IMPLEMENTING ACTIVE FORCE CONTROL TO REDUCE VIBRATION OF A SHORT LENGTH DRIVE SHAFT

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1.0 INTRODUCTION

Rotating shafts are used to transmit torque along one side to the other. During the operation, unwanted vibration can occur in rotating shafts, in which they are mainly due to unbalancing and lack of good support or constraint and unsuitable tolerances. There are three main types of unbalancing in rotational shafts or rotors [1-4] and they are, namely, static, couple and dynamic unbalance. The first
occurs where the mass axis is shifted parallel to the shaft axis, and in order to correct the unbalancing, only one axial plane should be taken into consideration. The second type of unbalancing takes place with the intersection of the mass axis and the running axis. The couple unbalance type often exists in two planes. The third type usually happens where there is no coincidence between the rotational axis and the mass axis. This type of unbalancing is a mixture of couple and static unbalances, and it usually gets corrected in two planes.

Usual methods that are implemented to improve and correct the effects of unbalancing are to remove, add, and to move some material (mass) of the rotatory shafts [4]. In order to determine the best method, it is important to analyze the physical properties of the device that is rotating. Another main reason for a drive shaft to suffer from vibration, is when the shaft is operating with support and constraint that are not fixed with suitable tolerances. A few samples of these conditions can be found in typical vibration textbooks [5]. In some machines and mechanisms, the support and constraint that are holding the shaft can get worn out and thus lose their suitable tolerance.

In this article, to reduce the noise and vibration that occur in a short length of rotating shaft, a closed loop control system is employed, known as active force control (AFC) along with a PID control element. One of the biggest benefits of the AFC method is its capability to reject the disturbances that are forcibly applied to the system through suitable handling of the designated parameters. Furthermore, the method requires far less computational task and has been well reported that it is readily applied in real-time [6]. The method of AFC was first presented by Hewitt and Burdess [7], and it turned out to be very robust and efficient in controlling a robot arm. Later on, other researchers successfully applied this method to a robot arm and structure incorporating artificial intelligence (AI) techniques [6, 8, 9] and employing various active elements (actuators) such as pneumatic artificial muscle actuator (PAM) [10], Voice coil actuator (VCA) [11] and piezoelectric actuator [12]. The AFC technique is also very much capable and successful in reducing friction induced vibration (FIV) [12-15] applied to various brake models in which the FIV is caused by the effects of negative damping and modal coupling. The methodology of AFC was also used and implemented for micro robots, in which the results were very satisfying [16-17]. The application of AFC to reduce vibration in various mechanical structures can be found in [18-21].

2.0 DYNAMIC MODEL OF THE SHORT LENGTH DRIVE SHAFT

A 3 DOF dynamic model of a short length of drive shaft is shown in Figure 1. The fact that the length of the shaft is short and the cross section area of the shaft is usually 10% of its length, for the purpose of vibration analysis, the shaft is considered as a lumped parameter instead of a continuous element. Also, since the material considered for this shaft has a high modulus of elasticity, continuous element analysis is not necessary. In this model, the driving shaft is caused to vibrate due to lack of suitable constraint and support or tolerances with the geometric shape and length of the shaft is assumed to be uniform, i.e., a cylindrical bar. As shown in Figure 1, there are two directions of movements, one in the x and y plane and the other, a rotational direction θ which is the rotation around the cross sectional axis of the shaft.

![Figure 1 A 3 DOF dynamic model of a short length drive shaft](image)

The designated specifications of this short length shaft are as follows:

- Length, \( L = 0.3 \) m
- Diameter of the circular bar, \( D = 0.03 \) m
- Modulus of elasticity, \( E = 186 \) GPa
- Shear modulus of elasticity, \( G = 72 \) GPa
- Mass, \( m = 1.7 \) kg

The dynamic equation of the above drive shaft model is written as follows:

\[
\begin{align*}
mx' + cx' + kx &= F_x \\
my' + cy' + ky &= F_y \\
J\ddot{\theta} + c_\theta\dot{\theta} + k_\theta\theta &= T_\theta
\end{align*}
\]

where:
- \( m \): the mass
- \( J \): is the mass moment of inertia
- \( x \) and \( y \): directions of movement in plane
- \( \theta \): angular movement of the cross section
- \( c \): damping coefficient
- \( k \): stiffness constant
- \( F \): applied force
- \( T \): applied torque
After obtaining the dynamic model and its equation of motion, the behavior of the model was simulated and studied using MATLAB and Simulink software. Due to the fact that a non-energy dissipating vibration of this shaft was required (for severe vibration), all the damping coefficients were deemed negligible. The values of the stiffness constants and external forces are as follows:

\[ k_x : 1000 \text{ kN/m} \]
\[ k_y : 400 \text{ kN/m} \]
\[ k_{\theta} : 5.71 \text{ kNm/rad} \]
\[ F_x : 1 \text{ kN} \]
\[ F_y : 0.60 \text{ kN} \]
\[ T_{\theta} : 0.10 \text{ kNm} \]

Figure 2 shows the Simulink block diagram of the 3 DOF model of the short length drive shaft, also known as the passive system.

![Simulink block diagram of the 3 DOF model of the short length drive shaft](image)

The simulation period was set to be 5 minutes (300 s) and the type of solver chosen for this study was Bogacki-Shampine (ode3 in MATLAB/Simulink) with a fixed step time sampling. Figures 3(a) and (b) show the results of this simulation in time and frequency domains, respectively for the horizontal \( x \) direction and axial loading. As can be seen, the horizontal direction of the vibration has an amplitude of 0.002 m (2 mm) and a frequency of 122 Hz.

![Vibration results in the horizontal (x) direction of the passive system in (a) time and (b) frequency domains](image)

Figures 4 (a) and (b) show the obtained results of the passive system simulation in time and frequency domains, respectively for the vertical \( y \) direction and normal loading. It is obvious that the vibration has an amplitude of 0.003 m (3 mm) and a frequency of 77 Hz.
3.0 CONTROL STRATEGY

After obtaining the 3 DOF model of the passive system and studying its behavior, the next step is to control and suppress the vibration that is produced in all directions. A good control strategy should be designed and implemented.

Thus, a strong and robust control strategy is presented, using an AFC scheme along with a conventional PID controller. A typical PID controller is of the following form:

\[ G_c(s) = (K_P + K_I/s + K_D)s \]

where \( K_P \), \( K_I \), and \( K_D \) are the proportional, integral and derivative gains, respectively, \( e(s) \) is the trajectory error and \( G_c(s) \) is the controller transfer function.

To tune the PID controller, firstly the Ziegler-Nichol method was employed to produce the initial values for the PID controller gains. Later on, to achieve good performance, more experimentations were carried out to determine a good combination of the gains. After tuning the PID controller, the AFC loop was applied to the control system in order to increase the efficiency and the robustness of the vibration attenuation process. The AFC scheme applied to the passive system is presented in Figure 6.

\[ F_d = F - EM \ddot{x} \]
where $F$ is the measured actuating force, $EM$ is the estimated mass and $\ddot{x}$ is the measured acceleration. The obtained parameter $F_2$ is later fed back through an inverse transfer function of the actuator and then summed up with the signal of the PID control. The hypothetical analysis and the stability of the AFC method has been appropriately and rigorously described in [22].

**4.0 SIMULATION OF THE CONTROL STRATEGY**

The same conditions and solver that were used in section 2 were implemented here through MATLAB/Simulink computing platform. At first, for every DOF in the passive model of the drive shaft, a PID controller with a linear actuator was considered. Note that it comprises a 3 DOF system, implying that it requires three separate controllers to control each DOF. Then after tuning the PID controller and obtaining a suitable value for the linear actuator coefficient, the performance of the driving shaft system for vibration suppression was tested. Afterwards, the AFC loop was added (cascaded in series) to each of the PID controller, and the values of the estimated mass ($EM$) for each AFC loop was obtained by crude approximation method. The PID controller gains and the estimated mass/inertia that were assigned to each DOF are given in Table 1. The values of $I$ and gains in $\theta$ direction are very small because the moment of inertia of the shaft is small and so is the applied torque.

![Figure 7 Simulink block diagram of the PID controllers and AFC loops for the 3 DOF passive system](image)

**Table 1 Parameters used in the simulation**

<table>
<thead>
<tr>
<th>Direction</th>
<th>EM/I for AFC</th>
<th>Actuator coefficient for AFC</th>
<th>PID controller gains</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x$</td>
<td>3.0 kg</td>
<td>0.20</td>
<td>0.75 0.50 0.11</td>
</tr>
<tr>
<td>$y$</td>
<td>2.7 kg</td>
<td>0.20</td>
<td>0.50 0.30 0.075</td>
</tr>
<tr>
<td>$\theta$</td>
<td>4 e-5 kgm$^2$</td>
<td>0.02</td>
<td>4 e-4 3.5 e-4 1.2 e-4</td>
</tr>
</tbody>
</table>

It should also be noted that the linear actuator coefficients were assumed and estimated based on the size of the physical motor obtained from a datasheet. As previously mentioned, the Ziegler-Nichol method for tuning the PID controller gains was used to obtain the initial values, and then further tuned experimentally (via numerical simulation) in order to come up with a suitable vibration reduction of the drive shaft. Figure 7 shows the Simulink block diagram of the passive system when three PID controllers and three corresponding AFC loops were directly added in series. In other word, for each DOF, an independent PID controller and AFC loop is considered.

**5.0 RESULTS AND DISCUSSION**

After obtaining the Simulink block diagram and tuning the PID and AFC parameters, the simulation was executed by first considering the PID controllers only, and then the PID plus AFC control loops. Figures 8(a) through (c) show the result of the two simulations (PID and PID+AFC) for the horizontal $x$ direction in time and frequency domains. It can be noticed that when the control system is operating with only a PID controller, the vibration is observed to show an upward offset but slow gradual decreasing trend with increasing frequency. On the other hand, when the AFC loop is engaged along with the PID controller, the frequency of oscillation is greatly reduced to less than 0.1 Hz, and the amplitude significantly shows a decreasing trend with much less magnitude. Note that the characteristics of the curve, though seemingly oscillatory, are for amplitudes within $20 \times 10^{-4} m$ region which is relatively very small in comparison to the dimension of the shaft.
Figures 8 and 10 show the results for the vertical y and θ directions, respectively. Again, similar trend was observed, implying that the vibration is also effectively suppressed at a much faster rate and bigger magnitude (lower frequency) when the AFC loop was engaged with the PID controller. The amplitude is seen largely reduced as it is hovering about the zero datum without any undesirable offset.

In order to evaluate the robustness of the control system in reducing the unwanted vibration in the drive shaft, another set of simulation with the same procedure was performed. But this time around, the external load for each direction (DOF) was increased to more than 50%. Figure 11 shows the result of reducing the vibration in the horizontal x direction when the horizontal external load is increased from 1 to 1.5 kN. It can be noticed that the PID system is indeed attenuating the vibration but consumes more time with high frequency fluctuation and that the amplitude of the vibration is relatively high especially within first two minutes. On the other hand, for PID+AFC scheme, the vibration is very much suppressed in magnitude and it produces very low frequency oscillation. For the vertical y direction, the normal load was increased from 0.60 to 0.90 kN, and for the θ direction, the external torque was increased from 100 to 150 Nm.
Figures 11 and 12 show the results when the external normal load and torque, respectively are increased. Again, it can be seen that the PID+AFC controller manages to minimize the vibration significantly, in terms of magnitude and frequency reduction compared to the PID controller only counterpart. From the results shown in Figures 8 to 13, it is very obvious that undesirable offsets, mainly due to static unbalancing, were produced by the PID controller for the rotating short drive shaft undergoing the given loading and operating conditions; i.e., the shaft is forcibly displaced for about 1.5 to 2 mm in translation and 0.015 to 0.025 rad in rotation and that it vibrates at relatively high frequency though towards the end, there is attenuation in vibration magnitude. It therefore signifies that the PID controller performance is sensitive, easily affected adversely and hence not robust against the disturbances. This also shows that the fixed PID controller gains could not facilitate or adapt to the changes in the disturbances. However, the performance was greatly enhanced with the onset of an AFC-based scheme cascaded in series with the conventional PID control scheme (same controller gains) whereby it totally eliminates the offsets and reduces the vibration magnitudes and their frequencies.

Table 2 Summary of the vibration results (normal loading) obtained for both with and without controller at 150 s of simulation period

<table>
<thead>
<tr>
<th>Controller (direction)</th>
<th>Amplitude</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without (x)</td>
<td>2.0 mm</td>
<td>122</td>
</tr>
<tr>
<td>Without (y)</td>
<td>3.0 mm</td>
<td>77</td>
</tr>
<tr>
<td>Without (θ)</td>
<td>0.035 rad</td>
<td>850</td>
</tr>
<tr>
<td>PID (x)</td>
<td>1.5 mm</td>
<td>122</td>
</tr>
<tr>
<td>PID (y)</td>
<td>2.2 mm</td>
<td>77</td>
</tr>
<tr>
<td>PID (θ)</td>
<td>0.025 rad</td>
<td>850</td>
</tr>
<tr>
<td>PID+AFC (x)</td>
<td>0.2 mm</td>
<td>below 0.5</td>
</tr>
<tr>
<td>PID+AFC (y)</td>
<td>0.3 mm</td>
<td>below 0.5</td>
</tr>
<tr>
<td>PID+AFC (θ)</td>
<td>0.004 rad</td>
<td>below 0.5</td>
</tr>
</tbody>
</table>
6.0 CONCLUSION

A 3 DOF model of a short length drive shaft was developed and numerically experimented. The behavior of the model was tested and simulated considering the PID and AFC-based schemes and introduced operating and loading conditions. It was noticed that the PID controller alone was able to reduce the vibration, but the process was a bit time consuming and resulted in noticeable frequency oscillation with an upward offset due to the effect of disturbances. On the other hand, when the AFC loop was added to the PID controller (PID+AFC), the vibration was significantly reduced at a much faster rate and the amplitude was heading towards zero, without any offset. In addition, the frequency of oscillation was greatly decreased to indicate that the vibration has been effectively suppressed and the offset produced by the PID scheme totally eliminated. The robustness of the proposed control strategy was also verified through analyzing the results obtained when the shaft is subject to an increase in the external loads of about 50% and yet the PID+AFC control scheme was able to reduce the vibration (with respect to its amplitude and frequency) very efficiently.

References