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ANALYSIS AND REJECTION OF OUT OF PLANE MOVING WEB VIBRATIONS IN WEB HANDLING SYSTEMS

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Abstract

The system under study, handling web material such as textile, is quite common in industry: it is composed of an unwinder, a traction motor, and a winder and presents the inherent difficulties of web transport systems. In web transport systems, roll eccentricity and non-circularity create web tension disturbances and thus longitudinal and out-of-plane web vibrations. This paper focuses firstly on the construction of a nonlinear plant model and its validation on a real benchmark, with longitudinal web vibrations. Out of plane web vibrations have been measured and the influences of free and forced vibrations have been observed, with resonance occurrence. The second part of the paper deals with the active control of the out of plane vibrations rejected using a new actuator strategy for such plant: a loudspeaker. A PI controller has been successfully experimented.

INTRODUCTION

Web systems handling material such as textile, paper, polymer or metal are very common in industry. The study of the modeling and the control of web handling systems are carried out for now several decades. But the increasing requirement on the running speed of web handling machines and the handling of thinner web material oblige us to search for more sophisticated control strategies [1, 16] and to take into account vibrations. A test bench has been set up in order to analyze web handling problems, characterize web vibrations and explore active control of vibration strategies. The system under study is quite common in industry: it is composed of an unwinder, a traction motor, and a winder and presents the inherent difficulties of web transport systems. The detailed model description is presented in [16]. In this paper only the main laws on which the model is based will be remained.
In this paper, the first part reminds the main laws of physical mechanics used for

modelisation of web handling systems. These laws enabled us to build a non-linear model which has been afterwards identified on our experimental bench composed of three motors (figure 1). This bench shows the inherent difficulties of web transport systems. The non-linear model built in Matlab/Simulink environment is used as a simulator. The second part presents simulated and experimental vibration results. The vast literature on axially moving material vibrations is reviewed in the papers by Ulsoy [2] *et al.* and Lee and Mote [3]. Analytical analysis and numerical implementation of the axially moving PDE have been developed in the articles by Ulsoy [2], Lee and Mote [3], Abrate [5] as well as recent tridimentionnal FEM analysis by Laukkanen [6]. Wickert and Mote [4] investigated the transverse vibrations of travelling strings and beams. Longitudinal vibrations have been observed on the three motor bench and compared to theoretical results using the simulation of the model. Then, out-of-plane vibrations will also be measured and some specific phenomena and resonance are pointed out. The third part of the paper deals with the active control of such out-of-plane vibrations. Several strategies have been proposed to reduce web vibrations, mostly concerning longitudinal vibrations: Rahn [8, 9] experimented counter vibrations generated by a mechanical actuator, Pagilla [13] and Bouscayol [11] use an active dancer, C. Shin [10], K. H. Shin [18] and Knittel [14, 15] focused on the unwinder command strategy. The idea presented here is to test the possibility of use of an acoustic actuator: the loudspeaker [12]. The first results using PI controllers are presented.

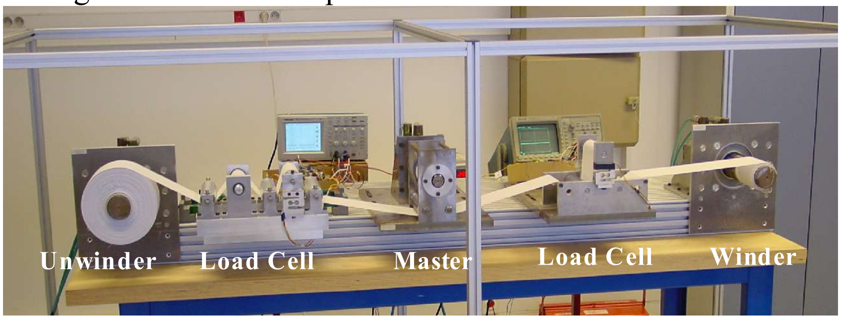


Figure 1 – Experimental setup with 3 motors and 2 load cells

PROCESS MODEL

The model of web transport systems, briefly presented, was derived from the model of the web tension between two consecutive rolls and the model of the velocity of each roll.

Web tension calculation : Web modeling is based essentially on three physical laws which allow us to calculate web tension between two rolls: Hooke's law of the web, Coulomb's law which describes the web tension variation due to friction between web and roll, the law of mass conservation which describes the coupling between web velocity and web tension.

Tension-velocity relation: the model of the web tension between two consecutive rolls was obtained since web length on the wrap angle can be neglected compared to the web length without contact between two rolls [1]:

$$L\frac{dT_k}{dt} = ES(V_{k+1} - V_k) + T_{k-1}V_k - T_k(2V_k - V_{k+1}) \qquad (1)$$

where T_k and V_k represent respectively the web tension and web velocity before the roll numbered k.

<u>Web velocity calculation:</u> the velocity of a roll can be obtained with a torque balance:

$$J_k \frac{dV_k}{dt} = R_k^2 (T_k - T_{k-1}) + R_k K_k U_k + C_f \tag{2}$$

where $K_k U_k$ is the motor torque (if the roll is driven) and C_f is the sum of the friction torque. We can notice at this point that at the unwinder and the winder the inertia J_k and the Radius R_k are time dependent and vary substantially during the process operation. The web behavior simulator has been constructed in the Matlab/Simulink software environment.

ANAYSIS AND MEASUREMENT OF WEB VIBRATIONS

Longitudinal vibrations

Geometric imperfections, due to the unwinding wound roll storage, as well as rotor unbalance are some of the non circularity defaults that will induce speed and tension perturbations of the web handling process. These perturbations have been observed on the experimental three motors bench and measured with load cells (figure 2):

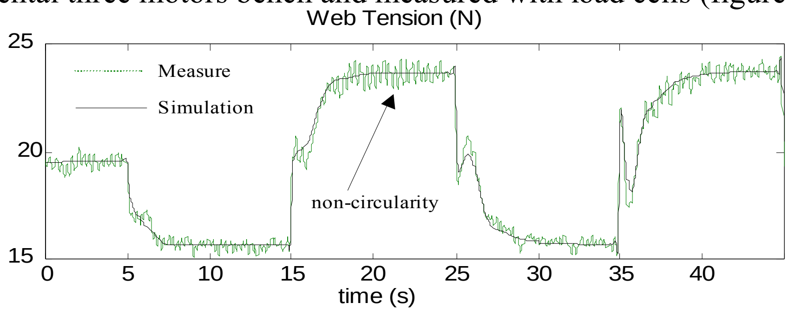


Figure 2 – Simulations and measures for a non circular unwinder

The calculation of a sliding Fast Fourrier Transformation (FFT) of the measured unwinder web tension is represented in figure 3. The winder and unwinder angular velocity are changing with time and therefore the forced vibration have time varying frequencies, as observed in figure 3 where up to six different harmonics of the unwinder frequency are represented. Nevertheless, the previous presented web model cannot reproduce these longitudinal forced web elongations (see figure 2). Thus, a more realistic unwinder roll radius has been modeled including eccentricity and non circularity as following:

$$R(t) = R_0(t) + \sum_{m=1}^{\infty} A_m \sin(\theta_m(t)) \tag{3}$$

where $R_0(t)$ is the nominal radius of the roll at time t and :

$$\theta_m(t) = m\int_{t_0}^{t} \Omega \, d\tau + \delta\theta_m(t) \tag{4}$$

$\theta_m(t)$ are the phases of the harmonic orders and Ω the angular velocity. With eqs. (3) and (4), the simulator can now reproduce the longitudinal web elongation variations.

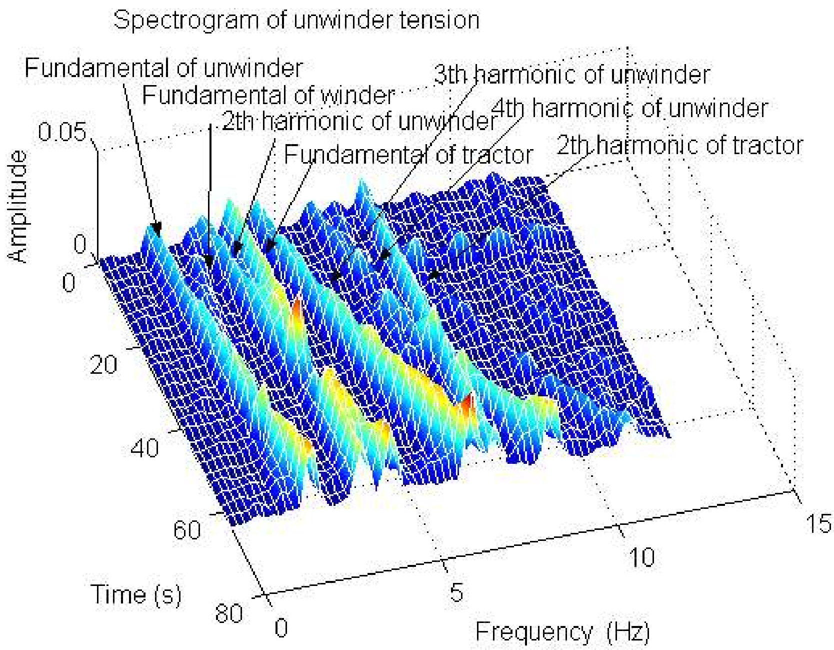


Figure 3 - Spectrogramm of unwinder web tension.

Out of planes vibrations

Geometric imperfections of the wound roll create not only longitudinal but also out of plane web vibrations. These last vibrations can be modeled by considering the vertical position y of a point x of a small width web:

$$(V^2 - c^2)\frac{\partial^2 y}{\partial x^2} + 2V\frac{\partial^2 y}{\partial x\, \partial t} + \frac{\partial^2 y}{\partial t^2} = 0 \tag{5}$$

where ρ , T , V represent respectively the web density, tension and linear speed. The first natural frequency f_1 is given by:

$$f_1 = \frac{c}{2L}\left(1 - \frac{V^2}{c^2}\right) \text{ with } c = \sqrt{\frac{T}{\rho}} \tag{6}$$

c is the out-of-plane wave velocity. The web width equals 0.1 m, the web thickness 260 µm, the web span between the unwinder and the next roll is 1,5 m. Transversal vibrations are measured by a laser sensor having 4 µm resolution and a sampling frequency of 1000 Hz. The web speed was kept constant. The unwinder angular rotation changed from 1.71 rounds per second to 1.97 rounds par second in 20s. The calculated spectrogram of the measured out-of-plane vibrations is given in figure 4. Several points can be pointed out:

- the harmonic’s frequencies of the measured forced vibrations are proportional to the angular velocity frequency of the roll close to the measured web point. But the influence of other rolls cannot be seen, contrary to the case of longitudinal vibrations. Therefore, the mechanical architectures of our web tension sensor and master entrainment, called “Omega architecture”, filter mechanically the out-of-plane vibrations, but not the longitudinal ones.

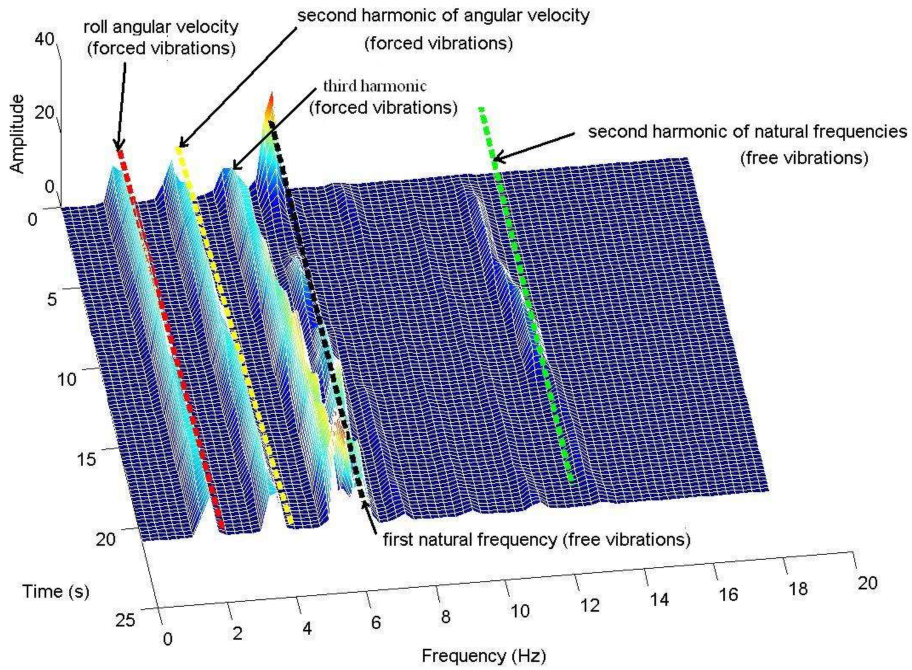


Figure 4 - Spectrogram of out-of-plane vibrations for T = 5N.

- for out-of-plane as well as for longitudinal vibrations, the amplitude of vibrations increases when the roll radius decreases (Fig 4). This can be explained by a higher non-circularity close to the core. In other words, for higher radius, the wound web smooths the non-circularity. For small variations ΔR_u and ΔT_u, the other variables staying constant, eq. (2) leads to the relation:

$$\Delta T_u \, R_u \; + \; T_u \, \Delta R_u \; = 0 \tag{7}$$

thus :

$$\left| \Delta T_u \right| = \left| T_u \frac{\Delta R_u}{R_u} \right| \tag{8}$$

Obviously, the tension perturbations of the unwinder increase when the radius R_u decreases.

- the interference of free and forced vibrations: as illustrated in figure 5, the superposition of free (first as well as second natural frequency) and forced vibrations creates a resonance of high amplitude. This has to be taken into account, from a practical point of view, in the unwinding-winding process adjustment.
- in reality, longitudinal and out-of-plane vibrations are coupled according to the following PDE equations [7]:

$$\rho\left(\frac{\partial^2 u}{\partial t^2} + 2V\frac{\partial^2 u}{\partial t\,\partial x} + V^2\frac{\partial^2 u}{\partial x^2} + \dot{V}\frac{\partial u}{\partial x}\right) - EA\frac{\partial^2 u}{\partial x^2} = -\rho\dot{V} \tag{9}$$

$$\rho\left(\frac{\partial^2 y}{\partial t^2} + 2V\frac{\partial^2 y}{\partial t\,\partial x} + V^2\frac{\partial^2 y}{\partial x^2} + \dot{V}\frac{\partial y}{\partial x}\right) - EA\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x}\frac{\partial y}{\partial x}\right) = p \tag{10}$$

where $y(x, t)$, $u(x, t)$, E, A, $\rho(x, t)$ represent respectively the transversal displacement, the longitudinal displacement, the Young modulus, the web section and load per length. A numerical study shows that the coupling in the vibrations through the web tension is not significant, and this was observed experimentally.

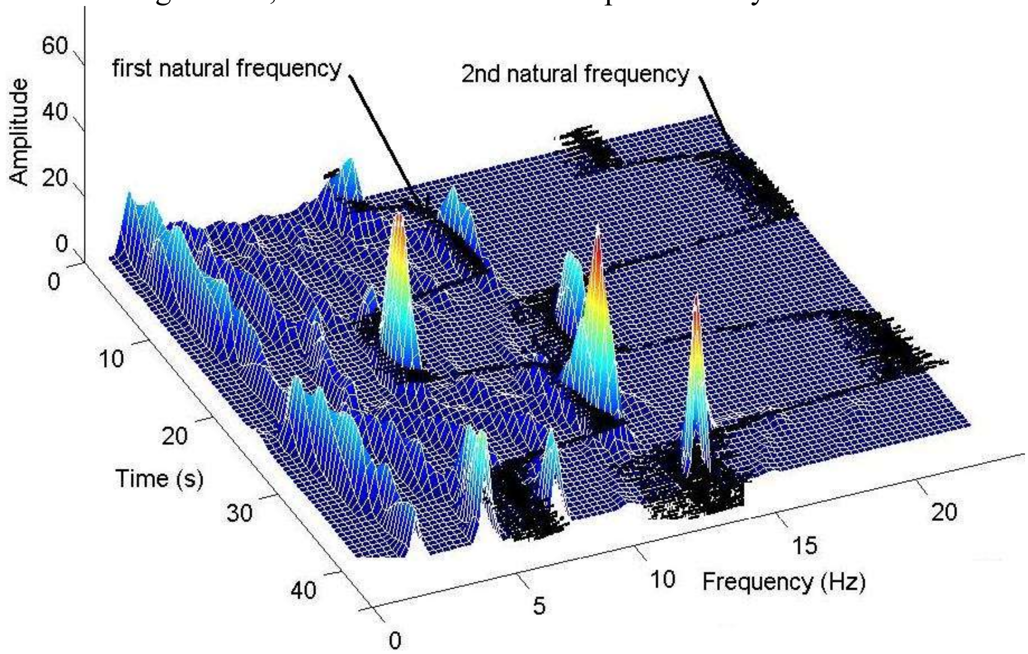


Figure 5 - superposition of free and forced vibration in the case of tension jumps between 5 N (first natural frequency at 6.5 Hz) and 15 N (first natural frequency at 13 Hz)

ACTIVE CONTROL: OUT OF PLANE VIBRATIONS REJECTION

Active control of web vibrations has been so far implemented through mechanical actuators such as jacks or active rolls. The idea presented here is to test the possibility of the use of an acoustic actuator: a 20 W loudspeaker. The out-of-plane web displacements are measured by a laser sensor.

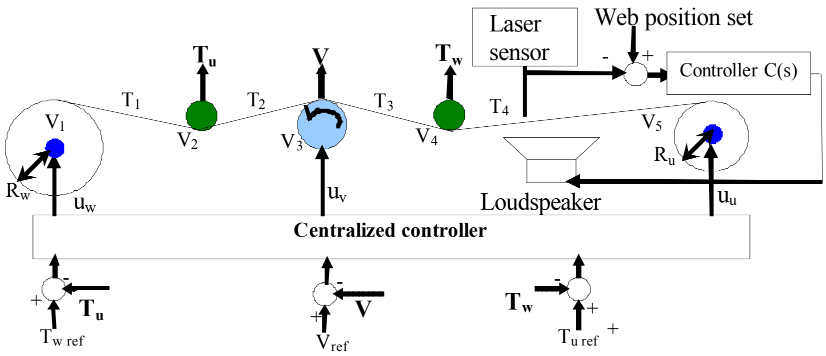
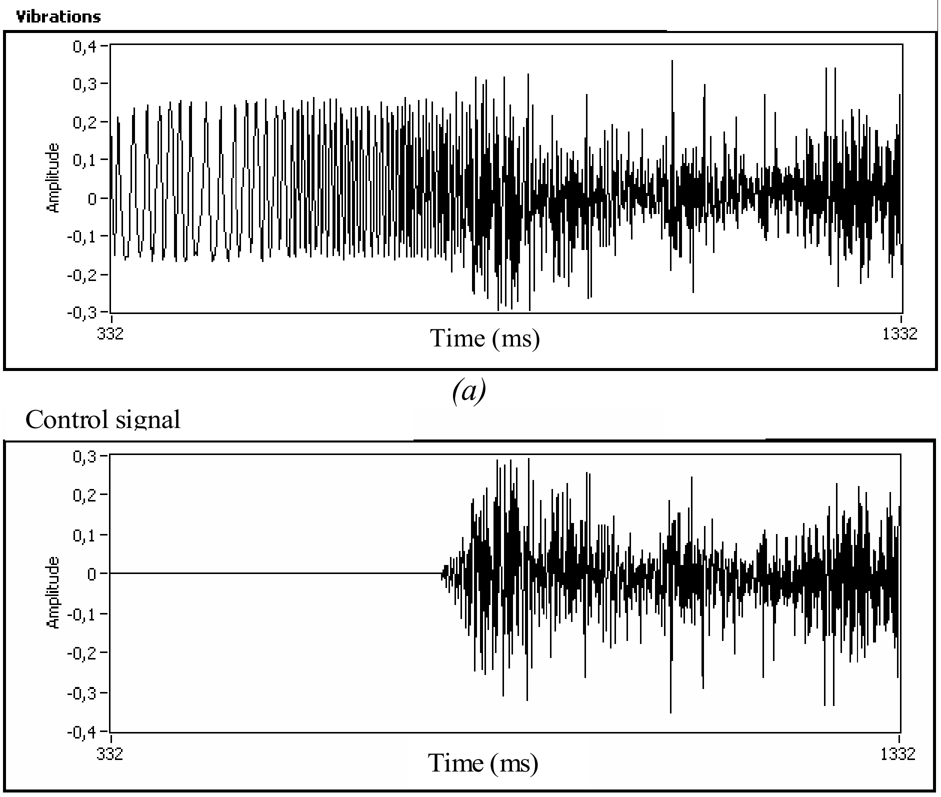


Figure 6 - principle of the test bench

The loudspeaker is controller by a PI controller implemented in a PC running in Labview® environment. The controller has been adjusted as follows :

$$C(s) = K_C \frac{(1+0{,}1s)}{s} \quad \text{with } K_C=75$$

Experimental results show the evolution of the out-of-plane vibrations without and with loudspeaker compensation. The evolution of the loudspeaker control signal is also presented.



(b)

Figure 7 - (a) Evolution of out-of-plane vibrations without and with loudspeaker compensation. (b) Evolution of the control signal.

The use of a PI controller roughly decreases the amplitude of out-of-plane vibrations by 50%. This experimental result presents high interest in industry applications and validates the approach of web active control using a loudspeaker for small web bandwidth. Further developments will focus both on acoustic improvements (better design of loudspeaker and inherent acoustic parameters) and on more sophisticated control strategies, such as phase-lock loop (PLL) or adaptive algorithm using feedforward.

CONCLUSION

We proposed and validated the use of a loudspeaker for an active control of web vibrations. This approach enables a 50% reduction of the out-of-plane vibrations. More sophisticated control strategies are in development on the test bed and with simulation tools in order to improve the performance of the system before an industrial implementation.

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