A study on electromagnetic driven bi-disc compensator for rotor auto-balancing and its movement control

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Abstract: A high-speed rotor system can be very sensitive to rotating mass unbalance which is harmful to rotating machinery. Therefore, many kinds of balancing device were created to reduce vibration in high-speed rotor system. In an electromagnetic driven balancing system, rotor unbalance is online identified and calculated by electromagnetic compensator, and then the balancing discs will be moved to the best compensating positions and the unbalance can be suppressed or eliminated. Unfortunately, industrial application using auto-balancing technology is limited by its long transition time and transient vibration enhancement. To solve these deficiencies on compensator movement control, a bi-disc freely rotation strategy was established and the solutions of two special cases were also given out. A test rig of electromagnetic driven bi-disc free rotation rotor auto-balancing system was established. Experiments using the new control strategy on this rig showed that the compensator could be moved faster and more accurate. At the meantime, the balancing time was reduced, and more important, enhancement on transient vibration was avoided.

Key words: Electromagnetic driven compensator; Balancing disc; Movement control; Rotor auto-balancing system

1. Introduction

Rotating machines such as machining spindles or turbo-machine are very commonly used in industry. One major problem faced by these machineries is vibration induced by unbalance mass. Many studies had been done on how to reduce this vibration. Rotor balancing technology has gone through three main stages: passive balancing, active balancing and auto-balancing [1, 2, 3, 4, 5]. Among them, the electromagnetic auto-balancing technology developed in the late 21th century is the most representative one. This technology can be further divided into two categories: electromagnetic force auto-balancing system (using magnetic bearings for example) and electromagnetic driven auto-balancing system (this study). The difference between them is that whether the electromagnetic force is directly used to compensate the rotor imbalance. In the first one, the electromagnetic force is directly used to compensate the vibration. While in the second one, the electromagnetic force is used to move the compensation masses to a correction distribution calculated from the measured vibration signal. In this way, a controlled correction force is generated, and the exciting force in an unbalance rotor system can be reduced by it. Therefore, the nature of electromagnetic driven auto-balancing still belongs to the mass redistribution method.

In 1964, J. Vande Vegte, Toronto University, Canada, developed the active mass redistribution balancer, in which the masses were driven by two small servos. After continuous improvement, a polar-style actuator was established in 1978, where the two adjustment masses could move along the circumference by the motor drive and generate compensation force by changing the angle between the two masses. [6,7]. In 1987, Lee and Kim of South Korea realized flexible rotor auto-balancing using radio control signals based on the works of J. Vande Vegte [8]. In 1999, Dyer of U.S designed an electromagnetic driven bi-disc balancing system for machine tool cutter [8, 9, 10, 11, 12]. In 2005, Jong-Duk Moon, South Korea, also developed an electromagnetic driven bi-disc balancing device. [13]

In China, electromagnetic driven balancing system was started at some universities and research institutes. But there were some problems in their devices. The compensator of a unidirectional electromagnetic auto-balancing device developed by Zhejiang University in 1999 could only be moved in one direction, and it was difficult to control [14, 15]; in 2001, its bi-directional solution was developed, but the mechanical structure was complicated, and the axial dimension was too large to installation. Its bearing may loosen due to the centrifugal force under high rotational speed, so it had a balancing speed limit. [16] The transition process of the electromagnetic balancing device developed in 2001 by National University of Defence Technology lasted in 184s, so its engineering application value was limited. [17, 18]

If electromagnetic driven compensator contains two or more balancing masses, there are several choices on their movement mode, and each one has different effects on the transition process. It has high research value on how to effectively avoid system short-time vibration increase during the balancing process, and minimize time consumption at the same time. In order to find the best movement control strategy, a test rig of electromagnetic driven bi-disc free rotation rotor auto-balancing system was established. Based on it, a mathematical model was given out. Simulations on the movement control of two balancing masses were finished using this model.

2. Bi-disc compensator for rotor auto-balancing

2.1 Concept of bi-disc auto balancing device

The previous studies have shown that a successful compensator should have following characteristics: low weight, small size, smooth operation even working in high-speed conditions, high accuracy and long running period. In this study, an electromagnetic driven bi-disc compensator was established which tried to satisfy all these characteristics. It achieved non-contact driving using the electromagnetic force between permanent magnet and electromagnetic coil.

This bi-disc auto-balancing system uses the spread angle balancing method. Each balancing disc contained same compensation mass which is built into the rotating part of the device. The principle of this method is shown in Fig.1. These two discs rotate freely on the rotational axis. The essence of electromagnetic driven auto-balancing is the process of moving the compensation masses to a proper place and generating the correction force which has the same magnitude but opposite phase as the unbalance force in the rotor system. In Fig.1, the compensation mass on each disc is equivalent to a compensation block.

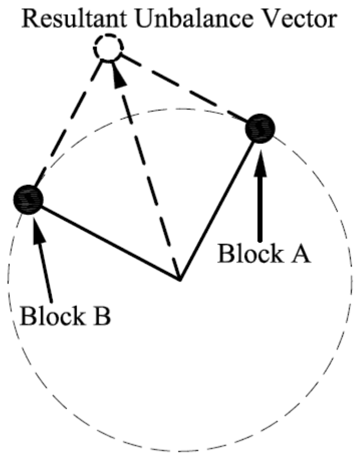


Fig.1 The generation of compensation forces

When the vibration measurement value is smaller than the set threshold, the two balancing discs will hold the last adjusted positions, and the resultant force generated by these discs is used to balance the inherent imbalance of the system; on the contrary, when the vibration amplitude is bigger than the set threshold, in order to neutralize the main vibration source, the balancing discs will be driven by the electromagnetic force to create a compensation force which is completely opposite to the rotor imbalance force. There are two special locations for balancing discs: one is the minima position, where the two balancing discs are at both ends of the same diameter and there is no compensation function; another one is the maximum position, where the two balancing discs coincide and the maximum compensation force is created.

2.2 Principle of bi-disc auto-balancing device

The construction of the whole balancing system is shown in Fig.2. The bi-disc compensator contains three main components: the balancing actuator, the vibration detector and the controller. The balancing actuator has a stator ring connected to the pedestal and a rotor ring connected to the rotator. The detector includes three sensors. One sensor is installed on the stator ring which provides speed and position information, and the other two near the rotor used to monitor the real-time vibration. If the vibration exceeds preset limits, an automatic balancing run is started and the rotor ring are moved by activation of the stator coils to the positions. These positions are calculated by the controller according to the detected unbalancing force.

Thereby, the balancing compensation is generated, and the unbalancing rotator will be balanced.

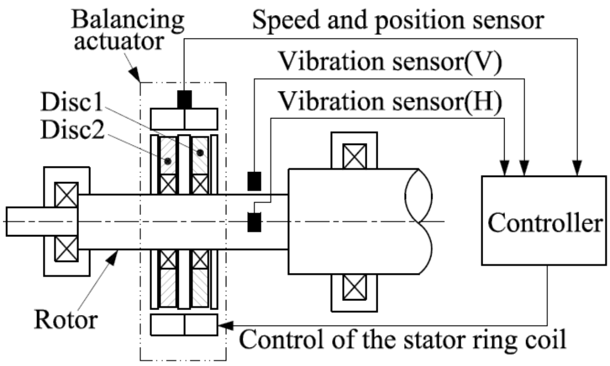


Fig.2 Components of the balancing system

2.3 The construction of electromagnetic driven bi-disc free rotation rotor auto-balancing test rig

Fig.3 is a real photo of the test rig. The balancing actuator is located in the basis and the rotor ring is settled on the rotor. There are magnetic gap between stator ring coil and rotor ring. The electromagnetic force is transferred by the gap. The vibration detector includes three sensors for vibration monitor and two sensors for discs positions monitor which installed on the stator ring coil. A PLC is used as the controller.

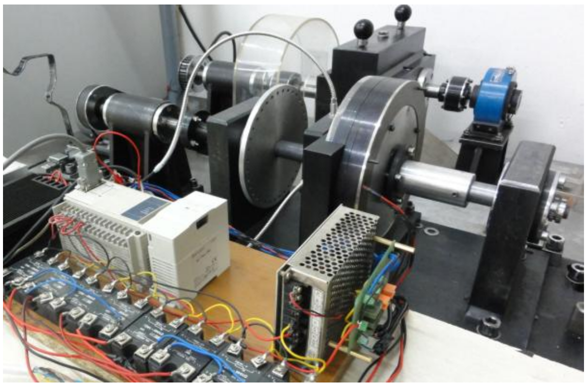


Fig.3 An electromagnetic driven bi-disc free rotation rotor auto-balancing test rig

2.3.1 The PLC controller

It is the first time that a PLC unit is used as the controller to manage the whole process of auto-balancing. The stator ring coil is controlled by proper pulses came from the PLC output. High speed pulse input channels are used for sensors. Normally, if the speed is 10,000r/min, the required frequency for speed and position sensor is:

$$f_1 = \frac{10000}{60} = 333.3 \text{ Hz}$$

The frequency required for H and V position sensors is:

$$f_2 = \frac{10000}{60} * 80 = 13.3K \text{ Hz}$$

Most Kinds of the PLC can provide 60K frequency by its high speed pulse input channels. Also, because of the widely applications and high reliability in industry, the PLC solution is more acceptable to the factories.

2.3.2 The rotor ring

The rotor ring is a core component in the electromagnetic driven auto-balancing system. This assembly rotates in synchrony with the rotator. It contains three main components: the balancing discs, the shell cover and the magnet plates, placed into a symmetric distribution, as shown in Fig.4. The movement step size of electromagnetic compensator is determined by the number of poles on balancing disc (in this design is 4.5°). Moreover, it determines the balancing absolute static error.

There are two adjustment masses settled on the balancing discs. The weight of this mass will determine the balancing capacity of the whole system based on the principle in section 2.1.

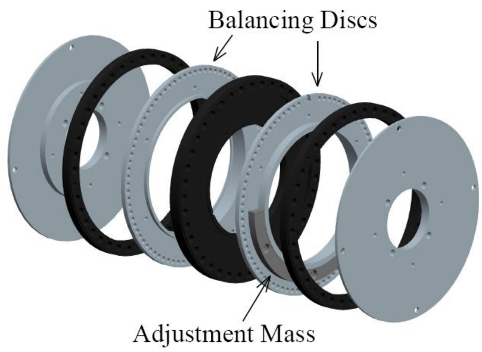


Fig.4 the structure of rotor ring

3 The driving algorithm of the bi-disc compensator

3.1 Forces Analysis and driving principle

In each control cycle, there are three kinds of forces in the bi-disc compensator system: the system's inherent imbalance force ($-\Sigma F_k$), the compensation force of two masses (ΣF_{k+1}), the system's residual unbalanced force ($-\Delta F_k$). All these forces and their causality are shown in Fig5. The first two forces are independent parameters while the residual unbalance force is the resultant force of the first two forces and rises up system's vibration; it is proportional to the vibration. In this case, auto-balancing process is trying to decrease the residual unbalanced force by adjusting the position of the balancing masses. With the movement of these masses, the compensation force and residual unbalance force are changed, only the inherent imbalance force is constant in a control cycle.

Assume that the compensator masses and system's inherent imbalance force in the same plane and the two masses have the same weight. If the system to be balanced, the two compensator masses must have two characteristics: first, both of the two masses located in the reverse direction of the inherent imbalance vector; second, they are symmetrical with the axis of inherent imbalance vector. Driving algorithm depends on the driving principle. Different principles will result in different algorithms.

During the control process, the residual imbalance force decreased in each calculation compared to the initial value is required, so the driving principle can be expressed as "the residual unbalance force showed a monotonic decline during the whole balancing process". This is important because any step which leading to the residual imbalance force increases during driving process, even though short time, is a negative impact on the rotation system. This is basic because there may be more than one solution satisfied the principle, and the best one needs be picked up. For example, in some cases the transition time may be different. If add the condition of "the shortest transition time" to the principle, the driving algorithm will be more complexity. In fact, the time for each step in the electromagnetic compensator is milliseconds; and the transition time will be several seconds for a cycle. Therefore, the deliberate pursuit of a few seconds to shorten the transition time will be not much sense for the actual demand.

3.2 Analysis of two consecutive balanced cycles

According to the above principle, compensation force for balancing is equal to the resultant force generated by composition mass carried on the balancing discs. Before moving the discs, the first thing to do is to determine the positions for each composition mass, called mass location. Then the discs will be driven to these specified positions by electromagnetic force. Since each disc is free rotation, there are multiple motion paths corresponding to every identified position. Mass movement control is required to determine the best one.

For a detailed description, a discs' motion model is established shown in Fig.5. To simplify the expression, the balancing discs are taken as two blocks of unit mass. And the rotor ring is regarded as a unit circle. A Cartesian coordinate system is established. The zero-phase is the positive X-axis, and the positive angle direction is the counter-clockwise.

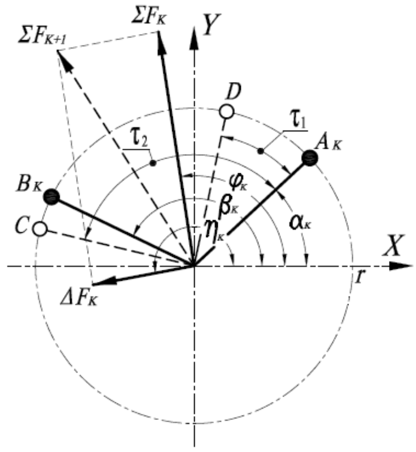


Fig.5 The motion model of balancing discs

Illustrations of graphical representation parameters in Fig.5:

A_k, B_k: positions of the two blocks after the k^{th} balancing;

C, D: positions of the two blocks after the $(k+1)^{th}$ balancing;

α_k, β_k: rotation angles of A_k and B_k under current coordinate;

ΣF_k, φ_k: amplitude and phase of resultant compensation force after the k^{th} balancing, it is the reaction force of system's inherent imbalance force during a control cycle;

ΣF_{k+1}: amplitude and phase of resultant compensation force after the $(k+1)^{th}$ balancing, it is the compensation force of two masses;

ΔF_k, η_k: amplitude and phase of new added compensation force, ΔF_k is the reaction force of system's residual unbalanced force;

τ_1, τ_2: angle between the two positions for each block in the two continuous balancing processes;

After the k^{th} adjustment, the positions of the two blocks are:

$$A_k\text{:}\quad (x_{A_k}, y_{A_k}) = re^{j\alpha_k}$$

$$B_k\text{:}\quad (x_{B_k}, y_{B_k}) = re^{j\beta_k}$$

And $\alpha_k, \beta_k \in [0, 2\pi)$ are the turn angles.

The equivalent counterbalance generated by A_k and B_k is $\sum F_k \left(x_{\Sigma F_k}, y_{\Sigma F_k}\right)$, and its corner is φ_k. Then:

$$\begin{cases} x_{\Sigma F_k} = r(\cos\alpha_k + \cos\beta_k) \\ y_{\Sigma F_k} = r(\sin\alpha_k + \sin\beta_k) \\ \varphi_k = \dfrac{\alpha_k + \beta_k}{2} \end{cases} \quad \text{Eq.1}$$

If the added unbalance after the k[th] adjustment is $\Delta f_k = \eta_k e^{j\theta_k}$, then the additional counterbalance will be:

$$\Delta F_k = \eta_k e^{j(\theta_k + \pi)} \quad \text{Eq.2}$$

And the equivalent counterbalance generated by the blocks in the (k+1)[th] adjustment will be:

$$\overrightarrow{\Sigma F_{k+1}} = \overrightarrow{\Sigma F_k} + \overrightarrow{\Delta F_k} \quad \text{Eq.3}$$

In this study, the step length of each movement is 4.5°, so the maximum error of counterweight is $e_{\max} = re^{j\frac{\pi}{40}}$.

Calculate the positions of balancing blocks after the (k+1)[th] adjustment. There are two conditions:

(1) $|\Sigma F_{k+1}| \geqslant 2r$

In this condition, the counterweight needed is out of the maximum balancing ability of the compensator, so the blocks will be overlapped at the corner of φ_k, and the residual unbalance is as follows:

$$\sigma_{k+1} = (|\Sigma F_{k+1}| - 2r)e^{j(\varphi_{k+1} - \pi)} \quad \text{Eq.4}$$

(2) $|\Sigma F_{k+1}| < 2r$

In this condition, the corner of the two blocks are α_{k+1} and β_{k+1}, then:

$$\begin{cases} \alpha_{k+1} + \beta_{k+1} = 2\varphi_{k+1} \\ \sin\alpha_{k+1} + \sin\beta_{k+1} = \sin\alpha_k + \sin\beta_k - \dfrac{\eta_k}{r}\sin\theta_k \end{cases} \quad \text{Eq.5}$$

Based on Eq.5, the positions of the blocks after the (k+1)[th] adjustment can be located. In Fig.5, they are marked as C and D.

4 Analysis of movement control for balancing discs

As illustrated above, the system's residual unbalanced force ($-\Delta F_k$) is proportional with the vibration. During the auto-balancing process, the residual unbalanced force will be decreased step by step following the principle in section 2. Therefore, a graph of the residual unbalanced force graph changing by balancing movement steps can indicate the differences between the different moving solutions.

4.1 Possible moving solutions corresponding to every identified position

There are several possible moving solutions for the blocks corresponding to every identified position. Generally, judging from the movement mode, it can be synchronous or asynchronous; judging from the movement direction, it can be identical or reverse. In the situation of asynchronous, moving control is too complicated for industrial applications, therefore, the control quality of identical synchronous moving and reverse synchronous moving will be highly focused on. As the situation in Fig.5, there were total eight different driven solutions for balancing blocks moving from points A_k, B_k to C, D.

4.1.1 Identical synchronous moving

The effects on system vibration cased by different moving solutions during the transient process were shown in Fig.6.

a) A_k counter clockwise moved to D, B_k counter clockwise moved to C: this was the best solution in this case;

b) A_k counter clockwise moved to C, B_k counter clockwise moved to D: the resultant force generated by the two blocks passed through the unbalance phase that cased the vibration enhancement in the first place and then reduced to the expectation.

c) A_k clockwise moved to D, B_k clockwise moved to C: the vibration change was the same as b).

d) A_k clockwise moved to C, B_k clockwise moved to D: the vibration change was also the same as b).

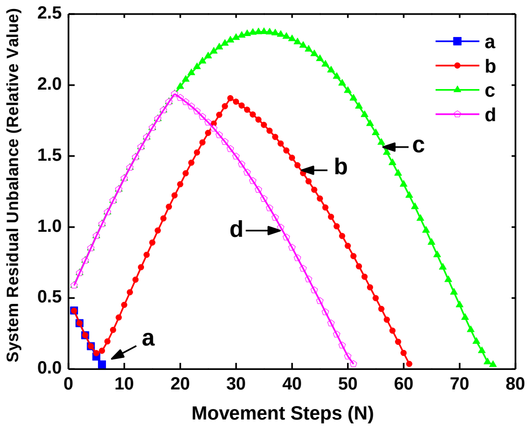


Fig.6 different vibration changes in identical synchronous moving

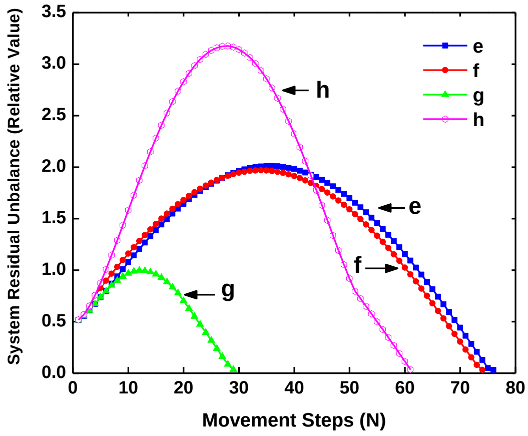


Fig.7 different vibration changes in reverse synchronous moving

4.1.2 Reverse synchronous moving

The effects on system vibration cased by different moving solutions during the transient process were shown in Fig.7.

e) A_k counter clockwise moved to D, B_k clockwise moved to C: the resultant force generated by the two blocks passed through the unbalance phase that cased the vibration enhancement in the first place and then reduced to the expectation.

f) A_k clockwise moved to D, B_k counter clockwise moved to C: not only the resultant force generated by the two blocks passed through the unbalance phase, but also the two blocks overlapped at a certain time which cased the maximum vibration enhancement. In engineering practice, this situation will be a big security risk.

g) A_k counter clockwise moved to C, B_k clockwise moved to D: there was a vibration fluctuation, and on the contrary of situation b), it was a decrease in the first place and then an increase to the expectation.

h) A_k clockwise moved to C, B_k counter clockwise moved to D: the same as situation g), a dangerous situation.

4.1.3 Conclusions on different moving solutions

As a conclusion, from the point of residual unbalance, the best moving solution was "A_k counter clockwise moved to D, B_k counter clockwise moved to C". Others cased temporally enhancement during the transient process. The worst ones leaded to a dramatically vibration enhancement which was a potential risk for the whole rotor system.

According to the driving principle: "the residual unbalance force showed a monotonic decline during the whole balancing process" which is put forward before, the best situation is that the residual unbalance after each adjustment will not be bigger than the last one during the balancing process. Based on this principle, rigorous analysis and calculation on blocks movement control were given out next.

4.2 Algorithm of blocks movement control

According to the principle and the parameters shown in Fig.8 and Fig.9, when the block's angular velocity is constant, two new variables are defined as:

$$\begin{cases} \tau_1 = \alpha_{k+1} - \alpha_k \\ \tau_2 = \beta_{k+1} - \alpha_k \end{cases} \quad \text{Eq.6}$$

Then, there are the conclusions:

(1) $|\tau_1| \leqslant |\tau_2|$

A_k will be fixed at D, and B_k will be fixed at C, Fig8. If $\alpha_k \leqslant \alpha_{k+1} \leqslant \alpha_k + \pi$, then A_k will be forward (counter clockwise) moved, otherwise A_k will be backward moved; If $\beta_k \leqslant \beta_{k+1} \leqslant \beta_k + \pi$, then B_k will be forward (counter clockwise) moved, otherwise B_k will be backward moved.

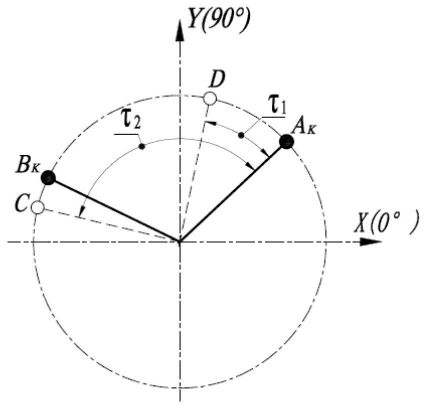


Fig.8 situation of $|\tau_1| \leqslant |\tau_2|$

(2) $|\tau_1| > |\tau_2|$

A_k will be fixed at C, and B_k will be fixed at D, Fig9. If $\alpha_k \leqslant \alpha_{k+1} \leqslant \alpha_k + \pi$, then A_k will be forward (counter clockwise) moved, otherwise A_k will be backward moved; If $\beta_k \leqslant \beta_{k+1} \leqslant \beta_k + \pi$, then B_k will be forward (counter clockwise) moved, otherwise B_k will be backward moved.

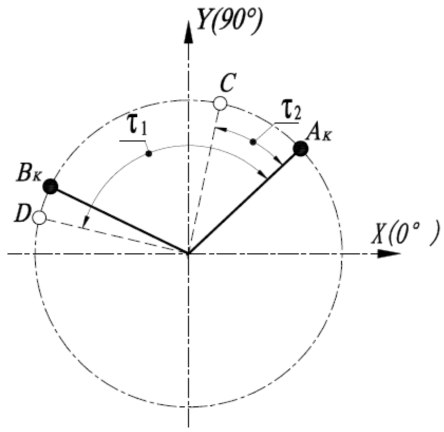


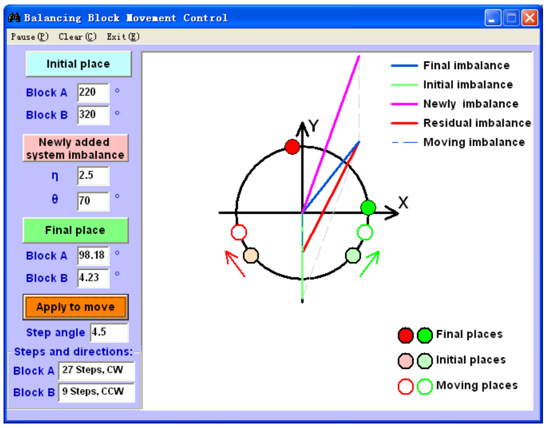
Fig.9 situation of $|\tau_1| > |\tau_2|$

5 The simulations of block movement control

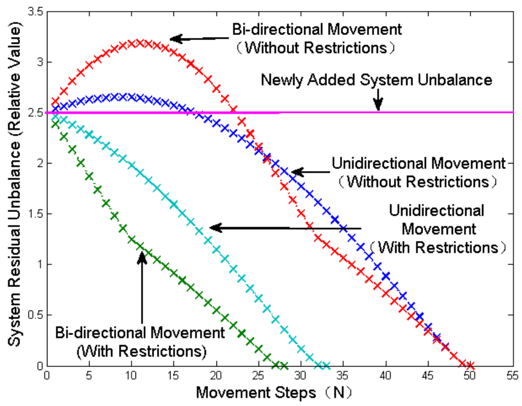
Set the counterweight created by a single block as one unit, which is used to quantify the system unbalance force. After the balancing adjustment, there may be a tiny system residual imbalance due to the existent of adjustment step. The definition of direction angle is the same as previous. The transition time of balancing adjustment depends on the maximum movement step number of the two blocks.

5.1 General situations

The simulation of general situations of balancing adjustment is shown in Fig.10. At the beginning, Block A is at 220°, and Block B is at 320°. The newly added system imbalance is $\Delta f_k = 2.5e^{j\frac{25}{18}\pi}$.



a. Balancing blocks movement



b. Comparison of different movement control

Fig.10 Simulations of balancing blocks movement control (general case)

There are two basic control schemes: unidirectional movement and bidirectional movement. In the first case, both the blocks can only be moved in one direction which was decided at the design stage; on the contrary, in the second case, each block can be freely moved in different directions at the same time. The final position is certain, so the rotor system can achieve balance eventually no matter in which way. But as shown in Fig.10 (b), there are significant differences between different movement schemes during the transient.

Without consideration of movement direction, in the unidirectional situation (clockwise), Block A was moved 47 steps and Block B was moved 49 steps; in the bidirectional situation, Block A was clockwise moved 32 steps and Block B was counter-clockwise moved 49 steps. Both solutions leaded to a worse vibration, and more seriously in bidirectional case. On the contrary, when the movement direction was carefully considered, in the unidirectional situation (counter clockwise), Block A was moved 32 steps and Block B was moved 30 steps; in the bidirectional situation, according to Eq.6, Block A was clockwise moved 27 steps and Block B was counter-clockwise moved 9 steps. This time, both solutions satisfied the movement principle, and the bidirectional one got a shorter transition time. In real auto-balancing system, the time of each step is determined by the frequency of electromagnetic pulse drive. In this simulation, the actual transition time was less than 4s.

5.2 Two special balancing cases

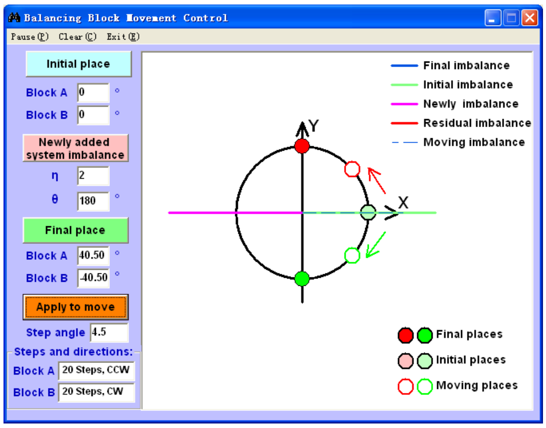
There are two special balancing cases for bi-disc electromagnetic compensator, which need to be considered separately.

(1) At time k, two discs is coincided at corner φ, and at time k+1, counterbalance needed is at corner $\varphi+\pi$ which is not smaller than the maximum balance ability.

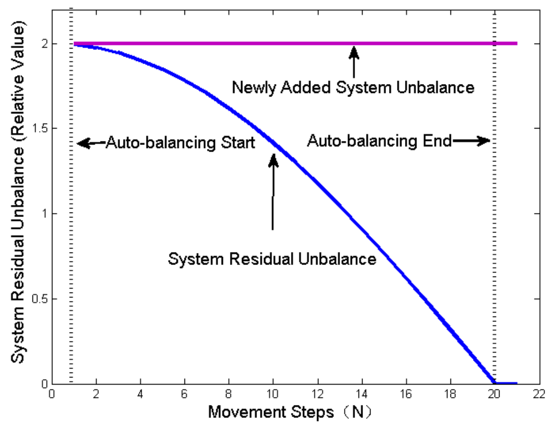
In this case, the two discs have the same rotation angle, if and only if the two be moved in different directions that the system vibration will not be enhanced. As shown in Fig.11, in the initial state, Block A and B are both at 0°, and the new unbalance force is $\Delta f_k = 2e^{0j}$, then the best movement will be that Block A is clockwise moved 20 steps and Block B is counter-clockwise moved 20 steps.

(2) At time k, rotor system is in balance, and disc A is at corner φ, disc B is at corner $\varphi+\pi$ that is the two ends of a same diameter. At time k+1, counterbalance needed is happened to be at corner φ or $\varphi+\pi$.

In this case, the system vibration will not be enhanced if and only if the two discs are moved in the same direction and not across the imbalance quadrant. As shown in Fig.12, in the initial state, Block A is at 0°, Block B is at 180°, the new unbalance force is $\Delta f_k = 0.6e^{0j}$, then the best movement will be that Block A is clockwise moved 23 steps and Block B is clockwise moved 16 steps. This situation is also the slowest balancing process for an electromagnetic compensator.

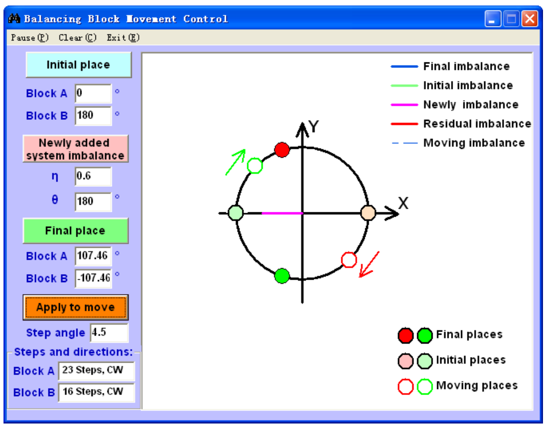


a. balancing discs movement

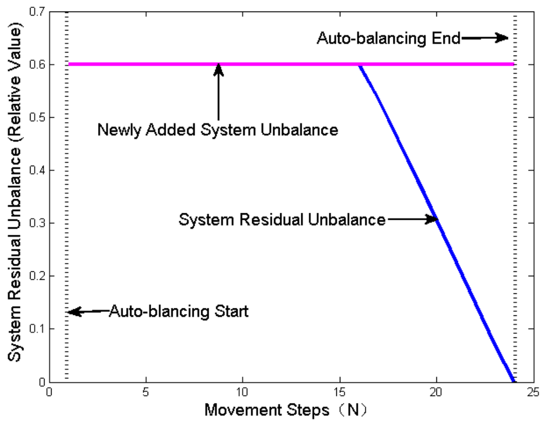


b. changes in residual imbalance force

Fig.11 Simulation of balancing blocks movement control (special cases)



a. balancing discs movement



b. changes in residual imbalance force
Fig.12 Simulation of balancing blocks movement control (special cases)

6 Conclusions

There are multi moving solutions during one control cycle of the bi-disc auto-balancing process for every identified position. One control algorithm was proposed based on movement principle in order to reduce the vibration during the balancing process. In this study, a bi-disc auto-balancing compensator was designed to confirm the algorithm.

As a new online balancing technology, there are many advantages and features in electromagnetic balancing [19, 20]. In particular, freely rotating bi-disc electromagnetic balance system has a bright application future. However, its technical difficulty lies not only in balancing algorithms, but also in the disc movement control. Multi-disc can speed up the balance process, but its possible movements also grow geometrically. Different solutions have different transition effects. It has a big chance to enhance system vibration. So the discuss of disc movement control has a very important practical significance. The principles and methods on balancing disc movement control mentioned in this paper are not only used to determine the bidirectional balancing process, but also speed up the transient process, making the electromagnetic compensator more practical.

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