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NOISE RELATED TO MECHANICAL VIBRATION AND REPEATED IMPACTS IN A COMPUTER FAN, AN EXAMPLE OF A CHAOTIC DYNAMIC SYSTEM

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ABSTRACT

Computer fans are a pervasive source of contaminant noise in everyday environments. Axial vibration of the fan causes impacts between its moving parts and generates a disturbing rattling noise. The impacts introduce strong nonlinearities into an otherwise linear system, thus leading to the excitation of high frequency harmonics and the onset of a chaotic response. A simple mathematical model is proposed and validated against experimental measurements. The model is then employed to study the sensitivity of the fan system to changes in the excitation leading towards the development of chaos.

INTRODUCTION

In today's world, computers are an essential part of life. Human-computer interaction is present in almost every activity of modern life, from the child playing video games and the storekeeper's cash machine to the highly specialized researcher performing the most complex calculations. Therefore, ergonomy is a key factor in computer design. Glare free monitors, ergonomic keyboards, left-handed mice, wrist-pads and lumbar support seats are just a few examples of the attention put into providing a comfortable interaction with the computer.

Nevertheless, computers are known to be a notorious source of noise, a disturbing contaminant that generates stress and attacks the human nervous system thus reducing concentration and effectiveness in the workplace (Pelton, 1993). While most of the computer's electronic components are static and noiseless, its moving parts (disk drives and cooling fans) can generate annoying operating noises. A PC user, however, expects a disk drive to produce a signal indicating that it is working when requested and an occasional sound is certainly welcome. On the other hand, cooling fans operating in optimum conditions generate a continuous noise between 40 and 50 dBa (NMB Technologies Inc., 1995), an acceptable noise level in the working place (OSHA, 1969). Yet, many computers are returned every year because of faulty or too noisy cooling fans, thus increasing production costs and evidencing the need for quieter fans.

Besides the intrinsic noise related to the aerodynamics of the air flow through the cooling fan (Wen-Shiang, 1989), computer fans are prone to the generation of noise caused by vibration, rub or impact of their moving parts. The present paper addresses the mechanism that leads to the occurrence of the disturbing rattling noise that comes and goes randomly in many desktop computers. Examination of a series of faulty power supply fans led to the discovery that the rattling noise is related to axial vibrations of the fan hub (San Andrés and Diaz, 1997). Imbalance, aerodynamic, and magnetic forces are strong enough to cause the loss of contact between the fan moving parts. The subsequent impact at contact recovery generates the noise and turns the fan system into a nonlinear dynamic system that could even present the phenomenon of chaos.

SYSTEM DESCRIPTION

Figure 1 depicts a typical power supply cooling fan unit whose size is approximately 14 cm x 14 cm x 2 cm. The fan is comprised of a rotating plastic fan hub mounted on a thin steel shaft supported on two small anti-friction ball bearings. The bearings fit loosely within the plastic frame structure. The inner side of the hub contains a two pole permanent magnet and a four pole motor winding is attached to the stationary frame. The combination of a coil spring and a retainer clip allows a mechanical preload on the assembly of the fan hub and the ball bearings.

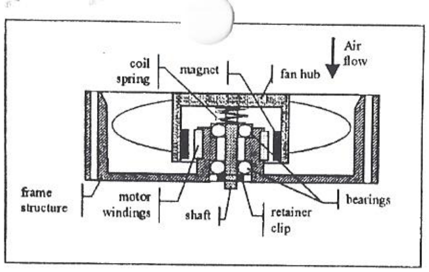

Figure 1 – PC cooling fan assembly

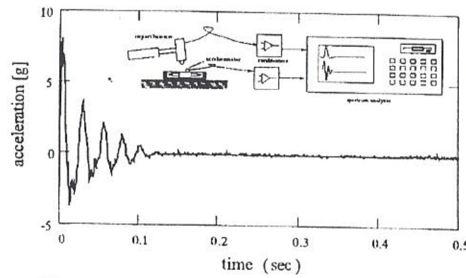

Figure 2 – Free axial vibration acceleration of fan hub

The stiffness of the coil spring measured statically equals 2,632 ± 90 N/m. With the retainer clip in place, the spring is preloaded to 0.7 mm. The fan hub (m_h) and structure frame (m_f) masses are 35.5 grams and 70.1 grams respectively. Measurements of the free axial vibration acceleration in a fan whose retainer clip has been removed suggests that the damping in the system is mainly due to dry friction (see Figure 2). An equivalent viscous damping ratio equal to 12.5 % of critical damping is estimated for use in the analysis. Spectral analysis of the free axial vibration confirms a fan hub natural frequency of 40 Hz versus the 43.3 Hz predicted from the mass and stiffness measurements.

An electromagnetic shaker is attached to the fan hub to study its forced axial vibration as shown in Figure 3. A function generator provides a harmonic excitation at a frequency identical to the fan operating speed in normal conditions (54 Hz). Two small (1 gram) accelerometers sense the vibration of the fan hub and frame. The RMS amplitude of the harmonic vibration of the hub (identical to that of the shaker head) is measured with a hand held voltmeter. The time trace of the frame acceleration is recorded digitally with a spectrum analyzer and stored for further analysis.

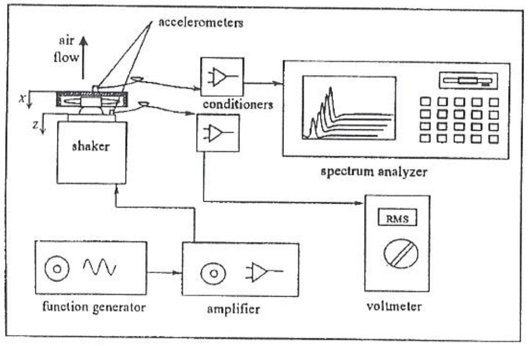

Figure 3 – Experimental set-up

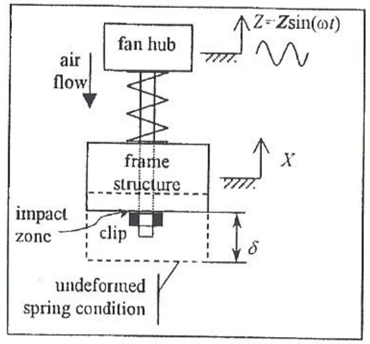

Figure 4 – Dynamic model of fan

MATHEMATICAL MODEL

Figure 4 represents the dynamic model for axial vibration of the cooling fan frame and hub system. The system behaves linearly when the retainer clip is not in contact with the ball bearing or when the contact is preserved at all times and both parts move as a single rigid body. When there is no contact, the frame motion is governed by the equation:

$$m_f\ddot{X} + C(\dot{X} - \dot{Z}) + K(X + \delta - Z) = 0 \qquad (1)$$

where C is the equivalent damping coefficient, δ is the spring preload equal to 0.7 mm, X is the absolute axial displacement of the frame structure, and Z=$Z\sin(\omega t)$ is the absolute displacement of the fan hub prescribed by the electromagnetic shaker. When an impact occurs the frame motion is no longer governed by equation (1) and the following kinematic relation establishes the frame velocity after the impact:

$$(\dot{Z}_a - \dot{X}_a) = -e(\dot{Z}_b - \dot{X}_b) \qquad (2)$$

where the subindexes b and a denote the conditions before and after the impact , respectively. The restitution coefficient (e) is estimated as 0.6 for metal to metal contacts (Meriam and Kraige, 1992).

Contact occurs when the values of Z and X are equal, although this does not necessarily imply an im[illegible]kes place. Therefore the introduction of a second condition assuring an approaching speed is required, i.e.:

$$\begin{cases} Z = X \\ \dot{Z} > \dot{X} \end{cases} \qquad (3)$$

The solution of equation (1) is computed using a fourth order Runge-Kutta integrator. The discrete nature of the solution forces the conditions of equation (3) to be defined in terms of finite tolerances. Therefore, in the integration algorithm both conditions in equation (3) are evaluated with a greater-than-or-equal-to sign (Diaz, 1993). The Runge-Kutta method is halted when the result of an integration step fulfills the impact conditions, the speed ($\dot{X}_a$) is recomputed with equation (2) and the position (X) is set equal to (Z).

RESULTS

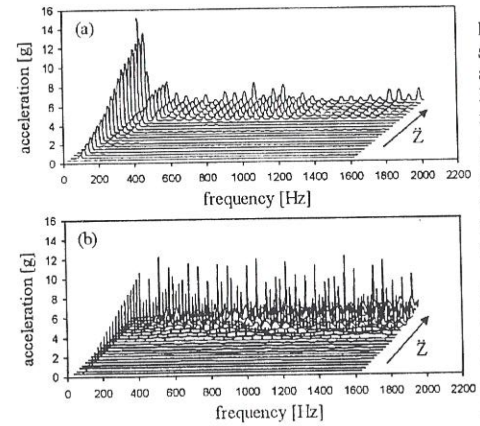

Figure 5 – Frequency spectra of fan frame acceleration for increasing fan hub excitation amplitude. (a) experimental (b) predicted

Figure 5 shows measured and predicted waterfall plots of the spectra of the frame acceleration response when the excitation acceleration of the hub ($\ddot{Z}$) increases linearly from 0 to 3 g RMS at intervals of 0.15 g. The similitude between the two results indicates that the numerical model represents well the physical system. In both cases, the response is purely harmonic for hub excitations up to about 1.8 g. Below this amplitude of excitation the spring preload is enough to keep the bearing and the clip in contact at all times. In this case, the base frame and the fan hub move together as shown by the linear increase of the synchronic amplitude at 54 Hz.

Separation (loss of contact) occurs when the inertial frame force overcomes the spring preload. Hence, an estimate of the minimum excitation that will result in loss of contact can be computed by equating the preload to the peak acceleration multiplied by the mass ($K\delta = m_f Z\omega^2$). This simple calculation predicts separation to occur for excitations above 1.87 g RMS, which is reasonably close to the value obtained experimentally.

For fan hub excitations above 1.8 g, separation occurs and the subsequent impacts excite higher harmonic multiples of the excitation frequency. The experimental observations also reveal the presence and persistence of a rattling noise similar to the one reported in faulty fans. This confirms the relation of the noise to the axial vibration and to the impacts between the retainer clip and the ball bearing. The fan system is piecewise linear in the sense that except for the instantaneous changes in velocity when an impact occurs, the motion is defined by a linear equation. However, the broad band of frequency responses generated by the single frequency excitation manifests the strong non-linearity that the impacts introduce.

Figure 6 shows displacement and velocity bifurcation diagrams of the fan system where the control parameter is the excitation amplitude expressed in terms of the hub RMS acceleration. It also presents several phase plots corresponding to different values of the hub acceleration. All system responses are computed from null initial conditions and plotted from the 12th to the 32nd periods after the transient has disappeared[1]. Up to about 1.8 g, the frame structure phase plots are smooth ellipses that follow exactly the prescribed motion of the hub. The responses are period one and the bifurcation plots follow straight lines. Between 1.8 and 2.35 g approximately, impacts only occur while the fan hub is moving downwards (negative speed) and contact is maintained while the hub moves up. Therefore, no trend change is appreciable in the bifurcation plots. As the excitation amplitude approaches 2.4 g, impacts start to appear during both upward and downward motions of the fan hub. As this happens and the excitation magnitude keeps increasing, chaos develops through a period doubling route (Nayfeh and Balachandran, 1995). Figure 6 includes the phase plots of period-two, period-four and chaotic windows. Figure 7 depicts the shape of the chaotic attractor generated for an excitation of 2.841 g. Above 2.87 g of excitation, the motion becomes again period-one with impacts only in the upward motion of the fan hub and with no contact during the downward motion. Identical phase plots are obtained using 528 or 1024 integration steps per excitation cycle, which confirms that the chaotic response region is a physical phenomenon rather than a numerical one.

[1] In most periodic cases the transient part of the response completely disappears after just two periods of excitation due to the significant energy dissipation by the viscous damping and inelastic impacts. The experimental results also reveal the same features.

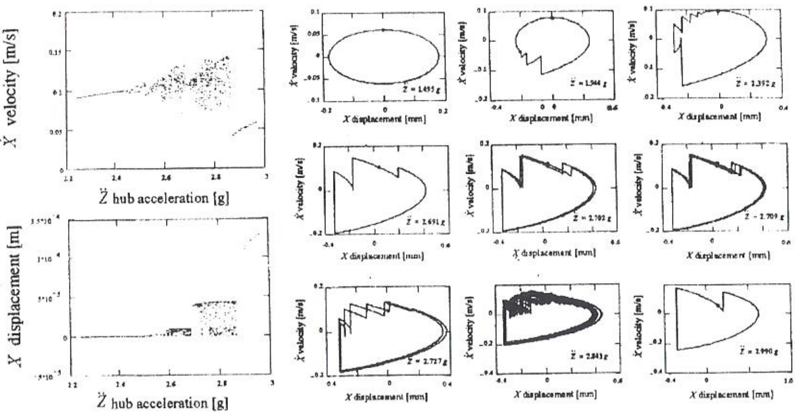

Figure 6 - Displacement and velocity bifurcation plots with their respective phase plots

CONCLUSION

The similarity between the experimental measurements and the numerical predictions of the fan system forced response confirms the validity of the assumptions made in the modeling. This reinforces the assertion of the axial vibration as the source of the reported rattling noise in the fans. The results show the hub-spring-frame system is strongly sensitive to excitation of axial vibration and able to demonstrate chaotic behavior within the fan operating range. Consideration of dry friction in the model instead of an equivalent viscous damping may result in an even stronger nonlinear behavior. Due to the discrete nature of the friction force, the fan hub could stick for some periods of time and be released in others thus explaining the random appearance and disappearance of the disturbing noise. A large enough increment of the spring preload will rise the onset of separation and could set the noisy regimes outside the fan normal range of operation. The use of softer materials in the impact zone between the retainer clip and the ball bearing would augment the dissipation of energy and result in softer impacts that might be not audible.

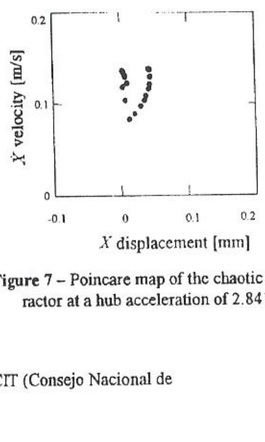

Figure 7 – Poincare map of the chaotic attractor at a hub acceleration of 2.841 g.

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