A nonlinear tuned vibration absorber for chatter mitigation

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Summary. A passive vibration absorber, termed the nonlinear tuned vibration absorber (NLTVA), is designed for the suppression of chatter vibrations. Unlike most passive vibration absorbers proposed in the literature, the NLTVA comprises both a linear and a nonlinear restoring force. Its linear characteristics are tuned in order to optimize the stability properties of the machining operation, while its nonlinear properties are chosen in order to control the bifurcation behavior of the system and guarantee robustness of stable operation. In this study, the NLTVA is applied to turning machining.

Introduction

Machine tool vibrations represent a main concern in industry, as they result, for instance, in wavy machined surfaces and decrease in tool and spindle life [1]. The so-called regenerative effect (where the tool interacts with its delayed displacement via the surface profile that is cut one revolution earlier) is considered to be one of the main reasons for these vibrations [1, 2]. Active controllers are a possible solution to this problem, but they have issues related to unpredictable instabilities and requirements of external energy source [3]. Several authors developed various kinds of passive vibration absorbers for the mitigation of such vibrations [4, 5]. Numerical and experimental studies exhibited their promising performance in terms of stabilization of machining operations, allowing larger depth of cut in stable conditions, which enables to increase production speed. However, the intrinsic nonlinearity of machine tool vibrations, combined with the regenerative time delay effect, generates robust oscillatory motions, which exist also within the stable region of machining operation, causing dangerous bi-stabilities [2]. These motions, generally related to subcritical bifurcations at the loss of stability, are overlooked by classical linear stability analysis, which makes them hardly predictable.
In this study, we propose a nonlinear passive vibration absorber, termed the nonlinear tuned vibration absorber (NLTVA), for the suppression of machine tool vibrations. It combines the beneficial effect of a linear absorber, able to enlarge the stable area of operation, with a nonlinear characteristic, which enables it to improve the robustness of the stable chatter-free motion and to avoid bi-stable conditions.
The optimal tuning of the NLTVA follows a classical stability analysis for the definition of the most convenient linear parameters, whereas its nonlinear characteristic is optimized through bifurcation analysis at the loss of stability. Doing so, the nonlinear restoring force of the absorber is designed such that the system loses stability through supercritical bifurcations, rather than subcritical ones. The NLTVA is tested numerically using a simplified model of turning operations.

Mechanical model and stability analysis

We consider a two degree-of-freedom (DoF) system, where the first coordinate refers to the motion of the cutting tool in the cutting direction, while the second coordinate refers to the movement of the attached NLTVA. The dimensionless equations of motion read

$$\begin{aligned}\ddot{x}_1 + 2\zeta_1\dot{x}_1 + x_1 + 2\zeta_2\gamma\mu\left(\dot{x}_1-\dot{x}_2\right) + \gamma^2\mu\left(x_1-x_2\right) + \alpha_3\gamma^2\mu\left(x_1-x_2\right)^3 \\ = p\left(\left(x_{1\tau}-x_1\right) + \eta_2\left(x_{1\tau}-x_1\right)^2 + \eta_3\left(x_{1\tau}-x_1\right)^3\right) \\ \ddot{x}_2 + 2\zeta_2\gamma\left(\dot{x}_2-\dot{x}_1\right) + \gamma^2\left(x_2-x_1\right) + \alpha_3\gamma^2\left(x_2-x_1\right)^3 = 0,\end{aligned} \tag{1}$$

where $x_{1\tau} = x_1(t-\tau)$ and τ is the time delay, ζ_1 and ζ_2 are, respectively, the relative damping factors of the primary system and of the absorber, γ and μ are, respectively, the natural frequency and mass ratio between the absorber and the primary system, p is the dimensionless depth of cut, η_2 and η_3 define the nonlinear part of the cutting force [2], α_3 is the coefficient of the cubic terms of the absorber restoring force.
Figure 1 shows the first lobe of the stability chart in the dimensionless space Ω_d, p (Ω_d is the dimensionless spindle speed) for a system without and with LTVA (Figs. 1a and 1b, respectively) and clearly illustrates the advantage given by the absorber in terms of stability (the solid lines in Fig. 1 refer to the stability limit). Equations for obtaining an almost optimal linear tuning of the absorber are given in [4].

Bifurcation analysis and nonlinear tuning of the absorber

Systems subject to time delay are generally prone to lose stability through subcritical bifurcations. This causes the existence of solutions, different from the trivial one, within the stable region, often generating bistability. In other words, while machining in chatter free conditions, it is possible that perturbations cause the system to jump towards another attractor which involves large chatter oscillations.

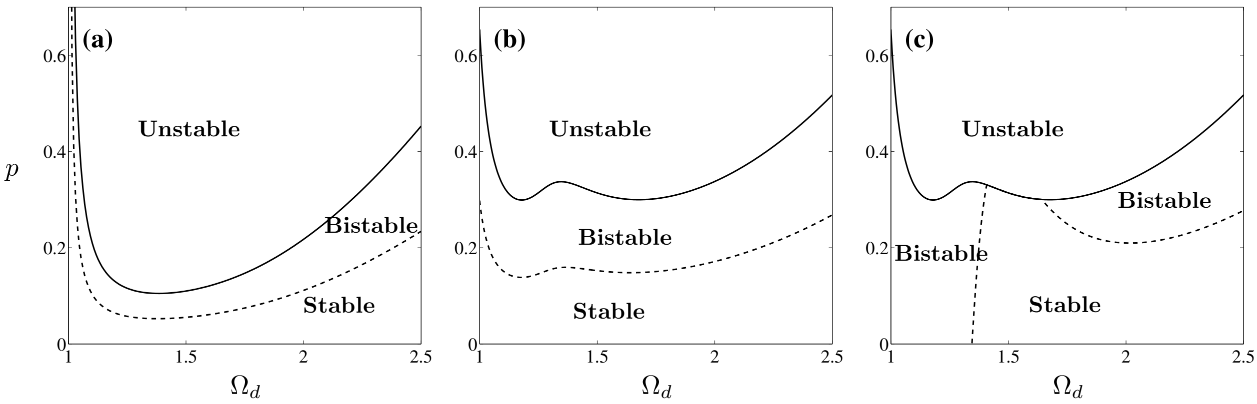

Figure 1: Illustration of stable, bistable and unstable regions (for a single stability lobe) for a system without absorber (a), with an attached LTVA (b) and with an attached NLTVA (c). Parameter values: $\zeta_1 = 0.05$, $\mu = 0.05$, $\gamma = 1.069$, $\zeta_2 = 0.1437$ and $\alpha_3 = 0.1$. Cutting force model according to [7] with nominal depth of cut $h = 0.07$ mm.

Adopting the multiple scales method [6], we investigate the nature of the bifurcations occurring at the loss of stability of the system under study. A lengthy analytical procedure provides the compact formula

$$x_1 \approx 2\sqrt{-\frac{p - p_{cr}}{\Delta}}e^{i\omega t} \tag{2}$$

which describes the amplitude of the periodic oscillations generated by the Hopf bifurcation at the loss of stability. p_{cr} indicates the critical depth of cut, while Δ characterizes the bifurcation. Namely, if $\Delta < 0$ the bifurcation is supercritical and if $\Delta > 0$ it is subcritical.

The results of the bifurcation analysis showed that, if an LTVA is adopted, Δ is always positive, therefore the bifurcations at the stability loss are always subcritical. On the contrary, if an NLTVA is implemented, through an accurate tuning of the absorber nonlinearity, it is possible to enforce a transition of the bifurcations from subcritical to supercritical, with a consequent elimination of the bistable regions. In this respect, the analytical formulation adopted is particularly convenient, since Δ is linearly proportional to α_3, which facilitate the tuning.

Interestingly, depending on the spindle speed Ω_d of operation, either a softening or a hardening nonlinearity is required to enforce supercriticality. This precludes the possibility of designing an absorber able to guarantee always supercritical behavior. Instead, it suggests that either the spindle speed of operation should be known and fixed (within a limited range), or the absorber nonlinearity should be adjustable with respect to the spindle speed. A comparison of the safe and unsafe regions for a system without absorber, with an LTVA or with a hardening NLTVA ($\alpha_3 = 0.1$) is depicted in Fig. 1. The extent of the bistable regions is estimated following the procedure outlined in [2]. Fig. 1 illustrates the advantage of the NLTVA over the LTVA for the spindle speed range $\Omega_d \in [1.4,\ 2]$ in terms of robustness of stable operation (elimination of bistable region). However, for $\Omega_d \in [1,\ 1.4]$ the bistable region is increased, compromising the usability of the absorber for this spindle speed range.

We note that this study is a first step, involving only local dynamics, towards the full understanding of the potentiality of the NLTVA for chatter mitigation. A future work will address the global behavior of the system, which might have an important role in a more accurate definition of the bistable region and in estimating the robustness of the system.

Acknowledgements

This work was financially supported by the European Union, H2020 Marie Skłodowska-Curie IF 704133 (G. Habib), ERC Starting Grant NoVib 307265 (G. Kerschen) and ERC Advanced Grant SIREN 340889 (G. Stepan).

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