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CONTROL OF A MULTIVARIABLE WEB WINDING SYSTEM

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Abstract — To guarantee a good productivity and a better quality of the industrial systems, it is necessary to know the evolution of its various parameters but also its operation mode. The main goal of this work is to carry out a study to improve a flatness control in a reversing cold rolling mill, precisely the web winding system part. We designate by a winding process any system applying the cycles of unwinding, transport, treatment and winding to various flat products. This system knows several constraints such as the thermal effects caused by the frictions, and the mechanical effects provoked by metal elongation, that generates dysfunctions due to the influence of the process conditions. For this installations type, the various automatisms functions, often very advanced, are realized in modular systems with distributed architecture. Our main goal is to obtain a precise thickness, with the best possible regularity. With this intention, we are preceded to the modelling and the control of the nonlinear dynamic behaviour of a web winding process.

Key-words: Web winding system (WWS), Modelling, Nonlinear Control, Linearizing Control, Diffeomorphism.

1. INTRODUCTION

In all cases of winding or unwinding of a material in strip, the flatness difficulty arises. This rolling problem is specific to the material type that it is more or less elastic, deformable or fragile. The rolling velocity determines also the velocity of metal deformation. Indeed, the principal motors behaviour of a web winding is defined by the intensity of the armature current and the evolution of the rotation velocity of the working rolls during one pass.

The dynamics of a web winding system (wws) is described by its strongly nonlinear behavior. Considering the complexity of the system due to nonlinearity and the strong coupling between the web velocity and the web tension, it is more convenient to linearize this wws. However, this model remains very depend on the set point considered and especially on the variation rate of nonlinearities. This situation pushed the researchers to be directed more and more towards the techniques of the nonlinear control based on the differential geometry theory. At present, several methods covering the subject are available (Slotine, *et al.*, 1991, Marino, *et al.*, 1995, Grcar, *et al.*, 1996). Among these methods, one finds the technique of linearization within the meaning of the input-outputs introduced by (Isidori 1989). Our work consists in presenting this principle of the technique in order to control both the web velocity and web tensions of wws and thereafter apprehending the industrial problems occurring of the rolling operation.

2. WWS DESCRIPTION

The web system is very important in a rolling mill, because its parameters determine the strip quality. Among its parameters, we quote:

- The entry and exit of the traction forces.

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- The entry and exit velocities ensured by the winders motors, and the work rolls velocity (Fig.1).
- The pressure force or the variations between the work rolls and their parallelism.

The variation of the exit strip flatness evolves because of the thermal dilation of the cylinders (El Hamzaoui, *et al.*, 2006; Rabbah and Bensassi, 2006), but also due to the elasticity forces (Schmitz and Herman, 1995). To avoid this phenomenon, the traction forces are applied to limit the elasticity of the rolled material.

The thickness control is ensured by programmable automats, which are called AGC (Automatic Gauge Control system) (Ueno and Sorao, 2004). Their goal is to maintain the strip thickness uniform in spite of the acting factors to change it. Considering the complexity of the Cold Rolling System (CRM), the modeling and the control of the WWS should be studied to minimize the flatness defaults. With this intention, we start with the development of a mathematical model describing the dynamic behavior of the system.

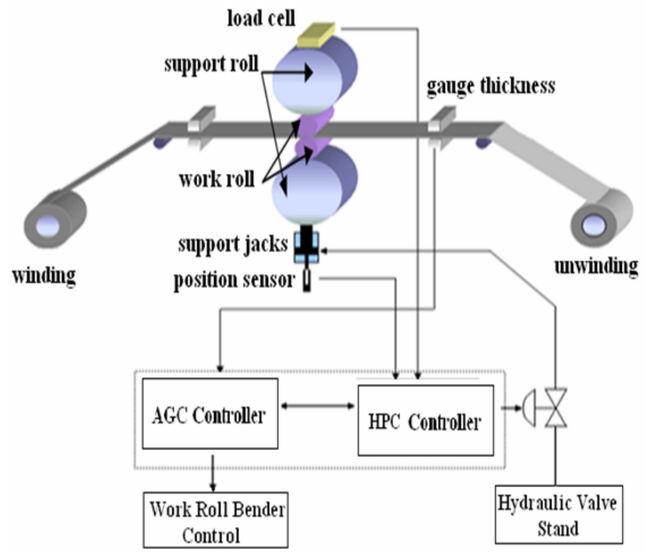

Fig.1. Interactions between the components of the cold rolling system.

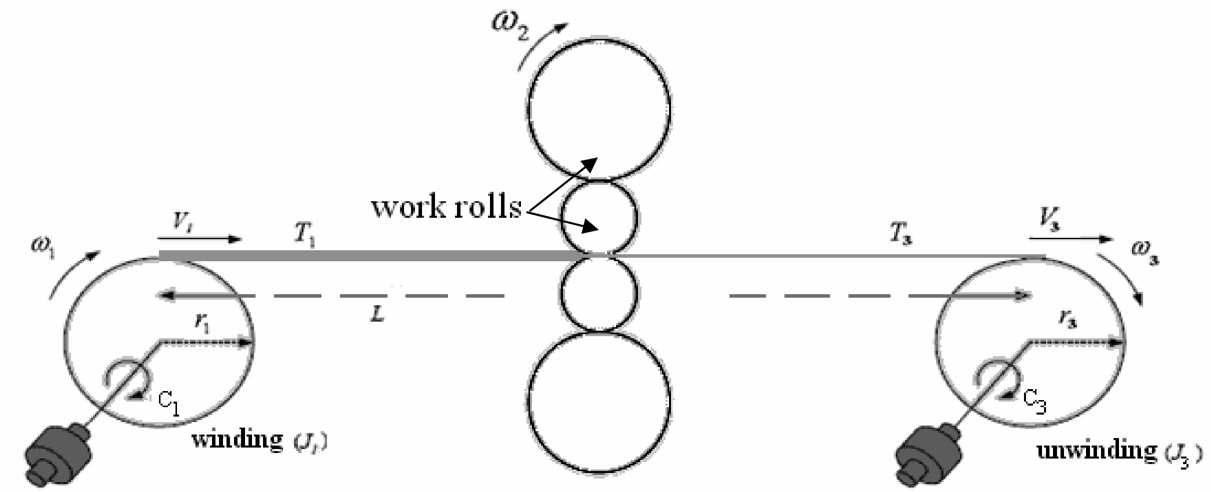

Fig.2. Synoptic model of the Web Winding System.

3. MATHEMATICAL MODEL

In this section, we present the constitutive model parts of the Winders/Strip during cold rolling (Rabbah and Bensassi, 2007).

In usual practice, the metal translates to the roll gap at a V_1 speed (Fig.2) lower than the linear circumferential speed of the work rolls V_2, and exit with $V_3 > V_2$. Close to the roll gap entry, the metal advances less quickly than the cylinder. Friction tends to involve it downstream. But close to the exit, the metal product goes more quickly. It is thus slowed down by the cylinders (resistant friction) (P. Montmitonnet, *et al.*, 2001). Therefore, when velocities change, traction is modified and results a supplementary elongation.

An empirical law (J. P. Louis, *et al.*, 1989) gives the web tension between two rolls driven by two electric motors:

$$(1)\quad \frac{dT_i}{dt} = -\frac{V}{L}.T_i + \frac{E.S}{L}.(V_{i+1} - V_i)$$

Where, V: is the linear web velocity (m/s); ω_i : is the rotational speed of i[th] roll (m/s); Ti: is the web tension between i and i+1 roll (N); L: is the web length (m); E: is the Young's modulus (Kg/mm²); S: is the web section (mm²).

3.1. Roll Dynamics Modelling

The winders radius and inertia are expressed as function of the time during the winding process (F. Claveau, 2005; S. Leirens, *et al.*, 2003). The variation of the radius *r* is given by the following equation:

$$\dot{r}(t) = -\frac{h}{2\pi}.\omega \quad (2)$$

Where *h* is the strip thickness.
As radius *r*, inertia *J* is time-varying too (Leirens, *et al.*, 2003):

$$\left(\dot{J.\Omega}\right) = J.\dot{\omega} + \dot{J}.\omega \quad (3)$$

with :

$$J = J_{motor} + J_{roll} \quad (4)$$

so :

$$J(t) = J_{ai} + \frac{\pi.\rho.l_a}{2}(r(t)^4 - r_{ai}{}^4) \quad (5)$$

Where l_a represents the strip width, J_{ai} and r_{ai} are the inertia and the radius of the i[th] motors axis respectively.
J can be written as: $J = J_{const} + J_{var}$ wich is the sum of a constant and variable inertia. Hence:

$$J_{const}(t) = J_{ai} - \frac{\pi.\rho.l_a}{2} r_{ai}{}^4 \quad (6)$$

$$J_{var}(t) = \frac{\pi.\rho.l_a}{2} r(t)^4 \quad (7)$$

Differentiating (7) with respect to time:

$$\dot{J} = 2\pi.\rho.l_a.\dot{r}.r^3 \quad (8)$$

The dynamic principle applied to a rotation movement gives:

$$\left(\dot{J.\omega}\right) = C - C_r - f.\omega \quad (9)$$

where: *C*: is the motor torque (N.m); *f*: is the coefficient of viscous friction (N.m.s/rad); C_r: is the resistive torque (N.m). Assuming that the inertias change slowly compared to the strip dynamics, therefore inertia is considered as constant now. The expression of rotational speed becomes:

$$J.\dot{\omega} = C - C_r - f.\omega \quad (10)$$

We consider that the winders Motors are linear, continuous, with constant coefficients, i.e., presenting any saturation effects or armature reaction. In this research an armature-controlled DC motor, traditionally used in industrial rolling mill, is used. The complete model of an armature-controlled DC permanent torque motor, with electric, mechanics and joint equations are presented below.

- *Electric equation :*

$$U = R.I + L.\frac{dI}{dt} + E \quad (11)$$

with: $E = k_e.\omega$; *U*: is the supply voltage of the armature (V); *R*: is the armature resistance (Ω); *I*: is the armature current (A); k_e: is the torque coefficient (N.m/A); *E*: is the electromotive force (V); ω : is the rotational speed (rad/s).

- *Coupling equation :*

$$C = k_c.I \quad (12)$$

with: $k_c = k_e = k$

By introducing equations (11) and (12) into (10) and applying the Laplace transform, we obtain:

$$\omega = \frac{\frac{k}{k^2+R.f}}{1+\frac{R.J+L.f}{k^2+R.f}.s+\frac{J.L}{k^2+R.f}.s^2}U - \frac{\frac{R+L.s}{k^2+R.f}}{1+\frac{R.J+L.f}{k^2+R.f}.s+\frac{J.L}{k^2+R.f}.s^2}C_r \quad (13)$$

We pose:

- Static gain: $k_m = \frac{k}{k^2 + R.f}$.
- Electric time-constant: $\tau_e = \frac{L}{R}$.
- Electromechanical time-constant: $\tau_{em} = \frac{R.J}{k^2 + R.f}$.

and we note: $\mu = \frac{R.f}{k^2 + R.f}$.

The expression (13) of rotational speed becomes:

$$\omega = \frac{k_m}{1+(\tau_{em}+\mu.\tau_e).s+\tau_{em}.\tau_e.s^2}U - \frac{\frac{k_m}{k}.R.(1+\tau_e.s)}{1+(\tau_{em}+\mu.\tau_e).s+\tau_{em}.\tau_e.s^2}C_r \quad (14)$$

While neglecting : μ and f; thus : τ_e in front of τ_{em}, we have : $\mu \approx 0$; $\tau_{em} = \frac{R.J}{k^2}$; $k_m = \frac{1}{k}$.

$$\omega = \frac{\frac{1}{k}}{1+\tau_{em}.s}U - \frac{\frac{R}{k^2}}{1+\tau_{em}.s}C_r \quad (15)$$

While applying inverse Laplace transform to the last expression, we obtain the final expression of the rotational speed as:

$$\frac{d\omega(t)}{dt} = -\frac{1}{\tau_{em}}.\omega(t) - \frac{1}{J}.C_r + \frac{1}{\tau_{em}.k}.U(t) \quad (16)$$

The resistive torque C_r consists of several couple: $C_r = C_i + C_p + C_{web}$, with:

C_i : inertia torque , C_p : losse torque , C_{web} : torque developed by the web , $C_{web} = (T_i - T_{i+1}).g.r_i$.
$0 < g < 1$: *sliding coefficient*.

While replacing it in the equation (16), it becomes:

$$\frac{d\omega(t)}{dt} = -\frac{1}{\tau_{em}}.\omega(t) - \frac{g.r_i}{J}.(T_i - T_{i+1}) - \frac{C_i + C_p}{J} + \frac{1}{\tau_{em}.k}.U(t) \quad (17)$$

Using results above (equations (1) and (17)), with neglecting the inertia and losses torque, the model below was built:

$$\frac{dT_1}{dt} = -\frac{V_2}{L}T_1 + \frac{E.S}{L}.V_2 - \frac{E.S}{L}.V_1 \quad (18)$$

$$\frac{dT_3}{dt} = -\frac{V_2}{L}T_3 - \frac{E.S}{L}.V_2 + \frac{E.S}{L}.V_3 \quad (19)$$

$$\frac{dV_1(t)}{dt} = -\frac{1}{\tau_{em1}}.V_1(t) + \frac{g.r_1^2}{J_1}.T_1 + \frac{r_1}{\tau_{em1}.k_1}.U_1(t) \quad (20)$$

$$\frac{dV_2(t)}{dt} = -\frac{1}{\tau_{em2}}.V_2(t) - \frac{g.r_2^2}{J_2}.(T_1 - T_3) + \frac{r_2}{\tau_{em2}.k_2}.U_2(t) \quad (21)$$

$$\frac{dV_3(t)}{dt} = -\frac{1}{\tau_{em3}}.V_3(t) - \frac{g.r_3^2}{J_3}.T_3 + \frac{r_3}{\tau_{em3}.k_3}.U_3(t) \quad (22)$$

with:

$$V_2(t) = \omega_2(t).r_2(t) \quad (23)$$

Table 1 contains the operating condition used in this analysis.

Table 1 Parameters of the operating conditions.

Web length between winder and unwinder	10.15 m
Young's modulus	0.16 10^9 kg/m²
Web section	0.19 mm²
Sliding coefficient	0.8
Strip or web thickness	1.6 mm
Strip or web width (largeur)	600 mm
Diameter or work rolls	0.45 m
Nominal torque of unwinding/winding motors	700 kN.m
Nominal velocity of unwinding/winding motors	120 Rpm
Rolling speed	1400 mpm

4. NONLINEAR CONTROL OF THE WWS TENSIONS AND VELOCITY

This work part illustrates a direct application of the nonlinear control system of the WWS concerning the reversing cold rolling mill. We determine the relative degree of each control output to establish the decoupling matrix then the development of the external set point which uncouples the three inputs.

Finally, the physical control design of the uncoupled and linearized system is carried out.

4.1. Variables to be controlled

Let us consider the definite nonlinear model by the state representation (E) which can be put in the general form of the nonlinear affine control system:

$$(E): \begin{cases} \dot{x} = f(x) + g(x).u & x \in \Re^n \quad u \in \Re^m \\ y = h(x) & y \in \Re^p \end{cases}$$

with :

$$x^T = [V_1 \;\; T_1 \;\; V_2 \;\; T_3 \;\; V_3] \quad y^T = [T_1 \;\; V_2 \;\; T_3] \quad u^T = [U_1 \;\; U_2 \;\; U_3].$$

$$f(x) = \begin{bmatrix} \frac{gr_1^2}{J_1}x_2 - \frac{1}{\tau_{em1}}x_1 \\ -\frac{1}{L}x_2x_3 - \frac{ES}{L}x_1 + \frac{ES}{L}x_3 \\ -\frac{gr_2^2}{J_2}x_2 + \frac{gr_2^2}{J_2}x_4 - \frac{1}{\tau_{em2}}x_3 \\ -\frac{1}{L}x_3x_4 - \frac{ES}{L}x_3 + \frac{ES}{L}x_5 \\ -\frac{gr_3^2}{J_3}x_4 - \frac{1}{\tau_{em3}}x_5 \end{bmatrix} \quad h(x) = \begin{bmatrix} x_2 \\ x_3 \\ x_4 \end{bmatrix}$$

$$g(x) = [g1, g2, g3] = \begin{bmatrix} \frac{r_1}{\tau_{em1}.k_1} & 0 & 0 \\ 0 & 0 & 0 \\ 0 & \frac{r_2}{\tau_{em2}.k_2} & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \frac{r_3}{\tau_{em3}.k_3} \end{bmatrix}$$

These parameters characterize the system: it is multivariable, strongly coupled, nonlinear and time-varying. The variables to be controlled are:

$$y = \begin{pmatrix} y_1(x) \\ y_2(x) \\ y_3(x) \end{pmatrix} = \begin{pmatrix} T_1 \\ V_2 \\ T_3 \end{pmatrix}$$

4.2. Linearizing Control of the WWS and decoupling problem

We know that the nonlinear system is affine control. We try to determine the control law: $u(x) = \alpha(x) + \beta(x)v$ so that each output y_i will be influenced by only one input v_i. For that, a successive derivative Lie is calculated $L_g L_f^k h_i(x)$ ($k=0, \ldots, k_i$) such as : $L_g L_f^{k_i} h_i(x) \neq 0$.

The linearization condition making it possible to check if a nonlinear system admits a linearization input-output is the relative degree order (r) of the system (Von Raumer, *et al.*, 1994; Belabbes, *et al.*, 2001). For the output system, we have:

$$\begin{cases} y_1(x) = x_2 = h_1(x) \\ y_2(x) = x_3 = h_2(x) \\ y_3(x) = x_4 = h_3(x) \end{cases}$$

The control appears in the first derivation of y_2, whereas for y_1 and y_3 just after the 2nd derivation, which gives:

$$(24)\quad \begin{cases} \ddot{y}_1 = \psi_1(x) + \psi_2(x).U_1 + \psi_3(x).U_2 \\ \dot{y}_2 = \psi_4(x) + \psi_5(x).U_2 \\ \ddot{y}_3 = \psi_6(x) + \psi_7(x).U_2 + \psi_8(x).U_3 \end{cases}$$

with :

$$\psi_1 = L_f^2 h_1 = \frac{1}{L^2} x_2 x_3^2 + \frac{E.S}{L^2} x_1 x_3 - \frac{E.S}{L^2} x_3^2 + \frac{E.S - x_2}{L}$$
$$* \left(-\frac{g.r_2^2}{J_2} x_2 + \frac{g.r_2^2}{J_2} x_4 - \frac{1}{\tau_{em2}} x_3 \right) + \frac{E.S}{L} . \left(\frac{1}{\tau_{em1}} x_1 - \frac{g.r_1^2}{J_1} x_2 \right)$$

$$\psi_2 = L_{g_1} L_f h_1 = -\frac{E.S}{L.\tau_{em1}.k_1}$$

$$\psi_3 = L_{g_2} L_f h_1 = \frac{1}{\tau_{em2}.k_2} \left(\frac{E.S - x_2}{L} \right)$$

$$\psi_4 = L_f h_2 = -\frac{g.r_2^2}{J_2} x_2 + \frac{g.r_2^2}{J_2} x_4 - \frac{1}{\tau_{em2}} x_4$$

$$\psi_5 = L_{g_2} h_2 = \frac{1}{\tau_{em2}.k_2}$$

$$\psi_6 = L_f^2 h_3 = \frac{1}{L^2} x_4 x_3^2 + \frac{E.S}{L^2} x_3^2 - \frac{E.S}{L^2} x_3 x_5 - \frac{E.S - x_4}{L}$$
$$* \left(-\frac{g.r_2^2}{J_2} x_2 + \frac{g.r_2^2}{J_2} x_4 - \frac{1}{\tau_{em2}} x_3 \right) + \frac{E.S}{L} . \left(-\frac{1}{\tau_{em3}} x_5 - \frac{g.r_3^2}{J_3} x_4 \right)$$

$$\psi_7 = L_{g_2} L_f h_3 = \frac{1}{\tau_{em2}.k_2} \left(\frac{E.S - x_4}{L} \right)$$

$$\psi_8 = L_{g_3} L_f h_3 = \frac{E.S}{L.\tau_{em3}.k_3}$$

The total relative degree of the output is ($r=r_1+r_2+r_3=5$); thus the system is completely observable, and consequently it is exactly linearisable by diffeomorphism and nonlinear state feedback.

According to the derivative Lie (24), we obtain the following form:

$$(25)\quad [\ddot{y}_1 \quad \dot{y}_2 \quad \ddot{y}_3]^T = \Delta_0(x) + \Delta(x).u$$

where:

$$\Delta_0(x) = \begin{bmatrix} L_f^2 h_1(x) \\ L_f h_2(x) \\ L_f^2 h_3(x) \end{bmatrix} = \begin{bmatrix} \psi_1(x) \\ \psi_4(x) \\ \psi_6(x) \end{bmatrix}$$

and:

$$\Delta(x) = \begin{bmatrix} L_{g_1} L_f h_1 & L_{g_2} L_f h_1 & 0 \\ 0 & L_{g_2} h_2 & 0 \\ 0 & L_{g_2} L_f h_3 & L_{g_3} L_f h_3 \end{bmatrix} = \begin{bmatrix} \psi_2(x) & \psi_3(x) & 0 \\ 0 & \psi_5(x) & 0 \\ 0 & \psi_7(x) & \psi_8(x) \end{bmatrix}$$

The matrix $\Delta(x)$ is called the decoupling matrix. So that the state feedback can exist, it is necessary that this matrix should be nonsingular (invertible) (Carlos Canudas de Wit, 2000).

$$\det(\Delta(x)) = \psi_2.\psi_5.\psi_8 \neq 0.$$

To linearize the system, we applie the nonlinear state feedback following (Isidori, 1989; Chibani, 2005):

$$(26)\quad u = -\Delta^{-1}(x).\Delta_0(x) + \Delta^{-1}(x).v = \alpha(x) + \beta(x).v$$

where:

$$\Delta^{-1}(x) = \begin{bmatrix} \frac{1}{\psi_2(x)} & \frac{-\psi_3(x)}{\psi_2(x)\psi_5(x)} & 0 \\ 0 & \frac{1}{\psi_5(x)} & 0 \\ 0 & \frac{-\psi_7(x)}{\psi_5(x)\psi_8(x)} & \frac{1}{\psi_8(x)} \end{bmatrix}$$

and: v is an external set point what leads to three subsystems mono variable, uncoupled and linear. By replacing the expression (26) in that given into (25), we obtain a linear system completely uncoupled from the form:

$$[\ddot{y}_1 \quad \dot{y}_2 \quad \ddot{y}_3]^T = [v_1 \quad v_2 \quad v_3]^T$$

To be able to make the synthesis of linearizing control law, we must determine a variables change using a diffeomorphism made up of the variables resulting from successive derivations from the output y.

Change of co-ordinates: $z = \Phi(x)\,;\, z \in \Re^n$

thus:
$$\begin{cases} z_1^1 = y_1 = x_3 \\ z_1^2 = \dot{y}_1 \\ z_2^1 = y_2 = x_4 \\ z_3^1 = y_3 = x_5 \\ z_3^2 = \dot{y}_3 \end{cases}$$

After transformation and looping:

$$\begin{cases} \dot{z} = Az + Bv \\ y = Cz \end{cases}$$

Such us:
$$\begin{cases} \dot{z}_1^1 = z_1^2 \\ \dot{z}_1^2 = v_1 = -k_1(z_1 - z_1^c) \\ \dot{z}_2^1 = v_2 = -k_2(z_2 - z_2^c) \\ \dot{z}_3^1 = z_3^2 \\ \dot{z}_3^2 = v_3 = -k_3(z_3 - z_3^c) \end{cases}$$

5. RÉSULTS AND DISCUSSION

The simulation results of the WWS control by the nonlinear control of the type linearization input-output by nonlinear state feedback are illustrated by the figures below. Figure 4 illustrates well the response of the D. C motor velocity through the work rolls, similar to that of a 1st order system without overshoot, with a response time of 0.05 seconds. We observe the perfect follow-up of the reference velocity (20m/s). Figure 5 present the strip tension (traction forces) of the work rolls upstream and downstream too. With starting, the upstream tension manifests a low oscillation which is cancelled quickly. However, the downstream tension shows a very good follow-up of the reference trajectory. The two tensions are stabilized at a response time of 0.05 seconds.

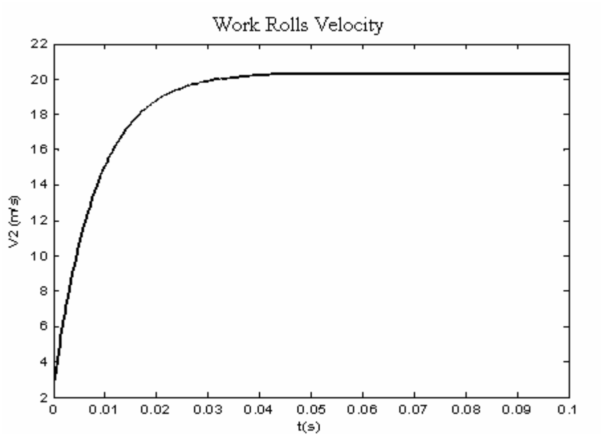

Fig.3. Velocity through the work rolls.

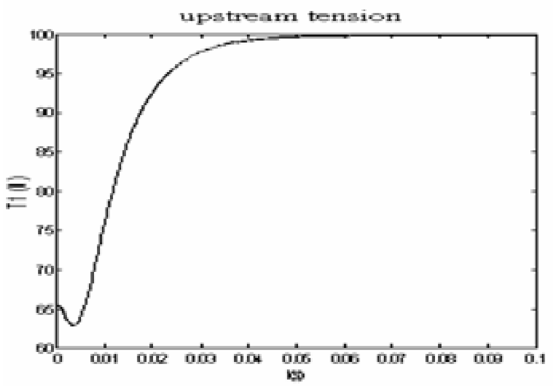

Fig.4. Upstream Strip Tension.

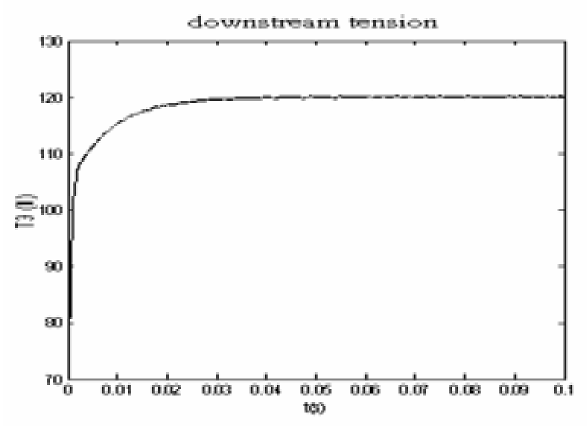

Fig.5. Downstream Strip Tension.

6. CONCLUSION

The nonlinear control of the type linearization input-output by nonlinear state feedback is an application of the differential geometry which is based on the derivative Lie of the output control, and then an adequate choice state-feedback control cancels the nonlinearity and allows a very good follow-up of the references trajectories. For our case, the nonlinear control is affected directly by the torques, which requires the development of an estimator to evaluate the disturbances and to make the robust control.

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