A Consideration about Oscillation Mechanism of a Cleaning Blade

Minoru Kasama, Yiemei Yu and Tomoyuki Itoh
Fuji Xerox Co., Ltd.
Ashigarakami-gun, Kanagawa/Japan

Masatsugu Yoshizawa and Yoshiaki Itoh
Department of Mechanical Engineering, Keio University
Yokohama, Kanagawa/Japan

Abstract

A mechanism of a cleaning blade squeaking due to the friction between the photoreceptor drum and the blade in a copy machine is presented in this paper. Although this problem has been known for many years, the details of oscillation mechanism are not clarified, so we have not found out the fundamental solution yet. Here, in order to clarify this oscillation mechanism, we took notice of the feature of the oscillation that the frequency changed discretely with a normal force of the blade. In other words, we found out that the oscillation is caused by couple of immanent modes in the cleaning blade, and it was dependent on the friction force which mode became unstable coupling. As a result, we confirmed that the discrete change of oscillation frequency was explained by the unstable coupled modes of the cleaning blade.

1. Introduction

Recently, laser printers via an electronic photograph system have come into wide use not only in offices but also in home use due to improvement of an image quality and printing speed. Therefore, consumers came to consider consumption of energy and quietness as a new value of these products. Especially, since a laser printer has many opportunities to be placed near users, it is very important to avoid noise, which makes users uncomfortable.

However, an unforeseen noise may occur from a cleaning blade in a laser printer. Although this blade squeaking has been known as one of typical unforeseen noise of a laser printer for many years, there is a dearth of information of a mechanism except that a self-excited vibration due to a dry friction between the photoreceptor drum and the blade causes this noise.

Thus, it is expected to clarify the mechanism of the self-excited vibration, which causes the unforeseen noise and to find out the fundamental solution to prevent the unforeseen noise. Until now, there are some conventional studies about the unforeseen noise due to the self-excited vibration in a copy machine or a printer. For example, Nakamura[1] clarified the mechanism that the friction force of the cleaning blade excited a twist vibration mode of the photoreceptor drum. In this case, the photoreceptor drum emits the noise. In another example, Kawamoto[2] focused the unforeseen noise arising just after the drum starts to rotate or just before a stop. The mechanism of oscillation is explained as the nonlinear phenomenon combined the self-excited vibration due to the negative damping and the forced vibration by AC charger. These studies indicate that the mechanism of the unforeseen noise is not only one.

In this paper, at first, we show a new unforeseen noise, which has different features from above studies. Next, observing the phenomenon in an actual machine, we clarify the peculiar features of this unforeseen noise quantitatively. Furthermore, we try to formulate the mechanism of the self-excited vibration, which causes this unforeseen noise. Finally, the numerical simulation applied the physical parameters of an actual machine is carried out, then the validity of presented the model is inspected.

2. Observation in an Actual Machine

2.1 Overview of this Unforeseen Noise Characteristic

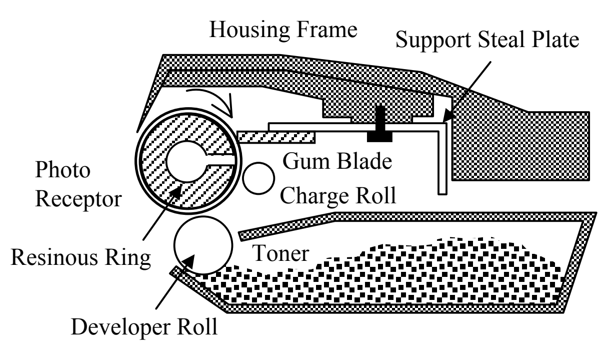

Fig.1 Schematic Diagram of Toner Cartridge in a Laser Printer

Fig.1 shows a schematic diagram of toner cartridge in a laser printer. The cleaning blade consists of a blade gum and a support steel plate. It is fixed on a cartridge housing at two points. In order to clean up the photoreceptor surface for the next imaging process, the cleaning blade is pressed on it in the counter direction to the photoreceptor rotation.

The unforeseen noise occurs at the part where the cleaning blade contacts with the photoreceptor on condition as follows.

- When temperature inside a machine becomes high.
- When the friction coefficient becomes high due to discharging products on the photoreceptor surface.

Fig.2 shows the result of frequency analysis of this unforeseen noise in an actual machine.

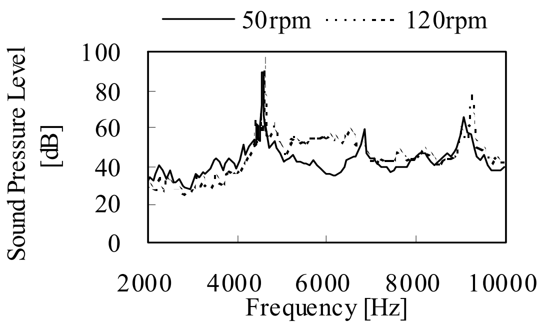

Fig.2 Frequency Analysis of the Unforeseen Noise

Fig.2 indicates that the unforeseen noise consists 4600Hz component and the second order harmonics. Furthermore, the frequency of this unforeseen noise does not vary for increasing of the photoreceptor rotating velocity from 50 rpm to 120 rpm.

Generally, a cause of a noise with particular frequencies is either a resonance or a self-excited vibration[3]. In the former case, the frequency of noise corresponds to that of an external exciter such as motors or gears. In the latter case, the frequency of noise corresponds to the natural frequency of an oscillating system. In the case of this unforeseen noise, the frequency does not depend on a motor revolution. Thus, it is evident that the self-excited vibration due to the dry friction causes this unforeseen noise.

2.2 Peculiar Features of this Unforeseen Noise

The self-excited vibration due to the dry friction usually occurs at low speeds just before a body in motion stops or immediately after a body starts to move. For a example, a squeaking noise from a brake pad of an automobile is a familiar case of the self-excited vibration[4].

However, this noise occurs at high speeds when the photoreceptor is rotating steady. This is the first peculiar feature of this unforeseen noise. Fig.3 shows the relationship between velocity of the photoreceptor revolution and amplitude of the noise.

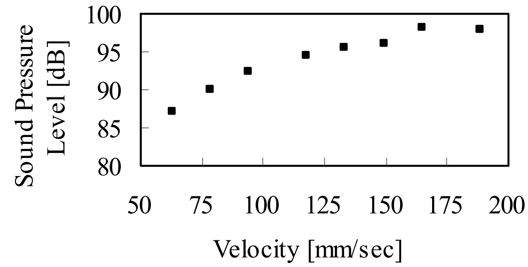

Fig.3 Relationship between Velocity of Photoreceptor Surface and Noise Amplitude

It is clarified that amplitude of the noise becomes large in proportion to velocity of the photoreceptor revolution.

Furthermore, the second peculiar feature of this noise, the frequency of this noise shifts lower with increasing temperature of the cleaning blade. Fig.4 shows the variation of frequency of the unforeseen noise at each temperature.

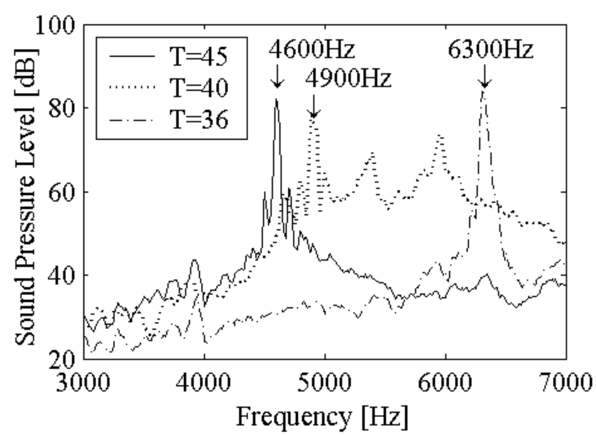

Fig.4 Frequency Shift due to the Variation of Temperature

When the temperature of blade reaches at 36 degree, the unforeseen noise occurs at 6300Hz. The temperatures inside machine continue rising up further, the frequency of noise varies from 6300Hz to 4900Hz, besides 4600Hz.

As mentioned above section, the frequency of noise due to a self-excited vibration corresponds to the natural frequency of the oscillation part.

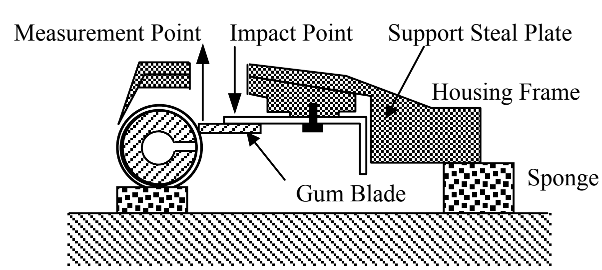

Fig.5 Measurement Point for Impulsive Response of the Cleaning Blade Assembly

We have then evaluated natural frequencies of the cleaning blade. The natural frequencies have been evaluated through the frequency analysis of free vibration of the cleaning blade when an impulsive force is given. Fig.5 shows the impact point and measurement point in the cleaning blade. A part of the housing was cut in order to expose the cleaning blade. We got the impulsive response at the edge of blade using a laser doppler velocity meter when the support steel plate was hit. Generally, it is known that the natural frequencies correspond to peak frequencies of the free vibration.

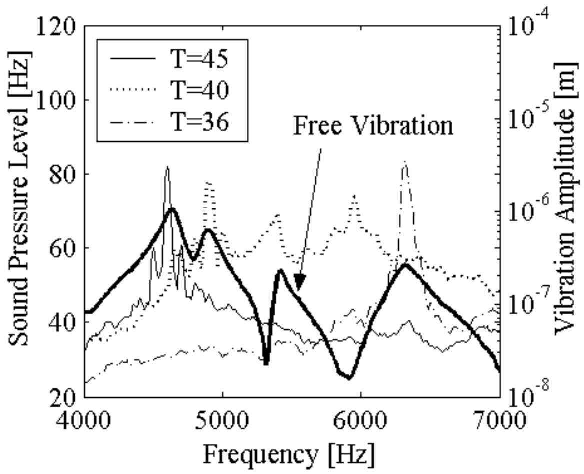

Fig.6 Comparison between the Free Vibration of Cleaning Blade Assembly and the Unforeseen Noise

A comparison of the noise frequency and the natural frequency is shown in Fig.6. Both peak frequencies show in good agreement. Thus, the frequency of unforeseen noise at each temperature corresponds to the natural frequency of the cleaning blade, and moreover, it depends on temperature of the cleaning blade which natural frequency causes unstable oscillation.

These features have not been seen in conventional studies[1, 2] about unforeseen noise of a cleaning blade.

3. Mathematical Formulation and Model

3.1 Conventional Model[5]

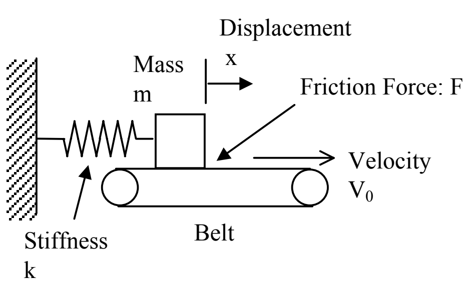

Fig.7 Conventional Model of the Self-Excited Vibration

In order to clarify the mechanism of the self-excited vibration, which caused this unforeseen noise, we consider a vibration model. A conventional model of a self-excited vibration due to friction is shown in Fig.7 In this model, consider a mass *m* on a moving belt, which is driving at constant velocity V_0 is connected through a spring k to a fix point. Between mass and belt, dry friction occurs, with the friction force F. Then, the dynamic equation about the displacement x is described as

$$m\ddot{x} + kx = F(V_0 - \dot{x}) \tag{1}$$

Here, suppose $|\dot{x}| << V_0$, the right term of the Eq. (1) is described by Taylor expansion as

$$F(V_0 - \dot{x}) = F(V_0) - \frac{dF(V_0)}{dV_0} + \cdots \tag{2}$$

From Eq. (1) and (2), we got

$$m\ddot{x} + \frac{dF(V_0)}{dV_0}\dot{x} + kx - F(V_0) = 0 \tag{3}$$

Then, the general solution is

$$x(t) = A_0 \mathrm{Exp}(-ct) \times \mathrm{Cos}\left(\sqrt{c^2 - \omega^2}\, t\right) \tag{4}$$

where A_0 is constant, c and ω are defined as

$$c = \frac{dF(V_0)}{dV_0} \qquad \omega^2 = \frac{k}{m} : \text{natural angle fequency}$$

Fig.8 indicates the relationship between the relative velocity and the friction force. From this figure, it is clarify that c has negative values in a region of low velocity.

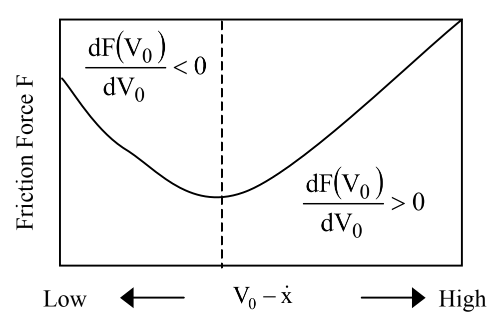

Fig.8 Relationship between Relative Velocity and Friction Force

Since c has negative value, displacement x in Eq. (4) increases exponentially. Thus, the self-excited vibration of this model has below features.

(1) Frequency of vibration has only one value, which corresponds to the natural frequency of system.

(2) This oscillation is caused in a region of low velocity where c has negative value.

However, these features are not in agreement with those of the unforeseen noise in section 2.2, namely this noise is caused in a region of high velocity and its frequency takes several values for temperature of blade. Thus, in order to explain these features, we have to consider another model.

3.2 Coupled Vibration Model

The most important feature of this unforeseen noise is that the noise is related to plural natural frequency. Therefore, a model of this noise also has to be able to treat plural natural frequency, namely it has multi degrees of freedom. Therefore, we direct our attention to a coupled vibration model in multi degrees of freedom system. This model has already been applied to the self-excited vibration of a contact head slider in magnetic recording equipment[6] or wiper blade in automobile[7].

As Fig.9 shows, we consider a model of the cleaning blade with a finite elements and nodes.

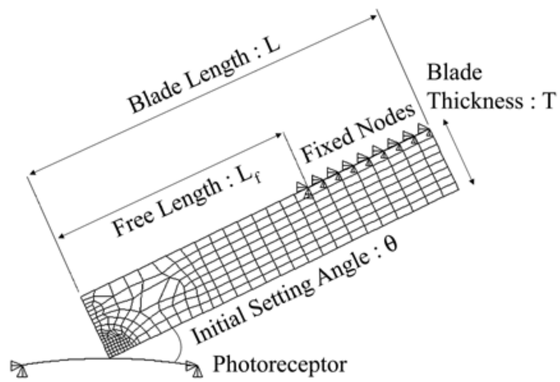

Fig.9 Finite Elements and Nodes Model of the Cleaning Blade

A dynamic equation of forced oscillation of the blade is described as

$$[\mathbf{M}]\ddot{\mathbf{x}} + [\mathbf{K}]\mathbf{x} = \mathbf{F} \tag{5}$$

where **x** is the vector, which consists displacement of each node from balanced point, **F** is the vector, which consists external force applied at each node, [**M**] is mass matrix and [**K**] is stiffness matrix.

According to the modal analysis theory, **x** is represented with the mass normalizing eigenvectors $\boldsymbol{\varphi}_i$ (i=1,...,n) as

$$\mathbf{x} = [\boldsymbol{\varphi}_1, \cdots, \boldsymbol{\varphi}_n]\begin{bmatrix} q_1 \\ \vdots \\ q_n \end{bmatrix} = \mathbf{\Phi q} \tag{6}$$

where, n is the number of mode, **q** is displacement in modal space whose base vector is $\boldsymbol{\varphi}_i$ .Substituting Eq.(6) into Eq.(5) leads to

$$\ddot{\mathbf{q}} + \mathbf{\Omega}^2 \mathbf{q} = \mathbf{\Phi}^T \mathbf{F} \tag{7}$$

where $\mathbf{\Omega}^2$ is the square matrix whose diagonal components correspond to the natural angle frequencies as.

$$\mathbf{\Omega}^2 = \begin{bmatrix} \omega_1^2 & 0 & \cdots & 0 \\ 0 & \omega_2^2 & 0 & \vdots \\ \vdots & 0 & \ddots & 0 \\ 0 & \cdots & 0 & \omega_n^2 \end{bmatrix} \tag{8}$$

In this case of the cleaning blade, the vector **F** consists friction force at the edge of cleaning blade. So it can be represented with a coefficient of friction μ and the normal force vector $\mathbf{F}_N$, i.e.,

$$\mathbf{F} = \mu \mathbf{F}_N \tag{9}$$

If we assume that the normal force depends on the displacement of nodes at contact area, the normal force vector $\mathbf{F}_N$ is represented as below.

$$\mathbf{F}_N = \begin{bmatrix} a_{11} & a_{12} & \cdots & a_{1n} \\ a_{21} & a_{22} & \cdots & a_{2n} \\ \vdots & \vdots & \vdots & \vdots \\ a_{N1} & a_{N2} & \cdots & a_{Nn} \end{bmatrix}\begin{bmatrix} q_1 \\ q_2 \\ \vdots \\ q_n \end{bmatrix} = \mathbf{Aq} \tag{10}$$

where N is the number of node, a is the coefficient which represents the reaction force at each node. In brief, we assume that the normal force $\mathbf{F}_N$ also can be represented in a linear combination of reaction force**,** same as displacement **x** in Eq.(6).

Substituting Eq.(9) and Eq.(10) into Eq.(7) leads to

$$\ddot{\mathbf{q}} + \mathbf{\Gamma}\, \mathbf{q} = \mathbf{0} \tag{11}$$

where

$$\mathbf{\Gamma} = \mathbf{\Omega}^2 - \mu \mathbf{\Phi}^T \mathbf{A} \tag{12}$$

When μ=0, then $\mathbf{\Gamma} = \mathbf{\Omega}^2$, Thus, all eigenvalues of Eq.(11) are pure imaginary numbers. When μ has some value, then **Γ** has asymmetric components. Thus, there is possibility that eigenvalue of Eq.(11) has a positive real part. In order to inspect such a possibility, we consider a case that two modes are coupled. If we assumed that n=2, Eq.(11) becomes

$$\begin{bmatrix} \ddot{q}_1 \\ \ddot{q}_2 \end{bmatrix} + \begin{bmatrix} \omega_1^2 - \mu\gamma_{11} & -\mu\gamma_{12} \\ -\mu\gamma_{21} & \omega_2^2 - \mu\gamma_{22} \end{bmatrix}\begin{bmatrix} q_1 \\ q_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{12}$$

where

$$\gamma_{lm} = \boldsymbol{\varphi}_l^T \boldsymbol{\alpha}_m \qquad l=1,2 \;\; m=1,2 \tag{13}$$

$$\boldsymbol{\alpha}_m = [a_{1m} \quad a_{2m} \quad \cdots \quad a_{Nm}]^T \tag{14}$$

The condition that the characteristic root of Eq.(12) has positive real part is described as

$$\left[\left(\omega_1^2 - \mu\gamma_{11}\right) - \left(\omega_2^2 - \mu\gamma_{22}\right)\right]^2 + 4\mu^2\gamma_{12}\gamma_{21} < 0 \tag{15}$$

Usually $\omega^2 >> \mu\gamma$ Thus, Eq.(15) is simplified as

$$\left(\omega_1^2 - \omega_2^2\right)^2 + 4\mu^2\gamma_{12}\gamma_{21} < 0 \tag{16}$$

From Eq.(16), we can write the unstable condition of Eq.(12) as

1) γ_{12} and γ_{21} have different sign each other.

2) ω_1 and ω_2 are close value.

3) The coefficient of friction μ is large

Where we note that the couple of modes, which satisfy the unstable condition, can change by a variation of the normal force with temperature, namely the variation of unforeseen noise frequency can be explained as the change of unstable mode coupling. Furthermore the unstable condition does not depend on the velocity. So according to this model, it is possible that the self-excited vibration is caused in region of high velocity.

Thus, this model is more suitable to explain this unforeseen noise features mentioned in section 2.2 than conventional one.

4. Numerical Simulation

In order to confirm the validity of the coupled vibration model, a numerical simulation with the same physical parameters as an actual cleaning blade was carried out. Table-1 shows the setting values used in the numerical simulation.

Table-1 Parameters of Numerical Simulation

Parameters	Values
Blade Length : L [mm]	11.5
Free Length : L_f [mm]	7.5
Blade Thickness : T [mm]	2.0
Initial Setting Angle : θ [degree]	27
Young's Modulus of Blade [MPa]	5.88
Density [kg/mm^3]	1.5×10^{-6}
Poisson Ratio	0.495

At first, an analysis for large deformation when the blade edge was pressed onto the surface of a photoreceptor with interference=1.2mm was carried out. The result is shown in Fig.10

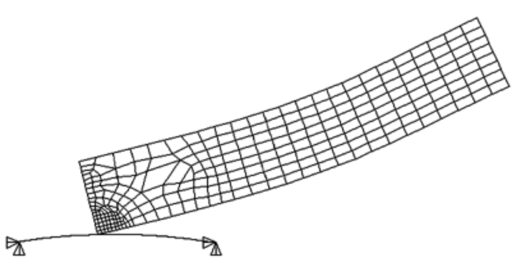

Fig.10 Analysis for Large Deformation with Interference=1.2mm

Analyzing eigenvalue for the deformed blade, we can obtain the natural frequency, the eigenvector and the reaction force of each mode. Because we assumed that the blade edge contacts onto the photoreceptor at only one point, the eigenvector and the reaction force are scalar value. Table-2 shows the results of eigenvalue analysis until the 6th mode.

Table-2 Natural Frequency, Eigenvector and Reaction Force

Mode	Natural Frequency [Hz]	Eigenvector at the Contact Point	Reaction Force at the Contact Point
1	1104.6	-1978.4	-925.9
2	1978.8	32106.0	-1173.8
3	2836.9	452.6	-4629.0
4	4430.5	9491.4	-5465.9
5	4687.5	-18053.0	-2226.4
6	6040.2	27568.0	2582.4

Since the necessary information for the stability analysis was obtained, we inspected of the eigenvalue of Eq.(11) and (12). Especially, it is important to clarify the distribution of eigenvalues in the complex plane for the variation of friction coefficient. Fig.11 shows the root locus of the eigenvalue for the variation of friction coefficient (from 0 to 1.0). Generally the real part of eigenvalue corresponds to the growth ratio of amplitude and the imaginary part corresponds to the natural angle frequency. When the eigenvalue has a positive real part, the oscillation becomes unstable[3].

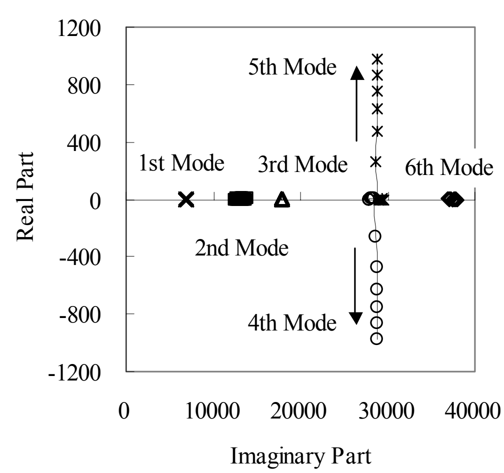

Fig.11 Root Locus of the Eigenvalue for the Variation of Friction Coefficient (0.0 $\rightarrow$ 1.0)

Fig.11 indicates that the 4th mode and 5th mode were coupled, and then the 5th mode moves to the region of a positive real part as the friction coefficient increasing. Furthermore the imaginary part of the 5th mode is 28793 [rad/sec] i.e. the natural frequency is 4582[Hz]. This is good agreement with the unforeseen noise frequency 4600 [Hz].

Because the coupled vibration model can explain destabilizing mechanism of the blade oscillation and calculate the unstable mode frequency, we can confirm the model is reasonable.

5. Conclusion

We considered the unforeseen noise of the cleaning blade in a laser printer. The self-excited vibration due to the dry friction causes this noise. However, this noise differs from a conventional one caused by the self-excited vibration in following features.

- This noise differs from the conventional one in that it occurs at high speeds when the photoreceptor is rotating steady.
- The frequency of this noise varies according to the temperature of blade.

These features indicate that the mechanism of the self-excited vibration, which causes this unforeseen noise, is distinct from the conventional one. In order to clarify the mechanism of self-excited vibration, we presented the coupled vibration model via finite degrees of freedom model and tried to formulate the phenomenon in an actual machine. Consequently, we reveal that the presented model can explain the occurrence of this noise in high velocity region and the frequency variation according to the temperature of blade.

Furthermore the numerical simulation of stability analysis via the coupled vibration model was carried out. As a result, the frequency of unstable mode was good agreement with that of the unforeseen noise. Since we confirmed the validity of the model, the condition of unstable mode was clarified using this model. Consequently, we obtained the following suggestions in order to avoid the unforeseen noise when the i th mode and j th one are coupled.

- Adjust the normal force of the blade so that γ_{ij} and γ_{ji} have the same sign in Eq.(13).
- Modify the configuration and material so that ω_i and ω_j are separated.
- Decrease the friction coefficient between the blade edge and the photoreceptor surface.

References

1. K.Nakamura, “Generation Mechanism of Unusual Noise in the Laser-Beam Printer”, Transaction of JSME-C, Vol.62, No.601, pp3428-3433, (1996).
2. H.Kawamoto, “Chatter Vibration of a Cleaner Blade in Electrophotography”, ibid, Vol.40, No.1, pp8-13, (1996).
3. L.Meirovich, “Elements of Vibration Analysis, 2nd Edition”, McGraw-Hill Book Company, (1986).
4. S.W.E. Earles, “Mechanism of Disk-Brake Squeal”, SAE paper 770181, (1977).
5. A.Nayfeh, “Nonlinear Oscillations”, Willey-Interscience Pub., (1995)
6. K.Ono et al., “Analysis of Friction-Induced Self-Excited Vibration of a Contact Recording Head Slider Using a Two-Degrees-of-Freedom Model”, Transaction of JSME-C, Vol.63, No.614, pp3343-3351, (1997).
7. K.Nagai et al., “Experiments on Modal Coupling in Selef-Excited Vibration of a Wiper-Blade”, Transaction of JSME-C, Vol.68, No.665, pp84-89, (2002).

Biography

Minoru Kasama has been concerned with the study of vibration and acoustic noise since he joined Fuji Xerox Co., Ltd. in 1991. He analyzed the image defects of various kinds that were occurred with a slight vibration on a Raster Output Scanner mirror and proposed new design parameters by a using structural analysis and modeling. He has been engaged in the search of a mechanism of the unforeseen noise of the cleaning blade due to the dry friction since 2002.