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Numeric and experimental study on several rattle noises from a generic system

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Rattle noises encountered in automotive systems such as seats or front panels are gaining more and more importance in the automotive industry. Recent progresses in the field of noise reduction in automotive cabins, combined with the emergence of new electrical/hybrid vehicles and the need for mass reduction, has emphasized the appearance of this new class of noise, which was masked in the past. Rattle noises are studied here using a continuous vibro-impacting system under forced excitation. An experimental set-up composed of a beam impacting barrier has been designed in order to reproduce this phenomenon. Periodic and apparent chaotic rattles are occurring at various combinations of environmental parameters such as excitation frequency, initial presence of clearance or pre-load between the impacting system and its associated barrier, and contact stiffness and damping. Simulated bifurcation diagrams are obtained representing the overall behaviour of the vibro-impacting system and are then compared with experiments.

1 Introduction

Noises that can be found in an automotive cockpit are generally induced by the following three class of noise sources:

- Aero-dynamical noise
- Road/tyre contact noise
- Engine noise

Due to the subsequent progress performed in the field of noise reduction focusing on these three above classes, a fourth category began to appear. This fourth class, commonly named ”squeak and rattle” noise is gaining increasing importance in today’s automotive market because of its irritating aspect and because it leads to a degradation of the overall build quality and durability perceived by the final user. Rattle noises [1, 2], caused by loose elements potentially impacting others, are generally modelled as normal contacts. Moreover, squeak noises caused by elements in friction are modelled as tangential contact[7]. Both of these two noises mostly appear under dynamic solicitation (e.g. forced road surface excitation) and are characterized by their random behaviour. They are caused by unwilling contact or free play that are inherent to complex assembly. This paper is specifically focusing on rattle noise.

Such phenomena are based on vibro-impact dynamics which were involved in many practical examples, such as heat exchanger in nuclear reactors [5], gear-pair systems with backlash [3] or a ship impacting a harbour wall [4]. Although impacting system is very common, the study of even one degree of freedom system is still wide spread, mainly due to the variety and the complexity of the dynamical behaviour of such systems. Impact oscillators include periodic and chaotic motions depending on initial conditions and system parameters. For example, Shaw and Holmes [6] examined a single-degree-of-freedom system comprising a concentrated mass that can impact a spring and that is otherwise free between collisions. This piecewise linear system response exhibited both periodic and chaotic response regimes.

This paper firstly describe an experimental approach which reproduces rattle found in automotive assembly, then a first model is introduced to validate experimental measurement. Experimental and numerical results are finally compared and discussed.

2 Experimental approach

2.1 Experimental set-up

The first objective of the experimental set-up proposed here is to reproduce as close as possible rattle noises encountered in automotive cabin. To do so, and as shown in figure 1, a beam is clamped on its two sides on a rigid support. The rigid support is submitted to an excitation through its base.

Measurement of the beam velocity is performed using a laser vibrometer. Vibrations of both rigid support and beam are measured using accelerometers. When submitted to a base excitation, the beam impacts a tip which is mounted with a piezo-electrical force sensor allowing a direct measurement of the impact force. An adjustment of the initial

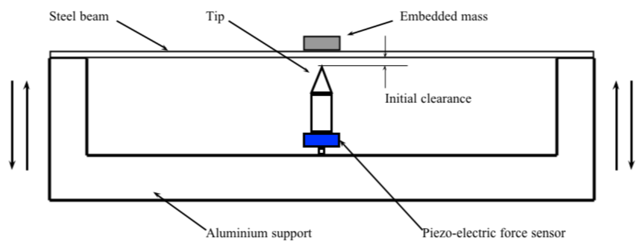


Figure 1: Experimental set-up

clearance (or pre-load) between the tip and the beam is done before performing the measurements. Overall behaviour of the experimental apparatus is observed using a bifurcation diagram. A bifurcation diagram is obtained by plotting a state parameter of the system according to an associated control parameter. One benefit of this method is to gather the whole dynamical behaviours exhibited by the set-up. In order to obtain this diagram, the acquisition chain is automated, the frequency of excitation ω is varied step by step and velocity before each impact is taken as a state parameter.

2.2 Measurement results

The bifurcation diagram shown in figure 2 is obtained by varying the frequency of excitation from $f = 20Hz$ to $f = 35Hz$ with a step of $0,1Hz$. The first resonance frequency of the beam is measured at $f_0 = 31.2Hz$. Due to the brief aspect of the phenomena observed, the sampling frequency is set at $102,4kHz$. An initial clearance between the tip and the beam is adjusted at $0.2mm$ and the velocity of each impact is measured.

This diagram can be firstly separated following two principal areas : a silent area with no rattle and a second area containing all rattle occurrences.

2.2.1 Chaotic motion

Several chaotic motions are shown on the bifurcation diagram. These are underlined using the dashed rectangle (1),

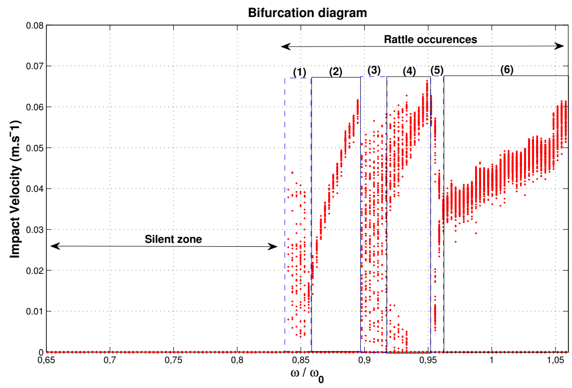
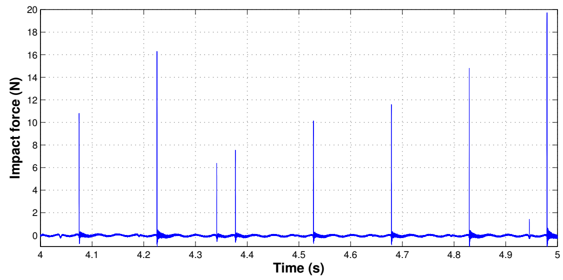
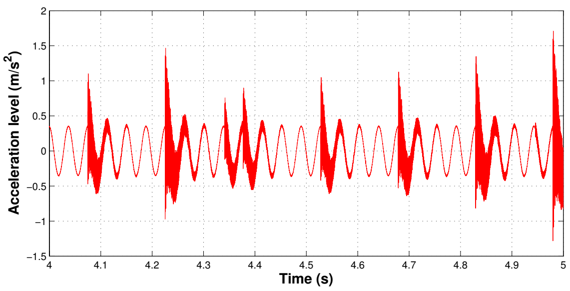


Figure 2: Experimental bifurcation diagram with presence of initial clearance

(3), (5). The impacts are randomly time spaced and of different force amplitude. A time domain acquisition is given in figure 3 and corresponds to the first area quoted (1) where erratic behaviour occurs.



(a)



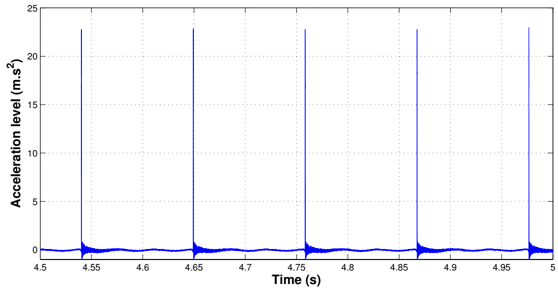
(b)

Figure 3: Apparent chaotic behaviour: (a) impact amplitude; (b) rigid support acceleration

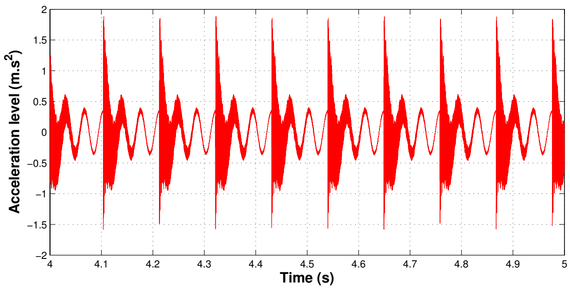
2.2.2 Periodic motion

In order to describe qualitatively the periodic orbits which are to be discussed, a notation is here introduced [8]. The periodic scheme of an impact oscillator must repeat after m impacts, where m is an integer greater or equal to 1. Furthermore, this periodic scheme is necessarily linked to the forcing function and will have a period quoted nT, where $T = 2\pi/\omega$ is the period of the forcing and n is another integer greater or equal to 1. The following periodic motion discussed below will be described using this pair of integers as (m, n) orbit.

On the given bifurcation diagram, three different periodic orbits are exhibited by the system. The first periodic orbit appears for a pulsation $0.87 < \omega/\omega_0 < 0.9$ and is quoted on figure 2 as (2). Time domain shown in figure 4 allows a definition of the considered orbit: one impact occurs for three periods of forcing, causing this one to be defined as a $(1, 3)$ orbit.

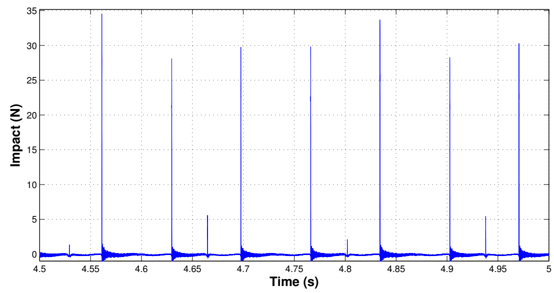


(a)

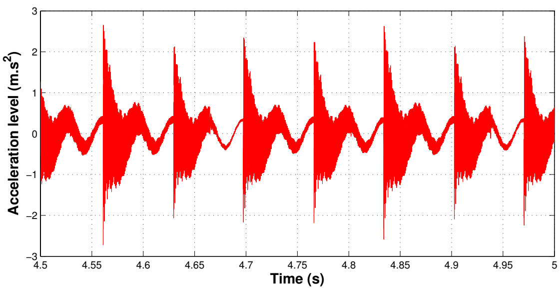


(b)

Figure 4: First periodic orbit (1,3): (a) impact amplitude; (b) rigid support acceleration



(a)



(b)

Figure 5: Second periodic orbit (6,8): (a) impact amplitude; (b) rigid support acceleration

Then, for a pulsation $0.92 < \omega/\omega_0 < 0.95$, another periodic sequence occurred (quoted (4) on the bifurcation diagram). As seen on the bifurcation diagram, the system branches out within two distinct branches : one with high velocity impact and one with low velocity impact. Corresponding

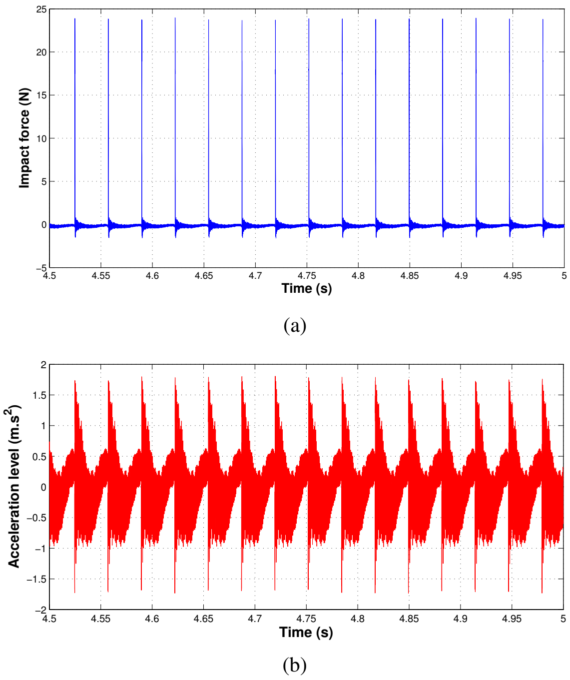


Figure 6: Third periodic orbit (1,1): (a) impact amplitude; (b) rigid support acceleration

time domain acquisition is shown in figure 5 and allows a more precise description of these two branches. The periodic scheme is actually composed of five different impacts : two of low amplitude force (between $2.5N$ and $5N$), and four of higher amplitude force (between $30N$ and $35N$). In fact, the bifurcation diagram should have shown six different branches instead of only two. This difference is mainly due to the fact that the difference of amplitude of the considered impact is very low, so that the branches are too close to each others to be seen. This orbit is a (6, 8) one, because six different impacts occur during eight periods of excitation pulsation.

The last one, quoted as (6) on figure 2, is a $(1, 1)$ orbit. The periodical scheme occurs at each period of the forcing as seen on figure 6 and is around the first resonance frequency of the system.

3 Numerical simulation

3.1 Model

In order to reproduce rattle phenomena, we study a simplified model of the experimental apparatus using a one degree of freedom oscillator. This oscillator of mass m shown in figure 7 is composed of a linear spring of stiffness k and is associated to a viscous damping c. The basis of this oscillator is submitted to periodic forcing $y(t) = Acos(\omega t)$, with A the displacement amplitude of the base.

The origin $x = 0$ is the equilibrium position of the oscillator and a rigid barrier is placed according to an initial clearance σ or pre-load. The presence of the rigid obstacle that will be impacted by the oscillator during its motion allows this kind of system to be named vibro-impact oscillator.

The process of impact is here modelled using an instantaneous rule based on a restitution coefficient [10, 9]. When the vibro-impact oscillator has an initial presence of clearance, the system is governed by:

$$m\ddot{x}(t) + c\left[\dot{x}(t) - \dot{y}(t)\right] + k\left[x(t) - y(t)\right] = 0, \qquad (1)$$

when $x(t) < \sigma$ and with:

$$\dot{x}(t+) = -r\dot{x}(t-) \qquad (2)$$

when $x(t) = \sigma$. Where $\dot{x}(t+)$ and $\dot{x}(t-)$ are the impact velocities respectively just after and before impact occurs.

Because practical measurement of the initial clearance on the experimental apparatus is assumed to be submitted to an important uncertainty, the initial clearance is adjusted using displacement transmissibility. On the experimental bifurcation diagram given in figure 2, the rattles almost begin at $\omega/\omega_0 = 0,84$. In order to have the same behaviour in the simulation, the clearance is defined and adjusted at $\sigma = 0.044mm$.

The modal parameters m, c and k, corresponding to the first mode were experimentally measured and then used in the numerical model. The parameters of this oscillator are $k = 1773,4N.m$; $c = 0.46N/m/s$; $m = 0.041kg$.

The calculations are then performed using a Runge-Kutta integration scheme of the fourth order [11].

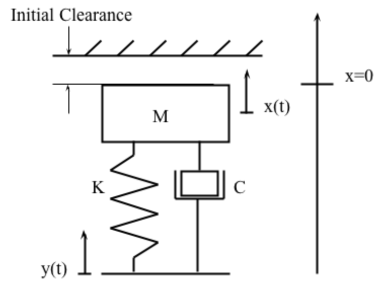


Figure 7: One degree of freedom impact oscillator

3.2 Simulation results

The simulated bifurcation diagram with its associated different motion zones is shown in figure 8. The state parameter which is observed is the beam velocity just before the impact and the pulsation of forcing is varied.

As in the experimental diagram, two principal areas are here separated: a silent zone from $\omega/\omega_0 = 0.65$ to $\omega/\omega_0 = 0.84$, and a zone containing all the rattle occurrences from $\omega/\omega_0 = 0.84$ to $\omega/\omega_0 = 1.05$.

Regarding the rattling area, the system also exhibits a number of different behaviours, mainly chaotic and periodic motions, which are to be discussed below.

3.3 Discussion

Several different frequency ranges of chaos are shown on the bifurcation diagram. The main ones are quoted (1) and (3) on figure 8. The first one occurs at around $\omega/\omega_0 = 0.85$.

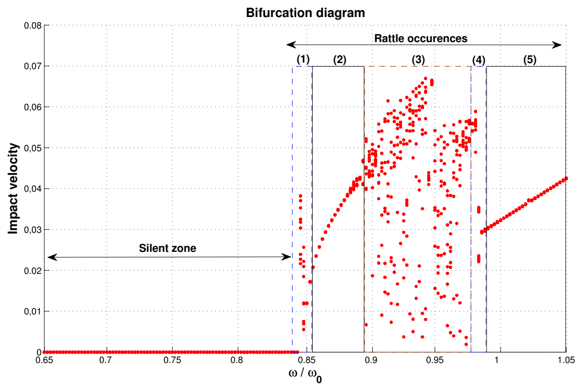


Figure 8: Numerical bifurcation diagram with a clearance of 0.045mm

The oscillator impact its rigid barrier with random impact velocity.

In frequency Range (2), a periodic branch is observed. This periodic orbit corresponds to forcing adimensionnal circular frequency $0.85 < \omega/\omega_0 < 0.88$. One contact occurs periodically for three period of forcing leading to a $(1, 3)$ orbit.

The area quoted as (3) on the bifurcation diagram is then more unclear. This zone contains only chaotic motion, and hence cannot be completely superposed to the corresponding experimental zones (quoted as (3) and (4) on figure 2).

This area is then followed by a brief new periodical orbit (quoted (4)) which is also found out on the experimental diagram (zone (5)). This brief periodical orbit then stabilizes to a new periodic regime around the resonance of the system. This orbit is a $(1, 1)$ one and corresponds to the experimental behaviour.

The whole experimental bifurcation diagram can be qualitatively simulated using a single degree of freedom vibro-impact oscillator. Although some differences are visible, the main zones are reproduced and the qualitative form of the periodic orbits (m, n) is well described by the simulation.

4 Conclusion

Rattle noises are frequently observed in automotive context. An academic experiment consisting in a beam impacting a rigid barrier has been experimentally studied in order to reproduce such noises. Measurements of impact force and beam velocity allow us to determine bifurcation diagrams which permits to define different frequency bands associated to silence or rattle noise, which can be periodic or random.

A numerical model is proposed using a single-degree-of-freedom vibro-impact oscillator which qualitatively reproduces the experiments: the existence of frequency bands associated to a given behaviour are predicted.

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