Payload Estimation for Electric Mining Shovels using a Load Sensing Pin

Slob, J.J. (Jelmer)

DCT 2007.052

Traineeship report

Coaches: Prof. P.R. McAree, The University of Queensland
         Dr. P.M. Siegrist, The University of Queensland

Supervisor: Prof. M. Steinbuch, Eindhoven University of Technology

Technische Universiteit Eindhoven
Department Mechanical Engineering
Dynamics and Control Technology Group

Commercial in confidence. Not for general distribution.

Brisbane, April, 2007
University of Queensland
## Contents

<table>
<thead>
<tr>
<th>Section</th>
<th>Pages</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>1  Payload Estimation</strong></td>
<td></td>
</tr>
<tr>
<td>1.1 Introduction</td>
<td>3</td>
</tr>
<tr>
<td>1.2 Electric rope shovel</td>
<td>4</td>
</tr>
<tr>
<td>1.2.1 Hoist Rope</td>
<td>4</td>
</tr>
<tr>
<td>1.3 Typical Production Cycle</td>
<td>4</td>
</tr>
<tr>
<td>1.4 Payload estimation for mining shovels</td>
<td>6</td>
</tr>
<tr>
<td>1.4.1 The continuous-discrete extended Kalman filter</td>
<td>7</td>
</tr>
<tr>
<td>1.4.2 The multiple model adaptive filter</td>
<td>8</td>
</tr>
<tr>
<td>1.5 Limitations</td>
<td>9</td>
</tr>
<tr>
<td><strong>2  Rope Force Measurement</strong></td>
<td>11</td>
</tr>
<tr>
<td>2.1 Introduction</td>
<td>11</td>
</tr>
<tr>
<td>2.2 Measuring Torque</td>
<td>12</td>
</tr>
<tr>
<td>2.3 Instrumented Clevis Pin</td>
<td>13</td>
</tr>
<tr>
<td>2.4 Implementation on P&amp;H-class mining shovels</td>
<td>16</td>
</tr>
<tr>
<td>2.5 Measuring strain gauge data</td>
<td>17</td>
</tr>
<tr>
<td>2.6 Calibration of the load-pin</td>
<td>18</td>
</tr>
<tr>
<td>2.7 Newton’s method</td>
<td>20</td>
</tr>
<tr>
<td>2.7.1 Remarks</td>
<td>20</td>
</tr>
<tr>
<td>2.7.2 First estimate</td>
<td>20</td>
</tr>
<tr>
<td>2.8 Analysis</td>
<td>20</td>
</tr>
<tr>
<td>2.8.1 Sensitivity to errors in alignment</td>
<td>24</td>
</tr>
<tr>
<td>2.9 Remarks</td>
<td>24</td>
</tr>
<tr>
<td><strong>3  Dynamic Modeling</strong></td>
<td>26</td>
</tr>
<tr>
<td>3.1 Introduction</td>
<td>26</td>
</tr>
<tr>
<td>3.2 The way it’s done</td>
<td>26</td>
</tr>
<tr>
<td>3.3 Adapting strain gauge measurements</td>
<td>29</td>
</tr>
</tbody>
</table>
List of Figures

1.1 P&H 4100A shovel ........................................ 3
1.2 Rope shovel overview ..................................... 5
1.3 Hoist ropes .............................................. 6
1.4 The multiple model adaptive filter .................... 7
2.1 Diagram of the sheave at the tip of the boom ........ 11
2.2 Instrumented load-pin .................................... 13
2.3 Strain gauges inside the load-pin ...................... 13
2.4 Load-pin as setup on the shovel ....................... 14
2.5 Electric scheme for one pair of strain gauges ....... 16
2.6 Bridge voltages ........................................ 17
2.7 Calibration setup ....................................... 18
2.8 Calibration data for varying L and β ................... 19
2.9 Payload Plus Approach ................................ 22
2.10 Lowpass filtered load-pin force and angle .......... 23
2.11 Error in alignment .................................... 24
3.1 Geometry and coordinate frames ...................... 27
3.2 Payload history calculated with load-pin data ....... 32
B.1 One rope-end parallel to x-axis ....................... 40
C.1 Calibration setup ..................................... 41
E.1 Lowpass filter .......................................... 43
F.1 Measured signals at the profibus .................... 44
List of Tables

A.1 P&H4100A load-pin specific specs .................................................. 37

D.1 Strain Gauge Input Signal Conditioner ............................................... 42
List of Algorithms

- Calculate $F = (P\beta)^T$ using a Newton-Raphson iteration.
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>Included hoist rope angle. Angle between the hoist ropes on either side of the sheave.</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Load angle. Angle between the x-axis of the load-pin and the applied net force $P$.</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>Nonlinear function: 2 dimensional lookup table that returns voltages $V$ for given $F$.</td>
</tr>
<tr>
<td>$\psi_1$</td>
<td>Angle of the hoist ropes at the cabin-side relative to the global x-axis.</td>
</tr>
<tr>
<td>$\psi_2$</td>
<td>Angle of the orientation of the load-pin relative to the global x-axis.</td>
</tr>
<tr>
<td>$\theta_1$</td>
<td>Shovel swing angle</td>
</tr>
<tr>
<td>$\theta_2$</td>
<td>Shovel pivot angle</td>
</tr>
<tr>
<td>$\theta_5$</td>
<td>Angle hoist rope makes to global x-axis</td>
</tr>
<tr>
<td>$\theta_c$</td>
<td>Crowd motor position</td>
</tr>
<tr>
<td>$\theta_h$</td>
<td>Hoist motor position</td>
</tr>
<tr>
<td>$\theta_s$</td>
<td>Swing motor position</td>
</tr>
<tr>
<td>$E_{x,y}$</td>
<td>Strain in x- and y-direction, used in load-pin.</td>
</tr>
<tr>
<td>$F$</td>
<td>Vector containing net load-pin force $P$ and load angle $\beta$.</td>
</tr>
<tr>
<td>$J$</td>
<td>Constraint Jacobian relating time derivatives of the configuration variables to the dependent variables</td>
</tr>
<tr>
<td>$k_t$</td>
<td>Torque constant for linear motor drives.</td>
</tr>
<tr>
<td>$K_{x,y}$</td>
<td>Constant gain that relates strain to force according to Hooke’s law.</td>
</tr>
<tr>
<td>$L$</td>
<td>Load applied in vertical direction during calibration of the load-pin.</td>
</tr>
<tr>
<td>$P$</td>
<td>Net load-pin force</td>
</tr>
<tr>
<td>$P_{x,y}$</td>
<td>Forces at in the load-pin in x- or y-direction of the strain gauges.</td>
</tr>
<tr>
<td>$R$</td>
<td>Electrical resistance</td>
</tr>
<tr>
<td>$r_s$</td>
<td>Sheave radius</td>
</tr>
<tr>
<td>$T_p$</td>
<td>Sheave torque</td>
</tr>
<tr>
<td>$V_b$</td>
<td>Bridge voltage of the load-pin in x or y-direction</td>
</tr>
<tr>
<td>$V_m$</td>
<td>Measured voltage of the load-pin at the profibus in x or y-direction</td>
</tr>
<tr>
<td>$\psi$</td>
<td>Vector of dependent kinematic variables</td>
</tr>
</tbody>
</table>
\( \theta \) Vector of configuration variables for shovel used for kinematics and dynamics

\( T_c \) Net torque applied by the crowd motor

\( T_h \) Net torque applied by the hoist motor

\( T_s \) Net torque applied by the swing motor

\( g \) Gravitational acceleration constant equal to 9.81 m/s\(^2\)
Abstract

Electric mining shovels are used in opencut mines to load dump-trucks with overburden (Figure 1.1). In order to operate as efficiently as possible, the dump-trucks should be filled, without being over- or underloaded. If a truck is not fully loaded, the available production capacity is not met. On the other hand, as trucks get overloaded, trucks get damaged and current site policies even require that overloaded trucks should be emptied right away and loaded again. This of course decreases the production rate and asks for a good estimation of the payload.

Currently static and dynamic models have been made to make on-shovel estimations of the payload using system models that have motor currents as their inputs. These models are limited by the transmission losses (i.e. friction). This causes the error level still to be too large. Since the tension in the ropes that support the bucket (Figure 1.2) has never been used as a source of information, a lot of information is neglected. As it is expected that this source might give a much better approximation of the payload, a load sensing pin has been developed for fitting to the boom point sheave. This pin allows the rope tension to be sensed.

This report addresses the use of this load pin for improving payload estimates. It starts with ways to measure torque in order to come to the adaption of the measured torque into current dynamic models. It is an extend to the PhD thesis of David Wauge [Wau07], who achieved to increase the accuracy of plus or minus 20% to 2%.
Chapter 1

Payload Estimation

1.1 Introduction

Before getting to the torque measurements at the sheave, first the P&H\textsuperscript{1} class mining shovels, of which one, the P&H4100A, is shown in Figure 1.1 will be introduced as well as the current status on payload estimation. Some elements of the shovel that are specifically needed in payload estimation (i.e. the ropes) are described in more detail.

\textbf{Figure 1.1:} P&H4100A shovel

\textsuperscript{1}P&H are a manufacturer of electric mining shovels, draglines and blast-hole drills and their machines are used in many Australian open-cut mines. P&H have manufactured more than 80% of the shovels currently operating around the world.
1.2 Electric rope shovel

All P&H class mining shovels have basically the same configuration, but are distinguished from each other by having different payload capacities. Figure 1.2 shows a shovel provided with terminology of its main parts. Some of the parts will briefly be discussed in the next paragraph. For this thesis experiments on the P&H4100A have been carried out. This machine has a dipper capacity up to $70\,m^3$ and is one of the larger shovels in the range. Specific specifications of this shovel can be found in the Appendix A.

Saddle & saddle point: The saddle is a joint assembly that allows the handle to freely pivot around the saddle point and by means of the crowd motor it is possible to translate the handle.

Boom: The structural support for the saddle, handle and dipper. The boom is pin jointed to the machinery house and is held up by the boom suspension cables.

Boom suspension cables: The boom suspension cables are cables of fixed length that hold the boom in its position at an angle of $45^\circ$.

Boom point sheaves: The hoist ropes wrap around the sheave. The sheave rotates freely over a clevis pin.

Handle: The handle is driven by the crowd motor that allows the bucket to be attracted and pushed out in what is called crowd direction.

1.2.1 Hoist Rope

A shovel is equipped with two hoist ropes. They both wind up on the same hoist drum, located in the machine house. The ropes on the P&H4100A approach the sheave from the cabin side at an angle of $33.5^\circ$ to the horizontal, in this thesis als referred to as the global x-axis. They wrap around the sheave and from there they go down to the bail pin where the left and right rope cross through the equalizer in order to equalize rope length at the bail pin position. An overview of how this is done is depicted in Figure 1.3.

“Eventually all wire rope in permanent service will fail.” [Wal04]. This certainly concerns the case of the rope shovels, where the ropes of a production shovel running 24 hours a day are replaced every 2 months. Rope selection is based on dynamic load. The causes of rope failure modes generally come from a combination of corrosion, fatigue, abrasive wear and excessive stress. Wear from abrasion, like fatigue, normally results from contact with sheave and drum and is the main cause of failure in this application.

Despite the strength of the selected cables, the hoist ropes change in length

$$\Delta L = \frac{\Delta F \cdot L}{A \cdot E} = \frac{G \cdot \Delta F \cdot L}{A \cdot E}$$

This change in length causes the rope to oscillate between the drum and the sheave. The effects it has on the proposed payload measuring system is discussed further on in Section 2.8.1.

1.3 Typical Production Cycle

Measuring payload on the truck is very hard. Current truck-scales only provide good information as the truck is driving and reaches second gear. Hence current measurements are inaccurate. For this reason on-shovel measurement is preferred.

A typical motion cycle consists of the following movements:
Figure 1.2: Rope shoval overview
Dig  Dirt is dug into the bucket

Swing  Shovel makes a swing towards the dump-truck

Dump  The dirt is dumped into the truck

Swing  Shovel makes an empty swing back to the digging area

The operator of the shovel controls these motions with two joysticks. The crowd and hoist motor drives are speed controlled. The swing motion is torque controlled. Commonly the latter of the three is used in a “bang-bang” mode. So full torque until a steady speed is reached and another “bang” to stop the shovel from swinging.

The payload is estimated during swing motion, because the swing motion because then the bucket carries the overburden and is not influenced by the peak forces that occur while digging. The bucket experiences the least dynamic load during swing. Still it is assumed that a static moment balance analysis will not satisfy.

1.4 Payload estimation for mining shovels

The existing technology for estimating payload on which this project builds is that proposed and evaluated in [Wau07] which uses a multiple model adaptive estimation (MMAF) scheme consisting of a bank of parallel Kalman filters and a conditional probability computation (hypothesis testing) as shown in Figure 1.4. The process models embedded within the Kalman filters all share the same model structure but incorporate different payload masses. The Kalman filters are provided with measurement sequence $z(t_i)$ and the input sequence $u(t_i)$, and each produces a state estimate $\hat{x}_k(t_i)$ sequence and an innovation sequence $r_k(t_i)$. The ensemble of the innovation sequences $r(t_i)$ give a relative indication of how close each model’s hypothesized payload is to the actual payload. These innovations are used by the hypothesis testing algorithm to assign relative probabilities $p_k(t_i)$ to each of the models and the converged probability distribution is used to estimate the most likely payload.
The process models used within the Kalman filters model the dominant rigid body dynamics of the rope shovel. The model inputs are the swing, crowd and hoist motor torques which are calculated using measurements of armature current and motor torque constant. The measurements are the swing, crowd and hoist motion (positions and velocities). A more detailed rationale for the use of this strategy is given in [Wau07].

### 1.4.1 The continuous-discrete extended Kalman filter

The Continuous-Discrete Extended Kalman Filter (CDEKF) is an extension of the widely used Kalman filter and described in detail in [Kal60, Bar88, Bro97, Gel74, May82a, May82b]. The discrete Kalman filter provides an optimal state estimate for discrete linear systems from discrete, linear measurements by minimizing the mean-squared error of the model output (the residual). The CDEKF extends the discrete, linear, basis of the Kalman filter to state estimation problems where the system dynamics are continuous and non-linear, using discretely sampled measurements.

To describe these non-linear problems, we define the state, $x$, and measurement set, $z$, as:

$$
\dot{x}(t) = f[x(t), u(t)] + Gw(t), \tag{1.1}
$$

$$
z(t_i) = Hx(t_i) + v(t_i), \tag{1.2}
$$

where $w$ is the process noise, and $v$ is the measurement noise, two independent, zero-mean, white,
Gaussian noise sequences, with covariances, $Q$ and $R$,

\[
E \{ w(t_i) w(t_j) \} = Q(t_i) \delta_{ij},
\]
\[
E \{ v(t_i) v(t_j) \} = R(t_i) \delta_{ij},
\]

where $\delta$ is the discrete delta function, otherwise known as the unit impulse. Note the system dynamics here is non-linear but the measurement equations are linear.

The CDEKF is founded in the linearization of Equations 1.3 and 1.4 about the current state estimate, and through a similar method to the derivation of the Kalman filter, the CDEKF can be derived \cite{Gel74}. The CDEKF is the best linear estimator of the corresponding non-linear system as measured by a minimum mean square error criteria, but the method is not without its problems. Despite this, the method is, in all probability, the most widely and most successfully used estimation method.

The calculations of the CDEKF are done in two stages. A prediction stage brings current state knowledge to the next time step so that it might be combined with measurements made at that time. The prediction step is summarized by the Equations 1.3 and 1.4.

\[
\dot{x}(t_i^-) = \dot{x}(t_{i-1}) + \int_{t_{i-1}}^{t_i} f(\dot{x}(t_{i-1})^+, u(t_{i-1})) dt,
\]
\[
P(t_i^-) = \Phi(t_i, t_{i-1}) P(t_{i-1}) \Phi(t_i, t_{i-1})^T + GQG^T,
\]

with

\[
\Phi(t_i, t_{i-1}) = \exp(F(t_{i-1}) \Delta t)
\]

\[
\Delta t = t_i - t_{i-1}
\]

\[
F(t_{i-1}) = \frac{\partial f(x,u)}{\partial x} \bigg|_{x(t)=\dot{x}(t_{i-1})^+, u(t)=u(t_{i-1})}.
\]

where $F$ is the linearized system matrix.

Equation 1.3 gives an estimate of the state $\hat{x}$, based upon the previous estimate of the state and the integration of the non-linear dynamics to the next time step. Equation 1.4 estimates the covariance, $P$, on that predicted state estimate.

The second stage of the calculation process is to update state estimates when new measurement data is available. The required calculations are given by Equations 1.5 to 1.9.

\[
r(t_i) = z(t_i) - H[\hat{x}(t_i)^-],
\]
\[
A(t_i) = H(t_i) P(t_i)^- H^T(t_i) + R(t_i),
\]
\[
K(t_i) = P(t_i)^- H^T(t_i) A(t_i)^{-1},
\]
\[
\dot{x}(t_i)^+ = \dot{x}(t_i)^- + K(t_i) r(t_i),
\]
\[
P(t_i^+) = P(t_i^-) - K(t_i) H(t_i) P(t_i^-).
\]

Here $r$ is the residual or innovation, giving the difference between the measurements and those predicted by our non-linear dynamic model, and $A$ is the covariance of the residual. $K$ is the Kalman gain.

### 1.4.2 The multiple model adaptive filter

The multiple model adaptive filter (MMAF), depicted in Figure 1.4, is an algorithm to provide accurate state estimation under parameter variation \cite{Mag65, May82a}. The algorithm additionally estimates
variable parameters and it is this property that is exploited in payload estimation, see \cite{Wau07}. The true parameter $M^*$ is assumed to be a member of a finite set of $n$ possible values, \{ $M_1, M_2, \ldots, M_n$ \}. A bank of Kalman filters is constructed with each filter based upon the hypothesis that $M^* = M_k$, $j = 1, 2, \ldots, n$. The filters run in parallel. The residual in each filter, $r_j$, is monitored and each hypothesis' conditional probability is calculated.

To summarize the implementation of the MMAF: given $n$ process models, each hypothesizing a specific payload, $M \in \{ M_j \}_{j=1}^n$ and a measurement set, $Z(t_i) = \{ z(t_0), z(t_1), \ldots, z(t_i) \}$ the probability of a residual $r_j(t_i)$ given a hypothesized mass $M_j$ and measurement sequence $Z(t_i)$ can be found from

$$ p (r_j(t_i) | M_j, Z(t_{i-1})) = \frac{1}{\sqrt{2\pi A_j(t_i)}} \exp \left( -\frac{1}{2} r_j(t_i)^T A_j(t_i)^{-1} r_j(t_i) \right) $$

where $A_j(t_i)$ and $r_j(t_i)$ correspond to the definitions given by Equations 1.5 and 1.6 for the filter hypothesizing the payload $M_j$.

This probability can be formed into a model likelihood through the hybrid pdf\footnote{Hybrid probability density function [pdf]: A pdf containing both continuous pdf’s and discrete probabilities.}

$$ \mu_j(t_i) = p (M_j | Z(t_{i-1})) = \frac{p (r_j(t_i) | M_j, Z(t_{i-1})) \mu_j(t_{i-1})}{\sum_{k=1}^n p (r_k(t_i) | M_k, Z(t_{i-1})) \mu_k(t_{i-1})} $$

where $\mu(t_0)$ is some initial discrete probability density series over the range $1 - n$. Wauge \cite{Wau07} assumes this initial distribution is uniform, i.e. $\mu_j(t_0) = \frac{1}{n}$.

We can refine the parameter estimate to a continuous resolution by taking the sum of the parameters from each model, weighted according to the corresponding residual probability:

$$ \hat{M}(t_i) = \sum_{j=1}^n M_j \mu_j(t_i). $$

We can also find an estimate of the variance in the parameter estimate by taking the square of the difference between the parameter sets of each model and the parameter estimate and multiplying through by the respective probability of that model:

$$ \hat{\sigma}_{M(t_i)}^2 = \sum_{j=1}^n \left( M_j - \hat{M}(t_i) \right)^2 \mu_j(t_i). $$

## 1.5 Limitations

When assessing measurements there are two relevant phenomena: precision and accuracy. Precision is the ability to reproduce consistently and accuracy is the mean distance to the true value. So precision is like a standard deviation and accuracy a mean value. Wauge improved the accuracy of the payload system from plus or minus 20\% to 2\%.
As mentioned in the introduction, the tension in the hoist rope is a potentially high source of valuable information in the process of payload estimation. Its largest benefit lies in the fact that the torque in the hoist motor can be replaced by the force in the rope. Currently the hoist torque $T_h$ is calculated as the hoist armature current multiplied by a torque constant $k_t$. Leaving out the hoist torque leaves out the complete hoist drive, with all its nonlinearities which are hard to model such as friction and other dynamics as well as inertia and transmission ratios, which are hard to predict.

The aim is to improve the accuracy as well as the precision. The reason to do this is to see what improvement is possible. Although little improvement seems to be possible, one should consider the payload capacity of the shovel. A large shovel has a dipper capacity up to 110t, which comes down to a fluctuation of 2.2t for the current payload systems. In order to get these results, the torque at the sheave should be measured more accurate than the estimation of the hoist torque at the hoist drive. How this is done is discussed in the following chapters. Chapter 2 introduces a method to measure rope force and Chapter 3 concerns the adaptation of the new measurements into the dynamic models.
Chapter 2

Rope Force Measurement

2.1 Introduction

In order to measure the rope force, first a suitable sensing method and sensing location have to be determined. When choosing such a method and location one should consider the operating conditions, accessibility, dynamic loads, vulnerability. This will be discussed in this chapter as well as the results of the chosen measurements.

2.2 Measuring Torque

The most reliable and direct measurements are obtained as close to the source as possible, which is the bucket for the purpose of load weighing. Therefore the forces at the boom tip sheave pin are considered. This comes down to the situation as sketched in Figure 2.1.

![Diagram of the sheave at the tip of the boom](image)

**Figure 2.1**: Diagram of the sheave at the tip of the boom

Generally torque can be classified into two subcategories: static and dynamic torque [Scho6]. The
torque in the sheave is considered to be static, although the inertia of the sheave itself might indicate otherwise (see Appendix A). The inertia of the sheave will dampen out dynamic torque effects produced by the hoist drive and in doing so it supports reliability of the static measurements. On the other hand, payload measurement takes place during swing motion, where generally the hoist motion is limited to a couple of meters and moves gradually. This supports the assumption that the torque should be considered to be static.

Another distinction in torque measurement can be made and that is reaction versus in-line measurements [Scho6]. The in-line method is very accurate because the sensor is placed between torque carrying components and torque is directly measured. For static torque measurement both in-line and reaction torque measurement will yield in the same results. Due to the enormous loads that are present in this application, measurement of reaction torque is required because of economical as well as practical thoughts. In-line torque measurement becomes complicated as it concerns rotational torque, because the rotational sensor has to be connected to the stationary world.

Torque is just a rotational force, that is a force through a certain distance. Here, in Figure 2.1 $T_h$ represents the torque in the hoist drum generated by the hoist drive. $T_p$ is the torque at the boom tip from the ropes wrapped around the sheave. Obviously these two torques (or forces) are equal to each other. However the torque in the drum is measured through the armature current (with all complications as mentioned in section 1.5) and the aim is to measure the static force $T_p$ using a rather direct approach using reaction force measurement.

Force can be measured in various ways. A very common way of measuring force is done with the use of strain gauges. One way of achieving this is to measure the reaction force on the clevis pin that supports the sheave and transmit the signal through a Wheatstone bridge to the channels of the profibus.

### 2.3 Instrumented Clevis Pin

A pin as described in the previous section is patented by A.U. Kutsay[1] in [Kut72]. It is an extension of the strain gauge apparatus patented in [Kut68], where a load cell in the shape of disc is discussed and the way the gauges are positioned. The instrumented clevis pin (here often shortened to load-pin) is a load cell application that aims to improve strain detection for machine junctions employing pin members that are subjected to shear stress. A pin is subjected to a shear force by placing it into i.e. a clevis and yoke combination. This is typically the situation at the boom tip sheave.

The load-pin (Figure 2.2) has short unsupported zones of slightly decreased diameter, to ensure the shear stress concentrates in this area. That is the place where the strain gauges will be inserted into the pin as depicted in Figure 2.3. Note that the top-end plate drawn in Figure 2.2 is used for calibration and is not used on the shovel. Its use is clarified in Section 2.6.

Area reduction increases the sensitivity, but it diminishes the strength of the pin. There’s a trade-off. Usually reductions of 10-30% are used. The length of these zones should be adequate to provide a nearly constant stress zone, preferably 2-4 times larger than the gauge length. Although here’s a trade-off to because of the loss of strength again as the length of the zones increases.

Strain gauges are placed inside the pin and measure the strain due to the applied force at the zones of decreased diameter. The gauges are close to the neutral axis as it concerns shear stress, to ensure maximum reliability of the measurements.

A. Yorgiadis[2] addresses two cases in his paper [Yor86]. Case (1) is along solely one axis and (2) is along two orthogonal axes (Figure 2.3).

---


1. **Constant wrap angle**: Determining the reaction force acting on the pin and hence the tension in the rope, using a set of 2 strain gauges (opposite in diametrical alignment at right angles to the known direction of the force).

2. **Variable wrap angle**: Determining the reaction force acting on the pin and hence the tension in the rope as well as the wrap angle using a set of 4 strain gauges ($2 \times X$, $2 \times Y$), where one of the ends of the rope is parallel to the x-axis.

### 2.4 Implementation on P&H-class mining shovels

The angle of the boom is fixed by the boom suspension cables, also referred to as gantry ropes, attached to the machinery house and doing so this also fixes the position of the sheave. This also fixes angle $\psi_1$ in Figure 2.4 of the hoist rope from drum to sheave with respect to the global x-axis. The other end of the hoist rope varies with the displacement of the handle as the crowd motor is actuated. Only case (2) would be capable in satisfying the needs for payload estimation at the rope shovel. An extra benefit of supplying the load-pin with two pairs of strain gauges (X and Y) is the ability to determine not only the magnitude of the reaction force, but also the direction of the force and through this determine the angle of wrap from the load-pin. The included hoist rope angle can be used to check the results in the design phase of this project, but is potentially capable of supplying information/data in the geometric calibration process of the shovel.

When using an instrumented clevis pin at the boom tip of P&H-class mining shovel, one should consider that the loads are very high compared to the applications from [Yor86]. For these heavy load designs, the zones of reduced diameter should be of relatively short length and the diameter needs little reduction. This has the disadvantage that non-linear effects will increase and that the repeatability of the measurement loses its probability. The accuracy of the measurements will suffer from these heavy load conditions.

---

* *A more detailed explanation of the geometry and kinematics of the shovel can be found in chapter [1].*
Another reason why diameter reduction of the pin should not be too large, is that the strength and lifetime are very important in this application. Safety issues prohibit this, as well as the complexity of (dis)assembling the sheave on site once the shovel is in production.

The clevis pin is almost continuously subjected to load cycles during its lifetime. Even in steady state and while carrying an empty bucket the pin is subjected to approximately 50% of the load of a full bucket, which is a considerably large amount. Because of the continuous loads during service life it is of great import to use large fillet radii at the unsupported zones of the pin in order to minimize the bending stress concentration. This comes down to less diameter reduction, or reduction over a longer distance.

For the purpose of payload estimation P&H have implemented strain gauges in the sheave pins of some of their shovels. The pin at the P&H4100A has a diameter reduction of 5.1% and the zones are only 44.45mm with a 19.05mm radius at the reduction. The geometry of the load-pin as well as the forces acting on the pin are depicted in Figure 2.4.

![Figure 2.4: Load-pin as setup on the shovel](image)

The x-axis of the load-pin, that is the direction of $P_x$ and the x-pair of gauges, is not parallel to the global x-axis, which is depicted in Figure 2.4. The pin is mounted at the boom tip at an angle $\psi_2$ of 15°. Figure 2.4 can be simplified taking into account that $\phi = \psi_1 - \psi_2$. This simplification aligns the x-axis of the load-pin to the global x-axis, this can be seen in Figure 2.4. Simplifying the layout is not
required but makes the calculations later less complicated.

\[
\begin{align*}
\psi_1 &= 33.496^\circ \\
\psi_2 &= 15^\circ \\
\phi &= \psi_1 - \psi_2 \\
&= 18.496^\circ
\end{align*}
\]

The strain gauges give an indication of the forces the pin is subjected to in x- and y-direction. From these forces the angle \( \beta \), relating the net force \( P \) to the local x-axis of the load-pin, can be calculated.

\[
\beta = \tan^{-1}\left(\frac{P_y}{P_x}\right)
\]  

The diagram (Figure 2.4) shows the included hoist rope angle \( \alpha \) defined as

\[
\alpha = 2(\beta - \phi)
\]

Which can easily be seen from Figure 2.4. This is not the wrap angle as mentioned in [Kut72 Yor86]. Though the wrap angle \( \theta \) and the included hoist rope angle are related through

\[
\theta = 180^\circ - \alpha
\]

The net force on the pin \( P \) can be calculated as \( P = P_x^2 + P_y^2 \). In order to derive the rope force \( T_p \), force balances with respect to the global x- and y-axis should be taken of the forces in Figure 2.4. Doing so results in the following derivation

\[
\begin{align*}
P_x &= T_p \cos \phi + T_p \cos (\phi + \alpha) \\
P_y &= T_p \sin \phi + T_p \sin (\phi + \alpha)
\end{align*}
\]

Squaring the forces gives a single relation of \( T_p \) and \( \alpha \). \( P_x, P_y \) and \( \phi \) are measured or given.

\[
\begin{align*}
P^2 &= P_x^2 + P_y^2 = 2T_p \left( T_p + T_p \cos \alpha \right) \\
&= 2T_p P_x \left( \frac{1 + \cos \alpha}{\cos \phi + \cos (\phi + \alpha)} \right)
\end{align*}
\]

And finally an expression for \( T_p \) is found, for given angle \( \alpha \)

\[
T_p = \frac{P_x^2 + P_y^2}{2P_x} \left\{ \frac{\cos \phi + \cos (\phi + \alpha)}{1 + \cos \alpha} \right\}
\]
2.5 Measuring strain gauge data

Section 2.4 deals with measures to obtain a rope force and included hoist rope angle from the load-pin forces in x- and y-direction. But the load-pin doesn’t measure forces. The strain gauges inside the pin (Figure 2.2) change in electrical resistance as they are deformed. The returned voltages over the resistance (the strain gauge) give an indication of the related force since strain and electrical resistance are linearly related \[ \varepsilon = \frac{\Delta l}{l} \sim \frac{\Delta R}{R} \] (2.6)

The change in resistance is very low. Therefore the strain gauges are setup in a Wheatstone-bridge configuration. The bridge is subjected to a 10V signal. Due to the change of electrical resistance of the gauges as they are stretched or compressed, the strain gauge input signal conditioner receives an input signal \( V_b \) (IN-, IN+) of 0-20mV (2.6). The signal conditioner is equipped with a current source and excites a 4 to 20 mA current according to the input, into the electric circuit on the right hand side of Figure 2.5. Over a 250Ω resistor this comes down to 1 to 5V, using Ohm’s Law, which is read into the analog channels of the profibus (Process Field Bus) and stored in data files.

\[ V_b = \left( \frac{V_m}{R} - i_{\text{min}} \right) \frac{V_{m,\text{max}}}{(i_{\text{max}} - i_{\text{min}})} \]

These voltages give an indication of the strain at the pin. The reaction force at the pin is defined as strain times a constant gain for a perfect linear spring according to Hooke’s law \[ \text{Mar00} \]. Section 2.7 deals with the solution to this problem.

The bridge voltages \( V_b \) [in mV], measured at the strain gauges, that go into the signal conditioner are shown in Figure 2.6. The blue line represents the voltage of the strain gauge pair in x-direction and the green line for y-direction. The noise level of the shovel being in a steady position (40-60s and 75-90s) is the same for x- and y-direction and about 8.5 to 10%.
2.6 Calibration of the load-pin

The load-pin is calibrated by applying a load in vertical direction for different orientations of the strain gauges relative to the global origin. The orientation is changed using the plate that is mounted onto the pin as shown in Figure 2.2. This plate is removed when mounted on the shovel.

In order to account for hysteresis and non-linear effect, the load is stepwise and increasingly applied until a certain maximum value and then decreased in the same number of steps. The actual calibration drawings are shown in Appendix C, but Figure 2.7 gives a much better idea in perspective of the previous diagrams of the sheave.

The 4 figures in Figure 2.8 show the calibration results. The figures on the left are the same as the one on the right, which implicates that no, or hardly any hysteresis is measured. The figures on the top are measurements of the strain gauges in x-direction and the ones on the bottom for y-direction.

On each figure the voltages are displayed for a combination of load and load angle. From the
isoclines on the ground level of each figure one can conclude that the surfaces are curved. This means that no linear relation exists between voltage and force/angle-combination. The influence of the angle should be investigated and is done in section 2.7.

2.7 Newton’s method

As mentioned in Section 2.6 it is not possible to give a linear relation between the measured voltage and the force in that same direction \( P_x = K_x \times V_x \) and \( P_y = K_y \times V_y \). The influence of the load angle \( \beta \) is too significant to ignore. Though it has been tried in order to give an estimation of the forces calculated from load-pin data using a least squares method:

\[
\begin{align*}
P_x &= K_x E_x \\
P_y &= K_y E_y
\end{align*}
\]

Force balances at the calibration setup (Figure 2.7 and C.1 where a load \( L \) is applied in vertical direction) in x- and y-direction comprise the following relations

\[
\begin{align*}
\sum F_x &= P_x \cos(\pi/2 - \beta) - P_y \cos \beta = 0 \\
\sum F_y &= -L + P_x \cos(\pi/2 - \beta) + P_y \cos \beta = 0
\end{align*}
\]

for \( L = -2T_p = P \)

The relations above are represented in matrix form in order to get a linear set of equations, which can be solved using a least squares method [Kre93].

\[
\begin{pmatrix}
E_{x,ij} \sin \beta_i \\
E_{x,ij} \sin \beta_i \\
E_{y,ij} \sin \phi_i \\
E_{y,ij} \sin \phi_i
\end{pmatrix}
\begin{pmatrix}
K_x \\
K_y
\end{pmatrix}
= 
\begin{pmatrix}
0 \\
L_{ij}
\end{pmatrix}
\]

(2.8)
\[ Ek = l \]
\[ E^T Ek = E^T l \]
\[ k = (E^T E)^{-1} E^T l \]

The result \( k = (K_x K_y)^T \) is a least squares solution to the linear set of equations. The solution for which \( Ek - l = 0 \) is called an exact to solution. To determine whether \( k \) is an exact solution simply compute \( Ek - l \) \[ \text{Mat06} \].

As expected there exists no exact solution for this set of equations. Solving this set of equations gives a rough estimate of the gains \( K_x \) and \( K_y \), but no solution to get forces related to the bridge voltages.

An alternative to get the pin force \( P \) and load angle \( \beta \) from the voltages is to use the calibration data as a lookup table. The pin force and load angle are denoted by \( P \) and \( \beta \) and collected in the vector \( F = (P \ \beta)^T \). The bridge voltages in x- and y-direction are stored in the vector \( V = (V_{b,x} \ V_{b,y})^T \). Two calibration surfaces \( (X \text{ and } Y) \) give the bridge voltages for applied forces and angles, i.e. there exists a function of the calibration curves of the form \( V = \Phi(F) \). This function is a lookup table that
interpolates bilinearly between the data points from the calibration data.

To determine the combination of pin force and load angle, \( F \), associated with specific voltages, \( V \), a solution for the roots of

\[
f(F) = \Phi(F) - V = 0
\]

has to be determined. In order to do so a multi-variable discrete Newton's method [Kre93] (also known as the Newton-Raphson method or the Newton-Fourier method) is considered. This method is known to be an efficient algorithm to find approximations to the roots of real-valued functions. The finally obtained algorithm is fully shown in Algorithm 1. The heart of this application of Newton’s iteration method is

\[
F_{\text{new}} = F - J^{-1}(V - [V_x V_y]^{T})
\]

where \( F \) is being updated. \( V \) is the result of the function \( \Phi(F) \). \( V_{b,x} \) and \( V_{b,y} \) are the bridge voltages and inputs to the algorithm. The \( 2 \times 2 \) Jacobian \( J \) is determined by numerical differentiation of the non-linear function \( \Phi(F) \) over a small step \( \Delta F_1 \) and \( \Delta F_2 \), which can be determined from the calibration data.

### 2.7.1 Remarks

The termination criteria for this application are the amount of convergence to the exact solution or until a certain and on the other hand a maximum number of iterations. The amount of convergence is set as the norm of the calculated bridge voltages minus the reference bridge voltages.

Whether \( F = (P, \beta) \) is actually updated depends on the boundaries of the calibration grid, spanned by the pin force and load angle, and is shown in the last two if-statements of the algorithm. If the estimate of \( F_{\text{new}} \) lies outside the calibration grid, that part of \( F \) will not be updated and the step size in that direction will be halved. This phenomena is also called damping and this algorithm the Damped Newton-Raphson method.

### 2.7.2 First estimate

Force is linearly related to the strain. Therefore the gains \( K_x \) and \( K_y \) derived from the least squares method mentioned earlier in this section is used to make a first estimate of the force and angle in the Newton-Raphson iteration method. When this estimate lies outside the calibration grid this point will be pushed inside the grid, 2% from the boundary.

### 2.8 Analysis

In order to obtain a rope force and angle of wrap from the data from the strain gauges the following approach is used. From the profibus output voltages \( V_{m,x} \) and \( V_{m,y} \) are collected. In order to compare these voltages to the bridge voltages from the calibration data, they are converted to bridge voltages \( V_{b,x} \) and \( V_{b,y} \) in stage I (see Figure 2.9 and 2.5). The next stage is the Newton-Raphson iteration. Here the net force on the pin \( P \) and the angle of this force with respect to the global x-axis \( \beta \) are solved for a combination of two bridge voltages \( V_{b,x} \) and \( V_{b,y} \). Finally, using force balances and geometry calculations, the rope force \( F_p \) and included hoist rope angle \( \alpha \) are calculated in stage III.

Figure 2.10 shows the results for rope force and included hoist rope angle for the input voltages shown previously in Figure 2.6. As expected a similar noise level can be seen as in the input voltages.
The noise level is about 5-6% for the forces and 8% for the load angles. The noise at the included hoist rope angle is approximately twice as high as the load angle, due to its definition (Equation 2.3).

From known weights in the right angled configuration comparisons are made to the assumed forces according to the load-pin. The difference in force is about 6% on average and the angle deviates by 5.5%. Because the shovel’s right angled configuration was a manual estimate, this information doesn’t give a reliable comparison, but confirms the credibility of the calculations.

Besides the load-pin forces, there is also a red line drawn in Figure 2.10. This is the rope force calculated from the armature and field current from the hoist drive. That is the input of the load weighing system designed in [Wau07]. The P&H4100A rope shovel has two hoist motors and a gear ratio from drum rotation to hoist extension of 66.19. A bilinear lookup-table gives the hoist torque of the hoist drive. Multiplication of this torque with the number of drives and the gear ratio gives the

---

4In the right angled configuration the handle is horizontal and the bail pin is situated right under the sheave.
hoist rope force.

\[ F_h = 2 \times 66.19 \times T_h(i_{a,h}, i_{f,h}) \]

Conclusions can be drawn that the load-pin gives a relatively smooth signal, despite the spiky behavior. The hoist force calculated from the hoist drives is much worse. Especially in cases where the operator switches on the hoist brakes. In that case the armature current drops to zero and gives a zero hoist force as an input to the load weighing system.

The spiky behavior of this noise intuitively asks for a low pass filter. A finite impulse response (FIR) Kaiser-window based, linear-phase filter is used with a normalized cutoff frequency at \( \omega_n = 18 \text{Hz} \). The filter is normalized so that the magnitude response of the filter at the center frequency of the passband is 0 dB [Opp99, Mat06]. Doing so, the filter can directly be applied as a gain to the load-pin signals. The normalized gain of the filter at \( \omega_n \) is -6 dB.

Increasing the order of the filter results in a phase lag. A suitable order is found with 8 poles, with a negligible phase lag for this application of merely 0.05s. The only trade-off might be the startup behavior, whereas the filtered signal by definition starts at zero. The startup problems only last the 0.05 sphase lag.

The low pass filter increases the quality of the signal as the noise level reduces to 3% for the forces and 5.5% for the load angles. Figure 2.10 shows the filtered signal on top of the non-filtered signal in the time domain, represented by the lighter colored lines. Although the filter is here applied to off-line data, it is possible to use the FIR filter for real-time applications.
Figure 2.10: Lowpass filtered load-pin force and angle
2.8.1 Sensitivity to errors in alignment

Orientation and fixation of the pin seems very important because the rope force is calculated as a function of the included hoist rope angle $\alpha$ (see Equation 2.5). Where $\alpha$ is defined as a function of the varying load angle and the fixed angles $\phi = \psi_1 - \psi_2$. Clearly an error in these fixed angles will effect the included hoist rope angle twice as much. But the effect on the rope force is more complicated. Figure 2.11 shows the rope force with an error $\Delta \phi$ relative to the same force without an error in the fixed angle, for $\Delta \phi$ between $\pm 5^\circ$. The different lines are for different loads applied in y-direction while the force in x-direction is kept static. The arrows indicate increasing values of the load. The lines, that appear to be linearly, are small intersections of highly non-linear functions as Equation 2.5. The errors become worse as the y-load is increased. If the pin is misaligned for $-5^\circ$, this comes down to an overestimation of the rope force of at least 10% in the worst case scenario of Figure 2.11. In a more general sense, the drawn error lines vary between 0.4-2.0% per degree. Fortunately errors of caused by misalignment during mounting of the pin ($\psi_2$) can be encountered for, since they will not change in time. The rope force angle $\psi_1$, however, can vary as the cable oscillates. This makes the error hard to predict and hence hard to compensate for. Experiments will have to point out whether this indeed is a relevant source of error.

![Figure 2.11: Error in alignment](image)

2.9 Remarks

This section deals with aspects of the design of the load-pin that have not been investigated, but which might be the cause of errors in the load angle and the net pin force.

**Temperature** Although the circuit typically includes temperature compensation for temperature changes [Strainsert catalog], it is recommended to examine the differences of the calibration temperature and the operating temperature and the influence on the strain in the pin.

**Fatigue** Although the direction of the force on the pin is merely always the same (about $55^\circ$), the effects of fatigue should be examined.
**Plastic deformation** Because the direction of the force on the pin is always the same, it is not unlikely that the pin tends to deform plastically.

**Cross-talk effects** The effects of cross-talk can simply be overcome by twisting the wires [Yor86], so special investigation on this subject doesn’t seem very important.
Chapter 3

Dynamic Modeling

3.1 Introduction

The previous chapters dealt with the measurements at the load-pin. Now this data can be obtained, it should be adapted to the dynamic model that is used to estimate the payload. First the current situation is introduced followed by several ways to implement the load pin measurements on the load weighing system.

3.2 The way it’s done

In order to come to a linear SISO state space model of the shovel dynamics [Wau07] first the kinematics of the shovel will be revealed.

Figure 3.1 shows the parameters and reference frames used to describe the geometry of P&H-class electric mining shovels. Lengths labeled $l$ and angles labeled $\phi$ are fixed by design; lengths labeled $d$ and angles labeled $\theta$ vary under machine motion. The derivation of relationships between the different frames using homogeneous transformation matrices can be found in [Wau07]. A set of independent variables, is chosen as the swing angle $\theta_1$, the pivot angle $\theta_2$ and the crowd extension $d_3$. They are collected in the vector $\theta$. Constraint equations relate the independent variables to the dependent variables $\psi = (\theta_s, \theta_c, \theta_h)$. They represent the displacement and rotation of the body fixed frames by the crowd, swing and hoist drive. The constraint equation is defined through the transmission ratios and the kinematics.

$\theta = J\psi$

Using the independent variables $\theta$ the Lagrange method is applied to write down the system’s equations of motion.

$$\frac{d}{dt} \left( \frac{dT}{d\theta} \right) - \frac{dT}{d\theta} = Q^{nc}$$

$$A\dot{\theta} + B\theta_{ii} + C\dot{\theta}_{ij} = \gamma + \tau$$

where $\gamma$ and $\tau$ together are the non-conservative forces. The vector $\gamma$ is created from the kinematic constraint equations and $\tau$ are the torques in $\theta_1$, $\theta_2$ and $d_3$ direction. Coupled to the motor drive torques $T$ through the Jacobian developed in the derivation of the constraint equations.

Besides the mechanical also the electrical dynamics of the drives are modeled, to relate the operator’s input to a displacement of the swing, crowd and hoist drives.
\[ L \frac{dI_a}{dt} + RI_a + K_v \frac{d\psi}{dt} = V \]

This is a general model of an electric drive with an electric motor, resistance and XXXcapacitor connected in series, as can be found in [Fra94]. Rewriting this equation results in an expression in terms of the dependent variables

\[ \frac{d\psi}{dt} = K_v^{-1} \left( V_a - L \frac{dI_a}{dt} - RI_a \right) \]

and the independent variables using the constraint equation

\[ \dot{\theta} = JK_v^{-1} \left( V_a - L \frac{dI_a}{dt} - RI_a \right) \]

For the state space definition the states \( x \) are defined as the independent variables \( \theta \) and their time derivatives \( \dot{\theta} \). The input \( u \) is chosen to be the torques as a function of the armature and field currents.

\[
\begin{align*}
  x &= \left( \begin{array}{c} \theta \\ \dot{\theta} \end{array} \right)^T \\
  u &= \left( T(I_a, I_f) \ g \right)^T \\
  y &= \left( \theta \ \dot{\theta} \ (I_a, \frac{dI_a}{dt}, V_a) \right)^T 
\end{align*}
\]

The system is written down as a time-linear system in state space

\[
\begin{align*}
  \dot{x} &= Fx + Bu \\
  y &= Cx
\end{align*}
\]

With the following system matrices

\[
\begin{align*}
  F &= \begin{pmatrix}
  \frac{\partial \dot{\theta}}{\partial \theta} & \frac{\partial \dot{\theta}}{\partial \ddot{\theta}} \\
  \frac{\partial \dot{\theta}}{\partial \ddot{\theta}} & \frac{\partial \dot{\theta}}{\partial \dddot{\theta}}
  \end{pmatrix} = \begin{pmatrix}
  0 & I_3 \\
  \frac{\partial \dot{\theta}}{\partial \theta} & \frac{\partial \dot{\theta}}{\partial \ddot{\theta}}
  \end{pmatrix} \\
  B &= \begin{pmatrix}
  \frac{\partial \dot{\theta}}{\partial T} & \frac{\partial \dot{\theta}}{\partial \gamma} \\
  \frac{\partial \dot{\theta}}{\partial \gamma} & \frac{\partial \dot{\theta}}{\partial \eta}
  \end{pmatrix} = \begin{pmatrix}
  0 & 0 \\
  \frac{1}{A^{-1}} \frac{\partial \tau}{\partial T} & \frac{1}{A^{-1}} \frac{\partial \gamma}{\partial \eta}
  \end{pmatrix} \\
  C &= \begin{pmatrix}
  I_3 \\
  0 \\
  0
  \end{pmatrix}
\end{align*}
\]

The system matrix is not completely written down in detail as it doesn’t really serve the purpose at this stage for this application. Wauge provides a detailed derivation in [Wau07].

The interesting variables that are needed as output \( y \) are the states and therefore the matrix \( C \) is equal to the identity matrix, to ensure that the output solely and directly depends on the states.

\[
y = Cx = x = \left( JFRK_v^{-1} \left( V_a - L \frac{dI_a}{dt} - RI_a \right) \right)
\]
3.3 Adapting strain gauge measurements

The linear system changes as an extra measurement equation for the torque $T_p = r_s F_p$ is added. There are several ways to adapt these measurements. The (dis)advantages of certain methods is discussed in the following sections.

3.3.1 Method I

\[
\begin{align*}
\dot{x} &= Fx + Bu \\
y &= Cx + Du
\end{align*}
\]

Because an extra measurement is to be adapted, one could consider this to be an extra input to the system and therefore extend the state space equation with an extra input in $u$. To extract the information, the output equation of the system could be equipped with another matrix $D$.

The vectors that describe the system become

\[
\begin{align*}
x &= (\theta \hspace{1mm} \dot{\theta})^T \\
u &= (T \hspace{1mm} T_p \hspace{1mm} g)^T \\
y &= (\theta \hspace{1mm} T_p \hspace{1mm} \dot{\theta} (I_a, \frac{dt_a}{dt}, V_a))^T
\end{align*}
\]

Accordingly to the changes, the matrices become

\[
F = \begin{pmatrix}
\frac{\partial \dot{\theta}}{\partial \theta} & \frac{\partial \dot{\theta}}{\partial \theta} \\
\frac{\partial \dot{\theta}}{\partial T} & \frac{\partial \dot{\theta}}{\partial T} \\
\frac{\partial \ddot{\theta}}{\partial \theta} & \frac{\partial \ddot{\theta}}{\partial \theta} \\
\frac{\partial \ddot{\theta}}{\partial T} & \frac{\partial \ddot{\theta}}{\partial T}
\end{pmatrix} = \begin{pmatrix}
0 & I_3 \\
0 & 0 \\
A^{-1} \frac{\partial \tau}{\partial T} & 0 \\
A^{-1} \frac{\partial \tau}{\partial T_p} & A^{-1} \frac{\partial \gamma}{\partial g}
\end{pmatrix}
\]

\[
B = \begin{pmatrix}
\frac{\partial \dot{\theta}}{\partial T} \\
\frac{\partial \dot{\theta}}{\partial T} \\
\frac{\partial \ddot{\theta}}{\partial \theta} \\
\frac{\partial \ddot{\theta}}{\partial \theta}
\end{pmatrix} = \begin{pmatrix}
I_3 \\
0 \\
0 \\
0
\end{pmatrix}
\]

\[
C = \begin{pmatrix}
I_3 \\
0 \\
0 \\
0
\end{pmatrix}
\]

\[
D = \begin{pmatrix}
0 & 0 & 0 \\
0 & I_3 & 0 \\
0 & 0 & 0
\end{pmatrix}
\]

When looking at the matrices the problem with the B-matrix becomes clear as $\tau$ depends on $T_p$, but their relation is not clearly defined. The possibility to use both $T_h$ and $T_p$ is attractive as it is not yet clear how the load-pin measurements will affect the payload estimation.

3.3.2 Method II

\[
\begin{align*}
\dot{x} &= Fx + Bu \\
y &= Cx
\end{align*}
\]

A very easy and intuitively logical approach is directly replacing the torque measured from the hoist armature and field currents by the torque at the sheave.

$T_h \leftarrow T_p$
The only thing that changes with respect to the original situation is the input vector. The torques in the input become \( T = (T_s, T_c, T_p, 0) \). So the \( T \) is now a function of the armature currents as well as the load pin voltages. The output equation remains the same.

The state, input and output vectors hardly change, except for a different calculation of the third element of \( T \). The input \( u \) becomes dependent of the load-pin voltages and no longer of the hoist drive currents. This is a very direct approach and very easy to implement on the current load weighing system.

\[
x = (\theta, \dot{\theta})^T
u = (T(I_a, V_e, V_p) g)^T
T = (T_s, T_c, T_p, 0)^T
y = Cx = x = \left( J K_v^{-1} (V_a - L \frac{dI_a}{dt} - RI_a) \right)
\]

State space matrices

\[
F = \begin{pmatrix} \frac{\partial \dot{\theta}}{\partial \theta} & \frac{\partial \dot{\theta}}{\partial \theta} \\ \frac{\partial \dot{\theta}}{\partial \theta} & \frac{\partial \dot{\theta}}{\partial \theta} \end{pmatrix} = \begin{pmatrix} 0 & I_3 \\ \cdots & \cdots \end{pmatrix}
B = \begin{pmatrix} \frac{\partial \dot{\theta}}{\partial T} & \frac{\partial \gamma}{\partial g} \\ \frac{\partial \dot{\theta}}{\partial T} & \frac{\partial \gamma}{\partial g} \end{pmatrix} = \begin{pmatrix} 0 & 0 \\ A^{-1} \frac{\partial \tau}{\partial T} & A^{-1} \frac{\partial \gamma}{\partial g} \end{pmatrix}
C = \begin{pmatrix} I_3 \\ 0 \end{pmatrix}
\]

### 3.3.3 Method III

When the system is expected to have measurement noise \( v \), the system is represented as

\[
\begin{align*}
\dot{x} &= Fx + Bu \\
y &= Cx + Du + v
\end{align*}
\]

The system equation remains identical to the original one. The output equation is extended with an extra relation in the last row

\[
T_p = kT_h + v
\]

where the aim is to minimize \( v \) and find a gain \( k \) that passes through the relevancy of \( T_p \) relative to \( T_h \).

The vectors in the system are all the same except for \( y \)

\[
x = (\theta, \dot{\theta})^T
u = (T, g)^T
y = (\theta, \dot{\theta} (I_a, \frac{dI_a}{dt}, V_a) T_p)^T
\]

The problem with this method might be the practical aspect. Especially in the situation when the hoist brakes are activated. In that case \( T_h(i_a, i_f) \) is much less than \( T_p(V_b) \), which makes it impossible to minimize the noise \( v \).

### 3.3.4 Method IV

One can also consider the torque at the load-pin to be another state of the system. The linear system equations remain the same it is just the state vector \( x \) that is extended with the measurements \( T_p \),
and the time-derivative \( \dot{T}_p \) of it. The output relation can remain the same and the C-matrix remains the identity matrix. This way the measurements are directly fed to the output of the system. The problem with this option is the time-derivative of the torque \( \dot{T}_p \), because it doesn’t exist. It might be seen as measurement noise. The largest disadvantage is the fact that this method changes the system dramatically since the states are the basics of the system. This cancels out the practical part of it.

### 3.4 Implementation

#### 3.4.1 Conclusion

Considering the enormous impact on the system changing, it is desired to have as little changes as possible in order to reduce errors while developing the new payload system. Because of its simplicity Method II is preferred to the other options. Without having to completely change the current system the load-pin measurements can be implemented. This method most likely requires tuning and calibration, but this should be an easy adaptation to the known technology.

#### 3.4.2 Adaptation

Method II is applied to the current load weighing system and uploaded to one of the P&H4100A shovels. The payload results are shown in Figure 3.2. These estimates are done for the same data that is used throughout the whole thesis and therefore the load-pin force in this figure is the same as the load-pin force in Figure 2.10. When the shovel is in a steady configuration, the hoist breaks are activated as can be seen in Figure 2.10 and F.1, where the armature current drops to zero and by that means the hoist motor torque drops as well. Here the advantage of the load-pin is confirmed once again. Unfortunately for this thesis no data is available with fair comparison (so without activated hoist breaks) of the current load weighing system and the one extended with the load-pin.

It is hard to draw any conclusions from these payload estimates since no comparison can be done without truck-scale\footnote{In-ground truck-scales are mentioned here. Not to be confused with the relatively inaccurate on-truck scales.} measurements. Where the scale measurements can represent the true payload. What can be concluded so far is that the estimates are definitely not bad. The relatively smooth input signal gives a nice estimate of the payload. The offset, the black line drawn in Figure 3.2 at 6t results in an (improving) accuracy of 1.8% for these three single configurations of the shovel.
Figure 3.2: Payload history calculated with load-pin data
Chapter 4

Conclusions & Recommendations

4.1 Conclusions

This thesis outlined the possibilities of applying a load sensing pin as an input to the load weighing system on electric rope shovels. The losses due to transmission ratios in the hoist torque are avoided by the use of the load-pin and as expected this results in a much smoother signal than calculated from the hoist drive torque as an input to the load weighing system. Further more, the signal is more reliable, especially when the operator puts on the breaks while swinging, which isn’t unusual. The torque calculated from the hoist drive torque is completely false, because the armature current drops to zero in this particular situation, causing the torque to drop to zero as well.

In order to achieve the force and angle from the strain gauges in the load pin a damped Newton-Raphson method has proven to be a successful algorithm. The algorithm converges fast to the desired solution and the included damping makes it a stable iteration method.

One should be aware of errors created by misalignment of the orientation of the load pin as Section 2.8.1 shows that a slight orientation mismatch can result in a substantial error in the calculation of the rope force depending on the values of the bridge voltages $V_x$ and $V_y$.

4.2 Future work

So far this extension of the current payload system has only run on off-line data and proven to be successful. The next stage in this application of the load weighing system is to apply the obtained models extended with the instrumented clevis pin on a shovel equipped with the load-pin. In order to assess the improvement in accuracy of the load pin information test cases have to be defined for empty and full buckets. When testing with full buckets, in-ground truck scales will have to prove the true payload so it can be compared to the payload system extended with load-pin. These experiments should be carried out for the P&H4100A but later on also for other rope shovels in the P&H range.

The system should be tuned and calibrated for static errors. These might be errors created while mounting the pin. If the influence of the oscillating rope on the cabin-side is substantial, research should be done in eliminating or predicting this behavior.
Bibliography


Index

boom, 4
boom suspension cables, 4, 13
CDEKF, see Kalman
clevis pin, see load-pin

FIR, 22
gantry ropes, 13
handle, 13
haul truck, see dump truck
hoist rope, 4, 13

instrumented clevis pin, see load-pin

Kalman, 6, 7
CDEKF, 7, 8
kinematics, 26

least squares method, 18
load pin, 33
load-pin, 13

MMAF, 6, see MMAF

Newton-Raphson, 20, 33
noise
  measurement, 7
  process noise, 7

P&H, 4
P&H4100A, 3, 4
P&H4100A, 33
profibus, 12, 16, 20

rope force, 11
rope shovel, 33

saddle, 4
sheave, 3, 13
  boom point sheave, 4
shovel
  rope, 13
signal conditioner, 42
  input signal conditioner, 16

torque

dynamic, 11
measurement
  in-line, 12
  reaction, 12
static, 11
dump truck, 2

Wheatstone bridge, 12, 16
wrap angle, 13
Appendix A

P&H4100A specs

<table>
<thead>
<tr>
<th>Table A.1: P&amp;H4100A load-pin specific specs</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Sheave:</strong></td>
</tr>
<tr>
<td>mass</td>
</tr>
<tr>
<td>( J )</td>
</tr>
<tr>
<td>2392.6 0 0</td>
</tr>
<tr>
<td>0 4724.0 0</td>
</tr>
<tr>
<td>0 0 2392.6</td>
</tr>
<tr>
<td>COG</td>
</tr>
</tbody>
</table>

| Load-pin:                                   |
| length                                     | 1.742m   |
| diameter                                   | 330mm    |
| orientation                                | 15°      |

| Electrical system:                         |
| Swing drive                                | Two k558a electrical motors |
| armature current limits -2250, 2250 A      |          |
| Crowd drive                                | One k700 electrical motor   |
| armature current limits -1500, 1500 A      |          |
| Hoist drive                                | Two k1690 electrical motors |
| armature current limits -2560, 2560 A      |          |
### Dipper Capacity

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dipper Capacity (Nominal)</td>
<td>62 yd$^3$</td>
</tr>
<tr>
<td>Dipper Capacity (Range)</td>
<td>40-80 yd$^3$</td>
</tr>
</tbody>
</table>

### ATTACHMENT

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boom Angle</td>
<td>45°</td>
</tr>
<tr>
<td>Boom Length</td>
<td>64 ft. 0 in.</td>
</tr>
<tr>
<td>Effective Dipper Handle Length</td>
<td>35 ft. 0 in.</td>
</tr>
</tbody>
</table>

### WORKING RANGES

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>A  Dumping Radius at Max. Lift</td>
<td>67 ft. 10 in.</td>
</tr>
<tr>
<td>A1 Dumping Radius (Max.)</td>
<td>68 ft. 5 in.</td>
</tr>
<tr>
<td>B  Height of Cut (Max.)</td>
<td>53 ft. 9 in.</td>
</tr>
<tr>
<td>B1 Depth of Cut (Max.)</td>
<td>5 ft. 8 in.</td>
</tr>
<tr>
<td>C  Digging Radius (Max.)</td>
<td>81 ft. 4 in.</td>
</tr>
<tr>
<td>D  Floor Level Radius</td>
<td>53 ft. 6 in.</td>
</tr>
<tr>
<td>E  Dumping Height (Max.) – Door Open</td>
<td>30 ft. 2 in.</td>
</tr>
<tr>
<td>E1 Dumping Height at Max. Radius – Door Open</td>
<td>16 ft. 1 in.</td>
</tr>
</tbody>
</table>

**NOTE:** Working ranges A through E may vary based on dipper selection. Sino each application varies, please consult Plant Mining Equipment for correct choice dipper capacity.
Appendix B

Sheave diagram

The following figure shows the simplification $\phi = \psi_1 - \psi_2$

Figure B.1: One rope-end parallel to x-axis
Appendix C

Calibration setup

The sketches of P&H provided with the calibration setup
Different loads are applied in vertical direction. The pin (and strain gauge orientation) is rotated and measurements are carried out for the following angles $\beta$ in degrees:
$\beta = [24 \ 30 \ 36 \ 40 \ 42 \ 44 \ 46 \ 48 \ 50 \ 52 \ 54 \ 56 \ 60 \ 66]$

![Diagram showing calibration setup with forces $P_y$ and $P_x$, angle $\beta$, and distance $L=2T_p$.](image-url)
Appendix D

Strain Gauge Input Signal Conditioner

The signal conditioner used is Dataforth’s DSCA38-19C with the following characteristics (source: www.dataforth.com)

<table>
<thead>
<tr>
<th>Description</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Excitation Voltage</td>
<td>+10.0V</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>2 mV/V</td>
</tr>
<tr>
<td>Input Configuration</td>
<td>Full Bridge</td>
</tr>
<tr>
<td>Input Range</td>
<td>0 to +20 mV</td>
</tr>
<tr>
<td>Output Range</td>
<td>4 to 20 mA</td>
</tr>
<tr>
<td>Isolation Voltage</td>
<td>1500 Vrms</td>
</tr>
<tr>
<td>Isolation Type</td>
<td>Transformer 3-way</td>
</tr>
<tr>
<td>Accuracy</td>
<td>± 0.03% Span</td>
</tr>
<tr>
<td>Supply Voltage</td>
<td>19 to 29 Vdc (+24V Nom)</td>
</tr>
<tr>
<td>Input Voltage Withstand</td>
<td>240 Vrms</td>
</tr>
<tr>
<td>Gain / Offset Adjust</td>
<td>± 5%</td>
</tr>
<tr>
<td>NMR (60 Hz) Rejection</td>
<td>100dB / decade above 3kHz</td>
</tr>
<tr>
<td>External I-to-V Resistor</td>
<td>N/A</td>
</tr>
<tr>
<td>Output Control</td>
<td>Always enabled</td>
</tr>
<tr>
<td>Output Resistance</td>
<td>&lt; 1 Ohm</td>
</tr>
<tr>
<td>Dimensions</td>
<td>2.95 x 0.89 x 4.13 inches</td>
</tr>
<tr>
<td>Interface</td>
<td>8 Pos terminal block</td>
</tr>
<tr>
<td>Customization</td>
<td>yes</td>
</tr>
</tbody>
</table>
Appendix E

FIR Lowpass filter

FIR lowpass filter with 8 poles.

Figure E.1: Lowpass filter
Appendix F

Collected signals at profibus

![Graphs showing various measured signals at profibus](image.png)

**Figure F.1:** Measured signals at the profibus