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Experiments on Active Control of Vibration of an Elevator Vable with Varying Length

Lars J. Teppo

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EXPERIMENTS ON ACTIVE CONTROL OF VIBRATION OF AN ELEVATOR CABLE WITH VARYING LENGTH

by

Lars J. Teppo
Bachelor of Science, University of North Dakota, 1996

A Thesis
Submitted to the Graduate Faculty
of the
University of North Dakota
in partial fulfillment of the requirements
for the degree of
Master of Science

Grand Forks, North Dakota
May
1999
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Degree                Master of Science

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Date                  7 May 1999
TABLE OF CONTENTS

LIST OF FIGURES ................................................................. vii
LIST OF TABLES ................................................................. ix
ACKNOWLEDGMENTS ....................................................... x
ABSTRACT ................................................................. xi

CHAPTER

I. INTRODUCTION ................................................................. 1
   Description of the Prototype ...................................................... 1
   Description of the Model ............................................................. 4

II. SCALING THE MODEL ............................................................. 7
   Scaling the Variables ................................................................. 7
   Scaling the Bending Stiffness ......................................................... 11
   Scaling the Tension due to Gravity .................................................. 17
   Scaling the Tension due to Acceleration ........................................... 19
   Scaling the Movement Profile ......................................................... 31

III. MODEL DESIGN AND MEASUREMENT ISSUES ......................... 36
   Controlling Band Slip ............................................................... 36
   Structure Vibration ................................................................. 37
   Accelerometer Bias Voltage Drift .................................................. 38
   Boundary Conditions ............................................................... 41
Band Warp

IV. ACTIVE CONTROL OF BAND VIBRATION

- Control Function
- Actuator Selection
- Actuator Mounting
- Filters
- Simulink Models for dSpace Control
- Determining Sample Rates

V. EXPERIMENTAL RESULTS

- Results Using Numeric Integration for Control
- Results Using Analog Integration for Control
- Results Using Numeric Integration and Control Mass Compensation

VI. RECOMMENDATIONS

APPENDIX

REFERENCES
LIST OF FIGURES

Figure | Page
--- | ---
1. Model elevator car with bearings removed | 4
2. Orthogonal view of the model elevator | 5
3. Prototype schematic showing notation for tension calculations | 17
4. Model schematic showing notation for tension calculations | 21
5. Band tension for car at start of upward movement | 28
6. Band tension for car at end of upward movement | 29
7. Prototype position | 33
8. Prototype velocity | 33
9. Prototype acceleration | 33
10. Prototype jerk | 33
11. Model position | 34
12. Model velocity | 34
13. Model acceleration | 34
14. Model jerk | 34
15. Side view of pulley modification to eliminate edge vibration | 37
16. Simulink model for active control using numeric integration with filtering, showing actuator force monitoring | 59
17. Simulink model for active control using signal conditioner analog integration, showing proximity sensor band displacement monitoring | 60
18. Simulink model for active control using mass compensation and numeric integration with filtering, showing band acceleration monitoring............................................................ 61

19. Control point velocity with no control applied......................................................... 64

20. Control point velocity with control gain of 144.5 N·s/m showing the effects of car rotation................................................................. 65

21. Control point velocity with a control gain of 144.5 N·s/m after an additional bearing was attached to the actuator.............................................. 66

22. Control point velocity with a control gain of 231.2 N·s/m using a 12 Hz filter................................................................. 67

23. Control point velocity with a gain of 231.2 N·s/m using a 6 Hz filter............... 67

24. Control point velocity with a gain of 231.2 N·s/m using a 2 Hz filter............... 68

25. Control point velocity with a gain of 231.2 N·s/m using a 12 Hz filter and constraint on the long return band. ......................................................... 69

26. Control point velocity with a gain of 231.2 N·s/m using a 6 Hz filter and constraint on the long return band. ......................................................... 70

27. Control point velocity from analog integrator with no control applied.............. 72

28. Control point velocity using analog integration and a control gain of 1734 N·s/m......................................................................................... 73

29. Band displacement data with no control applied.................................................. 76

30. Band displacement with a gain of 1734 N·s/m and a 6 Hz filter...................... 76

31. Stationary band vibration with no control............................................................ 77

32. Stationary band vibration with a control gain of 50 N·s/m............................... 78

33. Total actuator force with no control and mass compensation active............... 80

34. Total actuator force with a control gain of 150 N·s/m, mass compensation and a 12 Hz filter................................................................. 80

35. Total actuator force with a control gain of 150 N·s/m, mass compensation active and a 6 Hz filter................................................................. 81
<table>
<thead>
<tr>
<th>Table</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Key prototype parameters</td>
<td>2</td>
</tr>
<tr>
<td>2. Key model parameters</td>
<td>6</td>
</tr>
<tr>
<td>3. Comparison of natural frequencies for prototype cable acting as a beam and as a string</td>
<td>16</td>
</tr>
<tr>
<td>4. Comparison of natural frequencies for model band acting as a beam and as a string</td>
<td>16</td>
</tr>
<tr>
<td>5. Parameters used to calculate model tension changes in the band</td>
<td>27</td>
</tr>
<tr>
<td>6. Movement profile regions and descriptions</td>
<td>31</td>
</tr>
<tr>
<td>7. Prototype movement profile times and polynomial coefficients</td>
<td>33</td>
</tr>
<tr>
<td>8. Prototype movement profile times and polynomial coefficients</td>
<td>34</td>
</tr>
</tbody>
</table>
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ABSTRACT

The scaling laws used to build and analyze a model of a high speed, high-rise elevator were formulated, and the effect of each parameter on the model scaling was investigated. The model was used to investigate the active control of the cable vibration during upward elevator movement and decreasing cable length. A flat band was used to model the cable and to constrain transverse vibrations to a single plane. The motion profile of the elevator was successfully scaled, but it was found that bending stiffness of the cable, tension changes due to gravity and tension changes due to acceleration can not be fully scaled with a reasonably sized model.

Practical considerations in the design, fabrication and use of the scale model were addressed. The problem of band slip, structure vibration, accelerometer bias-voltage drift, boundary condition and band warp were investigated and possible solutions explored. Band slip was reduced by coating the motor pulley with a plastic substance and can be further reduced by increasing the wrap angle of the band around the motor pulley. Structure vibration can be improved with a more rigid structure, but the structural vibrations that were encountered had a minimal effect on the results. Accelerometer bias-voltage drift was minimized by insulating the accelerometer and by judicious combinations of signal conditioners and measurement devices. Boundary conditions of the band were met by creating a band guide to provide position and slope constraints.
Band warp had little effect on the active control and was minimized by using the band guide.

Active control of the band vibration was investigated using an accelerometer to sense band motion and a voice coil actuator to apply a force at the control point. The force required for vibration control is proportional to the velocity at the control point, which was obtained by integrating the accelerometer data numerically or using an analog integrator. The additional mass at the control point from the actuator and accelerometer counteract the damping effect of the active control on the band, and limit the damping to only the control point. A mass compensation force was added to the actuator output to counteract the effect of the mass. The best vibration reduction at the control point was obtained using the analog integrator, and the best vibration reduction of the entire band for the stationary case was obtained using the mass compensation force. Vibration control of the entire moving band was not seen, and more experiments combining the analog integrator with the mass compensation force are required.
CHAPTER I
INTRODUCTION

Description of the Prototype

The prototype elevator is an idealized high-speed elevator travelling 50 stories, using typical specifications obtained from an elevator manufacturer. Each story is assumed to be three meters so the length of the elevator movement is 150 meters. There are assumed to be approximately seven additional stories above the elevator at the top of the movement (20.45 m), which corresponds to the distance above the model elevator car required for the sensors and the actuator.

The vibration of the elevator cables was investigated, and in the prototype elevator, six identical parallel cables are used to suspend the elevator from a sheave at the top of the building. The elevator is attached to the guide rails of the shaft by a suspension system that includes a spring and a damper. If all cables act in unison, the cables will exert identical forces on the elevator car, and the car mass, spring constant, and damping coefficient can be divided by six to obtain the values for one cable. If the cables vibrate independently, some of the transverse cable forces will cancel with the forces from other cables acting on the cable mount. This will reduce the vibration forces transmitted to the suspension system, and the vibration of the elevator car and the movement of the ends of the cable at the top of the car will depend on the net forces of the cables on the car. For
this experiment it was assumed that the suspension system is rigid, and these effects are not investigated.

When a cable vibrates, there is a change in the tension of the cable. If the cables vibrate independently, an increase in tension in one cable could cause a decrease in tension in the rest of the cables. It was assumed that this effect is negligible, and the weight of the car is evenly distributed among the six cables. When the car is accelerating, there is an increase in tension in the cables equal to the mass of the elevator car times the acceleration. This change in tension is assumed to be evenly distributed among the six cables. Each cable therefore supports one-sixth of the weight of the car, and responds to one-sixth of the mass of the car during acceleration.

Table 1. Key prototype parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{op}$</td>
<td>length of elevator movement</td>
<td>150 m</td>
</tr>
<tr>
<td>$L_{endp}$</td>
<td>cable length above elevator car at the top</td>
<td>20.45 m</td>
</tr>
<tr>
<td>$m_{ep}$</td>
<td>mass of one-sixth of the elevator car (full)</td>
<td>756 kg</td>
</tr>
<tr>
<td>$T_{0p}$</td>
<td>initial tension in one cable at the top of the elevator car</td>
<td>7414 N</td>
</tr>
<tr>
<td>$\rho_p$</td>
<td>mass per unit length of cable</td>
<td>1.0047 kg/m</td>
</tr>
<tr>
<td>$v_{maxp}$</td>
<td>maximum velocity of elevator</td>
<td>5 m/s</td>
</tr>
<tr>
<td>$a_{maxp}$</td>
<td>maximum acceleration of elevator</td>
<td>0.75 m/s²</td>
</tr>
<tr>
<td>$E_{lp}$</td>
<td>bending stiffness of cable</td>
<td>1.3914 N·m²</td>
</tr>
<tr>
<td>$g$</td>
<td>gravitational acceleration</td>
<td>9.80665 m/s²</td>
</tr>
</tbody>
</table>

The elevator is driven by a motor connected to the sheave at the top of the elevator shaft. The motor provides a torque on the sheave to raise and lower the elevator. The cables pass over another sheave and are attached to a counterweight. The purpose of the counterweight is to offset the weight of the elevator and reduce the torque on the motor when the elevator is stationary or moving at constant velocity. In order to
accelerate the elevator car upward, the motor must supply additional torque to the sheave. This increases the tension in the cables above the car and decreases the tension in the cables above the counterweight. This causes the car to accelerate upward and the counterweight to accelerate downward. The net result is that the motor must provide enough additional torque to accelerate the entire mass of the system, including elevator mass, counterweight mass, and cable mass.

The cables are assumed to be bolted to fixtures at the top of the car, so the position (u) and slope (du/dx) of the cables must be zero at the car. If the cables are pinned at the top of the car, this would change the boundary condition so that position and curvature (d²u/dx²) are zero. As will be seen later, the beam effects of the prototype elevator cable are small, and the cable behaves very closely to a string, so the effect of changing this boundary condition is negligible. At the top of the shaft, the cable passes over the sheave, and the cable leaves the sheave parallel to the surface of the sheave at the point of first contact between the sheave and the cable. This point may change slightly as the cable vibrates, causing the cable to wrap around the sheave over a slightly different angle, but this effect is assumed to be negligible. Thus, the cable is assumed to have a fixed position and slope at the point of contact with the sheave. The boundary conditions of the cable in the prototype are therefore assumed to be fixed-fixed, with u=0 and du/dx=0 at both the top and bottom of the cable.
Description of the Model

The model consists of a steel frame approximately three meters tall. The elevator car is a block of aluminum with two linear bearings that slide on two steel guide rails.

The guide rails have a circular cross section, and are 25.4 mm in diameter and set 152.4 mm apart. A flat steel band is used instead of a round cable to constrain the vibration to only one direction. The band is 12.7 mm wide and 0.38 mm thick and is made of steel with a modulus of elasticity of 207 GPa.

The frame of reference for the model elevator is inverted for reasons that will be discussed later, and references to the "top" of the model elevator car mean the side closest to the floor of the building. Upward movement of the prototype elevator therefore corresponds to "upward" movement of the model toward the floor. The vibrations are studied in the part of the band that runs from the elevator car toward the floor, and this notation is used consistently so that an "upward" motion in both the prototype and the model corresponds to a decreasing cable or band length above the car. The band is bolted to the top of the car, giving it a fixed boundary condition, and it runs up (toward the floor of the building) to slide between a band guide, which consists of two flat steel bars pressed against the band and lubricated with silicon/teflon spray lubricant. This gives the band a fixed boundary condition at the top of the band. The position where the band passes between the steel bars corresponds to the point of contact between the cable and
the sheave on the prototype elevator. After the band passes between the steel bars, it passes over an idler pulley, a tensioning pulley, and another idler pulley. It then runs down (away from the floor) inside the steel frame to the opposite end of the structure, over the motor pulley, across to an idler pulley, and then up where it attaches to the bottom of the car, forming a closed loop.

The motor pulley, tensioning pulley, and idler pulleys are 65 mm in diameter and are made of aluminum with bearings pressed into the center.

Figure 2 shows the model elevator with both the elevator car and counterweight, but for this experiment, no counterweight is used in order to reduce band slippage. The model elevator car is able to travel 2.2 m from top to bottom, with 0.3 m of band above the
car at the end of travel to allow enough room to attach the sensors and actuator to the band. A Goldline Brushless Servomotor, Model Number B-204-A-21 was used to run the elevator model, controlled by an Acroloop motor controller board. The motor is capable of a maximum rotational speed of 1120 rpm.

Some of the model parameters given in Table 2 will be calculated in later sections.

Table 2. Key model parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>L_CM</td>
<td>length of elevator movement</td>
<td>2.2 m</td>
</tr>
<tr>
<td>L_endm</td>
<td>band length above elevator car at end of upward movement</td>
<td>0.3 m</td>
</tr>
<tr>
<td>m_em</td>
<td>mass of the elevator car (with actuator mounted)</td>
<td>1.1688 kg</td>
</tr>
<tr>
<td>T_op</td>
<td>initial band tension at the top of the elevator car</td>
<td>144 N</td>
</tr>
<tr>
<td>p_p</td>
<td>mass per unit length of band</td>
<td>0.039216 kg/m</td>
</tr>
<tr>
<td>v_maxp</td>
<td>maximum velocity of elevator</td>
<td>3.527 m/s</td>
</tr>
<tr>
<td>a_maxp</td>
<td>maximum acceleration of elevator</td>
<td>25.446 m/s²</td>
</tr>
<tr>
<td>EIp</td>
<td>bending stiffness of the band</td>
<td>1.201x10² N·m²</td>
</tr>
<tr>
<td>G</td>
<td>gravitational acceleration</td>
<td>9.80665 m/s²</td>
</tr>
</tbody>
</table>
CHAPTER II
SCALING THE MODEL

Scaling the Variables

In order to properly scale the model, each parameter in the prototype and model was converted to a dimensionless parameter using combinations of the initial cable length ($L_0$), the mass per unit length of the cable or band ($\rho$), and the initial tension at the top of the elevator car ($T_0$). These parameters were chosen because they are easily measured constants and they contain the three units that need to be scaled: length, mass and time.

Using these three parameters, the following dimensionless pi terms are obtained:

$$
\pi_1 = \frac{u(t)}{L_0} \quad \pi_2 = \frac{x}{L_0} \quad \pi_3 = \frac{L(t)}{L_0} \\
\pi_4 = \frac{t}{L_0} \sqrt{\frac{T_0}{\rho}} \quad \pi_5 = \frac{E l}{T_0 L_0^2} \quad \pi_6 = \frac{g L_0 \rho}{T_0} \\
\pi_7 = \frac{m e}{L_0 \rho} \quad \pi_8 = \frac{c}{\sqrt{\rho \cdot T_0}} \quad \pi_9 = \frac{k L_0}{T_0}
$$

(2.1)

A description of each parameter is as follows:

$L_0$ – length of elevator movement
$\rho$ – mass per unit length of the cable or band
$T_0$ – initial tension of cable or band at the top of the elevator car
$t$ – time
$u(t)$ – horizontal displacement of the cable at time $t$
\( x \) – distance along cable where \( u \) is measured
\( L(t) \) – length of cable at time \( t \)
\( E I \) – bending stiffness of the cable
\( g \) – gravitational acceleration
\( k \) – spring coefficient of the elevator car suspension
\( c \) – damping coefficient of the elevator car suspension
\( m_e \) – mass of the elevator car

Each pi term for the prototype must be equal to the corresponding pi term in the model to have a correctly scaled model. The first four pi terms \((\pi_1, \pi_2, \pi_3 \text{ and } \pi_4)\) are not constants; they indicate how the amplitude of the vibration, position of the measured point along the cable, length of the cable, and time correspond between the model and the prototype during the elevator movement. Since \( \pi_1 \) and \( \pi_3 \) are functions of time, which occurs in \( \pi_4 \), comparisons of \( \pi_1 \) and \( \pi_3 \) at corresponding times must be done using \( \pi_4 \).

The subscripts \( p \) and \( m \) are used for each variable to indicate prototype and model values respectively. Setting \( \pi_{4m} = \pi_{4p} \) and solving for \( t_m \) yields

\[
 t_m = t_p \frac{L_{0m}}{L_{0p}} \frac{\sqrt{T_{0m} \rho_{m}}}{\sqrt{T_{0p} \rho_{m}}} \tag{2.2}
\]

This allows calculation of the times in the model that correspond to the times in the prototype.

The prototype parameters \( v_{\text{max}} \) and \( a_{\text{max}} \) are not used in the pi terms because they can be found by differentiating \( \pi_3 \) with respect to time. Setting \( \pi_{3p} = \pi_{3m} \) and differentiating with respect to \( t_p \) gives

\[
 \frac{d}{dt_p} \left( L_p(t_p) \right) = \frac{1}{L_{0p}} \frac{dL_m(t_m)}{dt_m} = \frac{1}{L_{0m}} \frac{dt_m}{dt_p} \tag{2.3}
\]

or
\[
\frac{v_p(t_p)}{L_{0p}} = \frac{v_m(t_m)}{L_{0m}} \frac{dt_m}{dt_p}
\]  
(2.4)

Differentiating (2.2) with respect to \( t_p \) yields

\[
\frac{dt_m}{dt_p} = L_{0m} \frac{T_{0p} \rho_m}{L_{0p} \sqrt{T_{0m} \rho_p}}
\]  
(2.5)

Substituting and solving for \( v_m(t_m) \), gives

\[
v_m(t_m) = v_p(t_p) \frac{T_{0m} \rho_p}{\sqrt{T_{0p} \rho_m}}
\]  
(2.6)

Similarly, since \( \frac{d^2 t_m}{dt_p^2} = 0 \),

\[
\frac{a_p(t_p)}{L_{0p}} = \frac{a_m(t_m)}{L_{0m}} \left( \frac{dt_m}{dt_p} \right)^2
\]  
(2.7)

and

\[
\frac{a_m(t_m)}{L_{0m} T_{0m} \rho_p} = \frac{a_p(t_p)}{L_{0p} T_{0p} \rho_m}
\]  
(2.8)

The remaining \( \pi \) terms (\( \pi_5, \pi_6, \pi_7, \pi_8, \) and \( \pi_9 \)) are used in building the model. In this model, \( \pi_8 \) and \( \pi_9 \) were not used because it was assumed that the suspension of the car was rigid, but these terms are included for possible future modifications.

In order to have a model that will scale with the prototype, \( \pi_5, \pi_6 \) and \( \pi_7 \) for the model must be equal to the corresponding \( \pi \) terms for the prototype. The prototype values for these \( \pi \) terms are: \( \pi_{5p} = 8.34 \times 10^{-9}, \pi_{6p} = 0.199 \) and \( \pi_{7p} = 5.016 \).

Setting \( \pi_{6m} = \pi_{6p} \) and solving for \( T_{0m} \) gives
Setting \( \pi_{5m} = \pi_{5p} \), substituting (2.9), and solving for \( L_{0m} \), yields

\[
T_{0m} = T_{0p} \frac{L_{0m}}{L_{0p}} \frac{\rho_m}{\rho_p} \frac{m_p}{m_m} \frac{E_{1m}}{E_{1p}} \tag{2.9}
\]

(2.9)

Substituting the prototype values and the values for the model band into (2.9), (2.10), and (2.11) results in \( T_{0m} = 174 \) N, \( L_{0m} = 90.7 \) m, and \( m_{em} = 17.84 \) kg. Since \( L_{0m} \) is much too long for a model, all of the scaling laws cannot be satisfied with this choice of band.

\[ L_{0m} \text{ can be reduced if } E_m \text{ is reduced or } \rho_m \text{ is increased by using a different material, but reducing these factors by 0.5 only reduces the length by 0.79 because of the cube root. Better results are obtained through reducing } l_m \text{ by using a thinner band. Since the band has a rectangular cross section, reducing the thickness by 0.5 would reduce } l_m \text{ by a factor of 0.125 and } \rho_m \text{ by a factor of 0.5. This would reduce } L_{0m} \text{ by a factor of 0.63. To satisfy the scaling laws using a steel band and a model height of 2.2 m, the thickness would have to be reduced to 1.44 micrometers. Thus, it is not possible to satisfy all of the scaling laws unless the model is extremely tall. Since all of the scaling laws cannot be satisfied, each } \pi \text{ term will be examined to see if any can be satisfied, and if not, the effects of not satisfying them will be examined.} \]
Scaling the Bending Stiffness

The term $\pi_5$ describes the relationship between the bending stiffness of the prototype cable and the model band. Setting $\pi_{5m}=\pi_{5p}$ and solving for $T_{0m}$, describes the tension required to satisfy this pi term:

$$T_{0m} = T_{0p} \frac{E_{Im} l_{0p}^2}{E_{Ip} l_{0m}^2}$$  \hspace{1cm} (2.12)

Using the values for $E_I$, $T_0$, and $l_0$ given above for the model and the prototype, the band tension should be set to 297,496 N. This force is much too high for the model or the band to withstand. In addition to yielding the band and collapsing the pulley and motor bearings, it can be seen from (2.6) that velocity increases as model tension increases. Using $v_{\text{max}_p}$ from the prototype elevator, this tension would require a maximum model elevator velocity of 160 m/s, and the time for the model elevator travel would be approximately 14 milliseconds. Therefore, this pi term is not able to be satisfied.

Since this pi term cannot be satisfied and since the prototype value for the pi term is extremely small, an attempt should be made to make the pi term for the model as small as possible. The limiting factor is the maximum speed of the motor, so using the steel band described above with the current model length, the tension should be set as high as possible for the resulting model elevator velocity.

The maximum motor speed is 1120 rpm, and with a 65 mm diameter motor pulley, this allows a maximum elevator velocity of $v_{\text{max}_m}=3.76$ m/s. To avoid running at the absolute maximum motor speed, a maximum motor speed of 1050 rpm was used,
which yields a maximum elevator velocity $v_{\text{maxm}} = 3.527 \text{ m/s}$. Solving (2.6) for the tension and substituting $v_{\text{maxp}}$ and $v_{\text{maxm}}$ yields

$$T_{0m} = T_{0p} \frac{v_{\text{maxm}}^2 \rho_m}{v_{\text{maxp}}^2 \rho_p}$$

This gives an initial band tension in the model of $T_{0m} = 144 \text{ N}$. The value of the pi term for the model using these parameters is $\pi_{5m} = 1.723 \times 10^{-5}$.

Assuming tension is always set as high as possible for the maximum velocity, the factors to reduce $\pi_{5m}$ can be seen by substituting (2.13) into $\pi_{5m}$, giving

$$\pi_{5m} = \frac{EI_m}{v_{\text{maxm}}^2 \rho_m L_{0m}^2} \frac{v_{\text{maxp}}^2 \rho_p}{T_{0p}}$$

Since the prototype parameters cannot be changed, this equation shows that $\pi_{5m}$ can be reduced by decreasing $EI_m$ and increasing $v_{\text{maxm}}$, $\rho_m$, and $L_{0m}$.

E can be reduced and $\rho_m$ increased by using different materials, so band materials should be selected that have a low E and a high density. Gold appears to be one of the best choices, but it is not within the budget of most researchers. Copper is a good alternative. It has a lower E than steel and a slightly higher density. This is a possible improvement for future experiments.

$I_m$ can be reduced by using a thinner or narrower band. A flat band has the lowest $I_m$ for a given $\rho_m$, so this shape is ideal. Reducing the band thickness by half would reduce $I_m$ by a factor of 8 and $\rho_m$ by half, resulting in a reduction in the pi term by a factor of 4. In order to avoid exceeding the maximum velocity of the motor, this would also require $T_{0m}$ to be reduced by a factor of two, which could lead to problems of band...
slippage on the motor pulley, as discussed later. Decreasing the width of the band would
decrease $I_m$ and $\rho_m$ by the same factor, yielding no net reduction in $\pi_{5m}$, but since this
would also reduce $T_{0m}$, it could again lead to problems with band slippage.

Higher model elevator velocity would also reduce this $\pi$ term and could be
achieved with a new motor or motor sheave, but the faster elevator movement would
require higher sample rates and would result in higher structural vibrations. Increasing
$L_{0m}$ would require a taller model, which was also not possible given space constraints.

The effect of not satisfying this $\pi$ term is that the model band is stiffer than the
elevator cable. As the bending stiffness of a beam under tension decreases, the vibration
characteristics approach those of an ideal string. The prototype cable vibration closely
approximates an ideal string, and the degree to which the prototype acts as an ideal string
can be seen by comparing the natural frequencies of the cable using beam vibration and
string vibration theories. Beams under tension vibrate at higher natural frequencies than
strings of the same tension and mass per unit length. Using a pinned attachment instead
of a fixed attachment at the top of the model elevator car causes the natural frequency to
approach the string natural frequency, and is a possible area of future investigation. The
differences in natural frequencies from assuming that a cable vibrates as a string will be
highest when the length of cable is short, or when the elevator car is at the end of upward
movement.

The natural frequency (in Hz) of a vibrating string under tension is given by

$$f_n = \frac{v^2}{4 \rho L_0}$$

(2.15)
where $n$ is the mode of vibration, $T_0$ is the tension, $\rho$ is the mass per unit length, and $L_0$ is the length. A beam with both ends pinned will have natural frequencies of

$$f_n = \frac{n^2 T_0}{2\rho L_0^2} \sqrt{\frac{4\pi^4 E I}{4\rho L_0^2}}$$

(2.16)

where $E I$ is the bending stiffness. Although the boundary conditions for the model and the prototype are not pinned-pinned, this equation can be used to see the effect of the parameters on the natural frequencies. It can be seen from (2.15) and (2.16) that the natural frequencies of a pinned-pinned beam will approach those of a string when $E I$ is small and when $T_0$ and $L_0$ are large. For a fixed-fixed beam, no closed form solution is available, and the frequencies that cause the mode shape to satisfy the boundary conditions must be found numerically. The equation for the mode shape with the boundary conditions is given by

$$W(x) = C_1 \cosh(s_1 x) + C_2 \sinh(s_1 x) + C_3 \cos(s_2 x) + C_4 \sin(s_2 x)$$

$$s_1 = \frac{T_0^2}{\sqrt{4(EI)^2 + \frac{4\pi^2 \rho f_n^2}{E I} + 2EI}}$$

$$s_2 = \sqrt{\frac{T_0^2}{4(EI)^2 + \frac{4\pi^2 \rho f_n^2}{E I} - 2EI}}$$

(2.17)

$$W(0) = W(L_0) = 0$$

$$\frac{d}{dx}W(0) = \frac{d}{dx}W(L_0) = 0$$

Using the four boundary conditions gives the following four equations:
\[ C_1 + C_3 = 0 \]

\[ C_2 s_1 + C_4 s_2 = 0 \]  \hspace{1cm} (2.18)

\[ C_1 \cosh(s_1 L_0) + C_2 \sinh(s_1 L_0) + C_3 \cos(s_2 L_0) + C_4 \sin(s_2 L_0) = 0 \]

\[ (C_1 s_1 \sinh(s_1 L_0) + C_2 s_1 \cosh(s_1 L_0)) - C_3 s_2 \sin(s_2 L_0) + C_4 s_2 \cos(s_2 L_0) = 0 \]

Solving simultaneously yields

\[ C_4 \left[ 2 - 2 \cosh(s_1 L_0) \cos(s_2 L_0) + \left( \frac{s_1^2 - s_2^2}{s_1 s_2} \right) \sinh(s_1 L_0) \sin(s_2 L_0) \right] = 0 \]  \hspace{1cm} (2.19)

For a non-trivial solution, \( C_4 \) cannot be zero. The frequencies must therefore be found where rest of the equation is zero. Since \( \cosh \) and \( \sinh \) become very large, this could cause issues in the numerical solution, so \( \cosh \) and \( \sinh \) are decomposed into the exponential terms and the equation is divided by the largest exponential. From the definition of \( s_1 \) and \( s_2 \) it can be seen that

\[ \frac{s_1^2 - s_2^2}{s_1 s_2} = \frac{T_0}{2 \pi f n \sqrt{\rho E I}} \]  \hspace{1cm} (2.20)

Making this substitution and collecting terms yields

\[ \left( \frac{T_0}{4 \pi f n \sqrt{\rho E I}} \sin(s_2 L_0) - \cos(s_2 L_0) - 2 e^{-s_1 L_0} \right) \ldots = 0 \]

\[ + e^{-2 s_1 L_0} \left( \frac{T_0}{4 \pi f n \sqrt{\rho E I}} \sin(s_2 L_0) + \cos(s_2 L_0) \right) \]  \hspace{1cm} (2.21)

The frequencies that satisfy this equation can be found numerically for the prototype and the model.
Assuming constant tension in the prototype cable of 7414 N and a length of 20.45 m, the first five natural frequencies assuming a string model and a beam model and the percent differences between the beam and string models are given in Table 3.

Table 3. Comparison of natural frequencies for prototype cable acting as a beam and as a string

<table>
<thead>
<tr>
<th>Mode</th>
<th>Beam model frequency (Hz)</th>
<th>String model frequency (Hz)</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.6240</td>
<td>1.6211</td>
<td>0.17%</td>
</tr>
<tr>
<td>2</td>
<td>3.2480</td>
<td>3.2423</td>
<td>0.18%</td>
</tr>
<tr>
<td>3</td>
<td>4.8721</td>
<td>4.8634</td>
<td>0.18%</td>
</tr>
<tr>
<td>4</td>
<td>6.4962</td>
<td>6.4846</td>
<td>0.18%</td>
</tr>
<tr>
<td>5</td>
<td>8.1206</td>
<td>8.1057</td>
<td>0.18%</td>
</tr>
</tbody>
</table>

Because the bending stiffness of the model cannot be properly scaled, the band deviates more significantly from the string model. Assuming a constant tension in the band of 144 N and a length of 300 mm (when the elevator is at the end of an upward movement) the first five natural frequencies and percent differences are given in Table 4.

Table 4. Comparison of natural frequencies for model band acting as a beam and as a string

<table>
<thead>
<tr>
<th>Mode</th>
<th>Beam model frequency (Hz)</th>
<th>String model frequency (Hz)</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>108.05</td>
<td>100.99</td>
<td>6.5%</td>
</tr>
<tr>
<td>2</td>
<td>219.06</td>
<td>201.99</td>
<td>7.8%</td>
</tr>
<tr>
<td>3</td>
<td>335.81</td>
<td>302.98</td>
<td>9.8%</td>
</tr>
<tr>
<td>4</td>
<td>460.81</td>
<td>403.98</td>
<td>12.3%</td>
</tr>
<tr>
<td>5</td>
<td>596.21</td>
<td>504.97</td>
<td>15.3%</td>
</tr>
</tbody>
</table>

Although the frequencies observed in the model will not scale correctly – they will be higher in the model than in the prototype – the control function is based on a
beam model and does not depend on bending stiffness or natural frequencies, so this deviation will not affect the vibration control experiment.

Scaling the Tension due to Gravity

In the prototype elevator, the cable tension is not constant along the cable length. If the motor is holding the prototype elevator stationary, the tensions at the locations shown in Figure 3 are

\[
\begin{align*}
T_0 &= m_e g \\
T_1 &= \rho L g + T_0 \\
T_2 &= T_3 \\
T_3 &= \rho L_{cw} g + T_4 \\
T_4 &= m_{cw} g
\end{align*}
\]

(2.22)

where \( T \) is tension, \( L \) is the length of the cable, \( \rho \) is the mass per unit length and \( g \) is the acceleration of gravity. The tension increases along the cable from the top of the elevator car to the motor due to the weight of the cable.

When the car is at the start of its movement, the length of the cable above the car is 170.45 m, which has a weight of 1679 N. The tension varies linearly from \( T_0 = 7414 \) N at the elevator car to \( T_1 = 9093 \) N at the sheave, which is an increase of 22.6%. As the elevator rises with constant velocity, the tension remains at 7414 N at the top of the elevator but

Figure 3: Prototype schematic showing notation for tension calculations.
decreases at the top as the cable length gets shorter. When the elevator is at the end of an upward movement, the weight of the cable is only 201 N so the tension $T_1$ at the motor is 7615 N, or 2.7% higher than the tension at the car. In the model, the maximum weight of the cable above the elevator car is 0.96 N, so there is only a 0.67% maximum difference in tension due to gravity.

The scaling of gravitational effects is governed by $\pi_6$, which has values of $\pi_{6p}=0.199$ and $\pi_{6m}=0.00588$. Since the value of $g$ in $\pi_6$ cannot be changed, scaling tension changes due to gravity in the model would require an extremely dense band material, a large increase in the model height, or a decrease in the band tension. Decreasing tension would adversely affect the bending stiffness scaling, and the problems with increasing density and model height have already been discussed. The change in tension due to gravity in the prototype will have a small effect on the mode shapes and natural frequencies, but these effects were not investigated in this model.

Because the change in tension due to gravity in the model was so small, the model was run upside-down, with vibration measured and control forces applied to the section of band leading from the elevator car to the pulley closest to the floor. This offered three advantages: it allowed easier placement of and access to the actuator and sensors, it reduced band slip during acceleration because the weight of the car was acting in the same direction as the acceleration, and finally, as will be shown in the next section, it reduced the tension change during the acceleration and deceleration phases of the profile.
Scaling the Tension due to Acceleration

Using the equations for tension (2.22) it can be seen that the difference in tension across the motor is

\[ T_1 - T_2 = \left[ m_e - m_{cw} + \rho (L - L_{cw}) \right] \]  

(2.23)

This difference is maintained at the motor by the friction force of the cable on the sheave. To hold the elevator in position, the motor applies a constant torque equal to the tension difference times the radius of the motor pulley. During constant velocity motion upward, the force of friction between the car and the guide rails adds to the tension. The friction in the prototype elevator is assumed to be negligible.

To accelerate the prototype elevator upward, the motor applies an additional torque to the sheave that is transmitted by friction to the cable, which increases \( T_1 \) and decreases \( T_2 \). Assuming the cable does not stretch significantly, the elevator, cables and counterweight must all accelerate at the same rate. During upward acceleration of the elevator car, and neglecting the rotational inertia of the idler sheave, the new tensions are given as follows:

\[
\begin{align*}
T_{0a} &= m_e (g + a) \\
T_{2a} &= m_{cw} (g - a) + T_{3a} \\
T_{4a} &= m_e (g - a) + T_{3a} \\
T_{1a} &= \rho L (g + a) + T_{0a} \\
T_{3a} &= \rho L (g - a) + T_{4a}
\end{align*}
\]  

(2.24)

The new difference in tension across the motor is

\[
T_{1a} - T_{2a} = \left[ m_e - m_{cw} + \rho (L - L_{cw}) \right] g + \left[ m_e + m_{cw} + \rho (L + L_{h} + L_{cw}) \right] (g - a)
\]  

(2.25)

and it can be seen that the additional tension difference over the stationary or constant velocity case is equal to the acceleration times the total mass of the system.
During acceleration, the additional tension (over the stationary or constant velocity case) at \( T_1 \) required to accelerate the elevator car and the cable above the car is given by

\[
T_{1a} - T_1 = \rho \cdot L \cdot a + m_e \cdot a
\]  

(2.26)

This additional tension will decrease linearly from the top of the cable to the bottom of the cable where the additional tension at the top of the car is

\[
T_{0a} - T_0 = m_e \cdot a
\]  

(2.27)

Thus, for positive acceleration, there is a tension increase at the car, and a larger tension increase at the motor sheave, with a linear increase along the cable. To quantify the tension changes in the cable during acceleration, the maximum acceleration value for the prototype is used and the tension changes calculated for the car at the start and end of an upward movement. When the elevator is at the bottom, the stationary tensions are \( T_0 = 7414 \) N and \( T_1 = 9093 \) N, and the tensions when accelerating upward are \( T_{0a} = 7981 \) N, and \( T_{1a} = 7615 \) N. When the elevator reaches the top, the stationary tensions are \( T_0 = 7414 \) N and \( T_1 = 7615 \) N, and the tensions when decelerating (or accelerating downward) are \( T_{0a} = 6847 \) N and \( T_{1a} = 7033 \) N. These tension changes are increases or decreases of 7.6% throughout the cable.

For a physically similar model, the scaling of tension changes due to acceleration would be governed by \( \pi_7 \), which relates the inertial effects of the elevator car to the band. Since the model uses a closed loop of band to provide the required tension instead of the weight of the elevator car, this \( \pi \) term does not apply, and the tension in the band has
different characteristics. The actual changes in the model band tension must therefore be calculated and compared to the changes in tension of the prototype.

Tension changes are scaled using $T_0$, so the percent changes in tension can be compared. Figure 4 shows the notation for the lengths of the band segments and the tensions at the ends of each band segment with the effects of band weight on the tension neglected. When the elevator is stationary the band tension is constant from one side of the motor to the top of the car, and from the other side of the motor to the bottom of the car. Using the notation in Figure 4, $T_1$ through $T_9$ are equal to $T_0$, $T_{10}$ through $T_{13}$ are equal to $T_0 + m_c g$, and the force across the motor is simply the weight of the elevator car ($T_9 - T_{10} = -m_c g$).

Once the elevator moves, the tensions will change slightly due to friction, so comparisons of the tensions during acceleration should be made to tensions for constant velocity rather than for the stationary case. The direction of the tension changes due to friction depend on the direction of movement, and the calculations were based on

Figure 4. Model schematic showing notation for tension calculations
upward elevator movement. Denoting pulley friction by $F_p$, elevator car friction by $F_e$, and band guide friction by $F_g$, when the elevator car is moving upward (toward the band guide), the tensions are

\[
T_{1v} = T_{0v}
\]

\[
T_{2v} = T_{3v} = T_{0v} + F_g
\]

\[
T_{4v} = T_{5v} = T_{0v} + F_g + F_p
\]

\[
T_{6v} = T_{7v} = T_{0v} + F_g + 2F_p
\]

\[
T_{8v} = T_{9v} = T_{0v} + F_g + 3F_p
\]

\[
T_{10v} = T_{11v} = T_{0v} + m_e g - F_e - F_p
\]

\[
T_{12v} = T_{13v} = T_{0v} + m_e g - F_e
\]

The tension change across the motor is simply the sum of the friction forces and the weight of the car:

\[
T_{9v} - T_{10v} = F_g + F_e + 4F_p - m_e g
\]

During acceleration, the situation becomes considerably more complicated. There is a linear tension change across each band segment to accelerate that section of the band. There is also a tension change across the car to accelerate the car and to overcome friction, and there is a tension change across each pulley to provide angular acceleration to the pulley and to overcome friction. The friction of the pulley is expressed as the tension change at the surface of the pulley, so pulley friction causes a torque equal to $F_p r$. If the total tension change across the pulley is given by $\Delta T_p$, then setting the torque that remains equal to the angular acceleration of the pulley times the moment of inertia gives
Substituting for $I_x$ and $\alpha$ gives

$$\Delta T_p - F_p \cdot r = I_x \cdot a$$  \hspace{1cm} (2.30)

where $m$ is the mass of the pulley and $a_i$ is the acceleration of the band. Solving for $\Delta T_p$ yields

$$\Delta T_p = \frac{m}{2} \cdot a + F_p$$  \hspace{1cm} (2.31)

From this equation it can be seen that the pulley inertia causes a tension change equivalent to a linear inertia of half the mass. Thus the inertial effect of a pulley can be treated as linear inertia with an equivalent mass of $m_p=m/2$. The mass of the pulley is just the density times the volume, so using width=0.019 m, radius=0.032 m, and $\rho_{\text{aluminum}}=2800 \text{ kg/m}^3$, this yields $m=0.16932$ and $m_p=0.08466$ kg.

Writing the equations for the tension changes during acceleration yields

\[
\begin{align*}
T_{1a} - T_{0a} &= \rho \cdot L \cdot g \\
T_{2a} - T_{1a} &= F_p \\
T_{3a} - T_{2a} &= \rho \cdot L \cdot a \\
T_{4a} - T_{3a} &= m_p \cdot a + F_p \\
T_{5a} - T_{4a} &= \rho \cdot L \cdot a \\
T_{6a} - T_{5a} &= m_p \cdot a + F_p \\
T_{7a} - T_{6a} &= \rho \cdot L \cdot a
\end{align*}
\]

\[
\begin{align*}
T_{8a} - T_{7a} &= m_p \cdot a + F_p \\
T_{9a} - T_{8a} &= \rho \cdot L \cdot a \\
T_{10a} - T_{9a} &= \rho \cdot L \cdot a \\
T_{12a} - T_{11a} &= m_p \cdot a + F_p \\
T_{13a} - T_{12a} &= \rho \cdot L \cdot a \\
T_{0a} - T_{13a} &= m_{e} \cdot (a - g) + F_e
\end{align*}
\]  \hspace{1cm} (2.33)
Adding all of these equations together gives the tension change across the motor during acceleration:

$$T_0 - T_0 = 10a = \left[ \rho L_{\text{total}} + a + 4 + \frac{m_c}{m_p} \right] a + 4a + F_p + F_c + F_g - \frac{m_c g}{E}$$

where $L_{\text{total}}$ is the sum of all the lengths. This equation is simply the total mass of the system times acceleration, plus friction forces and the weight of the car.

$T_0$ is set initially, which allows the calculation of the tension at any part of the band for the stationary case, but during acceleration or constant velocity, the above equations only describe the relative changes in tension between the parts of the band. Since this is a statically indeterminate problem, in order to determine the tension at every point during acceleration and constant velocity, another equation must be added that contains the elongation of the band due to tension.

When the band is stationary, $T_0$ is set by raising the tensioner pulley, which stretches the band slightly. The initial elongation of the band is

$$\Delta L = \frac{T_0 L_{\text{total}}}{E A} + \frac{m_c g (L_6 + L_7)}{E A}$$

where $A$ is the cross sectional area of the band. During constant velocity upward, the elongation is given by

$$\Delta L = \frac{T_0 v L_{\text{total}}}{E A} + \frac{F_g (L_2 + L_3 + L_4 + L_5)}{E A} + \frac{F_p (L_3 + 2L_4 + 3L_5 - L_6)}{E A} + \frac{m_c g (L_6 + L_7)}{E A} + \frac{F_c (L_6 + L_7)}{E A}$$

Since the pulleys are fixed, the elongation during constant velocity must equal to the stationary elongation. Setting (2.35) and (2.36) equal and solving for $T_0v$ yields
Using this, the tensions in the other parts of the band can be found.

During acceleration, the tension changes along each length of the band, so the elongation of a segment of the band is given by

\[
\Delta L_a = \frac{T_a}{E \cdot A} \frac{\rho \cdot a \cdot L^2}{2E} \quad (2.38)
\]

where if upward acceleration is considered positive, \( T \) is the tension at the bottom of the segment. Using this equation, the total elongation of the band is

\[
\Delta L_{\text{total}} = \frac{1}{E \cdot A} \left( T_{0a} L_1 + T_{2a} L_2 + T_{4a} L_4 + T_{6a} L_6 + T_{8a} L_8 + T_{10a} L_6 + T_{12a} L_7 \right) - \\
+ \frac{\rho \cdot a}{2E} \left( T_{0a} (L_1^2 + L_2^2 + L_3^2 + L_4^2 + L_5^2 + L_6^2 + L_7^2) \right)
\]

Substituting the tensions from equations (2.33) to get this equation in terms of \( T_{0a} \) yields

\[
T_{0a} = T_0 - \frac{\rho \cdot a}{\left( m_e + m_e \right)} \frac{\left( L_1 + L_2 + L_3 + L_4 + L_5 \right)^2 - \left( L_6 + L_7 \right)^2}{L_{\text{total}}} - \\
\frac{F_e \cdot (L_2 + L_3 + L_4 + L_5)}{L_{\text{total}}} - \\
\frac{F_p \cdot (L_3 + 2L_4 + 3L_5 - L_6)}{L_{\text{total}}}
\]  

(2.40)

For the model elevator, measurements were performed to estimate friction values to use in this equation. The friction force for one of the pulleys was measured at about 1.37 N by wrapping a thin wire around it and hanging weights from the wire until the pulley started to turn. All of the pulleys were assumed to have approximately the same friction. Elevator car friction force was measured by dropping the elevator in free fall down the rails between two switches that triggered a timer. The deviation of the time from that calculated for frictionless free fall was assumed to be from a constant friction
force, and a regression analysis of the data yielded a friction force of 1.04 N with the regression R=0.97. The friction of the band sliding was measured by clamping a weight onto the band to see if it forced the band to slide. This yielded an estimate of 2.4 N. The motor friction was also measured using the same procedure as for the pulleys at 2.34 N.

Since all of the friction measurements were done without tension on the band, the system friction while under tension was measured by adding weight to the car. This additional weight added to the weight of the car allowed the calculation of the overall friction of the system at the working tension. Weight was also clamped to the band on the opposite side of the model until it overcame the weight and friction and caused the system to move in the other direction. These measurements yielded a total system friction of 25.4 N and 28.0 N respectively, so the average of 26.7 N was used. This test includes the motor friction, but when the motor is moving the system, this friction will not cause any tension changes in the band.

Summing the individually measured frictions yields 11.31 N, but each of these friction values was measured without tension in the band. When the band is under tension the friction forces may increase in the pulley bearings, motor bearings, elevator car bearings, and there may be additional band resistance from wrapping around the pulley. If the original values for pulley friction, motor friction, and band guide friction are doubled and remaining system friction is attributed to the elevator car, reasonable friction estimates are obtained for each component.

Using the values in Table 5, the changes in tension due to acceleration of the model are obtained. For maximum acceleration of the car at the start of an upward
movement, the tensions at each point are shown and compared to those for the stationary case and for constant velocity in Figure 5. The tensions are shown starting at the motor, where the motor causes a large tension change. The tensions are also compared for maximum deceleration with the car at the end of an upward movement in Figure 6.

The figures show that the largest tension change is across the car. During acceleration, the tension changes along each segment of band due to band inertia are relatively minor, and the tension changes across the band guide and across the pulleys are significant. The tension changes at $T_0$ and $T_1$ require the closest examination, because this is where the measurements are performed and vibration control is applied. When the car is decelerating, the tension in this section of band drops below the constant velocity tension by about 6.3% (compared to 7.6% in the prototype). When the car is

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Start of upward movement</th>
<th>End of upward movement</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_1$</td>
<td>2.5 m</td>
<td>0.3 m</td>
</tr>
<tr>
<td>$L_2$</td>
<td>0.127 m</td>
<td>0.127 m</td>
</tr>
<tr>
<td>$L_3$</td>
<td>0.229 m</td>
<td>0.229 m</td>
</tr>
<tr>
<td>$L_4$</td>
<td>0.229 m</td>
<td>0.229 m</td>
</tr>
<tr>
<td>$L_5$</td>
<td>2.9 m</td>
<td>2.9 m</td>
</tr>
<tr>
<td>$L_6$</td>
<td>0.406 m</td>
<td>0.406 m</td>
</tr>
<tr>
<td>$L_7$</td>
<td>0.209 m</td>
<td>2.409 m</td>
</tr>
<tr>
<td>$L_{\text{total}}$</td>
<td>6.6 m</td>
<td>6.6 m</td>
</tr>
<tr>
<td>$T_{0m}$</td>
<td>144 N</td>
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</tr>
<tr>
<td>$\rho_m$</td>
<td>0.039216 kg/m</td>
<td>0.039216 kg/m</td>
</tr>
<tr>
<td>$m_{\text{cm}}$</td>
<td>1.1688 kg</td>
<td>1.1688 kg</td>
</tr>
<tr>
<td>$a$</td>
<td>25.445 m/s$^2$</td>
<td>-25.445 m/s$^2$</td>
</tr>
<tr>
<td>$m_p$</td>
<td>.08466 kg</td>
<td>.08466 kg</td>
</tr>
<tr>
<td>$F_g$</td>
<td>4.9 N</td>
<td>4.9 N</td>
</tr>
<tr>
<td>$F_c$</td>
<td>6.2 N</td>
<td>6.2 N</td>
</tr>
<tr>
<td>$F_p$</td>
<td>2.7 N</td>
<td>2.7 N</td>
</tr>
</tbody>
</table>

accelerating, however, the tension in the model decreases by 4.0%, while the prototype tension increases 7.6%. This is because the elongation reduction in the portions with lower tension must equal the elongation increase in the portions with higher tension. Since $L_5$ is very long and has the highest tension increase, $L_5$ has a large elongation, and the tension
Figure 5. Band tension for car at start of upward movement
Figure 6. Band tension for car at end of upward movement
must decrease more in $L_6$ and $L_7$ to compensate. If the motor were moved to the other side (between $T_{11}$ and $T_{12}$), this effect would be increased, because the length of the band would be much shorter between the motor and the elevator car ($L_7$).

If the movement of the car were not inverted, the tension above the car would increase by 22.4% during acceleration and would decrease by 15.7% during deceleration, which is considerably more than in the prototype. Also, the weight of the car, friction forces and inertial forces would all act in the same direction during the initial acceleration, making the band more prone to slip. With the movement inverted, these three forces are never acting in the same direction: weight helps to overcome the friction and inertial forces during initial acceleration, and the friction helps to overcome the weight and inertial forces during deceleration.

A counterweight was not used primarily to reduce the band slip. Since the tension increase across the motor during acceleration depends on the total mass of the system, adding a counterweight would nearly double the mass of the system and would increase the friction and vibration caused by the linear bearings. A counterweight would, however, reduce the tension changes in the band during acceleration, provided the motor was on the opposite side of the model from the band being studied. A counterweight that has the same mass as the elevator car would cause the tension changes on either side of the motor to be more balanced, but the tension due to acceleration would still change as the lengths of the band segments change.
Scaling the Movement Profile

The prototype elevator acceleration, velocity and length of travel were given, so the next step was to create a movement profile using these values. In order to reduce the vibration caused by an instantaneous change in acceleration (infinite jerk) a profile was developed to have a continuous and finite jerk (rate of change of acceleration) throughout the profile. The movement profile was divided into seven regions as shown in Table 6.

A second order polynomial was used for the jerk in regions 1, 3, 5, and 7, and the duration of each region n was denoted by $t_n$. A symmetrical movement profile required that $t_1=t_3=t_5=t_7$ and $t_2=t_6$. The final constraint was to set $4t_1=t_2$. This was based on observations of an actual acceleration profile from the elevator industry (National Elevator Industry, Inc. Performance Standards Committee, 1994). Using these constraints, a position function of time was developed for the elevator position that was a fifth order polynomial for each region and had continuous and finite jerk. The maximum value of jerk in this profile agreed with the maximum jerk obtained from the elevator industry. The position function, where $(t-t_0)$ gives the time elapsed from the start of each region, is as follows:

Table 6. Movement profile regions and descriptions

<table>
<thead>
<tr>
<th>Movement Region</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Region 1</td>
<td>increasing acceleration to $a=a_{\text{max}}$</td>
</tr>
<tr>
<td>Region 2</td>
<td>constant acceleration at $a_{\text{max}}$</td>
</tr>
<tr>
<td>Region 3</td>
<td>decreasing acceleration to $a=0$, $v=v_{\text{max}}$</td>
</tr>
<tr>
<td>Region 4</td>
<td>constant velocity at $v_{\text{max}}$</td>
</tr>
<tr>
<td>Region 5</td>
<td>increasing deceleration to $a=-a_{\text{max}}$</td>
</tr>
<tr>
<td>Region 6</td>
<td>constant deceleration at $a=-a_{\text{max}}$</td>
</tr>
<tr>
<td>Region 7</td>
<td>decreasing deceleration to $a=0$, $v=0$, $y=150$ m</td>
</tr>
</tbody>
</table>
The start times for each region and the polynomial coefficients are shown in Table 7. The resulting position, velocity, acceleration, and jerk curves for the prototype can be seen on page 33.

The movement profile was scaled by multiplying the times and polynomial coefficients in each region by the appropriate combination of $L_0$, $p$, and $T_0$ for the model and the prototype:

$$t_{0m} = t_{0p} \frac{L_{0m}}{L_{0p}} \sqrt{\frac{p_m T_{0p}}{p_p T_{0m}}}$$

$$C_{nm} = C_{np} \frac{L_{0m}}{L_{0p}} \left( \frac{L_{0p}}{L_{0m}} \right) \frac{p_m T_{0m}}{p_p T_{0p}}$$

Thus, the model movement profile used the same polynomial as the prototype, but with different times and coefficients, shown in Table 8 on page 34. The resulting position, velocity, acceleration, and jerk curves for the prototype can also be seen on page 34.

Using these coefficients, a motion control program was developed using the Acroloop controller software. The Acroloop board receives encoder feedback from the motor amplifier and sends a voltage to the amplifier that is proportional to the desired velocity. The number of encoder pulses per millimeter of elevator travel was entered and programming of each movement was done directly in millimeters. Using the Acroloop software a target velocity was set and each movement command consisted of a position and an acceleration. For regions 1-4, the target velocity was set to the maximum elevator velocity, which the elevator reaches at the end of region 3. Regions 1 and 3 were divided into ten segments each, and the acceleration and position at the end of each segment was
Table 7. Prototype movement profile times and polynomial coefficients

<table>
<thead>
<tr>
<th>Region</th>
<th>$t_0$ (sec)</th>
<th>$C_{0p}$ (m)</th>
<th>$C_{1p}$ (m/s)</th>
<th>$C_{2p}$ (m/s²)</th>
<th>$C_{3p}$ (m/s³)</th>
<th>$C_{4p}$ (m/s⁴)</th>
<th>$C_{5p}$ (m/s⁵)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0.1055</td>
<td>-0.03164</td>
</tr>
<tr>
<td>2</td>
<td>1.33</td>
<td>0.2</td>
<td>0.5</td>
<td>0.375</td>
<td>0</td>
<td>0</td>
<td>-0.1055</td>
</tr>
<tr>
<td>3</td>
<td>6.67</td>
<td>13.53</td>
<td>4.5</td>
<td>0.375</td>
<td>0</td>
<td>0.03164</td>
<td>0</td>
</tr>
<tr>
<td>4</td>
<td>8</td>
<td>20</td>
<td>5</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>30</td>
<td>130</td>
<td>5</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>6</td>
<td>31.33</td>
<td>136.47</td>
<td>4.5</td>
<td>-0.375</td>
<td>0</td>
<td>0</td>
<td>-0.1055</td>
</tr>
<tr>
<td>7</td>
<td>36.67</td>
<td>149.8</td>
<td>0.5</td>
<td>-0.375</td>
<td>0</td>
<td>0.10547</td>
<td>-0.03164</td>
</tr>
<tr>
<td>End</td>
<td>38</td>
<td>150</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 7. Prototype position.

Figure 8. Prototype velocity.

Figure 9. Prototype acceleration.

Figure 10. Prototype jerk.
Table 8. Prototype movement profile times and polynomial coefficients

<table>
<thead>
<tr>
<th>Region</th>
<th>t₀ (sec)</th>
<th>C₀m (m)</th>
<th>C₁m (m/s)</th>
<th>C₂m (m/s²)</th>
<th>C₃m (m/s³)</th>
<th>C₄m (m/s⁴)</th>
<th>C₅m (m/s⁵)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>8277.4</td>
<td>-119433</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.028</td>
<td>0.003</td>
<td>0.35</td>
<td>12.72</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>0.139</td>
<td>0.198</td>
<td>3.17</td>
<td>12.72</td>
<td>0</td>
<td>-8277.4</td>
<td>119434</td>
</tr>
<tr>
<td>4</td>
<td>0.166</td>
<td>0.293</td>
<td>3.53</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>0.624</td>
<td>1.907</td>
<td>3.53</td>
<td>0</td>
<td>0</td>
<td>-8277.4</td>
<td>119434</td>
</tr>
<tr>
<td>6</td>
<td>0.651</td>
<td>2.002</td>
<td>3.17</td>
<td>-12.72</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>7</td>
<td>0.762</td>
<td>2.197</td>
<td>0.35</td>
<td>-12.72</td>
<td>0</td>
<td>8277.4</td>
<td>-119433</td>
</tr>
<tr>
<td>End</td>
<td>0.790</td>
<td>2.2</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 11. Model position.

Figure 12. Model velocity.

Figure 13. Model acceleration.

Figure 14. Model jerk.
calculated. Each of these segments had a duration of about 2.7 milliseconds. For
movements in region 1, the board was programmed to move the motor to the position at
the end of each segment in region 1 using the acceleration for that segment. For region 2,
the acceleration rate was set to the maximum model acceleration and the board was
programmed to move the motor to the position at the end of region 2. The same method
was used for region 3 as for region 1. At the end of region 3 the elevator had reached the
specified target velocity, so for region 4 a move was programmed to the position
calculated for the end of region 4, and during region 4 the controller maintained the target
velocity. The same method was used for regions 5-7 as in regions 1-3, except the target
velocity was set to 5 mm/s instead of zero. Otherwise, if the controller were slightly off
at the end, it would try to correct the position using a zero velocity move and spend an
ininitely long time doing so. The movement control program can be seen in the
appendix.
CHAPTER III

MODEL DESIGN AND MEASUREMENT ISSUES

Controlling Band Slip

The friction force between the band and the motor pulley develops a tension ratio in the band across the motor pulley that is governed by the following equation:

\[
\frac{T_9}{T_{10}} = e^{\mu \varphi}
\]

(using the notation of Figure 4 on page 21). For the model elevator, the maximum tension ratio across the motor occurs at the start of region 2, when \(T_9=158.2 \text{ N}\) and \(T_{10}=102.9 \text{ N}\). The band wraps \(\pi/2\) radians around the motor pulley, so the coefficient of friction must be at least 0.274. Typical coefficients of friction between the motor pulley and the band were 0.07 to 0.12. These were too low, and the model exhibited considerable slip. In order to solve the problem without major modification to the apparatus, the motor pulley was coated with a plastic substance used to coat tool handles. This coating performed well, and very little slip was observed while using the coating. The coating lasted about 200 runs of the model before it needed to be reapplied.

Increasing the wrap angle would also reduce band slip. If the wrap angle around the motor pulley were increased to \(\pi\) radians (180 degrees) by using another idler pulley, the friction coefficient could be as low as 0.13 before slip occurs. Surprisingly, however, attempting to reduce band slip by increasing the tension has no effect. This is because in
order to satisfy the scaling laws, when tension is increased, the acceleration must also be increased, and the tension ratio is unchanged.

Structure Vibration

Vibrations of the overall model structure were observed to come from several sources. One source of vibration was the linear bearings of the car. The bearings have several rows of ball bearings contained in a plastic race. As the car moves, each row of bearings can roll parallel to the guide rail in a small loop in the race. When the car is moved by hand, slight vibration caused by the rolling action of the bearings can be seen, heard, and felt. Other bearings are available that might reduce the vibration, such as sliding bearings with a Teflon insert. These would be smoother, but the alignment of the guide rails may be more critical with this type of bearing.

The bearing clearance also allows the car to wobble and rotate slightly, and designing a car with two bearings on each guide rail would reduce the wobble and rotation of the car. If the bearings were set some distance apart on each guide rail, the car would not be as free to rotate on the bearings. A modification of this method was used to mount the actuator.

There was also some vibration caused by interaction between the band and the pulleys. The pulleys were originally machined with a 6 mm lip on both sides to prevent the band from running off the side of the pulley. As the band moved over to one side

Figure 15. Side view of pulley modification to eliminate edge vibration
of the pulley, the edge of the band rubbed against the lip before it contacted the flat part of the pulley. This caused considerable vibration, noise and wear. Reducing the size of the lip did not correct the problem, so a 2 mm radius fillet was machined on both sides of the pulley. The fillet was successful in keeping the band away from the edges of the pulleys.

The entire structure also flexed slightly from side to side. The base was made from 6 mm steel plate, but the frame, guide rails, top plate and motor were heavy enough to cause the plate to flex visibly if the structure was rocked from side to side. This was a relatively low frequency motion, and it didn’t significantly affect the measurements.

Accelerometer Bias Voltage Drift

One of the most persistent problems was bias drift of the accelerometer signal used to measure vibration of the band. A PCB Piezotronics Model 352C68 ICP accelerometer with a ±50 g range was bonded to the band with adhesive. The accelerometer measured transverse acceleration of the band at the control point, and since the vertical motion of the band was perpendicular to the direction of measurement, there was no net motion of the accelerometer in the direction of measurement. The integral of the accelerometer data should therefore remain close to zero. The accelerometer has a transverse sensitivity of 0.9%, and since the transverse acceleration was less than 3 g, this should not significantly affect the accelerometer output. The accelerometer was powered by a PCB Model 480B10 signal conditioner that used two nine-volt batteries. When the signal conditioner was connected to a DSP Technologies Siglab Model 20-22A hardware
measurement module, the accelerometer signal could be monitored using the software that interfaces with the Matlab mathematical analysis program.

When the motor was engaged there was a large bias drift in the signal conditioner output. This was present even when the elevator was not moving, presumably because there was still a current flowing to the motor to hold the elevator stationary. With the motor stationary, the bias voltage stabilized at around 1 V. When the elevator executed the movement profile, the bias voltage dropped to around 0.5 V and returned slowly to 1 V after the elevator stopped moving. When the accelerometer output was numerically integrated to get velocity, this yielded a value of 10 m/s at the end of the elevator movement, when actual velocity should be zero. This error was caused by the integration of the bias voltage.

Several methods were tried to eliminate the bias voltage, including electrically isolating the accelerometer, removing it from the elevator model entirely and suspending it in the air. Even with the accelerometer suspended in midair, the bias voltage settled at 53 mV with the elevator stationary, and decreased at a constant rate when the elevator moved. Bias changes decreased when the accelerometer and the signal conditioner were moved farther away from the model frame and from the cement floor, and increased when the accelerometer, signal conditioner or insulated accelerometer wire touched the floor, frame or band. It was thought that the problem might be caused by a charge accumulating on the model, but when an additional ground wire was attached from the motor amplifier to the frame or directly to Siglab, it made the problem worse. With the accelerometer mounted to the band on an insulating layer of epoxy and the signal
conditioner elevated and on an insulating material, the bias shift during elevator movement was reduced to around 180 mV. This was still unacceptably high, and repeated conversations with engineers at PCB and DSP Technologies failed to resolve the problem.

The bias drift appeared to be related to the interaction between the signal conditioner output and the Siglab input. Similar bias drifts were observed when the signal conditioner output was measured with a Fluke Model 2620 Data Acquisition Unit. Both the Fluke and Siglab have differential inputs with impedances of 1 MΩ. When the signal conditioner was plugged into an oscilloscope, however, no bias drift whatsoever was observed. The oscilloscope input also has an impedance of 1 MΩ, but the shield of the input is grounded through the outlet. Connecting the oscilloscope ground to the shield of the Siglab input did not reduce the bias drift.

When a different type of signal conditioner was used with Siglab, the bias drift was nearly eliminated. Using the PCB Model 482A16 Signal Conditioner, which is AC rather than battery powered, the bias output only shifted slightly during elevator movement. The vibration signal was much larger than the bias shifts, so bias shifts were estimated by integrating the accelerometer output and looking at the slope of the velocity curve, since a constant bias voltage will produce an apparent linear velocity. It appeared that with this signal conditioner the bias started very close to zero volts, jumped to −6 mV when the elevator started to move, jumped to +9 mV at 0.5 seconds, jumped to −4 mV at 0.75 seconds, and increased to zero volts again when the elevator stopped. When integrated twice to get position, a 6 mV shift held for 0.5 seconds will indicate a 7.4 cm
position change. These are step changes, rather than continuous drifts as seen with the battery powered signal conditioner. The relatively large effect of these shifts can be seen by noting that the resolution of the accelerometer is supposed to be 0.00016 g, and a 1 mV shift in the accelerometer signal is equivalent to 0.01 g.

Large increases in noise were also observed from both signal conditioners when the motor was engaged. Without the motor engaged, the noise in the battery powered signal conditioner was around 0.09 mV rms, and the noise in the AC signal conditioner was around 0.14 mV rms. When the motor was engaged, the noise from the signal conditioner increased to 10.6 mV rms and 35.4 mV rms respectively.

In order to reduce the effects of the bias drift, the AC signal conditioner was used when taking data with Siglab. Since integration of the acceleration signal to obtain velocity is required for the control scheme, a high pass digital filter was also designed to keep the velocity values centered at zero.

Very little bias drift was observed when measurements were taken using dSpace, the digital signal processor used for active control, so both signal conditioners were successfully used with the dSpace board.

Boundary Conditions

The band vibration was measured at the control point with the accelerometer, but vibration measurements at other points were required to monitor the effectiveness of the active control. A Lion Precision non-contact proximity sensor was used to measure vibrations close to the top of the band. This sensor uses changes in capacitance to measure the position of the band to within 1 μm. Since this sensor only has a ±0.64 mm
range, this probe could only measure vibration close to the end of the band where the amplitude was small. From measurements taken without using a band guide, it was found that when the elevator started to move, the change in tension caused the band to wrap tighter around the pulley, causing a large position change at the sensor. This was because the stiffness of the band prevented the band from forming a straight line between the car and the pulley. In an attempt to eliminate this effect, a pinch roller was designed that would force the band to remain in contact with the roller until it was vertical. The pinch roller was machined from aluminum, with the same diameter and bearings as the idler pulley. When the pinch roller was mounted and tightened against the idler pulley, it did eliminate the movement of the band due to the wrapping action around the pulley, but it also caused several other problems.

The first problem was that the pinch roller negated the centering action of the pulley fillets. The forces on the band from the pinch roller caused the band to continue moving off the edge of the pulley, and as it rolled up the fillet, the pinch roller put a large longitudinal kink in the band. To prevent this the fillets were removed on that pulley and the pulley edges were again machined to a sharp lip. This was still not enough to prevent the band from rolling off the edge, and the pulley had to be carefully adjusted so contact with the pinch roller was exactly flat. This had limited success, and several runs of the elevator model were possible before the band rolled off the edge again. When an attempt was made to press the kinks back out of the band, it was found that the deformation had caused one side of the band to become slightly longer than the other. This made it curve sideways when viewed flat, and required a replacement band.
The second problem was seen in the motion of the band near the pulley during elevator movement. The band oscillated back and forth several times during the elevator movement with a frequency very close to the rotation frequency of the pulley and an amplitude of about 0.1 mm. When a dial indicator was mounted on the frame to check the pulley, it was found that the pulley was out of round by 0.07 mm. This could not be improved by turning the pulley on the lathe, and trying to improve the surface of the pulley manually with a file and sandpaper tended to create an n-sided polygon.

The third and largest problem was that the pinch roller greatly increased the friction at that pulley. The increased friction could be felt as the band was pulled through the pinch roller, and it was obviously the largest friction in the system. Furthermore, this friction was not constant, but it could be felt to pulse as a section of band was pulled through. This was probably caused by interaction between the bearings of the two pulleys, and this pulsing of the friction force could cause large additional vibrations in the band. The pinch roller was thus removed and another method of keeping the band from flexing with the tension changes was sought.

It was decided to create a band guide that would hold the position and slope constant just above the pulley, but would allow the band to slide through. Hard plastic was considered as a sliding surface, but there were concerns over the wear characteristics and the friction. It was decided simply to try two bars of flat steel. These were clamped together on the band to set the clearance and then clamped to the frame. The clamp on the band was then removed to allow the band to slide between the bars, and the band and
guide were lubricated with silicon-Teflon spray. This system was found to work quite well, and as noted above, the friction was only about twice that of one of the pulleys.

Band Warp

The proximity sensor was used to measure vibration by measuring the distance to the band as it traveled past the sensor. Any band warp due to bends, kinks, or curvature of the band would also cause a distance change from the sensor as the kink traveled past. This position data was large scale and very slow when compared to the vibration, and it always occurred at the same time in the profile. A large portion of the band warp that was experienced was caused when the band was damaged by the pinch roller, but even a new band is not perfectly flat. This effect can be corrected during the analysis of the data by averaging several data sets to obtain a curve corresponding to the band warp. Since the position changes due to vibration should be largely independent between elevator movements, any large-scale changes that remain in the average should be caused by band warp. A curve can be developed for band warp error at each position. This can be subtracted from each set of vibration data to obtain the position changes caused by vibration. If the vibration data is instead analyzed in the frequency domain, this will not be necessary, because the position changes due to band warp are slow relative to the vibration, so the frequency of these effects will be very low.
CHAPTER IV
ACTIVE CONTROL OF BAND VIBRATION

Control Function

In order to reduce the vibration of a translating beam with varying length, a force must be applied to the band. If the location of the applied force moves with the band, the force to reduce the total vibration energy in the band must be proportional to and in the opposite direction of the velocity at that point. In mathematical terms:

\[ F_c(t) = -G \frac{du(t)}{dt} \]  \hspace{1cm} (4.1)

where \( F_c(t) \) is the control force, \( u(t) \) is the band displacement at the control point, and \( G \) is a gain term (Zhu et al, 1998). This is equivalent to a simple damper acting at the control point. If the control force is applied at a point that is fixed to the structure with the band travelling past the control point, the control force also depends on the slope of the beam at that point. This would require two proximity sensors close to the control point to measure both the slope and the velocity at the control point, resulting in a more complex control scheme (Zhu et al, 1998). For this reason, the first method was used by attaching an accelerometer to the band at the control point to measure acceleration. This was integrated to obtain velocity, and an actuator was placed at the same point to apply the control force proportional to the velocity. By using the accelerometer and actuator combination, the effect of different gains was investigated.
In the continuous implementation of (4.1), the force is always proportional to the velocity, so when the velocity is zero, the force will also be zero. In this case, the gain can be set arbitrarily high. In the discrete time sampled implementation that was used, this is not the case. If the total mass of the actuator, accelerometer, and length of the band affected by the control force are given by $m_c$, the momentum of this mass is $m_c \frac{du}{dt}$. In a discrete system, the velocity is measured every $T_s$ seconds, multiplied by the gain term, and this force is sent to the actuator. This force is held until the next sample. The impulse imparted to the mass at the control point is equal to $F_c T_s$ and is equal to the change of momentum of the control mass. Suppose this change of momentum during $T_s$ is equal and opposite to the momentum of the control mass. If the velocity changes during $T_s$ from band forces are neglected, the control force will bring the control mass to a complete stop during each sample time. Setting the momentum and the impulse equal and solving for $F_{c, \text{stop}}$, this equation takes the same form as (4.1)

$$F_{c, \text{stop}} = \frac{m_c}{T_s} \frac{du(t)}{dt}$$

(4.2)

with a gain of $m_c/T_s$ bringing the control mass to a complete stop during each sample time. If the velocity changes due to band forces in each sample time occur equally in each direction, $m_c/T_s$ should be the ideal gain to use. Since this gain becomes larger as $T_s$ becomes smaller, higher gains may be used with a faster sample time, and this should achieve better control. In practice, $m_c$ is difficult to measure, because the mass of the band that acts at the control point is unknown.
If a gain is used that is equal to twice this amount, the impulse would be enough to reverse the velocity during each sample period. A gain higher than this will cause the velocity to increase in magnitude for each sample time, causing an unstable feedback. Although this is to be avoided in active control, this phenomenon can be used to find $m_c$ and the ideal gain. If the gain is increased to the point where the onset of unstable feedback is observed, and then decreased by half, this should be the ideal gain for control.

For the control of the model band vibration, a control point was used that was 38 mm above the attachment of the band to the car. This corresponds to 2.6 m above the prototype elevator car. Control points farther from the car would provide greater control, but in a real elevator, the control equipment must fit in the space above the elevator car when the car is on the top floor.

As control experiments proceeded, it became apparent that although the vibration at the control point was successfully reduced, vibrations at all other points along the band were unaffected. It has been shown that even a small mass at the control point reduces the damping effect in a translating string (Zhu et al, 1997) or beam (Zhu et al, in review) with fixed length and speed. It seems likely that a similar effect is occurring in this experiment, although length and speed are not fixed. In order to compensate for this, a force was applied to the control point to attempt to cancel effect of the mass of the actuator coil and the accelerometer. When the band is accelerating at the control point, there is a force between the band and the control mass, where the control mass is the sum of the actuator coil mass and accelerometer mass. This force is equal to the control mass times the acceleration at the control point, and is the force required to cause the control
mass accelerate with the band. Since the acceleration of the control point is measured and the control mass is known, the actuator can apply this force to the control point and cancel the force between the band and the control mass. This force is applied in the same direction as the acceleration, and effectively "helps" the band accelerate the control mass, reducing the effect of the control mass on the band. This force is sustained throughout each sample period.

**Actuator Selection**

The linear actuator to provide the control force should have a fast response time, wide frequency range, small size and low mass. It should allow the band to move freely if no control signal is applied, and should be capable of applying the required force for any band position. Previous experiments in active control have used electromagnetic non-contact actuators to apply a force to a translating band (Yang, 1989) or chain (Dishan et al, 1996). Since there is no contact between the actuator and the band, this type of actuator has zero mass and completely free movement for zero force, so it ideally satisfies these two requirements. The response time and frequency range depend on the impedance of the electromagnet and the size of the amplifier, so these requirements can usually be satisfied as well. The force requirement is more difficult. The electromagnetic force on the band depends on the distance between the actuator and the band, and the force is not linear. If an proximity sensor is used to measure position and this is differentiated to get velocity, the since both position and velocity are known, the control routine could compensate for nonlinearities in the force due to position. One disadvantage of this method is that when the positions are numerically differentiated,
there is a large increase in the level of noise. Another disadvantage of electromagnetic non-contact actuators is that they are not readily available for purchase, and usually must be designed, built, tested and calibrated by the experimenter. Despite these difficulties, control experiments using a proximity sensor and an electromagnetic non-contact actuator have been successful in the previous experiments by Yang (1989) and Dishan (1996).

Other types of linear actuators include motorized screw-type actuators, inertial actuators, and voice coil actuators. Motorized screw-type actuators have a slow response time and do not allow the band to move freely in the absence of a control force. They are typically used to move to a given position rather than to apply a given force. Inertial actuators attach to the object being controlled and apply inertial forces by generating internal vibrations. These have a relatively high mass that is used as a reaction mass to generate the forces, and this would significantly change the vibration at the control point both with and without active control. They are also not capable of low frequency vibration.

Voice coil actuators, which are based on audio speaker technology, provide an excellent combination of the requirements of this experiment. Voice coil actuators consist of a tubular magnet that is closed at one end with a post in the center, leaving an air gap between the center post and the side of the tube. The top of the post at the open end forms one pole of the magnet, and the inner surface of the tube forms the other pole, creating a magnetic field across the air gap. The actuator coil is a tube wrapped with wire that fits into the air gap, so the magnetic field lines cut across the wire. When a current is
applied to the coil, the coil generates a force into or out of the magnet that is proportional to the current. Voice coil actuators come in a variety of sizes, forces and ranges of motion, and exhibit linear force characteristics throughout their range of motion. The coil has a relatively low mass and moves freely when no current is applied, apart from a small inductance created by the coil moving through the magnetic field.

In selecting the actuator for this application, the size requirements of an actuator that could mount on the elevator car and fit between the band and the guide rail were first examined. Next, the required range of motion was calculated. To find this, a set of acceleration data, taken with the AC signal conditioner to reduce the effects of bias drift, was numerically integrated. The remaining bias drift was then manually removed by finding lines that approximately fit the different portions of the velocity data. These were subtracted from the velocity data to get velocities that oscillated around zero. This was again numerically integrated to get position data, and from this set of data it was seen that the band position ranged by approximately ±0.5 mm at the control point. An actuator was desired with a range of ±1.5 mm, in order to allow enough flexibility to try different control points if necessary.

Next, the acceleration data was examined to find the maximum acceleration at the control point. Since the control gain can be set arbitrarily and $m_c$ is unknown, the control law cannot be used to size the actuator. Instead, a force was required that would provide the mass at the control point with the maximum observed band acceleration during the elevator movement. To estimate the mass at the control point, the masses of the accelerometer, the actuator coil, and 300 mm of band were added. This band length was
chosen to be roughly ten times the distance between the car and the actuator, with the assumption that this is an overestimate and will result in a safety margin in the force. The maximum observed acceleration was approximately 300 m/s², and actuator data from BEI’s Kimco Magnetics Division was used to select an appropriate actuator. The coil mass data for each potential actuator was added to the accelerometer mass and band mass, and this was multiplied by the maximum observed acceleration to obtain the force required. The actuator was suitable if it could produce this force. After examining the data for several models, a BEI Model 15-16-001A actuator was purchased, which is capable of producing a continuous (stall) force of 12 N and a ten-second peak force of 34 N. The coil mass is 37.7 g, so the force required to produce an acceleration of 300 m/s² is 17.1 N. This actuator has a ±2.5 mm stroke and a force constant of 5.78 N/amp. This met the requirements of the experiment and left enough flexibility to allow for future changes.

The actuator was powered by a Techron Model 7521 Power Supply Amplifier that was capable of operating in either constant voltage or constant current mode. In constant current mode, the current output is proportional to the voltage input. Since the desired actuator force is proportional to the current, this made configuring the output of the digital control straightforward. In a normal (constant voltage) amplifier, consideration of the back EMF created by the coil is required, since this will change the current produced by the amplifier. The amplifier also had a dial on the front that controlled the amplifier gain. This facilitated experimentation with different gain settings.
Actuator Mounting

To attach the actuator to the elevator car, an aluminum mount was created that bolted to the car. The mount consisted of one piece that bolted to the car and another piece that clamped around the magnet of the actuator. The actuator piece fit into a slot on the elevator piece that kept it aligned, and the pieces were held together with a screw in a vertical slot. This allowed vertical adjustment of the actuator so the control point could be set exactly. Another advantage of having the mount in two pieces is that selecting another control point would require machining only a new elevator piece, rather than a whole new mount.

The diameter of the coil was 28 mm, or over twice the width of the band, so an aluminum tip was machined for the coil that would apply the actuator force to an area 7 mm across. This was the same diameter as the accelerometer, which was mounted to the band directly opposite the actuator. The actuator coil tip and the accelerometer were attached to the band using epoxy, after applying an initial layer of epoxy under the accelerometer to provide electrical isolation. The magnet was then adjusted in the clamp to center the coil at mid-stroke.

Filters

A highpass digital filter was required to eliminate any bias shifts from the accelerometer. Since the accelerometer output is integrated, any constant bias shift will result in a ramped velocity when the accelerometer data is integrated. The filter therefore must be able to remove constant and ramped velocity signals. Several filters were investigated, and a second order Butterworth filter was found to be the best choice.
Higher order filters required more processing time by the digital signal processing board and caused oscillation in response to a step input, and first order filters were not able to remove a ramp input. Second order Chebyshev filters that allowed ripple in the passband caused significant attenuation in frequencies above the corner frequency, and Chebyshev filters that allowed ripple in the stopband did not completely eliminate a DC (0 Hz) signal. The Butterworth filter has a maximally flat passband response but slower rolloff characteristics.

The selection of the corner frequency for the filter was a tradeoff between the phase shift of low frequencies and the settling time in response to a step or ramp input. Higher corner frequencies provide faster settling times but cause a phase shift in frequencies below the corner frequency. A phase shift of more than 90° in the velocity would cause a positive feedback in the velocity control, so a phase shift of less than 90° was required at the first natural frequency of the longest part of the band, or 10.5 Hz. A 12 Hz corner frequency was found to have a phase shift of approximately 90° at 10.5 Hz, so this was used as the upper limit for the corner frequency. A 6 Hz corner frequency was found to have a 90° phase shift at approximately 3 Hz, and experiments were performed using both 12 Hz and 6 Hz corner frequencies. Corner frequencies below 3 Hz took more than 0.2 seconds to adequately respond to a ramp signal, so this was used as an effective lower limit for the corner frequency.

The filter transfer function was calculated using Matlab in terms of the zeros, poles and gain of the filter, which yielded two zeros at $z=1$. Since the accelerometer data must be integrated as well as filtered, the transfer function of the integrator was
multiplied with the filter transfer function. A trapezoid rule digital integrator transfer function is

$$H(z) = \frac{Ts (z + 1)}{2(z - 1)}$$

(4.3)

where $Ts$ is the sample period. When this integrator is multiplied by the highpass filter, the zero at $z=1$ cancels, the zero at $z=-1$ is added, and the gain is multiplied by $Ts/2$. The resulting transfer function is used in the velocity control to provide integration and filtering. The Matlab commands to accomplish this were

```matlab
[z,p,k] = butter(2,Fc*Ts*2,'high');
z(1) = -1;
k = k*Ts/2;
```

where $Fc$ is the corner frequency. The variables $z$, $p$ and $k$ were referenced in the Simulink model's transfer function block.

An analog integrator built into the signal conditioner was also used to provide velocity data for the active control. This integrator had a built-in highpass filter with a corner frequency of around 0.5 Hz, based on phase shift measurements with the accelerometer mounted to a shaker table. Since the output velocity was filtered by the signal conditioner, no additional digital filtering was necessary, and this velocity could be used directly in active control. The velocity values from this analog integrator were more accurate than those from the numeric integration, and gave the active control better characteristics. During shaker table testing of the signal conditioner's analog integrator it was noticed that the velocity output was opposite in sign to the acceleration output. This property of the integrator was not mentioned in the signal conditioner operating guide.
and it is fortunate that this was discovered before using the analog integrator in the control, since a reversed sign in the velocity would create unstable feedback.

A second order Butterworth filter was also used to remove noise from a second accelerometer used to measure vertical acceleration of the elevator car. This was a lowpass filter with a corner frequency of 100 Hz, since the acceleration of the car should be a relatively slow movement with low frequency components. The transfer function was created using the Matlab command

\[ [b,a]=butter(2,100*Ts*2); \]

and the variables a and b were referenced in the Simulink model. This accelerometer was primarily used for triggering during measurements, providing a change in signal when the elevator car moves. This provided a consistent start time for all measurements, and allowed a minimum of data to be taken before and after the elevator movement. Since the acceleration of the elevator car was comparatively low, the filter was required so that the accelerometer noise did not trigger the measurements prematurely.

**Simulink Models for dSpace Control**

The active control algorithm was implemented on a dSpace model 1102 digital signal processing board. Although the board had four inputs (labeled ADC#1-4) and four outputs (labeled DAC#1-4), only three of the inputs and one of the outputs were used. One input was used to monitor either acceleration or velocity or acceleration at the control point, and one used to monitor elevator car acceleration for triggering. The third input channel was used to monitor either amplifier current, band displacement from the
proximity sensor, or band acceleration from a third accelerometer mounted to the band. The triggering input had a 12 bit resolution, and the other two inputs had 16 bit resolution. The output from dSpace was connected to the input of the amplifier used to power the actuator. The analog to digital converters (ADC) used for the dSpace inputs could accept ±10 V input range, which was scaled to ±1 and sent to the digital signal processor. The digital to analog converter (DAC) used for the dSpace output received values between ±1 from the dSpace processor and scaled them to ±10 V output range.

The control program was created by building a Matlab Simulink model and converting it into C code using Matlab’s Real Time Workshop. The C code was compiled with a Texas Instruments compiler to make an executable program for the dSpace board, which was downloaded and executed by the dSpace board’s processor. All of these functions were integrated into Real Time Workshop’s “Build” command, making changes easy to implement. Several Simulink models were used to explore the different characteristics of the active control algorithm, and each point in the model could be measured and recorded using the Trace program that came with dSpace. Conversion of the input data to appropriate units was performed by multiplying the signals in the model by unit conversion gains so the recorded data using Trace would be in meaningful units.

The models used ADC#1 to process the control point data and send commands to amplifier through DAC#1. The accelerometer used at the control point was a PCB Piezotronics 352C68 ICP accelerometer with a ±50 g range and an output of 98.9 mV/g. It was powered with a PCB 480B10 signal conditioner that could output acceleration,
velocity or position by using two analog integrators. Converting the units and accounting for the scaling, the acceleration input to ADC#1 was multiplied by a conversion gain of 991.572 to obtain the acceleration in m/s². In the first model, this was multiplied to the gain (k) of the integrator/filter so the output of this integrator/filter would be in m/s. In later models this was left as a separate gain for clarity, since it was found that processing additional gain terms in the model required little processor overhead.

Experiments were also performed using the analog integrator of the 480B10 signal conditioner. The output units of the signal conditioner when used in this mode were 989 mV/in/s. This results in a conversion gain of 0.2568 to obtain velocity in m/s.

The maximum band velocity at the control point with a zero control gain was found to be approximately 0.08 m/s. The output signal of the model was multiplied by an output gain of 10 in order to scale this to ±0.8. This kept the signal to DAC#1 within the ±1 limits while utilizing nearly the full range of the output. Control function gain for the actuator was set with the dial on the amplifier, which controlled the output current for a given input voltage. The dial had a range of 0-1000, and a setting of 100 caused the amplifier to deliver a one-amp output in response to a one-volt input. Thus, the control function gain (G in equation (4.1)) was (5.78 N·s/m)×(dial setting).

In later experiments, the control function gain was set within the model, since the actuator was applying forces for both control and mass compensation. In this case, the output gain and amplifier dial settings were set to limit the force to the maximum peak force allowed for the actuator. This corresponded to 5.85 amps to the actuator, and since dSpace is capable of 10 volts output, a dial setting of 58.5 was used on the amplifier.
This caused the amplifier to provide 5.85 amps in response to an input signal of 10 V. To convert signals in the model from Newtons to the appropriate output signal, the output gain was set at 0.029586. This caused a direct correspondence between the model signal (expressed in Newtons) and the actuator force in Newtons.

The accelerometer used for measuring car acceleration on ADC#3 was a PCB Piezotronics Model 339M12 ICP triaxial accelerometer with a ±100 g range and an output of 48.3 mV/g for the x-axis (aligned with the car movement). This yielded a unit conversion gain of 2030.0 to obtain the elevator acceleration in m/s^2.

ADC#2 was connected to either the amplifier current-monitor output to monitor actuator current (and therefore the actuator force), the proximity sensor to measure band vibration displacement, or an additional accelerometer mounted on the band to measure band vibration acceleration. When connected to the amplifier, the output of the amplifier current-monitor was one volt for ten amps. Converting units and using 5.78 N/amp for the actuator, this yields a unit conversion gain of 578.3 to convert the input into Newtons.

The proximity sensor used was a Lion Precision Model PX405HB probe using a DMT12 driver used in the low sensitivity (higher range) mode. The output of the proximity sensor was 0.0635 mm/V, so the unit conversion gain when monitoring the band displacement is 0.635 to convert the input to millimeters.

The accelerometer that was used to monitor band vibration at a point other than the control location was the PCB Piezotronics Model 352A10 ICP accelerometer with a ±500 g range and an output of 10.53 mV/g. This was powered with a PCB 482A10
signal conditioner, and the output was multiplied by 10 using a gain switch on the signal conditioner. This yielded a unit conversion gain of 931.306 to convert the input to m/s².

The Simulink model that was used for control with numeric integration is shown in Figure 16. This figure shows the input signal from ADC#1 going to a transfer function block that integrates and filters the acceleration as described above. The unit conversion for acceleration in m/s² is included in k. The output of this transfer function is the velocity in m/s, and the signal is multiplied by the output gain (Output Convert) for proper scaling before it is sent to DAC#1. The actuator and accelerometer are mounted so that a positive actuator force will cause a negative acceleration of the accelerometer, so the sign of the output is positive. Positive velocity signals therefore cause a force opposing the velocity. The amplifier current-monitor signal comes into ADC#2 and is converted to N. The accelerometer mounted on the car sends a signal to ADC#3 where it is filtered and converted to m/s². The Trace program was able to measure the output of any block in the model during elevator movement, and the elevator acceleration signal

![Figure 16. Simulink model for active control using numeric integration with filtering, showing actuator force monitoring.](image-url)
Figure 17. Simulink model for active control using signal conditioner analog integration, showing proximity sensor band displacement monitoring.

value was monitored to trigger the start of data collection. When the elevator exceeded the trigger value, the data collection started, with 0.1 seconds of pretrigger measurements to ensure capture of all pertinent data.

Figure 17 shows the Simulink model used for control with the signal conditioner in velocity mode. Since the signal conditioner analog integrator also contains a filter, no additional filtering was necessary, and the velocity conversion was simply a unit conversion gain block rather than a transfer function. The sign of the velocity conversion is negative because the signal conditioner inverts the velocity signal. This figure also shows monitoring of the hand displacement vibration data from the proximity sensor, which replaces the actuator force monitoring in the model.

Figure 18 shows the Simulink model used for control with compensation for the actuator coil and accelerometer mass. The acceleration data from ADC#1 is converted to \( \text{m/s}^2 \), and then splits for mass compensation and active control. The mass compensation gain block, labeled “Comp Force (N)”, is simply the mass of the actuator coil and control
accelerometer in kilograms. An acceleration measured by the control accelerometer in m/s² results in a mass compensation force in Newtons that will sustain that acceleration for the sample period. Since both acceleration and velocity data are required for this model, the analog integrator on the signal conditioner cannot be used. The active control uses the same integrator/filter as previously, but the control gain is now set within the model rather than by the amplifier dial. This is because changing the dial setting would also change the mass compensation force.

Since the compensation force must act in the same direction as the acceleration, it is subtracted from the control force by the sum block. The output is scaled and sent to the amplifier through DAC#1. This model also shows monitoring of the band acceleration by an additional accelerometer mounted directly on the band away from the

![Simulink model](image_url)

Figure 18. Simulink model for active control using mass compensation and numeric integration with filtering, showing band acceleration monitoring.
control point. This accelerometer signal is monitored through ADC#2.

Determining Sample Rates

The sample rate was limited by the capabilities of the dSpace digital signal processing board. Complex models decrease the maximum sample rate, so the sample rate can be increased slightly if the model is simplified by combining gains and transfer functions wherever possible. This was done with the filter and integrator, but when this is done, it is no longer possible to measure at intermediate points in the model. Gain blocks were seen to have a relatively small effect on the maximum sample rate, so these were left in the model for clarity and ease of analysis. The dSpace board is capable of running models that contain continuous function blocks such as integrators, but these were found to decrease the maximum sample rate without substantial improvement in output quality.

The dSpace documentation has formulas for estimating the maximum sample rate based on the number of variables in the model, but it recommends that one simply create the model and try to load it onto the board. Using this method, the sample period could be decreased to 70 microseconds, for a sample rate of 14,286 Hz. A sample period of 60 microseconds (16,667 Hz) was possible with some Simulink models, but this was approaching the limit of the speed of the board. In some cases the board would successfully load the model, but when measurements were attempted with the Trace program, the board was not fast enough to handle the additional processing overhead for storing the measurements in the buffer.
Higher sample rates also meant that more data was collected during the elevator movement. The buffer on the dSpace board could store approximately 30,000 measurements so during the measurement time of one second only two parameters could be measured using a sample period of 70 microseconds. If the sample period was increased to 100 microseconds, three parameters could be measured. Since shorter sample periods meant improved control, the shorter sample period was used for most of the later experiments.
CHAPTER V
EXPERIMENTAL RESULTS

Results Using Numeric Integration for Control

The first measurements were performed using numeric integration of accelerometer data to obtain velocity data for the active control (using the Simulink model in Figure 16). This was done in conjunction with a high pass filter to remove any bias signal, starting with a filter corner frequency of 12 Hz. Data was taken with a sample period of 100 microseconds and the pinch roller was used to set the boundary condition at the top of the band.

Figure 19. Control point velocity with no control applied.
The velocity of the band at the control point when the control gain is zero (no active control) is shown in Figure 19. When the control gain was increased to 144.5 N·s/m, the high frequency vibrations were reduced, but large amplitude, low frequency oscillations in the velocity were seen (Figure 20). Some of these oscillations, especially those at the end of the elevator movements, were due to the entire car rotating in the plane parallel to the guide rails. As the actuator exerted a force on the band, the reaction pushed the actuator away from the band, causing the entire car to rotate due to excessive bearing clearance. As the car rotated it also moved the band where it was attached to the car and caused the control accelerometer to sense this movement.

To counter this rotation, an additional linear bearing was added to the guide rail next to the actuator, and the actuator was fastened tightly to the bearing with steel wire.
This allowed the reaction forces from the actuator to be transmitted directly through the bearing to the guide rail, rather than through the elevator car. This yielded a reduction in the large velocity spikes and the oscillation at the end of the car movement, but no overall reduction in the velocity oscillations (Figure 21).

Another possible source of the remaining large scale, low frequency velocity oscillations was the low frequency phase shift caused by the high pass filter. Measurements were taken with the corner frequency reduced to 6 Hz and 2 Hz to see if this had any effect. A comparison of the results for a control gain of 231.2 N·s/m using corner frequencies of 12 Hz, 6 Hz, and 2 Hz are shown in Figure 22, Figure 23, and Figure 24. It can be seen from these figures that the oscillations are reduced for a filter...
Figure 22. Control point velocity with a control gain of 231.2 N·s/m using a 12 Hz filter.

Figure 23. Control point velocity with a gain of 231.2 N·s/m using a 6 Hz filter.
Figure 24. Control point velocity with a gain of 231.2 N·s/m using a 2 Hz filter.

with a corner frequency of 6 Hz and 2 Hz, but the overall drift of the velocity signal using the 2 Hz filter is as large as the oscillations with the 12 Hz filter.

It was also possible that the velocity oscillations were coming from the long return band on the other side of the elevator, since this band has a first natural frequency of approximately 10.5 Hz. (These measurements were actually taken with a slightly higher band tension, so the first natural frequency of this segment of the band was closer to 11.5 Hz.) In order to eliminate possible low frequency vibrations from this band, the band on that side of the model was constrained at three locations using wire wrapped with cloth. The locations were chosen to divide the long return band into four segments to eliminate vibration in the first two fundamental frequencies of this segment of the
band. Two wires were tightened against the band on opposite sides at each point to press the cloth against the band and constrain the band from transverse vibration. Since the wires pressed the cloth against the band in opposite directions, there was no deflection that could increase the band tension. The cloth allowed the band to slide freely as the car moved and caused negligible friction. Measurements were again taken using 12 Hz and 6 Hz filters, and the results for a control gain of 231.2 N·s/m are shown in Figure 25 and Figure 26.

It can be seen from Figure 25 that constraining the long return band greatly reduced the velocity oscillations. Thus, the regular oscillations seen when using the 12 Hz filter shown in Figure 22 were probably caused by vibration of the long band on the other side of the model. This indicates that the pulleys do not completely isolate the

![Figure 25. Control point velocity with a gain of 231.2 N·s/m using a 12 Hz filter and constraint on the long return band.](image-url)
segment of the band that we are examining from the band segments in other parts of the system. The velocity oscillations when using the 6 Hz filter with the long band constraints (Figure 26) have a slightly smaller amplitude than those seen with the 12 Hz filter, but the difference is not significant. This indicates that constraining the long part of the band accomplishes the same results as using a lower filter frequency, and the large oscillations coming from the long band are maintained by the filter phase shift.

If the remaining velocity oscillations were caused by bias shifts in the accelerometer output, the 12 Hz filter should show lower amplitudes, since a higher corner frequency filter will remove bias signals more effectively. If the oscillations were caused by a phase shift of the low frequency vibrations, the 6 Hz filter should show lower amplitudes, since there will be less phase shift with a lower filter corner frequency. Since

Figure 26. Control point velocity with a gain of 231.2 N·s/m using a 6 Hz filter and constraint on the long return band.
the oscillations using the two filters show approximately the same amplitude and frequency of oscillations, the oscillations are probably true velocities, and some of the vibrations are probably caused by band warp. This hypothesis is strengthened by noting the negative spike in velocity toward the end of the movement. This is seen in the uncontrolled case (Figure 19) at 0.62 seconds, the 12 Hz filter controlled case (Figure 25) at 0.67 seconds, and the 6 Hz filter controlled case (Figure 26) at 0.54 seconds. The slightly different times could be caused by the warped portion of the band contacting the pulley at slightly different times in the profile. (These measurements were taken with the pinch roller installed.) This is possible because if the band slips slightly between elevator runs, the elevator will start the movement profile at a slightly different location.

The oscillations also seem to have larger amplitudes toward the end of the elevator movement. This is probably because the actuator is closer to the pulley when the band is shorter. Pulley and band warp effects therefore have more of an effect at the end of elevator movement. Another possible cause is that the tension in the band is slightly lower during deceleration, allowing more transverse band movement, although the shorter band should counteract this effect by constraining transverse movement.

The oscillations limited the maximum control function gain because at higher gains the large amplitude oscillations would cause the actuator coil to travel past its range and contact the bottom of the magnet. The highest gains possible were approximately 231.2 N·s/m.
Results Using Analog Integration for Control

The analog integrator of the signal conditioner was used to provide velocity data for control in the next set of experiments (using the Simulink model in Figure 17). According to the operation manual, this integrator had a built in filter with a corner frequency of less than 3 Hz, and measurements of the phase shift using a shaker table indicated it was less than 1 Hz. The band guide also replaced the pinch roller for these experiments.

Figure 27 shows the control point velocity with no control applied. The cause of the initial jump in velocity is unknown, but it could be caused by transverse sensitivity, a bias error that the signal conditioner is not capable of filtering or movement of either the car or the coil. There is also a smaller rise in velocity at the end of the profile, and these were present in all measurements and at all gains.

![Figure 27. Control point velocity from analog integrator with no control applied](image-url)
When the gain was increased, the control function reduced the motion at the control point so effectively that higher gains could be used without exceeding the maximum actuator force or range. The optimum reduction of the control point vibration was seen at a gain of 1734 N·s/m, shown in Figure 28. This was much higher than was possible using the numeric integration, where the highest gain possible was only 231.2 N·s/m. The control function reduced the vibration velocities by at least a factor of ten at the control point, and the large-scale drift dominates the velocity signal.

The level of the initial velocity rise is approximately the same level as in the measurements without active control. If this is a true band velocity, the resulting actuator force should be able to bring it to a halt and should not overshoot. Also, if this were a true band velocity, integration reveals that it would result in a 1 mm band displacement at

![Figure 28. Control point velocity using analog integration and a control gain of 1734 N·s/m.](image)
0.2 seconds. This is too large to attribute to band warp, considering that the band is at its longest at the start of the elevator movement, and the effects of a warped band going through the band guide should have the least effect during this time. This could be a true band movement if it were caused by movement of the car or the actuator coil. This is plausible, considering that most of the first velocity rise occurs during acceleration.

In contrast, the final velocity rise occurs after the elevator has come to a halt. For this signal to be caused by bias drift, the output voltage would have to drift by 0.4 V. Bias drifts of this size were seen when interfacing with Siglab, but not with dSpace, so this should not be possible unless the analog integrator is magnifying the bias drift problems.

If the accelerometer were not mounted perpendicular to the band but was mounted at a slight angle, a component of the elevator velocity would be seen as band velocity. A mounting error of only one degree would result in a velocity component along the accelerometer of 0.06 m/s. Unless the accelerometer angle changes, the velocity should rise to a constant during acceleration and drop back to zero during deceleration, but the filtering characteristics of the signal conditioner could bring the velocity signal back to zero during the constant velocity portion of the movement profile. This could explain the velocity “humps” from 0-0.2 seconds and from 0.65-0.8 seconds, but it can’t explain the large negative velocity from 0.2-0.4 seconds or the velocity “hump” after the elevator stops.
At gains of approximately 2300 N·s/m, the actuator caused a large, high frequency vibration. This is probably related to the unstable feedback discussed earlier, but if the velocity is reversed every sample period, the feedback frequency should be half of the sample frequency. The frequency of this unstable feedback vibration was approximately 1560 Hz, and the sample frequency as 10000 Hz, so the interaction between the mass and the sample time is probably more complicated than was discussed earlier.

Despite the excellent control of vibrations at the control point during the elevator movement, there was no detectable reduction in vibration at other parts of the band. Measurements were taken with the proximity sensor located approximately 51 mm from the band guide, and with an accelerometer on the band 43 mm from the control point. These data sets were analyzed by looking at the maximum amplitude of the band vibration positions and accelerations. A spectral analysis was also performed by dividing the data into short time intervals and performing an FFT of the data in each interval. The spectral analysis for the controlled case was compared to the uncontrolled case, and no difference was detectable. The band position data for the uncontrolled case is shown in Figure 29 and for the controlled case in Figure 30 for comparison.

Results Using Numeric Integration and Control Mass Compensation

The final set of experiments was performed using the acceleration of the control point and the mass of the control system (the actuator coil and the accelerometer) to compensate for the mass effects on damping (using the Simulink model in Figure 18). The same integrator/filter was used to obtain velocity data, and the filter corner frequency
Figure 29. Band displacement data with no control applied.

Figure 30. Band displacement with a gain of 1734 N-s/m and a 6 Hz filter.
was set to 12 Hz. The sample period for these tests was set to 0.0007 seconds, and the mass of the actuator coil and accelerometer in the Simulink model was set to 44.3 grams.

The damping of the vibration was first tested with a stationary band by exciting the first mode of vibration and observing the decay rate. Using the previous model (Figure 17) for active control, no reduction in the vibration of a stationary band was observed, and reduction for the stationary case would indicate that the control function was successfully reducing vibrations along the entire band. The band length was set at 1.37 m and the proximity sensor measured the vibration 51 mm from the band guide. The vibration with the gain set to zero is shown in Figure 31. The control function gain was steadily increased and the effect on the decay rate was recorded. The optimum gain was found to be approximately 50 N·s/m, which resulted in significantly faster decay of
the vibration in response to the same initial band displacement. This is shown in Figure 32.

The mass of the actuator coil and accelerometer could not be determined exactly because the cable used to connect the accelerometer added a slight mass, but it was not clear how much of the cable mass contributed to the mass at the control point. The sensitivity to the control mass value was tested by increasing the mass slightly to 49.3 grams. This reduced the optimum gain for vibration control to approximately 40 N·s/m. When the gain was increased past the optimum, the decay rate decreased until the band vibration was actually sustained for a longer time at a gain of 240 N·s/m. When the mass compensation control method was used during elevator movement, no discernable reduction of the vibration was observed at the proximity sensor or at the

![Graph showing stationary band vibration with a control gain of 50 N·s/m.](image)
accelerometer mounted 51 mm from the control point, regardless of the gain used. The compensation force required was approximately 25 N with no control, and at a gain of 50 N·s/m the control function forces were much smaller than the mass compensation forces. The total actuator force did not increase appreciably until the control function force approached the level of the mass compensation force at a gain of 150 N·s/m. The total actuator force for zero control gain but with the mass compensation active is shown in Figure 33, and the total actuator force for a gain of 150 N·s/m is shown in Figure 34. The active control force in Figure 34 shows large scale, low frequency oscillations similar to those observed with the numerical integration results before, with the high frequency mass compensation force superimposed.

Since the actuator force resulting from the control function looked similar to the results obtained in previous experiments using numeric integration, a 6 Hz corner frequency was also tested. This resulted in a small reduction in the low frequency oscillation, as shown in Figure 35.
Figure 33. Total actuator force with no control and mass compensation active.

Figure 34. Total actuator force with a control gain of 150 N·s/m, mass compensation and a 12 Hz filter.
Figure 35. Total actuator force with a control gain of 150 N·s/m, mass compensation active and a 6 Hz filter.
CHAPTER VI
RECOMMENDATIONS

Further investigation into the mass compensation should take place in order to validate the use of contact sensors and actuators for active control. This method was successful in producing damping for the stationary band, which indicates that it might also reduce moving band vibrations under the right conditions. Since the best results were obtained at the control point using an analog integrator, a separate analog integrator should be used instead of the numerical integration on the digital signal processor. This should reduce errors associated with the numerical integration, but will require an additional input to the signal processor board.

The model should be improved by increasing the motor pulley size and the wrap angle of the band around the motor pulley. This will require a new motor pulley and motor mount, and an additional idler pulley. This should significantly reduce problems with slip. Different bearings for the elevator car should be investigated, and two pair of bearings should be spaced vertically apart to reduce car rotation. The effects of using a counterweight should also be investigated. A counterweight could help decrease the change in tension due to acceleration in the band above the car. The motor should be placed on the bottom of the model instead of the top if the elevator model is to be run right side up. This will decrease tension changes during acceleration, but will increase the tendency of the band to slip. Efforts should also be made to reduce the friction present in
the model, since small friction forces can result in comparatively large tension changes in
the band when the model is in motion.

Future models should be built as tall as possible. This will improve the scaling
between the prototype and the model by reducing the effect of the bending stiffness and
allowing slower movement profiles. Since the quality of the active control depends on
the speed of the digital signal processor, and since taller models will result in lower
frequency vibrations, the active control should improve with taller models. Actuators
should be chosen to have a minimum coil mass to reduce the effects of the mass on the
damping characteristics. A lower coil mass will result in lower mass compensation
forces.

The causes of the bias voltage drift from the signal conditioners should be
investigated further and eliminated if possible. This could improve the control using
dSpace because lower corner frequencies could be used on the filters.

Implementation of this type of control in the prototype would require two sensors
and two actuators for each elevator cable, since vibration control in two directions would
be required. The actuators would travel with the elevator car and apply a force to the
cable. The required sampling times would be slower, since a 70 microsecond sample
period for the model corresponds to a 3.3 millisecond sample period in the real elevator.
The forces required would be much higher in the prototype elevator, since a 34 N
maximum model actuator force corresponds to a 1750 N actuator force in the prototype.
APPENDIX
ACROLOOP MOTION CONTROL PROGRAM

PROG 0
HALT
DETACH ALL
ATTACH MASTER0
ATTACH SLAVE0 AXIS0 "X"
MULT XI
RES X
PPU X20.32
ACC 700 DEC 700 STP 700
VEL 1000

90 VEL 3527 STP 0
100 ACC 712 DEC 712 X 0.00
110 ACC 2646 DEC 2646 X 0.01
120 ACC 5496 DEC 5496 X 0.03
130 ACC 8957 DEC 8957 X 0.11
140 ACC 12723 DEC 12723 X 0.24
150 ACC 16489 DEC 16489 X 0.48
160 ACC 19949 DEC 19949 X 0.85
170 ACC 22799 DEC 22799 X 1.36
180 ACC 24733 DEC 24733 X 2.05
200 ACC 25446 DEC 25446 X 198.49
300 ACC 24733 DEC 24733 X 207.39
310 ACC 22799 DEC 22799 X 216.47
320 ACC 19949 DEC 19949 X 225.73
330 ACC 16489 DEC 16489 X 235.15
340 ACC 12723 DEC 12723 X 244.69
350 ACC 8957 DEC 8957 X 254.33
360 ACC 5496 DEC 5496 X 264.03
370 ACC 2646 DEC 2646 X 273.78
380 ACC 712 DEC 712 X 283.56
390 ACC 83 DEC 83
400 X 1906.67
499 VEL 5
500 ACC 712 DEC 712 X 1916.44
510 ACC 2646 DEC 2646 X 1926.22
520 ACC 5496 DEC 5496 X 1935.97
530 ACC 8957 DEC 8957 X 1945.67
540 ACC 12723 DEC 12723 X 1955.31
550 ACC 16489 DEC 16489 X 1964.85
560 ACC 19949 DEC 19949 X 1974.27
570 ACC 22799 DEC 22799 X 1983.53
580 ACC 24733 DEC 24733 X 1992.61
600 ACC 25446 DEC 25446 X 2197.07
700 ACC 24733 DEC 24733 X 2197.95
710 ACC 22799 DEC 22799 X 2198.64
720 ACC 19949 DEC 19949 X 2199.15
730 ACC 16489 DEC 16489 X 2199.52
740 ACC 12723 DEC 12723 X 2199.76
750 ACC 8957 DEC 8957 X 2199.89
760 ACC 5496 DEC 5496 X 2199.97
770 ACC 2646 DEC 2646 X 2199.99
780 ACC 712 DEC 712 X 2200.00
790 ACC 83 DEC 83
800 X 2200.00
900 DWL 1
1000 ACC 700 DEC 700 STP 700
1010 VEL 1000
1020 X-85
1030 X0
1040 END
REFERENCES


