University of Southern Queensland

Faculty of Engineering and Surveying

Automation of the Rollingstock Twist Test

A dissertation submitted by

Carson Keith Holzheimer

in fulfilment of the requirements of

Courses ENG4111 and 4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: November, 2007

Abstract

This dissertation is an investigation into the feasibility of automating the static twist test procedure currently performed at Queensland Rail. The aim is to select equipment that can perform this procedure automatically while staying within the confines of a budget.

The test requires synchronous jacking of train wheels and it was concluded from the research carried out that synchronous lifting of heavy objects can be successfully achieved through the use of 'digital hydraulics', a term explained in detail within this dissertation. The concept of digital hydraulics was tested for application to the twist test and was found to be completely successful.

A final system design is specified, including equipment such as position transducers, hydraulic cylinders and a PLC. The equipment exceeds the budget specified by Queensland Rail. However it proves that a successful system can be constructed for reasonable cost.

University of Southern Queensland

Faculty of Engineering and Surveying

ENG3111 & ENG4112 *Research Project*

Limitations of Use

Prof F Bullen
Dean
Faculty of Engineering and Surveying

Certification

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Carson Keith Holzheimer

Student Number: 0050026259

____

Signature

____

Date

Acknowledgements

Firstly, I would like to acknowledge and thank Selvan Pather at USQ for his excellent supervising throughout the course of my project work. His extra guidance with formatting, proof reading and the structure of my dissertation was invaluable.

Also, thanks must go to Ian Burns, my supervisor at QR. His helpfulness, support and patience was very welcome and his initiation of the project allowed me to work on something interesting for the year. The rest of the engineers in the Vehicle and Track Section were also extremely helpful in providing data and answering questions.

I also thank the many industry suppliers for their willingness to provide their time and information, whilst knowing that I was not likely to purchase any equipment.

Thanks Kraig for those few discussions we had about my project. Your experience helped.

Table of Contents

List of figures

1 Introduction

1.1 Client Introduction

This project was completed for the benefit of Queensland Rail and to satisfy the requirement of a final year research project for a Bachelor of Mechanical Engineering degree.

Queensland Rail Limited (QR) is a government owned company providing rail services to the state of Queensland. They are a progressive company and conduct business in the divisions of passenger, freight and coal transportation. Their portfolio is expanding to include freight and coal haulage in the states of Western Australia, New South Wales and Victoria.

Most of their business is centred in Brisbane where their head office, some smaller offices and a number of workshops are located. The workshops are used for maintenance, modifications, new construction and train performance testing procedures.

1.2 Client Brief

The need for this project was stipulated by QR. Engineers in the Vehicle and Track Engineering section of Rollingstock Engineering required that a feasibility study be done on automating the twist test, a rollingstock performance test undertaken by employees of this section. A simple description of this test is that it lifts wheels of the rollingstock to differing heights and the ability of the rollingstock to cope with this is assessed.

The test is to be automated in the sense that the lifting of the rollingstock will be controlled without human interaction.

These guidelines were also specified by QR engineers.

- Determine if the automated test is achievable for realistic cost. Aim for $20,000 maximum.

- Complete research into possible existing equipment that can adequately perform the test or be tailored to perform the test.
- Specify what equipment is required. This will include independent validation of ram size, pump capacity, power requirement, valves, control etc.
- Analyse safety considerations.

1.3 Need for Project

The twist testing procedure that QR follows currently is tedious, slow and prone to human error. The test takes a long time to set up. Verbal communication is used to synchronise the lift, which is difficult to coordinate. The test is a repeatable process that appears predisposed to automation which could improve some of the shortcomings of the current test.

1.4 Project Aim

The aim of this project is to design a system of twist testing that is more accurate, faster and less prone to human error than the current manual method used at QR. This will enable QR to reduce the time of train registration, reduce the amount of personnel engaged in performing the test and permit more information to be gathered from the test.

1.5 Project Objectives

These are the objectives of this research project.

1. Identify outcomes QR expects from an automated twist test.
2. Research applicable regulations.
3. Develop clear specifications to which equipment has to perform.
4. Research existing systems that may satisfy requirements.
5. Research and select possible control schemes and software control equipment.
6. Research and select appropriate lifting equipment.
7. Simulate or build model system.
8. Analyse performance of the specified system.

9. Calculate cost of entire system and compare with budget.

1.6 Methodology

The respective project objectives will aim to be achieved using these methods.

1. To identify the outcomes QR requires from the test, negotiation will be undertaken with QR engineers. Specifically it must be known exactly what information to what accuracy will be required to be output from the test.
2. With any large project involving public transport there will be regulations and guidelines. These will be obtained through contacts at QR.
3. A large amount of information will be gathered to completely specify the performance requirements of an automated test. This will be done thorough negotiation with QR, researching the rollingstock to be tested and reviewing the current testing method documentation.
4. To develop system concepts, other areas where heavy lifting is required will be looked at for ideas. Lifting equipment and the power these devices require will be researched by contacting suppliers and checking the capabilities and compatibility of their products.
5. Control schemes will be compared and assessed on factors such as cost, accuracy and setup time. Again, suppliers of automation control hardware will be contacted and the suitability of their products assessed.
6. Lifting equipment will be selected by completing the necessary calculations to estimate sizing. Product catalogues of system components will be inspected to select appropriate models.
7. The system will be tested by either building a physical model or by computer simulation analysis.
8. The performance of the model will be analysed in terms of its ability to satisfy the performance requirements.

9. Costs will be calculated by requesting quotes from the suppliers of the various system components.

1.7 Conclusion

This dissertation aims to describe the process involved in designing an automated twist test. The research and design conducted will hopefully result in a successful system which the QR Vehicle and Track Engineering section can propose to higher management in the hope of gaining funds for further development. The first project objective to deal with is to indentify clear outcomes which the test needs to satisfy. Before this is attempted, background details of the twist test will be covered.

2 Background

2.1 Introduction

This chapter serves the purpose of covering the background information needed to understand the context of the twist test in the rail industry. The concept of twist is explained and the need for twist testing shown. It describes the procedure used to undertake twist tests currently and problems with the test will become evident.

2.2 Track Twist

Track twist in simple terms is the vertical variation of rail height. Twist is induced by the track in particular at the transition into or out of a curve. At this transition, there is a cant ramp. Cant is the difference in heights between the two rails so that the centrifugal forces of negotiating the curve at speed are balanced by the train tilting in the opposite direction – much like camber on a road surface. Naturally there needs to be some sort of ramp to gradually raise and lower the outside rail as the vehicle enters and exits a curve - the cant ramp. It is when the vehicle is on this ramp that it is subject to substantial twist.

2.3 Train Derailments

A derailment is an accident where a train partially or entirely leaves the tracks. They can range in severity from one wheel jumping the track, to rollover and possible loss of passengers or drivers lives. Derailment can occur for a number of reasons, but the particular reason we are interested in for this project is the inability of rollingstock to cope with track twist. This is a reasonably prevalent cause of derailment.

2.3.1 Categorisation of Past Derailments

To give an indication of how notable twist is in causing derailments, a report prepared by Risk Solutions for the UK Rail Safety and Standards Board will be examined (Rail Safety & Standards Board 2004). It is a study of the causes of all derailments over the past 10 years in the UK. The data may not be directly transferable to the Australian rail industry because of the differing track widths and types of rollingstock but the interaction between the rails and rollingstock wheels will be similar. The most interesting data from this report is the breakdown of specific causes of derailment, and it has been presented in Figure 2-1.

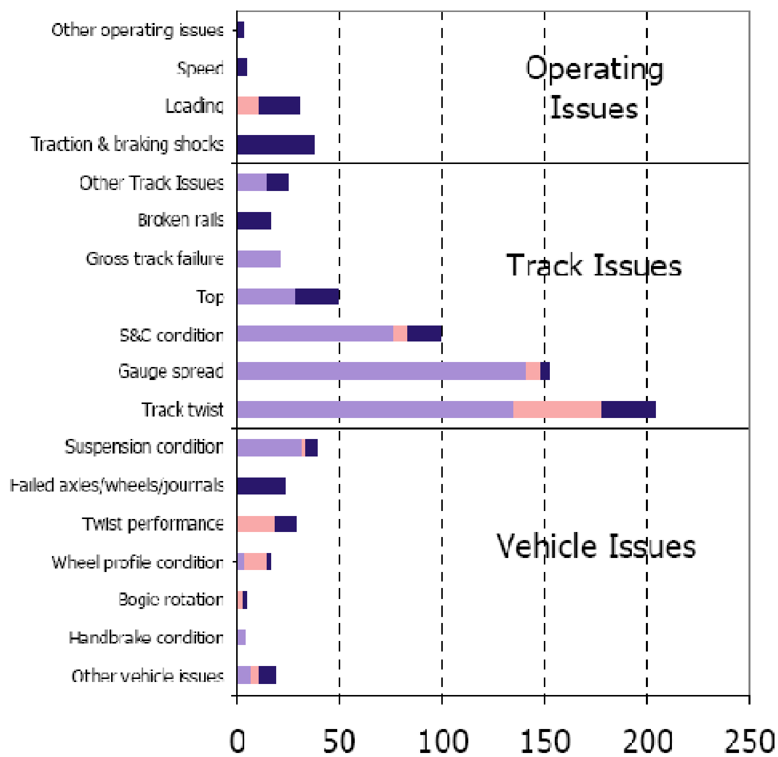


Figure 2-1 Categorisation of past derailments (adapted from RSSB report)

Notice that most derailments occurred due to track twist. This is a track maintenance issue and cannot be diminished by better twist testing of the cars. However it highlights very clearly that track twist outside maintenance tolerances is a common occurrence. Therefore vehicles need to be carefully tested for twist performance because during their service they will almost certainly come into contact with excessively twisted track. Please ignore the different colours as this was

Twist performance of the vehicle itself was the second most common vehicle issue, once again emphasizing the importance of an accurate twist test. Suspension issues, the vehicle issue causing most derailments, can often be detected by QR's twist test. Better highlighting of these suspension faults may well be the main advantage gained from producing more accurate and detailed test data.

2.4 The Twist Test

The twist test is designed to ensure that any new or substantially modified vehicle is compatible with the track twist induced by the rail network. It certifies that this twist does not cause an unacceptable level of wheel unloading, as this may cause derailment.

A twist test is performed on any vehicle where the torsional stiffness is suspected to have increased due to modifications such as the following:

- Increased suspension spring rates
- Increase in bogie frame torsional stiffness
- Increase in underframe torsional stiffness
- Change in vehicle equipment or mass distribution

The bogie frame can be viewed in Figure 2-2.

Figure 2-2 Bogie frame

2.4.1 Wheel unloading

Wheel unloading is important because it can cause the wheel to 'climb the rail' and ultimately derail. The critical measurement is the L/V ratio, which is the ratio of lateral to vertical forces. While the train is cornering there is a high lateral force exerted on the outside wheel. Vehicles and track are designed to keep the L/V ratio below 1 as this has been calculated as the safe point above which there is high risk of the wheel jumping the rail. Figure 2-3 shows the effect of an excessive L/V ratio.

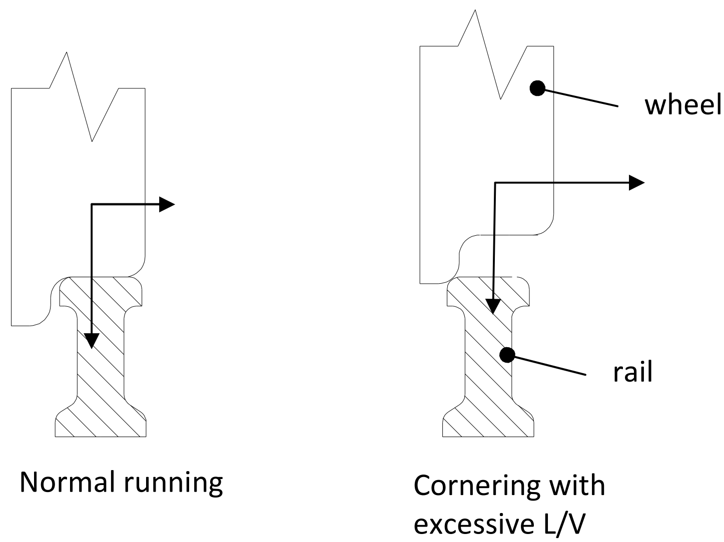


Figure 2-3 Excessive L/V ratio can cause a wheel to jump off the track while cornering

As a train is cornering the vertical downward force will also increase due to the centrifugal forces trying to tilt the train and this will do somewhat to keep the L/V ratio at acceptable values. Unfortunately it is during this curve negotiation that track twist comes into play and causes significant vertical unloading, which may cause excessive L/V ratios.

2.4.2 The Current Testing Procedure

The current procedure is governed by the QR standard – Rollingstock Dynamic Performance. Here it is stated that the maximum wheel unloading shall not exceed 90% and individual wheel L/V shall not exceed 1.

It is difficult to measure the wheel unloading dynamically on the track because of the need to set up a test track of the exact dimensions. For this reason a static twist test is used. Obviously dynamic unloading will be larger than that which occurs in a static test and therefore the limit for acceptable unloading is reduced to 60% for the static twist test.

Currently the test procedure is quite basic. The procedure is to lift wheels to heights replicating the position the wheels will be in while travelling over the cant ramp. A representation of a vehicle negotiating a cant ramp can be viewed in Figure 2-4.

To simulate the worst case of unloading, the possibility of track irregularities is included in the test. A dip is superimposed on the cant ramp at the critical front wheel. This is to simulate degraded track that, however, is still within maintenance limits.

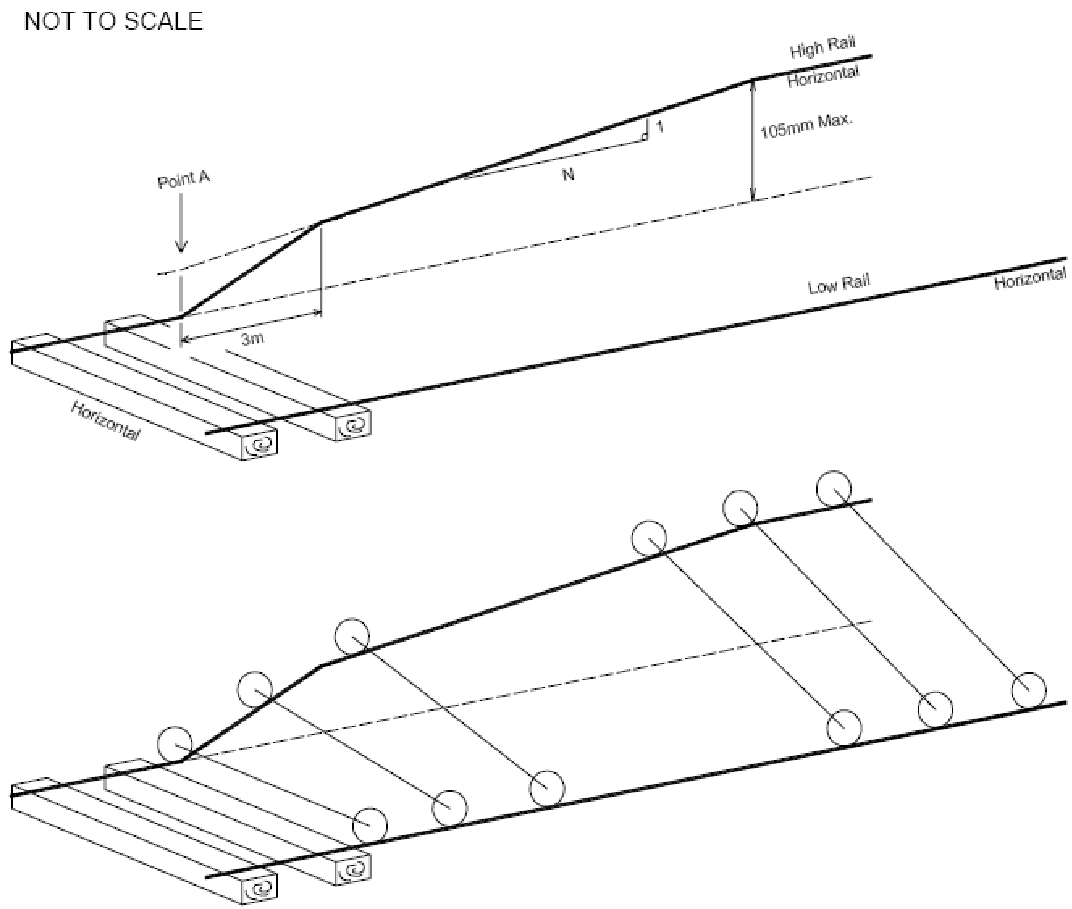


Figure 2-4 A representation of a vehicle negotiating a cant ramp (QR standards)

The wheel at Point A in Figure 2-4 is the wheel undergoing most unloading. This wheel will be called the 'critical wheel' throughout the rest of this dissertation and the twist test is designed to measure the maximum unloading that occurs on this wheel.

The actual heights are calculated from the bogie wheelbase, bogie centre distance, cant ramp slope and dip dimensions. The following figure shows these dimensions. Bogies are the structure providing the link from the axles to the vehicle body.

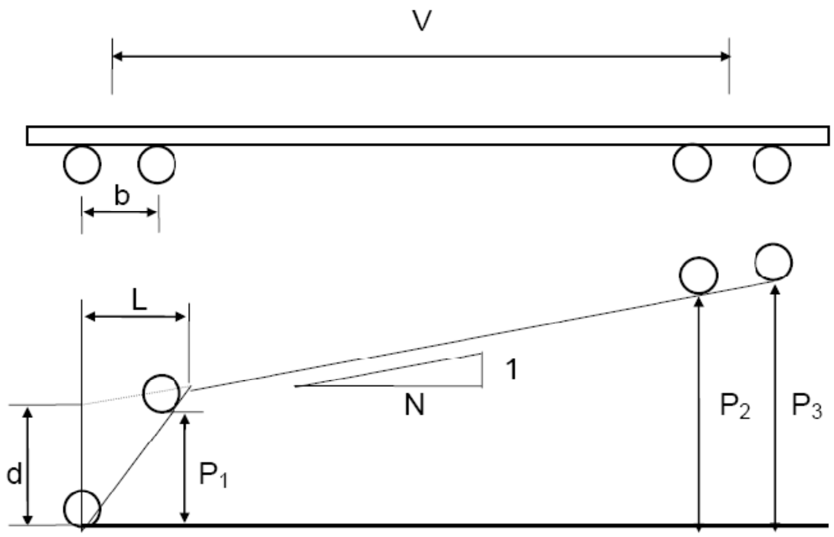


Figure 2-5 Twist test dimensions (QR standards)

Figure 2-5 is taken from QR standards and the dimensions are explained below:

- N = Slope of cant ramp (1 in N)
- b = Bogie Wheelbase (mm)
- V = Bogie centre distance (mm)
- d = Dip depth (mm)
- L = Length over which dip acts (mm)
- P1 =(d+L/N)b/L (mm)
- P2 =d+V/N (mm)
- P3 =d+(V+b)/N (mm)

The wheels are lifted to these heights with hand operated hydraulic bottle jacks or a pump and cylinder set. Heights are measured by comparing a mark on the wheel with a steel ruler sitting against the wheel, visible in Figure 2-6. Note that the heights are different for each wheel.

To achieve accuracy of the load data gathered, it is desired that the cylinders be lifted synchronously, meaning cylinders should reach their respective required heights at the same time. Synchrony is achieved in the current test by verbal communication between personnel operating the bottle jacks. For example, phrases like “I’m nearly there” and “You jack slower” are used. This is quite obviously not going to be repeatable or accurate and will cause different load reading due to the hysteresis effect described in Section 2.5. Sometimes a mistake is made by one of the personnel operating the jacks and the test has to be restarted.

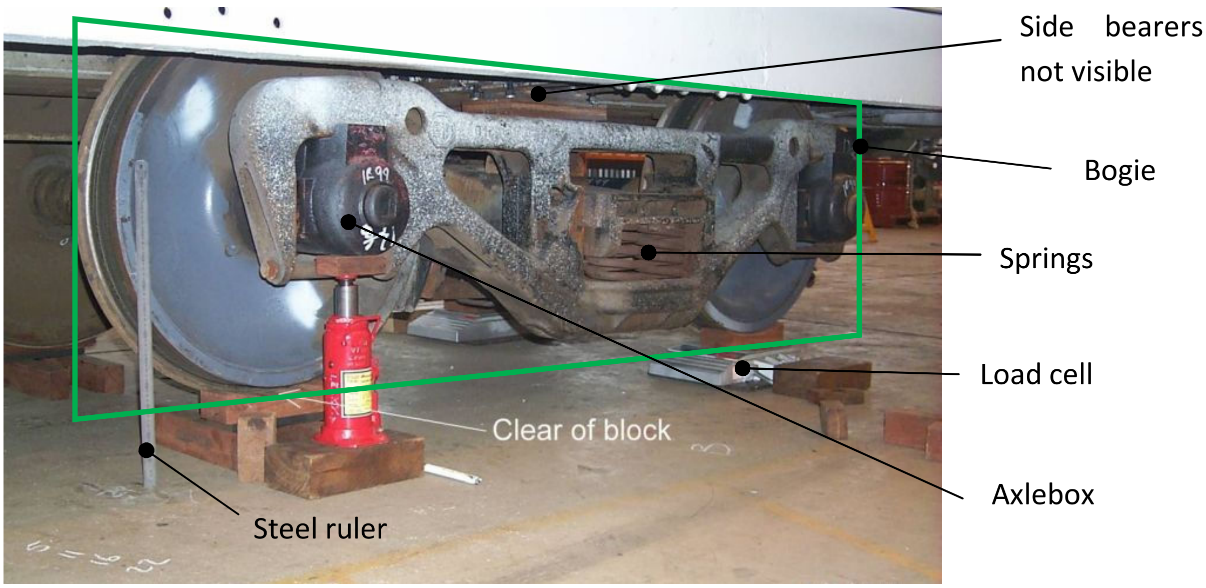


Figure 2-6 The current manual testing method

2.4.3 Load measurement

As seen in Figure 2-6, load cells are positioned under the critical wheel to measure the amount of unloading once the wheels have been jacked to their respective heights. The lifting is stopped at intermediate heights, at 25% 50% and 75% of total stroke. This is so a plot of load vs displacement can be generated and from this, considerable information about the suspension system can be viewed.

2.4.4 Other measurements

During the lift more readings than just the load are taken. To determine how much each component of the suspension is contributing to absorbing the rail height, measurements of these component deflections are taken at all intermediate heights. The two components measured are spring and side bearer deflections and the location of these components can be viewed in Figure 2-6. Side bearers are devices installed either side of a centre pin of a bogie and they are needed for the train body to lean against when cornering. The side bearers sometimes include friction wedges and provide an amount of friction damping similar to shock absorbers in a car.

2.4.5 Test set up

When a train needs to be tested, QR engineers pack the equipment needed in a 4WD and drive to the workshop where the train is situated. This could be at any number of QR's workshops around Brisbane depending on where it is easiest to drive the train to. There are a few different ways that the test is currently set up. The method used depends on the locations available.

Method 1:

The vehicle is lifted off the rails inside the workshop via the use of two overhead cranes and placed on a level concrete floor. Every wheel has to be placed on wood blocks to stop damage to the concrete and to keep them level with the front wheels, which will be placed on load cells. This method is being used in Figure 2-6.

Method 2:

The train is left on the rails and a steel plate placed to straddle the rails so that the load cells can have a flat base to rest on. The vehicle is moved onto the load cells by lifting with cranes or lifting with jacks and dragging the load cells underneath the wheels.

Method 3:

In some workshops there is a pit between the rails and a train weigher can be used. The train weigher is positioned as shown in Figure 2-7. The nuts are tightened and this pushes the weighing plates outwards against the rails, thus wedging it in to a secure position. The vehicle can be easily rolled onto the weighing plates of the train weigher. This is the preferred set-up method because it does not require the use of cranes or any flat concrete floor space.

Figure 2-7 Train weigher

2.5 Twist Test Data

Twist test data is gathered and presented in an Excel spreadsheet. The Excel sheet contains many calculations to find values such as body twist in degrees, rail height absorbed by the body and the critical, percent unloading. From this data sheet it can be analysed which suspension components are ‘going solid’; very useful if the train fails the test and it has to be decided what modifications are needed. There have been a quite a number of rollingstock fail the test in just the last 5 years.

Another outcome of the test is the hysteresis graph. Two examples of these are given in Figure 2-8 and Figure 2-9. This again is a tool to analyse the performance of the suspension. The definition of hysteresis is the lagging of an effect behind its cause. Here the cause is the lifting of the wheel and the effect is the load measured at the critical wheel. It can be seen on the graphs that load measurements are taken on an upstroke and a downstroke. Notice from the two graphs below that the load measured on the way down is always less that the load measured on the upstroke. The area between these lines is a measure of the energy absorbed by the suspension.

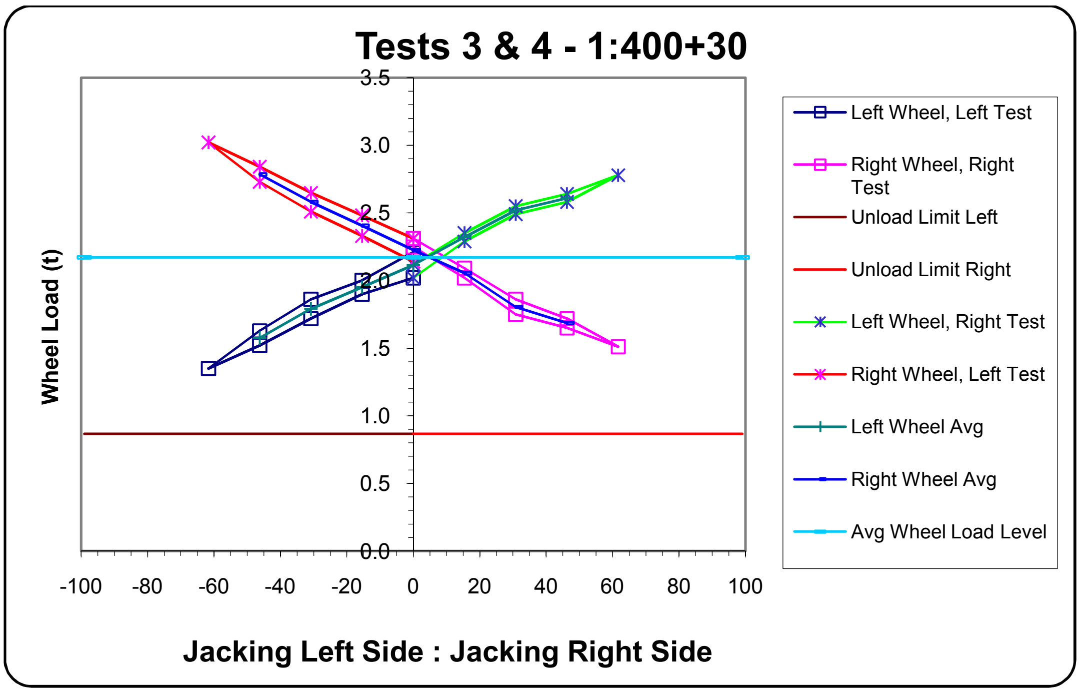


Figure 2-8 KLEX Wagon hysteresis graph (QR test data)

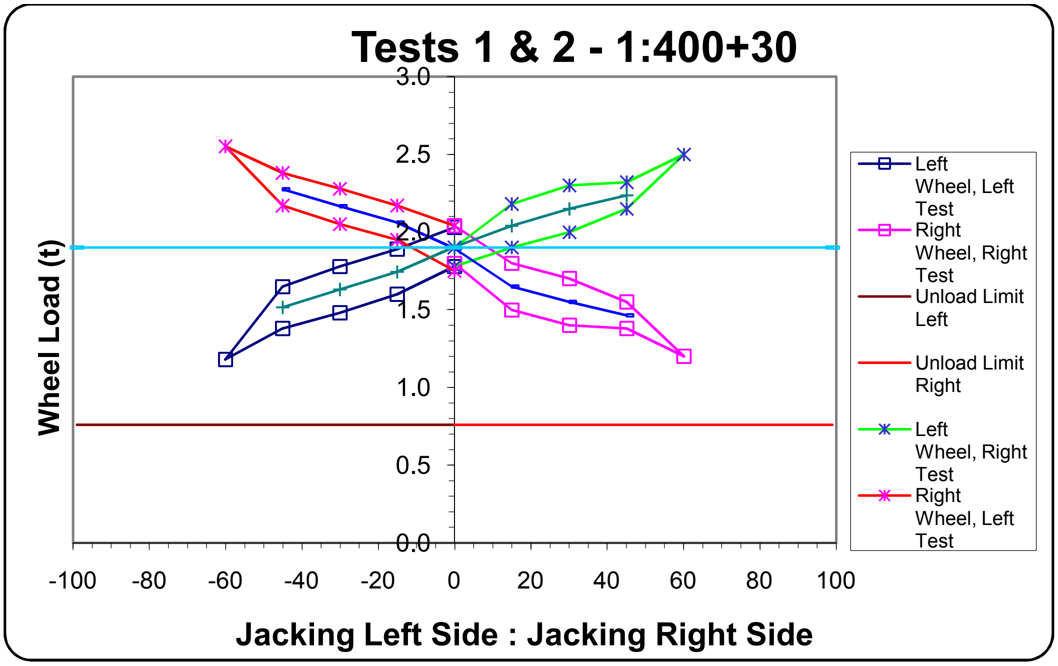


Figure 2-9 VFMQ Wagon hysteresis graph (QR test data)

The right wheel, right test and the left wheel left test are the two important curves because they are the ones measuring the critical wheel. The horizontal axis of the graph is the jacking height. The marked points are positions where the lifting has been stopped to take load measurements.

A comparison of these two graphs indicates that the amount of hysteresis varies widely. From the data observed, the VFMQ wagon is one of the vehicles best exhibiting the effect.

An automated test will enable more accurate production of these important graphs. More intermediate steps can be easily entered into the control software and the jacks will stop at these heights to generate more points on the graph. Also the hysteresis effect will be more accurately measured with an automated test if the cylinders can be kept in near perfect synch.

2.6 Conclusion

This chapter provides all the required background information for the commencement of the design of a new automated rollingstock twist test. The current procedure has been examined first hand by the author and some downfalls of this procedure recognised. These downfalls are hoped to be mitigated by the new design. However more detailed information has to first be researched to determine the complete constraints of this design.

3 Requirements of an Automated Twist Test

3.1 Introduction

This chapter will present the findings of the research conducted into the requirements for a new automated twist test. It is essential to gather all the performance requirements together into a complete list so that no significant requirement is left out and discovered at a later date, causing the need for much redesign. Requirements were discovered through examining documents and communicating with QR engineers.

3.2 Current Twist Test Guidelines

There are many guidelines in relation to the twist test QR performs currently. Most are contained in the internal document, Work Instruction - Static Twist Test. This document has been created to satisfy the requirements set out in QR Standard Rollingstock Dynamic Performance and the document Rail Infrastructure Corporation Standard – RSU 283 – Static twist test. The latter of these two documents describes the procedure for testing rollingstock which is to travel on standard gauge track. Standard gauge track is wider between the rails than the narrow gauge track used in most of Queensland. Most other states in Australia use standard gauge track.

The Work Instruction sets out exactly what is required for the current test. The design of an automated test will need to adhere to many of these guidelines. Specifically, these phrases will have a large impact on the new design:

- **"All brakes should be released including handbrake and spring applied parking brakes"**

This is because the test needs to simulate the suspension as it is freely rolling along the track. Chocks of some sort will be required to keep the tested vehicle stationary as in the current test.

- **“Jacks should be placed under the axleboxes or bearing cups”**

This specifies where the jacks will be positioned which leads to the need for research into the height of these axleboxes so the space available for jacks can be ascertained.

- **“Raising the wheels should be performed in a manner so that they all reach the required height simultaneously.”**

It is stated in the Work Instruction that this is because friction damping within the suspension will cause different load cell readings. The test needs to best simulate the actual conditions on the track, which is having the heights reached simultaneously. Motion control of the lifting equipment will definitely be required to achieve this. The two parameters requiring control will be speed and height.

- **“It is generally advantageous to take the load cell readings at 25%, 50%, 75% and 100% of the required wheel heights.”**

The current test uses this method because extra information about suspension elements reaching their travel limits can be gained. A hysteresis graph is generated from this data and is an important tool in analysing the performance of the suspension.

Using only 4 steps is a limitation of the manual method. With software control it would be easy to measure loads at any number of steps. However this is specified as a static test and the vehicle should be at rest before load readings are taken, or at least there should be no dynamic behaviour induced by the test and recorded.

- **“Before jacking to the next height, the security of all wheels on their chocks should be checked.”**

If the load readings are taken automatically and the test automated to the extent that it can be done “with one press of a button”, then the chocking method used should be checked that it will be secure throughout the whole test.

- **“The side that has just been jacked up should be set down at the same 75%, 50%, and 25% intermediate heights, and readings taken”**

The significance of this is that the system will need speed and height control in both directions because the lowering will have to be stopped at these exact intermediate heights, the same as for the lifting stroke.

3.3 Types of Rollingstock

This is an overview of the types of rollingstock that have been previously tested and are a good indication of what is expected to be tested in the future. The research into these vehicles was specific to the wheel configurations, tare weights, some overall maximum and minimum dimensions of axlebox heights and distance between wheels. These dimensions are relevant to the design of an automated twist test.

3.3.1 3700 Class Locomotive

The 3700 class locomotive is a re-manufactured version of the older 3200 class locomotives, which were built in 1986. It is QR’s heaviest vehicle running on narrow gauge track, with a tare tonnage of approximately 126 tonnes. This is currently the heaviest vehicle that a new twist test system would expect to lift. However some factor of safety for the event of future models being even heavier will be considered.

Many locomotives, including the 3700 class have three bogies with two axles on each bogie. This means that, referring to Figure 2-4, there will be 5 wheels to lift. Some locomotives have two bogies with three axles, exactly as depicted in Figure 2-4, still meaning there will be 5 wheels to lift.

3.3.2 Harsco Track Technologies Stoneblower

The Stoneblower is a track maintenance vehicle. It pneumatically injects ballast under the sleepers to repair the height of the rail. The twist test previously undertaken on this vehicle was quite a difficult procedure. This is because of its unusual configuration of two cars sitting on three bogies. This means there is a distance of greater than 26 meters from leading to trailing wheel. To correctly simulate the dip and ramp angle of 1:400 the trailing wheel has to be jacked to a height of 96mm.

3.3.3 Cairns Tilt Train

The significant dimension of the Tilt Train is its low clearance between the axlebox and the ground. The Tilt Train trailer cars have wheels of diameter 810mm when new and down to 730mm when worn. The distance between the centre of the wheel and the bottom of the axlebox is 180mm. This means the minimum clearance under the axlebox for the positioning of a jack is:

$$\frac{730}{2} - 180 = 186\text{mm}$$

This dimension provides a minimum clearance for the positioning of jacking devices.

3.3.4 Inter-urban Multiple Unit

The IMU is an electric passenger car. It is significant in relation to twist testing it that it has one of the longest spans between bogie centres at 17 meters.

3.4 Key Requirements of the System

From the information gathered so far we can now formulate a list of key requirements.

3.4.1 Load

The heaviest vehicles to be tested are locomotives. In particular the 3700 class is the current heaviest at 126 tonnes. This locomotive has 12 wheels so the approximate load per wheel, in the level position, is

$$126/12 = 10.5\ tonnes$$

However as the wheels are lifted varying amounts, uneven loading will occur and more force will be required. Exactly how much more is difficult to determine because of the complicated suspension system. To find an approximate value it will be assumed that the 5 wheels being lifted will share equal load. Of course this is not the case in practise and a correction factor of 1.2 will be added to account for dissymmetry. To lift one side of the locomotive it will require a force of half the total weight. This force will be divided between 5 lifting points. Another factor of safety of 1.2 will be added to account for future vehicles being heavier

$$(126/2)/5 * 5 * 1.2 * 1.2 \approx 18\ tonnes = 180\ kN$$

The number of wheels to be lifted will never exceed 5 because the maximum number of wheels on any of QR's vehicles is 12. The other configurations will need one or three wheels lifted.

- Lift 180 kN at up to 5 points

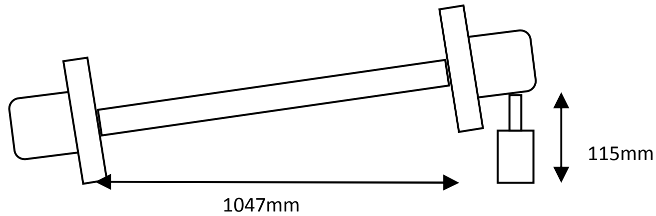
3.4.2 Stroke

- Lift these points to a maximum stroke of 105 mm

QR's standards dictate that the maximum height of the outside rail on a cant ramp is 105mm, see Figure 2-4. The majority of trains tested will require a stroke less than this; in fact the largest stroke required for a test to date is 96mm.

- The lifting device must safely accommodate the tilt of 5.6° created by this stroke.

The train will tilt as one side is jacked up. The significance of this is that the interface between the jack and the bottom of the axlebox should rotate with this tilt. Otherwise all the force may end up on one edge of the jack bearing surface, causing damage to the axlebox and possibly to the jack because of the off-axis loading. The maximum tilt can be calculated from the maximum stroke.



3.4.3 Dimensions of device

- The lifting device is required to fit within dimensions approximately 400 by 400 by 185mm high.

These dimensions are not fixed, but more of a general feel for how much space is available. It would take too much time to research exact clearance dimensions for all the different train types.

The vertical clearance is more critical because it will dictate the collapsed height of lifting devices. As mentioned in Section 3.3, the minimum height from floor to axlebox found on any of QR's fleet is 185mm. A flat floor will be available. Non-metallic material should be used to sit against the axlebox so as not to damage it.

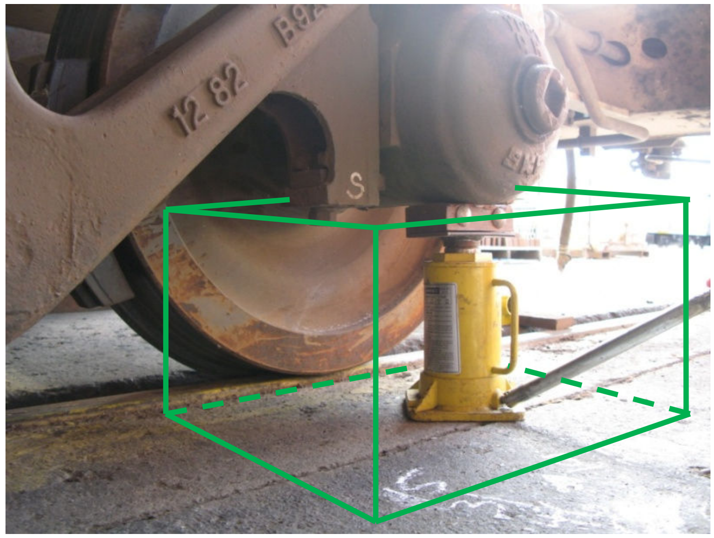


Figure 3-1 Space available for lifting device

- Maximum distance between lifting points is 26 meters
- The distance between lifting points will vary
- Complete system must be transportable in a 4WD to and from the Vehicle and Track engineers' office.

Again this is a general constraint on dimensions. It also means that the system must be designed to be portable i.e. handles required on equipment for easy carrying.

3.4.4 Time constraints

- Desired lifting speed of 115 mm/min

This converts to a stroke of 115mm completed in 60 seconds. An incomplete report by a QR employee specified that the lift should take less than 60 seconds. The lift rate is purely a question of convenience for the personnel running the test. However it is stated in the project aim that the test should be faster than the current manual method.

- Desirable set up time of less than 20 minutes

Set up time is critical because decreasing the total time of completing a test is part of the project aim. Therefore any methods to minimise set up time should be implemented. The equipment will have to be moved from one side of the train to the other as both have to be tested. The time required to this should also be minimised.

3.4.5 Automation

- Stop the lift automatically at four or more stages i.e. 25%, 50%, 75% and 100% of full height. These heights vary from vehicle to vehicle so a means of inputting the dimensions into the control software is required.

As mentioned in Chapter 2 the calculation of the heights to lift each wheel is made from the bogie wheel base and bogie centre distance dimensions and the track conditions. This calculation can be either integrated into the system or pre-calculated and input simply as a height for each lifting point.

- Minimise the amount of input by the operator.

This point is to help reduce the overall time taken for the test. Ideally the parameters of the individual tests would be input into the software beforehand in the office, then only a few presses of buttons required once the system is in place in the workshop.

- Stop the lift within 1mm of the required height. The required heights will be different for each point.

- Lift and lower the points simultaneously to differing heights. The points should all reach their specified height within approximately 2mm of each other.

This is better accuracy than you would expect from the current manual twist test. From looking at previous hysteresis graphs, a lag of 2mm by one cylinder will not affect accuracy of measured data.

- Measure load readings of the critical wheel of interest at all five heights.

The readings are simply taken from the portable Train Weigher or load cells once the lift has stopped at intermediate heights. Load readings will not be taken automatically.

3.5 Conclusion

A complete list of performance requirements for an automated twist test has been established. The next step in this design is to start to examine equipment which will fulfil the above requirements.

4 Existing Systems used for Synchronous Heavy Lifting

4.1 Introduction

The basic function of the system required to complete the twist test is that a heavy load has to be lifted synchronously. It is important with any new design to research the pre-existing products that are available. There are two methods of lifting heavy loads which are already used extensively at QR for lifting rollingstock; power screws and hydraulics. This chapter aims to review these two options and to identify products that use these methods of actuation and incorporate electronic control.

4.2 Hydraulic Synchronous Lifting Systems

There are a few hydraulic lifting products on the market that have already been developed as a package with electronic position control. They are called synchronous lifting systems.

However after examination of their capabilities it is obvious they will not be suitable for direct application to twist testing because they are all designed towards keeping heavy loads balanced. This means the displacement of each lifting point is controlled so that it moves upwards as close as possible to exactly the same rate, thus keeping all cylinders at the same level. The twist test needs points to be lifted to varying heights. They are also much too expensive for this application. However studying these systems gives valuable insight into possible methods of lifting synchronously.

4.2.1 Enerpac SLS

Enerpac is a worldwide producer of hydraulics equipment. They sell a system called the Synchronous Lift System (SLS). The SLS has been used in bridge launching, dragline maintenance and tunnel jacking. Some extremely heavy objects have been lifted with this system such as a 3500 tonne dragline in a recent project at Carragh coal mine, Queensland. The dragline had to be lifted 200mm so that the main pivot bearing could be replaced. A picture of the SLS in action on this project is shown below.

For this project the hydraulic system was composed of 80 high pressure hydraulic cylinders. They can be seen grouped together in four's with stabilising structure in Figure 4-1. Also visible in the photo is the pump, control equipment, hydraulic hose and electrical cable for feedback from the cylinders.

Figure 4-1 Dragline lifting with the Enerpac SLS (Enerpac catalogue)

The system makes use of a Programmable Logic Controller (PLC) to very accurately control the movement of up to 64 hydraulic cylinders. A PLC is a dedicated hardware device that controls and monitors equipment using electrical input and output ports. The purpose of such an accurate system is primarily that the object is lifted safely with little risk of the load slipping horizontally and it ensures that no excessive stresses are placed on the object and cylinders.

As described in the document Lift Anything, (Hohensee, 2005) cylinders are positioned and extended to a specific initial load, the user specifies a tolerance and a target, then the system is put in automatic mode and lifting/lowering commences. While in automatic mode the user has no input. Lifting is controlled by feedback from pressure sensors and Linear Variable Differential Transformers (LVDT's). LVDT's are a device which outputs a voltage proportional to its armature displacement and thus can provide position feedback.

On this system LVDT's are connected parallel to each cylinder. They are not placed on the actual cylinder because in the initial stages of lifting very heavy objects the displacement of the cylinder is not exactly the same as the movement of the object. There will always be a small amount of slack in the cylinder. Therefore the LVDT's are connected to the load and to a stationary platform next to the cylinder.

The pressure sensors are used to measure load at each cylinder. If information about the position of the cylinders is combined with this load measurement, the centre of gravity can be calculated continuously as the object is lifted. The centre of gravity information is vital in ensuring the object is balanced. The pressure sensors are also used in initial positioning of the cylinders. The cylinders are told to extend until they reach a certain load.

This system is interesting in that control is performed via 'digital hydraulics'. In this sense digital hydraulics simply means that every cylinder is controlled by a 2/2 valve that only has on and off positions. No proportional valves are used. The reason that this is acceptable is that they use high performance valves that can switch from closed to open in 30 milliseconds and the lifting will be done appropriately slow. The rate of reporting from the LVDT's is also very fast.

The software control works with the use of a protocol file which indicates how often the measured values have to be achieved. In effect this is dividing the stroke into many small targets for the software. These divisions can be set to a fraction of an inch or second. The LVDT’s start tracking the position of the cylinders when the system enters automatic mode. Once a cylinder reaches one of these intermediate targets it is switched off. If one cylinder is lagging behind the others outside the tolerance, the 2/2 valve opens briefly, followed by a new reading. Once all cylinders have reached the increment specified in the protocol file, the software moves on to the next target. This correction process happens very rapidly and it appears that the cylinders are moving upwards continuously.

Enerpac uses a hydraulic rectifier on each of its cylinder circuits so that oil can only flow in one direction. This lets each cylinder act independently and thus means that different sized cylinders can be used at each lifting point.

The Enerpac system allows input and output of results through the PLC/touch screen input device. Figure 4-2 is a picture of the components used in this system.

Figure 4-2 Enerpac SLS system components - note touch screen in bottom left corner

This system costs approximately $120 000 for the control system alone, without cylinders. This seems ridiculously priced but being one of the few systems of this kind on the market, Enerpac is at liberty to set their prices high.

4.2.2 Euro Press SLS

The Euro Press SLS appears to work in a similar fashion to the Enerpac version. It also uses the concept of digital hydraulics with feedback from LVDT's. The tolerance of variation between cylinders can be set, equivalent of the protocol file on the Enerpac SLS. Also an accuracy value can be input, which specifies at what percentage error the valve will be opened.

This system also uses a PLC but relies on a laptop running Microsoft Windows to provide input and output. It ships with necessary software into which the user can input how many cylinders are being used, effective areas of these cylinders, whether they are single or double acting and accuracy tolerances. The controlled lift is also started and stopped from the laptop.

4.2.3 Hydra-Capsule SLS

Hydra-Capsule is an English company who supply another line of the same high pressure jacks and control systems. However they sell a few different variations of these synchronous lifting systems.

They sell a fully functioned system exactly like the two above using the same digital hydraulics concept and a laptop for input. (Hydra-Capsule 2007)

They also sell a unit called the multiple-flow unit. This synchronous system has no electronic control and just relies on multiple pumps driven from the same motor to provide even flow to the separate cylinders, regardless of pressure differences. Obviously the same size and number of jacks will have to be used on each pump outlet.

The other SLS variation they sell is their method using single-flow pumps. This system is used for operations carried out over large distances. A number of individual pumps are used and each individual pump is monitored and its speed controlled so that cylinders move synchronously.

Their method of gaining position feedback is described on their website. LVDT's are mounted in a mechanism that serves to provide attachment to the load. A plumb bob weight is suspended from the load by a wire and the LVDT is secured to a stationary object. The wire is kept taut by the spring return mechanism inside.

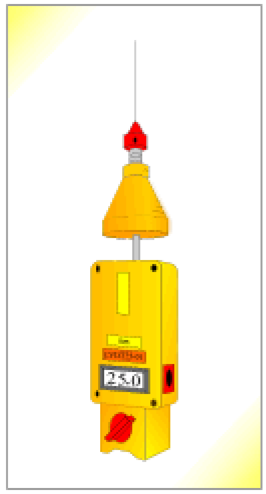


Figure 4-3 LVDT mounting used on Hydra-Capsule systems (Hydra-capsule 2007)

4.3 Power Screws

A power screw is sometimes called a linear actuator, screw jack or a translation screw. Their purpose is to create linear motion from rotary motion. A depiction of a power screw is shown in Figure 4-4.

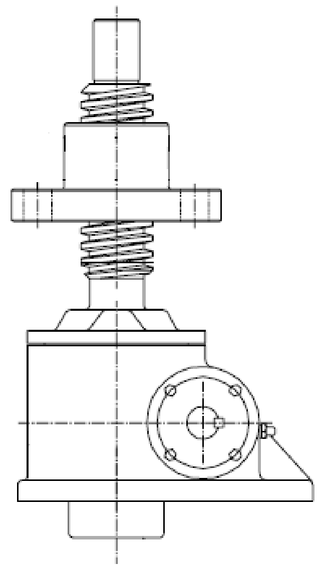


Figure 4-4 A power screw

Mechanical advantage is gained through the screw mechanism, which in turn means that the nut or the screw has to be rotated relatively quickly to produce the linear motion of the load. This leads to the idea of using an electric motor as the means of generating rotation as electric motors are most efficient when running at high speeds.

An electric motor drive will be essential for power screws in their application to the twist test because of the high precision needed. The speed of an electric motor can be controlled by altering the voltage applied to it and the linear position can be kept track of by recording how many fractions of a rotation the motor has turned. Feedback is provided by encoders or resolvers. Encoders work by converting their position to a digital or analogue code. The code is generated by mechanically or optically recognising a pattern on the rotating shaft. Resolvers are a type of rotary electrical transformer and produce an analogue output (Wikipedia 2007).

Motors incorporating position control are very common because they are the primary source of power used in robotics. There are two types of electric motors, servomotors and stepper motors. The stepper motor, as its name suggests, has its rotation broken into a large number of steps. The required angle of the shaft can be reached by electronically informing the motor which step it should be at. Feedback is not needed to control position, although it can be used. Stepper motors have the disadvantage of sometimes missing steps if there is a large torque applied or when driven at high speeds. If the motor starts to miss steps the accuracy of its position control will obviously suffer. The twist test application will make high torques because of the heavy loads and therefore stepper motors are not suitable.

Servo motors on the other hand rely on feedback from resolvers or encoders to accurately move to a certain angle. The feedback device is often built into the servo motor package. Servo motors are much more commonly used in high load applications such as the twist test and for that reason there are a range of high powered motors on the market exceeding the power required to lift a locomotive in acceptable time.

A search of the internet revealed no ready-made synchronous lifting systems incorporating power screws. However there was plenty of evidence of their suitability for the twist test and the components of a likely system are readily available.

4.4 Conclusion

The most important information gained from this study is the fact that all synchronous heavy lifting systems common on the market today use hydraulic cylinders as the lifting mechanism and 'digital hydraulics' as the control scheme.

Interestingly all systems also use high pressures of 700bar. This is most likely because it is more cost effective to manufacture high load cylinders for high pressure since a smaller effective diameter is needed. The cylinders will also be lighter for transporting and positioning.

The method used by Hydra-Capsule of synchronising the lift by flow adjustment without electronic control could possibly be developed into a viable system for the twist test.

Power screws coupled with servo motors also provide a feasible lifting system at this stage.

The study of these synchronous lifting systems has provided invaluable insight into possible designs for a twist test system. The next chapter will deal with determining the best control method and check feasibility of using such components as power screws and hydraulic cylinders.

5 System Concept

5.1 Introduction

There are a number of options and schemes when considering components and control systems for the twist test. The two areas that will govern the rest of the system design are the lifting mechanisms and the control system. Once a decision has been made on the best choice of options in these areas the rest of the design for the system should be straightforward. This chapter aims to introduce concepts and discern which appear viable at this stage.

5.2 Lifting Devices

A decision must be made on which lifting device will be most appropriate for the twist test so that the rest of the system can be designed around the chosen device. From the review of lifting mechanisms in the previous chapter the two viable options of are:

- Power Screws
- Hydraulic cylinders

Other devices, such as winches or scissor lifts, fail to meet performance requirements because they are not portable.

We already know that hydraulic cylinders are viable because they are used currently on the manual twist test as described in Chapter 2.

Figure 5-1 Hydraulic cylinder are viable for the twist test

The same cannot be said for power screws and their viability still has to be checked.

5.3 Power Screw Viability

Linear Bearings, a power screw supplier located in Brisbane, was contacted and their range of power screws assessed. For heavy loads, they recommended their Powerjacks range. A couple of different configurations of jacks available in this range are shown in Figure 5-2.

Figure 5-2 Powerjack E series screw jacks

There were two options for achieving the required load rating of 180 kN. There is a 300 kN version of Powerjack, but this has a collapsed height of 235mm or more depending on the type of top plate used, and it weighed 86 kg. This was not acceptable because the inadequate collapsed height would not have allowed the jacks to fit under a large quantity of vehicles to be tested. Also 86kg cannot be considered easily portable. The other option was utilising two 100 kN jacks. These have a collapsed height of 147mm and only weigh 25kg each.

The next problem was to specify the appropriate configuration and placement of these screw jacks. There are two options, rotating or non-rotating.

5.3.1 Rotating Type

The rotating version of a screw jack rotates the screw while a nut is driven up and down the screw. Observably, the nut has to be restrained so that it cannot rotate.

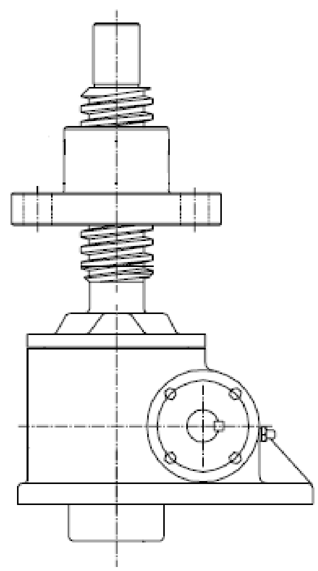


Figure 5-3 Rotating type screw jack (Power Jack catalogue)

The problem with this configuration is the screw protruding from the top, which is where we want to rest the axlebox. However as there are two jacks needed they could be configured as in the conceptual drawing, Figure 5-4.

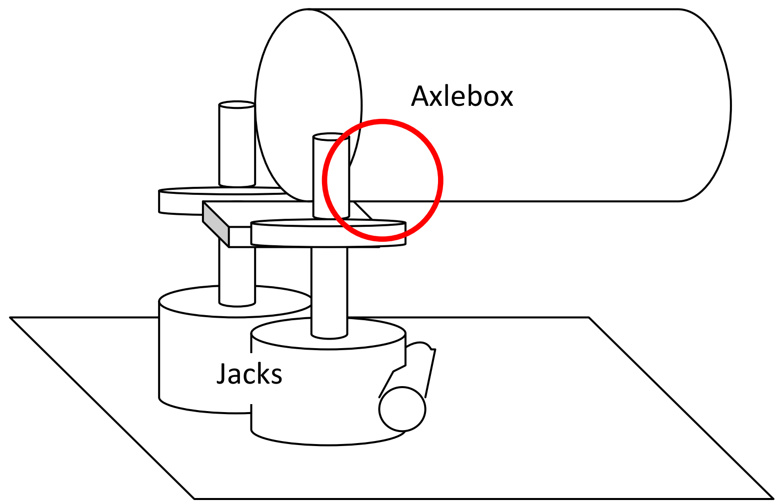


Figure 5-4 Rotating type jack placement

Unfortunately the jacks will foul with other parts of the bogie on some trains. Certain types of bogies have structural members in the area of the red circle shown in Figure 5-4, which will not allow the jacks to be positioned as they are here. Therefore there is no viable solution using rotating type jacks.

5.3.2 Non-rotating Type

A non-rotating screw jack has a worm gear mechanism inside which allows the screw to raise and lower without rotating. This is useful because we do not want the surface touching the axlebox to rotate. However the disadvantage is that it means the screw needs to be stored below the motor when in the lowered position, as seen in Figure 5-5.

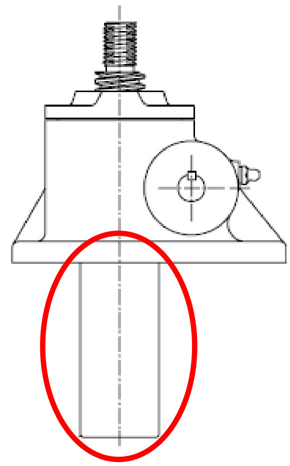


Figure 5-5 Non-rotating power screw (adapted from Power Jack catalogue 2004)

This type of jack does not appear to be designed with the purpose of jacking from the floor. However, Linear Bearings came up with the solution shown in Figure 5-6, using the non-rotating screw jacks:

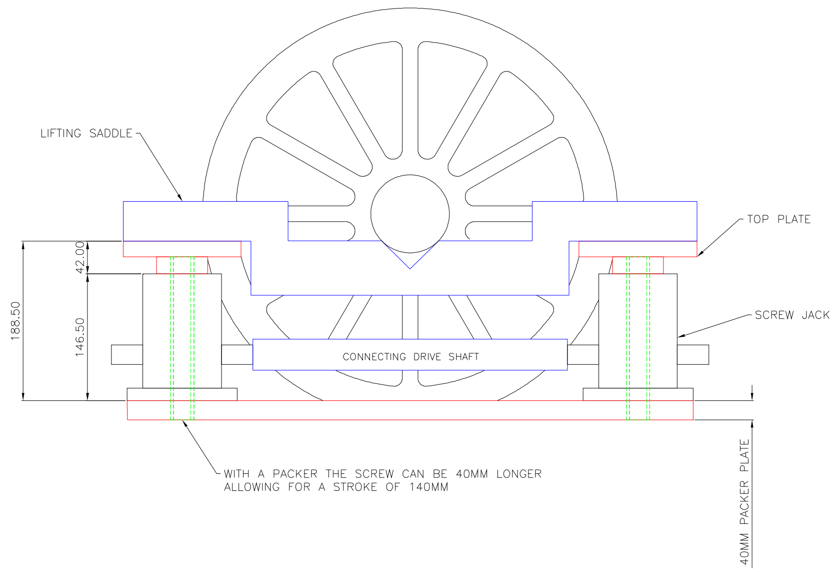


Figure 5-6 Linear Bearings solution for jack placement

Again, similar dimensional constraint issues as with the rotating type concept could arise. Also the whole lifting mechanism is starting to become quite large. By the time servo motors are added, this device would weigh at least 70kg, which is not easily portable.

5.3.3 Power screw viability conclusion

The viability of power screws as lifting devices for the twist test has been analysed and it can be concluded that they are not feasible. The issue of portability was the main concern and hydraulic cylinders are much more favourable in this area. Therefore from this point forward only hydraulic systems will be considered.

5.4 Traverse-Time Graph

The cylinders have to move in the fashion shown in Figure 5-7. Each cylinder has to move to different positions but in a synchronous manner.

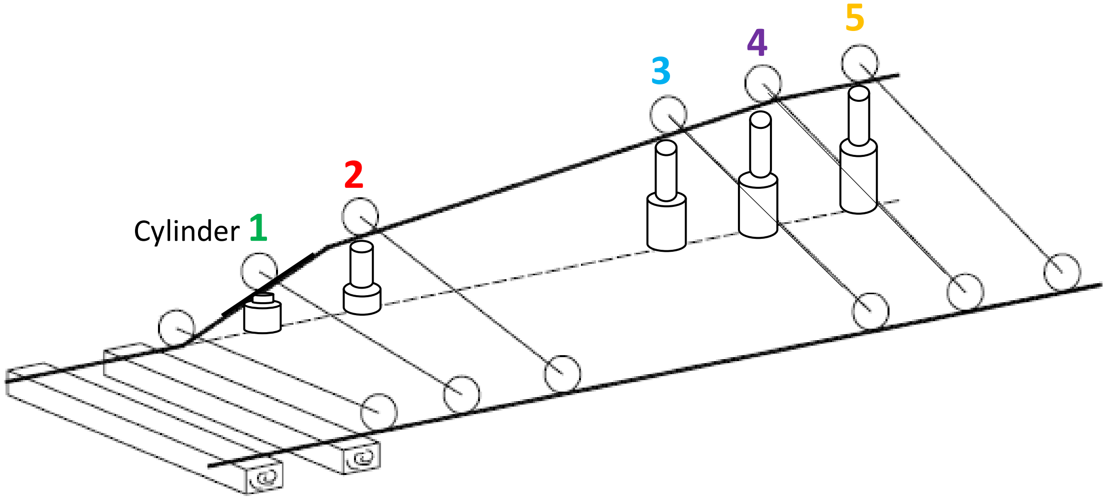


Figure 5-7 Cylinder positions (adapted form QR standards)

The stroke and load required has been calculated in Chapter 3. We can put this together with the synchrony required in the traverse-time graph shown in Figure 5-8.

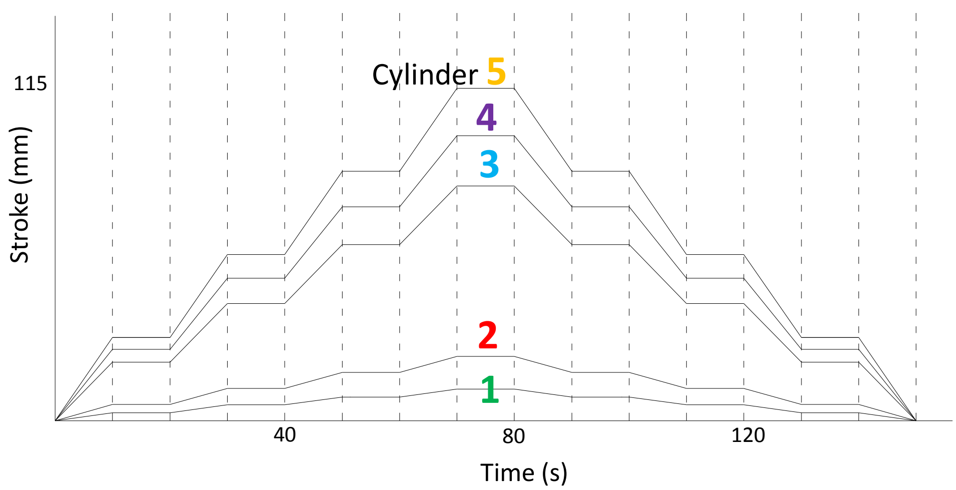


Figure 5-8 Traverse-time graph. Each line represents a lifting device's position

The traverse-time graph gives an easily readable picture of what movement of the cylinders is required. This is the particular traverse-time graph for rollingstock with 6 axles and a length long enough to require maximum stroke. Obviously the graph will vary depending on the number of wheels and dimensions of the train.

Traverse-time graphs are usually used in hydraulics to visualise the sequencing of cylinders. The twist test system does not require any sequencing in the sense of cylinders hitting limit switches and activating another cylinder. However the graph depicts clearly that there needs to be some sort of control to move the cylinders relative to each other i.e. 'micro-sequencing'.

The time delay between strokes is not fixed as shown on the graph as the horizontal portions of the stroke; it will be controlled by the press of a button to start the next stage of the stroke.

5.5 Control Methods

Control theory can be split into two distinct methods, open-loop and closed-loop. Open-loop control means that a device is controlled without feedback. The output is simply proportional in some way to the input. An example of open-loop control is a heating system where the output of the heater is calculated from the outside temperature only. Closed-loop control requires a device to provide feedback, from which an error from the target can be obtained. Input into the device to be controlled is then based on this error. An example of closed-loop control is where the aforementioned heating system adjusts its heat output according to the difference between the inside temperature and a desired temperature, such as 20°C.

The twist test system needs a closed-loop system. The foremost reason for this is the variance in the load between different rollingstock. In fact the accuracy required would be difficult to obtain even if the same load was lifted every time. Open-loop systems are not generally used in position control where any accuracy is required because of the uncertainties of friction and other unforeseen disturbances.

An example of an open-loop hydraulic system is one where the position of an actuator is controlled by opening a valve for a certain amount of time, see Figure 5-9.

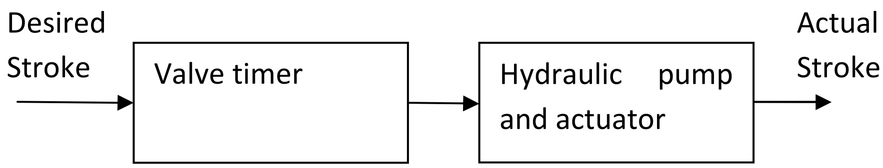


Figure 5-9 Open-loop control example

The amount of time is the input and the displacement of the cylinder is a function of such parameters as: flow rate, cylinder diameter and cylinder leakage. Obviously it will only take a small variation in parameters, such as the voltage applied to the pump and the flow rate will change, causing considerable error in the result from the predicted displacement. A system such as this will certainly not generate accuracy of better than 1mm.

An example of a closed-loop hydraulic system is where the same actuator has a position transducer parallel to it, generating a signal that can be fed back to a Programmable Logic Controller, see Figure 5-10.

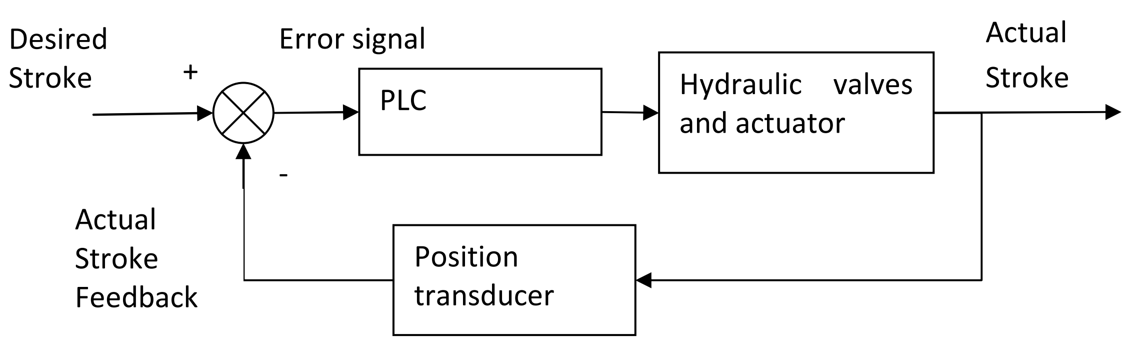


Figure 5-10 Closed-loop control example

The PLC contains code to analyse the feedback and output a suitable position of the hydraulic valve to minimise the error in the actual stroke compared to the desired stroke. This type of system should provide the accuracy necessary for the twist test application.

5.6 Control Concepts for a Hydraulic System

There are three concepts of control for a hydraulic system, ranging from simple and probable low accuracy, to complex with expected high accuracy. We will give them these names to distinguish between them:

- Limited feedback
- Pump speed control
- Digital control
- Proportional control

5.6.1 Limited feedback system

The purpose of this concept is to minimise the amount of control hardware needed, thus reducing equipment cost and design complexity.

The idea is to pre-set the flow entering each separate cylinder before the lift is started. The flow should be set, using manual flow control valves, to a rate proportional to the stroke required for that cylinder. Once the lift is started the cylinders should then move up at the correct rate so that they reach the end of their stroke at the same time. This will mitigate the need for position feedback transducers and these are quite expensive so this is a worthwhile concept to analyse.

The question arises of how the lift will be stopped at exact intermediate heights without position transducers. Cam-operated valves could be placed so that they are tripped when the cylinders are at the correct height. A mechanism would need to be designed to house these valves and allow the cam stop to be slid to new positions, see Figure 5-11. The concept is basically to use hydraulic logic rather than electronic. It would have to be tested but it may be possible to achieve 1mm accuracy using this method.

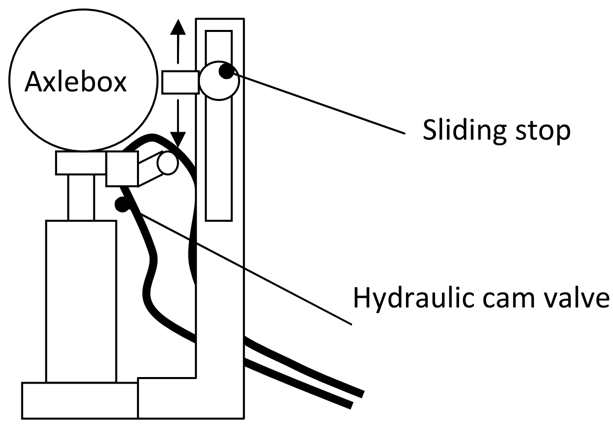


Figure 5-11 Hydraulic logic concept

The other concern with this system is setting the flow to exactly the right amount will be difficult. A table would have to be generated to specify the amount of turns the flow control valve should be turned for a specific total stroke. This is not expected to be very accurate and to analyse whether this form of operation is acceptable we need to look at the effect of hysteresis on the accuracy of results. Below in Figure 5-12 is a hysteresis graph exhibiting a large hysteresis effect.

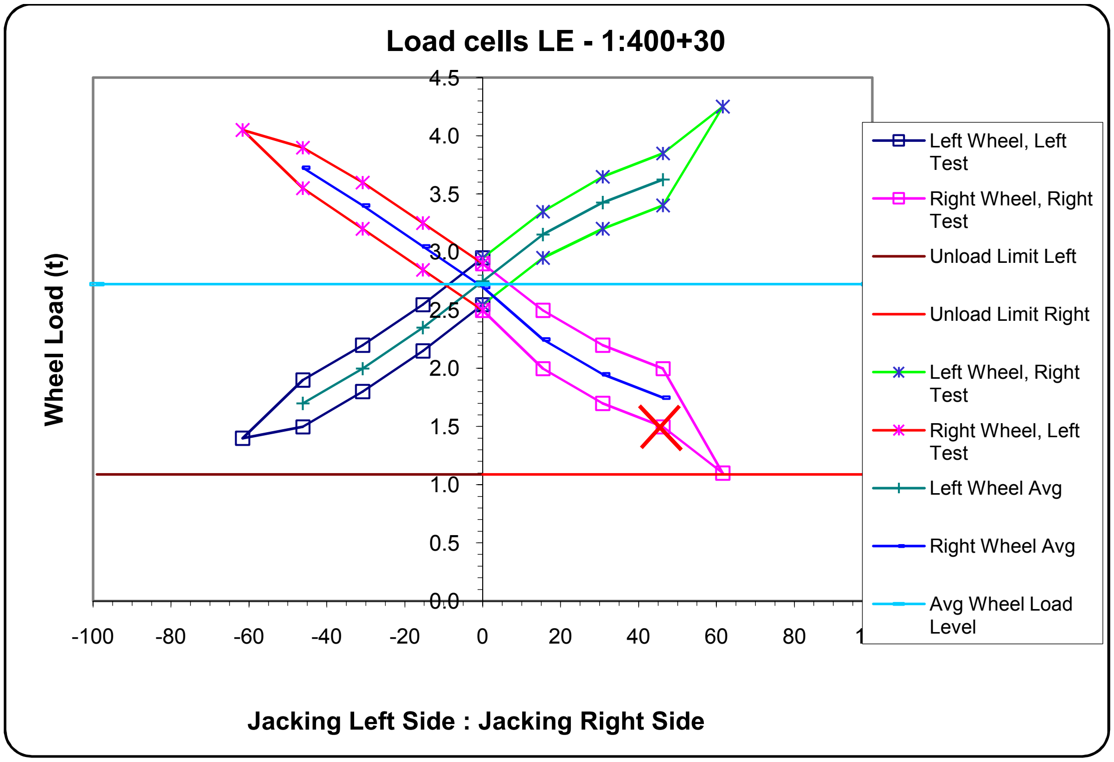


Figure 5-12 Hysteresis graph of a DSOP 43181 wagon (QR test data)

It is impossible to get a completely accurate value from this data but we will attempt to analyse the graph and see how much effect a lagging cylinder will have.

If we take the point marked on the graph at 75% stroke on the right side, the difference between loads on the up-stoke and down-stroke for the right wheel is 0.5 tonnes. This means that 0.25 tonnes of the load is being caused by the hysteresis effect. The previous value on the up-stroke (50%) is 0.2 tonnes more than the load at 75%. There were three cylinders contributing to the lift in this test. The average cylinder distance moved for the stroke from 50% to 75% was 11mm.

Now assume that this rear wheel lags the maximum amount behind the other two i.e. it only starts moving after the other two have reached 75%. This will have the same effect as releasing and re-raising the rear cylinder and this will be releasing approximately 1/3 of the load caused by the hysteresis effect, or 0.083 tonnes.

So as a guide we could say that for this test the effect on the load per asynchrony mm is:

$$\frac{0.083}{11} = 0.0075 \text{ tonnes/mm}$$

That is, the load recorded will be inaccurate by 7.5 kg for every mm that a cylinder lags behind the others. For this test, since it was already so close to the 60% unloading limit, this means that a cylinder only had to lag 2mm for the test to have passed when it really should have failed.

Of course this all depends on the particular test and other rollingstock will exhibit greater inaccuracies due to asynchrony. The actual justification for synchrony of 2mm or better is to be more accurate and repeatable than the manual jacking method. These calculations just serve to provide a judgement on the cost of asynchrony. Whether an inaccuracy of 20 kg on a 20 tonne wagon will be the difference between derailing or not is very doubtful.

In conclusion, probably the greatest concern with this system is the inability of stopping the lift within 1mm of the desired height with a hydraulic cam switch. Also the system does not satisfy the requirement set out in chapter 3 of minimising the amount of input by the operator. It would take a few minutes for each vehicle, to look up a table for the amount of turns the flow valve should be opened and

5.6.2 Pump speed control

This concept was conceived when looking at systems for synchronous lifting. Hydra-capsule mentioned on their website that they sometimes use variable-flow pumps as the control method in their lifting systems. The idea is to move the control back to the flow of the pump in the control loop as seen in Figure 5-13.

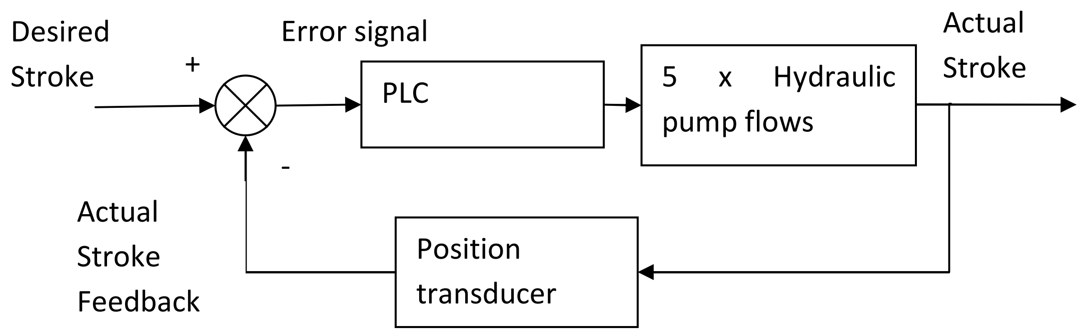


Figure 5-13 Pump speed control feedback scheme

The PLC will control the flow of five separate pumps, with each pump thus controlling the speed of a cylinder. Ideally no valves will be needed, with even the lift to be stopped at a height with only the control of the pump. The advantage in this is power will be saved because valves are not used to dissipate energy as heat.

Flow from a pump is usually controlled by varying the displacement, not the speed of the motor. Displacement can be controlled manually, electronically with servos, or with internal hydraulic controls (Rohner, 1995). Obviously the option we need is servo control.

There are few models of variable-displacement pumps with servo control available on the market. Parker sells a model, but it is designed with the purpose of being driven by a power source such as a tractor P.T.O. Designing a whole new pump is not desirable for the twist test system.

No hydraulic pumps driven by variable speed motors could be found on the market as a standalone unit. This is probably because it is not a practical option and design of this type of pump system will not be looked into. The concept has been used before but the lack of available products means that extensive design work would need to be completed.

The issue with a variable displacement pump is that it will be difficult to stop the cylinder movement completely. The motor will still be rotating while the cylinder is stopped and we will need a complex control system managing the variable displacement to keep the cylinder steady at a certain height.

Another negative for this concept is that variable displacement pumps, due to their complicated internal mechanisms are much more expensive than fixed displacement pumps (Rohner, 1995).

With so many disadvantages we can conclude that this concept can be immediately discounted. It appears that variable-displacement pumps are not suitable for a portable type system such as is needed for the twist test. They are more suited to large production applications where power savings are more critical. Hydra-Capsule must have used this type of system where conventional valve control was not viable because of the large distances between lifting points. In this situation hydraulic pipes would cause large amounts of friction and the use of electrical lines to transmit power is a more economical option.

5.6.3 Digital control

Again this concept is spawned from the review of synchronous lifting systems. All three systems reviewed used the digital hydraulic concept, leading to the notion that there is merit in this method.

The digital hydraulic system uses the closed-loop control scheme shown in Figure 5-10. The valves used to control the cylinders are simple directional control valves, but are switched on and off according to the control algorithm programmed into the PLC.

The synchronous lifting systems used software to split the lift stroke into tiny segments and ensured all cylinders had travelled this segment before opening all the valves again to progress to the next segment. The twist test requires that cylinders progress at different rates. The obvious solution is to make the segments of varying size for depending on the stroke required at a particular cylinder. This is the same as saying that each cylinder's stroke should be divided into the same number of segments.

The issues with this concept may be the difficulty of sourcing valves with quick response times and this system is expected to be slower than the other methods because of the restriction of flow that the constant switching will cause.

5.6.4 Proportional control

The proportional control method uses the same control loop as shown in Figure 5-10. Indeed it is very similar to the digital hydraulic method, but the directional control valves are replaced with proportional valves or servo valves. There are a wide variety of proportional valves with different internal mechanism and slightly different performance, but their basic function is to allow a flow though the valve that is proportional to the input voltage applied to the solenoid.

The proportional valves will allow for very precise control of speed and position of the cylinder.

A negative of proportional control is that proportional valves are somewhat less robust than directional control valves. They are more susceptible to particulate contaminants in the hydraulic oil (Rohner 1995). They are also more expensive and design of software for the analogue valves will be more complex.

5.7 Conclusion

Reviewing all of these concepts, the digital and proportional hydraulic are both viable at this stage. If the digital hydraulic system works acceptably the proportional hydraulic valves have no advantage and the higher cost and more complex software means that they are less desirable. Hence a digital hydraulic system will attempt to be applied to the application of the twist test. The next stage of design is to somehow appraise the performance of a digital hydraulic system.

6 Analysis of Digital Hydraulics

6.1 Introduction

The digital hydraulics system seems to be a suitable control system for the twist test. However there are a few issues that need to be analysed: whether the response time of valves will affect accuracy and if the switching nature of the system restricts flow too much. It will be attempted to answer these by the use of a computer simulation. Flow restriction is important because it will take a lot of power to lift a locomotive and therefore we may need a large pump. We want the pump to lift the train quickly but to still be small enough to lift into the back of a 4WD. Another purpose of the computer analysis is to determine how difficult it is to write code for the application of digital hydraulics.

6.2 Model Description

The model was constructed with Matlab code. The desired output was an animation of the position of the cylinders as the model stepped through time. This animation was easily generated using Matlab's inbuilt plot command. Only three cylinders were simulated as more would not help in analysing digital hydraulics. The strokes for these cylinders used in most of the analysis were 120, 80 and 40mm and the space between cylinders was constant so that the success of the simulation could be checked visually by ensuring that a straight line could always be drawn between the points as shown in the plot below.

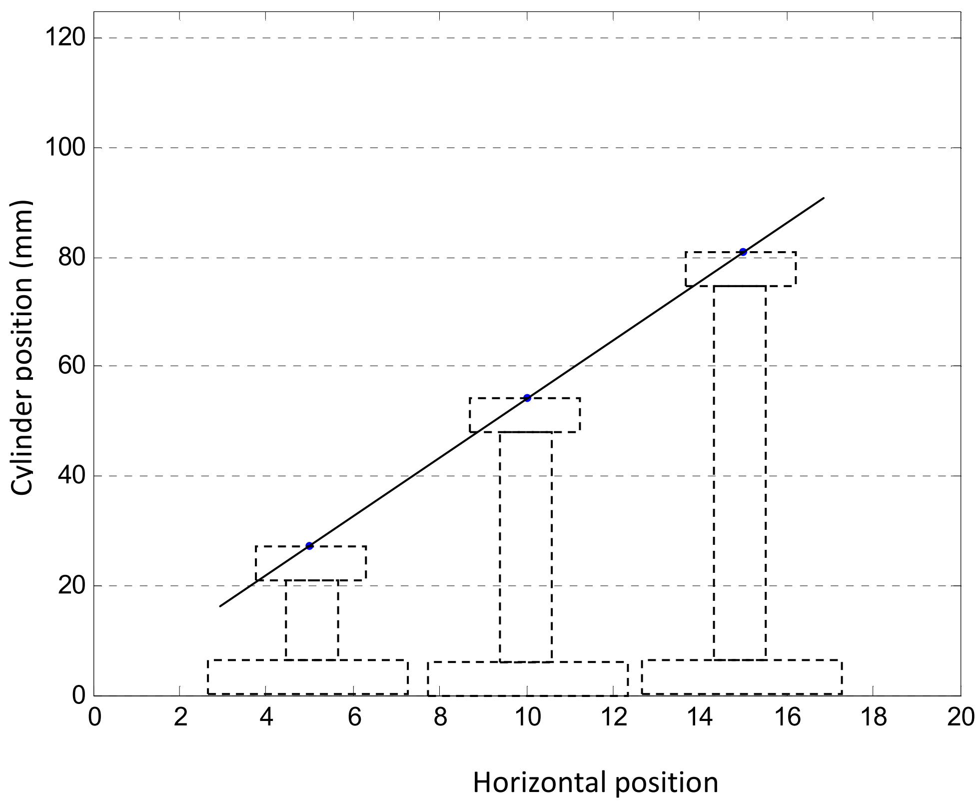


Figure 6-1 Cylinder positions mid-traverse showing synchronous behaviour

To test the animation code it was initially set up to simply step through time and raise the dots representing cylinders to equal heights.

From there the stroke was broken into miniature targets, called synchrony targets in the software. As the name indicates these are the targets each cylinder has to reach before they all progress to the next target. As the model loops a displacement is added to the current displacement. If a cylinder has not reached its synchrony target its valve is left open. The model continues to loop until the synchrony target is reached or exceeded and then the valve is turned off. The number of these synchrony targets that a stroke is split into is called the resolution. Increasing the resolution is expected to increase the time for a lift but increase the accuracy.

The velocity of the cylinders when a valve was fully open was specified in the code as 2.7mm/s. This value was calculated as a ballpark figure by looking at pump flow rates of some portable pumps and cylinder sizes from an Enerpac catalogue.

The intermediate targets are where the lift is stopped the suspension component measurements are taken and the load under the critical wheel recorded. Once an intermediate target is reached all valves are turned off and the cylinder position is recorded so it can be checked for accuracy. A key is pressed to progress to the next target much like what is expected on the real system.

Once the cylinders reach the final stroke the signs of the cylinder position and the targets are reversed. This way, the same code to check cylinder position against targets can be used for the downstroke. At this stage the code in the real system would specify that the downstroke has begun and that each time a valve is turned on, this would mean the valves are actually moved to the release position.

The model behaved appropriately and the “cylinders” could be seen moving up the screen at varying rates but in synchrony. However the model was not realistic because there were no inaccuracies of the LVDT’s or valves included. This was giving a forgiving picture of the system because sometimes the cylinder positions would stop *exactly* on the target. Aside from the fact that this is unlikely to happen in real life it meant the accuracy of the system could not be meaningfully assessed.

LVDT and valve flow inaccuracies were simulated by adding different random amounts to the cylinder position and the cylinder velocity each time the model looped. The errors were changed each loop instead of having just a static error as this will be a much better simulation of real life.

To test the importance of valve response time, code had to be written to simulate the position of the valve between its open and closed position. This was done by creating a new variable for valve position and if the command is given for the valve to move for closed to open position the valve moves the correct amount evaluated from the value of the time step and the valve response time. For this simulation it was assumed that acceptable accuracy would be gained by relating flow to valve position in a direct linear manner.

6.3 Results

Two plots were produced; the animation of cylinder positions as they moved up and down, and a plot of cylinder position vs time. The animation is hard to view on paper but it was essential for checking the sequence of the code and ensuring that the cylinder did stop at intermediate heights. An example of the animations is given in Figure 6-2. The position vs time graph was useful for analysing the accuracy of the model and an example of these graphs is given in Figure 6-3.

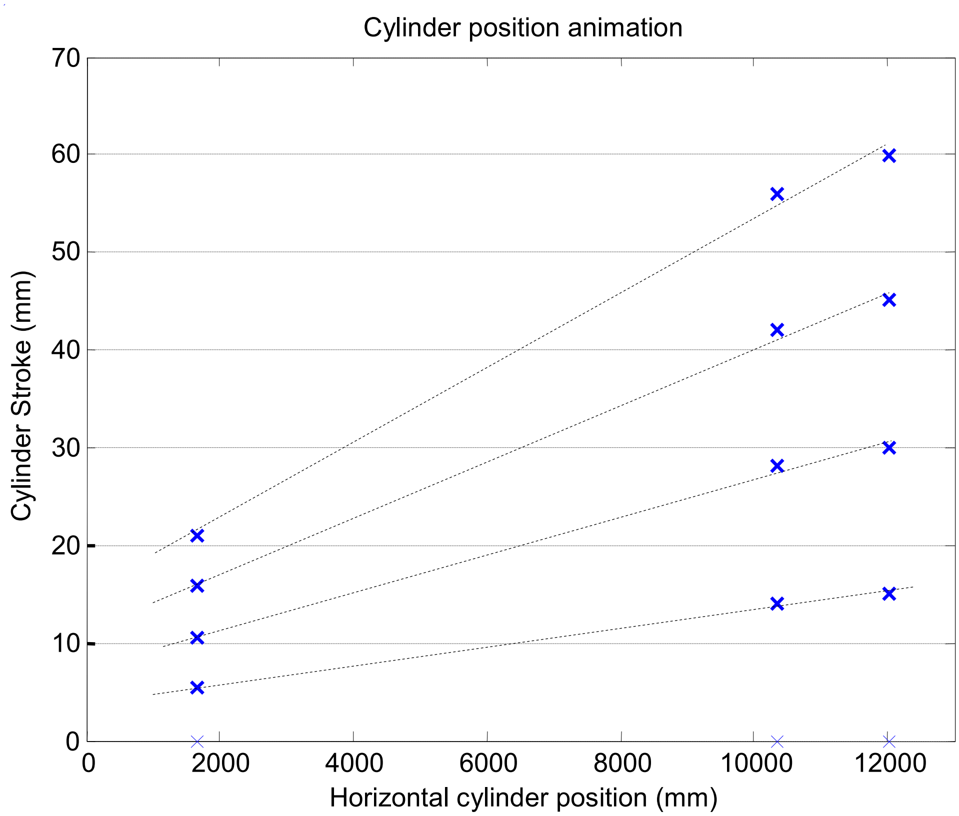


Figure 6-2 Animation plot

Figure 6-2 shows the actual intermediate positions of a test previously done on a VFMQ wagon. As described in Chapter 3, the cylinders have to be stopped at intermediate heights to take load readings. From each position shown here, a key is pressed and the dots move to the next position on the screen. The four positions on the graph depict the synchronous behaviour that is required.

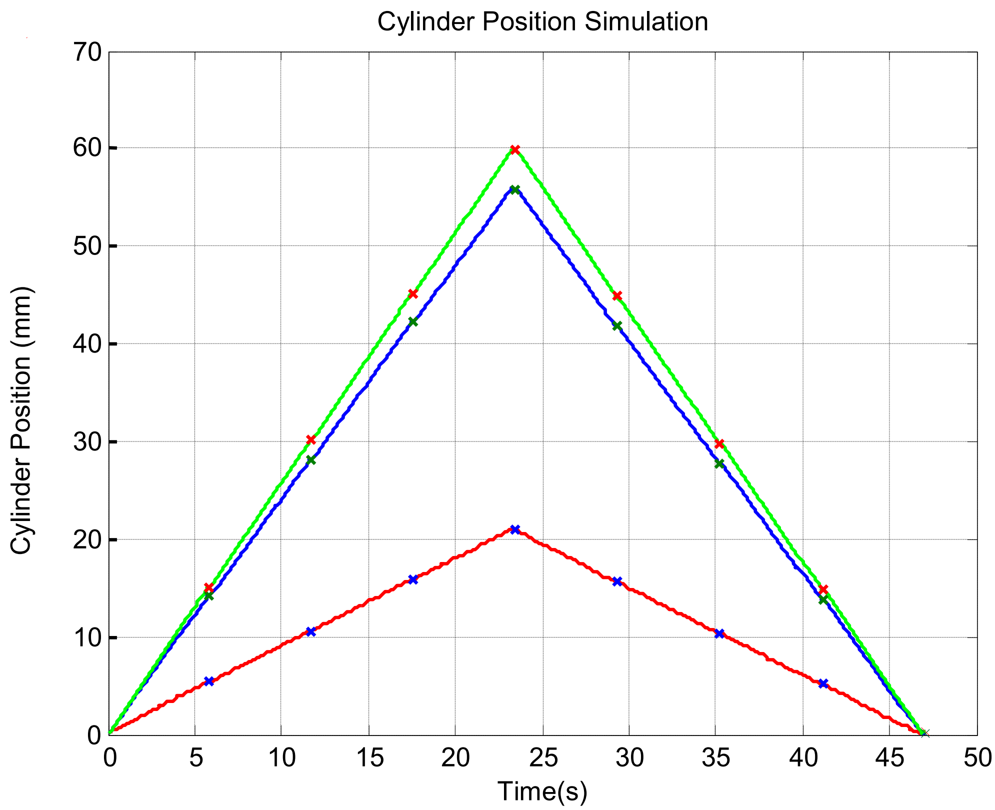


Figure 6-3 Cylinder position plot

In Figure 6-3 we can see that the shape of the graph matches the transverse-time graph specified in Chapter 5. It is interesting to note how the lowest cylinder moves in a bumpier fashion. This is obviously because this cylinder's valve has to be in the off position more regularly to slow it down. The small crosses are the positions where the cylinders have stopped so load cell readings and other data can be gathered.

With these graphs the following scenarios can be analysed.

6.3.1 Effect of resolution on lift time and accuracy

The accuracy we are concerned about is how accurately the cylinders stop at the intermediate heights. The tolerance on variation from the desired height is set at 0.5mm and hopefully by achieving this accuracy, we can achieve better than 1mm on the real system. For the comparison, valve response time is set constant at 20ms, which is about standard for directional valves.

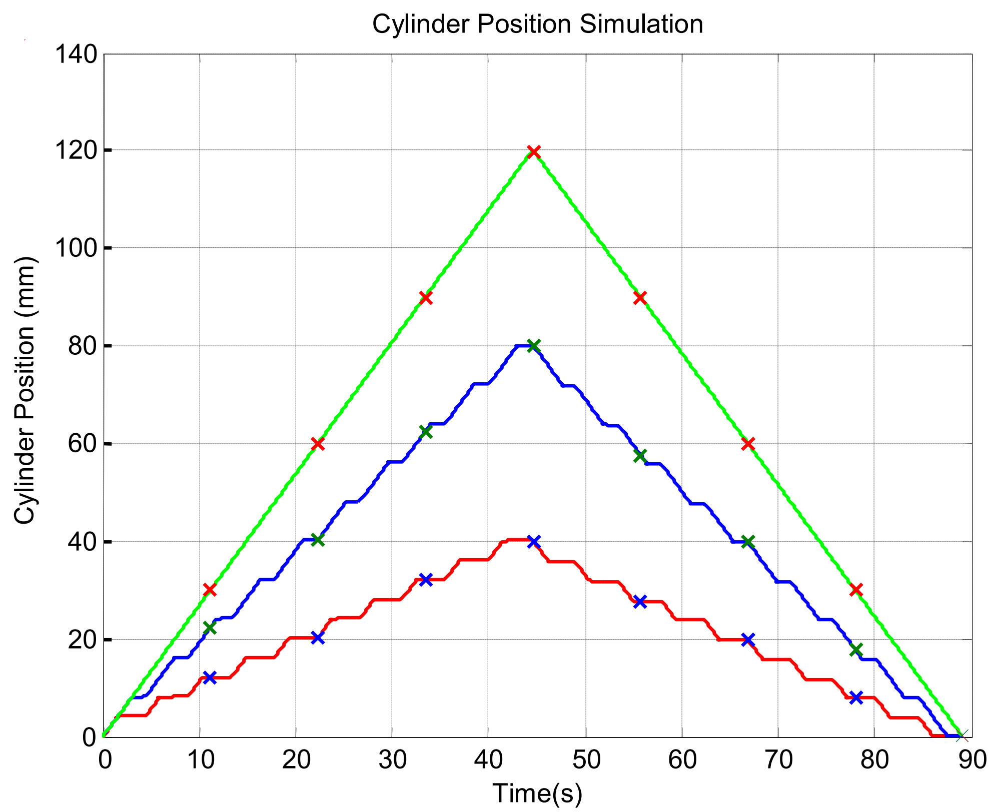


Figure 6-4 Resolution: 10

At this extremely low resolution the maximum error occurring at any intermediate height was 2.57mm, an unacceptable error. The 10 steps can clearly be seen on two of the cylinders while the third remains on continuously. The error of intermediate heights is not the only concern here. It can be seen in some places that the lowest cylinder reaches its intermediate height approximately 8mm before the highest cylinder. This will have an effect on the load measured because of the hysteresis effect discussed in Chapter 2.

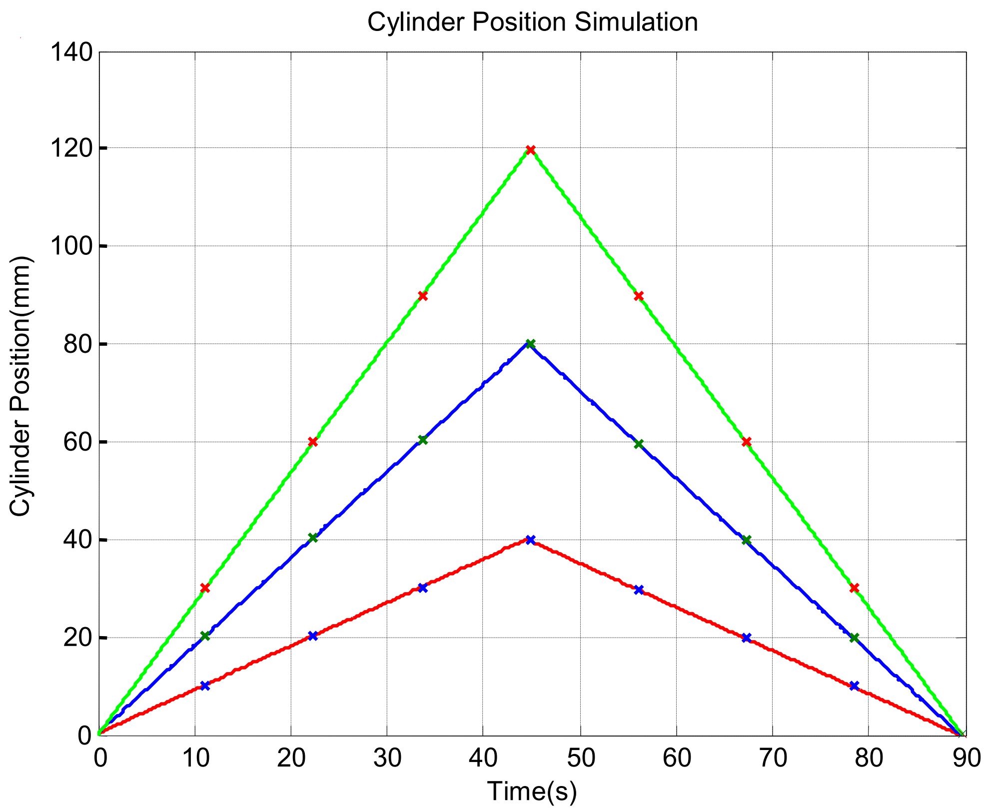


Figure 6-5 Resolution: 120

It might be expected that a resolution of 120 synchrony targets, for a stroke of 120mm will give an accuracy of approximately 1mm. However this is not the case. The intermediate targets are calculated separate from the resolution. The resolution only serves to keep the cylinders in synch. The maximum error for this particular scenario occurred at 75% on the downstroke on the middle cylinder and was 0.263mm.

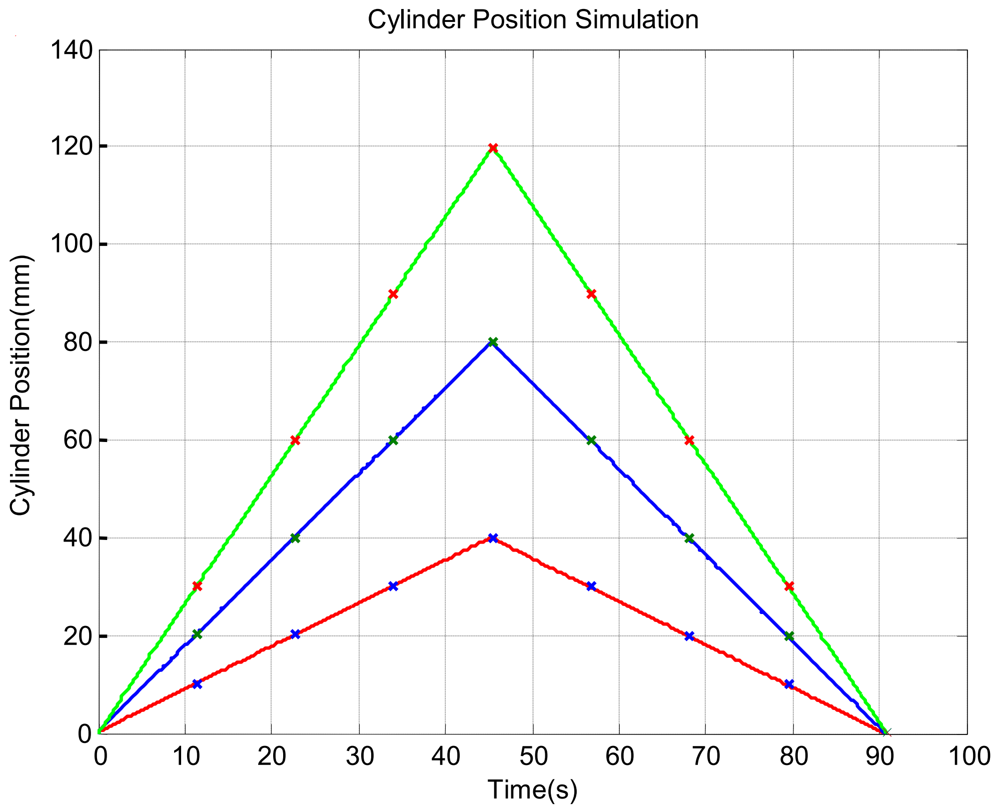


Figure 6-6 Resolution: 240

At twice the resolution there is a small increase in accuracy with the maximum error being 0.21mm. The better synchrony at high resolution means there is less time for cylinders to move because of LVDT inaccuracies while they are waiting for the other cylinders to catch up. This is probably an effect only seen in the simulation and accuracy is not expected to improve this much in the real system. Time for the complete lift has increased by a few seconds.

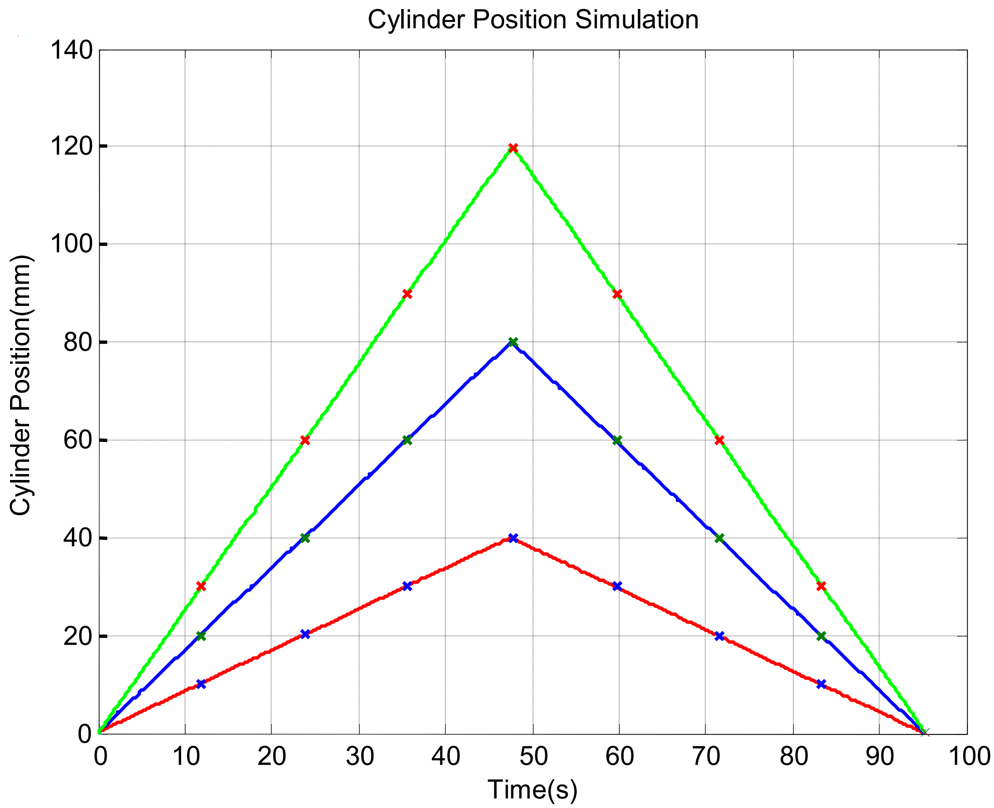


Figure 6-7 Resolution 480

At even higher resolution we achieve a maximum error of 0.12mm. However lifting time increases by about 5 seconds.

The comparison has shown that resolution does not affect accuracy greatly except if it is extremely low. As resolution is increased the time taken for the lift only increases marginally. It is recommended that a resolution of twice the largest stroke be used.

6.3.2 Effect of valve response time on accuracy

If we zoom in on the transverse-time graphs that we have been analysing we can see the response of the valve. For this comparison the number of simulation steps is increased so that the response curve is smooth. Below, in Figure 6-8, is the response of a cylinder turning off after it reaches the height of 30mm and then turning on again once a key has been pressed. In this simulation the delay while reading the value into the PLC, processing and sending the signal to the solenoid is ignored.

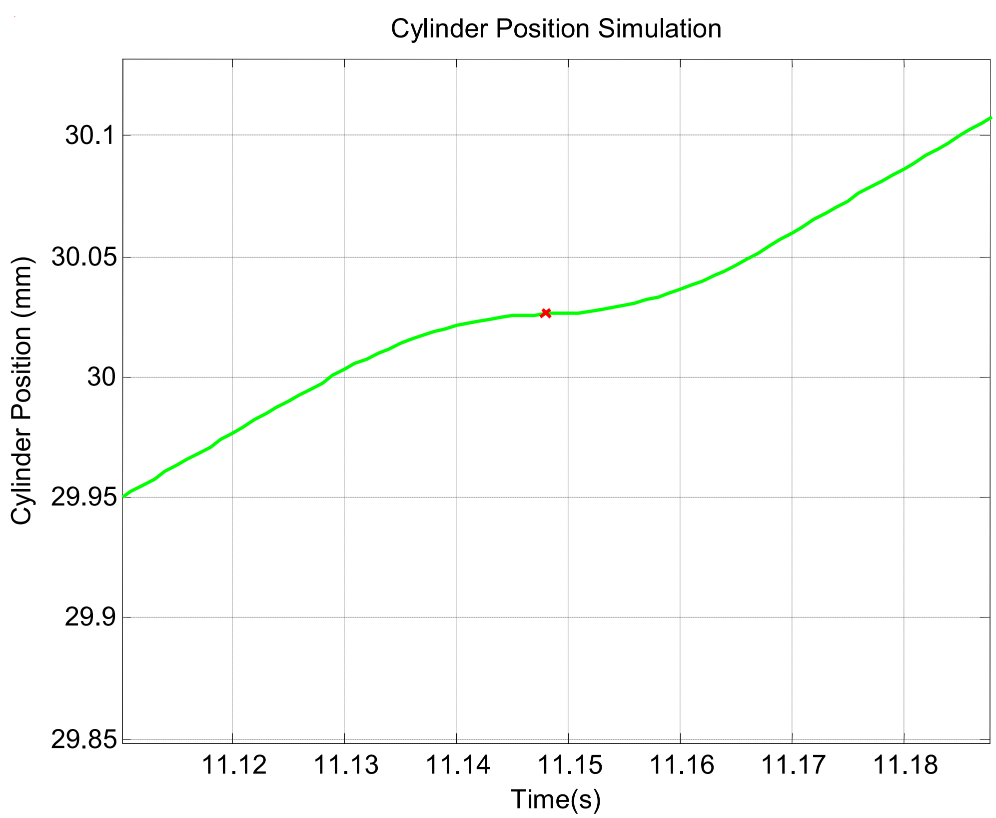


Figure 6-8 Typical valve response curve

For the purpose of this comparison the resolution is set constant at 240.

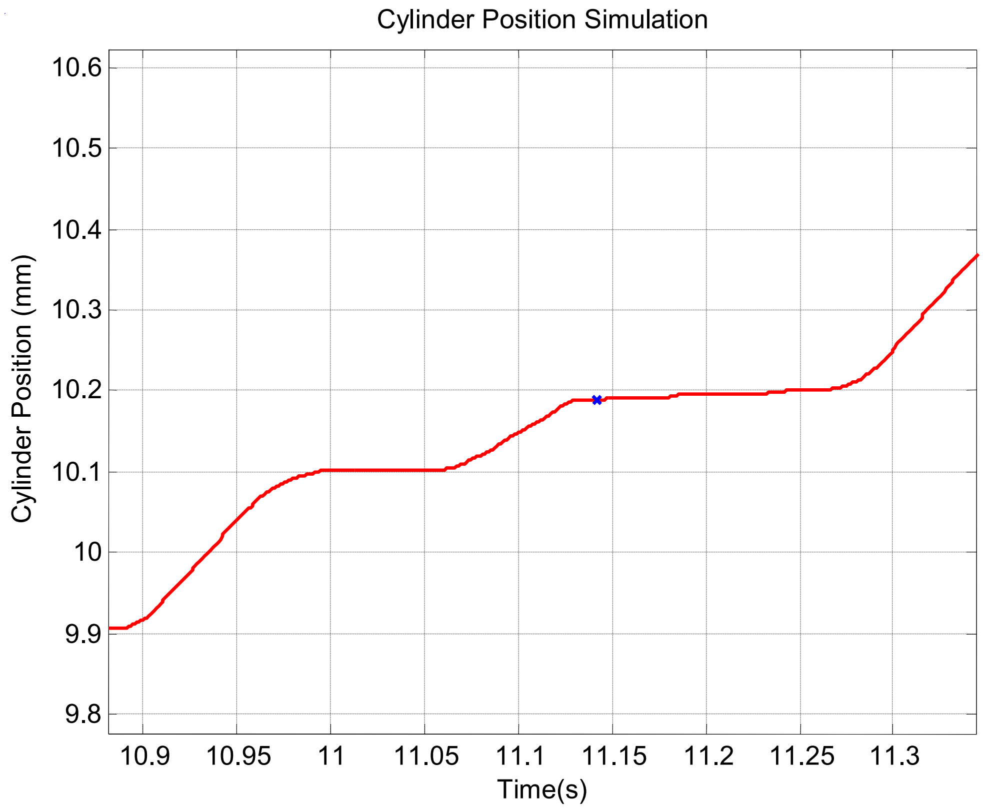


Figure 6-9 Response time 20ms

In Figure 6-9 the target height is 10mm. There is an error of 0.19mm. Intriguingly the cylinder stops, then starts to move again, even after it has travelled past 10mm. Again this is probably just an abnormality of the simulation. The valve is staying open because of the simulated errors in the feedback, whereas it should be closing and producing a smooth curve as shown in Figure 6-8.

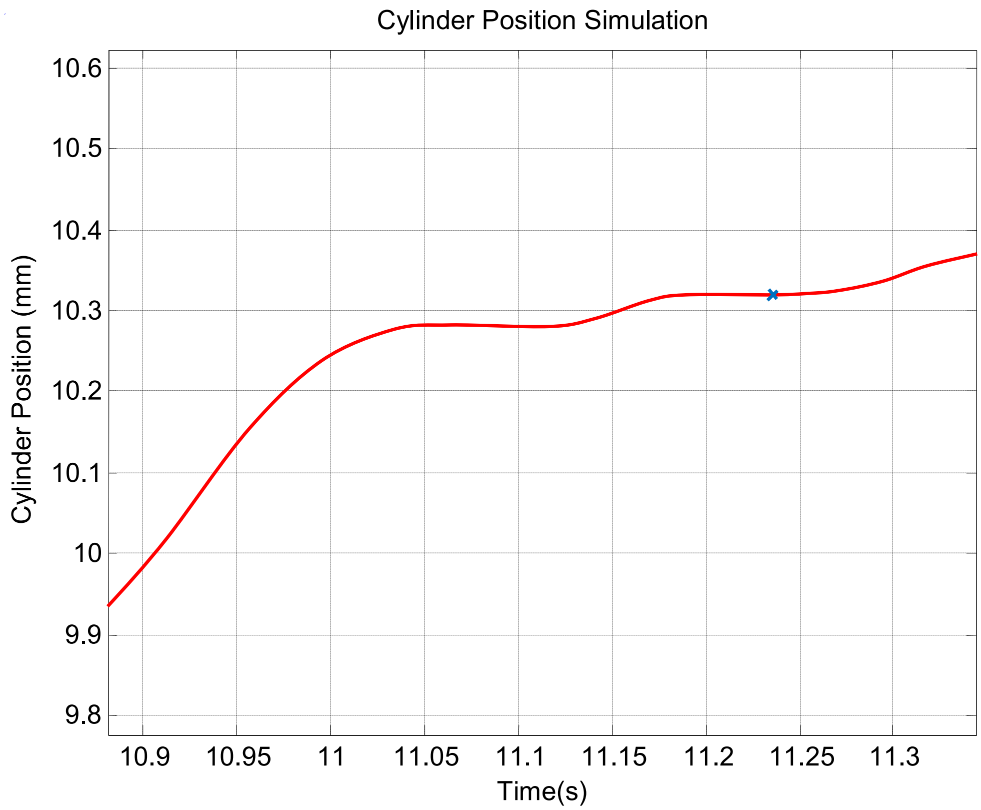


Figure 6-10 Response time 50ms

For the response time of 50ms shown in Figure 6-10, the error was 0.132mm, which is smaller than the error of the previous response time. However the maximum error over the whole lift was 0.29mm whereas the maximum for the 20ms valve was 0.22mm.

At 100ms the maximum error was 0.4mm. This is getting close to our tolerance of 0.5 mm and therefore we can conclude that if the real system exhibits these strange valve control behaviours then the valve response needs to be at least 100ms.

As expected valve response time had very little effect on the overall lifting time.

6.4 Effect of Step Size on Solutions

In this simulation increasing the step size gives a better approximation of the real system due to the response curves being smoother. It also has the effect of increasing the frequency of feedback and the aim of this comparison is to estimate the minimum frequency the feedback loop should operate at in the real system.

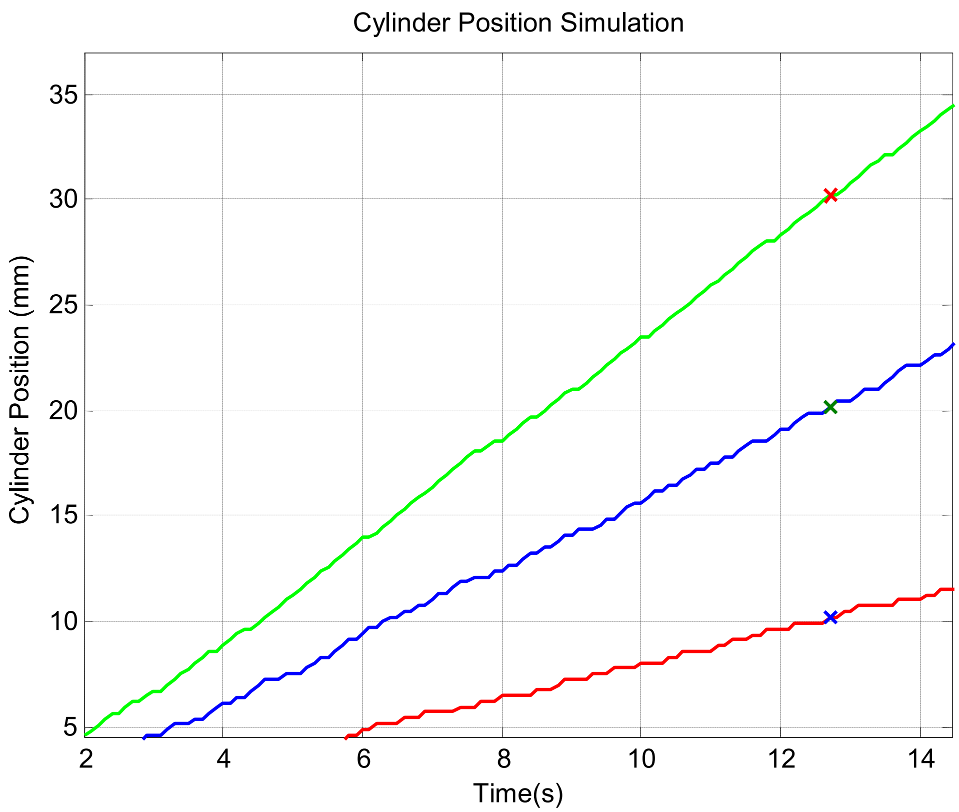


Figure 6-11 Time step: 0.1 seconds

The interesting result about Figure 6-11 is how the highest cylinder is stationary quite a few times throughout the stroke, shown by the bumpy top line. This is slowing down the lift and it has taken more than a second longer to reach the first intermediate height. So, being somewhat unintuitive, the lift time is actually increased when an exceptionally slow feedback frequency is used. The maximum error measured at the intermediate heights was 0.55mm.

When the step size was decreased to 0.01 seconds the error was reduced to 0.21mm.

As the step size was further reduced to 0.001 seconds the error was 0.38mm. It is difficult to explain why the error has actually increased and this may reflect inaccuracy of the simulation. However it means that the solution has converged and that decreasing the time step further will not be beneficial.

In conclusion the best performance was observed when feedback cycle time was 0.01 seconds, or 100 Hz. PLC's can operate at frequencies much higher that this. It should be recommended to alter the cycle time if the high cylinder appears to be stopping at all.

6.5 Problems with the Model

The model was a useful tool in analysing the concept of digital hydraulics. However there were quite a few problems and it did not seem to match the performance of a real system in some aspects.

One issue was trying to get the valve of the highest cylinder to be open all the time. This is the ideal situation where flow is not restricted at all and the lift will be completed in the shortest time possible. Before the code was improved the simulation was outputting graphs like the one in Figure 6-12.

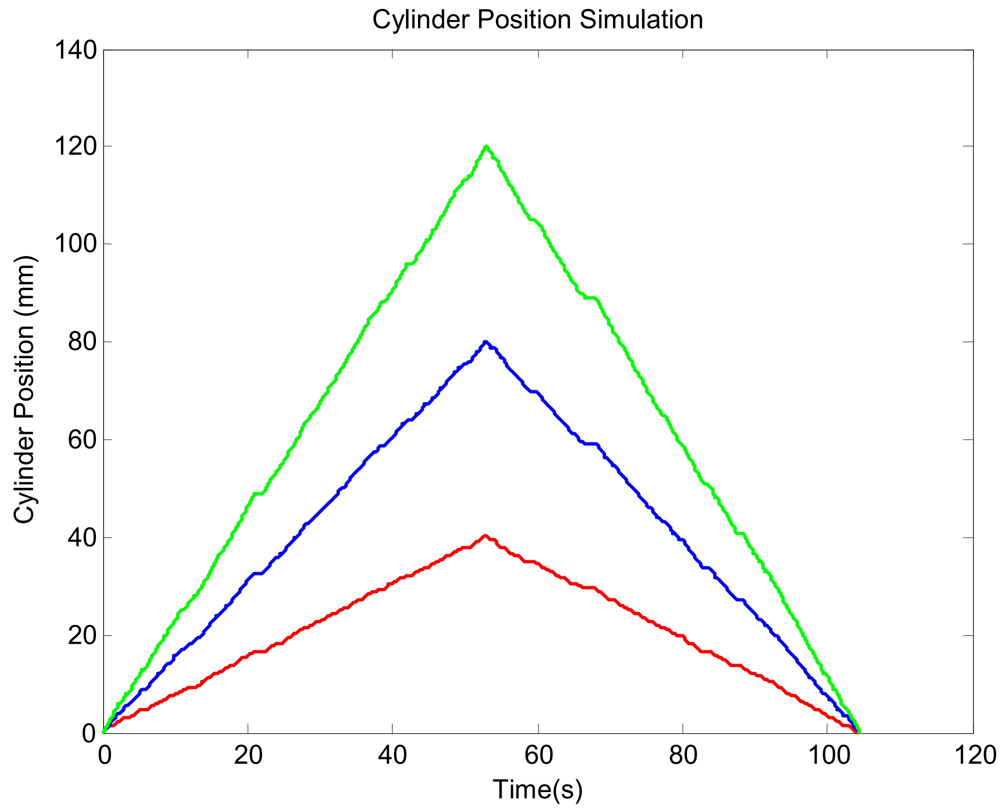


Figure 6-12 Inefficient code example

This example shows that the highest cylinder is stopping at numerous times along its stroke. The inefficiency was in the sequence of the code execution. Initially the code was written to check all valves were stopped before moving to the next synchrony target. Obviously this entails that all the cylinders will be stopping frequently. The code was then improved to check that all cylinders had reached or exceeded their synchrony targets before continuing to the next target. The next target was specified immediately after this check, allowing the high cylinder to always be moving towards the next target, with no stoppages.

6.6 Real Program Considerations

The simulation program developed for this analysis can be used as a model for the real program in the PLC. However there are many considerations in the generation of complete working code for the PLC. Much of the simulation code can be left out as it is used to generate values that are already part of a real system, such as the error of LVDT readings. Also the simulation code is quite simple and better code for stopping the cylinder accurately would be desired.

6.6.1 Implementing more cylinders

The simulation code is set to use three cylinders. A twist test may require five. The simulation code could easily be modified to simulate five cylinders. It would only need a few more 'if' statements for each cylinder. An input from the user would be required to ask, 'How many cylinders?', and the appropriate code then run.

6.6.2 Using more intermediate steps

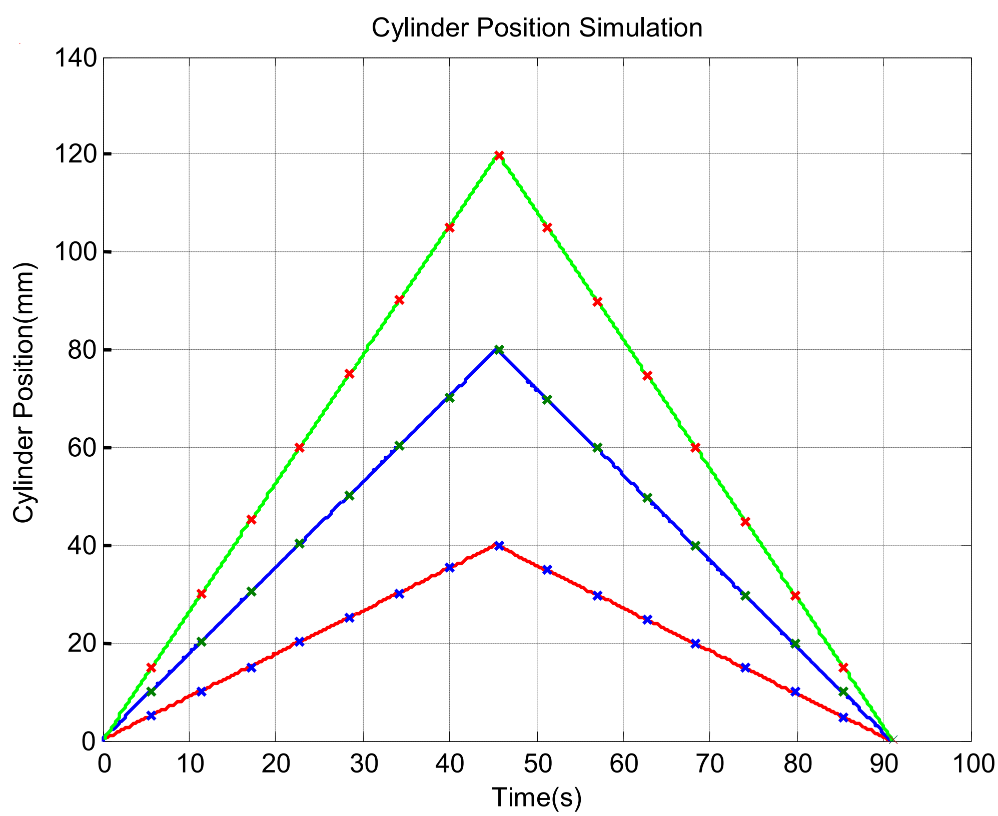


Figure 6-13 Simulation with 8 intermediate steps

The simulation code already handles any number of intermediate steps. Again a simple question should be put to the user in the real code; 'How many intermediate steps?'

6.6.3 Calculating the stroke heights

The stroke heights should be calculated from the bogie wheelbase, bogie centre distance, cant ramp slope and dip depth. The equations for these heights can be seen in Chapter 2 and these can easily be written into the real code.

6.6.4 Configuration code

There will be much more to the actual PLC program than the code to control the cylinders throughout the lift. Some features that will need to be built in are:

- Emergency stop buttons
- A routine to zero the position transducers once they have been placed in position
- A calibrating routine which will extend the cylinder to a certain height to check the system accuracy before lifting.

6.6.5 Difference in programming language

There is considerable difference between Matlab code and PLC code. However the coding concepts used in the simulation should transfer nicely to the real PLC code. PLC's are coded using ladder logic. Ladder logic is a graphical method of representing electrical logic schemes and PLC's are commonly programmed using this graphical interface from a PC (Wikipedia). The simulation code is based on *if-else* statements which transfer to the *and, not and or* statements which are used in ladder logic. Manufacturers also can supply software which will transform C type code like Matlab into equivalent PLC code.

6.7 Conclusion

The concept of digital hydraulics has been analysed and it has passed with flying colours. With the right code sequence and step time there is zero flow restriction of the highest cylinder. Accuracy was less than 0.3mm using the optimum parameters. This is entirely acceptable as an error of 0.5mm was hoped for in the simulation so that we create a factor of safety of two for the real system accuracy of 1mm.

The code to actually apply the concept of digital hydraulics was very simple and could be written in a few lines of code. The real code programmed into the PLC should be quite simple to create.

The next step in evaluating the digital hydraulic system is to check that all the components of a hydraulic system can meet performance specifications and to design a hydraulic diagram that will allow for the control of cylinders in a manner similar to this simulation.

7 Hydraulic system design

7.1 Introduction

The next step in determining the feasibility of a digital hydraulic system is to look at the individual components. Equipment will be selected that will fulfil all performance requirements. The equipment selected may not necessarily be the cheapest brand or model but will prove that the overall system is feasible. As mentioned in Chapter 1 the maximum budget is $20 000 and the cost of components will be established and the overall system cost checked against this budget.

7.2 System Pressure

It is important to first establish what maximum pressure the system should be able to handle. A point of interest is that pumps create flow, not pressure (Rohner 1995). It is only when there is a resistance to this flow that pressure is created. Obviously the system has to be designed to handle the maximum pressure which will be created when a heavy locomotive has to be lifted.

The maximum force a cylinder can apply is governed by the maximum system pressure and the cylinder effective area. The effective area is the area available for the oil to act against. For a double acting cylinder this means the rod area has to be subtracted from the bore area to calculate area in the secondary direction.

Here is a comparison of cylinder sizes to lift 200 kN, the force required to lift a locomotive, using the common pressure of 200 bar compared to high pressure of 700 bar.

<u>200 bar</u>

$$\text{Effective cylinder area} = \frac{\text{Force}}{\text{Pressure}} = \frac{200\ 000\ \text{N}}{20\ \text{MPa}} = 10000\text{mm}^2$$

$$\text{Cylinder Diameter} = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 10000}{\pi}} = 113\text{mm}$$

<u>700 bar</u>

$$\text{Effective cylinder area} = \frac{\text{Force}}{\text{Pressure}} = \frac{200\ 000\ \text{N}}{70\ \text{MPa}} = 2857\text{mm}^2$$

$$\text{Cylinder Diameter} = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \times 2857}{\pi}} = 60\text{mm}$$

Using high pressure means the cylinders will only need to be approximately half the diameter, making them more portable. Looking at some general figures in product catalogues this could mean an approximate weight difference of 10kg compared to 20kg.

Of course this weight saving is not the all important consideration when comparing systems pressures. High pressure pumps are likely to be much more expensive because of the mechanism used; they are usually piston pumps compared to the cheaper gear pumps used in 200 bar systems.

Another consideration is availability. While there might be more pumps with a limit of 200 bar on the market overall, there is a portable pump niche in the market that seems to be standardised on 700 bar. These pumps are supplied by companies such as Enerpac, Simplex and SGS Engineering. Using a portable pump already set up with power supply, electric motor and oil reservoir will save huge amounts of needless design. This is reason enough alone to specify that the system will be designed with a maximum pressure of 700 bar.

7.3 Cylinder Selection

The first consideration is to specify whether the cylinders should be double or single acting. Single acting means there is only one hydraulic connection and force can only be applied in one direction. For the twist test system this is fine and only a single acting cylinder is needed. When pressure is released the weight of the train will force the cylinder closed again.

Single acting cylinders also have the option of including a spring return – an internal spring to force the cylinder closed when pressure is released but no load is applied. This may be useful to counteract any rogue pressure that may build up in the return line. If there is any pressure at all in the return line it will be difficult to slide the cylinders out from underneath the axlebox because it will be still forced against the load.

The other parameters that need to be examined for cylinder selection are collapsed height, stroke and load capacity. We have already specified the system pressure so load capacity can be calculated from the bore diameter. These parameters were looked at in an Enerpac catalogue and other catalogues online. It was more difficult than expected to find a match. There was the same difficulty of achieving a stroke of 115mm with a collapsed height of 200mm as with the power screws. An example cylinder is the Enerpac RC-254. It is a single acting spring return cylinder having a stroke of 102mm and a collapsed height of 215mm. This meets neither of the dimension constraints, especially when the interface between the cylinder and axlebox is added to the collapsed height. The question arises of which of these two dimensions is more important.

All twist tests previously completed have not required a stroke above 96mm. The average stroke from the twist test data that was examined was approximately 60 to 70mm. Therefore a stroke of 80mm may be acceptable. However a stroke of 50mm (this appears a standard stroke size in the catalogues) will never be acceptable because practically every test will then require lifting in stages. If exceptionally large strokes are required again in the future it is not a major concern to lift in stages occasionally. In fact SGS Engineering supply purpose built cribbing blocks for their RSS 'Shorty' cylinder series.

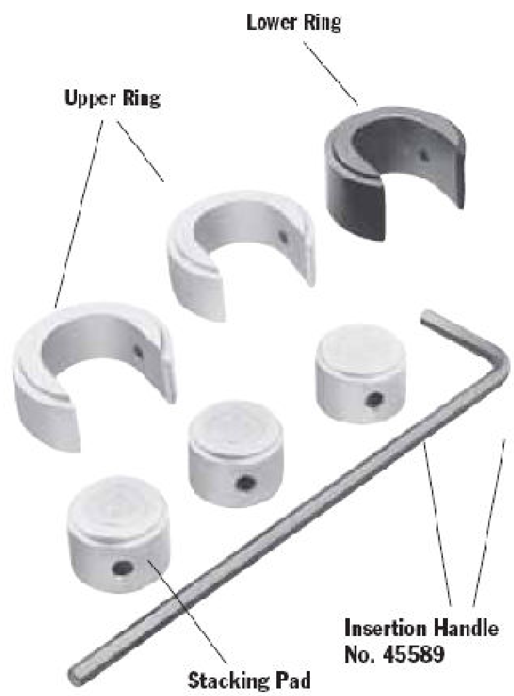


Figure 7-1 Specialised cribbing blocks (SGS 2005)

These allow for quick and safe lifting in stages by inserting an outer ring, releasing a cylinder, inserting a stacking pad, then re-lifting. These would be a wise investment.

If the collapsed height is not small enough for certain rollingstock the consequences are more serious. It means that the rollingstock can only be tested in a workshop with raised rails and it could be difficult to schedule time at these workshops. Therefore a cylinder which meets the collapsed height constraints will be chosen.

Two sets of cylinders could be purchased to cover all possible strokes and clearance requirements. However this will push up the price of the overall system considerably and will be avoided.

An Enerpac CLS-504 cylinder will be selected and it has a stroke of 100mm and a collapsed height of 178mm. It is rated to lift 50 tons giving a safety factor of more than 2 for load capacity, but this is the lightest cylinder in this model range. It is a specialised model with reduced collapsed height for heavy lifting applications. It achieves the reduced collapsed height partly by forgoing a spring return mechanism. As mention above this may present issues in that the cylinders may still be wedged underneath the load when they have been fully retracted. Of course the system can be turned off completely and pressure bled to ensure that the cylinders can be removed. The CLS-504 weighs 18kg, just suitable for one person to lift. RTP price was given as $1535 each. It is worth noting here for interest's sake that equivalent power screws would have cost more than $4000 each.

7.4 Cylinder – Axlebox Interface

The interface between cylinder and axlebox has to be designed so as not to damage the axlebox by marking the surface. The other function of this interface is to provide a larger surface for the axlebox to rest on and prevent it slipping off the cylinder. Currently, with the manual test method there is a wood block inserted as shown in Figure 7-2.

Figure 7-2 Wood block used as cylinder - axlebox interface

The problem with this design is that it takes up too much of the collapsed height and it does not allow for the 5° that the rollingstock will tilt if it is lifted to the maximum stroke.

The tilt can easily be accounted for by purchasing a tilt saddle with the Enerpac cylinder. A tilt saddle fits to the top of the plunger and allows tilting up to 5° in any direction, as seen in the conceptual drawing in Figure 7-3. It will add another 24mm to the collapsed height.

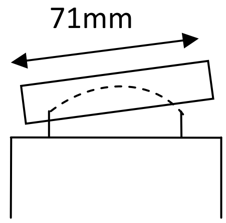


Figure 7-3 Tilt saddle

The saddle for this cylinder is already 71mm in diameter, which should be wide enough to be safe from slipping. To guard against marking the axlebox surface a high strength plastic can simply be placed over the top of the saddle as shown in Figure 7-4.

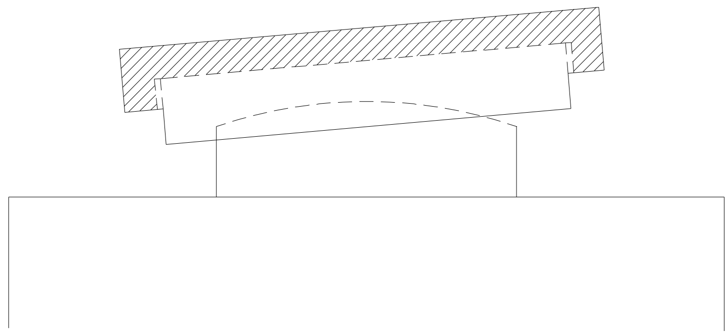


Figure 7-4 Polymer cap shown shaded

The cap could be purchased from Modern Engineering, a common supplier of bearing composites to QR, for less than $50 each. Because of its high strength the composite should only need to be 5mm thick. This means the overall collapsed height will be 207mm.

7.5 Pump Selection

As mentioned in Section 7.2 the pump selected will be a portable version with a maximum pressure of 700 bar. The criteria governing the selection of a pump is that it has to be driven by 240VAC and has to provide enough flow to lift the cylinders 105mm in less than a minute. A simple calculation will tell us how much power this requires.

Now, if the highest cylinder has to be lifted 105mm, the average cylinder extension over all 5 will be approximately half that – 55mm. The 3700 class locomotive will exert an average force of 105 kN per wheel. The power required is then:

$$Power = 5 \times Force \times \frac{Stroke}{time} = 5 \times 105000 \times \frac{0.055}{60} \approx 480 \text{ Watts}$$

Assuming a typical efficiency of 50% for the combined electric motor and hydraulic pump, we are looking for a pump rated at better than 1 kW. One such pump is the Enerpac ZU4 rated at 1.25 kW. This is a portable hydraulic piston pump, driven by an electric motor which requires only single phase electrical power.

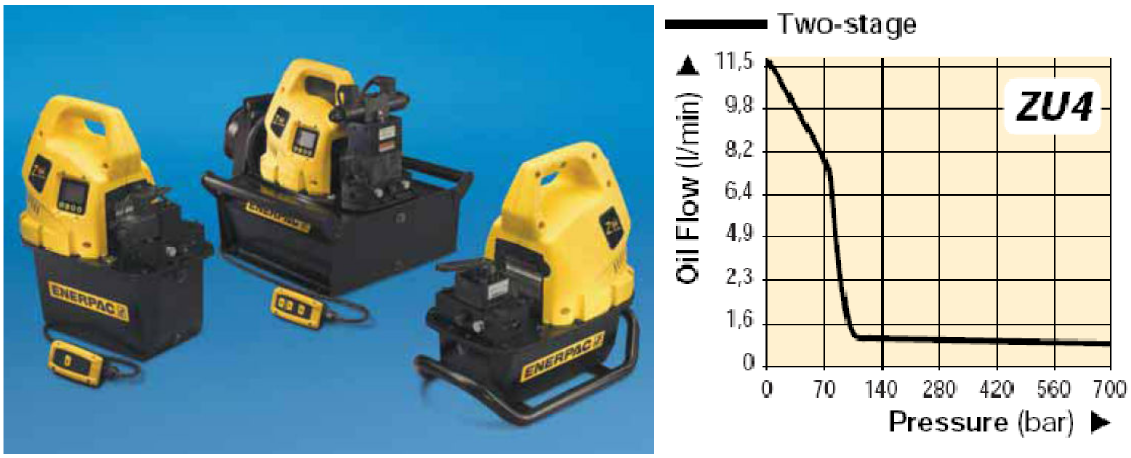


Figure 7-5 ZU4 models and performance chart (Enerpac product catalogue)

The speed this pump will operate the cylinders at will now be checked. The CLS-504 cylinders are rated for much larger loads than 105kN, meaning they will be operating at pressures much less than 700 bar. The effective area of these cylinders is 71cm^2 so they will be operating at this pressure:

$$P = \frac{F}{A} = \frac{105000}{0.0071} = 14.7 \text{ MPa} = 147 \text{ bar}$$

An examination of the ZU-4's performance chart reveals that the pump can output approximately 1.3 litres/min at 147 bar. This translates to a velocity of:

$$V = \frac{Q}{A} = \frac{1\text{l/min}}{71\text{cm}^2} = \frac{1.66 \times 10^{-5}\text{m}^3/\text{s}}{0.0071\text{m}^2} = 2.3\text{mm/s}$$

Therefore the cylinders theoretically will traverse 105mm in 45 seconds. Of course there will be many losses in the system due to restrictions from valves and friction losses in the hydraulic hose. However these extra losses will be counteracted by the fact that most rollingstock to be tested will not be this heavy and at reduced pressures the flow output will be greater. In fact, the ZU-4 is a two stage pump, meaning flow increases dramatically at low pressures. A 20 ton wagon will allow the pump to output flow at approximately 7 l/min. A ZU-4 pump will cost $4770.

7.6 Reservoir Size

The ZU-4 is supplied with oil reservoirs ranging in size from 4 to 40 litres. We can calculate the amount of oil required be adding the maximum oil capacities of the cylinders. Oil capacity of a cylinder is found by multiplying the stroke by the effective area.

$$\text{Total oil capacity} = 5 \times \text{A} \times \text{Stroke} = 5 \times 71\text{cm}^2 \times 10\text{cm} = 3550\text{cm}^3$$
$$= 3.55 \text{ litres}$$

The oil capacity of the hoses is actually more than that of a cylinder over distances of 15m. However the hoses can be filled first and then the reservoir topped up. An 8 litre reservoir will be selected as this will provide enough oil so that it does not have to be topped up often.

7.7 Hose Selection

Judging by a list of previous twist tests, most tests will be on rollingstock with lengths of approximately 10 to 15m. Therefore, for a 6 axle vehicle, three 15m hoses and two 6m hoses will be selected. The diameter is important for this system, not because of the losses, but because of the back pressure that hoses will create when cylinders are retracting. The cylinders selected have no spring return, which makes back pressure more critical. Therefore the larger of Enerpac's two choices of hose diameter will be used. These hoses have an internal diameter of 9.7mm. Exact cost of these hoses was not confirmed but scaling up from prices obtained for shorter hoses, we can guess that cost would be:

3 x 15m hose @ $600 each = $1800

2 x 6m hose @ $300 each = $600

Total cost of hose = $2400

For exceptionally long vehicles such as the Stoneblower, more hose will have to be purchased.

7.8 Hydraulic System Schematic

The layout of the hydraulic system can now be designed and appropriate valves selected. There are many points to consider for a successful system, such as incorporating inherent safety. The paramount safety issue is the risk of cylinders losing pressure and causing the load to drop rapidly, which is likely to cause extensive damage to the cylinders, and more importantly, the rollingstock.

The hydraulic schematic can be viewed in Appendix A.

7.8.1 Valves

The directional valves used to control the single acting cylinder are specified as 3-way, 3-position, spring centred, closed centre, solenoid valves, with pilot operated check valves. For reference their diagram is shown below in Figure 7-6.

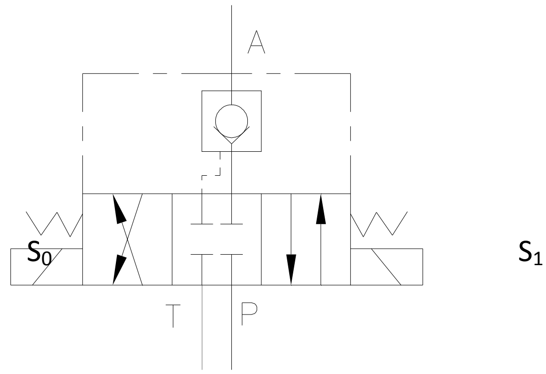


Figure 7-6 Directional control valve required

There are three positions because we need one each for the functions; extend, hold and retract. Solenoid S_0 will give the command to retract the cylinder while solenoid S_1 will give the command to extend. If no solenoid is actuated the springs will centre the valve and the cylinder will be held stationary.

The piloted operated check valve can be inbuilt into the directional valve. It is shown as the extension on the top the valve in Figure 7-6 and it serves to provide secure load holding capability. Check valves give a better seal than the spool within the directional control valve because they use oil pressure to seal them selves off. When the valve is in the centre position the check valve seals itself. When the valve is in the retract position, pressure is routed to release the check valve and allow flow back to the tank.

Usually valves have a tandem centre where the oil under pressure is routed back to the tank when the directional valve is in the centre position. However for this system this will cause a problem in that pressure will escape out one valve when the other cylinders still need it to extend. Therefore a closed centre is need, but this will mean that a pressure relief valve must be installed so as not to cause excess pressure build up. A relief valve will allow oil to pass back to the tank when it exceeds a certain pressure.

The other valve needed is a counterbalance valve, sometimes called a back pressure valve. This is depicted on the schematic as the valve just before the cylinder. The counterbalance valve controls the pressure on the retract stroke so that flow is proportional to the pressure, ensuring that the cylinder never ‘free falls’. This valve is essential for safety reasons but it will increase the chance of the cylinder retaining pressure on the retract stroke and getting jammed under the axlebox. If this system does not operate successfully a spring return cylinder may be needed.

Enerpac was asked for a quote on the directional control valve and a price of $4500 each was given. This is incredibly expensive and would exceed the budget. Therefore cheaper versions were sought and Hystar valves can be purchased for approximately $500 each. These valves are only rated at 315 bar but the relief valve can be set below this because the cylinders will lift the maximum load at 147 bar as calculated in Section 7.5. Counterbalance valve prices are unknown so we will assume they are $500 as well. A manifold will be used to mount the valves and one will cost $500. Miscellaneous fittings and connectors could cost a further $200.

This hydraulic system is not proven to work and a prototype system should be tested first to make sure it operates successfully.

7.9 Position Transducer Design

The first thing to be considered is the actual locating of the transducer. Optimally we want it to be located as close as possible to the train wheel as this will give an accurate displacement of the wheel.

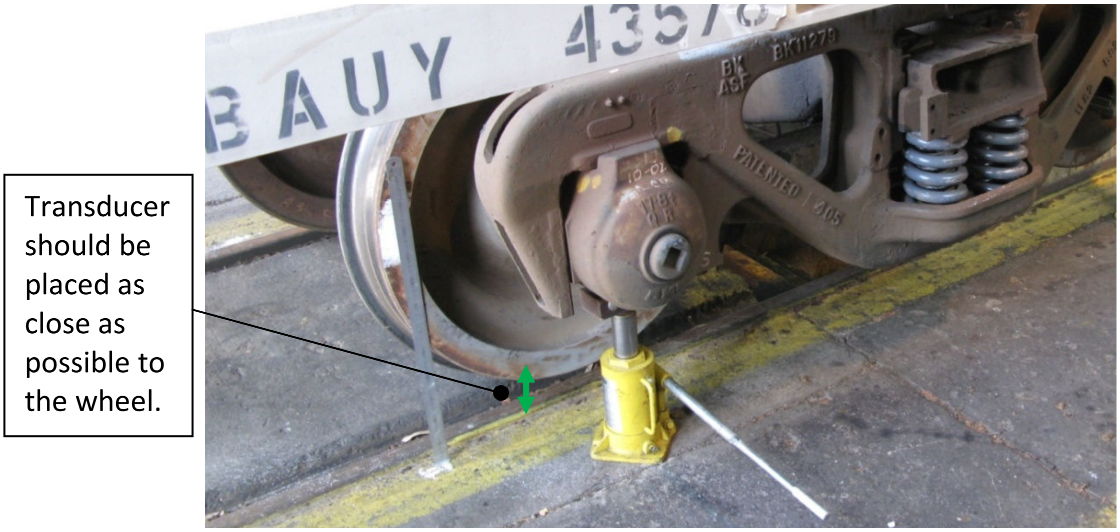


Figure 7-7 Locating the transducer

The other consideration is that in a closed-loop system the accuracy of the feedback is the more critical when compared to the controlling mechanisms (the valves and software code) (Wen et al. 2007).

Although we have already selected a cylinder model it is interesting to note that there are hydraulic cylinders available with displacement transducers integrated, from companies such as Hydro-Line. However, housing the electronics means that the collapsed height of these cylinder are much larger than standard.

There are a few options for position transducers, which vary considerably in their underlying concept. There are two possibilities mechanically; the transducer can operate in a rotary or linear fashion. A rotary encoder would need a string and spring return system, similar to a self retracting tape measure. The actual transformation from physical movement to an electrical signal can be performed by a myriad of devices, such as: encoders, potentiometers and Linear Variable Differential Transformers (LVDT's).

The option selected for this feasibility study is a draw-wire potentiometer, which uses the rotary spring mechanism just mentioned. This type of position transducer may not be quite as accurate as an LVDT but it is much easier to locate it near the wheel and it still produces accuracy that should be acceptable. Below is schematic of a probable method of mounting the draw-wire potentiometer:

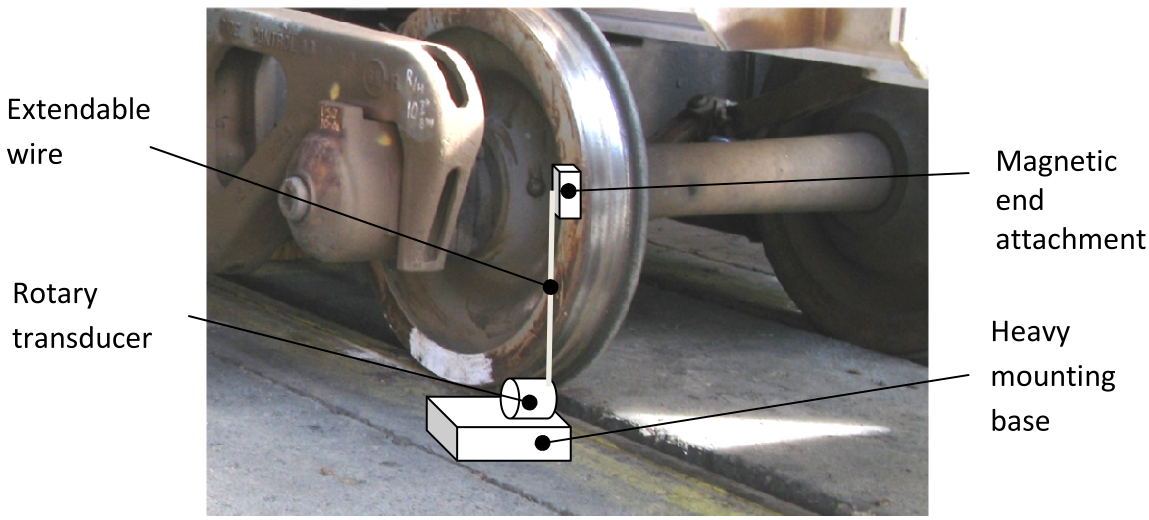


Figure 7-8 Draw-wire potentiometer placement

Issues with this design are that it is susceptible to being bumped out of position and the wheel may rotate and ruin the measurement. However models can be purchased with string tensions of only a few Newtons so the wheel is not likely to shift. This design is advantageous because it will be extremely quick to attach to the wheel, which satisfies the primary goal of the new automated test of a quick setup time.

A particular model that will be suitable is the Micro-Epsilon wireSENSOR WDS-MPM. This model has a non-linearity of 0.2% of full scale output (Micro-Epsilon n.d.). For the 150mm stroke model this will mean an accuracy of 0.3mm, which should keep the accuracy within the 1mm tolerance. This particular brand was more accurate than some others researched. It requires an input of up to 32VDC and outputs a proportion of this voltage. This is relevant to PLC selection. wireSENSOR WDS-MPM’s cost approximately $1800 each.

7.10 PLC Selection

As with the hydraulic components different brands of electronics will not be compared and a suitable model from Koyo's DirectLogic line of PLC's will be selected (Automation Direct 2007). Information we have relative to PLC selection has been gathered over the last two chapters.

- 5 inputs required at 0 up to 32VDC from the draw-wire potentiometer.
- 5 outputs required to operate Hystar solenoid valves. These valves can be purchased in a range of voltages; 100VAC-240VAC or 12-48VDC. From the Hystar documentation AC powered valves will have faster response times.
- Very little memory space or processor speed required, because the code to implement digital hydraulics is simple and high speed scanning is not required for better accuracy.

The second base model, the DL06 PLC, will be selected because it meets all of these requirements. This model has 20 inputs and 16 outputs. In particular the D0-06DR-D base unit should be selected as it has inputs of 12-24VDC which should successfully interface with the draw-wire potentiometer. The outputs are relay type with a voltage of 240VAC at 2 amps, which will operate the Hystar valves at their maximum response time.

The DL06 includes an auxiliary 24VDC power supply with 300 mA. This will be plenty of power to supply the draw-wire potentiometers. The DL06 itself is powered by normal 240AC mains power. It costs less than $300. Electrical signal cables should cost less than $100

7.11 Total Equipment Cost Breakdown

Item	Cost per item	Quantity	Total cost
CLS-504 Cylinder	$1535	5	$7675
Load bearing cap	$50	5	$250
ZU-4 Pump	$4770	1	$4770
Hoses	$2400	1	$2400
Directional Valves	$500	5	$2500
Manifold	$500	1	$500
Miscellaneous fittings	$200	1	$200
DL06 PLC	$300	1	$300
Draw-wire pot	$1800	5	$9000
Electrical cables	$100	1	$100
		Grand total	$27695

This system exceeds the budget. However it is likely that this equipment cost could be reduced considerably. As mentioned at the start of this chapter no price comparisons between product brands were undertaken. An example of reducing the price by selecting another brand is that cheaper 200 bar hose can be obtained for less than $100 for a 15m length. Indeed for price reasons it will probably be beneficial to reduce the overall maximum system pressure to 200 bar and source an appropriate pump. Also only one cylinder with a stroke of 150mm needs to be purchased, the others could be shorter. By doing this it should be possible to reduce the equipment cost below the budget of $20 000.

7.12 Conclusion

This chapter has proven that equipment can be selected that meets all performance requirements of an automated twist test. However, the total cost of the system exceeds the specified budget of $20 000. This cost may be reduced by selecting cheaper brands and reducing the maximum system pressure.

A maximum system pressure of 700 bar was specified because of the wide range of portable pumps operating at this pressure. However after analysis of the costs it is recommended that this pressure be reduced to 200 bar. Over-capable cylinders were selected meaning the maximum load can still be lifted at this reduced pressure.

The hydraulic system still needs to be tested as it is uncertain if the valves will operate entirely successfully. The electrical system also needs to be tested to ensure compatibility between components and to check accuracy.

8 Conclusions and Further Work

8.1 Introduction

This chapter will draw conclusions on the success of the whole project. It will also describe the further work that has to be done before the new twist testing system can enter service. The main project objective of selecting equipment that will perform an automated twist test has been achieved.

8.2 Discussion

The most important conclusion from this project is that it appears that equipment could be selected for the construction of an automated twist test for approximately $20 000, meaning it is successful from QR's perspective. The real cost to QR will be significantly more than this when final design costs are considered.

Another achievement of this project was to apply the concept of digital hydraulics to a new application. This type of hydraulic control was proved to work for the application of the twist test. Accordingly, it is proved that it will also work for applications with similar functions to the twist test. Digital hydraulics, as the term is used in this dissertation, is a relatively new and unheard of concept. However it seems to be growing in popularity, especially in the business of heavy duty synchronous lifting.

8.2.1 Time savings

It is unlikely that the digital hydraulic system defined will save any time compared to the manual jacking method. Setup time will be longer. The draw wire potentiometers will have to be set up and calibrated. Hoses, cables and cylinders will have to be dragged into position and connected. A rough estimate of setup time is around 30 minutes once employees have been trained to use the system. It will also take longer to transfer the equipment to the other side of the train for the second half of the test.

However, once the test has been set up, the actual lifting part of the test will be slightly quicker. The system should be more reliable than manual jacking and therefore restarting the test due to synchrony errors will be avoided, thus saving time.

8.2.2 Accuracy and repeatability benefits

The best feature of the new test will be its accuracy and repeatability. The final PLC control software will be more accurate than the simulation conducted in Chapter 6, and therefore the goal of accuracy better than 0.5mm for the stopping height of cylinders should be achieved. This will be much better than the accuracy achieved with manually jacking while judging height on a steel ruler. This method would struggle to achieve accuracy of better than 1mm because of parallax errors and the difficulty of controlling the jacking perfectly. Better height accuracy will mean that the load data collected will be more reliable.

More accuracy will also be achieved in terms of synchrony. The cylinders will trace their positions very accurately and reach their required heights at almost the exactly the same time. This will be a substantial benefit it terms of the accuracy of hysteresis graphs produced from the test data. Hysteresis has a substantial effect on the load recorded and the better synchrony accuracy will mean load data is more accurate again. Also, hysteresis graphs could be compared between tests with confidence, knowing that each test is exactly the same.

8.3 Further Work and Recommendations

The next action that needs to be taken in the line of further work is an appraisal of the advantages and disadvantages of the system by QR engineers. They will have to weigh the cost and the extra setup time of this new test system, against their judgement on the benefits of better accuracy and repeatability. If they deem it worthwhile the process of preparing a business case for proposal to higher management will begin.

If the new test project proceeds this far, there will be further work to do on the design in these areas:

- Detailed design of draw-wire potentiometer mounting and bearing cap.
- Maintenance considerations
- Safety considerations
- PLC programming
- Selection of most cost effective equipment
- Prototype testing

A prototype could be constructed from a single cylinder and draw-wire potentiometer, a PLC and a pump already in possession of QR. The system could be proved to work with this equipment.

8.4 Summary

The project was successful because a system was specified which will meet performance requirements for a cost of approximately $20 000. It was determined that time savings would not be significant and would probably be negative. However the major advantage of a new system was specified as improved accuracy and comparability of test data. There is much further work to be completed to realize a new automated twist test, however this is dependent on QR's commitment to this project in the future.

The project objectives have been achieved and a feasible concept design for an automated twist test has been developed.

University of Southern Queensland

FACULTY OF ENGINEERING AND SURVEYING

ENG4111/4112 Research Project
PROJECT SPECIFICATION

FOR: *Carson Holzheimer*

TOPIC: *Automation of the Rollingstock Twist Test*

SUPERVISORS: Selvan Pather
Ian Burns, QR Vehicle and Track Engineer

SPONSORSHIP: Queensland Rail, Rollingstock Engineering Division

PROJECT AIM: Design a machine to automate the Rollingstock Twist Test.

PROGRAMME: (**Issue B, 10-9-07**)

1. Identify outcomes QR expects from an automated twist test.
2. Research applicable regulations.
3. Develop clear specifications to which equipment has to perform.
4. Research existing systems that may satisfy requirements.
5. Research and select possible control schemes and software control equipment.
6. Research and select appropriate lifting equipment.
7. Simulate or build model system.
8. Analyse performance of the specified system.
9. Calculate cost of entire system and compare with budget.

AGREED ______________________(student) ______________________(supervisor)

Date: / / 2007 Date: / / 2007

Co-examiner:______________________

Appendix B – Digital Hydraulics Simulation Code

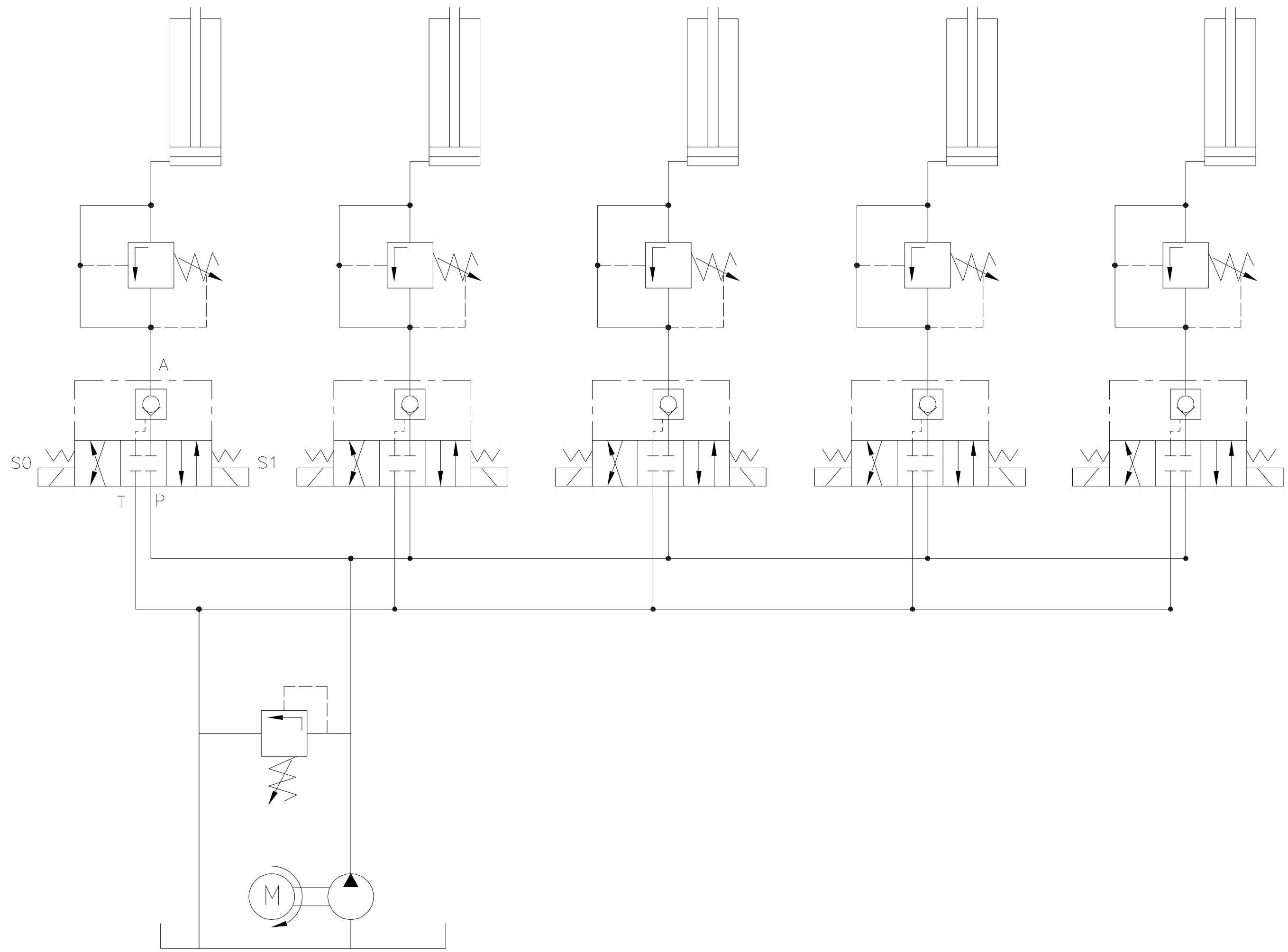
```
% Calibration code need here
clc; clear all;


%stroke(1)=40;stroke(2)=80;stroke(3)=120;
stroke = [21 56 60]
intsteps=4;                                   % the amount of
intermediate heights to stop and read loads
strokestop=stroke/intsteps;
n=240;                                      % the resolution of
synchrony target strokes
cpos(1)=0;cpos(2)=0;cpos(3)=0;
dt=0.01;
starget=[0 0 0];
currentstep= 0;
downstroke=0;
counter=0;
stopping=0;
valvestate=[0 0 0];
valvestroketime=[0.05 0.05 0.05];       % in seconds
valvemove=1./(valvestroketime./dt);
x=[1675 10362 12037];

p = plot(x,cpos,'x','EraseMode','xor','MarkerSize',8);
axis([0 13000 0 70])
set(gca,'YGrid','on')
hold off
xlabel('Horizontal cylinder position (mm)')
ylabel('Cylinder Stroke (mm)')
title('Cylinder position animation')


for t = 0:dt:120;
    % Velocity variance simulated in this line
    vel=[2.3+normrnd(0,0.1),2.3-
normrnd(0,0.2),2.3+normrnd(0,0.3)] ;
    % position transducer inaccuracies are simulated in this
line
    lvdterr=[normrnd(0,0.1),normrnd(0,0.1),normrnd(0,0.1)];
    % Check if all cylinders are at their synchrony targets.

    if  cpos(1)  +  lvdterr(1)  >=  starget(1)  &  cpos(2)  +
lvdterr(2) >= starget(2)...
            & cpos(3) + lvdterr(3) >= starget(3)

    %if valvestate==0
    starget = starget + stroke/n;
    end
```

```
    if cpos > stroke  & downstroke == 0        % Downstroke -
reversal of signs. At this stage a signal is sent to change to
the
                                              %     hydraulic
scheme for lowering.
    starget = -abs(starget);
    cpos = -abs(cpos);
    strokestop = -abs(strokestop);
    downstroke = 1;
    end

    if cpos >= strokestop
        valvestate=valvestate - valvemove;
        stopping=1;

        if valvestate <= 0
        fprintf('Press  enter  to  move  to  next  stroke
position\n')
        pause
        strokestop=strokestop+stroke/intsteps;
        currentstep = currentstep + 1;
        stopping=0;


        % stored variables to check tolerance on cylinder
stopping position
        cposstop(currentstep,:)=abs(cpos)
        tstop(currentstep)=t-dt;

        end
    end


    if currentstep >= intsteps*2
        'Testing complete'
        break
    end

    if stopping == 0
    if cpos(1) + lvdterr(1) < starget(1)
        valvestate(1)=valvestate(1) + valvemove(1);
    else
        valvestate(1)=valvestate(1) - valvemove(1);
    end

    if cpos(2) + lvdterr(2) < starget(2)
        valvestate(2)=valvestate(2) + valvemove(2);
    else
        valvestate(2)=valvestate(2) - valvemove(2);
    end

    if cpos(3) + lvdterr(3) < starget(3)
```

```
        valvestate(3)=valvestate(3) + valvemove(3);
    else
        valvestate(3)=valvestate(3) - valvemove(3);
    end
    end
    % Limit valvestate to real values between 1 and 0
    if valvestate(1) > 1
        valvestate(1)=1;
    elseif valvestate(1) < 0
        valvestate(1)=0;
    end
    if valvestate(2) > 1
        valvestate(2)=1;
    elseif valvestate(2) < 0
        valvestate(2)=0;
    end
    if valvestate(3) > 1
        valvestate(3)=1;
    elseif valvestate(3) < 0
        valvestate(3)=0;
    end



    % Cylinder position simulation assuming that velocity of
cylinder is
    % directly proportional to the position of valve spool.

    cpos(1)= cpos(1) + vel(1)*dt*valvestate(1);


    cpos(2)= cpos(2) + vel(2)*dt*valvestate(2);


    cpos(3)= cpos(3) + vel(3)*dt*valvestate(3);

     % Move dots on graph to new position
    set(p,'XData',x,'YData',abs(cpos))
    %pause(0.1)
    drawnow


    %Store variables for later plotting
    counter=counter + 1;
    cpos1plot(counter)=abs(cpos(1));
    cpos2plot(counter)=abs(cpos(2));
    cpos3plot(counter)=abs(cpos(3));
    t2plot(counter)=t;

    end
    figure
```

```
plot(t2plot,cpos1plot,'r',t2plot,cpos2plot,'bl',t2plot,cpos3pl
ot,'g')
    xlabel('Time(s)')
    ylabel('Cylinder Position (mm)')
    title('Cylinder Position Simulation')
    grid on
    hold on
    plot(tstop,cposstop,'x','MarkerSize',8)

    %Error at stopping positions

    for k = 1:intsteps*2;
    if k <= intsteps
        err(k,:)=cposstop(k,:)-((stroke/intsteps)*k);
    else
        err(k,:)=cposstop(k,:)-(stroke/intsteps)*(intsteps-(k-
intsteps));
    end
    end
    max(abs(err))
```

Appendix C – Hydraulic system schematic



References

1. Automation Direct 2007, *PLC Hardware products* Automationdirect.com, Cumming, GA, USA viewed 19 September 2007, <http://web5.automationdirect.com/adc/Shopping/Catalog/PLC_Hardware>.
2. Enerpac 2006, *Hydraulic Power for all Industrial Applications*, product catalogue, sourced from <www.enerpac.com.au>
3. Euro Press Pack n.d., *Documentation of EURO PRESS Synchronised Lifting System*, company product use instruction, sourced through Specialised Force, Brisbane.
4. Hitchcox, A (ed.) 2005, 'World-record lift succeeds with pinpoint precision', *Hydraulics & Pneumatics*, issue March 2005, viewed 29 September 2007, <http://www.hydraulicspneumatics.com/200/GlobalSearch/Article/False/9505/world+record+lift>.
5. Hohensee, P 2005, *Lift Anything!*, Enerpac, Milwaukee, viewed 23 April 2007, <http://www.infolink.com.au/articles/35/0C03B535.aspx>.
6. Hydra-Capsule 2007, Hydra-Capsule Limited, Worcestershire, UK, viewed 15 September 2007,< http://www.hydra-capsule.com>.
7. Micro-Epsilon n.d., *wireSENSOR Draw-wire sensors,* product catalogue, Micro-Epsilon Messtechnik GmbH & Co., Ortenburg, Germany, sourced from < http://www.micro-epsilon.com>
8. Power Jacks 2004, *E-SERIES Metric Machine Screw and Ball Screw Jacks,* product catalogue, Power Jacks Limited, Fraserburgh, Scotland, sourced from <www.powerjacks.com>
9. Queensland Rail 2003, *Rollingstock Dynamic Performance*, version 1.0, QR Safety and Security Management System standard
10. Rail Safety & Standards Board 2004, *Derailment mitigation – Categorisation of past derailments,* version 1.05, London, viewed 20 May 2007

<http://www.rssb.co.uk/pdf/reports/research/Derailment%20mitigation%20-%20categorisation%20of%20past%20derailments.pdf>.

11. Rohner, P 1995, *Industrial Hydraulic Control*, 4th edn, John Wiley and Sons, Milton.
12. SGS Engineering 2005, *Shorty Cylinders RSS Series,* product catalogue, SGS Engineering UK Ltd, Derby, UK, sourced from < http://www.sgs-engineering.com>
13. Wen, P , Phythian M, Pax, R, Black, J 2007, *ELE 2103 Linear systems and control: Study Book,* University of Southern Queensland, Toowoomba
14. Wikipedia contributors 2007, 'Ladder Logic', *Wikipedia, The Free Encyclopedia,* viewed 20 October 2007 <http://en.wikipedia.org/w/index.php?title=Ladder_logic&oldid=166982247>.
15. Wikipedia contributors 2007, 'Resolver (electrical)', *Wikipedia, The Free Encyclopedia,* viewed 5 October 2007 <http://en.wikipedia.org/w/index.php?title=Resolver_%28electrical%29&oldid=153339232>.
16. Wong J 2005, *Static Twist Test*, version 1.1, work instruction, Queensland Rail – Rollingstock Engineering, Brisbane