Banding Mechanism Caused by Self-Excited Vibration in Contact Development System

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Abstract

Previous studies of the banding mechanism have proposed various reduction methods for the electrophotographic process. The banding mechanism is generally caused by vibration transmitted by the drive system and the resulting resonance. We analyzed banding that occurred in a single-component contact development system, and it became clear that banding was caused by self-excited vibration. Our self-excited vibration model consists of the following elements: a spring element (the rubber layer of the development roller), a mass element (the development unit), and an exciting force (the friction between the development roller and the OPC drum). We also observed that the model's vibration frequency varies nonlinearly, an interesting characteristic caused by the nonlinear hardening properties of the spring. Since self-excited vibration can be prevented by stabilizing the system, we propose the use of appropriate damping components. We also statistically verified the banding reduction effect by conducting quantitative analysis of vibration intensity.

Introduction

Halftone uniformity in the electrophotographic process is very important in meeting demand for higher quality images. Since the appearance of banding makes uniformity worse, several studies have been conducted on banding to achieve banding-free images. Kawamoto investigated chatter vibration of a cleaner blade induced by negative damping between the cleaner blade and photoreceptor, and by electrostatic force between the charge roller and photoreceptor.[1] Chen conducted detailed studies of OPC drum speed regulation to reduce banding.[2]

Major banding is caused by photoreceptor velocity fluctuations which are induced by periodic vibrations of the transmission, such as gear train noise and mechanical resonance.

In this paper, we analyze a banding mechanism which occurs in a single-component contact development system, and clarify that the banding is caused by self-excited vibration. We propose the use of appropriate damping components, and show the banding reduction efficiency statistically.

Experimental

Banding in a Single-Component Contact Development System

Figure 1 shows a schematic of our single-component contact development system. The circumferential speed of the photoreceptor drum (OPC) is 117 mm/sec, and the development roller, which has a rubber layer, is driven with a different circumferential speed. The development roller contacts the photoreceptor drum with some pressure due to the outside spring.

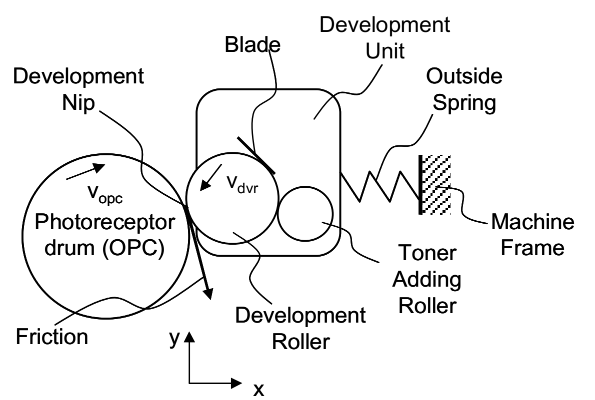

Figure 1. Schematic of a single-component contact development system

In this single-component contact development system, periodic banding non-uniformity rarely appeared. The period was 1.39 mm and OPC speed was 117 mm/sec, so the frequency of the vibration was 84 Hz. Vibration measurement using an acceleration pickup and rotary encoder yielded the following two results. 1) There was no periodic vibration peak in the driving mechanism around the frequency 84 Hz. 2) The whole development unit vibrated in the x axis direction.

Figure 2 shows the vibration intensity of the development unit in the x axis direction, as measured by an acceleration pickup. The time domain chart is shown above, and the frequency domain chart with FFT is shown below. The chart shows strong periodic vibration around 84 Hz. This result suggests that the periodic banding is due to the natural vibration of a spring-mass system (where the mass element is the development unit).

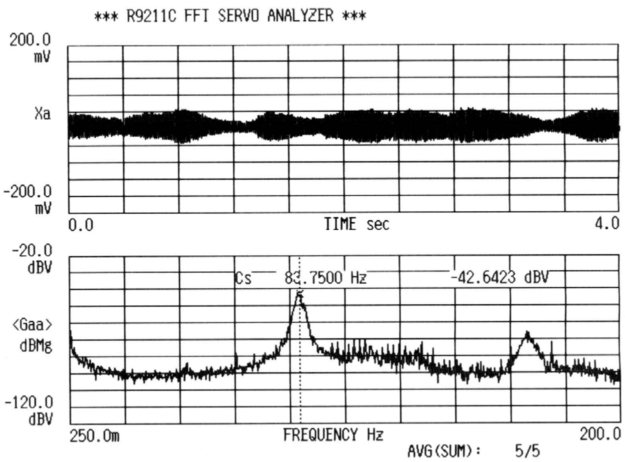

Figure 2. Vibration intensity of the development unit in the x axis direction, as measured by an acceleration pickup

The natural frequency f_n is defined by Eq.1. The mass m of the development unit was 1.4 kg and the frequency f_n was 84 Hz. Therefore the spring constant k is calculated to be 390 kN/m. The spring constant of the outside spring in Fig.1, which provides pressure between the development roller and the photoreceptor drum, was 1 kN/m. Therefore the outside spring has no relevance to the 84 Hz vibration. This calculation result suggests that the spring element is very hard.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{1}$$

Spring Constant k of the Development Roller

Figure 3 shows the relation between load and deformation of the development roller, which has a rubber layer and an effective length of 30 cm. The slope of the curve corresponds to the spring constant k. Deformation increased nonlinearly with the load. This indicates that the spring is a nonlinear hardening spring.

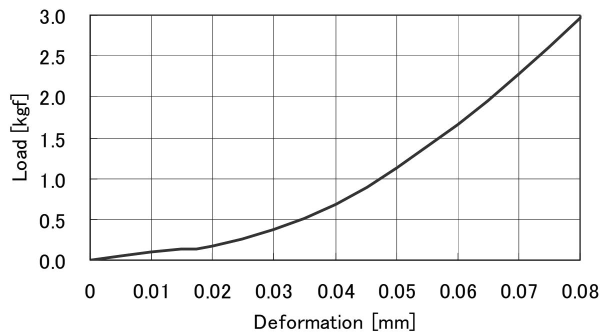

Figure 3. Relation between load and deformation of the development roller

The pressure of the development nip was set to approximately 30 gf/cm by the outside spring. At that tine, the load to the development roller was 0.9 kg. We can determine the spring constant k of the development roller to be 390 kN/m from Fig. 3. This result corresponds to the calculation result using Eq. 1. These results suggest that the spring element of the 84 Hz vibration is the development roller.

In addition, experimental results using a heavier development unit (with 160 g of added weight) showed that the vibration frequency shifted approximately 4 Hz lower. The calculation result using Eq. 1 was 79.6 Hz when the mass was 1.56 kg. These results prove that banding is due to the natural vibration of a spring-mass system, where the mass element is the development unit and the spring element is the development roller.

Lastly, when the development nip pressure was set to 17 gf/cm, the vibration frequency shifted about 10 Hz lower. In this case, we can determine the spring constant k to be 295 kN/m from Fig. 3. The frequency is calculated to be 73 Hz by substituting this value into Eq. 1. It is clear that the banding is based on a spring-mass system.

Self-Excited Vibration Model

We must clarify the source of the force to the spring-mass system. We assumed that the force was the friction force at the development nip, and that the vibration was due to self-excited vibration. The grounds for this assumption are the following experimental results: 1) There was no periodic vibration peak in the driving mechanism around the frequency 84 Hz. 2) When the thickness of the toner layer on the development roller was varied, the intensity of the vibration varied accordingly. 3) When the relative velocity between the OPC and development roller was varied, the intensity of the vibration varied.

In the case of a single-component contact development system, like that in Fig. 1, the friction is generated at the development nip. General friction is called Coulomb friction and is not a function of the relative velocity, as shown in Fig. 4. Coulomb friction F_c is given by Eq. 2.

$$\begin{aligned} F_c(v) &= c_c \quad \{v > 0\} \\ &= -c_c \quad \{v < 0\} \end{aligned} \tag{2}$$

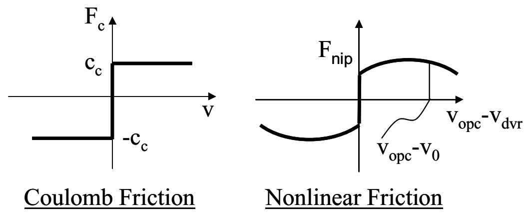

Figure 4. Concept of Coulomb friction and nonlinear friction

Given that friction at the development nip F_{nip} is a function of relative velocity (v_{opc}-v_{dvr}), as indicated in Fig. 4, the friction is given by Eq. 3, where v_{opc} is the circumferential velocity of the OPC and v_{dvr} is the circumferential velocity of the development roller.

$$F_{nip} = f(v_{opc} - v_{dvr}) \tag{3}$$

Equation 4 gives a Taylor series approximation to Eq. 3 for v_{dvr} close to v_0, up to terms of order v_{dvr}, where v_0 is the set velocity of the development roller.

$$F_{nip} = f(v_{opc} - v_0) - f'(v_{opc} - v_0)(v_{dvr} - v_0) \tag{4}$$

F_x is the X axis component of the nip friction F_{nip} and is given by Eq. 5, where v_x is the velocity in the X axis direction, and α is the angle between the F_{nip} direction and Y axis direction.

$$\begin{aligned} F_x &= F_{nip} Sin(\alpha) \\ &= f(v_{opc} - v_0) Sin(\alpha) - f'(v_{opc} - v_0)(v_{dvr} - v_0) Sin(\alpha) \\ &= f(v_{opc} - v_0) Sin(\alpha) - f'(v_{opc} - v_0) v_x \\ v_x &= (v_{dvr} - v_0) Sin(\alpha) \end{aligned} \tag{5}$$

The equation of motion for the development unit is defined by Eq. 6, where m is the mass, c is the damping coefficient, k is the spring constant, and the right hand side is the friction given by Eq. 5.

$$\begin{aligned} & m\,a_x + c\,v_x + k\,x_x = f(v_{opc} - v_0) Sin(\alpha) - f'(v_{opc} - v_0) v_x \\ & \therefore m\,a_x + \{c + f'(v_{opc} - v_0)\} v_x + k\,x_x - f(v_{opc} - v_0) Sin(\alpha) = 0 \end{aligned} \tag{6}$$

Equation 6 us simplified to Eq. 8 by substituting x_1, as defined Eq.7. The standard form is given by Eq. 9, where ζ is the damping ratio and ω_n is the natural frequency.

$$x_1 = x_x - \frac{f(v_{opc} - v_0) Sin(\alpha)}{k} \tag{7}$$

$$m\,a_x + \{c + f'(v_{opc} - v_0)\} v_x + k\,x_1 = 0 \tag{8}$$

$$\begin{aligned} & a_x + 2\zeta\omega_n v_x + \omega_n^2 x_1 = 0 \\ & \omega_n = \sqrt{\frac{k}{m}}, \qquad \zeta = \frac{c + f'(v_{opc} - v_0)}{2\sqrt{k\,m}} \end{aligned} \tag{9}$$

$$c < -f'(v_{opc} - v_0) \tag{10}$$

Consequently, if the nip friction is a function of relative velocity (v_{opc}-v_{dvr}) and the value of f'(v_{opc}-v_0) satisfies Eq. 10, the damping ratio ζ assumes a negative value, and the system starts self-excited vibration.

Results and Discussion

Banding Reduction

For general resonance, there are various vibration reduction methods: changing the natural frequencies of the vibrated parts or the outside noise, diminishing the outside noise level, and providing a damping element. For self-exciting vibration, the basic reduction method is stabilizing the system. That is, the damping ratio ζ has a positive value. The nip friction F_{nip} depends on the condition of the toner layer and the relative velocity between the OPC and the development roller. So it is difficult to change the value of f'(v_{opc}-v_0). Figure 5 shows the concept of a friction load to reduce self-excited vibration. This friction load in the X axis direction can make the damping ratio positive and stabilize the system. Figure 6 shows experimental results using the friction load.

For vibration intensity, we had to set a vibration reduction target. Analysis of the vibration intensity and the output halftone images showed that banding became visible when intensity became larger than -50dB. So we had to reduce the vibration intensity below -50dB. The results in Fig. 6 showed the friction load was enough to reduce the visible vibration.

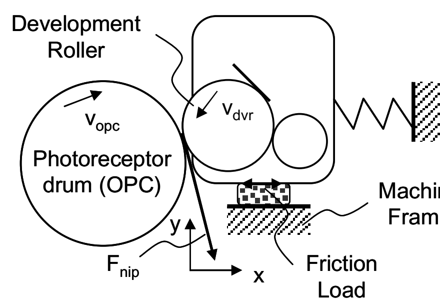

Figure 5. Concept of a friction load to reduce self-excited vibration

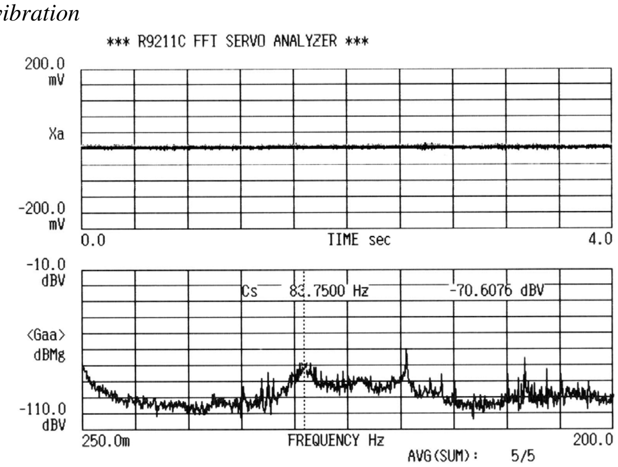

Figure 6. Vibration intensity of the development unit using a friction load in the x axis direction.

Figure 7 shows the effect of vibration reduction using a friction load in the X axis direction. When the system did not use the friction load, the intensity of the vibration was about -40 to -30 dB. Using a damping element, vibration intensities were reduced about -70dB. Variation of the friction load element was as follows: the circular dots indicate initial conditions, the triangular dots indicate a leaf spring, square dots indicate a PET film, and the diamond dots indicate urethane foam.

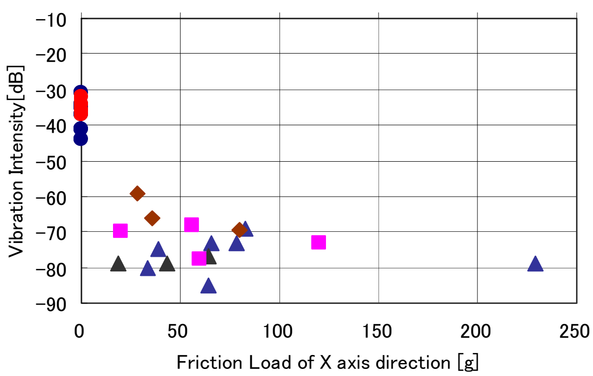

Figure 7. Relation between friction load and vibration intensity

Figure 7 shows that the effect of vibration reduction was not related to the magnitude of the friction load. A small friction load was very effective in reducing vibration. The experimental results show that banding due to the self-excited vibration system and the friction load could stabilize the system. A friction load which was too large prevented the development unit from moving smoothly in the x axis direction and the development nip pressure became non-uniform. So this small friction load was suitable for avoiding bad effects on the image.

Statistical Evaluation

We evaluated the manufacturing dispersion of vibration statistically by using a number of test development units and establishing many different test conditions. Figure 8 shows the distribution of the vibration intensity, which was assumed to follow a Gaussian distribution. The solid curve is the initial vibration distribution when not using a friction load element. The average intensity of the vibration was -63.8 dB and the standard deviation σ was 7.2. The -50 dB point corresponds to 1.92 σ. The probability that visible banding will occur is calculated to be 2.7%. The distribution when using a friction load, given by the broken line, shows the effect of the vibration intensity reduction. The average vibration intensity was reduced to -72.6 dB and the standard deviation σ became 5.6. Consequently, the -50 dB point corresponds to 4.02 σ and the probability that visible banding will occur becomes 0.003%.

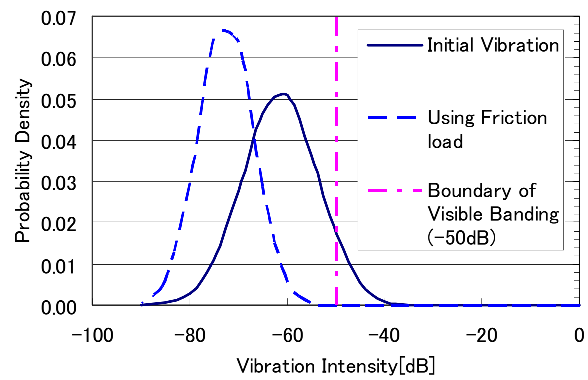

Figure 8. Statistical effect of the friction load element on vibration intensity reduction

Conclusion

We analyzed periodic banding, which rarely occurred in our single-component contact development system, and clarified that the banding was due to the self-excited vibration of a spring-mass system, where the mass is the development unit and the spring is the development roller. The force which increased the vibration intensity was friction between the OPC and the development roller, and this had nonlinear characteristics with regard to the relative velocity. Self-excited vibration caused negative damping due to this nonlinear friction.

To stabilize the self-excited vibration, we proposed a small friction load, which made the damping ratio positive, and verified the effect of vibration reduction statistically.

References

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Biography

Tadashi Iwamatsu received his B.S. degree in mechanical engineering from Kyoto University in 1986. After graduation, he joined the Sharp Corporation in 1986. He has been engaged in the design of servomechanisms, optical design of hologram scanners, and research on electrophotography. He belongs to the Production Technology Development Group. His current research interests include simulation of electrophotography. He is a member of the IS&T.