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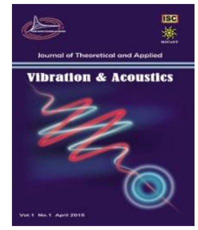

New Dynamic Balancing System based on Magnetic Interaction and Software Removal of some Perturbations

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K E Y W O R D S

Magnetic interaction

Perturbation Removal

Computational method

A B S T R A C T

The purpose of this paper is to present a new type of dynamic balancing system, having a driving solution of the rotating part based on magnetic interactions. The magnetic system also plays the role of an elastic bearing. The structure has some important advantages: an easy to design and build mechanical system; no influence of the environmental factors on the meas-uring accuracy; low production costs; no direct transmission from the driving motor to the ro-tor that has to be balanced. In the first part of the article it is presented the technical solution which allows the dynamic balancing evaluation depending on the radial displacement be-tween two disks with permanent magnets, creating a magnetic coupling. It is presented the results obtained on the experimental way, that validated both the numerical simulation as well as the analytic calculation. Using a 2D model, the resultant magnetic force was analytically calculated, whose value depends on the misalignment of the balanced part, against the equilibrium position. Due to the specific geometry, to validate the 2D analytical calculation model, it was necessary to create a FEM model of the magnetic system. A simulation was performed to evaluate the dependence between the radial displacements and the magnetic forces. It was used a 3D simulation software, specific for these kind of problems - the IN-FOLYTICA software. The final results show that there is a similarity between 2D analytical calculation model, 3D simulation and practical measurements. In the second part of the article it is presented a practical application for this type of balancing system, used to build a two plane dynamic balancing machine for card an shafts. It is also shown how it can be eliminated the disturbing unbalance introduced by the clamping system of the balanced part using a software method.

1. Introduction

Balancing is the process of attempting to improve the mass distribution of a body so that it rotates in its bearings without unbalanced centrifugal forces [1]. The motion transmission from the driving motor to the balanced rotor is always accompanied by damaging vibrations that affect the main vibration due to the rotor unbalance. Considering

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these undesired issues, new technical approach was used in order to reduce the damaging influences of external factors. An improved solution, [2], [3], consisted in an active balance of the rotors using active magnetic bearings.

In this article we propose to introduce a dynamic balancing system, based on a new principle, which uses motion transmission and bearings for the balancing part using passive magnetic interaction, without mechanical contact. No active magnetic bearings were used.

2. Technical solution

The basically sketch of the chosen technical solution is presented in Fig. 1 [4, 5].

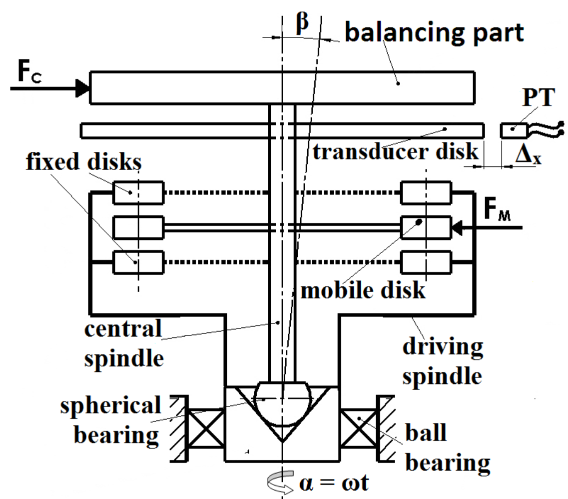


Fig 1. The basically sketch of the dynamic balancing system

The option was for a special structure with permanent magnets in radial arrangement on the surface of three concentric disks. Each disk has the same number of permanent magnets in a hetero-polar arrangement on the circumference.

The magnets are set one in front of the other, so that axial gaps occur. The mobile disk is fixed with the central spindle. This disk is between the fixed disks, which are fixed on the driving spindle. Together they achieve an axial magnetic coupling.

If the tested part, a disk subjected to balancing, has a dynamic unbalance, this creating a centrifugal force.

Due to the finite stiffness of the magnetic coupling, a displacement Δ in the radial direction will occur; the displacement Δ will increase as the centrifugal force will increase.

The magnetic system is design so as the magnetic interaction force, which occurs between the centres of the permanent magnets, will be in opposition with the centrifugal force. The radial displacement is stopped when equilibrium between the mechanical and magnetic forces occur.

3. Modeling and simulation

The magnetic forces are analytically calculated, using a 2D model, the value of these forces de-pending on the misalignment of the balanced part, against the equilibrium position. The calculation shows that the unbalance value is not dependent on its angular position, relative to permanent mag-nets arrangement on the disks circumference. Also, the magnetic force, which oppose to the centrifugal force caused by the dynamic unbalance, does not give a torque to the rotation centre of the system. Due to the specific geometry it was necessary to create a FEM model of the magnetic sys-tem to validate the 2D analytical calculation model. It was used a 3D simulation software, specific for these kind of problems - the INFOLYTICA software. The results allow the building of a numerical model of the magnetic sub-systems and confirmed the feasibility of the proposed idea**.**

3.1. The analytical calculation model

The analytical calculation model for the radial force between disks 1 and 2 as shown in Fig. 1 was approached in [6]. It has been shown that the resultant force module is the sum of each forces' module of the force between the permanents magnets pairs and it is not dependent on its angular position, relative to permanent magnets arrangement on the disks circumference. Considering n=12 magnets on each disk, the resultant force module can be expressed as in Eq. (1)

$$F_{REZ} = 2\sum_{i=1}^{12} F_i = F_1 + \ldots + F_{12} = 24\frac{B^2 \delta \pi^2}{4\mu_0} r_c \tag{1}$$

where B is the magnitude of the fundamental value of magnetic flux density, δ is the thickness of the air gap between the disks, rC is the displacement of the mobile disk.

3.2. Numerical simulation

For numerical simulation it was considered a model having 12 magnets per disk, with 15 mm diameter and 5 mm height. The magnet material is NdFeB with remnant magnetization (Br) of 1.4 T. The air gap between disks is 3 mm.

The geometry was meshed in cells up to 1.5 mm; the refining rate of the curves was 0.0218. The resulting system was solved using 3D Static mode for 41 positions, corresponding to the central shaft tilt with an angular increment of 0.0125°, resulting a final deviation of the central disk of 1.03 mm. It was used a 3D simulation software, specific for these kind of problems - the INFOLYTICA software.

3.3. Experimental validation

Experimental validation was performed for the radial force between the displacement of the disks with permanent magnets and was approached in [7] The test stand was built using a dynamic balancing machine made by ICPE-CA, which was replaced the elastic bearing with main spindle, with the new magnetic balancing system described above.

The measurements for the radial force between the disks with permanent magnets were made in a static regime, for more angular position of the displacement direction.

It was used a dial gauge type Knuth (measuring range 0-1.27mm; smallest increment 0.002mm) and a mechanical dynamometer type ADEXX FA (±1% accuracy; measuring range 0-100N). For all angular positions of the displacement direction, the same result was obtained.

Comparing the solutions for radial force (from Fig. 2) given by all the three methods (analytical calculation, numerical simulation and experimental measurements), it can be noticed a good agreement, i.e. the maximum value of the force obtained by analytical calculation was 71.35 N, the value of the force obtained by 3D numerical simulation was 66 N, while the value of the measured force on experimental stand was 64 N.

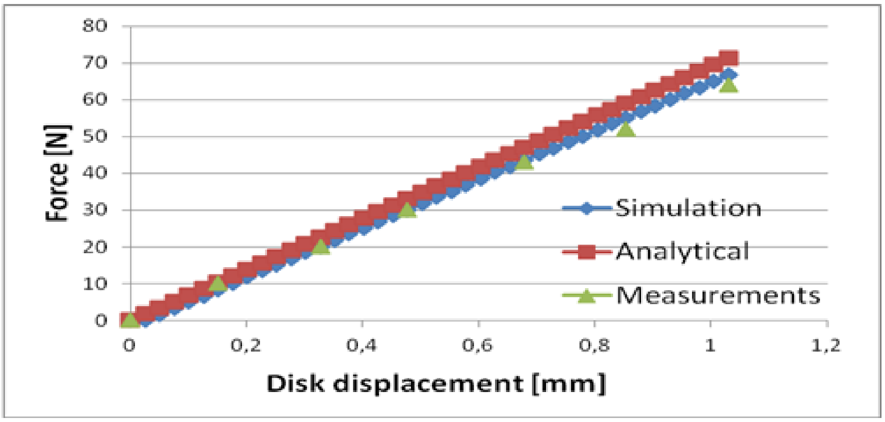


Fig 2. Force given by analytical, simulation and experimental method.

Similar to radial displacement simulation, another set of measurements were run, having as parameter the angular displacement of the mobile disk relative to vertical axis, while the radial displacement remains constant at 1.03 mm.

In Fig. 3 was plotted the value of reacting magnetic force, proportional to the centrifugal force created by the unbalance D, different angular positions. Fig. 3 shows that the unbalance value is not dependent on its angular position, relative to permanent magnets arrangement on the disks circumference, according to all three used methods (analytical calculation, numerical simulation and experimental measurements).

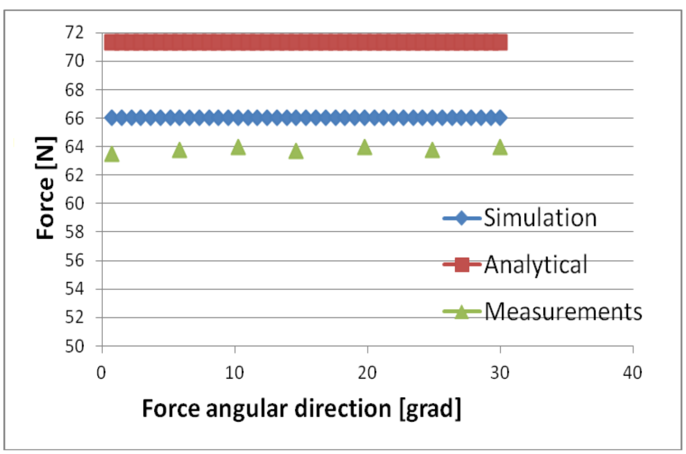


Fig 3. Force module versus angular direction

3.4. Practical applications and software elimination (removal) of some perturbations (disturbances)

The most suitable application to achieve a two planes balancing machine, using the magnetic balancing devices, is to balance the card an shafts. Fig. 4 is showing schematically such a machine.

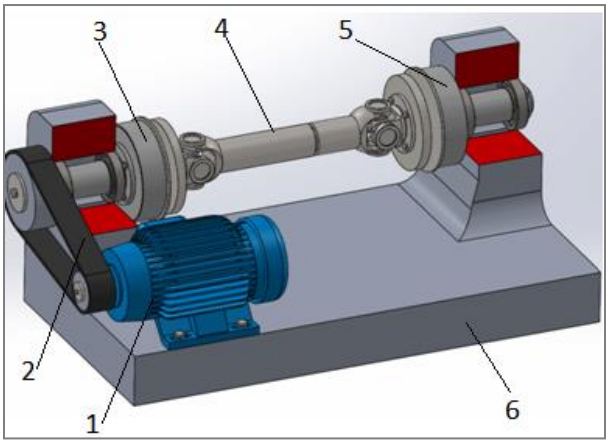


Fig 4. Balancing machine for cardan shaft

The machine is made up of two systems with magnetic interaction (3) and (5), put into uniform rotation by an electric motor (1) and a wide belt (2). When measuring the unbalance of the card an shaft 4, the measurement system will determine both the unbalance of the card an shaft 4, as well as the unbalances of the clamping system that are part of the magnetic system 3 and 5.

Due to the alignment of the clamping system made by magnetic interaction, it is difficult for them to be perfectly balanced.

The further proposed method removes the influence of the radial non-uniformity of the magnetic fields developed between the disk with permanent magnets.

In this way the deviations concentricity between fixed and mobile disks from Figure 1 will be compensated.

The removal of this disturbing unbalance will be done using a mathematical method included in a specialized software program.

This method involves removing of the unbalance vector induced in the system by fixing components. The fixing components rotate with the same angular velocity as part witch will be balanced.

From the point of view of the designer, this means clamping the balanced part in two different angular positions, with the phase difference between these also known.

The machine is launched and the first data acquisition is made.

The equations which are obtained:

$$\begin{aligned} \overline{U}_1 &= \overline{H}_{11}(\overline{D}_1 + \overline{N}_1) + \overline{H}_{12}(\overline{D}_2 + \overline{N}_2) \\ \overline{U}2 &= \overline{H}_{21}(\overline{D}_1 + \overline{N}_1) + \overline{H}_{22}(\overline{D}_2 + \overline{N}_2) \end{aligned} \tag{2}$$

where $\overline{U}_1$ and $\overline{U}_2$ is the vectorial value of the acquired signal from plane 1, respectively 2; $\overline{H}_{ij}$ is the transfer factor of the signals from unbalance in the plane i to the acquisition system for plane j: $\overline{D}_1$ and $\overline{D}_2$ is the unbalance of the balanced part in plan 1 respectively 2; $\overline{N}_1$ and $\overline{N}_2$ is the disturbing unbalance introduced by the clamping system of the balanced part in the two measuring plane.

After data acquisition that was made above, $\overline{U}_1$ and $\overline{U}_2$, stops the balancing machine, and the balanced part is rotate with a 1800 relative to the clamping system.

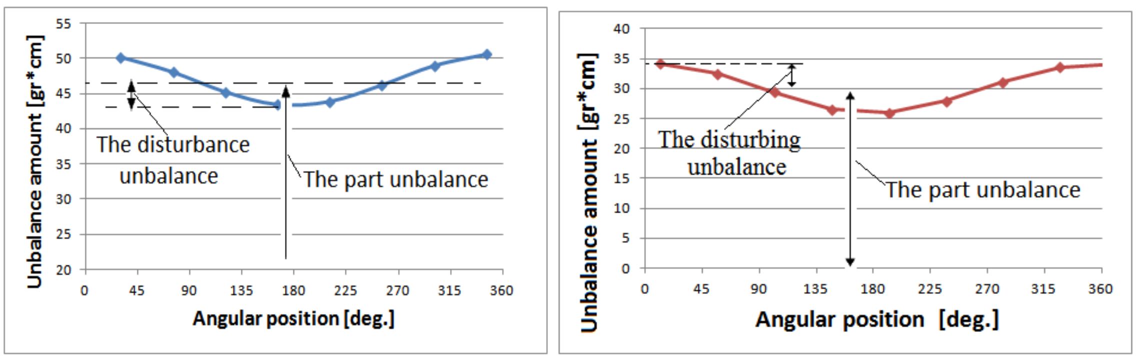
The second launch is made. The second equations data, $\overline{W}_1$ and $\overline{W}_2$, are obtained - Eq. (3).

$$\begin{aligned}\overline{W}_1 &= \overline{H}_{11}(\overline{D}_1 - \overline{N}_1) + \overline{H}_{12}(\overline{D}_2 - \overline{N}_2)\\ \overline{W}_2 &= \overline{H}_{21}(\overline{D}_1 - \overline{N}_1) + \overline{H}_{22}(\overline{D}_2 - \overline{N}_2)\end{aligned} \tag{3}$$

It is solved the mathematical system of four equations with the four unknowns in relation to the disturbing imbalance $\overline{N}_1$ and $\overline{N}_2$. The system solutions will be used as a correction term for all subsequent measurements.

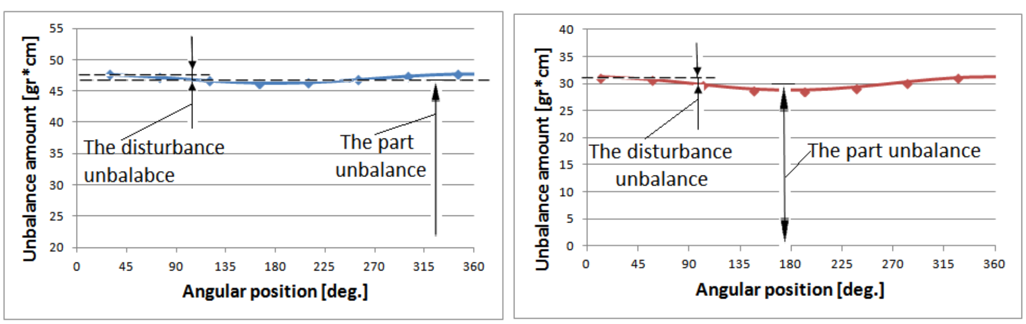
Because the balancing machine uses two balanced devices with the magnetic interaction, this procedure must be performed for each system, and then shall be deducted the unbalance vectors, applying the superposition principle.

There have been obtained the following results, shown in Fig. 5 and Fig. 6.



Disturbance unbalance for the right plane Disturbance unbalance for the left plane

Fig 5. The unbalance distribution before software correction



Disturbance unbalance for the right plane Disturbance unbalance for the left plane

Fig 6. The unbalance distribution after software correction.

The measured parts were fixed in 8 different positions to 45 degrees, making the measurement in each position, both for the plan 1 and for plan 2.

For the measurements without correction software, the displayed unbalance level was between 34.2 gr.*cm. and 25.8 gr.* cm., for plane 1 and between 50.7 gr.*cm. and 43.3 gr.*cm. for plane 2.

For the measurements with implemented correction software, the displayed unbalance level was between 31.3 gr.*cm. and 28.7 gr.*cm., for plane 1 and between 57.8 gr.*cm. and 46.2 gr.*cm. for plane 2.

From measurements it can be seen that the disturbing unbalance decreased from 4.2 gr.*cm to 1.3 gr.*cm. for plane 1 and from 3.7 gr.*cm. to 0.8 gr*cm. for plane 2.

Although the disturbing unbalance was not completely eliminated, the method can be applied successfully to achieve high accuracy measurement.

4. Conclusions

The new balancing system is viable, the force value obtained by analytical calculation and by numerical simulation are in good agreement with curves obtain by measurement on the test stand.

The values of the force obtained at end of the displacement range by the all three methods (analytical, numerical, measurements) are in a confidence limit of ~10 %.

This proves that the analytical equations and numerical model are adequate to use in design of such systems. It is obvious that the removal of the disturbances introduced by clamping systems is much cheaper to be made by software method compared with the use of the high precision mechanical machining.

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