Active Vibration Reduction of Rotating Machinery Using a New Mass Distribution Control System Design

Ozan Enginoglu1, Hasan Ozturk2

1Gediz University, Mechatronics Program, Menemen, Izmir, Turkey
2The Graduate School of Natural and Applied Sciences, Dokuz Eylul University, Tinaztepe, Buca, Izmir, Turkey

Dokuz Eylul University, Department of Mechanical Engineering, 35397 Buca, Izmir, Turkey

ABSTRACT

In this study a new mass distribution control system (MDCS) is designed and it is simulated under different conditions in order to minimize the dynamic vibration of rotating machines. The design consists of two MDCS placed at the both ends of the rotating shaft and it consist of three masses that can rotate around a plane which is perpendicular to the shaft. As a consequence of rotations of the masses, Center of Gravity (COG) of the MDCS moves to a point where it can be beneficial as a counterforce to suppress the imbalancing forces. In this study, using the forces applied on the bearings, both MDCS\' mass angles are determined and successfully applied for reducing vibration significantly.

Keywords: Active Balancing, Rotating Machinery, Vibration Reduction, Imbalance Compensation

1. INTRODUCTION

Machines used in the industrie cause vibration because of their internal structure which consist of parts such as gear, shaft, cam, pulley etc. These machine parts cause vibration on the whole machine due to various reasons. Many factors cause vibration in machine operation but usually the lack of balance stands out as the main cause. Under these circumstances, the extra vibration cause all machine parts to be exposed to extra dynamic load. The shaft bearings under constant exposure to these dynamic forces are subject to decreased life expectancy because of the wear down on the ball bearings. As a result, fatigue occurs on the machine parts which in turn reduces the efficiency of the whole machine dramatically. The proposed balancing system is aimed to minimize the dynamic forces caused by the balance problems.

Researchers have studied the imbalance vibration problem since the beginning of 20th century. To achieve the desired result, researchers have experimented with various balancing methods. Incorporating the use of balls [1, 2], electromagnetic field [3–5], rollers and even liquid moving in a groove or ring [6–10] are among some examples. Most of these papers either consist of a rigid shaft balanced by placing balls on one or two planes or replacing the rigid shaft with a flexible body to achieve the balance [11–13].

A smoother and quieter operation with the advantage of increased lifetime [14–20] is obtained by applying the balancing methods to modern equipment such as optic discs and hand grinders.

Review of literature on vibration reduction shows that the achieved results are not satisfactory. For instance, the reduction efficiency of passive ball balancers is not high and stable. The instability manifests itself as the change in dynamic forces in almost all cases when system is started over even though the initial conditions prior to the operation are kept the same. Also in some rare cases, the vibration is observed to get higher than normal. When magnetism is used to force the balls to move on a groove, the balancing vector force is limited by the number of grooves and their length. The lubrication of the grooves plays an important role for all balancing techniques incorporating the use of balls. In order to fill the gap in the design of a more efficient and stable balancing system, a different solution is proposed suggesting a new mechanical design.

1email: ozan.enginoglu@gediz.edu.tr
2email: hasan.ozturk@deu.edu.tr
2. CONCEPT OF AN ACTIVE BALANCING SYSTEM

In the context of rotating machines, external mass input/output during rotation is frequently witnessed. This causes the COG of the machine to move leading to serious vibration problems. For instance, a small chunk of stone stuck into the thread of a car tire can damage the internal components. On the other hand, for systems such as washing machines, the movement of COG is considered as normal. The laundry mass inside the drum causes the COG to change arbitrarily even if the machine is perfectly balanced. In such cases, an active balancer is required to return the COG back to its original rotation axis to reestablish the balance. This can be made possible by using various sensors to detect instantaneous force changes at the bearings.

When the balancing vector is calculated using the sensor feedbacks, the active balancing system creates a counter force vector to compensate the imbalance inside the machine. In order to do this, a balancing mass is moved to a location where the centrifugal force caused by the balancer cancels out the centrifugal force created by imbalance.

3. SIMULATION AND METHOD

Simulating and observing the effects of a new mechanical balancing system on a rotating machinery is the main scope of this paper. For this purpose, a simulation model shown in Fig. 1 is used. This model is easy to implement and includes all the necessary elements to represent a much more sophisticated structure such as a rotor, a fan, a washing machine or similar.

Table 1. Nomenclature of Fig. 1

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>( D_1, D_2 )</td>
<td>-</td>
<td>Balancing Planes</td>
</tr>
<tr>
<td>( G_\alpha, G_\beta, G_\gamma )</td>
<td>m</td>
<td>COG of the flaps</td>
</tr>
<tr>
<td>( m_a, m_b )</td>
<td>kg</td>
<td>Masses causing imbalance</td>
</tr>
<tr>
<td>( a, c )</td>
<td>m</td>
<td>Lengths between balancing planes and bearings</td>
</tr>
<tr>
<td>( b )</td>
<td>m</td>
<td>Length between two balancing planes</td>
</tr>
<tr>
<td>( \vec{F}_A, \vec{F}_B )</td>
<td>N</td>
<td>Forces acting on the bearings due to vibration</td>
</tr>
<tr>
<td>( \omega )</td>
<td>rad/s</td>
<td>Angular velocity of the shaft</td>
</tr>
</tbody>
</table>

To understand the structure of the model, it can be divided into three separate parts; the shaft, the disks and the balancers. Among these, balancers are explained in a subsection.

The Shaft is connected to the ground via supports through bearings and it is assumed that there are three axis force sensors at both supports. During the operating, the dynamic forces that occurs due to imbalance are fed back to the control system simultaneously from both of the sensors. Along with imbalance vibrations, when the shaft rotates, it makes transverse oscillations. If the shaft is out of balance, the centrifugal force causes vibration and this force increases as the rotational speed approaches the natural frequency of transverse oscillations to the multiples of resonant speed. These large vibrations lead to the whirling of the shaft. In this study, in order to demonstrate the influence of whirling effect, the shaft is simulated as being flexible in...
one simulation and rigid in another. After the simulation in which this flexible shaft is used, the radius of the shaft is increased from 10 mm to 15 mm in order to demonstrate the effects of whirling on balancing.

The disks can be visualized as the simplified representations of main mechanical components such as gear, fan or similar. These components may cause undesired imbalance side effects if not maintained properly. These side effects can be created by attaching various imbalance masses to different positions on the disks. This procedure makes it possible to test the system under different conditions. When a mass is placed on a disk as shown in Fig. 2, the COG of the whole system shifts by an amount \( e \).

![Figure 2. Centrifugal Force Acting on a Mass](image)

When the system starts operating, a centrifugal force \( F_s \) occurs on the whole system \( m_s \) which can be calculated by the Eq. (1).

\[
F_s = m_s \omega^2 e
\]  

(1)

The number of disks attached to the shaft can be increased without altering the system. The information needed to calculate balancer flap angles are obtained from the sum of all vector forces acting on the supports. In this study, two disks are used for testing purposes.

3.1 The Active Balancers

The proposed balancing system consists of three flaps shown in Fig. 1.b and also in Fig. 3 as a schematic representation. In the \( xy \) plane, the flaps are uniformly placed with \( \theta_\alpha, \theta_\beta, \theta_\gamma \) angles each being equal to \( 120^\circ \). These flaps can rotate relative to the \( xy \) plane in both directions which gives the balancer the ability to adjust its COG position. The rotation angles are shown as \( \phi_\alpha, \phi_\beta, \phi_\gamma \).

![Figure 3. Schematic Representation of the Balancing System](image)
When the flap angles are changed, the COG of the balancer moves to a position with cartesian coordinates of \( r_x, r_y, r_z \).

This balancing system has two major improvements in respect to the existing balancing systems in use. The first one of these is that by using three flaps, it is possible to balance a wide area with no deadzone left inside. The workzone of the balancers is shown in Fig. 4. The three flap structure causes the workzone to appear in a hexagon shape with its incircle radius given by Eq. (2). Therefore, if the point of imbalance lies within that radius, it can be balanced.

\[
r = \frac{l\sqrt{3}}{6} \approx 0.29l
\]

where \( l \) is the length between the balancer’s center point and the COG of the balancer.

The second major improvement is that, if the flaps’ masses are increased, it is possible to balance much bigger systems such as rotors, turbines, washing machines etc. An increase in mass of the flaps, increases the radius of the incircle workzone, making it possible to balance greater imbalance forces without compromising its ability to balance smaller ones.

### 3.2 Methodology

The method used for automatic balancing is similar to the traditional balancing methods with the difference of using flaps to create the compensation masses. Initially, two masses are placed on disks as shown in Fig. 1. The purpose of placing masses to different disks and different positions is to simulate different balancing scenarios. For such a system, balancing would not be possible using static balancing techniques. Static balancing techniques balance the system when it’s not operating but when it starts to rotate, dynamic forces occur on the bearings due to the asymmetric placement of the masses. By using these forces, it is possible to find the compensation mass positions. Two different types of sensors are used for measuring the forces. The first one is a three axis force sensor that is mounted directly below the bearings to measure the force. The second sensor is a hall effect sensor which is placed on the shaft used to get a reference of \( x \) axis.
At the initial stage of the rotation, the balancers’ flaps are stretched out, giving the possibility to measure the imbalanced system force outputs. This ensures the center of gravity of the balancers to become coincident with the centerline of the shaft making them dynamically not effective over the system. At this stage, balancing equations are calculated and flap angles $\phi_1, \phi_2, \phi_3$ are found using the information acquired from the sensors. Later on, the flaps are adjusted to rotate at the calculated angles thus making the COG of the balancers to become coincident with the centerline of the shaft that was disturbed by imbalance masses. As a result, the new configuration balances the system and reduces the vibration. If one or both imbalance masses change their positions (like clothing in a washing machine), in order to balance the machine once again, the flaps are returned to their initial position for the recalculation of the flap angles. This infinite loop provides a quality balancing mechanism that can balance the system even if it is somehow disturbed by any reason.

4. MATHEMATICAL MODELING

The proposed balancing mechanism depends on the flap angles but since the variables used in these equations depend on the other parts of the system, it is not possible to calculate the angles separately. It is necessary to merge the equations obtained from the body of the system. For this purpose force and moment equations are derived and then merged together to obtain the balancing positions $r_{1x}, r_{1y}, r_{2x}, r_{2y}$ on the $D_1$ and $D_2$ planes.

To calculate the flap angles, balancing position equations that are derived by force and moment equilibriums are merged with the equations that are derived by finding the COG of the flaps and the following non-linear equations are obtained. For the first balancing plane, these equations are as follows:

$$m_1 l_1 \cos(\theta_1) \cos(\phi_1) + m_1 l_1 \sin(\theta_1) \sin(\phi_1) = \frac{-(m_1 + m_2 + m_3) [(a + b + r_2) F_{A_y} + (c - r_2) F_{B_y}]}{\omega^2 m_1 (b - r_{1x} + r_{2x})}$$  [3]

$$m_1 l_1 \cos(\theta_1) \sin(\phi_1) + m_1 l_1 \sin(\theta_1) \cos(\phi_1) = \frac{-(m_1 + m_2 + m_3) [(a + b + r_2) F_{A_y} + (c - r_2) F_{B_y}]}{\omega^2 m_1 (b - r_{1x} + r_{2x})}$$  [4]

$$m_1 l_1 \sin(\phi_1) = m_1 l_1 \sin(\phi_1)$$  [5]

Derived non-linear equations for the second balancing plane are given below.

$$m_2 l_2 \cos(\theta_2) \cos(\phi_2) + m_2 l_2 \sin(\theta_2) \sin(\phi_2) = \frac{(m_1 + m_2 + m_3) [(a + r_1) F_{A_y} - (b + c - r_1) F_{B_y}]}{\omega^2 m_2 (b - r_{1x} + r_{2x})}$$  [6]

$$m_2 l_2 \sin(\theta_2) \cos(\phi_2) + m_2 l_2 \sin(\theta_2) \sin(\phi_2) = \frac{(m_1 + m_2 + m_3) [(a + r_1) F_{A_y} - (b + c - r_1) F_{B_y}]}{\omega^2 m_2 (b - r_{1x} + r_{2x})}$$  [7]

$$m_2 l_2 \sin(\phi_2) = m_2 l_2 \sin(\phi_2)$$  [8]
The COG position variables $r_1z$ and $r_2z$, which are the distance of the COG in $z$ plane are assumed to be equal to zero. With this assumption, six unknown $\phi$ variables remain which can be solved with the six equations given above. Since these equations are nonlinear, it is not possible to reduce these equations further. Instead, Matlab application’s nonlinear function solver is used to solve and find all flap angles.

The $r_z = 0$ assumption provides the static balance on the whole system but due to the flap structure, the system still remains dynamically unstable. With these active balancers, the effects of the centrifugal forces are minimized but some of the moment forces that occur due to the asymmetrical positioning of the flaps remain as a residue. These residue moment forces are much smaller than the centrifugal forces in comparison and they will be the only imbalanced forces left along with the forces caused by whirling effect after the balancing is applied on the system.

5. CASE STUDY

In order to test the accuracy and the efficiency of the balancers, a simulation rig shown in Fig. 5 is designed in Solidworks and then exported to MSC Adams application for analysis.

Figure 5. Isometric View of the Simulation Rig in MSC Adams Analysis Software

The shaft, flaps and discs are assumed to be made of steel in the simulation rig. The shaft and its bearings are designed as flexible objects in order to acquire realistic results. With this assumption, it is possible to observe the whirling of the shaft. Other numerical assumptions and variables used for the simulation are given in Table 3.

<table>
<thead>
<tr>
<th>Variable Name</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flap Mass</td>
<td>0.130</td>
<td>kg</td>
</tr>
<tr>
<td>Shaft Mass</td>
<td>1.910</td>
<td>kg</td>
</tr>
<tr>
<td>Disc Mass</td>
<td>0.589</td>
<td>kg</td>
</tr>
<tr>
<td>Left Disturbance Mass</td>
<td>$2.5e -2$</td>
<td>kg</td>
</tr>
<tr>
<td>Right Disturbance Mass</td>
<td>$2.5e -2$</td>
<td>kg</td>
</tr>
<tr>
<td>Shaft Length</td>
<td>0.8</td>
<td>m</td>
</tr>
<tr>
<td>$a, c$</td>
<td>0.1</td>
<td>m</td>
</tr>
<tr>
<td>$b$</td>
<td>0.6</td>
<td>m</td>
</tr>
<tr>
<td>$\omega$</td>
<td>200</td>
<td>rpm</td>
</tr>
<tr>
<td>$m_{1a}, m_{1b}, m_{1c}$</td>
<td>0.130</td>
<td>kg</td>
</tr>
<tr>
<td>$m_{2a}, m_{2b}, m_{2c}$</td>
<td>0.130</td>
<td>kg</td>
</tr>
<tr>
<td>$l_{1a}, l_{1b}, l_{1c}$</td>
<td>$48.4e -3$</td>
<td>m</td>
</tr>
<tr>
<td>$l_{2a}, l_{2b}, l_{2c}$</td>
<td>$48.4e -3$</td>
<td>m</td>
</tr>
</tbody>
</table>

Table 3. The Simulation Rig Properties

In this simulation, imbalance mass positions are unknown, providing a good example of the mechanical problems of real world rotating machinery. Therefore, imbalance masses with a mass of 25 grams each are randomly placed on the discs.
In order to understand the reaction of the system under different conditions, further simulations will be made. For instance, to figure out the effects of gravity, some simulations will be performed with gravity whereas it will be removed in others. The use of gravity will also enable the simulation to show the whirling effect of the flexible shaft.

Initially, the output of the left and right force sensors when the balancers are stretched out and the gravity is not present are shown in Fig. 6. The forces are observed to be periodic and they form a sinusoidal wave as expected. The period for 200 rpm angular speed corresponds to a 0.3 second time interval. The flap angle calculations are made using the force outputs of the system at the startup positions. With this rotation speed, the system returns to its startup position at every period. The simulation charts are trimmed to show from 0.9 s to 2.0 s to remove noise.

![Dynamic Forces Acting on the Left and Right Bearings at Both Axes](image)

**Figure 6: Dynamic Forces Acting on the Left and Right Bearings at Both Axes (200 rpm, without Gravity, imbalanced)**

When no gravity is present, the peak to peak amplitude of the dynamic forces at the left bearing $x$ axis is equal to 0.644 N. When the same simulation is repeated with gravity present, the flexible shaft causes additional oscillations and the peak to peak amplitude increases to 1.038 N. This corresponds to a significant increase of 68.16% which also proves the importance of the whirling effect on the flexible shaft. Comparison of the force outputs with or without the presence of gravity can be seen in Fig. 7.

![Dynamic Forces Acting on the Left Bearing X axis](image)

**Figure 7. Dynamic Forces Acting on the Left Bearing X axis (200 rpm, imbalanced)**

Two variables are related directly with the whirling. These are the length and the radius of the shaft. The radius and the whirling are inversely proportional; when the radius is increased, the whirling decreases. This is the reason why rigid materials are preferred for shaft design in rotating machines.

On the $y$ axis, the mass located above the three axis force sensors cause a static force due to gravity. This causes the graph to shift downwards as shown in Fig. 8. The magnitude of this static force is approximately 18.9 N.
Figure 8. Dynamic Forces Acting on the Left Bearing Y axis (200 rpm, imbalanced)

The output with the presence of gravity shown above has more ripple due to the whirling effect which is similar to the Fig. 6,7. The peak to peak amplitude for y axis output under gravity is equal to 0.913 N which corresponds to an increase of 42% when compared to no gravity.

If the dynamic load outputs are taken from Fig. 6 at 0.3 s intervals, the forces at the supports are measured as shown in Table 4.

<table>
<thead>
<tr>
<th>Variable Name</th>
<th>Value (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{A_x}$</td>
<td>0.269</td>
</tr>
<tr>
<td>$F_{A_y}$</td>
<td>0.1735</td>
</tr>
<tr>
<td>$F_{B_x}$</td>
<td>0.2263</td>
</tr>
<tr>
<td>$F_{B_y}$</td>
<td>0.2102</td>
</tr>
</tbody>
</table>

Table 4. Dynamic Forces on the Bearings

By using these reaction forces, the COG positions of the balancers can be calculated. A small Matlab script is written for this purpose. The script takes force inputs, solves the equations and prints out the solution. For the current simulation, the output of the script is shown below in Table 5.

<table>
<thead>
<tr>
<th>Balancing Positions</th>
<th>Balancer 1</th>
<th>Balancer 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r_x$</td>
<td>-0.001615 m</td>
<td>-0.001282 m</td>
</tr>
<tr>
<td>$r_y$</td>
<td>-0.000979 m</td>
<td>-0.001265 m</td>
</tr>
</tbody>
</table>

Table 5. Calculated Balancer COGs

Knowing the intended COG positions provides the possibility to calculate the flap angles. The solutions for these nonlinear equations are provided with another Matlab script calculating the list of $\phi_{1x}$, $\phi_{1y}$, $\phi_{2x}$, $\phi_{2y}$ and $\phi_{2\gamma}$ angles. The results of the current simulation are given in Table 6.

<table>
<thead>
<tr>
<th>Flap Angles</th>
<th>Balancer 1</th>
<th>Balancer 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\phi_{\alpha}$</td>
<td>31.05 deg</td>
<td>-29.18 deg</td>
</tr>
<tr>
<td>$\phi_{\beta}$</td>
<td>-22.82 deg</td>
<td>24.87 deg</td>
</tr>
<tr>
<td>$\phi_{\gamma}$</td>
<td>-7.35 deg</td>
<td>3.84 deg</td>
</tr>
</tbody>
</table>

Table 6. Calculated Flap Angles

Adjusting the flap angles to these output values should be able to move the COG of the whole system to a position which will minimize the vibration level. When these angles are entered into MSC Adams to see the difference, a much lower vibration is obtained as a result. The outputs of the balanced system with no gravity is shown below in Fig. 9.
Figure 9: Dynamic Forces Acting on the Left Bearing at Both Axes (200 rpm, without Gravity)

The reduction of vibration forces in $x$ axis is 92.1% and 91% in $y$ axis. A small amount of force caused by the asymmetrical positioning of the flaps remains as a residue force on both axes.

If the same simulation is repeated with the gravity enabled, the simulation outputs change as in Fig. 10.

Figure 10. Dynamic Forces Acting on the Left Bearing at Both Axes (200 rpm, with Gravity)

The outputs are slightly different when gravity is applied. The vibration reduction rate is observed to be 50.3% in $x$ axis and 39.1% in $y$ axis. This change is related to the whirling of the shaft and as stated before this effect cannot be balanced. In this run, the shaft is especially selected to be as flexible as possible to show the impact of whirling on the system clearly.

6. CONCLUSION

This paper proposes a new type of mechanical balancing system to provide a comprehensive solution for balancing rotating machinery. For testing the accuracy of the balancers, a simulation model and a case is created by attaching two imbalance masses arbitrarily. The model is simulated under various conditions to see the effects of rotational speed and gravity on the system and the following conclusions are drawn.

1. The balancing of the system is accomplished by rotating the flaps of the balancers. By adjusting the flap angles, it is possible to sensibly move the COG of the balancers to an appropriate location which compensates the vibration caused by imbalance masses.

2. The formulas to solve the balancers’ flap angles are derived and presented in the paper. By using these formulas repeatedly, it is possible to run the system in close-loop and balance it once the position of the imbalance masses changes during operation.

3. When the simulation results are examined, the results clearly show the success of the system. The proposed active balancing system has prominent advantages over existing balancers since it has a much larger workzone and has the ability to move its COG to an exact position within its workzone.

4. It is also possible to balance large masses using heavier flaps. Increasing the weight of the flaps does not change the ability of the balancers, on the contrary, it increases its workzone and improves its capability of balancing heavier imbalance masses.
5. In the study, the vibrations due to imbalance masses are greatly reduced. On the other hand, the vibrations due to whirling of the shaft is not possible to balance but can be made negligible by selecting a shaft with greater radius, shorter length or with a different material which has more rigidity.

7. **ACKNOWLEDGEMENTS**

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