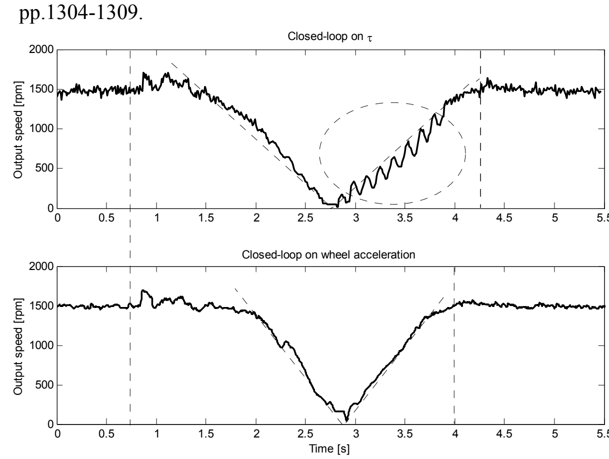

Fig.9a. Inversion of the vehicle in nominal conditions.

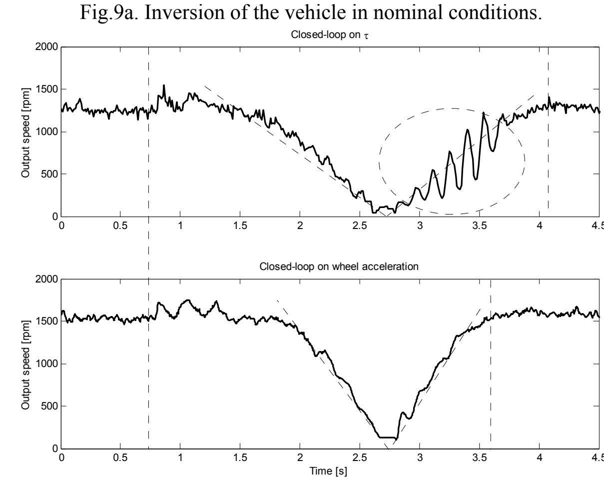

Fig.9b. Inversion of the vehicle with an additional load of 1000Kg.

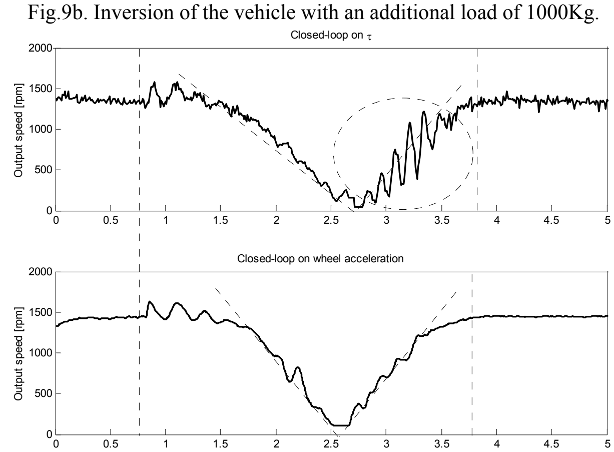

Fig.9c. Inversion of the vehicle with an additional trailer of 5000Kg.