SYNTHESIS OF DEMAND SIGNALS FOR HIGH SPEED OPERATION OF A PACKAGING MECHANISM

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ABSTRACT
The aim of the work described in this paper is to improve the dynamic performance of a one-degree-of-freedom packaging mechanism through demand signal shaping to minimize the peak to peak motor torque. This enables the mechanism to operate at higher speeds with lower vibration and noise levels, and hence with higher accuracy. Initial experimental tests have shown the motion of the Woodpecker mechanism to suffer from dynamic, vibration problems synonymous with a mechanism possessing large amounts of harmonic content in its output motion. The dynamics of the Woodpecker mechanism and the accompanying servo system are developed and the likely causes of the dynamic issues experienced are identified. A computer model of the complete system drive unit is developed utilizing experimental data. The intention is to use the model in further detailed analytical work to shape the velocity demand signal passed to the system. Inverse dynamics are used to derive the variation in driving torque, which must be exerted on the mechanism crank by the drive motor for the mechanism to achieve a constant speed over the complete cycle. Based on the computer model, a novel technique to shape the speed demand signal is developed and it is shown that significant performance improvements can be achieved without re-synthesizing the mechanism or altering the existing industrial controller.

Nomenclature
\( \tau \) Motor time-constant
\( \theta_1 \) Crank angle
\( \theta_m \) Motor angle
\( \theta_r \) Motor reference angle
\( b \) Viscous friction coefficient at the crank shaft
\( I_m \) Effective inertia of the motor and drive unit at the crank shaft
\( K_i \) Integral gain
\( K_m \) Motor/amplifier gain (Nm/V)
\( K_p \) Proportional gain
\( N \) Gear ratio

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**INTRODUCTION**

The mechanism design or mechanism synthesis is a topic which has been the subject of a great deal of research over the years. With ever increasing reliance on automation in industry, the ability to design mechanisms which can follow desired output motion paths with accuracy and repeatability at high speeds has become an invaluable tool. The traditional mechanism synthesis method is to formulate the problem as a multi-parametric optimization problem. It is a highly complex numerical task requiring a large amount of computational work to achieve a result due to the large number of variables.

Traditional parametric optimization techniques such as Newton-Raphson method or Euler’s method do not possess the ability to search multiple solution neighborhoods and are easily fixated by local minima [1]. Identifying global minima is a difficult task since the behavior of the cost function is not known in space [2]. To overcome this issue a lot of work has been carried out to create new optimization techniques. The most common include the use of population based, stochastic, evolutionary numerical methods such as genetic algorithms [3–6]. Others include differential evolution [7], and artificial immune searching [8]. An optimization method inspired by the principle of pheromone deposition by ants is developed in [9]. New numerical optimization techniques have been developed to search a solution space, such as the Geometric Centroid of Precision Positions technique [10], and the Time Varying Dimensions method [11].

The majority of the synthesis methods mentioned above, are designed to solve the dimensional synthesis problem, primarily to generate mechanism designs which can follow a desired output path with accuracy. One aspect of function generation, which is of particular interest, is the dynamic behavior of a mechanism when it is actuated at high speeds. When actuated at high speeds, physical imbalances in the mechanism induce harmonic content in the output motion of the mechanism, which is manifested as physical vibrations, the amplitude of which increase with actuation speed [12,13]. These vibrations have the potential to be very damaging to the performance of the system. They can result in a wide variety of problems including loss of motional accuracy, increased energy consumption and a decrease in mechanical life. A common method used to minimize the loss of performance is simply to limit the maximum operational speed of a servomechanism; although a higher operational speed would be desirable.

A direct link between harmonic content in the output motion and peak-to-peak actuation torque has been implied in [13].

The problem of harmonic content was tackled in [3] through the use of a cost based optimization method involving penalizing higher order (2nd harmonic and above) frequencies. The application of a mechanism synthesis program, which uses Fourier transforms to perform both dimensional and functional synthesis is described in [14]. Smart materials and cams are considered in [12, 15] to improve the dynamic performance. The influence of physical parameters of a linkage on the resonant frequency of a mechanism is analyzed in [16]. They concluded that varying the width of a linkage changes its resonant frequency considerably whereas as varying its height did not. This complements work reported in [17], which describes a method of defining the minimum cross-section of mechanism linkages with respect to physical, inter-component loading.

This paper tackles the problem of reducing the harmonic content through a model based approach to shape the velocity demand signal as a function of the crank angular position. Inverse and forward dynamic modeling of the mechanism together with experimental determination of the drive system parameters are used in the proposed method.

**SYSTEM DESCRIPTION AND INVERSE DYNAMIC ANALYSIS**

As a basis for experimental work, a candidate servo mechanism called the Woodpecker Mechanism is considered. The purpose of the mechanism is to push thin products into packaging held in a neighboring hopper. It is intended that the mechanism should be able to operate a nominal speed of 450rpm or above. A CAD model of the mechanism is shown in Fig. 1.

Dynamic modeling of the mechanism is carried out by using the program Dysim, which is explained in detail in [18, 19]. The program is energy based and utilizes Lagrangian dynamics; it can be used within the Matlab/Simulink environment. Using the CAD model of the mechanism it was possible to estimate the...
Table 1. PHYSICAL DATA FOR THE WOODPECKER MECHANISM

<table>
<thead>
<tr>
<th>No</th>
<th>Name</th>
<th>Mass (kg)</th>
<th>Inertia (Nm² × 10⁻³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Crank</td>
<td>0.927</td>
<td>0.901</td>
</tr>
<tr>
<td>2</td>
<td>Triangular inter-connector</td>
<td>0.310</td>
<td>1.42</td>
</tr>
<tr>
<td>3</td>
<td>Upper grounded pivot</td>
<td>0.414</td>
<td>1.31</td>
</tr>
<tr>
<td>4</td>
<td>End-effector</td>
<td>0.174</td>
<td>0.38</td>
</tr>
<tr>
<td>5</td>
<td>Spine</td>
<td>0.482</td>
<td>10.55</td>
</tr>
<tr>
<td>6</td>
<td>Lower grounded pivot</td>
<td>0.123</td>
<td>0.29</td>
</tr>
</tbody>
</table>

masses, inertias, locations of the centers of gravity and overall geometric dimensions of the individual linkages of the mechanism. The mechanism is modeled as 6 objects, whose physical parameters are shown in Table 1.

The connection data were also read from the CAD model. The upper and lower grounded pivots are positioned with respect to motor shaft at \((x = -123\text{mm}, y = 60\text{mm})\) and \((x = 15\text{mm}, y = -274\text{mm})\) respectively. The simulation package Dysim automatically generated a dynamic model for the mechanism by using 18 generalized coordinates representing the absolute position of the center of gravity of each object and the angle with the horizontal plane. The constraint equations were also automatically generated. The crank angle is selected as the independent generalized coordinate. The program automatically calculates the initial conditions of all dependent coordinates and their derivatives given the initial condition of the crank angle and its velocity.

Inverse dynamics was used to analyze the mechanism in order to determine the variation in torque which the drive motor must exert on the mechanism crank to enable it to rotate with a constant speed over a complete cycle. The mechanism was simulated with the crank rotating at constant speeds of 600, 450 and 300 rpm. For ease of comparison of mechanism behavior at different speeds, the concept of normalized time was adopted. For a given rotational speed, the ratio between elapsed time and revolution time \(T_{rev}\) is used as a dimensionless time value, which represents the proportion of the motion cycle completed.

Using the program, it was possible to simulate the orbital path of the end-effector of the Woodpecker mechanism as shown in Fig. 2, and to predict the ideal input driving torque profile needed to propel the mechanism at different constant speeds throughout the cycle as shown in Fig. 3. For inverse dynamics analysis, it was not necessary to include the servo system, and the controller. These are added later to develop controllers to improve the system dynamics. All external influences such as friction, gear ratios and aerodynamic effects are ignored at this stage.

The motion as depicted in Figs 2 and 3 possessed a series of important operational characteristics. The most critical portion of the path exists between normalized time values of 0.1 and 0.5 since this is the portion of the cycle in which the mechanism interacts with the product and with the product feeding sub-mechanism. During this portion of the motion, the end-effector should follow an approximately straight path. Vertical deviation should be minimal. Ideally, the end effector should move with a constant speed in this region although this is not critical. Between normalized time 0.5 to 0.1 (in the next cycle), the end-effector should recoil. Neither the geometric form nor the end-effector velocity is critical in this portion of the motion.

Analysis of the simulated response highlighted the presence of motional features which could cause potential problems when
running the mechanism at high speeds. Of particular concern
was the presence of a torque spike, between normalized times
0.8 to 0.93, followed immediately by a torque trough, between
normalized times 0.93 to 0.07 (in the next cycle). It can be seen
that as cyclic rate increases so does the magnitude of the torque
spike and trough. The magnitude and rate of change of torque in
this region was identified as being the most likely cause of vibra-
tions in the system. In contrast, throughout the remaining por-
tion of the cycle, the torque response is smooth with little deviation.
The magnitude of the torque demand is also small. Such a torque
profile is more dynamically desirable and is easy to follow with
standard servos and drives. It is implied in [12] that reducing
the harmonic content in the output motion of the mechanism re-
duces the peak-to-peak magnitude of the actuating torque. Hence
a torque response with a large peak-to-peak magnitude indicates
an output motion rich in harmonic content and vibration. This
correlates well with observations of the mechanism.

To improve the motion of the mechanism, work is concen-
trated to minimize the peak to peak magnitude of the torque re-
sponse. This can be done either through modifying the control
strategy of the system or by modifying the physical construc-
tion of the system itself. The former approach is adopted in here.

EXPERIMENTAL DETERMINATION OF DRIVE SYSTEM
PARAMETERS

Experimental work was carried out in order to create a re-
alistic mathematical model of the Woodpecker mechanism and
accompanying servo and drive units. The Woodpecker mech-
anism was connected to the drive motor via a belt drive transmis-
sion with a 3:1 gear ratio, \( N = 3 \). The servo motor used in the
test rig was an Allen Bradley MPL 540K-MJ22AA controlled by
an Allen Bradley Kinetix6000 drive unit. The servo drive unit
was connected to an external computer via Ethernet loaded with
a drive interface software package called RSLogix 5000. Using
this program, the user is capable of monitoring the state of the
motor and drive units in real time as well as passing commands
to the drive unit to dictate the behavior of the motor. The re-
sponse of the motor to a given input signal can be logged by the
interface software and dumped to a text file. The RSLogix5000
control software consists of PI control loops, and allows the user
to alter controller parameters, such as gains and tolerances.

The velocity control system for the Woodpecker mechanism
can be represented by the block diagram shown in Fig. 4. The
servomotor and the drive system is assumed to be a first order
lag with a time constant of \( \tau \) and a gain \( K_m \).

\[
G_m = \frac{K_m}{1 + \tau s} 
\]  (1)

The Dynsim model was also updated to simulate the energy
dissipation in the system through friction. The friction torque,

\[
T_f = -b \dot{\theta}_1 
\]  (2)

where \( b \) is the effective viscous friction coefficient at the crank
shaft, and \( \theta_1 \) is the crank angle.

Experiments were run through a command file to perform
certain manoeuvres and log certain signals for processing off-
line. The following system behavioral variables were logged by
the control software; velocity demand, velocity output, acceler-
atation output, position demand, position output, motor torque. In
all cases, these measurements were taken with relation to the mo-
tor output shaft. The system was again excited from rest using
ramp and step input velocity signals. Various combinations of
values of \( K_p \) and velocity demands were tested. Through analysis
of the responses of the system it was possible to derive values for
the unknown system parameters namely controller time-constant
\( \tau \), controller/motor gain \( K_m \), effective drive system inertia on the
 crank shaft \( I_m \), and the coefficient of friction \( b \).

To estimate the effective friction coefficient, the steady state
response of the complete system was considered. The torque and
velocity response of the complete system over a single complete
revolution of crank is shown in Fig. 5, where the PI controller
gains are set to \( K_p = 20 \) and \( K_i = 0 \). The high frequency oscil-
lations (of approximately at 15 Hz.) in the measured response is
considered to be due to the belt dynamics, which is not included
in the simulation. By using the approximate constant speed sec-
tion of the curve, the following relationship can be used to esti-
mate friction coefficient \( b \).

\[
\frac{T_m}{\dot{\theta}_m} = \frac{N^2}{b} 
\]  (3)

where \( T_m \) and \( \dot{\theta}_m \) are the average steady state motor torque and
speed respectively. This gives a representative friction coeffi-
cient of \( b = 0.255 \). The motor/amplifier gain \( K_m \) can be found by
considering steady state gain of the closed loop

\[
\frac{\dot{\theta}_m}{\dot{\theta}_r} = \frac{N^2 K_m K_p}{b + N^2 K_m K_p} 
\]  (4)
Again using the approximate constant velocity section of the cycle, the motor/amplifier gain is estimated as $K_m = 0.0883$. Motor torque and motor speed transient data are used to determine the effective inertia of motor shaft and associated pulleys acting on the crank shaft as $I_m = 0.0071$ kgm$^2$. Observation of the transient motor torque and the error signal suggested that the time constant $\tau$ can be taken as zero. This may be explained by the high-gain internal current feedback in the motor drive circuit. The final model is built in Simulink including the Dysim forward dynamic model of the mechanism as shown in Fig. 4. The model is tested at various speeds and controller parameters, and showed good agreement with the experimental recordings. For example, Fig. 6 shows the experimental and simulation results for a constant speed reference of 300 rpm with $K_p = 20$ and $K_i = 0$.

SYNTHESIS OF SPEED DEMAND SIGNAL

With a good model of the complete test rig, this can be used to derive a method for smoothing the torque profile by replacing the constant velocity input signal by a variable velocity signal. First of all the steady state error in the response is reduced. As it can be seen in Fig. 5, in response to a 300rpm velocity reference signal a resultant velocity response was generated with a cyclic rate of just 222.72rpm. To reduce the magnitude of this steady state error, a feed forward gain of 1.347 (300/222.72) could be introduced to the demand signal. As an alternative to using a feed forward gain an integral gain was enabled.

The demand signal shaping technique used here involves introducing a virtual "Torque Saturation" sub-system containing a "saturation" block and a "first order filter" block at the motor torque in the block diagram in Fig. 4. The "saturation" allows the maximum and minimum values of the output block to be restricted to a user definable range, and the "first order filter" smooths the truncation of the signal and aid numerical stability. There are three steps in the process:

1. With no torque limits initially imposed on the system, the velocity and torque responses of the system over a complete cycle to a constant velocity demand signal are measured (i.e. original response without velocity demand shaping).
2. With reference to the resultant torque response, upper and lower torque limits are defined to truncate the torque profile. With these torque limits now active, the constant value velocity demand signal is now fed to the system and the resultant torque and velocity responses are measured.
3. The resultant velocity was logged against position over a complete rotation of crank. The torque limits were then removed. This logged velocity is then passed to the system as a velocity demand signal this time using crank position as the independent variable. The resultant torque and velocity response were then measured (response with velocity demand shaping) and compared with the constant velocity demand results in (1) above.

Initially, torque saturation limits were chosen, so that the truncated torque peak and trough did not impact on the dynamically non-challenging portion of the motion. Later, tighter torque limits were experimented with. In all cases, the peak-to-peak magnitudes of the torque responses were markedly reduced. The torque saturation method was also applied by using a variety of integral gain $K_i$ values.

A typical result of applying the above three-stage synthesis
method is shown in Fig. 7 with $K_p = 20$, $K_i = 400$, and cycling rate demand of 300 rpm. Changing the constant velocity demand signal to a variable velocity demand as a function of the position resulted a torque profile with a significant reduction in the peak to peak magnitude. Most noteworthy is the flattened torque responses achieved with peak to peak torque signal magnitudes of only about 0.005Nm.

CONCLUSION

The work described in this paper deals with a prototype mechanism, which was reported to suffer from unacceptable levels of vibration when run at high speeds. Using dimensional and physical data extracted from a CAD model of the mechanism, a dynamic simulation model of the mechanism was created. The mechanism was analyzed using inverse dynamics to predict the variation in input driving torque that the motor must exert on the mechanism crank to propel the mechanism at a constant speed over the course of a complete cycle at different constant speeds. This analysis identified the source of the problem and indicated a region in the motion where a sharp torque spike followed by a torque trough would be needed to maintain a constant running speed.

A model of the complete test rig, of which the Woodpecker mechanism is part, was tested experimentally for various input signals and controller configuration settings. Experimental data were used to complete the modeling of the overall system and the drive unit. Using the model, work was carried out to reduce the peak-to-peak magnitude of the torque response thus theoretically reducing the harmonic content of the output motion.

A method was developed to synthesize a variable, position dependent velocity demand signal. The torque response of the system to this input signal had a significantly reduced peak-to-peak value. Integral control was used alongside proportional control to compensate for steady state error. The smoothness of the resultant torque profiles were vastly improved with peak to peak magnitudes of only 0.005Nm. The resultant cyclic rate was also very close to that of the demand signal.

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