Vibration Induced in Driving Mechanism of Photoconductor Drum in Color Laser Printer

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An experimental and analytical investigation has been performed on vibration induced in a driving mechanism of a photoconductor drum in a color laser printer. The mechanism consists of a flywheel supported by a dry journal bearing and driven through a belt and a pulley that is connected to a shaft of the photoreceptor drum. From the results of the investigation, the following characteristics were deduced: (1) Pulley runout induced a vertical harmonic pull force to the flywheel through the belt and caused the vibration of the flywheel. (2) When the pull force is large enough to support the flywheel, the shaft of which is always in contact with the stator ring of the dry bearing, the harmonic vertical forced vibration takes place by the harmonic vertical force due to the eccentricity of the pulley through the belt. (3) On the other hand whirling takes place when the pull force is low. The shaft vibrates in the air gap not only in vertical but also horizontal directions. (4) The driving mechanism must be constructed to have small runout of the pulley and sufficient static tension of the belt.

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Introduction

Photoreceptor drums in color laser printers have been driven by a driving mechanism as shown in Fig. 1(a). A flywheel was connected directly to the drum in order to guarantee stable and constant speed that is important essentially to reduce banding.[1] Because the flywheel had to be large and heavy to maintain enough inertia with the low rotational speed that was common with the drum speed, this design was not suitable for compact desktop printers. Then a new driving mechanism shown in Fig. 1(b) was developed. A relatively small flywheel is driven through a belt and a pulley that is connected to a shaft of the photoreceptor drum. Because the rotational speed of the flywheel is increased, sufficient inertia is maintained even with the small flywheel, and high motion quality was realized without feedback control. Figure 2 shows a photograph of a low speed desktop color laser printer with the new driving mechanism, and Fig. 3 is the flywheel, pulley, belt, and motor of the new mechanism. A compact and low cost color laser printer was realized with this mechanism.[2,3]

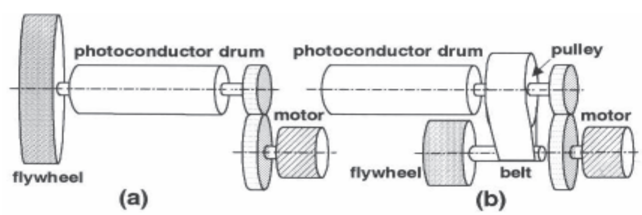

Figure 1. Driving mechanisms of photoconductor drums in color laser printers. (a) traditional design, (b) new design.

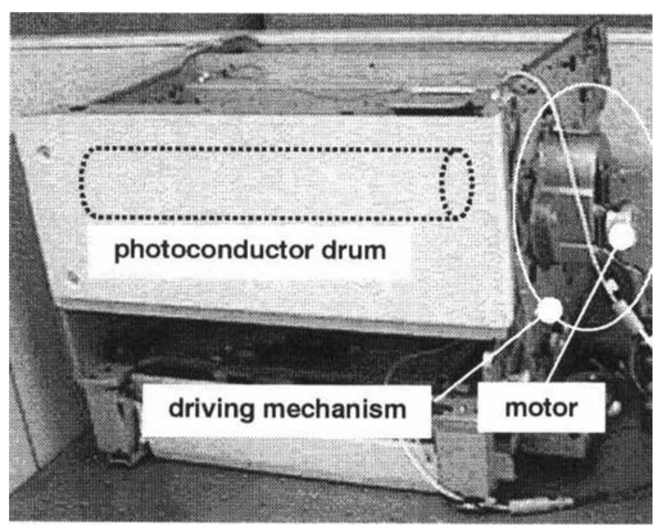

Figure 2. Photograph of low speed desktop color laser printer with new driving mechanism.

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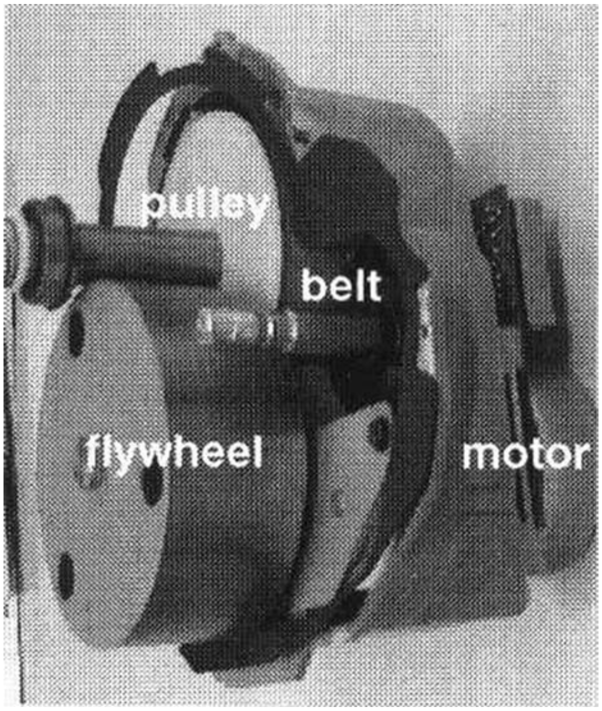

Figure 3. Photograph of flywheel, pulley, belt, and motor of new driving mechanism.

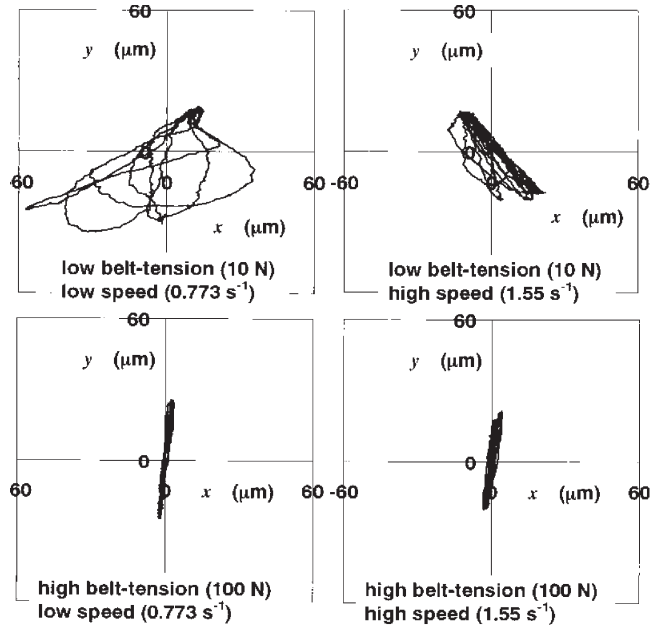

Figure 4. Measured Lissajou figures of flywheel vibrations.

The bearing system of the new flywheel is designed to be very simple, that is, only one end of a flywheel shaft made of stainless steel is inserted into a stator ring made of fluorocarbon polymer without lubricant, and the other end is free. Although this system is cost effective, chatter vibration sometimes takes place due to the dry friction between the shaft and ring. The vibration shortens the lifetime of the bearing, and if the vibration magnitude is large, it induces intolerably large acoustic noise and image defects.[2,3]

Although much literature exists on whirling in the dry bearing system with a large air gap,[4,5] there has been no literature, to the author's knowledge as of today, on the vibration of the rotor supported by the dry journal bearing and driven with the belt. The objectives of the present investigation are to clarify the vibration mechanism and to propose effective countermeasures against its cause in order to realize high quality and low cost desktop color laser printers.

Experimental

A series of experiments were performed to clarify fundamental characteristics of the vibration, and demonstrate the adequacy of the model presented in the next section. The lateral vibration of the flywheel was measured by two eddy current non-contact vibration sensors, which were attached at right angles to measure Lissajou figures of the vibration. Transducer signals were introduced to an oscilloscope and to an FFT analyzer. Figure 4 shows measured Lissajou figures as a function of the belt tension and the rotation speed of the flywheel. Here, signal noise due to the deformation of the outer surface of the flywheel was eliminated numerically and since the sensors were attached not exactly in the vertical and horizontal directions, the coordinates were numerically transformed to show that the x-axis corresponds to horizontal and the y-axis to vertical.

It was observed that although the rotor moved simply up and down in case of relatively high belt tension, whirling vibration took place with low belt tension. In any case, main frequency was not coincident with a critical frequency of the rotor-bearing system nor with the rotational frequency of the flywheel but almost equal to the rotational frequency of the pulley as shown in Fig. 5. (The critical frequency of the rotor-bearing system is about 58 – 69 Hz depending on the belt tension, the rotational frequency of the flywheel is 0.773 s^{-1} in a low speed mode and 1.55 s^{-1} in a high-speed mode, and the rotational frequency of the pulley is 0.133 s^{-1} in the low speed mode and 0.267 s^{-1} in the high-speed mode.)

Figure 6(b) shows the runout of the pulley surface to the pulley shaft, measured by a dial gauge, and Fig. 7 shows this runout and the vertical vibration of the flywheel. Although high frequency noise was included in the runout due to the experimental error, the magnitude of the vibration was the same order of the runout. From these experimental results it was found that in the case of the high belt tension, a phase of eccentricity of the pulley coincided with that of the flywheel vibration and the magnitude of the vibration was of the same order as the eccentricity. Therefore it is assumed that in the case of the high belt tension, periodic vertical force synchronized with the rotation of the pulley was applied to the flywheel and the vertical forced vibration was induced through the belt. On the other hand, the Lissajou figures of the vibration in case of the low belt tension were similar with those of the whirling vibration of a rotor supported by a dry bearing with a large air gap.[4,5] Not only the vertical but also horizontal coupled vibration was induced in this condition. That is, if the belt tension is high, the flywheel shaft is always in contact with the stator ring and the vertical forced vibration, simple up and down motion, takes place around the statically equilibrium position due to the periodic force synchronized with the rotation of the eccen-

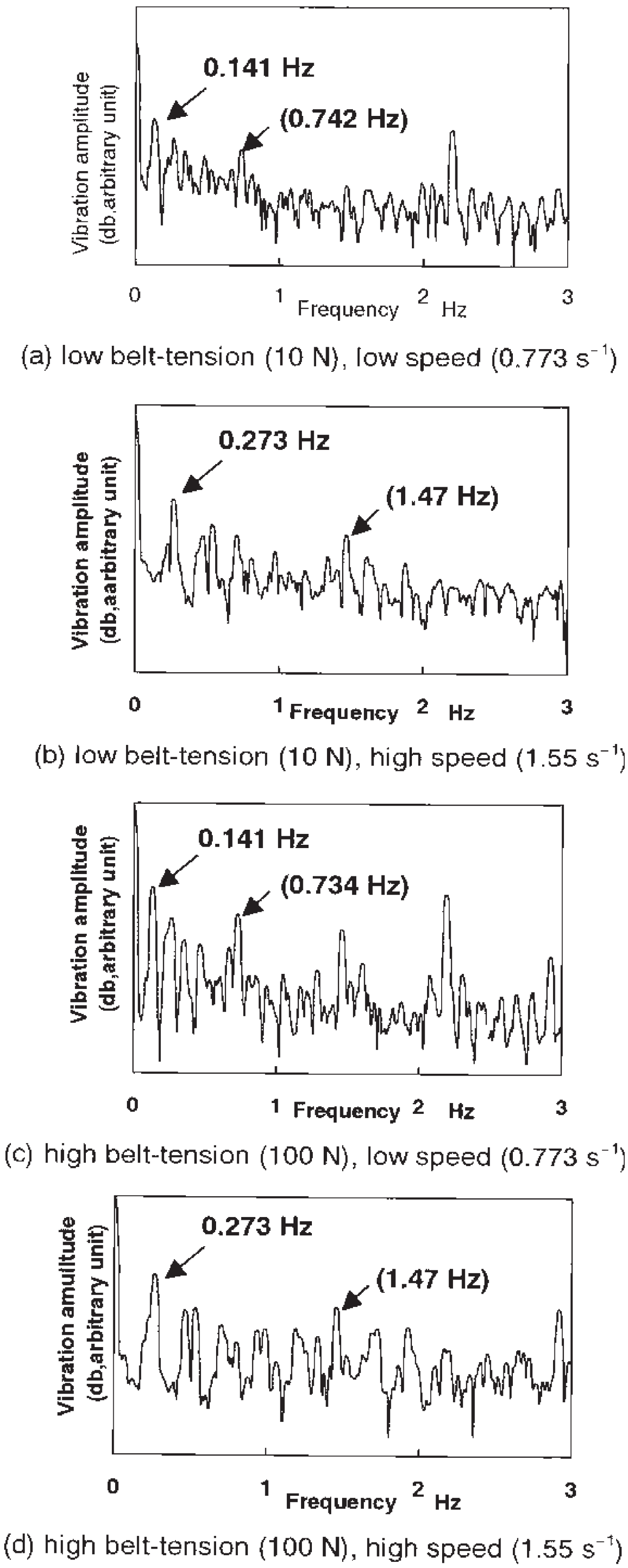

Figure 5. Measured spectrum of vibration.

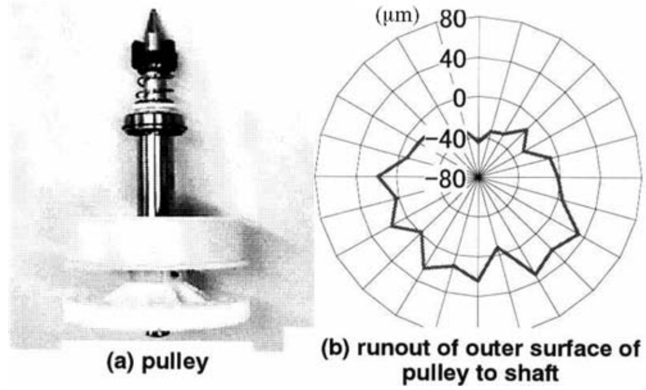

Figure 6. Photograph of pulley and measured runout of outer surface of pulley to shaft.

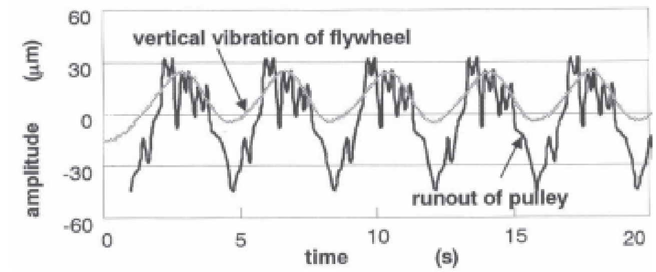

Figure 7. Vertical vibration of flywheel and runout of outer surface of pulley.

tric pulley; on the contrary, if the belt tension is low, the flywheel shaft is not always in contact with the ring and whirls in the gap.

Modeling

Based on the experimentally observed characteristics of the vibration, a vibration model shown in Fig. 8 was introduced. It is a two degree of freedom system with respect to the lateral displacement of the flywheel (x, y). It is assumed that the flywheel is pulled upward by the belt of which stiffness and viscous damping are k_2 and c_2, respectively. Vertical displacement of the belt is $Y_0 \sin\Omega\, t$, where Y_0 is a magnitude of the eccentricity, and Ω is the rotational angular velocity of the pulley. The stiffness and damping of the stator ring are k_1 and c_1, respectively. Friction force that causes coupled vibration is applied to the rotor. The the stiffness and damping of the stator ring and the friction force, are effective only when the rotor shaft is in contact with the ring. The vibration equations of the model are as follows when the rotor is in contact with the ring.

$$\ddot{x} + 2\zeta_1\omega_1\dot{x} + \omega_1^2\left(1 - \frac{\Delta}{\sqrt{x^2+y^2}}\right)x - \mu\omega_1^2\left(1 - \frac{\Delta}{\sqrt{x^2+y^2}}\right)y = 0 \tag{1}$$

$$\ddot{y} + 2(\zeta_1\omega_1 + \zeta_2\omega_2)\dot{y} + \left\{\omega_1^2\left(1 - \frac{\Delta}{\sqrt{x^2+y^2}}\right) + \omega_2^2\right\}y$$

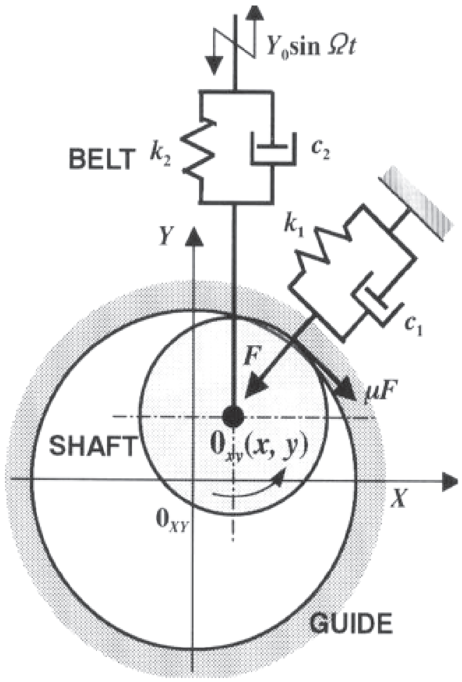

Figure 8. Vibration model of flywheel supported by journal dry bearing of gap Δ and driven by pulley through belt.

$$+\mu\omega_1^2\left(1-\frac{\Delta}{\sqrt{x^2+y^2}}\right)x=\omega_2^2\left(Y_0\sin\Omega t+S\right)-g \quad (2)$$

where $\omega_1^2 = k_1/m$, $2\zeta_1\omega_1 = c_1/m$, $\omega_2^2 = k_2/m$, $2\zeta_2\omega_2 = c_2/m$, $(\ddot{\ }) = d^2(\ )/dt^2$, $(\dot{\ }) = d(\ )/dt$ (t: time), m: mass of flywheel, μ: coefficient of friction, S: static pull, and g: gravitational constant. In the case of no contact, that is, when $\sqrt{x^2+y^2} < \Delta$, $\omega_1 = 0$, $\zeta_1 = 0$, and $\mu = 0$. The vibrations in x and y directions are coupled with each other through the friction. Although the friction coefficient between fluorocarbon polymer and steel shows a slightly positive speed dependence in the very low slip speed region, it is assumed to be constant.[5–7]

Because we are not able to derive analytical solutions of the nonlinear Eqs. (1) and (2) without neglecting the gap, numerical calculations were conducted using the Runge–Kutta method. This was done to investigate the characteristics of the vibration and the effects of all the parameters. Initial conditions are assumed to be $(x, y, \dot{x}, \dot{y}) = (0, \Delta, 0, 0)$ at $t = 0$, that is, the flywheel is pulled by the belt and it is in contact with the stator ring and starts to vibrate from the offset displacement Δ in the y-direction. Parameters for calculations were determined by separate experiments:

$\mu = 0.26$	Measured value between fluorocarbon polymer and steel
$\omega_1/2\pi = 66.4$ Hz	Measured by impact test method (Because the critical frequency of the rotor-bearing system depended on the belt tension and its value was in 58 – 69 Hz, an averaged value was employed.)
$\omega_2/2\pi = 114$ Hz	Deduced from belt stiffness measured by static tension test
$\zeta_1 = \zeta_2 = 0.0394$	Measured by impact test method
$Y_0 = 40$ μm	Measured runout of the pulley; refer to Fig. 6.
$\Omega/2\pi = 0.267$ s^{-1}	Rotational speed of the pulley

Numerical calculations were conducted with parameters of the initial pull force w (= k_2S) and the gap Δ. Calculated Lissajou figures of the flywheel vibration are shown in Fig. 9, where the diameter of the circles in Fig. 9 indicate the gap, 2Δ. The shaft of the flywheel is in contact with the stator ring when its locus is out of this circle, and it is free when the locus is inside the circle. A constant time step of the numerical integration was set to be small enough so that the numerical error due to changes of contact status is negligibly small. Figure 10 shows vibration responses of some typical cases; (a) no pull force and large gap, $w = 0$ N, Δ = 60 μm, (b) moderate pull force and moderate gap, $w = 100$ N, Δ = 40 μm, and (c) large pull force and no gap, $w = 200$ N, Δ = 0 μm.

The following characteristics are deduced from these results: (1) Main frequency of the vibration is not the critical frequency of the rotor-bearing system, $\omega_1/2\pi$, but coincided with the rotational frequency of the pulley $\Omega/2\pi$ except for the vibration in the x-direction in the case of no pull force and large gap (a). This result agrees with the experimental result. (2) The flywheel vibrates harmonically up and down, synchronized with the runout of the pulley, if sufficient pull force is applied to the rotor. Because the stiffness of the belt is almost two times larger than that of the bearing, the amplitude of the vibration is almost same with the runout. That is, if the pull force is large enough to support the flywheel, of which the shaft is always in contact with the stator ring, the harmonic vertical forced vibration takes place driven by the harmonic vertical force due to the eccentricity of the pulley through the belt. In this case the coupling vibration is small. These characteristics agree with the experimental observation in the case of high belt tension shown in the lower two cases of Fig. 4. (Whirling centers, equilibrium positions of vibrations, were centers of the measured Lissajou figures in Fig. 4, but the center of the ring is selected as the origin in Fig. 9.) (3) Whirling takes place in case of low pull force and large gap. The shaft is not always in contact with the ring and vibrates in the air gap not only in the y-direction but also in the x-direction. This also coincides with the experimental observation in the case of the low belt tension shown in the upper two cases of Fig. 4. Although the system has the potential to be dynamically unstable due to a pseudo-gyroscopic term in Eqs. (1) and (2), caused by the friction,[8] the observed and calculated vibrations are both stable on the whole. If the friction coefficient is larger and/or the damping is smaller, catastrophic whirling vibration will take place.[5–7]

Conclusions

An experimental and analytical investigation has been performed on vibration induced in a driving mechanism of a photoconductor drum in a color laser printer. The mechanism consists of a flywheel supported by a dry journal bearing and driven through a belt and a pulley that is connected to the shaft of the photoreceptor drum. From the results of the investigation, the following characteristics were deduced: Pulley runout induced a vertical harmonic pull force to the flywheel through the belt and caused the vibration of the flywheel. When the pull force is large enough to support the flywheel, the shaft of which is always in contact with the stator ring

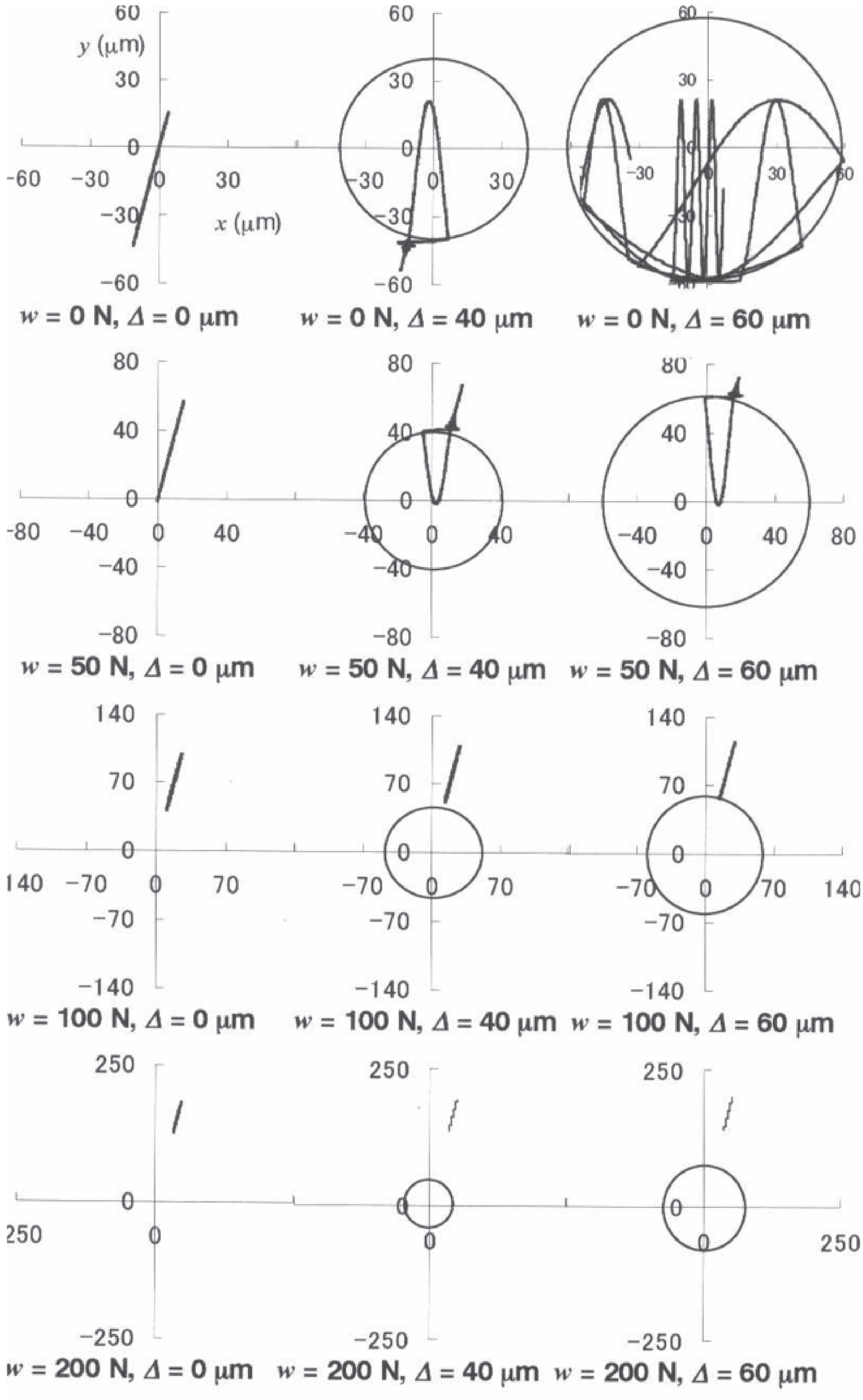

Figure 9. Calculated Lissajou figures of flywheel vibration.

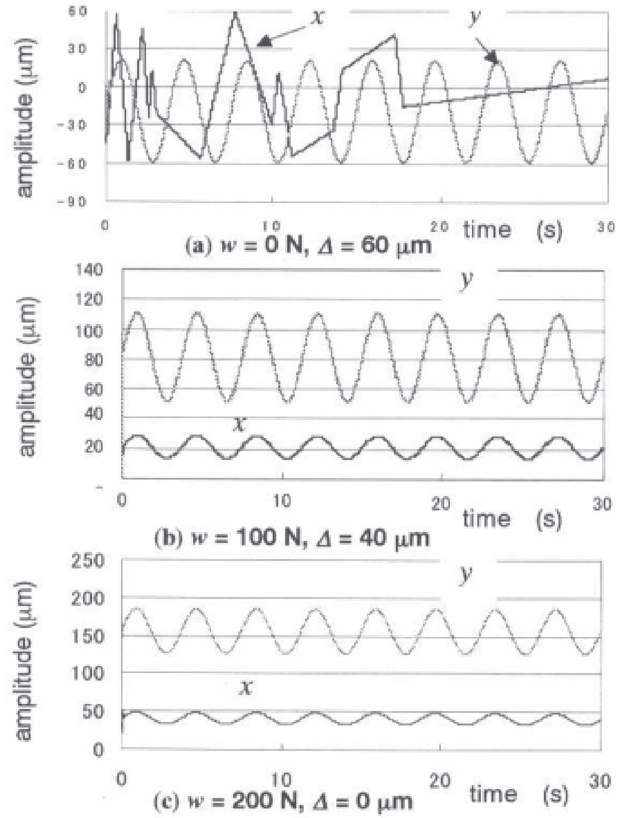

Figure 10. Calculated vibration responses of typical cases.

of the dry bearing, the harmonic vertical forced vibration takes place by the harmonic vertical force due to the eccentricity of the pulley through the belt. On the other hand whirling takes place when the pull force is low. The shaft vibrates in the air gap not only in vertical but also horizontal directions. Likewise, we deduce the following countermeasures to suppress the vibration: The driving mechanism must be constructed to have small runout of the pulley and sufficient static tension of the belt. The friction coefficient must be small and high damping must be applied to stave off catastrophic whirling vibration.

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