Chatter Vibration of a Cleaner Blade in Electrophotography*

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Chatter vibration of a cleaner blade charged by a contact charger roller was mathematically investigated. From the results of the investigation, the following points were deduced: (1) The chatter vibration is basically induced by nonlinear negative damping due to negative speed dependence of the friction coefficient between the cleaner blade and a photoreceptor drum. Parametric excitation and forced vibration are also generated by the photoreceptor vibration induced by the alternating electrostatic force of the charger roller. (2) Calculated results based on the present model qualitatively agreed with experimental observations, and several methods to suppress the chatter vibration were proposed.

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Introduction

In a cleaning subsystem of electrophotography, the photoconductor is cleaned of any untransferred toner by means of brushes or scraper blades. In low-end applications, a blade cleaner is widely used because it is simple, inexpensive, and compact.[1] However, several types of failures are associated with cleaner blades. One of the most serious problems is chatter vibration due to dry friction between the blade and the photoreceptor. Chatter vibration, sometimes called "stick-slip" vibration, has been extensively investigated in the field of mechanical engineering, because clarification of the mechanism and knowledge of measures that can be taken to suppress the vibration are extremely important for industrial applications, such as in frictional brakes and automotive wipers. The essence of the mechanism, according to former investigations, is that chatter vibration is a self-excited vibration caused by negative damping due to negative speed dependence of the friction coefficient in the low slip-speed zone. In addition to this typical chatter vibration, we have also recently observed a slightly different vibration in our low-speed laser printer, which uses a contact roller charging subsystem.[2] The vibration took place only when ac voltage was applied between the contact charger roller and the photoreceptor drum. The objective of this paper is to explain the mechanism of the vibration, and several methods are proposed to realize reliable laser printers and copiers that adopt the blade cleaner subsystem together with the contact roller charging subsystem.

Chatter Vibration of Cleaner Blades

Chatter vibration of the cleaner blade was observed in our low-speed (4 A4 prints/min) laser printer, which uses a contact roller charging subsystem.[2] The contact roller charging subsystem is also suitable for low-end machines because it produces extremely low ozone emissions[3] and lowers the charging voltage compared to that of conventional charging devices such as corotrons and scorotrons. Therefore, it is usually used with the blade cleaner subsystem. A schematic drawing of a xerography engine using the contact charger roller and the cleaner blade is shown in Fig. 1. The urethane blade is glued to a metal bracket and is compressed to the photoreceptor by the 1.25-mm deformation. The blade is 2 mm thick and 8 mm in length beyond the bracket. The charging subsystem consists of the charger roller and a power supply. An ac voltage superposed on dc voltage is applied between the roller and the organic photoconductor (OPC). Electrical microdischarge takes place and the photoconductor is charged or discharged when the instantaneous magnitude of the gap voltage is larger than Paschen's threshold voltage of electrical breakdown. The ac voltage is applied to realize uniform charging.[2] Characteristics of the observed vibration were as follows:

1. The chatter vibration began to be observed in accordance with the increase of print volume, typically at more than 200 to 800 A4 prints. The vibration was also observed using an intentionally worn OPC drum but was not observed for a new drum without abrasion.
2. The main resonance frequency of the vibration spectrum was 2.3 kHz, which was measured by a strain gauge pasted on the blade and a sound-level meter.
3. The vibration arose just after rotation started and just before it stopped. The vibration did not continue at a normal speed.

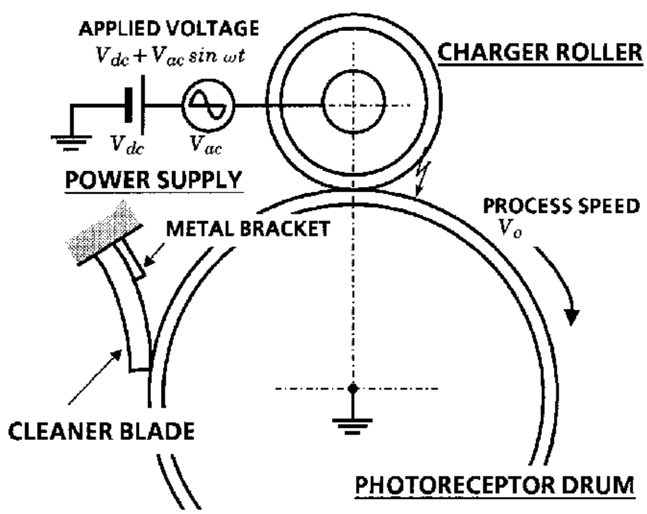

Figure 1. Contact charger roller and cleaner blade in electrophotography engine.

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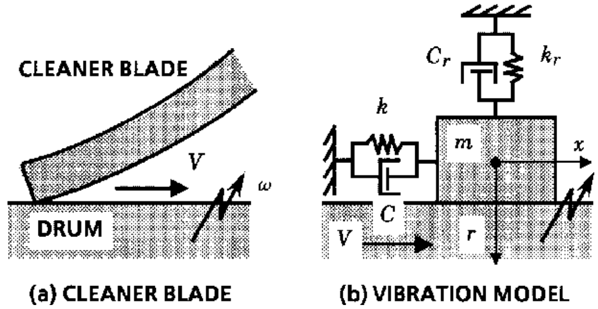

Figure 2. (a) Cleaner blade and (b) vibration model.

4. The vibration did not always occur without applying ac voltage between the contact charger roller and the photoreceptor drum.
5. The vibration rarely occurred with the toner between the blade and the drum.
6. The vibration also rarely occurred at low temperatures.

Modeling

Vibration Model. A typical cleaning subsystem with the cleaner blade is shown in Fig. 2(a). A vibration model of this system has been simplified to single degree of freedom, as shown in Fig. 2(b). Here, m is the effective mass of the blade; k and C are the stiffness and viscous damping coefficient of the blade in the moving direction of the drum (x direction), respectively, k_r and C_r are those perpendicular to the x direction (r direction); and V denotes the drum speed. Because ac voltage induces a high electrostatic field in the vicinity of the nip between the charger roller and the drum, alternating electrostatic force is produced between the roller and the drum and this force induces the vibration of the drum.[4] The main angular velocity of the drum vibration Ω is twice that of the ac applied voltage ω as reported in Ref. 4, i.e., $\Omega = 2\omega$.

Vibration Equation. The force to the blade due to dry friction F_x is

$$F_x = -\mathrm{sgn}(s)\mu F_r, \tag{1}$$

where $\mathrm{sgn}(s) = s/|s|$, s is the relative slip speed between the blade and the drum, $s = V - \dot{x}$, μ is a coefficient of friction, and F_r is the compression force of the blade to the r direction. The coordinate x is determined to be zero where the force by the stiffness k is neutral, and ($\cdot$) refers to a time differential $d(\ )/dt$. The alternating electrostatic force causes drum vibration of a magnitude r_o in the r direction.

$$r = r_o \sin \Omega t,\ \dot{r} = r_o\, \Omega \cos \Omega t. \tag{2}$$

The compression force F_r is

$$F_r = -C_n \dot{r} - k_r (R+r) = -C_r r_o \Omega \cos \Omega t - k_r (R + r_o \sin \Omega t), \tag{3}$$

where R is the nominal deformation of the blade to the r direction due to the compression of the blade to the drum. Equation 3 means that the compression force is not constant, but an alternating component due to the electrostatic vibration of the drum[4] is superposed on the constant component $k_r R$. The coefficient of friction is generally not constant but is a function of the slip speed:

$$\mu = \mu(|s|) = \mu(|V - \dot{x}|). \tag{4}$$

Since the drum vibrates not only to the r direction but also to the x direction,

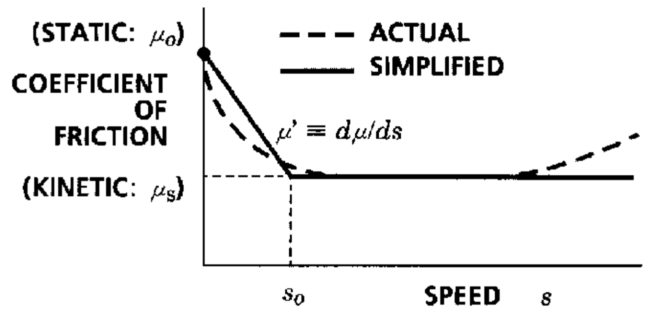

Figure 3. Speed dependence of the friction coefficient.

$$V = V_o + x_o\, \Omega \cos(\Omega t + \alpha). \tag{5}$$

Here V_o is the nominal process speed, x_o is the magnitude of the drum vibration to the x direction, and α is the phase shift between the r and x directions. The angular velocity Ω is common with that of the r direction. Substituting Eq. 5 into Eq. 4,

$$\mu = \mu(|V_o + x_o \Omega \cos(\Omega t + \alpha) - \dot{x}|). \tag{6}$$

Speed dependence of the friction coefficient is simplified as shown in Fig. 3:

$$|s| < s_o:\ \mu = \mu_o + \mathrm{sgn}(s)\{V_o + x_o \Omega \cos(\Omega t + \alpha) - \dot{x}\}\ \mu', \tag{7}$$

$$|s| > s_o:\ \mu = \mu_s, \tag{8}$$

where μ' denotes $d\mu/ds$ and is usually negative when the slip speed is slow; μ_o and μ_s are the static and kinetic coefficients of friction, respectively, and s_o is the threshold speed between the speed-dependent and constant regions of the friction coefficient. Substitution of these relationships into a vibration equation of a single degree of freedom, $m\ddot{x} + C\dot{x} + kx = F_x$, gives

$$\begin{aligned} |s| < s_o:\ & \ddot{x} + [2\zeta\omega_n + \mu'\{\omega_r^2 R + 2\zeta_r \omega_n \Omega r_o \cos\Omega t + \omega_r^2 r_o \sin\Omega t\}]\dot{x} \\ & + \omega_n^2 x \\ & = [\mathrm{sgn}(s)\mu_o + \mu'\{V_o + x_o \Omega \cos(\Omega t + \alpha)\}] \\ & \times (\omega_r^2 R + 2\zeta_r \omega_n \Omega r_o \cos\Omega t + \omega_r^2 r_o \sin\Omega t) \end{aligned} \tag{9a}$$

$$\begin{aligned} |s| < s_o:\ & \ddot{x} + [2\zeta\omega_n \dot{x} + \omega_n^2 x = \\ & \mathrm{sgn}(s)\mu_s\ (\omega_r^2 R + 2\zeta_r \omega_n \Omega r_o \cos\Omega t + \omega_r^2 r_o \sin\Omega t, \end{aligned} \tag{9b}$$

where $\omega_r^2 = k/m, \omega_r^2 = k_r/m,\ 2\zeta_r\omega_n = C/m,\ 2\zeta_r\omega_n = C_r/m$. Because the fundamental vibration, Eq. 9, is a nonlinear forced vibration system including negative damping and parametric excitation terms,[5] specific vibrations can develop under certain specific conditions. In the case of the constant friction coefficient ($\mu' = 0$), Eq. 9 is simplified to a forced vibration system without the negative damping and the parametric excitation. If the drum does not vibrate ($r_o = 0$, $x_o = 0$), the periodic terms in the damping and the force terms vanish and the system is reduced to a free vibration system with the negative damping in $|s| < s_o$ under certain conditions.

Estimation of Parameters

Estimated vibration parameters are as follows:

1. Damping: Figure 4 shows the measured dynamic viscous elasticity, tanδ, of the urethane used for the blade.

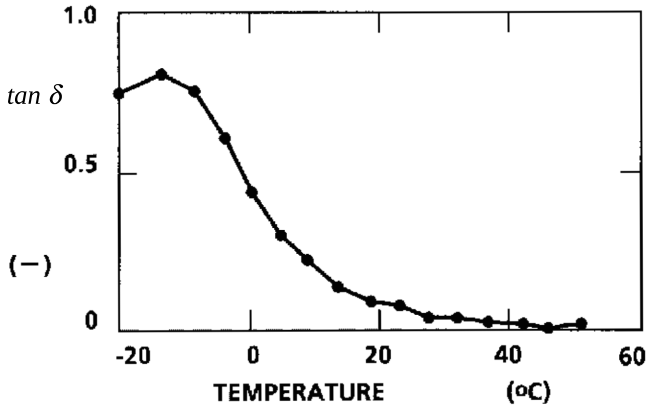

Figure 4. Damping of urethane cleaner blade.

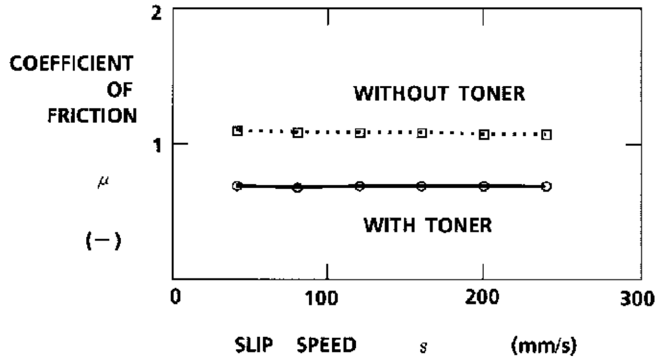

Figure 5. Measured coefficients of friction between cleaner blade and stainless steel.

The damping ratio was estimated to be 0.1 at room temperature and 0.02 at an elevated temperature.

2. Effective Mass and Stiffness: Because the tip of the blade in contact with the drum is largely changed in shape,[6] it is difficult to deduce the effective mass and stiffness of the blade. However, the angular velocity to the x direction ω_n could be determined to be 14,500 rad/s from experimental evidence that the major frequency of the chatter vibration was 2.3 kHz (14,500 rad/s). Because the stiffness to the r direction seems to be small compared with that to the x direction, it is assumed in this report that $\omega_r = 10^{-2}\ \omega_n$.

3. Coefficient of Friction: Figure 5 shows the measured kinetic coefficient of friction in the constant region. It was estimated that $\mu_s = 0.7$ with toner and $\mu_s = 1.1$ without toner. These values coincide approximately with the measurement by Lindblad of $\mu = 1$.[6] On the other hand, because reliable data are not available to date on the coefficient of friction in the low-speed region, we assumed in this report that $\mu' = -100$ s/m and $s_o = 20$ mm/s with toner, and $\mu' = -157$ s/m and $s_o = 20$ mm/s without toner.

4. Nominal Deformation of Blade: The nominal deformation of the blade R is designed to be 1.25 mm.

5. Angular Velocity of Drum Vibration: Because the frequency of the ac voltage ω applied to the contact charger roller is designed to be 160 Hz, the angular velocity of the drum vibration $\Omega = 2\omega = 2000$ rad/s.

6. Magnitude of Drum Vibration: Because the measured acceleration of the electrostatic vibration was of the order of 1 G,[4] the magnitude of the vibration was estimated to be $r_o = x_o = 2.5$ μm at 2000 rad/s.

7. Process Speed: The process speed V_o of the laser printer is 23 mm/s.

Characteristics of Vibration and Stability

1. The external force term, the right side of Eq. 9, includes the constant component. This induces static displacement X_o:

$$|s| < s_o: X_o = \{\mathrm{sgn}\,(s)\mu_o + \mu' V_o\} R(\omega_r/\omega_n)^2, \tag{10a}$$

$$|s| < s_o: X_o = \mathrm{sgn}\,(s) R(\omega_r/\omega_n)^2, \tag{10b}$$

X_o is less than 0.14 μm, which is small compared with the drum vibration r_o and x_o.

2. The damping term, the second term on the left side of Eq. 9a, includes the periodic component. That is, the system is parametric excitation,[5] and therefore it has the potential to be dynamically unstable at $\Omega \doteq 2\omega_n$. The self-excited vibration due to the parametric excitation of the main frequency Ω may, however, not occur, because $\Omega \ll 2\omega_n$ in the present case, but the parametric excitation of the higher mode is possible, because the electrostatic vibration of the drum includes ultraharmonic components.[4] This issue will be investigated more thoroughly in a separate work.

3. Because the external force term of Eq. 9a includes periodic components of frequencies Ω and 2Ω, the frequency of the forced vibration includes Ω and 2Ω components in $|s| > s_o$. The natural frequency ω_n should be designed to be apart from Ω and 2Ω to avoid resonance vibration.

4. Because the friction force is nonconservative, the system has the potential to be dynamically unstable. The following condition, which satisfies dynamic stability, is derived from the condition that the damping term, the second term in the left side of Eq. 9, should be positive.

$$|s| < s_o: 2\zeta\omega_n + \mu'\ \omega_r^2 R > 0,\ \ |s| > s_o: \text{stable.} \tag{11}$$

From this condition, the following important information has been deduced, which can be utilized for the design of the cleaner blade:

a. The negative speed dependence of the friction coefficient should be small.
b. The compression of the blade should be small.
c. The stiffness of the blade in the compression direction should be small.
d. On the other hand, the stiffness of the blade in the moving direction should be large.
e. Damping of the blade should be large.

5. Since the system is stable in $|s| > s_o$, the self-excited vibration may be not divergent but steady in a limit cycle.

Parameter Survey

Because we cannot derive an analytical solution of Eq. 9 without neglecting some important terms, such as the speed dependence of the friction coefficient or the drum vibration, numerical calculations were conducted using the Runge-Kutta-Gill method to investigate the effect of vibration parameters. Calculated results are shown in Figs. 6 through 9. The vibration parameters estimated in the former section were otherwise used unless specified in the figure legends. The original points of the abscissas, time t, are merely the start of the calculation, that is, the figures show transient vibration responses. A unit scale of the abscissa indicates one period corresponding to ω_n. The ordinates are the relative displacement between the blade and the drum, $x - x_o \sin(\Omega t + \alpha)$. Longer period $(2\pi/\Omega)$ vibrations correspond to the forced vibration due to the electrostatic vibration of the drum, and the shorter ones $(2\pi/\omega_n)$ correspond to the self-excited vibration in the limit cycle due to the negative damping. The following information is deduced from the figures. These calculated results qualitatively agree with the experimental observations listed in the former section.

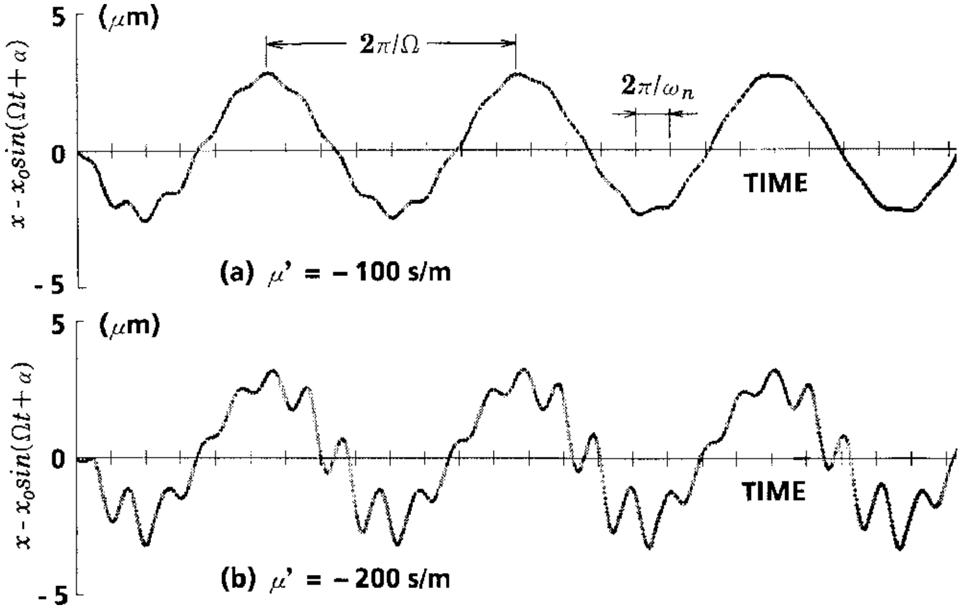

Figure 6. Effect of speed dependence of the friction coefficient for V_o = 10 mm/s, at room temperature with toner.

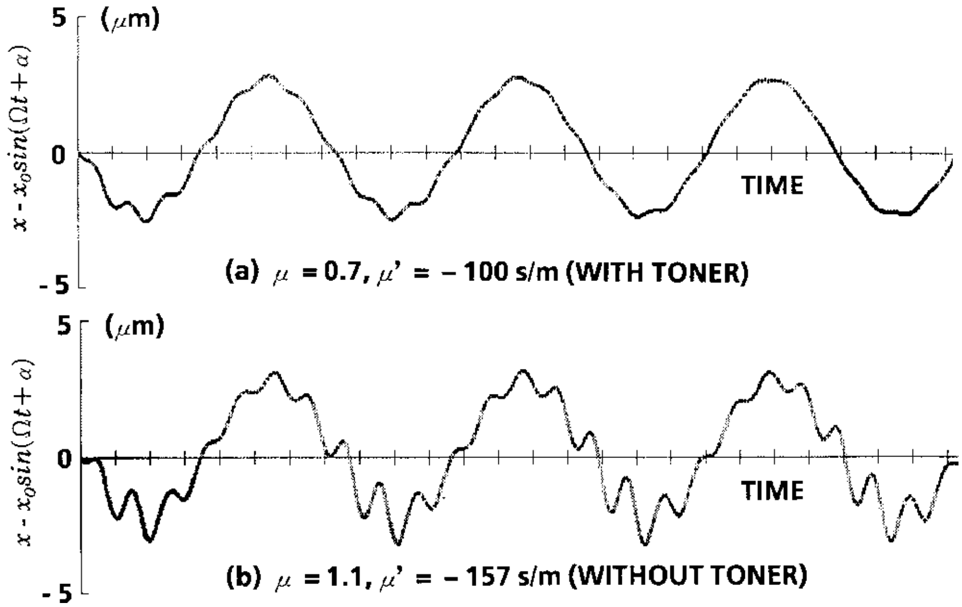

Figure 7. Effect of toner for V_o = 10 mm/s, at room temperature.

1. Effect of Speed Dependence of the Friction Coefficient (Fig. 6): The self-excited vibration does not take place if the speed dependence of the friction coefficient is smaller than the threshold value, – 110 s/m, in the present case. The self-excited vibration is not divergent but convergent to the limit cycle even if the speed dependence is larger than the threshold. The present numerical calculation confirms the theoretical inference discussed in the former section.

2. Effect of Toner (Fig. 7): The untransferred toner stabilizes the self-excited vibration.

3. Effect of Speed (Fig. 8): The self-excited vibration takes place at the low-speed region. Low-end printers have a disadvantage with respect to the chatter vibration of the cleaner blade.

4. Effect of Temperature (Fig. 9): The self-excited vibration is apt to take place at a higher temperature.

Concluding Remarks

Chatter vibration of the cleaner blade charged by a contact charger roller was investigated. From the results of the investigation, the following points were deduced:

1. Chatter vibration is basically induced by nonlinear negative damping due to the negative speed dependence of the friction coefficient between the cleaner blade and the photoreceptor drum. Forced vibration and parametric excitation are also generated by the photoreceptor vibration induced by the alternating electrostatic force of the charger roller.
2. From the results of calculations and experimental observations, the characteristics of the vibration are assumed to be as follows:
   a. Chatter vibration does not take place at an early stage of print volume, because the speed dependence of the friction coefficient between the drum and the blade is small for the fine drum and the blade without abrasion.
   b. Chatter vibration began to be observed in accordance with the progress of abrasion, because the speed dependence of the friction coefficient is increased by abrasion.

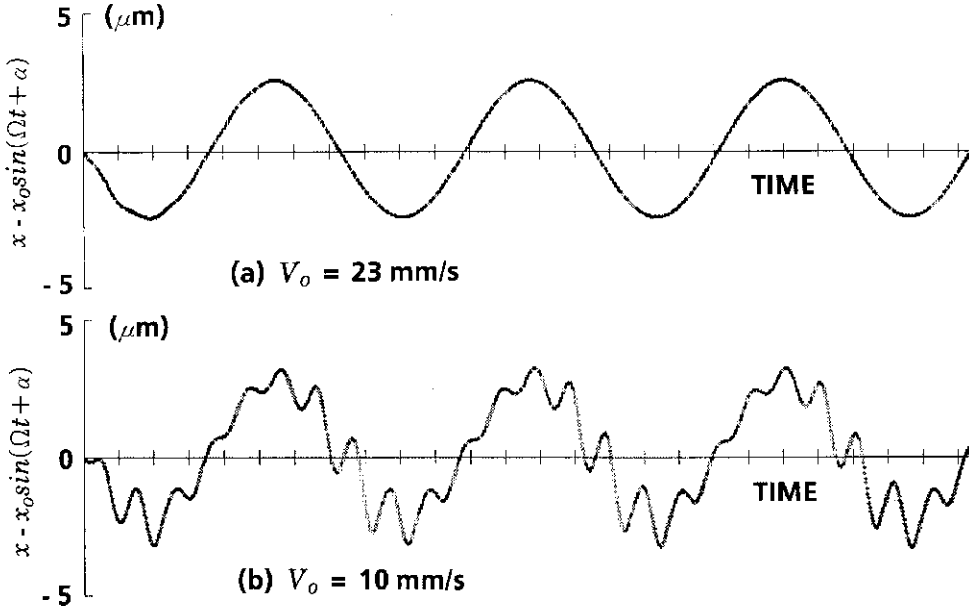

Figure 8. Effect of speed for μ'= -200 s/m, at room temperature with toner.

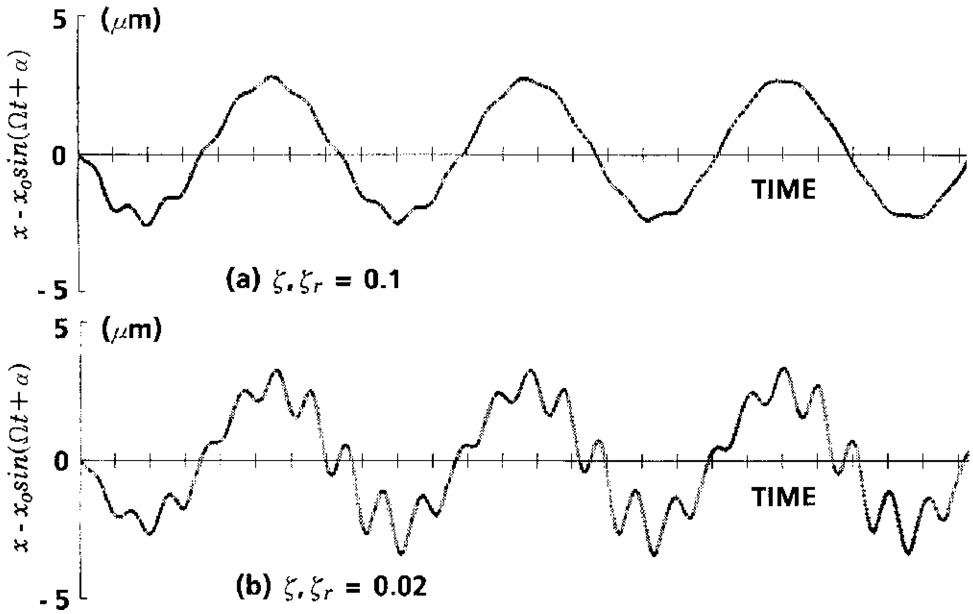

Figure 9. Effect of damping (temperature) for V_o= 10 mm/s, μ' = -200 s/m, with toner.

  c. Chatter vibration is apt to take place just after rotation starts and just before it stops, but the vibration does not continue at a normal speed, because the speed dependence of the friction coefficient is small in a relatively high-speed region.
  d. The existence of untransferred toner reduces the coefficient of friction and stabilizes the self-excited vibration.
  e. The chatter vibration rarely occurs at a low temperature, because the damping of the blade is larger under higher temperatures.
3. The following methods are effective to suppress the chatter vibration:
  a. The speed dependence of the friction coefficient should be small.
  b. The compression of the blade should be small.
  c. The stiffness of the blade in the compression direction should be small.
  d. On the other hand, the stiffness of the blade in the moving direction should be large. The natural frequency of the blade to the x direction should be designed to be different from one and two times the frequency of the drum vibration to avoid resonance vibration.
  e. Damping of the blade should be large. The existence of untransferred toner and operation under low temperature are preferable.
  f. The forced vibration of the drum should be small. Another effective means of suppressing chatter vibration is to adopt some proposed methods[4] that can reduce the electrostatic vibration induced in the contact charger roller.

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Nomenclature

C = viscous damping coefficient of blade in x direction, $2\zeta\omega_n \equiv C/m$ — Ns/m
C_r = viscous damping coefficient of blade in r direction, $2\zeta_r\omega_n \equiv C_r/m$ — Ns/m
F_r = compression force of blade to r direction — N
F_x = force to blade due to dry friction — N
k = stiffness of blade in x direction, $\omega_n^2 \equiv k/m$ — N/m
k_r = stiffness of blade in r direction, $\omega_r^2 \equiv k_r/m$ — N/m
m = effective mass of blade — kg
R = nominal deformation of blade to r direction due to compression of blade to drum — m
r = coordinate perpendicular to x direction — m
r_o = magnitude of drum vibration in r direction — m
s = relative slip speed between blade and drum, $s \equiv V - \dot{x}$. — m/s
s_o = threshold speed between speed dependent and constant region of friction coefficient — m/s
V = drum speed — m/s
V_o = nominal process speed — m/s
X_o = static displacement of blade in x direction — m
x = coordinate to moving direction of drum, determined to be zero where the force by the stiffness k is neutral — m
x_o = magnitude of drum vibration in x direction — m
α = phase shift of drum vibration between r and x directions — deg
μ = coefficient of friction between drum and blade — —
μ' = speed dependence of friction coefficient, $\mu' \equiv d\mu/ds$. — s/m
μ_s = kinetic coefficient of friction — —
μ_o = static coefficient of friction — —
Ω = angular velocity of electrostatic drum vibration — rad/s
ω = angular velocity of ac voltage applied to contact charger roller — rad/s

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