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THE ELECTROMAGNETIC BALANCING REGULATOR AND THE AUTOMATIC BALANCING SYSTEM*

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A new kind of automatic balancing system is presented with the electromagnetic balancing regulator. The system provides a vibration feedback and a correction mass position feedback, which results in the adoption of a new balancing method. The method first obtains the influence coefficients and unbalance vector of the rotor system, then calculates the optimum positions of the correction masses and controls them to move to the exact positions. All the operations are at working speed without interruption. Moreover, the method's feasibility and efficiency have been verified by an experiment on a model fan.

1. INTRODUCTION

Since Van de Vegte for the first time put forward the conception of on-line automatic balancing in 1964 [1], various kinds of such balancing systems have been presented [2–5]. Observing the current on-line balancing systems, we find that each of them is made up of four parts: the balanced rotor system, the sensor, the controller and the balancing regulator, among which the former two are well-known, while the regulator and the controlling method are still being researched.

The controlling method of the on-line balancing system is designed according to the structure of the balancing regulator, which, in the light of whether the position of the correction mass is able to be detected, can be divided into two groups: the optimum seeking method and the fixed phase one.

It is the optimum seeking method which is applied by Van de Vegte's balancing regulator. In his complex regulator, two correction masses are mounted on concentric gear, driven via reduction gearing by permanent-magnet DC motors. The correction mass can rotate in two directions. No provision is made to measure the positions of the correction masses, which just meets the demand of the optimum seeking method and makes its application wider and more convenient. This method may be described as "adding trial weight—comparing—deciding". Here, "adding trial weight" is to rotate the correction mass in the balancing head in arbitrary or deliberate direction: "comparing" means to compare the goal function value calculated before and after rotating the correction mass; and "deciding" is to determine whether the rotating direction of the last mass is right or not to infer the moving direction of the next one, or whether the balancing has been finished. Obviously, the method, just like "the blind climbing hill", will be time-saving if only one balancing head is adopted, while it will be time-consuming with multiple heads.

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The fixed phase method may be applied with the introduction of a mechanism which can measure the positions of the correction masses. In this case, the requirement for the correction mass being able to rotate in two directions will be reduced, that is to say, a single rotating direction is enough. In addition, the fixed phase method can be separated into the influence coefficients and the modal ones. Their common characteristics are to calculate the magnitude and phase of the needed correction weights from the previous information or through one trial, and then to directly drive the masses from their initial positions to the exact calculated ones. Here the iteration of moving of mass and comparison is unnecessary. It is appreciated that this method is time-saving, however, the application of this method for on-line balancing is still a long way off for the time being.

In this paper, a new kind of balancing regulator—the electromagnetic balancing regulator—is introduced, and then a block diagram of a balancing system with the regulator above and its balancing method are described. The test rig set-up and the experimental results are also presented.

2. THE ELECTROMAGNETIC BALANCING REGULATOR

A new kind of balancing regulator, which operates in accordance with the motor principle is proposed. Its structure is shown in Figure 1: just like a motor, it consists of a stator (2) and a rotor (4, 5, 6). In the stator, there are teeth with coils (not shown in Figure 1). Generally, the stator is fixed on the base and the slide plate seat (6) of the rotor is installed tightly on the balanced shaft and rotates with it. Simultaneously, the slide plate (5) which is of tooth structure is pressed on the slide plate seat by a pressing-board (4) which prevents the slide plate from moving axially, but leads its sliding circumferentially. However, the joint torque between the slide plate and the slide plate seat can be changed through adjusting the screws and the springs on the pressing-board. There is a hole drilled in the proper position of the slide plate, which results in a correction mass produced on the opposite side, signed with a reflection film (3). When the electromagnetic torque generated by the electric current which is fed into the coils on the stator is more powerful than the joint torque between the slide plate seat, the slide plate and the correction mass rotate circumferentially and there is an angle displacement with respect to the balanced shaft. In order to detect the position of the correction mass, a photo-electricity transistor, whose circuit is shown in Figure 2, is set up on the stator. Subject to correct utilization, as shown in Figure 3, when the reflection film rotates to the same place as that of the

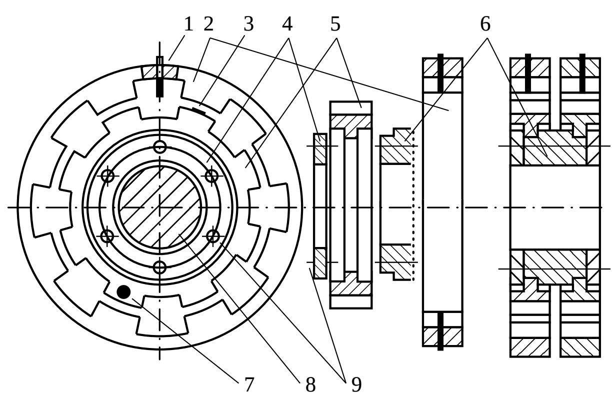

Figure 1. Design of the balancing regulator. 1: sensor for detecting the position of mass; 2: stator; 3: reflection film; 4: press board; 5: slide plate; 6: slide plate seat; 7: correction mass; 8: balanced shaft; 9: screw and spring.

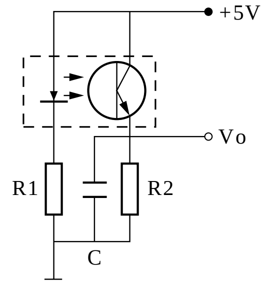

Figure 2. Circuit of photo-electricity transistor.

sensor, a pulse will be output. In this way, the position of the correction mass can be determined by comparing this pulse with the key-phase one.

Furthermore, it must be pointed out that a regulator with only one set of the mechanisms is insufficient to perform as a balancing regulator, while two or more sets disposed in parallel can work. If there is no initial unbalance on the balanced shaft, the correction masses on the slide plates have to be positioned to offset each other. For example, for a regulator with two sets, the correction masses are distributed with an interval of 180° in circumference, while the one with three sets, has an interval of 120°. When balancing is expected, the correction masses on different slide plates, driven by the electromagnetic torques which are produced correspondingly, rotate to the proper sites, and the suitable composed correction weight is prompted to set-off original unbalance. Finally the goal of on-line balancing is achieved.

Here, in order to reduce the axial size and simplify the controlling circuit, the regulator is designed to make the slide plate and the correction mass on it only able to rotate in the direction opposite to that of the balanced shaft.

3. THE SCHEMATIC OF THE BALANCING SYSTEM

Figure 4 illustrates schematically the main parts of the balancing system with the electromagnetic regulator. Through vibration and key-phase sensors, both vibration signals and key-phase signals are transmitted to a computer. The computer must detect

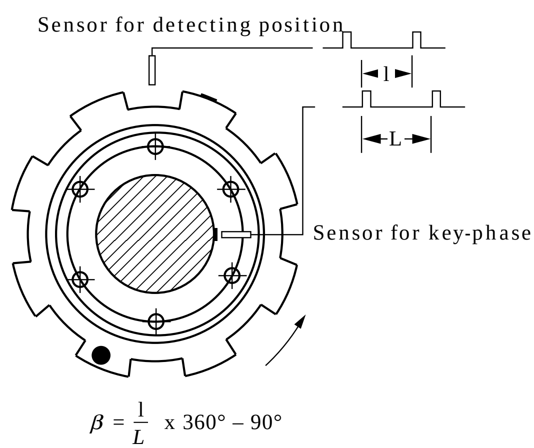

Figure 3. Sketch of measuring position of correction mass.

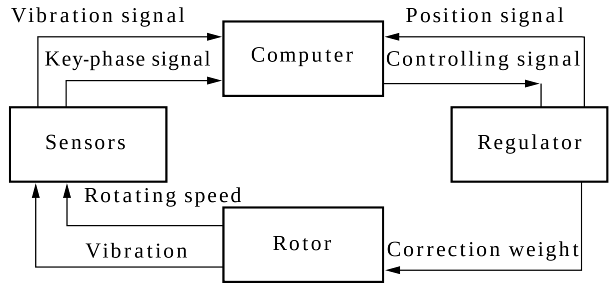

Figure 4. Block diagram of balancing system.

the rotating speed of the shaft, obtain the vibration wave-form, and band pass filter this at the rotating frequency to give synchronous amplitude and phase lag relative to the key-phase signal. When the vibration of the rotor increases to a certain extent, it must judge whether to perform balancing or not, detect the positions of the correction masses, calculate the unbalance on the rotor and determine the optimum position of each correction mass. Hereafter, further control of each correction mass from its initial position to the optimum one is also one of its functions.

A particularly attractive feature of the system is that it provides a vibration feedback and a position feedback, so that after the correction masses are driven to move a detectable angle displacement separately, their influence on vibration can be determined. This is analogous to the trial weight and to the influence coefficient calculation process of influence coefficient balancing, but avoids the need to stop the rotor for each trial weight addition. Various algorithms with the position feedback can be applied to the system, and so the balancing time will be greatly reduced.

We will now discuss when the parameters of a rotor system are unknown, how to make out the influence-coefficients, unbalance and eliminate unbalance on-line with the help of the electromagnetic balancing regulator. This is useful in practice, since usually many parameters of the rotor system are unknown before balancing and have to be determined through trial weights. How to avoid these trials is the fundamental aim of this paper.

4. THE BALANCING METHOD FOR THE ELECTROMAGNETIC BALANCING REGULATOR

In view of the fact that the majority of rotors work at a relatively constant rotating speed whether they are rigid or flexible, the on-line balancing is carried out at the same constant speed; this assumption is preserved.

Consider an isotropic rotor system with Q discs, N regulators and two bearings, as shown in Figure 5. Using only the vibration at the measurement points, the extensive motion equation can be expressed as:

$$M_{(P\times P)}X_{(P\times 1)} + C_{(P\times P)}X_{(P\times 1)} + K_{(P\times P)}\mathbf{X}_{(P\times 1)} = (B\mathbf{u} + D_{(P\times N)}\mathbf{f}_{(N\times 1)})\omega^2\,\mathrm{e}^{i\omega t} \tag{1}$$

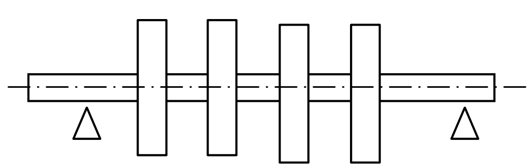

Figure 5. A rotor system.

where $\mathbf{X}$ is the extensive vibration at measurement points; M, C and K are the extensive mass, extensive damping, and extensive stiffness matrices, respectively; B is the position matrix of initial unbalance, whose element is equal to 1 when unbalance exists in this position, and 0 otherwise; $\mathbf{u}$ is the initial unbalance vector; D is the position matrix of balancing head, whose element is equal to 1 when head exists in this position and 0 otherwise; $\mathbf{f}$ is the balancing force vector; N is the number of balancing heads; and P is the number of measurement points.

In vector $\mathbf{f}$, suppose the product of weight and radius of each correction mass in the balancing regulator is U_0, there are two correction masses in a regulator and the angular positions of the two masses in the nth balancing head is β_{n1}, β_{n2} $(n = 1, 2, \ldots, N)$ respectively, then:

$$\mathbf{f} = [U_0(e^{i\beta_{11}} + e^{i\beta_{12}}), \ldots, U_0(e^{i\beta_{n1}} + e^{i\beta_{n2}}), \ldots, U_0(e^{i\beta_{N1}} + e^{i\beta_{N2}})]^T. \tag{2}$$

The balancing aim is to adjust β_{ni} $(n = 1, 2, \ldots, N; i = 1, 2)$ to minimize the goal function $\|\mathbf{X}\|$. Note that the correction mass can rotate only in a direction opposite to that of the balanced shift. If the direction opposite to that of the balance shift is positive, the rotating direction is negative and β_{ni} will only increase.

An assumption which will be maintained throughout is that the system vibration is essentially harmonic, and the transient response due to the motion of correction mass is negligible. In the virtue of the linear equation (1) with the impel forces of initial unbalance $\mathbf{u}$ and balancing force $\mathbf{f}$ on the right side, which generate vibrations $\mathbf{X}_u$ and $\mathbf{X}_f$ respectively, the whole vibration $\|\mathbf{X}\|$ is:

$$\mathbf{X} = \mathbf{X}_u + \mathbf{X}_f. \tag{3}$$

When introducing the influence coefficient matrix $A_{P \times N}$, whose element α_{ij} $(i = 1, 2, \ldots, P;\ j = 1, 2, \ldots, N)$ representing the exclusive vibration at the ith measurement point caused by unit mass in the jth balancing head, equation (3) becomes:

$$\mathbf{X} = \mathbf{X}_u + A\mathbf{f}. \tag{4}$$

In equation (4), $\mathbf{X}$ and $\mathbf{f}$ are measurable, $\mathbf{f}$ is controllable and $\mathbf{X}_u$ and $A_{P \times N}$ are unknown. What we should do is to identify $\mathbf{X}_u$ and $A_{P \times N}$ and adjust $\mathbf{f}$ to make $\mathbf{X}_f$ counterbalance $\mathbf{X}_u$.

While utilizing the electromagnetic balancing regulator, the identification of matrix $A_{P \times N}$ at a rotating speed is simplified compared with that of the general influence coefficient balancing method, in which $(N + 1)$ trial weights are expected. The procedure here is as follows: at a rotating speed, the first step is to measure the initial vibration vector $\mathbf{X}_0$ and the correction mass position vector $\mathbf{f}_0$; next move either correction mass of the first balancing head, and again measure the vibration vector $\mathbf{X}_1$ and correction mass position vector $\mathbf{f}_1$; thereafter, move either correction mass of the second head, and measure $\mathbf{X}_2$ and $\mathbf{f}_2, \ldots$. This step is repeated continuously to the Nth head.

Let

$$\mathbf{X}_i' = \mathbf{X}_i - \mathbf{X}_{i-1}, \qquad \mathbf{f}_i' = \mathbf{f}_i - \mathbf{f}_{i-1} \quad (i = 1, 2, \ldots, N)$$

then

$$[\mathbf{X}_1', \mathbf{X}_2', \ldots, \mathbf{X}_N'] = A_{P \times N} \times [\mathbf{f}_1', \mathbf{f}_2', \ldots, \mathbf{f}_N'] \tag{5}$$

because the correction masses are moved on by one, $[\mathbf{f}_1', \mathbf{f}_2', \ldots, \mathbf{f}_N']$ is a diag matrix, and its inverse matrix $[\mathbf{f}'_1, \mathbf{f}'_2, \ldots, \mathbf{f}'_N]^{-1}$ exists. The influence coefficients matrix $A_{P \times N}$ is determined by

$$A_{P \times N} = [\mathbf{X}'_1, \mathbf{X}'_2, \ldots, \mathbf{X}'_N][\mathbf{f}'_1, \mathbf{f}'_2, \ldots, \mathbf{f}'_N]^{-1}. \tag{6}$$

Substitution of equation (6) into equation (3) results in

$$\mathbf{X}_u = \mathbf{X}_0 - A\mathbf{f}_0. \tag{7}$$

The vibration caused by initial unbalance, which is constant during a process of balancing, is now determined.

$\mathbf{X}$ in equation (3) should be minimized as much as possible. In the special case when the number of measurements is equal to the number of balancing heads, $\mathbf{X}$ can be made to equal zero by solving equation (3) directly

$$\mathbf{f}_{(opt)} = A^{-1} \times (\mathbf{X}_0 - \mathbf{X}_u). \tag{8}$$

In the general case, however, where P is greater than N, $\mathbf{X}$ cannot be made to vanish. Instead, the sum S of the squares of the residual amplitudes can be minimized

$$S = \sum_{i=1}^{P} |\delta_i \cdot x_i|^2 \tag{9}$$

where δ_i are weight coefficients, which represent the importance of the measurement. S is minimized by solving the following equation, which will not be discussed here any further:

$$\frac{\partial S}{\partial \beta_{(opt)ij}} = 0 \quad (i = 1, 2, \ldots, N; j = 1, 2). \tag{10}$$

It is found that the balancing method mentioned above will finish balancing at working state without the interruption of the operation and the knowledge of the influence coefficients. The influence coefficient is a function of rotating speed, so the balanced state may be destroyed when the speed changes, especially for a flexible shaft. In this case, the rotor system will be brought back to the balancing state by repeating the procedure. After the procedure has been repeated four times, the influence coefficients at different rotating speeds are known, which can be fitted into a curve and used by later balancing processes. At this time, the controller has known something about the rotor system. So the time spent to identify the influence coefficients may be saved and the whole balancing time is reduced.

But during the identification process for mass adjustment of the nth balancing head ($n = 1, \ldots, N$), the goal function $\|\mathbf{X}\|$ may increase, because the correction masses can rotate only in one direction. In order to minimize this effect, the optimum seeking method is introduced as follows: for the nth ($n = 1, \ldots, N$) balancing head, first, move either correction mass; then if the goal function $\|\mathbf{X}\|$ decreases, move it continuously; but if the goal function $\|\mathbf{X}\|$ increases, move the other correction mass for the same angle displacement. If the increase is not weakened, the mass is moved in such a way that makes $\|\mathbf{X}\|$ increase less.

We can then control each mass to rotate to its optimum position to finish the balancing. At any time of the process, the effect of the movement of each mass in balancing heads on goal function $\|\mathbf{X}\|$ may be calculated:

$$\frac{\partial \|\mathbf{X}\|}{\partial \beta_{ij}} \quad (i = 1, 2, \ldots, N; j = 1, 2) \tag{11}$$

in equation (3). The masses which make $\|\mathbf{X}\|$ decrease are given the priority to rotate.

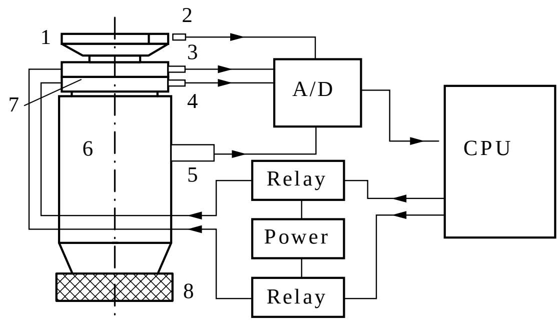

Figure 6. Sketch of test rig. 1: wheel of a fan; 2: sensor for key-phase; 3 and 4: sensors for detecting the position of mass; 5: sensor for detecting vibration; 6: motor; 7: balancing head; 8: rub pad.

5. TEST RIG

A sketch of the test rig is shown in Figure 6, which is a fan model. The balancing head is installed on the vicinity of the wheel, just at the edge of the motor.

As mentioned above, a simple controlling circuit is applied, which consists of two relays and a DC power. Each relay connects DC power with the coil on a stator. Two relays are controlled separately by a computer. When a slide plate is expected to move, a signal must be transmitted to the corresponding relay to make it switch on, then the direct current fed into the coil to drive the slide plate. Generally the signal is intermittent, so the plate is moved step by step. The angular displacement of each step depends on the sustaining time and the magnitude of the current.

Some geometrical and physical parameters of the balancing head used for the experiment are: axial size of a head = 36 mm; weight of the whole head ≈ 4·8 kg; diameter of slide plate = 100 mm; width of each slide plate = 15 mm; radius gap between stator and rotor 0·57 mm; rotating inertia of slide plate = $5{\cdot}7 \times 10^{-4}\ \mathrm{kg \cdot m^2}$; conjunction torque (shear torque) between slide plate and its seat = 1 N · m; product of weight and radius of correction mass in each slide plate = 5·8 g × 42 mm; and the weight of the motor = 79 kg. In this test rig, there is a head and a measurement, so equation (1) can be simplified as

$$mx + cx + kx = bu + dm_c \cdot r \cdot \omega^2(e^{i\beta_1} + e^{i\beta_2})\, e^{i\omega t} \tag{12}$$

or

$$x = x_u + \alpha' \cdot m_c \cdot r \cdot (e^{i\beta_1} + e^{i\beta_2})$$

where x is the vibration at the measurement point; x_u is the exclusive vibration caused by initial unbalance; β_1, β_2 are positions of masses; $m_c \cdot r$ is the product of weight and radius of correction mass; and α' is the influence coefficient.

If the product of weight and radius is merged with α', let $\alpha = \alpha' \cdot m_c \cdot \mathrm{r}$, then

$$x = x_u + \alpha \cdot (e^{i\beta_1} + e^{i\beta_2}). \tag{13}$$

Among them, x, β_1, β_2 can be measured at any time and α is unknown but can be determined by moving the correction masses.

After the unbalance is artificially added to the wheel, the system with the balancing method discussed in Section 4 is started in order to carry out balancing. The results of four experiments are shown in Figure 7 and listed in Table 1. In Figure 7, the point marked by an arrow in each figure indicates identification is finished, that is to say, the identification of the influence coefficient may be achieved by comparing the initial vibration with that of this point. The norm for selecting the point is: the amplitude or phase of

Table 1

Experiment results

	Rotating speed (rpm)	Initial positions of masses, angles β_1, β_2 (deg)		End positions of masses, angles, β_1, β_2 (deg)		Initial vibration (μm)	End vibration (μm)	Balancing time (s)	Identified influence coefficient α	Identified initial vibration x_0
1	1659	68	21	214	132	11.1	1.03	95	$3{\cdot}84e^{-130i}$	$5{\cdot}07e^{-132i}$
2	1785	266	235	117	33	12·99	0·61	70	$4{\cdot}0e^{-126i}$	$5{\cdot}3e^{132i}$
3	1845	193	195	117	45	11·81	0·91	160	$3{\cdot}86e^{-130i}$	$6{\cdot}32e^{130i}$
4	1770	205	240	105	44	12·1	1·09	180	$3{\cdot}84e^{-134i}$	$5{\cdot}81e^{132i}$

vibration of this point should change to some extent to minimize the error caused by measurement noise.

In the experiment, the balancing is thought to be finished when the correlation masses in the head rotate to the vicinity of the calculated optimum positions (the angle difference between actual position and optimum position not greater than 10°) or vibration amplitude decreases by 90 per cent.

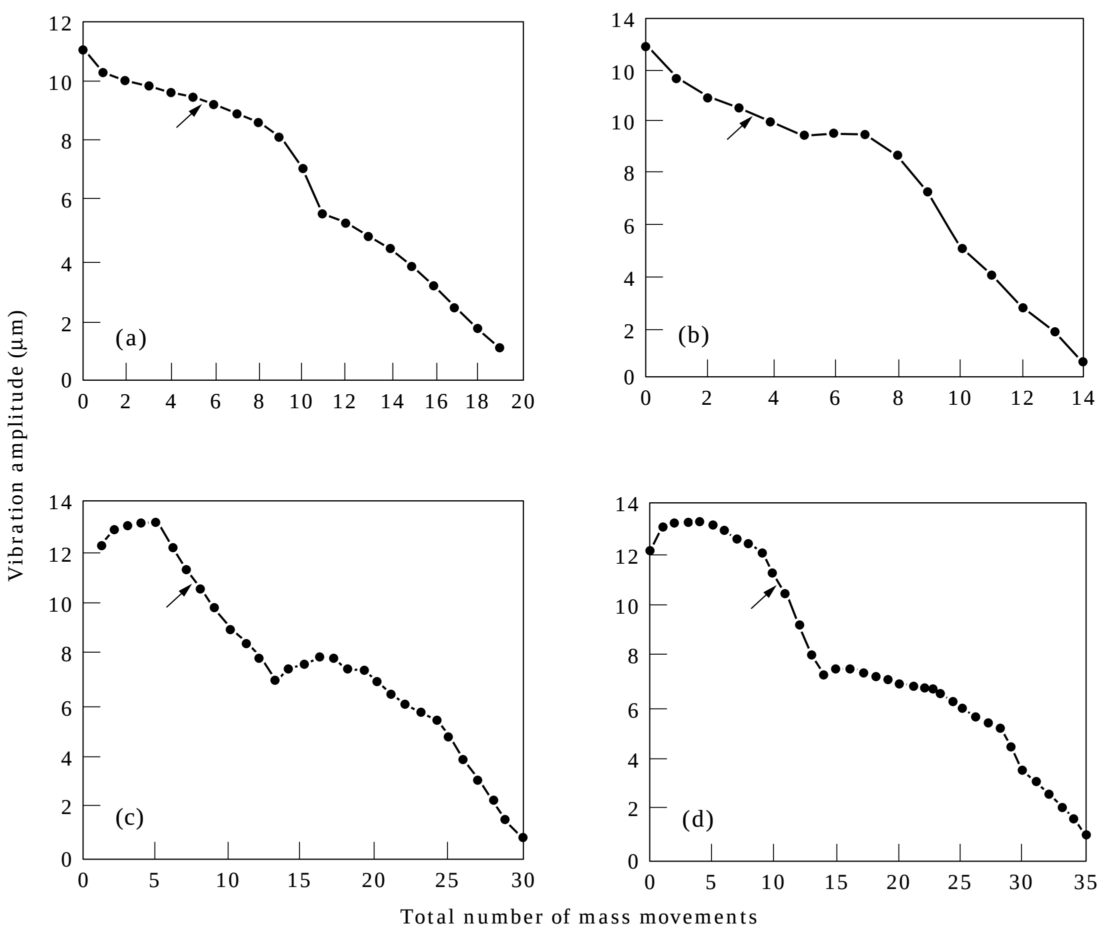

Figure 7. Vibration change during balancing process.

Because the correction mass can rotate in one direction, in some cases, for example in the third and fourth experiment, the vibration will increase for a while, then decrease to a satisfying range.

6. CONCLUSION

The balancing of a rigid rotor system can be achieved satisfactorily with the aid of the electro-magnetic balancing system which has vibration feedback and the correction mass position feedback.

The simple structure regulator can be controlled to generate the expected correction weight and the control circuit is simple and reliable. The angle displacement of the correction mass depends on the current feeding time of the coil and the magnitude of the current.

The presented balancing method can work efficiently, even when the parameters of the balanced rotor system are unknown. The current working speed range of the regulator is below 2000 rpm, and it may be improved to 3000 rpm at least.

Finally, it should be noted that this paper only applied the electromagnetic balancing regulator and its balancing method to the rigid rotor system such as the fan. Experiments on the flexible rotor will be studied later.

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