



**HAL**  
open science

## **Analysis of a turning process strongly coupled to a vibro-impact nonlinear energy sink**

Tao Li, Etienne Gourc, Sébastien Seguy, Alain Berlioz

### ► **To cite this version:**

Tao Li, Etienne Gourc, Sébastien Seguy, Alain Berlioz. Analysis of a turning process strongly coupled to a vibro-impact nonlinear energy sink. Joint Conference on Mechanical, Design Engineering & Advanced Manufacturing, 18-20 juin 2014, Toulouse, France, Jun 2014, Toulouse, France. ⟨hal-02054141⟩

**HAL Id: hal-02054141**

**<https://hal.science/hal-02054141v1>**

Submitted on 1 Jul 2021

**HAL** is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.



HAL Authorization

# Analysis of a turning process strongly coupled to a vibro-impact nonlinear energy sink

Tao LI <sup>1</sup>, Etienne GOURC <sup>1</sup>, Sébastien SEGUY <sup>1</sup>, Alain BERLIOZ <sup>2</sup>

(1): Université de Toulouse; INSA; ICA (Institut Clément Ader); F-31077 Toulouse, France  
E-mail : {tli, seguy}@insa-toulouse.fr  
etienne.gourc@gmail.com

(2): Université de Toulouse; UPS; ICA (Institut Clément Ader); F-31062 Toulouse, France  
E-mail : alain.berlioz@univ-tlse3.fr

**Abstract:** Recently, it has been demonstrated that a vibro-impact nonlinear energy sink (VI-NES) can be used to efficiently mitigate vibration of a Linear Oscillator (LO) under transient loading by irreversible energy transfer from LO to VI-NES. In this paper, the possibility of application of VI-NES to control the instability of chatter in turning as a passive control method is studied as a typical delay system. The dynamic responses of machine tool strongly coupled to VI-NES in turning are investigated theoretically by supposing that the tool just vibrates in the predominant direction and numerically with the help of Matlab dde23 algorithm. Three typical response regimes are obtained through numerical simulation under different parameters, that is, stable oscillation and complete suppression of regeneration chatter, relaxation oscillation and partial suppression, and unstable oscillation. Finally, a developed experimental setup is presented and the first test to acquire the stability lobe without VI-NES is accomplished.

**Key words:** Dynamics, Turning, Chatter, Nonlinear Energy Sink, Vibro-Impact

## 1- Introduction

The surface quality of parts produced by machining operation is strongly affected by the well know regenerative chatter. The chatter instability is induced by the time delay between two consecutive cutting teeth, linked to spindle speed. By the effect of some external disturbance, the tool start damped oscillation relative to the workpiece, and the surface roughness is undulated. For the consecutive cutting teeth, the chip thickness is modulated. This regenerative mechanism is well known and presented first by Tobias [TF1]. Since this work, many researcher have improved the knowledge by the well know stability lobe representation, see e.g. [AB1] [MI1] [ML1].

Various techniques for chatter suppression have been studied. The idea behind various techniques is to disturb the time delay by variable pitch tool or spindle speed variation [SI1]. The use of magnetic bearing spindle is also a promising way for chatter

reduction [GS1].

Another approach to reduce chatter is the use of linear tuned vibration absorbers, presented first by Ormondroyd and Hartog in 1928 [OH1]. More recently, an analytical optimized method was presented for linear absorbers in the context of chatter [S1]. These linear absorbers are successfully applied on boring process [MB1]. Active absorbers have been also proposed with piezoelectric tool [HD1]. However all this linear absorbers are limited by the small frequency bandwidth, and in practice the efficiency is not interesting for the machinist.

The idea of attaching a nonlinear oscillator to a turning machine is relatively recent [W1] [GS2] [NS1]. It has been showed that various systems consisting of a primary structure and of a strong nonlinear oscillator (SNO) are able to transfer irreversibly the energy from the primary structure to SNO with almost any mode of the primary system and therefore over broad frequency bands through resonance capture [GG1] [VR1] [GM2]. Such a feature has revealed interesting behavior for vibration mitigation. In recent studies, it has been demonstrated that systems comprising with a Nonlinear Energy Sink (NES) can exhibit regimes which are not related to fixed points, and cannot be explained using local analysis. These regimes are related to relaxation oscillation of the slow flow and can be efficiently analyzed by applying the multiple scale method. A VI-NES coupled to linear systems and convenient for engineering application is analytically treated using multiple scale method in [G1] and further studied experimentally in [GM1].

The dynamics of the turning machine have been studied by applying different and efficient methods. A systematic approach comprised of a path-following scheme, the method of multiple scales, the method of harmonic balance, Floquet theory, and numerical simulations to investigate its local and global dynamics and stability has been proposed in [NN1]. Specially, a method mixed with the method of multiple scales and that of harmonic balance is proposed in [LZ1] to

analyze the delay system like the turning system. Moreover, the analytic approach to obtain the complete solutions for systems of delay differential equations like turning is presented in [AU1].

In this paper, the possibility of regenerative chatter suppression using a Vibro-Impact Nonlinear Energy Sink (VI-NES) is analyzed for turning process. The paper is organized as follow. In the next section, the model considered in this study is described. Then, different response regimes accompanied with numerical simulation are presented. Finally, the experimental setup is presented.

## 2- Mechanical Model with Vibro Impact NES

The model developed herein consists of a cutting tool on a lathe with an embedded VI-NES. The cutting tool is assumed to vibrate only on its first flexible mode and the workpiece is considered to be rigid. A schematic of the model is given in Fig. 1.

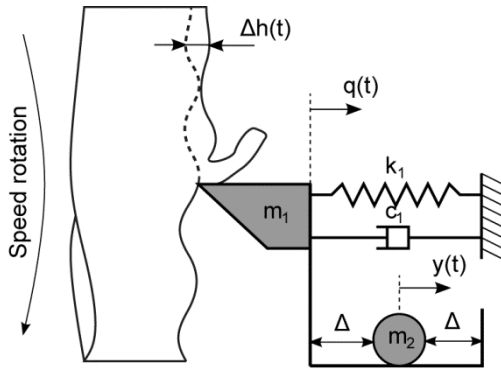


Figure 1: Mechanical model.

The governing equation of motion, between impact, are expressed as:

$$\begin{cases} m_1 \frac{d^2 q}{dt^2} + c_1 \frac{dq}{dt} + k_1 q = F_c \\ m_2 \frac{d^2 y}{dt^2} = 0 \quad \forall |q - y| < \Delta \end{cases} \quad (1)$$

where  $q$  and  $y$  represent the tool tip and VI-NES displacement respectively.  $m_1$ ,  $c_1$  and  $k_1$  are the mass, damping and stiffness of the tool and  $m_2$  is the mass of the VI-NES.

$F_c$  is the cutting force taking into account the regenerative effect. The cubic polynomial form due to Shi and Tobias [ST2] is expressed as follow:

$$F(\Delta h(t)) = p(\rho_1 h(t) + \rho_2 h(t)^2 + \rho_3 h(t)^3) \quad (2)$$

where  $p$  is the chip width (depth of cut),  $h(t)$  is the chip

thickness and  $\rho_i$  ( $i=1...3$ ) are specific cutting coefficients obtained by fitting the experimental cutting force measurement with a third order polynomial [DB1]. The chip thickness  $h(t)$  is expressed as follow:

$$h(t) = h_0 + \Delta h(t) \quad (3)$$

$h_0$  represent the nominal chip thickness in the absence of vibration and  $\Delta h(t)$  is the chip thickness variation gives by:

$$\Delta h(t) = q(t - \tau) - q(t) \quad (4)$$

where  $q(t - \tau)$  is the delayed position of the tool and  $\tau$  is the time delay between two workpiece revolutions  $\tau = 2\pi/\Omega$ , with  $\Omega$  the spindle speed. In the steady state,  $q(t) = q_0$ . Substituting Eq. 3 and Eq. 2 into Eq. 1 gives:

$$k_1 q_0 = p(\rho_1 h_0 + \rho_2 h_0^2 + \rho_3 h_0^3) \quad (5)$$

In the unsteady state, the displacement would be the sum of the steady state component  $q_0$  and an unsteady component  $u(t)$

$$q(t) = q_0 + u(t) \quad (6)$$

Substituting Eq. 3, Eq. 5, Eq. 6 into Eq. 1 gives:

$$\begin{cases} m_1 \frac{d^2 u}{dt^2} + c_1 \frac{du}{dt} + k_1 u = p(\alpha_1 \Delta \tilde{h} + \alpha_2 \Delta \tilde{h}^2 + \alpha_3 \Delta \tilde{h}^3) \\ m_2 \frac{d^2 y}{dt^2} = 0 \quad \forall |q - y| < \Delta \end{cases} \quad (7)$$

where  $\Delta \tilde{h} = u(t - \tau)$ ,  $\alpha_1 = 2\rho_1 h_0 + 3\rho_3 h_0^2 + \rho_1$ ,  $\alpha_2 = 3\rho_3 h_0 + \rho_2$ ,  $\alpha_3 = \rho_3$ . Changes of variables are introduced as follow:

$$\begin{aligned} T &= \omega_1 t, \quad \omega_1^2 = \frac{k_1}{m_1}, \quad 2\mu_1 = \frac{c_1}{m_1 \omega_1}, \quad \varepsilon^2 = \frac{m_2}{m_1} \\ \psi &= \frac{p\alpha_1}{m_1 \omega_1^2}, \quad \eta_1 = \frac{p\alpha_2}{\psi m_1 \omega_1^2}, \quad \eta_2 = \frac{p\alpha_3}{\psi m_1 \omega_1^2} \end{aligned} \quad (8)$$

Substituting Eq. (8) into Eq. (7), the compact form of equations of motion is as follow:

$$\begin{cases} \ddot{u} + 2\mu_1 \dot{u} + u = \psi(\Delta \tilde{h} + \eta_1 \Delta \tilde{h}^2 + \eta_2 \Delta \tilde{h}^3) \\ \varepsilon^2 \ddot{y} = 0, \quad \forall |ru - y| < \Delta \end{cases} \quad (9)$$

where  $\varepsilon^2 \ll 1$  represents the mass ratio between the VI-NES and the modal mass of the first bending mode of the tool,  $r$  is the influence coefficient, depending on the position of the VI-NES on the tool and the dots denotes differentiation with respect to the non-dimensional time  $T$ .

A detuning parameter  $\sigma$  representing the nearness of  $\psi$  to the critical value  $\psi_c$  is introduced as  $\psi = \psi_c + \varepsilon^2 \sigma$ .

When  $|ru-y| = \Delta$ , a collision occurs. The state of the system after impact is obtained using the simplified shock theory and the condition of total momentum conservation:

$$\begin{aligned} u_+ &= u_-, & y_+ &= y_- \\ r\dot{u}_+ - \dot{y}_+ &= R(r\dot{u}_- - \dot{y}_-) \\ \dot{u}_+ + \varepsilon^2 \dot{y}_+ &= \dot{u}_- + \varepsilon^2 \dot{y}_- \end{aligned} \tag{10}$$

where  $R$  is the restitution coefficient of impact and the superscripts + and - denotes time immediately after and before impact.

### 3- Numerical simulation

In this section, different scenarios of passive control of chatter are presented. The equations of motion Eq. 9 were numerically integrated using Matlab dde23 algorithm [ST1] together with the event function to detect impacts. The dynamic parameters of the system used for the numerical simulations are:

$$\begin{aligned} \mu_1 &= 0.03; & \varepsilon^2 &= 0.01; & r &= 0.8 \\ R &= 0.65; & \Delta &= 2 \times 10^{-5} \end{aligned} \tag{11}$$

#### 3.1 – Stabilization of chatter

For  $\sigma=0.15$ , various numerical simulations were performed. In this case, the amplitude growth of the main system is limited by the random impact of the ball on the walls. An example of a numerical integration is shown in Fig. 2.

It is observed that the oscillation amplitude of the main system increases, which reflects the fact that the system is unstable, but the growth of these is hampered by the chaotic impacts of the free mass of the VI-NES on the walls. Furthermore, it is observed that the system has other sub-harmonic attractors with type responses  $z=2/3$  according to the classification of Peterka [P1]. This phenomenon is not detailed here.

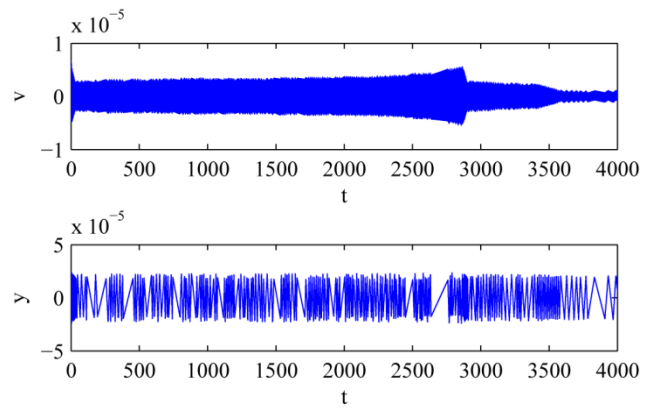


Figure 2: Numerical integration for  $\sigma=0.15$ , stabilization of chatter.

#### 3.2 – Passive control of chatter

If we increase the depth of cut to  $\sigma=1$ , the system is unstable and the amplitude of the oscillations of the main system increases sufficiently, to activate the movement of the ball. In this case, the VI-NES is synchronized with the main system, the energy in the system is dissipated by the successive impacts and the oscillation amplitude decreases to a lower level. The numerical integration shown in Fig. 3 illustrates this behavior.

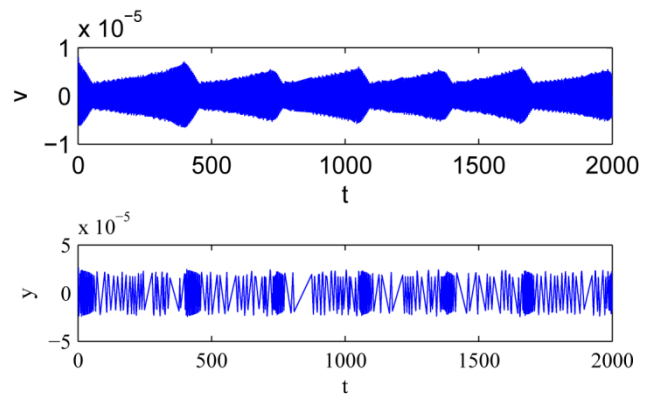


Figure 3: Numerical integration of motion equation for  $\sigma=1$ , passive control of chatter.

#### 3.3 – Limit of passive control of chatter

When the value of  $\sigma$  further increases, the system is very unstable and the amplitude is very important. During cycles of relaxation the system is no longer controlled. The result of the numerical integration in this case is shown in Fig.4, for  $\sigma=4.16$ . It confirm predicted scenario.

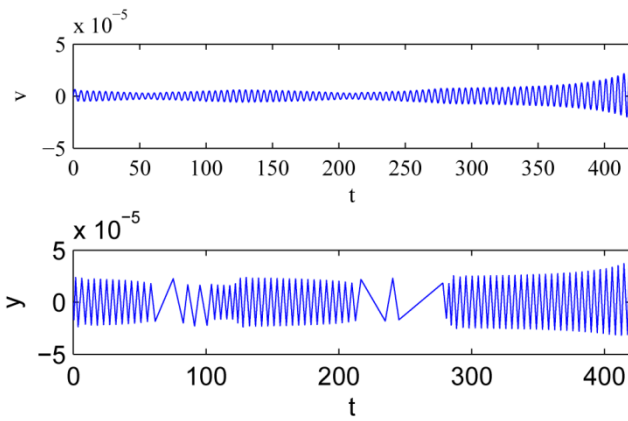


Figure 4: Numerical integration of motion equation for  $\sigma=4.16$ , chatter not controlled.

4- Experimental tests

In this section, the experimental setup developed is presented. After the first test for stability identification are presented.

3.1 – Experimental setup

The cutting tests are performed on a lathe Cazeneuve three axes (CT210). The experimental setup is show in Fig. 5. It's composed by steel bar 40mm diameter in XC38. The part is considered as infinitely rigid compared to the tool. Cutting tool, modeled as a single degree of freedom, is a boring bar to insert, with a length of 250mm. It has been deliberately modified by embedding (in the tool holder), in order to activate the bending mode in the feed direction. In addition, a mass was added on the tool to reduce the natural frequency of this mode. The operation of straight turning is conducted with a feed per revolution set to  $h_0 = 0.1$  mm/rev.



Figure 5: Experimental setup.

The VI-NES is mounted on the mass tool tip. Machining operations consist of turning operations with different cutting parameters (speed and depth of cut).

The vibrations of the tool are measured using a laser velocimeter. The cutting coefficients are derived from the literature [Do]. The dynamic characteristics of the tool are obtained by acter. The data are summarized in Table 1.

Table 1 : Dynamic parameter of the experimental setup.

$m_1$	3.1 kg	$\mu_1$	3 %
$f_1$	95 Hz	$m_2$	0.032 kg
R	0.8		

3.2 – Stability lobe without VI NES

Preliminary tests without the ball of the VI-NES have been made in order to find the behavior of the dynamical system. Determining stability or instability of a test can be difficult. In fact, the tool is subjected to forced vibration, as they may be due to a lack of concentricity for example. The direct or indirect measurement of the vibration does not allow itself to detect the presence of regenerative vibrations. However, there are simple methods in the time domain to detect. A first step is to resample the measured frequency of rotation of the workpiece signal [MI1]. In this case, a stable machining is in the form of a line, since the relative position between the tool part is always the same. On the other side, unstable machining present a mess. A second possibility is to plot the Poincare section of the measured signal. Stable machining then present an attractor point type, then an unstable machining present an attractor type circle [MI1]. Finally, the comparison between the stability lobes obtained theoretically and those obtained experimentally is shown in Fig. 6.

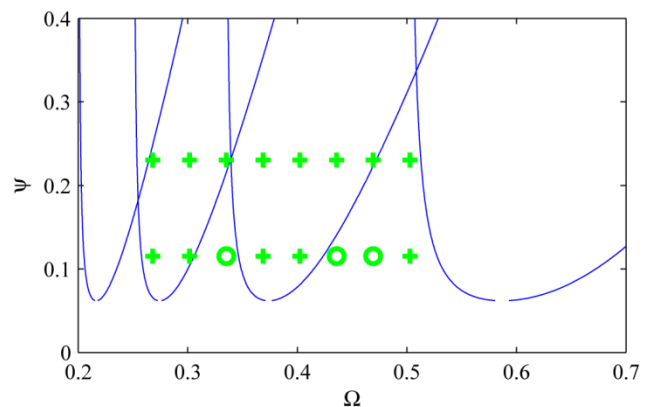


Figure 6: Experimental stability lobe Circles and crosses correspond respectively to stable and unstable machining.

## 5- Conclusion

The purpose of this paper is to analyze the dynamic response regimes of the machine tool strongly coupled to VI-NES in turning theoretically and numerically and to acquire the clues for further research in the future. Three different response regimes, that is, stable oscillation and complete suppression of regeneration chatter, relaxation oscillation and partial suppression, and unstable oscillation are identified through numerical simulation. The experimental setup is established for tests of different conditions and goals. A preliminary test to get stability lobe is accomplished and the result between theory and experiment is consistent.

## 6- References

- [AB1] Y. Altintas, E. Budak, Analytical prediction of stability lobes in milling, *Ann CIRP* 44, (1995), p. 357–362
- [AU1] F.M. Als and A.G. Ulsoy: Analysis of a system of linear delay differential equations, *ASME J. Dyn. Syst., Meas., Control* Vol. 125, (2003), p. 215–223
- [DB1] Z. Dombovari, D.A.W. Barton, R.E. Wilson, and G. Stepan: On the global dynamics of chatter in the orthogonal cutting model. *Int. J. Non-Linear Mech.* Vol. 46, (2011), p. 330–338
- [G1] O.V. Gendelman: Analytic treatment of a system with a vibroimpact nonlinear energy sink. *J. Sound Vib.* Vol. 331, (2012), p. 4599–4608
- [GG1] O.V. Gendelman, E. Gourdon, and C.H. Lamarque: Quasiperiodic energy pumping in coupled oscillators under periodic forcing. *J. Sound Vib.* Vol. 294, (2006), p. 651–662
- [GM1] E. Gourc, G. Michon, S. Seguy and A. Berlioz: Theoretical and experimental study of an harmonically forced vibro-impact nonlinear energy sink. *Proceedings of the ASME-IDEETC, Oregon, USA, 2013*
- [GM2] E. Gourc, G. Michon, S. Seguy, A. Berlioz: Experimental investigation and design optimization of targeted energy transfer under periodic forcing, *J. Vib. Acoust.* Vol. 136, (2014), p. 021021
- [GS1] E. Gourc, S. Seguy, and L. Arnaud: Chatter milling modeling of active magnetic bearing spindle in high-speed domain. *Int. J. Mach. Tools Manuf.* Vol. 51, (2011), p. 928–936
- [GS2] E. Gourc, S. Seguy, G. Michon, and A. Berlioz: Chatter control in turning process with a nonlinear energy sink, *Adv. Mater. Res.* Vol. 698, (2013), p. 89–98
- [HD1] A. Harms, B. Denkena, and N. Lhermet: Tool adaptor for active vibration control in turning operations. In *9th International Conference on New Actuators, Brême, Germany, 2004*
- [LZ1] A. Luongo, and D. Zulli: Dynamic analysis of externally excited NES-controlled systems via a mixed Multiple Scale/Harmonic Balance algorithm, *Nonlinear Dyn.* Vol. 70, (2012), p. 2049–2061
- [MB1] H. Moradi, F. Bakhtiari-Nejad, and M.R. Movahhedy: Tuneable vibration absorber design to suppress vibrations: an application in boring manufacturing process. *J. Sound Vib.* Vol. 318, (2008), p. 93–108
- [MI1] B.P. Mann, T. Insperger, G. Stepan, and P.V. Bayly: Stability of up-milling and down-milling, part 2: experimental verification. *Int. J. Mach. Tools Manuf.* Vol. 43, (2003), p. 35–40
- [ML1] M. Mousseigne, Y. Landon, S. Seguy, G. Dessein and J.M. Redonnet: Predicting the dynamic behaviour of torus milling tools when climb milling using the stability lobes theory, *Int. J. Mach. Tools Manuf.* Vol. 65, (2013), p. 47–57
- [NN1] A.H. Nayfeh and N.A. Nayfeh: Analysis of the cutting tool on a lathe. *Nonlinear Dyn.* Vol. 63, (2010), p. 395–416
- [NS1] A. Nankali, H. Surampalli, Y.S. Lee, and T. Kalmar-Nagy: Suppression of machine tool chatter using non-linear energy sink. *Proceedings of ASME-IDEETC, Washington DC, USA, 2011*
- [OH1] J. Ormondroyd, D. Hartog: The theory of the dynamic vibration absorber. *J. Appl. Mech.* Vol. 49–50, (1928), p. A9–A22.
- [P1] F. Peterka; Bifurcations and transition phenomena in an impact oscillator. *Ch. Sol. Fract.* Vol. 7, (1996), p. 1635–1647
- [S1] N.D. Sims: Vibration absorbers for chatter suppression: a new analytical tuning methodology. *J. Sound Vib.* Vol. 301, (2007), p. 592–607
- [SI1] S. Seguy, T. Insperger, L. Arnaud, G. Dessein, and G. Peigne: Suppression of period doubling chatter in high-speed milling by spindle speed variation. *Mach. Sci. Technol.* Vol. 15, (2011), p. 153–171
- [ST1] L.F. Shampine, S. Thompson: Solving ddes in matlab. *Appl. Num. Math.* Vol. 37, (2001), p. 441–458
- [ST2] H. Shi, and S. Tobias: Theory of finite amplitude machine tool instability, *Int. J. Mach. Tools Manuf.* Vol. 24, (1984), p. 45–69
- [TF2] S.A. Tobias and W. Fishwick: Theory of regenerative machine tool chatter. *Eng.* Vol. 205, (1958), p. 199–203
- [VR1] A.F. Vakakis and R.H. Rand: Non-linear dynamics of a system of coupled oscillators with essential stiffness nonlinearities. *Int. J. Non-linear Mech.* Vol. 39, (2004), p. 1079–1091
- [W1] M. Wang: Feasibility study of nonlinear tuned mass damper for machining chatter suppression. *J. Sound Vib.* Vol. 330, (2011), p. 1917–1930