Design Specification

Precisionsreglering av gaffeltruck

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Version 1.0

Reviewed	Jonatan Gustafsson	2019-11-08
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Course name:	Reglerteknisk projektkurs	Group E-mail:	jongu926@student.liu.se
Course code:	TSRT10	Responsible for the document:	Jonathan Carlin
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Project Identity

Group E-mail:	jongu926@student.liu.se
Orderer:	Erik Hedberg, Linköpings universitet Phone: +46 13 28 13 38 E-mail: erik.hedberg@liu.se
Supervisor:	Olov Holmer Phone: +46 13-28 16 17 E-mail: olov.holmer@liu.se
Course Responsible:	Daniel Axehill Phone: +46 13 28 40 42 E-mail: daniel.axehill@liu.se

Participants of the group

Name	**Responsibility**	**E-mail**
Jonatan Gustafsson	Project leader (PL)	jongu926@student.liu.se
Jonathan Carlin	Responsible for the documentation (DOC)	jonca673@student.liu.se
Sebastian Haglund	Responsible for the testing (TEST)	sebha953@student.liu.se
Jennie Bäcklin	Responsible for the design (DES)	jenba497@student.liu.se
Carl Hynén Ulfsjöö	Responsible for the software (SOFT)	carhy681@student.liu.se
Henrik Johansson	Responsible for the quality (Q)	henjo648@student.liu.se
Linus Roos	Responisbe for the public relations (PR)	linro231@student.liu.se

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1 INTRODUCTION

This is a design specification that discusses in detail how the requirements in the requirement specification will be met, see *Kravspecifikation*. This includes a general description of the system and how the three main modules (modeling, control and IMU modules) will be designed.

2 SYSTEM OVERVIEW

The truck is an autonomous sliding stand truck model BT Reflex RAE250. It is equipped with a Speedgoat real-time computer that makes it possible to control the truck from a Simulink model. In this project the truck will lift a given load from a given height A to another given height B.

A functional Simulink model of the truck has been provided by Toyota Material Handling (TMH) that will be used for modeling (see Modeling module) and control (see Control module). In addition, an IMU module will be used to measure the oscillations of the forks during a lift (see IMU Module).

A more detailed overview of the system is presented in the requirement specification, see *Kravspecifikation*.

2.1 Signal interface

There are plenty of signals coming from the CAN-bus, but the most relevant ones are:

- Free lift height
- Main lift height
- Reference height
- Reference velocity
- Main cylinder pressure

Relevant signals that will be sent to the actuators:

- Current to the 2 proportional valves
- Speed reference to the hydraulic pump motor

3 MODELING MODULE

The model of the forklift is created in the environment Simulink where the mechanincal and hydraulic parts of the model are implemented in Simscape.

The mast is currently modeled as three solid bodies with prismatic joints in-between. As a result the model only has freedom in one direction. If the horizontal oscillations in the real system are significant, more degrees of freedom are required. The stiffness in the prismatic joints and in the solid bodies implies that mechanical oscillations are not modeled. The modeling is divided in two parts: Evaluation of the Current Model and Modeling of System Dynamics.

3.1 Evaluation of the Current Model

The first step in evaluating the current model is to compare the model with a measured output data from the forklift. This will be done by sending in a given input signal to both the model and the forklift. The two different sets of data can then be compared. The objective is to execute different experiments to evaluate the current model and to measure the oscillations created by the load.

A complete lift from bottom to the top is divided into three sections. The first section is the free lift, the second section is the transfer phase between the first section and the last section which is called the main lift. The different sections requires different types of tests described below.

3.1.1 *Free Lift and Main Lift*

The first lift from the truck is the free lift. During the free lift only the valve denoted Q_3 is used. The last lift is the main lift. During this lift only the valve denoted Q_2 is used. Both the free lift and the main lift has the same structural behaviour. The type of experiment used to determine the model behaviour of the free lift and the main lift is to do a lift on each section where the input signals are the current to each valve and the speed to the hydraulic pump motor. By doing different lifts with different weight of the load, the free lift model and the main lift model can be evaluated.

3.1.2 *Transfer Phase*

The transfer phase is the phase between the two lifts. During this lift the active flow valve for the free lift, Q_3, is slowly closing and the valve, Q_2, used for the main lift is increasingly opening. The planned experiment to determine what happens during the transfer phase is to do a top lift, that is a lift from the bottom to the top.

3.1.3 *Experiments to Determine the Oscillations due to the Load*

At TMH a positioning system is available to determine the positioning of the carriage. By doing lifts with different weights the oscillative behaviour can be measured. Simulations will be done with the same type of lifts and then compared with the given data from the positioning system.

3.2 Modeling of System Dynamics

The modeling of the system dynamics is described in the subsections below.

3.2.1 *Index for Measuring Oscillations*

One simple way to measure oscillations would be to look at the total amplitude offset from its steady state value of a point p.

$$J = \int_{t_0}^{\infty} ||(p(t) - p(t \to \infty))||^2 dt \tag{1}$$

Here the system is assumed to be stable, which means that p approaches a constant value as time goes to infinity. The position of p depends on which oscillations are most relevant. Here p is assumed to be on the tip of the forks. The boundary time t_0 starts when the vertical component of p passes the vertical component of $p(t \to \infty)$ for the first time in order to avoid large integral wind-ups during the initial step.

3.2.2 *Mechanical Dynamics*

It is difficult to model the dynamics of the full system in a way that can be implemented in the current model without knowing how all the background calculations are handled in the model provided by TMH. Therefore, rather than modeling the entire system, the dynamics are split into two separate parts: the vertical oscillations in the mast and the horizontal oscillations.

Elasticity in the Mast
Since the current model only supports movement in the vertical direction the simplest way to include dynamic effects would be by including springs and dampeners in the system. Whether it is most accurate to place this spring/dampener between the mast and the middle guide, or between the guides needs to be investigated. The different placements alternatives can be seen in Figure 1. If the real structure has several resonance frequencies, more springs are needed to represent this.

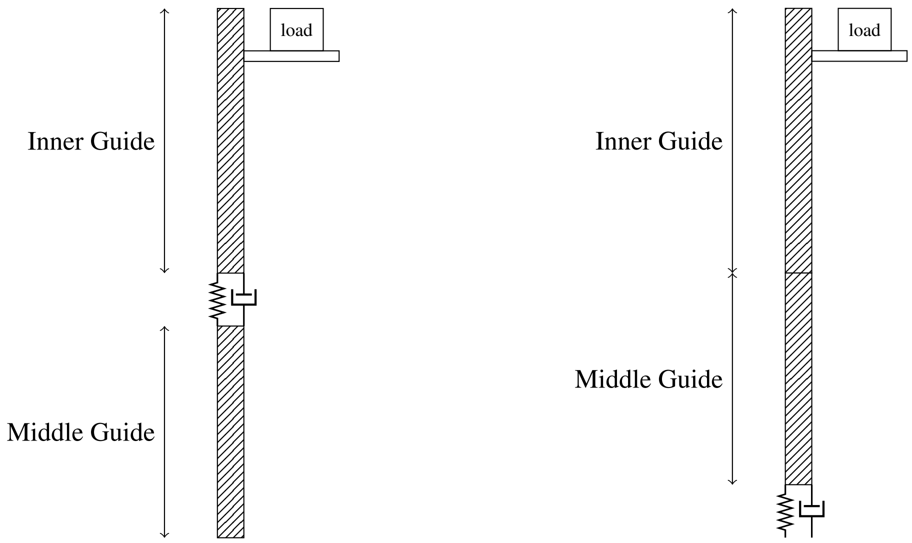

Figure 1: Possible positions for the spring/dampener system.

Adding more Degrees of Freedom

It is visible in the real system that oscillations also occur in the horizontal direction. In the model, this effect could be achieved by allowing the mast to tilt slightly. Rotational springs and dampeners would have to be included to guarantee that the equilibrium position of the mast is vertical and that a clear resonance frequency is modeled. In the event that this single rotational spring/dampener system would not be sufficient to describe the real system, it is possible to include a second spring dampener system so that both the mast and the guides are allowed to tilt. The alternatives for the position are shown in Figure 2.

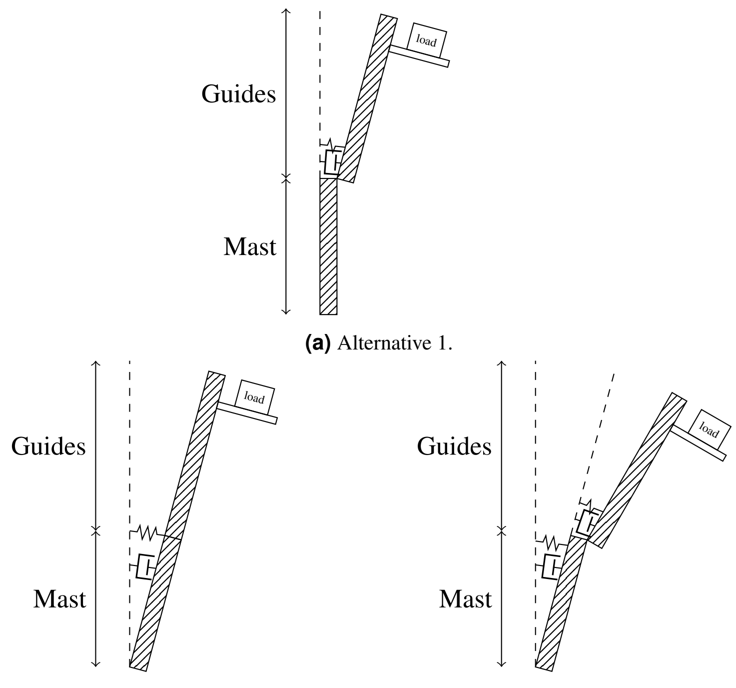

Figure 2: Possible positions for the rotational spring/damper system.

Load Impact on Resonance Frequencies

By adding a load on the forks, the resonance frequency should be affected. It is however assumed that the spring and damper parameters are not load dependent. Whether this is true or not needs to be investigated. Collecting data using different loads could be used to investigate where the springs should be placed.

3.3 Parameter Estimation and Model Choices

The modeling will be carried out in three steps. The first step is to analyse the behavior of the system. The second step is to model the dynamics in Simulink/Simscape, and the third step is to verify and fine-tune the model.

Step one:

- Determine the number of resonance frequencies for each direction.
- Analytically approximate vertical spring position for the main resonance frequency.
- Analytically approximate rotational spring position for the main resonance frequency.
- Analytically approximate vertical spring/damper parameters to match the main resonance frequency.
- Analytically approximate rotational spring/damper parameters to match the main resonance frequency.

Step two:

- Add one vertical spring/damper system to model vertical oscillations without load.
- Add one rotational spring/damper system to model horizontal oscillations without load.
- Tune parameters empirically after the main resonance frequencies of the real system, using the analytical solutions as a base.

Step three:

- Model other potential resonance frequencies.
- Determine if the behavior of the transfer stage is accurate.
- Determine if lift height affects the spring/damper parameters.
- Determine if there are better ways to separate the mechanical and the hydraulic parts of the system.

4 CONTROL MODULE

The control system will have the structure that can be seen in Figure 3. The control system will have two main parts:

- A trajectory planner
- A feedback loop

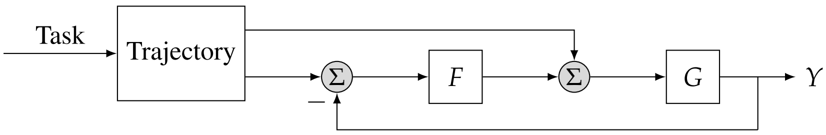

Figure 3: A block diagram of the system with controller.

4.1 Control Oriented Models

The full Simulink model is far to complex to use in the trajectory optimization. Because of this a simpler model has to be developed. The components in the model and how they are related to each other can be seen in Figure 4. The dynamics of the Electric Engine, the Hydraulic Pump and the Valves are not considered. The dynamics will only be introduced if there is a need for a more accurate model. In the following subsections the equations for describing the model are presented.

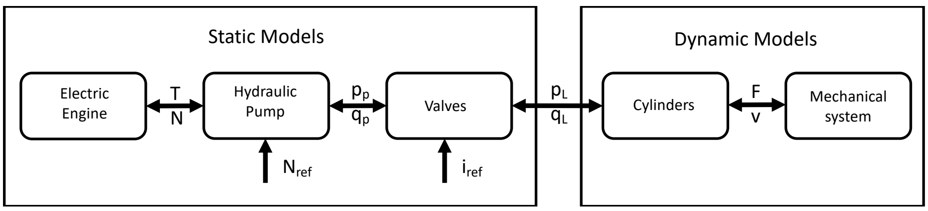

Figure 4: A block diagram showing the models of the system and how they are connected. The parameters are intensity and flow in the different domains.

4.1.1 *Electrical Engine*

The engine will be modeled using a motor map of the efficiency $\eta(T, N)$, where T is the torque and N is the speed.

$$P_{in} = \frac{TN}{\eta(T,N)} \frac{2\pi}{60} \tag{2}$$

4.1.2 *Pump*

The pump will be modeled as an optimal pump with the displacement D_p, the volumetric efficiency η_{volp} and the hydraulic mechanical efficiency η_{hmp} as parameters and the speed N_p and pressure p_p as variables.

$$q = D_p N_p \eta_{volp} \tag{3a}$$

$$T = \frac{D_p p_p}{\eta_{hmp}} \tag{3b}$$

4.1.3 *Proportional Valves*

The proportional valves will be model as ideal valves with the flow coefficient C_q and oil density ρ as parameters. The variables are the opening area A, the pump pressure p_p and the pressure from the load p_L. The open area of the valves is a linear function of the current sent to it.

$$q = C_q A \sqrt{\frac{2}{\rho}(p_p - p_L)} \tag{4a}$$

$$A = ki \tag{4b}$$

4.1.4 *Cylinders*

The cylinders will be modeled as ideal cylinders, where the force and velocity will be calculated as:

$$F = p_L A \tag{5a}$$

$$\dot{h} = \frac{q}{A} \tag{5b}$$

4.1.5 *Mechanical structure*

The oscillation of the load will be modeled as a spring-mass-damper system, with degrees of freedom in both the height and angle, see the sketch of the system in Figure 5. The spring and damper parameters will be modeled as functions of the respective cylinder heights. The mechanical system is connected to the hydraulic system by Equations 5, these equations give h_f and h_m.

The equations obtained by using Newton's law is presented below. The indices are presented in Table 1.

Index	Description
r	Rotation
l	Load
f	Free lift cylinder
m	Main lift cylinder
mg	Middle guide
ig	Inner guide
fk	Forks
1	Free lift encoder reading
2	Free lift encoder reading

Table 1: All the indices used in the mechanical model.

$$J\ddot{\theta} = -k_r\theta - c_r\dot{\theta} + m_f(l + \theta h_{tot})(g + \ddot{h}_{tot}) \tag{6a}$$
$$m_f\ddot{h}_1 = -m_f g + F_f \tag{6b}$$
$$m_m\ddot{h}_2 = -m_m g + F_m - 2F_f \tag{6c}$$

where:

$$F_m = k_m(h_2)(h_m - h_2) + c_m(h_2)(\dot{h}_m - \dot{h}_2) \tag{6d}$$
$$F_f = k_f(h_1)(h_f - h_1) + c_f(h_1)(\dot{h}_f - \dot{h}_1) \tag{6e}$$
$$m_m = m_{mg} + 2(m_{ig} + m_f) \tag{6f}$$
$$m_f = m_l + m_{fk} \tag{6g}$$
$$h_{tot} = h_1 + 2h_2 \tag{6h}$$

4.2 Trajectory and Feed Forward Control

Given the lift height, the mass and center of gravity of the load, the control module has to calculate a trajectory. The trajectory will be calculated using optimization where a goal function that minimizes the time, energy consumption and oscillation will be used. This will be done beforehand for several different scenarios. These trajectories with their respective control signals will be saved in a lookup table. When performing a lift the controller will interpolate a trajectory with control signals. The trajectory will be used as reference signal to the system and the control signals will be used in a feed forward control.

4.2.1 *First Approach on Trajectory*

The first approach will generate a trajectory based on the concept of minimizing jerk. This trajectory will not have any control signal attached to it. This means that the feed forward controller has to use an inverse model to get the control signals from the trajectory. This solution will be used as a benchmark for the optimal trajectory generator.

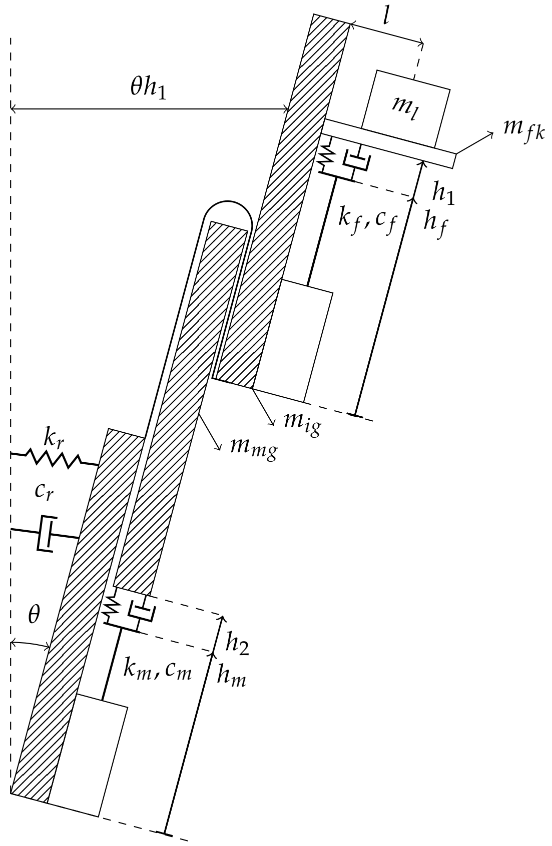

Figure 5: The model of the mechanical system.

The method will be based on minimizing the square of jerk during the motion.

$$\min_{x(t)} \int_0^T \dddot{x}^2(t)dt \tag{7}$$

Given a set of boundary conditions on position, velocity and acceleration, it can be proven that the optimal solution is a fifth order polynomial [1].

The constants a_i in Equation 8 will be determined by the boundary conditions. It is possible to choose the total lift time T but it cannot be chosen arbitrary small since there is a maximum allowed velocity. The resulting appearance of the curves can be seen in Figure 6.

$$x = \sum_{i=0}^{5} a_i t^i \tag{8}$$

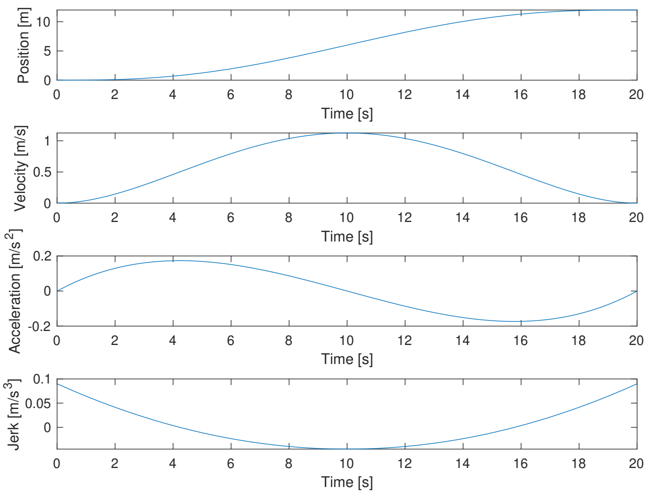

Figure 6: A graph showing the trajectory that minimizes the jerk.

4.2.2 *Trajectory Optimization*

To perform the trajectory optimization a direct collocation method will be used. This is one type of a transcription method that allows the optimal control problem to be transcribed to a parameter optimization problem. These problems can then be solved with non-linear optimization.

Direct collocation methods work by discretizing the state and control and then approximating them with polynomial splines. They ensure that the problem fulfils the system dynamics by introducing constraints on the state and control at points along the trajectory called collocation points. See [2] for a detailed introduction.

These methods could be used to convert the optimal control problem:

$$\min_{x(t),u(t),T} \quad \underbrace{\omega_1 T}_{\text{Time}} + \int_0^T \underbrace{w_2 \frac{i(t)u(t)}{m_l g v(t)}}_{\text{Energy}} + \underbrace{w_3 h_{tot} \ddot{\theta}^2 + w_4 \ddot{h}_{tot}^2}_{\text{Oscillation}} dt \tag{9a}$$

$$\text{subject to} \quad \dot{x}(t) = f(x(t), u(t)), \tag{9b}$$

$$x(t_0) = x_0, \quad x(t_f) = x_f, \tag{9c}$$

$$u_{min} \leq u(t) \leq u_{max}, \tag{9d}$$

$$x_{min} \leq x(t) \leq x_{max}. \tag{9e}$$

to a nonlinear program, typically formulated as:

$$\min_{z} \quad J(z) \tag{10a}$$

$$\text{subject to} \quad f(z) = 0 \tag{10b}$$

$$g(z) \leq 0 \tag{10c}$$

$$z_{min} \leq z \leq z_{max}. \tag{10d}$$

which can then be defined in the symbolic framework Casadi and then solved with a nonlinear solver such as IPOPT or FMINCON.

The goal function contains 4 parts all weighted with w_i.

- Time: T
- The inverse of the efficiency: $\frac{i(t)u(t)}{m_l g v(t)}$
- The horizontal oscillations: $\ddot{\theta}$
- The vertical oscillations: $\ddot{h}_{tot}$

4.3 Feedback Control

Feedback control will be used to minimize the difference between the calculated trajectory and the measured output. The controller will be a LQ-controller (linear-quadratic controller), which will minimize a quadratic cost function. The main advantage with this approach, in contrast to using a PID-controller or a regular state feedback-controller, is that it gives an easy way to determine the feedback gains.

Since the main idea for the feedback loop is to take care of model errors and disturbances a simplified control model will be used. Compared to the model described in Section 4.1 the influence of the rotation θ will not be considered. The resulting controller will measure the free and main lift heights and will control the pump reference speed and the valve currents.

The state space form will then be:

$$x = \begin{bmatrix} h_1 & \dot{h}_1 & h_2 & \dot{h}_2 \end{bmatrix}^T \tag{11a}$$

$$\dot{x} = Ax + Bu \tag{11b}$$

$$y = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} x \tag{11c}$$

$$u = \begin{bmatrix} N_{ref} & i_m & i_f \end{bmatrix}^T \tag{11d}$$

Where the matrices A, B and C are found by linearizing the equations in Section 4.1 under the assumptions described above. The system will be linearized at two different points, one halfway up the free lift and one halfway up the main lift:

$$x_{lin,0} = \begin{bmatrix} \frac{h_{1,max}}{2} & v_{max} & 0 & 0 \end{bmatrix} \tag{12a}$$

$$x_{lin,1} = \begin{bmatrix} h_{1,max} & 0 & \frac{h_{2,max}}{2} & v_{max} \end{bmatrix} \tag{12b}$$

This will result in two controllers, the first will be used below the transition and the second will be used above the transition.

5 IMU MODULE

The requirements on the IMU module is to evaluate the required performance of an IMU for effective control and to find an appropriate mount point on the forklift. The IMU will be fastened on the forklift to measure the acceleration of the load and transfer the signal to the control unit.

5.1 Design

Two different implementations will be considered, a wired and a wireless approach.

- In a wireless approach, the IMU needs to be connected to an independent system with its own power supply and control unit that could communicate with the forklift through WiFi. Wireless alternatives include:
  - Smart device:
    A smart device such as a smartphone or a smartwatch has built in sensors, a power supply and a control unit that could be used to communicate with the forklift.
  - Wireless IMU Module:
    An IMU *box* of separate modules that provide the IMU with the necessary characteristics such as power supply and communication unit.
- In a wired approach, the IMU will be supplied with power and communicate through a separate wire network that will hang alongside the forklift.

The approach depends on the ease of implementation, but the first approach will be to use a smart device. Furthermore, the output from the IMU will then be band-pass filtered in order to remove sensor bias and low/high frequency noise.

5.1.1 *Mounting*

The IMU will initially be mounted at the heel of the forks on the forklift where the amplitude of the oscillation is most noticeable. If the result proves to be inadequate, the IMU will be mounted on the mast instead.

5.2 Evaluation

To evaluate the required performance of the IMU, a first approach would be to use TMHs camera positioning system to measure the oscillations of the load. The data will then be used to determine the necessary range of acceleration, bandwidth and accuracy of the IMU.

Another approach would be to evaluate the data from a smart device IMU to determine the required IMU characteristics. By observation it is known that the IMU needs to measure at least two axes; vertical and horizontal.

5.2.1 *Estimation of IMU Characteristics*

By observing the system during a lift an initial estimate is that the IMU should have the following characteristics:

- Range of acceleration:
  The IMU should have a range of around 1-2g. With a higher range, the measurements will become less accurate, and with lower range will result in invalid measurements.
- Bandwidth:
  The IMU should have a bandwidth of around 20 Hz to be able to measure the low frequency oscillations that occur during a lift.
- Accuracy:
  As accurate as possible.

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[2] M. Kelly, “An introduction to trajectory optimization: How to do your own direct collocation,” *SIAM Review*, vol. 59, no. 4, pp. 849–904, 2017.