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# New electromechanical balancing device for active imbalance compensation

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## Abstract

New type of electromechanical balancing device that can be used for active compensation of variable imbalance of a rotational machine is proposed. The balancing device is in a shape of ring with a number of grooves displaced around the axis of rotation. Balancing members (steel balls) are placed into the grooves, and a driving electromagnetic device is used to actively control the movement of the balancing members. The balancing members are moved between radially inner position and radially outer position of the ring. When the proposed device is attached to a rotating machine, the driving electromagnetic device is activated to generate a force on a selected balancing member, so as to displace it and to vary the balancing status. The main advantage of the proposed device is in its capability to reduce rotational imbalance in applications where the value and position of imbalance is variable. The paper presents both finite element analysis results and experimental results of prototype design.

## 1. Introduction

This paper proposes a new type of electromechanical balancing device that can be used for varying balance of any rotational machine. In particular, it relates to reducing a rotational imbalance of machines that have fixed or variable imbalance at a relatively low level.

### 1.1. Principle of balancing

Owing to uneven distribution of mass on rotational parts, machines generally encounter an imbalance when rotating [1–3]. Each part of the machine is subject to centrifugal force given by Eq. (1):

$$F_c = mR\omega^2 = I\omega^2, \tag{1}$$

where $m$ is the mass of the rotating part, $R$ the radial distance of the rotating part from the center of the rotation, $\omega$ the angular rotating speed and $I$ the imbalance.

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If the rotating parts are perfectly symmetrical and centralized, the centrifugal forces acting on all rotating parts will cancel each other and the total centrifugal force acting on machine will be zero, resulting in zero vibrations. However, any misalignment, asymmetry or imperfection will cause the total centrifugal force to be non-zero and result in vibrations. The passive compensation method is based on placing compensating mass at $180^\circ$ opposite the imbalance. The centrifugal force acting on the compensating mass will cancel out the centrifugal force caused by the imbalance of the machine.

Another option how to change the value of compensating force is to change the radial distance of the compensating mass from the center of rotation, as utilized in the balancing device proposed in this paper.

### 1.2. *Effect of imbalance*

Imbalance generates vibrations that can cause wear and damage to machine parts such as bearings, bushings, shafts and gears. In magnetic disc drives, such as hard disc drives (HDDs) and servo pack writers (SPWs), the imbalance can contribute to track misregistration and hence affect positioning accuracy of the read/write head element [4]. In addition, imbalance can result in disc slippage and excessive wear of bearings. In these applications, the rotational imbalance is undesirable, hence reduction of the imbalance becomes necessary.

Various balancing technologies have been developed to reduce rotational imbalance in order to reduce vibrations, and hence reduce wear and damage of machine parts. In a recent survey [5], various research work on active balancing methods was reviewed. Some of current active balancing methods eliminate the imbalance induced vibration using mass distribution actuators mounted on the spindle, which may not be suitable for some applications [6]. Other methods use magnetic bearings to apply synchronous force to the shaft to control the imbalance response [7]. In some applications, autobalancing of rotating machines using moving correction balls has been proposed [8].

### 1.3. *Balancing technologies for hard disks*

Many patents related to HDD balancing have been published. Conventional balancing technologies include measurement of the imbalance value and position, and adding a balancing mass onto or removing an excessive mass from the relevant area of the machine so that the imbalance can be reduced and/or compensated [9,10]. In one attempt, as disclosed in Ref. [11], a device for balancing a disc pack assembly provides imbalance correction by use of precision manufactured plugs which are attached to a spindle motor at various selected locations. Unfortunately, because the plugs are permanently fixed to the disc pack, the balancing process cannot be reversed, hence this attempt is not suitable for imbalance reduction in the event where the imbalance status is changed during operation. In addition, the cost of utilizing this approach is relatively high.

Patent [12] discloses a disc rotating device comprising of a turntable and a holding frame with an annular groove that is circumferentially partitioned to form a number of accommodating portions to accommodate steel balls. Owing to rotation of the turntable or disc, each steel ball is displaced within the accommodating portion and moves from the inner side of the accommodating portion to the outer side, thereby correcting the imbalanced state wherein a center of gravity of the disc is deviated from an axis of a rotary shaft of turntable. According to this patent, balancing balls are displaced by the centrifugal force due to the distance of deviation of the center of the gravity from the axis of rotation. However, in order for the balancing balls to move to the balancing positions to reduce imbalance, the centrifugal force must be sufficiently high. Accordingly, balancing operation in this manner has a relatively low sensitivity, and hence is not suitable for the compensation of imbalance at a relatively low level, as for example in the case of an HDD or SPW. Furthermore, it is very sensitive to speed variation and external vibrations; there is no control over how much the imbalance is reduced; it will not operate correctly in vertical position; and it works only at one pre-determined speed.

There is therefore a need to provide a balancing device and method having a high sensitivity and robustness, and which is suitable for imbalance reduction in applications where the initial imbalance value is relatively low, such as in HDDs. There is also a need to provide a balancing device and method having a relatively high sensitivity, which is suitable for imbalance reduction in applications wherein the initial value of imbalance changes frequently, such as in SPWs.

### *1.4. Advantages of the proposed electromechanical balancing device*

Main advantages of the proposed balancing device are in its high sensitivity, complete control over the balancing process, and low sensitivity to operational speed. In addition, the balancing effect can be either permanent or reversible; no power is required to maintain the balanced state; and it can operate in both vertical and horizontal position [13].

Furthermore, the device can be used in applications where intentional introduction of imbalance can be desirable.

## 2. Description of the proposed balancing device

The proposed balancing device comprises of a base body (ring) with a number of grooves displaced around the axis of rotation. Balancing members (steel balls) are placed inside the grooves, and electromagnetic device (electromagnet) is used to actively control the movement of the balancing members [13], Fig. 1. Each of the balancing members is able to move between the first (radially inner) position and the second (radially outer) position, Fig. 2.

When the device is attached to a rotating machine and rotates together in an initial state, before the balancing process, the balancing members are initially retained at the first position, Fig. 1. The balancing members are retained at the first (inner) position by the attractive force of the permanent magnet. The electromagnetic device can be controllably activated to generate electromagnetic force on a selected balancing member, so as to help to displace the selected balancing member from the first position to the second position in order to vary the balancing status.

Vibration sensor is used to determine the balancing status before and after the movement of the balancing members. One or more balancing members can be displaced by the electromagnetic device (electromagnet) until the desired imbalance reduction level is achieved. If the imbalance status of the rotating device changes, the rotational speed of the device can be reduced, so as to lower the centrifugal force acting on balancing members, and thus allow the balancing members to return to the first position. In this way, the initial state can be restored, and the balancing procedure can be repeated by displacing one or more balancing members in different positions in a similar manner.

The electromagnetic device (electromagnet) has to be able to generate a pulse of force with the duration that is approximately equal to the time a groove faces the electromagnet. This prevents displacements of a neighbouring ball.

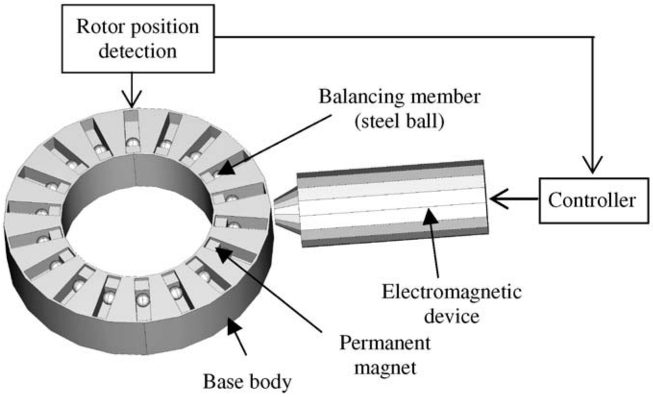

Fig. 1. Illustration of the proposed electromechanical balancing device for active imbalance compensation.

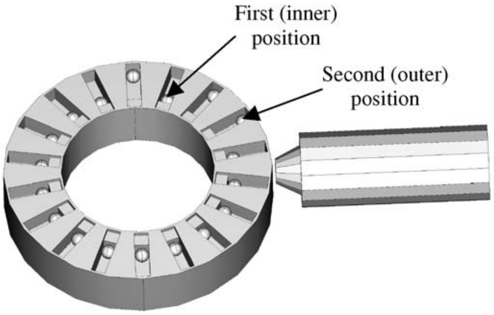

Fig. 2. Illustration of the displacement of three balls to the second (outer) position.

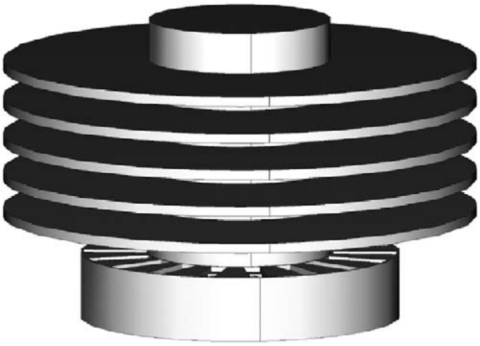

Fig. 3. Example of application of the electromechanical balancing device for active imbalance compensation in servo pack writers.

Displacement of the balls does not have to be done in sequence. For example, one ball is displaced during the first revolution, another ball is displaced one revolution later and so on. This way, the current of the electromagnet can be easily controlled in order to displace the required ball.

Fig. 3 shows an example of application of the electromechanical balancing device for active imbalance compensation in SPWs.

## 3. Principle of operation

When in the stationary state, steel ball is attached to the first position by attractive force $F_{\mathrm{pm}1}$ of permanent magnet, as shown in Fig. 4. The permanent magnet is configured such that the attractive force $F_{\mathrm{pm}1}$ applied to steel ball is large enough to overcome centrifugal force $F_{c1}$ acting on the steel ball when base body and steel balls rotate under a predetermined operational speed range. This state is referred to as "initial state". When base plate and steel balls rotate in the initial state, the total force $F_{t1}$ acting on each steel ball is given by Eq. (2):

$$F_{t1} = F_{\mathrm{pm}1} - F_{c1}. \tag{2}$$

Electromagnet is placed adjacent to the second position of grooves and does not rotate with the base plate. When supplied with electrical current, electromagnet generates an electromagnetic force $F_e$ acting on the selected steel ball.

During rotation, relative position of grooves with respect to the electromagnet can be detected through a rotor position detection sensor. The rotor position sensor is coupled to a controller that controls activation and deactivation of the electromagnet.

Upon starting of the balancing process, electromagnet is activated when one of the grooves rotates to a position in alignment with the electromagnet. Electromagnet is configured such that when activated, generated

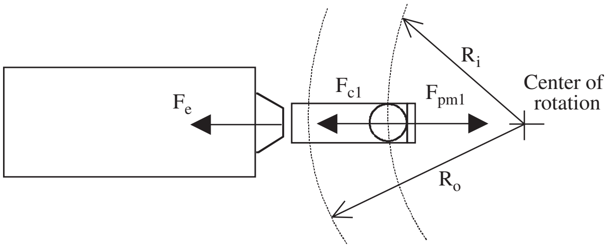

Fig. 4. Steel ball at the first (radially inner) position.

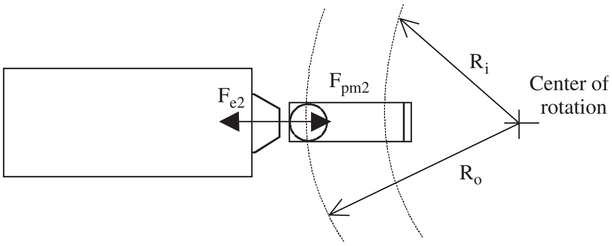

Fig. 5. Steel ball at the second (radially outer) position.

electromagnetic force $F_e$ is greater than total force $F_{t1}$. As a result, the steel ball in groove will be displaced from the first position towards the second position.

When located at the second position, centrifugal force $F_{c2}$ is acting on steel ball, Fig. 5. Since the second position is located further away from the axis of rotation than the first position, $F_{c2}$ is greater than $F_{c1}$. When ball is being displaced to the second position, centrifugal force $F_{c2}$ applied on steel ball overcomes attraction force $F_{\mathrm{pm}2}$ of permanent magnet and therefore, when electromagnet is deactivated to remove the electromagnetic force $F_e$, steel ball continues to move to the second position provided that the base body rotates under the predetermined operational speed range.

Hence, the centrifugal forces and force created by the permanent magnet have to meet conditions (3)–(5):

$$\frac{F_{c1}}{F_{c2}} = \frac{R_i}{R_o}, \tag{3}$$

$$F_{c1} < F_{\mathrm{pm}1}, \tag{4}$$

$$F_{c2} > F_{\mathrm{pm}2}. \tag{5}$$

When a steel ball is displaced from the first position to the second position, and other remaining steel balls remain at the first position, the balance status is changed. This state is referred to as “altered state”.

By comparing the balance status of the “initial state” and the “altered state”, the effect of the balancing can be determined. Further steel balls may be displaced in a similar manner until the desired imbalance compensation level is achieved.

If the imbalance status is changed after the steel balls are displaced to the second position, the rotational speed of the device can be lowered so that the centrifugal force acting on steel balls is reduced. When centrifugal force $F_{c2}$ is less than the attractive force $F_{\mathrm{pm}2}$ of permanent magnet, steel balls will return to the first position. The balancing process can then be repeated by displacing different steel balls.

In applications where the axis of rotation is vertically oriented, grooves may be tilted towards the axis of rotation, with the first position lower than the second position, so that to assist steel balls to return and attach to the respective first position.

Fixing means may be provided at the second position of the grooves to permanently or temporarily attach steel balls at the second position. For example, adhesive can be provided at the second position. In cases where axis of rotation is vertically oriented, grooves may be tilted away from the axis of rotation, that is, the first

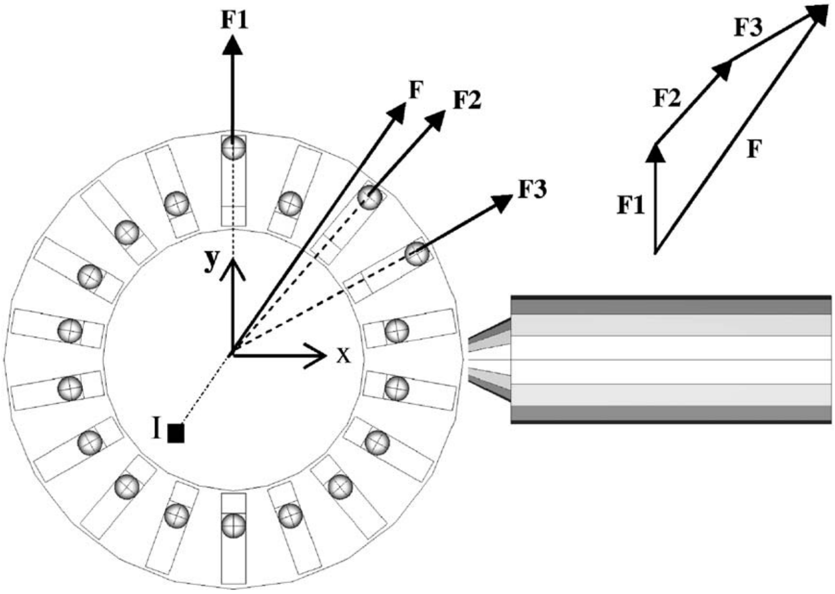

Fig. 6. Centrifugal forces $\boldsymbol{F_1}$, $\boldsymbol{F_2}$, and $\boldsymbol{F_3}$ acting on each steel ball displaced to the outer position, and resultant force $\boldsymbol{F}$ which acts opposite to, and reduces imbalance $\boldsymbol{I}$.

position is higher than the second position. Hence, when the rotational speed is reduced, the tilted configuration will assist steel balls to remain attached at the second position. Another option is that grooves have a recess or step adjacent to the second end so that when the rotational speed is reduced, the recess or step assists to retain steel balls at the second ends.

Fig. 6 shows an example, where three steel balls are displaced to the second position in order to reduce an imbalance $I$ at an angle of 236.3°. Centrifugal forces $\boldsymbol{F_1}$, $\boldsymbol{F_2}$, and $\boldsymbol{F_3}$ acting on each steel ball form a resultant force $\boldsymbol{F}$ which acts opposite to, and reduces the imbalance $\boldsymbol{I}$. The total force $\boldsymbol{F}$ in Fig. 6 is given as a vector sum of vectors $\boldsymbol{F_1}$–$\boldsymbol{F_3}$ which can be described mathematically as (6)

$$\boldsymbol{F} = \boldsymbol{F_1} + \boldsymbol{F_2} + \boldsymbol{F_3} = f\mathrm{e}^{\mathrm{j}90^\circ} + f\mathrm{e}^{\mathrm{j}50^\circ} + f\mathrm{e}^{\mathrm{j}30^\circ} = 2.722f\mathrm{e}^{\mathrm{j}56.3^\circ}, \tag{6}$$

where $f = m_b(R_o - R_i)\omega^2$ is the magnitude of the centrifugal force created by steel ball of mass $m_b$ displaced by $(R_o - R_i)$ at angular speed $\omega$.

Hence, the total imbalance compensation $I_{\text{comp}}$ at angle of 56.3° is given by Eq. (7):

$$I_{\text{comp}} = 2.722m_b(R_o - R_i). \tag{7}$$

### 3.1. Graphical representation of all possible combinations of displaced balls

In order to visualize all achievable imbalance compensations, graphical representation showing resulting imbalance compensations for all possible combinations of displaced balls can be used. For example, in Fig. 7 two balls are displaced. Ball 1 provides imbalance compensation $I_b = m_b(R_o - R_i)$ in [g mm] at angle $\alpha$, and ball 2 provides the same imbalance compensation at angle $\beta$. In total, these two balls provide total imbalance compensation given by Eq. (8):

$$\mathbf{I}_{\text{comp}} = \mathbf{I_1} + \mathbf{I_2} = I_b\mathrm{e}^{\mathrm{j}\alpha} + I_b\mathrm{e}^{\mathrm{j}\beta} = I_{\text{comp}}\mathrm{e}^{\mathrm{j}\gamma} \quad \text{where } I_{\text{comp}} = \sqrt{I_x^2 + I_y^2}. \tag{8}$$

Then, each "dot" in the figure represents the end point of the vector $\mathbf{I}_{\text{comp}}$ originating from the origin [0,0]. Subsequently, for a given maximum number of displaced balls, all possible combinations of their displacement

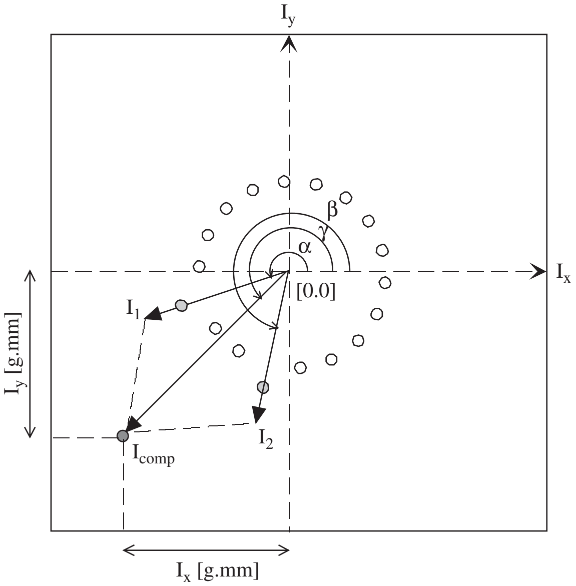

Fig. 7. Illustration used for explanation of graphical representation of all combinations of displaced balls.

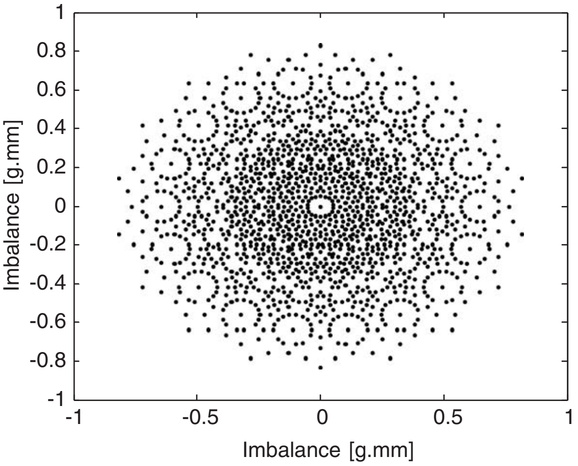

Fig. 8. Graphical representation of all possible resulting compensations if mass of the steel ball is 0.064 g, the ball is displaced by 3.5 mm, total number of balls is 18, and maximum number of displaced balls is 4.

can be found and the vectors representing resulting imbalance compensations can be calculated and drawn as "dots" [14]. Fig. 8 shows graphical representation of all possible imbalance compensations if maximum number of displaced balls is 4, i.e., 1, 2, 3 or 4 balls are displaced. If the maximum number of displaced balls is increased, for example to 6, then the number of all possible combinations of displaced balls will increase, which will result in better resolution and balancing sensitivity.

Ref. [14] proposed algorithm for imbalance compensation using a balancing ring with 24 equidistant holes that are used for placement of compensating weights. Using the algorithm, optimum position of a given maximum number of compensating weights is found in order to achieve the best possible compensation of the detected imbalance. The same algorithm can be used for the balancing device proposed in this paper, where imbalance compensation is achieved by displacing the balls instead of by placing compensating weights into equidistant holes.

Another option is to find all possible combinations of ball displacements together with corresponding resulting imbalance compensations and choose the best match.

## 4. Sensitivity analysis

To evaluate the sensitivity of the proposed balancing device, a set of 1000 uniformly distributed random initial imbalances of 0–0.8 g mm was tested. Computer algorithm [14] was used to find the best possible compensation of random initial imbalance. The algorithm was used to find all possible combinations of ball displacements together with corresponding resulting imbalance compensations and the best match for compensation of random initial imbalance was chosen. Percentile reduction of the random initial imbalance by using the best match was then calculated. In the test, displacement of 1 ball results in imbalance of 0.224 g mm. Fig. 9 illustrates percentile reduction in the magnitude of the random initial imbalance by choosing the best match out of all possible ball displacements (maximum of 1, 2, 3, or 4 balls can be displaced). As expected, larger number of displaced balls results in better reduction of imbalance. Furthermore, by using less balls, the magnitude of maximum imbalance that can be compensated is reduced, which results in poorer sensitivity in compensation of larger imbalance (Fig. 9).

Fig. 10 illustrates the effect of different number of used grooves on the sensitivity of balancing. As expected, the larger the number of used grooves the better is the reduction of the initial random imbalance. This is due to the fact that the increase in number of grooves results in larger number of possible combinations of displaced balls and hence in better sensitivity.

Based on these results it can be concluded that the sensitivity of the proposed balancing device is a function of number of grooves and maximum number of displaced balls. Reasonable sensitivity can be achieved if maximum number of displaced balls is 3 or 4 and at least 10 grooves are used. Precise determination of optimum number of displaced balls and optimum number of grooves is a complex problem due to the large number of combinations which can be obtained by displacing the balls.

The maximum value of imbalance that can be compensated is a function of the weight of the used balls and the displacement range. The displacement value is a design parameter which has to be taken into account during the design of the electromagnet in order to ensure that sufficient electromagnetic force can be generated

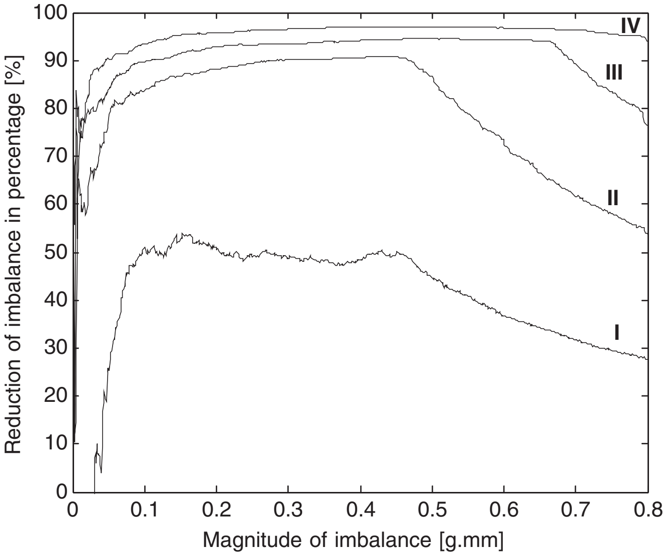

Fig. 9. Illustration of balancing sensitivity for different maximum number of displaced balls, while the number of grooves is 18 (I—1 ball is displaced, II—up to 2 balls are displaced, III—up to 3 balls are displaced, IV—up to 4 balls are displaced).

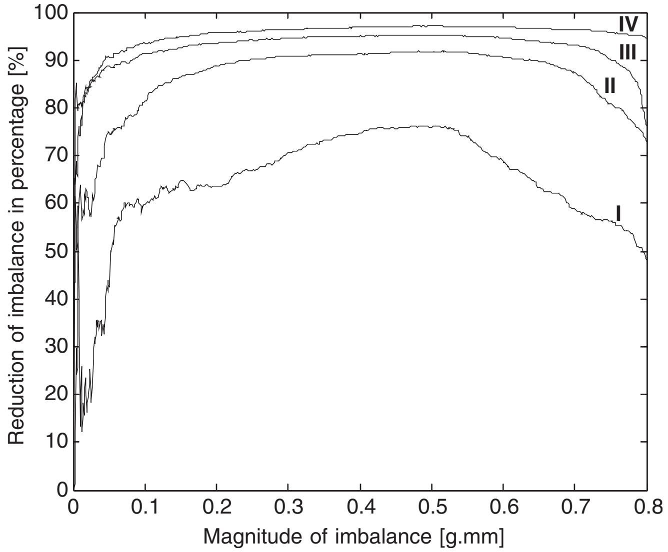

Fig. 10. Illustration of balancing sensitivity for different number of grooves, while maximum number of displaced balls is 4 (I—6 grooves, II—10 grooves, III—14 grooves, IV—18 grooves).

to help to displace the ball. The weight of the ball can be easily selected using vector analysis once the maximum number of displaced balls is decided.

## 5. Finite element analysis of the prototype

To further analyse the operation of the balancing ring a three-dimensional model was created and finite element (FE) analysis was performed. The main purpose of the FE analysis was to determine what is the variation of the total force acting on a ball for different values of displacement of the ball with respect to the permanent magnet. Parameters used for the model are given in Table 1.

Fig. 11 shows the flux density distribution obtained by simulating one quarter of the ring, and the current density of $4 \times 10^7\,\mathrm{A/m^2}$ in the coil. As it can be seen the effect of the energized coil on the neighbouring ball is negligible and hence only half of one ball, half of permanent magnet and coil were used for subsequent simulations. This simplification reduced the simulation time.

Fig. 12 shows magnetic flux lines if the current density in the coil is 0 and $4 \times 10^7\,\mathrm{A/m^2}$, respectively, and ball is displaced by 0.25 mm. Fig. 13 summarizes the FE analysis results. The $y$-axis represents the total force acting on the ball, i.e., the sum of the permanent magnet force, force created by the electromagnet and the centrifugal force at 3200 rev/min; and the $x$-axis represents displacement of the steel ball with respect to the permanent magnet. Traces 1 and 2 in Fig. 13 show the total force acting on the ball when the rotational speed is zero and the current density in coil is 0 and $4 \times 10^7\,\mathrm{A/m^2}$, respectively. Using these two traces it is possible to determine the rotational speed during the balancing process.

For trace 1, the total force acting on the ball is equal to the attractive force of the permanent magnet $F_{\mathrm{pm1}}$. The value of $F_{\mathrm{pm1}}$ can then be used to determine the maximum possible rotational speed of the balancing ring speed$_{\max}$ using (9):

$$\mathrm{speed}_{\max} = \frac{30}{\pi}\sqrt{\frac{F_{\mathrm{pm1}}}{m_b R_i}} \quad [\mathrm{rev/min}]. \tag{9}$$

If speed of the ring exceeds speed$_{\max}$ then all balls will move to the outer position of the ring since the centrifugal force will exceed the attractive force of the permanent magnet.

Table 1
Dimensions and parameters of the balancing ring and electromagnet

| | |
|---|---|
| Inner diameter of the ring | 25.8 mm |
| Outer diameter of the ring | 41 mm |
| Thickness of the ring | 8.7 mm |
| Groove (width × height) | (2.5 × 3) mm |
| Material of the ring | Aluminum |
| Magnet dimensions (width × height × length) | (2.3 × 2.6 × 1.5) mm |
| Type of magnet | SmCo ($B_r = 0.88$ T) |
| Steel ball diameter | 2.3 mm |
| Steel ball mass | 0.064 g |
| Iron core of electromagnet (diameter × length) | (8 × 30) mm |
| Permeability of iron core | 2500 |
| DC resistance of coil winding | 2.2 Ω |
| Inductance of coil winding | 0.83 mH |
| Number of turns of coil | 190 |
| Resistance of external resistor connected in series with coil | 4.7 Ω |

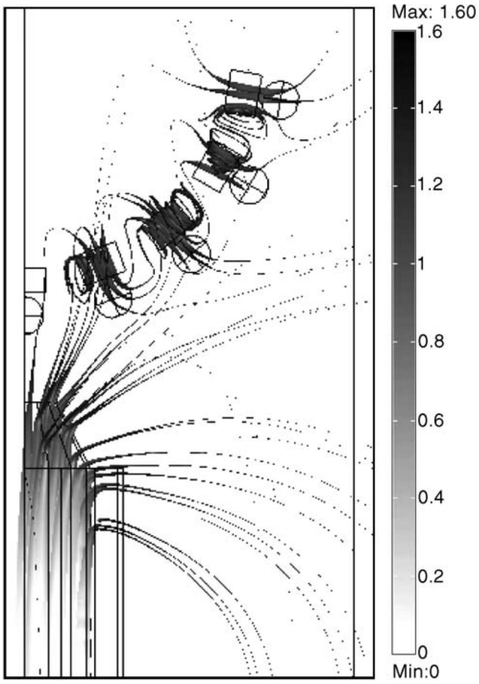

Fig. 11. Streamlines of flux density distribution if the current density in the coil is $4 \times 10^7\,\mathrm{A/m^2}$, ball is displaced by 0.25 mm, quarter of the balancing ring is simulated.

For trace 2, the total force acting on the ball is equal to the sum of attractive force of the permanent magnet $F_{\mathrm{pm1}}$ and the force $F_e$ created by the coil energized by the current density of $4 \times 10^7\,\mathrm{A/m^2}$. These values can be used to determine the minimum possible rotational speed$_{\min}$ during the balancing process using (10):

$$\mathrm{speed}_{\min} = \frac{30}{\pi}\sqrt{\frac{F_{\mathrm{pm1}} - F_e}{m_b R_i}}\ [\mathrm{rev/min}]. \tag{10}$$

If speed of the ring is lower than speed$_{\min}$ then the total force acting on the ball will not be sufficient to displace the ball. Hence, the rotational speed during the balancing process should be within the range

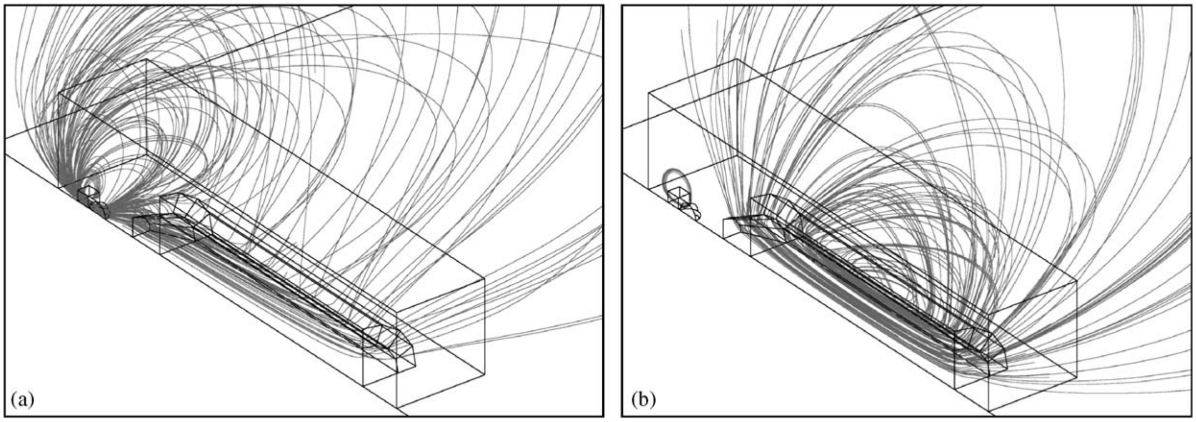

Fig. 12. Illustration of magnetic flux lines if the current density in the coil is (a) $0\,\mathrm{A/m^2}$ and (b) $4\times10^7\,\mathrm{A/m^2}$, ball is displaced by 0.25 mm.

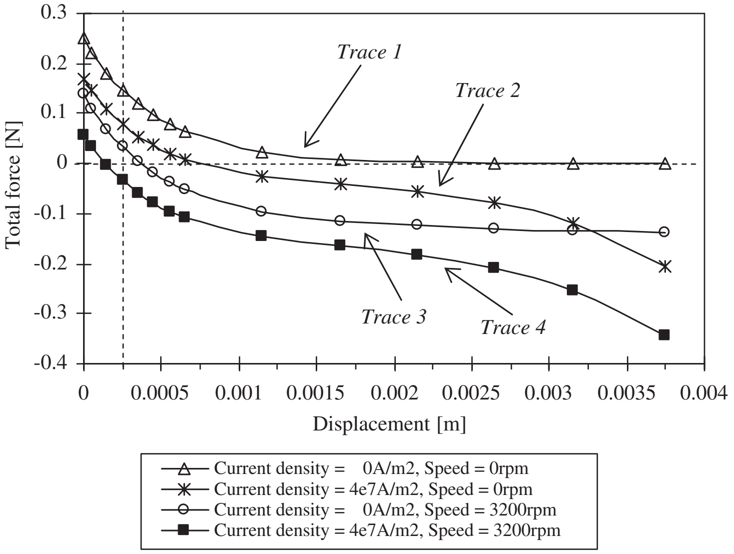

Fig. 13. Illustration of the total force acting on the ball as a function of the its displacement with respect to the permanent magnet, for the current density of 0 and $4\times10^7\,\mathrm{A/m^2}$, and for the rotational speed of 0 and 3200 rev/min.

$\mathrm{speed_{min}} < \mathrm{speed} < \mathrm{speed_{max}}$. Once the required balls are displaced, the rotational speed can be reduced below $\mathrm{speed_{min}}$, but it can never exceed $\mathrm{speed_{max}}$.

For the simplicity, the optimal balancing speed $\mathrm{speed_{bal}}$ was chosen using (11):

$$\mathrm{speed_{bal}} = \frac{\mathrm{speed_{min}} + \mathrm{speed_{max}}}{2}. \tag{11}$$

Fig. 14 shows the optimal balancing speed $\mathrm{speed_{bal}}$, minimum balancing speed $\mathrm{speed_{min}}$, and maximum balancing speed $\mathrm{speed_{max}}$ for different value of the initial displacement of the ball with respect to the permanent magnet.

Based on (11) the balancing speed was selected to be 4400 rev/min (for 0 mm displacement). However, during the experimental tests it was found that although the steel ball was displaced to the outer position at 4400 rev/min when the coil was energized, the ball was bouncing back to the inner position of the ring. This is believed to be due to the fact that the total force acting on the ball was too high and the ball rebounded from the outer side of the ring and returned back to the inner position. To reduce the force acting on the ball the

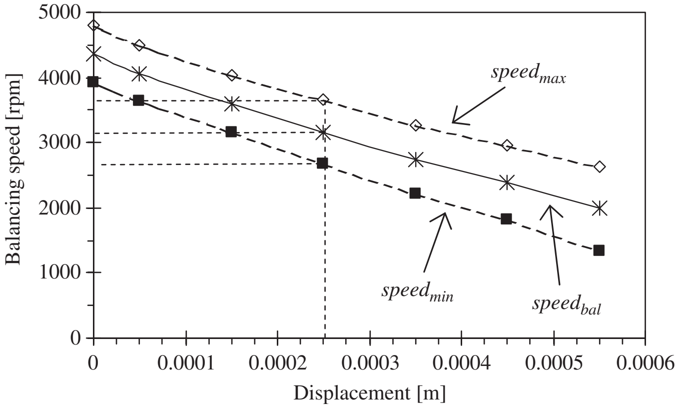

Fig. 14. Illustration of optimal balancing speed $speed_{bal}$, minimum balancing speed $speed_{min}$, and maximum balancing speed $speed_{max}$ for different value of the initial displacement of the ball with respect to the permanent magnet.

rotational speed had to be reduced. At a speed of 3200 rev/min the ball behaved as expected. However, since the balancing speed was reduced to 3200 rev/min, the ball had to be pre-displaced by 0.25 mm in order to meet the conditions for the optimal balancing speed, see Fig. 14. In experimental tests, this was achieved by inserting 0.25 mm plastic film between the ball and the permanent magnet.

Traces 3 and 4 in Fig. 13 show the total force acting on ball at the rotational speed of 3200 rev/min and for the current density of 0 and $4 \times 10^7$ A/m$^2$, respectively. As mentioned in the previous paragraph, the ball had to be pre-displaced using plastic film by 0.25 mm. As shown in Fig. 13, at the current density of 0 A/m$^2$ (Trace 3) the total force acting on the ball is positive and hence the ball is attached to the inner position of the ring. At the current density of $4 \times 10^7$ A/m$^2$ (Trace 4) the total force acting on the ball is negative and hence the ball will be displaced to the outer position.

## 6. Experimental tests of the prototype

To verify the FE results and the operational principle of the proposed balancing ring a simple prototype ring with one groove and one ball was built, see Fig. 15. Simplified prototype using one ball is sufficient to fully prove the concept, since as proved by the experimental results presented later, both the pulse duration and generated force are sufficient to displace the ball. The parameters of the ring are given in Table 1. At the rotational speed of 3200 rev/min and below, and at 0 A current in the coil, the steel ball remained attached to the inner position of the ring as shown in Fig. 16(a). At a speed of around 3800 rev/min the ball was displaced to the second position because the centrifugal force exceeded the attractive force of the permanent magnet. This result is approximately in agreement with the FE analysis results shown in Fig. 14, where $speed_{max}$ for displacement of 0.25 mm is 3655 rev/min.

When the coil was energized with a current of 3.2 A at a speed of 3200 rev/min, the ball was displaced to the second position, and remained at the second position after the current was reduced to 0 A, Fig. 16(b). The ball behaved in this manner in the speed range of 2900–3700 rev/min. At the speeds below 2900 rev/min the force created by the electromagnet was not sufficient to reduce the attractive force of the permanent magnet and the ball was not displaced. This is approximately in agreement with the results shown in Fig. 14, where $speed_{min}$ for displacement of 0.25 mm is 2667 rev/min.

### 6.1. *Evaluation of the dynamic behaviour of the electromagnet*

In order to avoid displacement of a neighbouring ball, the electromagnet has to be able to generate a pulse of force with duration that is approximately equal to the time a groove faces the electromagnet. The electromagnetive force has to be sufficient to temporarily reduce the attractive force of the permanent magnet. Once the ball is slightly displaced away from the permanent magnet, the centrifugal force acting on the ball

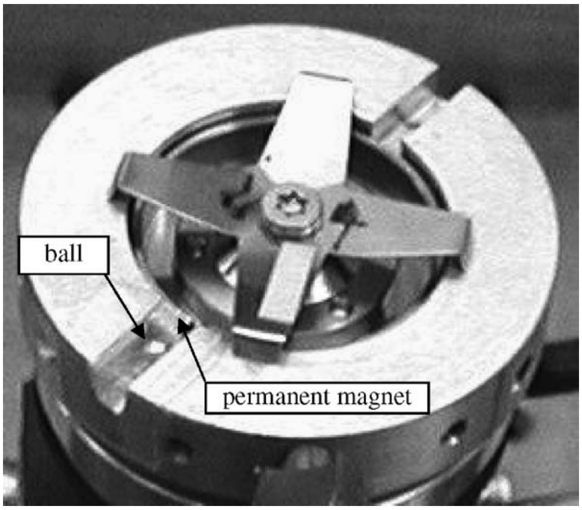

Fig. 15. Photograph of the prototype balancing ring at zero speed.

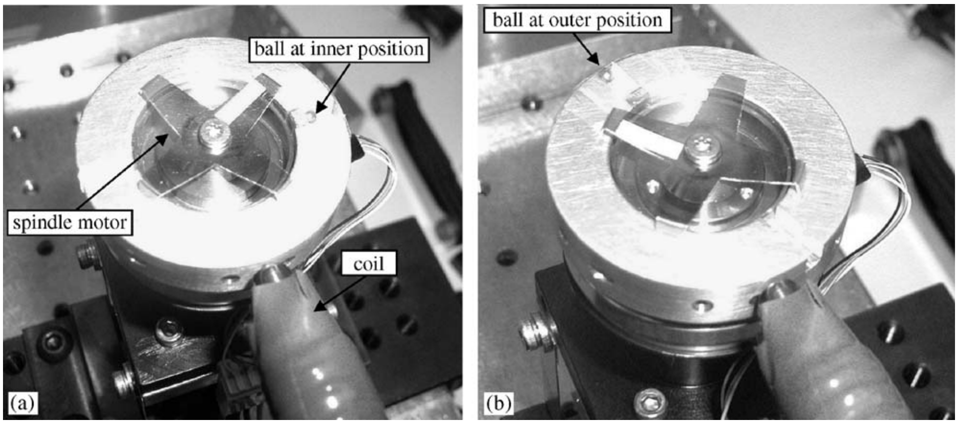

Fig. 16. Experimental results obtained at the rotational speed of 3200 rev/min: (a) steel ball at inner position, (b) steel ball displaced to outer position.

will prevail and the ball will move to the outer position. From results shown in Fig. 13 it can be found that once the ball is displaced away from the permanent magnet by only about 0.1 mm, the centrifugal force acting on the ball will prevail, and the ball will be displaced even if the electromagnetic force created by the electromagnet is reduced to zero.

Displacement of the balls does not have to be done in sequence, one ball at a time can be displaced. This way the current of the electromagnet can be easily controlled to displace the required ball.

Fig. 17 shows measured pulse shaped current in the electromagnet. To reduce the time required for the current to build up, an external resistor of 4.7 Ω was connected in series with the coil. This enabled to further reduce the time constant of the coil. As it can be seen, sufficient current can be built up in approximately 1 ms, which corresponds to the time $t_d$ one ball is facing the groove, as given by Eq. (12):

$$t_d = \frac{60}{\text{speed in rev/min} \times \text{number of grooves}} = \frac{60}{3200 \times 18} = 1\,\text{ms}. \tag{12}$$

The coil of the electromagnet can be ‘pumped’ with such a high current without any danger of overloading since the duration of the pulse is very short and a period of few hundreds or thousands of milliseconds can be

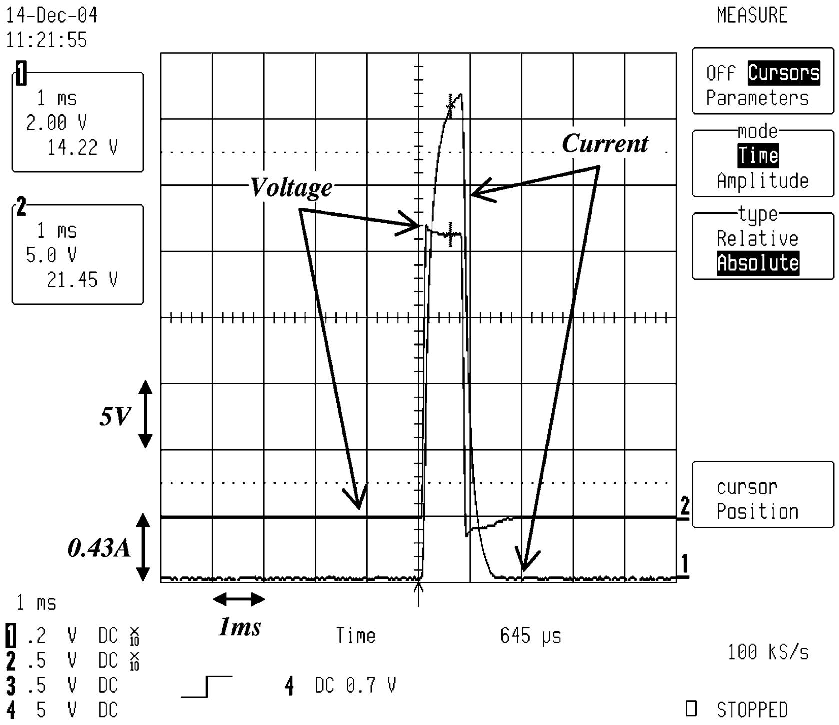

Fig. 17. Measured current in the coil and total voltage across the coil of electromagnet (external resistor of 4.7 Ω is connected in series with the coil)·

used before it is energized again to displace another ball. Parameters of the coil of the electromagnet are given in Table 1.

## 7. Conclusion

This paper proposed new type of electromechanical balancing apparatus for active imbalance compensation with many advantages over existing designs. Simple prototype was built and tested to verify the concept. Future work will focus on design of electromagnetic actuator with better performance, in order to reduce the value of excitation current and to increase the generated electromagnetic force.

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