University of Southern Queensland
Faculty of Engineering and Surveying

Development of a Mechanical Testing System to Investigate Road Deformation

A dissertation submitted by

Ryan Domrow

In fulfilment of the requirements of

Courses ENG4111 and 4112 Research Project

towards the degree of

Bachelor of Engineering (Mechanical)

Submitted: October, 2005

Abstract

Damage to road pavement is a common occurrence in areas near road intersections. This type of road damage is caused by frequent decelerations and accelerations of vehicles near the intersections. When this occurs, the road is said to have failed due to repeated surface tractions. Through the development of a mechanical testing system to investigate road deformation, a better understanding of the effect of surface traction forces on road pavements will be acquired.

The purpose of this project is to develop a mechanical testing system for investigating surface traction forces, and to provide engineers with a better understanding of these forces that can lead to improved road design. Improved road design will not be investigated in this project, and neither will the performance of various road designs.

The effect of surface traction forces has not been explicitly considered in the design of current road pavements. There is also very little literature in this area, indicating a potential for development in this unexplored research area. There is a need to urgently study the effect of these repeated surface tractions on road pavement designs to develop a better understanding of the performance of the road pavement, and also to increase knowledge in the road design industry.

University of Southern Queensland

Faculty of Engineering and Surveying

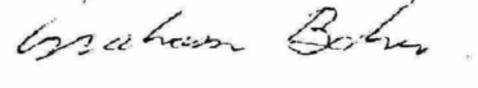
ENG 4111 and ENG 4112 Research Project

Limitations of Use

The Council of the University of Southern Queensland, its Faculty of Engineering and Surveying, and the staff of the University of Southern Queensland, do not accept any responsibility for the material associated with or contained in this dissertation.

Persons using all or any part of this dissertation do so at their own risk, and not at the risk of the Council of the University of Southern Queensland, its Faculty of Engineering and Surveying or the staff of the University of Southern Queensland. The sole purpose of the unit entitled "Project" is to contribute to the overall education process designed to assist the graduate enter the workforce at a level appropriate to the award.

The project dissertation is the report at of an educational exercise and the document, associated hardware, drawings, and other appendices or parts of the project should not be used for any other purpose. If they are so used, it is entirely at the risk of the user.



Prof G Baker
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.

Ryan Domrow
Student Number: Q12221617

____
Signature

________27 October 2005_________________
Date

Acknowledgements

I would like to acknowledge and thank my supervisor Dr. Selvan Pather for his contribution and assistance throughout the entirety of my project

I would also like to thank Dr Jim Shiau for the original project idea, and also for allowing me to work on the project.

I would also like to acknowledge my family and friends for their continual help and support throughout the year.

RYAN DOMROW

University of Southern Queensland
October 2005

Contents

List of Figures

List of Tables

Chapter 1

Introduction

> "Determining the condition of pavement structures is not easy. It is also extremely important to our daily lives and helps determine the amount we spend on pavements and their repair." (Bandara, N. & Briggs, R. C., 2004)

There are only a limited number of machines available capable of determining the condition of a pavement structure. For this reason, it has been decided to design a facility capable of applying accurate, simulated traffic loads to a pavement section, and also capable of allowing investigation into the deformation of the layers of the pavement under the surface.

The facility to be constructed must consist of a laboratory-scale driven wheel guided by a rail system. The system must be driven by the wheel in order to properly simulate surface traction forces on the section of test pavement. Power is supplied to a motor, which will in turn drive the wheel through both a gearbox and belt drive system. The model will allow the wheel to move across the section of test track, lifted above the track, returned to the start, and placed down on the track to being another cycle. These parameters are needed to ensure an accurate simulation of traffic conditions while conducting the experiments on the road pavement.

1.1 Overview

The aim of this project is to design and build a mechanical system to investigate the deformation of road surfaces. The system will be assembled inside a laboratory where conditions can be controlled and altered to simulate variations in weather experienced by road surfaces.

This will involve a mechanical system composed, most probably, of a motor, gearbox, belt drive, and a driving wheel. The wheel will complete a run over a test section of road, be lifted clear of the test track, returned the start of the track, and placed back down on the track. This process will repeat for a large number of cycles, using a process called accelerated pavement testing to create road deformation in this experiment.

In order to do this, all current methods of measuring road deflection will have to be investigated, with research into the areas of accelerated pavement testing (APT) and non-destructive testing (NDT). Research on individual components will also be conducted in order to achieve the best overall design.

As well as causing and measuring the deformation of road surfaces, the deformation of specific layers will be studied through using the system in conjunction with a test section of road. This test section will be able to be removed from the test track and will be taken apart to allow for investigation into the deflection of each specific layer.

1.2 Goals

The overall goal of this project is to develop a mechanical system to investigate the deformation of roads. However, there are certain constraints that need to be considered when choosing the final design.

The project is constrained by a budget, meaning the best-designed system may not be the system that will get built. Consideration will be given to the performance of each component, with the best value for money components being chosen for the final design.

Availability of components is also another issue to be considered. As the project is also constrained by a timeline, preference will be given to components that are readily available rather than those that must be delivered, from either interstate or overseas.

The project also aims to develop a test facility to provide data for better road design. The results obtained from these experiments will allow for the calculation of the elastic modulus of the road, and therefore the fatigue life and maximum stress can all be found from further experimentation and calculation. The structural adequacy of the road can also be obtained from the experiments, as well as an overall estimate of the remaining life of the road.

1.3 Need

There is a need for a system that allows the user to investigate the deformation of the road in more detail. Current systems that use either accelerated pavement testing or non-destructive testing do not allow the user to actually inspect the layers that are deforming under the load. They merely look at the surface damage caused to the road. If a structural weakness were present deep within the road, the current systems would not show this being the reason for the surface deformation. The new mechanical system I am designing will allow the user to see clearly the layers that are deforming, and therefore the results obtained could lead to better road design.

A similar test track located at the Sydney University Pavement Testing Facility is inefficient in a number of ways (see Section 3.6) and also utilises outdated technology. In order to better simulate road traffic, and therefore achieve more accurate road deformation, it has been decided to improve upon the Sydney design with the new mechanical system.

1.4 Methodology

In order to achieve the best final design, ideas from a number of different areas must be looked at. These areas definitely include looking specifically at the current methods of simulation of road traffic. New ideas could also be found in equipment not related to traffic simulation, so any components of machinery that undergoes similar loading or movement to the proposed testing system will be considered.

The current mechanical testing system in Sydney at the Sydney University Pavement Testing Facility also featured strongly when choosing both components and an overall system design. However, this system was designed almost 15 years ago and has become somewhat outdated.

After collating all the data found while researching different designs, a number of designs for a mechanical testing system were proposed. Each of these systems has advantages and disadvantages, and to choose the most beneficial system a decision matrix was used.

Once the final design had been chosen, further component research was conducted. This involved looking into cost and availability, as well as structural properties such as fatigue life and maximum strength. The need for a low maintenance system that could run for an extended period without the need for human supervision forced the choosing of components with high reliabilities.

1.5 Conclusion

Although a seemingly straightforward task of specifying components, a lot of research must be done in order to choose the most accurate design, and also to avoid the shortcomings of other similar systems.

Chapter 2

Background

There are two main areas of pavement testing that apply to this project. They are accelerated pavement testing (APT) and non-destructive testing (NDT). Both methods of testing have strengths and weaknesses, and the final design for my mechanical system will need elements from both methods.

2.1 Accelerated Pavement Testing

Accelerated pavement testing (APT) is a controlled application of wheel loading to a pavement structure for the purposes of simulating long-term loading conditions. Factors critical to this simulation are that the loading configuration and method of loading be realistic with that encountered by regular road surfaces or pavements.

APT allows for the monitoring of pavement performance and response to accelerated accumulation of damage within a shorter time frame, providing road engineers with valuable information concerning the behaviour, performance, and life expectancy of pavement structures. This ultimately results in enhanced understanding of pavement structures and improved, cost effective design and rehabilitation construction methods.

APT furthermore provides a controlled testing environment in which innovative pavement designs and new materials can be tested and validated in a short period without the financial risks associated with failures of in-service experimental pavements.

The costs involved with full-scale APT testing on test roads using actual traffic initiated research into alternative methods of APT. Currently APT is differentiated into three main categories namely: full scale test roads; mobile APT units; and fixed APT units.

Full-scale test roads are APT facilities at which actual vehicle traffic is utilised for the application of loading.

Mobile APT units are load frames with variable axle and wheel configurations designed to simulate the load application of actual vehicle traffic on a limited test section. Mobile APT units can be transported to various locations for testing and are applicable to the investigation of in-service pavements in addition to specifically constructed test sections.

Fixed APT units also utilise load frames for load application but have a fixed location and cannot be transported easily to various testing locations. [4]

2.2 Non-Destructive Testing

Non-destructive testing (NDT) of pavement structures has gained popularity in the recent past. Non-destructive equipment operates by applying a load to the pavement surface and measuring the resulting surface deflections. The results from this equipment can be used to calculate the elastic modulus of asphalt, pavement structural adequacy, and the remaining structural life.

There are three main types of non-destructive deflection testing, which are static or slow moving deflection testing, steady state or dynamic vibratory deflection testing, and impulse deflection testing. [4]

2.2.1 Static or Slow Moving Deflection

Static deflection equipment is used to measure pavement surface deflections under static or slow moving loads. The most common piece of equipment used for this is the Benkelman beam. This method of testing provides deflection measurements at any number of points under a non-moving or slow moving load. The Benkelman beam test procedure involves the measurement of a pavement surface rebound with a cantilevered beam as a truck loaded to 80 kN on its rear axle moved from rest. Measurements are made between dual tires on the rear axle at specified intervals in the outer wheel path.

The main advantages of these static or slow moving deflection testing devices are simplicity, low instrument cost and the possibility of utilising realistic load levels. The disadvantages of these devices are that they are slow, labour intensive, do not provide a true deflection basin, and suffer relatively poor precision and bias. [4]

2.2.2 Steady State or Dynamic Vibratory Equipment

Steady state or dynamic vibratory equipment uses a relatively large static preload and a sinusoidal vibration to the pavement with a dynamic force generator. With some devices, it is possible to change the magnitude and the frequency of the applied load. A major problem with this equipment is that the relatively large static preload may adversely affect the accuracy of the test.

However, these systems are generally highly reliable with low maintenance and can produce a full deflection basin. Disadvantages include a significantly low dynamic load, relatively large static preload, susceptibility to errors due to pavement resonance effects and inadequate dynamic load to test heavy pavements. [4]

2.2.3 Impulse Deflection Equipment

Currently, impulse deflection equipment is the most popular and widely used pavement deflection measurement technology. All impulse type NDT devices produce a transient load to the pavement surface typically lasting 25 to 30 ms. The impulse load is generated by a falling mass from one or more predetermined heights. The resulting load pulse is transmitted to the pavement as a half sine wave. The peak deflections and load magnitude are captured, reported and automatically stored

Impulse load devices can apply loads from 1360 to over 22 700 kg based on the device used. This equipment has a relatively low preload so its influence on the pavement response is negligible.

Deflections are most commonly measured with velocity transducers (seven or more), which are mounted on a bar and automatically lowered to the pavement surface with the loading plate. One transducer is located in the centre of the loading plate and others are located at different distances from the loading plate.

Advantages of impulse deflection testing equipment include high productivity, realistic pavement loading levels, low static preload, rapid data acquisition and the ability to measure and record a deflection basin. However, initial costs for the impulse equipment are higher than static and vibratory devices and they are more complex in nature. [4]

2.3 Conclusion

Although there are only two main areas of measuring pavement deformation, each area is quite broad. However only accelerated pavement testing will be investigated fully for this project. Several machines that use non-destructive testing to measure pavement deformation will be researched to ensure the design for this project does not have the same shortcomings as these devices.

Chapter 3

Current Equipment

There are several current machines specifically designed to measure the deflection in a road surface. These include the Heavy Vehicle Simulator, which fits into the APT area, as well as the Rolling Weight Deflectometer, Falling Weight Deflectometer, and Heavy Weight Deflectometer, which are all in the NDT area of road deflection measurement.

3.1 Heavy Vehicle Simulator

The South African Council for Scientific and Industrial Research (CSIR) designed the Heavy Vehicle Simulator (HVS) almost 30 years ago, and it was primarily designed to simulate heavy road traffic. Newer versions of the HVS have been modified to simulate aircraft. The HVS provides a mobile laboratory with the ability to accurately and cost effectively test and monitor pavements under a variety of environmental and loading scenarios. [15]

Figure 3.1: Heavy Vehicle Simulator [15]

3.1.1 Specifications

The HVS is electrically powered with hydraulic motors for use inside specific facilities. It is also equipped with a diesel engine to move from test-point to test-point. It weighs approximately 46 tons with dimensions of 22.6 m x 3.5 m. It is capable of applying wheel loads varying from 4 tons to 30 tons per axle (nearly four times the national legal maximum), through a standard half axle dual truck tyre configuration or modified aircraft wheel configuration.

Figure 3.2: Heavy Vehicle Simulator Tyre Configuration [15]

A total of 14000 single directional passes can be completed in 8 hours at a speed of between 13 and 16 km/h. This enables the HVS to apply 20 years worth of heavy traffic in only four or five months. [15]

3.1.2 Advantages

The HVS has several advantages over other simulators, such as being able to simulate a wide range of loading possibilities. The HVS is also the only simulator able to simulate environmental conditions such as temperature and moisture variations above the road surface. This is useful when the local weather is quite unstable with large variances in temperature and humidity, either throughout the day, or over a longer period of time. As mentioned above, the HVS is capable of uni-directional passes to properly simulate road traffic. This ability is an incredible advantage when trying to precisely simulate traffic conditions, and is one of the main design points for the system I am designing. [15]

3.1.3 Disadvantages

Size is the main limiting factor of the HVS. The newest model is almost 23m long and has a mass of 46 tons. Therefore the mobility of the HVS is directly affected. The HVS must be towed to the test site, but it is equipped with a diesel engine to manoeuvre about the test site rather slowly. The other main drawback of the HVS is that it cannot investigate the deformation caused during the test. Separate equipment is used to measure the deformation, but most importantly, the specific layers of road base that have deformed throughout the test are unknown. [15]

3.2 Falling Weight Deflectometers

These highly accurate, well supported, and reliable products are a proven load or deflection measurement solution for many engineers worldwide. The Dynatest FWD technology provides a measurement foundation for the analytical-empirical pavement engineering methodology, a system of advanced automated pavement measurement, analysis and management engineering services and products. [8]

3.2.1 Specifications

There are two main types of falling weight deflectometers, the Falling Weight Deflectometer (FWD) and the Heavy Weight Deflectometer (HWD). The two are similar and use the same technology to measure the deflection of road surfaces. The FWD is capable of applying loads from 7 to 120 kN, whereas the HWD can apply loads up to 240 kN. The range of Falling Weight Deflectometers were designed primarily for multi-purpose pavement applications ranging from unpaved roads to airfields. However, the HWD is the only one in the range capable of use on airfields. [8]

Figure 3.3: Falling Weight Deflectometer [8]

Figure 3.4: Heavy Weight Deflectometer [8]

3.2.2 Method of Measurement

The use of a FWD enables the user to determine a deflection basin caused by a controlled load with accuracy. The FWD applying a dynamic impulse load to an area of pavement causes the deflection basin. Measurements are taken by optical sensors and collated by a computer before the load is released and the pavement returns to its original position. The optical sensors measure the difference in time it takes to reflect a beam from the pavement in its original position to the time it takes in the deflected position. Knowing the velocity of the optical beams sent out, the distance the pavement has deformed can be calculated. [8]

3.2.3 Advantages

There are several advantages the FWD and HWD have over other pavement testing machines. They can both be operated by a single person and are non-destructive testing machines. They are also highly accurate and have excellent repeatability with precise control over both the dynamic impulse load and the optical sensors used in measuring the deformation basin. [8]

3.2.4 Disadvantages

The main limitation of the FWD and the HWD is the cycle time. While stationary above one test point, the FWD and HWD have fairly good cycle times, but the limitation becomes evident the machinery must be frequently moved between a number of test points. Each time they must be disassembled at the previous point, moved into position by a separate vehicle, and then reassembled and calibrated at the new point. This amounts to a best cycle time of around 60 test points in each hour. Again, the FWD and HWD can only apply the loads and give no indication into which layers in the road deform under certain types of loads rather than others. [8]

3.3 Rolling Weight Deflectometer

The Rolling Weight Deflectometer (RWD) is a device designed to measure and record road deflections at highway speeds, thereby eliminating the need for road closure or specific test facilities. It also helps officials target funding and projects to the segments of road that need structural improvement and rehabilitation. However, the RWD is still in the prototype stage in America and production models are unavailable. [17]

Figure 3.5: Rolling Weight Deflectometer [17]

3.3.1 Specifications

The RWD is actually a regular semitrailer 16m long with an aluminium beam running underneath the length of the trailer. The beam measures 7.8m long x 51mm wide x 216mm high and is specifically designed to house four lasers spaced 2.6 metres apart. The beam is mounted on the right side of the trailer, which in America is the side closest to the weakest part of the road, the outside edge. The semitrailer places over 80kN of dead weight over the rear axle of the dual axle trailer. [17]

3.3.2 Method of Measurement

The Harr approach is the most commonly used method to measure the deflection of a road surface. Put simply, it involves gauging the distance from a reference datum to the road surface at three equally spaced points ahead of the load wheel. Once the load wheel has moved forward a distance equal to the spacing between the measurement points, the same three points on the pavement are measured again using the second, third, and fourth sensors. In the case of the prototype RWD, this distance is 2.6m.

The only disadvantage of the Harr approach is the need for the reference datum to be absolutely straight at all times. Obviously, with temperature and loading variations it is impossible to keep the aluminium beam perfectly straight. It is for this reason a set of optical alignment sensors is mounted on each pavement height laser. With constant measurements being taken, the accuracy of the pavement height laser is around 20 µm. [17]

3.3.3 Advantages

The main advantage the RWD has over the HVS is the fact it can measure the deflection of road surfaces on normal roads while maintaining the flow of traffic. This, of course, eliminates the need for lane closures, reducing traffic congestion and also work zone safety issues. Again, unidirectional passes are used because the RWD has no additional loading mechanism; rather it uses the weight of the semitrailer to which it is attached. [17]

3.3.4 Disadvantages

The first disadvantage of the RWD is that it is a relatively new technology. This means it will not be commercially available for some time, and will also probably need several iterations to find and fix flaws in the system. For instance, it has been found the RWD experiences a warming up effect prior to the stabilisation of readings, where the first one or two runs over a test section show markedly higher deflections than the others. Also, when compared with the HVS, it has a relatively slow rate of loading. While the HVS can simulate 20 years accumulation of traffic in only a few months, the RWD only simulates one heavy vehicle for each pass completed over a test section. While the results may arguably be more accurate, the HVS would still be the preferred choice of heavy traffic simulator for the sheer volume of traffic it can simulate. [17]

3.4 Accelerated Loading Facility

The Accelerated Loading Facility (ALF) is an increasingly popular machine for accelerated pavement testing for several reasons. While expensive, the ALF has many advantages, which will be discussed shortly, that make it an attractive option when looking for a vehicle simulator. [12]

Figure 3.6: Accelerated Loading Facility [12]

3.4.1 Specifications

The ALF consists of a reciprocating rolling wheel load mechanism used to test pavements. The wheel assembly travels at 8km/h while testing in one or both directions, and with optional random lateral wheel wander of up to 25cm. The 12m wide pit permits testing at several locations across a full-scale pavement facility. The ALF offers considerable flexibility in testing pavements. Not only can different pavement, base, and subgrade materials be tested and compared under known conditions, the heavy duty loading mechanism permits the evaluation of various tire configurations and load levels on performance. These features make it easily the most advanced vehicle simulator when the loading mechanism alone is considered. Current worldwide projects utilising the ALF include studies of ultrathin concrete, verification of three-dimensional pavement models, dowel bars, materials, and other aspects of flexible and rigid pavement structures.

The loading mechanism in the environmentally controlled chamber of the ALF helps engineers and researchers evaluate the effects of various environmental conditions, materials, and load levels on a variety of pavement structures. The ALF has several attributes that set it apart from similar machinery around the world. Major features include:

- Bitumen and concrete testing capability,
- Full access for construction equipment to place pavements in accordance with standard highway specifications,
- Multiple test paths across the 6m wide pavement,
- Broad loading versatility with the ability to use various wheel loading configurations, loads from 4000kg to 14000kg, as well as optional random lateral wander of the loaded wheel or wheels, depending on the configuration, to better simulate road traffic,
- Full environmental control within the enclosed test facility to regulate air temperature and humidity. Moisture also can be added to the subsurface pavement structure,
- Optional instrumentation to monitor pavement response to environmental changes and/or dynamic loading.

All of these features together make one of the most advanced traffic simulators throughout the world; making it a popular choice when large scale accelerated pavement testing is needed. [23]

3.4.2 Method of Measurement

While it does have an advanced loading mechanism, the ALF itself does not measure the road deformation it causes. The current idea is for separate facilities to construct their own test pits of pavement and hire an ALF to conduct the accelerated pavement testing. Once the tests are finished, the measurements will be undertaken at the facility to determine the extent of the deformation.

However, dedicated facilities are becoming increasingly present throughout mainly the U.S. where an ALF is permanently located at a facility in a completely dedicated building. An example of this is at Ohio University, where a substantial grant of $1.35 million was given for the construction of a fully enclosed accelerated pavement loading facility. The facility itself consists of the environmental room housing the ALF, and also a dedicated test pit. To reiterate though, while the ALF can be used effectively in conjunction with dedicated facilities to both apply various loading patterns and to measure the deformation, the ALF itself is not capable of measuring the deformation, and is certainly not capable of investigating the deformation further. [23]

3.4.3 Advantages

As mentioned before, the main advantage of the ALF is the advanced loading mechanism. While they vary slightly from machine to machine, the exact specifications for one ALF were available and are listed below.

- Load Range: 40kN to 140kN
- Test Directions: Uni or Bi directional
- Test length: 10.5m
- Tires: Standard single, standard duals, and wide-base single
- Lateral Wheel Wander: Random up to 25cm optional
- Test Speed: up to 8km/h
- Load Applications per Hour: 250 for unidirectional tests, 500 for bidirectional test. [23]

3.4.4 Disadvantages

Like the HVS, the main disadvantage of the ALF is it size and lack of mobility. While the HVS has an engine to maneuver slowly about the test site, an ALF lacks even this basic function. The ALF basically is a 29m long structural frame with a moving wheel assembly controlled by a computer. The lack of mobility of the ALF means it must be transported by truck to the test site or facility, and then positioned by crane. These factors lead to increased costs in order to conduct the experiments.

The other major disadvantage of the ALF is the need for extra equipment to both measure and investigate the deformation caused during the pavement testing. As already shown, many pavement-testing machines throughout the world do not investigate the deformation they cause, and in cases such as the HVS and ALF, they do not even measure the deformation caused. This is the area that will be rectified by this project through the design of a system that can both effectively measure and investigate the deformation caused during the test. [12]

3.5 The Wheel Tracker

The Wheel Tracker, shown below in figure 3.7, is a machine designed specifically to assess the resistance of road surfaces to deformation from simulated traffic conditions. Slightly different versions of the Wheel Tracker are available due to the number of different companies manufacturing the system. [20]

Figure 3.7: The Wheel Tracker [20]

3.5.1 Specifications

The Wheel Tracker consists of loaded wheel that travels on a test sample of pavement held on a moving table. The table moves with simple harmonic motion through a distance of 230mm. This differs from the other accelerated pavement testing machinery in that it is the only system where the wheel is held stationary. The table has a frequency of 42 passes per minute. The diameter of the loaded wheel is 200mm and has a thickness of 50mm. A load of 520N is applied to the tyre under standard operating conditions.

The actual cabinet in which the wheel and test section are enclosed can be used to vary both temperature and humidity. Some variations of the Wheel Tracker allow a range of 30°C to 60°C, whereas other models range from 30°C to as high as 65°C.

The Wheel Tracker encompasses a safety device to prevent injury during operation of the machine. Safety switches are fitted to the polycarbonate doors of the cabinet that prevent operation of both the drive and temperature control systems when the doors are opened. [20]

3.5.2 Method of Measurement

The Wheel Tracker is fitted with a LVDT to monitor the depth of the rut during the experiment. The unit can also be fitted with two thermocouple inputs for optional specimen temperature measurement.

Both the deformation and temperature are recorded onto an IBM compatible computer fitted with a data acquisition and control (DAC) board. The computer is used to start and stop the Wheel Tracker, as well as controlling speed and acquiring the deformation and temperature data. Once an experiment is completed, the results are converted to a text file and can be printed to produce a hard copy. [20]

3.5.3 Advantages

The Wheel Tracker offers several advantages over the other traffic simulators investigated. It is the first design to actually scale down the sizes and forces to make operation both simpler and less expensive. Also, another excellent advantage of the Wheel Tracker is its computer compatibility. This compatibility means the entire system can be controlled externally by a single operator and also eliminates the need for any immediate human presence during operation, eliminating the possibility of workplace injury.

Another advantage offered by the Wheel Tracker is the extremely low cycle time. Completing 42 passes in 60 seconds deform the pavement extremely quickly, meaning useful results can be obtained in a short period of time. [20]

3.5.4 Disadvantages

While the cycle time of the Wheel Tracker is high, moving the pavement sample with simple harmonic motion underneath the wheel will result in bi-directional passes. While this results in a high cycle time, it is not really an accurate simulation of forces on a road surface as traffic generally travels in only one direction over an area of pavement.

Although the Wheel Tracker is cheaper than its large-scale counterparts, the price of the system is outside that of the budget for this project. The Wheel Tracker retails at QCQA.com for $27 800 US, or about $39 700 AUD. While purpose-built facilities such as that at Ohio University in America cost more than 1 million dollars, the price of the Wheel Tracker is still high enough to push it outside the boundaries of this project. [20]

3.6 Sydney Pavement Testing Facility

The Sydney Pavement Testing Facility is located on campus at Sydney University. It was developed in 1994 to test model pavements and further modified in 1998 in order to randomly change the position of the tyre across the pavement. The facility consists of a test bed, an overhead track, and the loading carriage. [26]

Figure 3.8: Sydney Pavement Testing Facility [26]

3.6.1 Specifications

A plan of the Sydney University Pavement Testing Facility test track is shown below in figure 3.9.

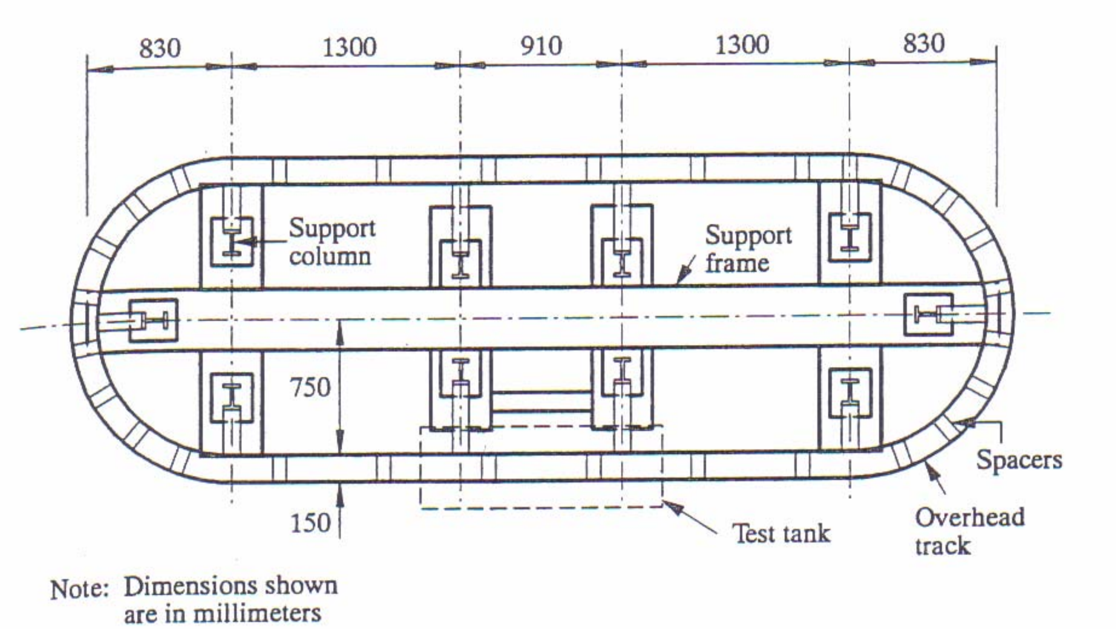


Figure 3.9: Plan of Test Track at Sydney Pavement Testing Facility [26]

The track itself is an oval configuration measuring an overall length of 5170mm by 1500mm across. The test tank measures 1.4m long by 0.5 m wide. It consists of two hollow steel boxes bolted together to allow the lower boxed to be used to prepare a soil layer if required, and for different materials to be placed in the upper box. A false bottom can be placed into the tank to simulate different pavement thickness if required.

The loading mechanism consists of a wheel driven by an electric motor through a belt drive and loaded by four compression springs. Each spring is capable of applying up to 90kg, but in order to increase fatigue life, a value of only 70kg is applied. The wheel used is a regular pneumatic tyre with a steel rim of 220mm diameter. Other specifications available are listed below. [26]

Specifications of Sydney Pavement Testing Facility	
Feature	Specification
Speed Range (km/h)	0 – 7.2
Wheel Load (kN)	0 – 1.4
Maximum Tyre Pressure (kPa)	500
Tyre Width (mm)	45
Length of Test Section (m)	1.4
Width of Test Section (m)	0.5
Maximum Depth of Tank (m)	0.8
Length of Test Track (m)	12.15
Cycle Time at 1 km/h (s)	44.0
Motor Power (kW)	0.75

Table 3.1: Specifications of Sydney Pavement Testing Facility [26]

3.6.2 Method of Measurement

Linear Voltage Transducers are responsible for both measuring subsurface settlements and also monitoring the variation in compression for each spring in the loading mechanism.

The design at the Sydney Pavement Testing Facility is the first to be able to investigate the deformation caused by accelerated pavement testing and repeated surface tractions. This can be done by disassembling the test bed for an actual look at the deformation caused to each layer under the road surface. Measurements can also be taken, both at the surface and also for each layer.

This enables engineers and researchers a much better frame of reference when comparing the deformation of different pavement designs. It can be used effectively because although surface deformation may by less for a certain pavement design, the deformation caused underneath the surface in this same design may be greater and lead to more serious pavement damage further along in the testing process. [26]

3.6.3 Advantages

The greatest advantage the Sydney Pavement Facility has over other vehicle simulators is obviously that it is the first design to allow the engineer to investigate the deformation of each layer underneath the road surface. This feature was much needed after the plethora of accelerated pavement testing machines provided good data on surface deformation, but none really looked into deformation in the layers underneath.

Another advantage of the Sydney Pavement Facility is the small scale of the design. Reducing the size of components and forces also reduces the space required to build the facility, and also reduces the costs involved. It may also improve such factors as time involved in construction and testing, and all the while provide accurate test results in the same amount of test time.

The Sydney design also offers other good qualities such as a high rate of data reading and the ability to applied varied loads through the spring loading mechanism. The reliability of the system is also high due to the relative simplicity of the components involved. [26]

3.6.4 Disadvantages

The disadvantages of the Sydney Pavement Testing Facility can be broken into two categories. These are disadvantages when compared to other vehicle simulators and disadvantages when compared with the proposed design for this project.

Firstly, when compared with other vehicle simulators, the Sydney design lacks use of current technology. While vehicle simulators such as the HVS have been released in new incarnations every few years to match the advancements in technology, the Sydney design has remained the same despite these advancements.

When compared with the proposed design for this project, the disadvantages of the Sydney design become increasingly numerous. Most of these disadvantages stem from the use of outdated technology that will be rectified in this project.

When the design was first constructed in Sydney, there was not sufficient technology available for the engineers to consider a linear test track. The linear test track presents a need for the wheel to be stopped at the end of a run, lifted above the track, returned to the start, and then placed back onto the track. Because this technology was lacking, it left the engineers with a choice between a circular track and an oval shaped track. An oval track was chosen as it eliminated the need for lifting the wheel off the track, and also allowed a straight test section to be retained.

However, the oval test track does present several disadvantages. The track itself takes up a lot of extra space when compared with a linear test track, and also the test section is very small when compared with the overall size of the track.

The last major disadvantage of the Sydney design when compared with the proposed design of this project is the cycle time. As stated in Table 3.1, the cycle time of the Sydney rig is 44 seconds at 1km/h. Although small when compared with current vehicle simulators like the ALF, this cycle time definitely has room for improvement and is a major factor when experimenting using accelerated pavement testing. A reduction of only five seconds in the cycle time would result in an extra 1760 completed in a week if continuous testing was undertaken. Obviously, completing the cycles faster will cause the pavement to deform faster, allowing for useful results in a shorter period of time. [26]

Chapter 4

Conceptual Design

An important stage of any design process is the conceptual design phase. To get started on the conceptual design of the new system, several broad ideas were investigated. After collecting several decent ideas, a decision matrix was constructed to choose the best overall design. Once the final design is chosen, the selection of components can commence.

4.1 Initial Conceptual Design

The initial conceptual design process involved looking at systems that involved lifting and dropping of a mass in the vertical direction, as well as systems that moved horizontally. While by no means common, there are systems used for a variety of tasks that exhibit this behaviour.

An overhead crane, as show below in figure 4.1, can be found in most workshops and foundries. The crane is quite a good model to investigate as it can move freely both forwards and backwards along the girders. The crane can also obviously lift and lower a mass independently of its horizontal travel.

Figure 4.1: Overhead Crane

One idea that evolved from looking at an overhead crane was the possibility of having two rails running parallel to the test track. The carriage would roll forwards and backwards along the two rails while being driven by the loaded wheel rolling along the actual test track and test bed. There are many variations of loading mechanisms that could be used in conjunction with this system, such as spring loading, hydraulic loading or simply dead weights.

Another type of system with movements similar to the proposed mechanical testing system is a skill-tester game. These games are usually located in shopping centres or arcades and involve an electric motor moving a mass along a single rail, then dropping a claw or magnet before retracting it again and returning to the original position. This system is slightly more complex than the proposed system for this project because it is required to move in two directions as well as dropping and lifting a mass, rather than moving in only one direction like the carriage for this project.

The main idea that grew from looking into this area involved moving the system along an overhead rail, much like the rig at the Sydney Pavement Testing Facility. Seeing as the design of this system is really the redesigning of the Sydney Pavement Testing Facility rig, a lot of work would have already been completed.

So, after the initial conceptual design process, the two main ideas to emerge were:

- a design based upon the system at the Sydney Pavement Testing Facility with modifications made to correct faults, and
- a new system involving a set of parallel rails running the length of the test track with the carriage spanning the gap and being driven by the loaded wheel in the centre.

Both of these concepts were investigated thoroughly before a decision matrix was used to decide upon the best overall design.

4.2 Overhead Rail Concept

The basic idea for the overhead rail concept was an improvement upon the carriage used in the Sydney design. The concept design is shown below in figure 4.2.

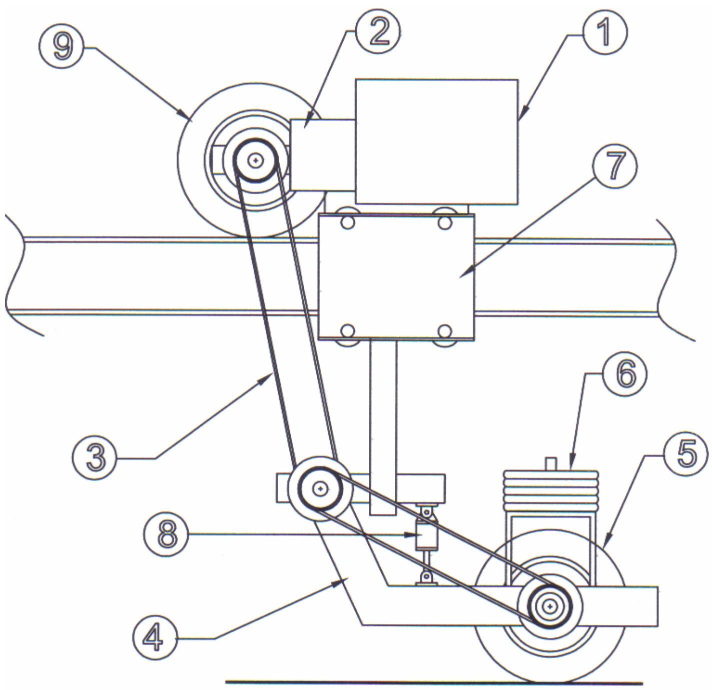


Figure 4.2: Overhead Rail Concept Design

As can be seen from the picture, the most important components of the design are labeled. These labels are explained further in Table 4.1.

Number	Description
1	Motor
2	Gearbox with electric clutch
3	Belt drive
4	Pivot arm rotating about pulley centre
5	Drive wheel
6	Load
7	Free rolling bogie on gantry track
8	Lifting mechanism
9	Reverse wheel

Table 4.1: Legend for Figure 4.2

The overhead rail concept involves a motor and gearbox fixed to a bogie. The bogie is free to move in both directions along the gantry beam shown. While in forwards motion, the bogie will be driven by the drive wheel at all times to properly simulate traffic, rather than having the loaded drive wheel simply rolling over the test section.

Figure 4.2 shows a few additions from the Sydney design. Additions such as a wheel for reverse motion and a cylinder to lift the drive wheel were necessary due to the slightly more complicated design. The entire driving wheel assembly can rotate about the pulley centre when the lifting mechanism is activated. The use of a belt drive has eliminated the possibility of using the drive wheel on the end of a retractable arm.

A simplified 3D view of the overhead rail concept is shown below in figure 4.3.

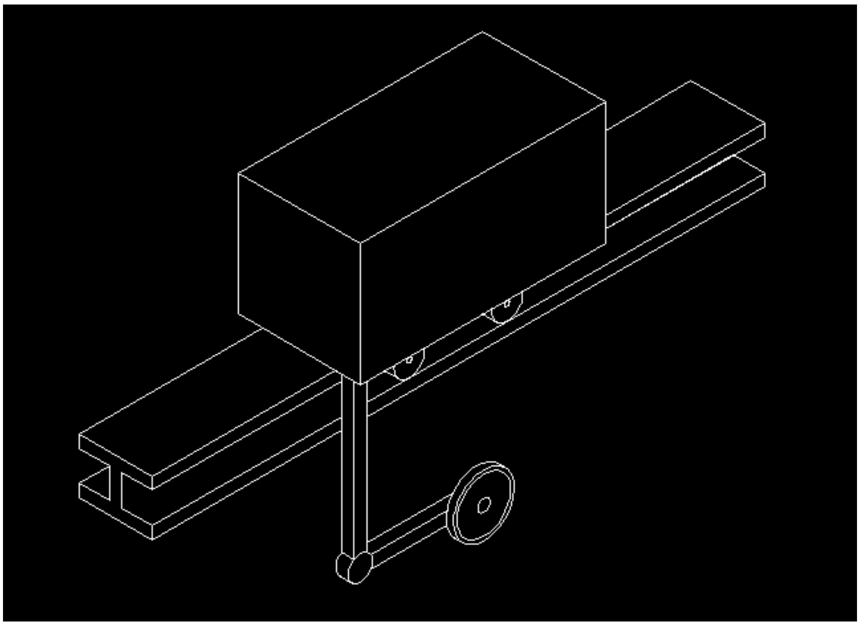


Figure 4.3: 3D view of Overhead Rail Concept

4.3 Rails Concept

As stated earlier, the rails concept involves a set of parallel rails running alongside the test track. The carriage will be required to span the gap between the rails, and will be driven by the loaded wheel running over the test track. Figure 4.4 shows a 3D view of the carriage.

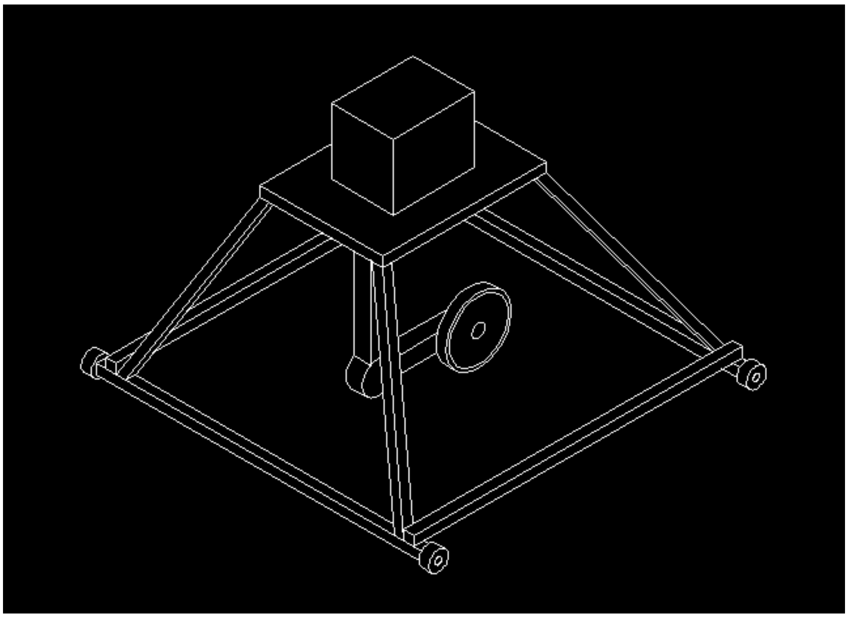


Figure 4.4: Carriage for Rails Concept

It can be seen from figure 4.4 that the motor and gearbox will sit on the top of the carriage with the driving wheel directly underneath. The loaded wheel assembly will still pivot about the centre of the pulleys. However, there is no reverse wheel for the rails concept so a completely new return system will be needed if this design is chosen. The small wheels the carriage uses to run along the rails will need to be specified to allow almost frictionless motion in both directions.

Also, the thickness of the members in the frame will be critical, as small members may have a high deflection, and larger members will cost extra money. The method of joining members will also have to be specified, with alternatives including welding and bolting.

4.4 Decision Matrix for Carriage Design

A decision matrix is a chart that allows analysis and a final rating to be given to a number of different alternatives. The nature of a decision matrix allows a direct comparison between alternatives with the highest scoring alternative being considered the best option for implementation.

After the two alternatives were decided upon, a set of key criteria has to be selected. The qualities deemed important in the carriage design are as follows:

Criterion
Low cost
Reproducibility of results
Ease of Assembly
Life
Maneuverability
Weight
Ease of Maintenance

Table 4.2: Criterion for Carriage Conceptual Design

Although there may be other factors influencing the design process such as personal preferences, the list of criteria above was considered to be the most useful qualities when deciding upon the final carriage design.

Once the list of criteria was finalised, a weighting was given to each quality. This was to ensure the most important qualities were given a higher importance, and the less important qualities would not affect the final design choice as severely. The weightings, with a high weighting signifying a high importance, were assigned as follows:

Criterion	Weighting
Low cost	3
Reproducibility of results	3
Ease of Assembly	2
Life	3
Maneuverability	1
Weight	1
Ease of Maintenance	2

Table 4.3: Criterion and Weighting for Carriage Conceptual Design

In table 4.3, it can be seen that both low cost, reproducibility of results, and the life of the carriage are considered the most important characteristics. These are fairly self-explanatory as the importance of low cost stems from the restrictive budget, the reproducibility of results because of the high number of cycles to be conducted with the system, and the life of the carriage, again because of the high number of cycles to be undertaken.

Ease of assembly and maintenance were considered fairly important as the design should be fairly easy to assemble, and it should also be easy to conduct maintenance upon the equipment.

Maneuverability and weight of the carriage were considered less important because once the system is assembled and in place, it should not require any further adjustments. However, the maneuverability and weight of the carriage may have a direct impact on the positioning of the system, which is why they must be taken into account when deciding upon the final design.

Finally, a score from 1 to 5, with 1 being the lowest and 5 the highest, was assigned to each of the criteria for both alternatives, allowing an overall score to be given to both, hence allowing for a decision to be made upon the carriage design. The completed decision matrix is shown below in Table 4.4.

Criterion	Weight	Rails	Overhead Rail
Low Cost	3	3	2
Reproducibility of results	3	5	5
Ease of Assembly	2	5	2
Life	3	4	3
Manoeuvrability	1	5	1
Weight	1	3	4
Ease of Maintenance	2	4	3
Total Rating		**62**	**45**

Table 4.4: Decision Matrix for Carriage Conceptual Design

To get the total rating given for each of the two alternatives, simply multiply the score from each category by the weighting of that category and sum them.

4.5 Scoring Explanation

As shown in the decision matrix in table 4.4, the scores for each of the alternatives vary significantly throughout the criteria. The reasons for this are outlined below.

While the overall cost of the proposed system will be significantly lower than systems such as the Wheel Tracker, it will still be a fairly large investment. For this reason, the rails concept was given a mid-range score of 3. Most of the components in the rail concept that differ from those in the overhead rail concept such as the members and caster wheels would be only minor costs. The overhead rail concept would be slightly more expensive because of the need for more complex components such as the free rolling bogie. The extra cost involved with the bogie would be for factors such as precise machining with lower tolerances and a higher grade of steel to deal with the larger forces associated with hanging the entire assembly from an overhead track. For the reasons listed above, the overhead rail concept was given a slightly lower score of 2.

Both of the concepts were given a perfect score of 5 for reproducibility of results. The overhead rail concept has been shown to provide excellent results at the Sydney Pavement Testing Facility, so it can be inferred that the slightly improved concept design for this project would also produce accurate results. Although the rails concept is a completely new idea for used in accelerated pavement testing, a perfect score has been given because the simplicity of the design would allow for excellent repeatability, and therefore the reproducibility of results would also be very good.

The rails concept was also given a perfect score for ease of assembly. The concept would require only basic mechanical skills to assembly as all the joints are very simple and the wheels and motor are easily fixed to the frame. Weight distribution is not critical either as the nature of the design would allow more weight to be placed on either one side or one end without adversely affecting the life of the carriage or ability of the design to produce accurate results. However, the overhead rail design would be harder to assemble, mainly because of the nature of the design. The loaded wheel assembly will have to be placed to one side of the overhead rail, creating a larger force on one side of the bogie. Also, aligning the wheels of the bogie to run freely in both directions will require extra time and effort, whereas it is not as important for the wheels on the rails concept to run almost without friction. For these reasons, the overhead rail concept scored only a 2 for ease of assembly.

The life of the carriage is important because of the high number of cycles to be completed. Also, having a carriage with a long life will reduce ongoing costs such as maintenance costs. The rails concept was given a fairly high score of 4 for this category, as it would be possible to keep the motor running at all times. With the introduction of an independent return system, the drive wheel could be raised off the track and the motor could be left running while the carriage is returned to the start of the test track. Allowing the motor to continue running, even at a slower speed, rather than switching it off and reversing the direction of travel, will increase the longevity of the motor, and also lead to lower maintenance costs. The overhead rail concept scored slightly lower than the rails concept because of the fact the motor must be reversed to drive the carriage back to the start of the track. This process will significantly shorten the life of the motor. Also, the larger moments associated with the overhead rail design will produce increased wear and tear on components such as the free wheeling bogie.

While not very important, the maneuverability of the carriage was a quality that was needed in the final design. If the entire system ever needed to be moved for any reason, such as maintenance or relocation, a carriage with high maneuverability would be advantageous. The maneuverability of the rails concept was considered extremely high, as the carriage can be lifted directly from the track and either carried or rolled to a new destination. The overhead rail concept, however, has very little mobility. In order to move this system, either the entire assembly with the overhead rail must be moved, or the carriage must be disassembled to free it from the gantry beam. For this reason, the overhead rail concept was given a poor score in this category.

Again, while not very important, the overall weight of the carriage does need to be taken into consideration. While both carriages will have basically the same components, the rails concept has the extra mass of the framework to take into account, whereas the overhead rail design has no real framework to consider. Having said this, both carriages will still have a fairly sizeable mass because of the mass of components such as the motor, as well as the amount of steel used in both carriages. The overhead rail concept scored slightly higher in this category than the rails concept because of the lack of a distinct frame.

Finally, the rails concept scored slightly higher in the final category. The rails concept would be easier to maintain because of its mobility and simplicity. It is relatively easy to move the carriage to undertake maintenance on either the carriage itself or the test track, and the carriage is also easy to disassemble, allowing for easy access to various components for maintenance. The overhead rail concept, however, is more difficult to separate from the test track; meaning undertaking maintenance on the track would be more complicated. Also, if maintenance of components such as the motor or bogie were needed, the carriage would have to be disassembled in order to allow the motor or bogie to be taken away and worked on separately. So for these reasons, the rails concept again scored slightly higher than the overhead rail concept.

4.6 Conclusion

As shown in table 4.4, the rails concept has the highest score of the two alternatives after finishing the decision making process. Therefore, the rails concept, shown below in figure 4.5, has been chosen as the final design, and the remaining components will be designed to be compatible only with the rails concept.

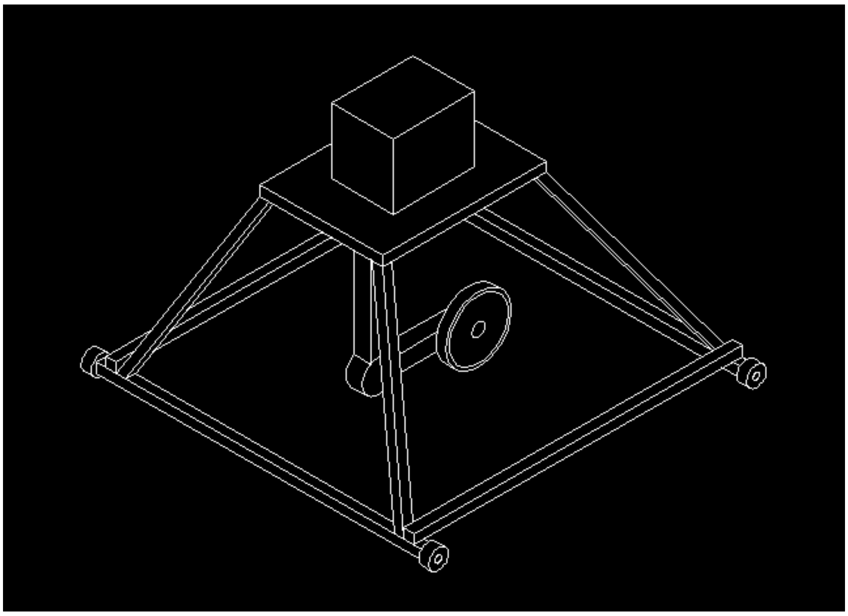


Figure 4.5: Rails Concept - Final Design

Chapter 5

Component Selection

Although the carriage design has been chosen, the rest of the components must now be selected. These include the motor, the return system, the loading mechanism, the lifting mechanism, the drive system, and also specifying the wheels and rail system. The control of the system must also be investigated to ensure a computer can control the entire system.

5.1 Motors

While an electric motor seems like an obvious choice, there are many different types available. These include single phase AC, three phase AC, and DC motors. Each type of motor can also have many different configurations, such as an induction motor, polyphase motor, wound rotor, and multispeed AC. Also, because of the possibility of using a hydraulic loading system for the design, the advantages and disadvantages of a hydraulic motor were also investigated.

Before a choice can be made on the type of motor for this project, research into the advantages and disadvantages of the more common motors must be undertaken to ensure the final choice is made correctly.

5.1.1 Electric Single Phase AC Motors

Induction AC motors are the simplest and most rugged electric motor and consist of two basic electrical assemblies: the wound stator and the rotor assembly. The induction AC motor derives its name from currents flowing in the secondary member (rotor) that are induced by alternating currents flowing in the primary member (stator). The combined electromagnetic effects of the stator and rotor currents produce the force to create rotation.

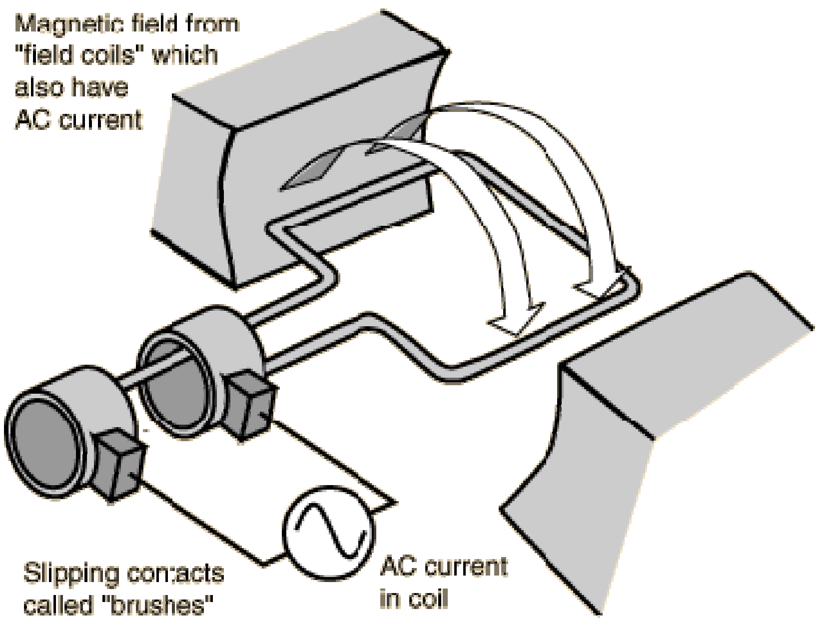


Figure 5.1: AC Motor [10]

AC motors typically feature rotors, which consist of a laminated, cylindrical iron core with slots for receiving the conductors. The most common type of rotor has cast-aluminum conductors and short-circuiting end rings. This AC motor, which is referred to as a squirrel cage, rotates when the moving magnetic field induces a current in the shorted conductors. The speed at which the AC motor magnetic field rotates is the synchronous speed of the AC motor and is determined by the number of poles in the stator and the frequency of the power supply:

$n_s = 120f/p$, where
n_s = synchronous speed,
f = frequency, and
p = the number of poles.

Synchronous speed is the absolute upper limit of AC motor speed. If the rotor turns exactly as fast as the rotating magnetic field, then the rotor conductors cut no lines of force, and torque is zero. When AC motors are running, the rotor always rotates slower than the magnetic field. The rotor speed is just slow enough to cause the proper amount of rotor current to flow, so that the resulting torque is sufficient to overcome windage and friction losses, and drive the load. The speed difference between the rotor and magnetic field, called slip, is normally referred to as a percentage of synchronous speed:

$s = 100\,(n_s - n_a)/n_s$, where
s = slip,
n_s = synchronous speed, and
n_a = actual speed.

Polyphase squirrel-cage AC motors are basically constant-speed machines, but some degree of flexibility in operating characteristics results from modifying the rotor slot design. These variations in AC motors produce changes in torque, current, and full-load speed. Evolution and standardization have resulted in four fundamental types of AC motors:

- Designs A and B

- Design C
- Design D
- Design F

Designs A and B: General-purpose AC motors with normal starting torques and currents, and low slip. Fractional-horsepower polyphase AC motors are generally design B. Because of the drooping characteristics of design B, a polyphase AC motor that produces the same maximum torque as a single-phase AC motor cannot attain the same speed-torque point for full-load speed as single-phase AC motors. Therefore, maximum torque must be higher so that full-load speeds are comparable.

Design C: High starting torque with normal starting current and low slip. AC motors are normally used where breakaway loads are high at starting, but which normally run at rated full load and are not subject to high overload demands after running speed has been reached.

Design D: High slip, AC motor starting torque, low starting current, and low full-load speed. Because of the high slip, speed can drop when fluctuating loads are encountered. This AC motor design is subdivided into several groups that vary according to slip or the shape of the speed-torque curve.

Design F: Low starting torque, low starting current, and low slip. These AC motors are built to obtain low locked-rotor current. Both locked-rotor and breakdown torque are low. Normally these AC motors are used where starting torque is low and where high overloads are not imposed after running speed is reached. [10]

Squirrel-cage AC motors are relatively inflexible with regard to speed and torque characteristics, but a special wound-rotor AC motor has controllable speed and torque. Application of wound-rotor AC motors is markedly different from squirrel-cage AC motors because of the accessibility of the rotor circuit. AC motor performance characteristics are obtained by inserting different values of resistance in the rotor circuit.

Wound-rotor AC motors are generally started with secondary resistance in the rotor circuit. The AC motor resistance is sequentially reduced to permit the motor to come up to speed. Thus, AC motors can develop substantial torque while limiting locked-rotor current. This secondary AC motor resistance can be designed for continuous service to dissipate heat produced by continuous operation at reduced speed, frequent acceleration, or acceleration with a large inertia load. External resistance gives AC motors a characteristic that results in a large drop in rpm for a fairly small change in load. Reduced AC motor speed is provided down to about 50% rated speed, but efficiency is low.

Two-winding AC motors have two separate windings that can be wound for any number of poles so that other speed ratios can be obtained. However, ratios greater than 4:1 are impractical because of AC motor size and weight. Single-phase multispeed AC motors are usually variable-torque design, but constant-torque and constant-horsepower AC motors are available.

Power output of multispeed AC motors can be proportioned to each different speed. These AC motors are designed with output horsepower capacity in accordance with one of the following load characteristics:

- Variable torque
- Constant torque
- Constant horsepower

Single-phase induction AC electric motors are commonly fractional-horsepower types, although single-phase integral-horsepower are available in the lower horsepower range. The most common fractional-horsepower single-phase AC motors are split-phase, capacitor-start, permanent split-capacitor, and shaded pole.

The AC motors come in multispeed types, but there is a practical limit to the number of speeds obtained. Two, three, and four-speed motors are available, and speed selection may be accomplished by consequent-pole or two-winding methods.

Single-phase AC electric motors run in the direction in which they are started; and they are started in a predetermined direction according to the electrical connections or mechanical setting of the starting means. General-purpose AC motors may be operated in either direction, but the standard AC motor rotation is anti-clockwise when facing the end opposite the drive shaft. AC motors can be reconnected to reverse the direction of rotation.

Universal AC motors operate with nearly equivalent performance on direct current or alternating current up to 60 Hz. However, in Australia mains frequency is 50 Hz, meaning this is well within the motors capabilities. AC motors differ from DC motors due to the winding ratios and thinner iron laminations. DC motors runs on AC, but with poor efficiency. Universal AC motors can operate on DC with essentially equivalent AC motor performance, but with poorer commutation and brush life than for an equivalent DC motor.

An important characteristic of universal AC motors is that it has the highest horsepower-per-kg ratio of any AC motor because it can operate at speeds many times higher than that of any other 50-Hz electric motor.

When operated without load, universal AC motors tend to run away, speed being limited only by windage, friction, and commutation. Therefore, large universal AC motors are nearly always connected directly to a load to limit speed. On portable tools such as electric saws, the load imposed by the gears, bearings, and cooling fan is sufficient to hold the no-load speed down to a safe value.

With a universal AC motor, speed control is simple, since electric motor speed is sensitive to both voltage and flux changes. With a rheostat or adjustable autotransformer, ac motor speed can be readily varied from top speed to zero.

Synchronous AC motors are inherently constant-speed electric motors and they operate in absolute synchronism with line frequency. As with squirrel-cage induction AC motors, speed is determined by the number of pairs of poles and is always a ratio of the line frequency.

Synchronous AC motors are made in sizes ranging from sub fractional self-excited units to large-horsepower, direct-current-excited AC motors for industrial drives. In the fractional-horsepower range, synchronous ac motors are used primarily where precise constant speed is required.

In large horsepower sizes applied to industrial loads, synchronous AC motors serve two important functions. First, AC motors provide highly efficient means of converting AC energy to mechanical power. Second, AC motors can operate at leading or unity power factor, thereby providing power-factor correction.

There are two major types of synchronous AC motors: non-excited and direct current excited electric motors.

Non-excited electric motors are made in reluctance and hysteresis designs. These electric motors employ a self-starting circuit and require no external excitation supply.

DC-excited electric motors come in sizes larger than 1 hp, and require direct current supplied through slip rings for excitation. Direct current may be supplied from a separate source or from a DC generator directly connected to the AC motor shaft.

Single-phase or polyphase synchronous electric motors can't start without being driven, or having their rotor connected in the form of a self-starting circuit. Since the electric motor field is rotating at a synchronous speed, the electric motor must be accelerated before it can pull into synchronism. Accelerating from zero speed requires slip until synchronism is reached. Therefore, separate starting means must be employed.

Although the DC-excited electric motor has a squirrel cage for starting, called a damper winding, the inherent low starting torque and the need for a DC power source requires a starting system that provides full electric motor protection while starting, applies DC field excitation at the proper time, removes field excitation at maximum torque, and protects the squirrel-cage winding against thermal damage under out-of-step conditions. [10]

5.1.2 Electric Three Phase AC Motors

Three phase motors have several distinct advantages over single-phase motors. There are also differences between how three phase motors work and how single-phase motors work. These are outline in the following paragraphs.

For higher-power applications where a polyphase electrical supply is available, the three-phase AC induction motor is used. The phase differences between the three phases of the polyphase electrical supply create a rotating electromagnetic field in the motor.

There are two types of rotors found in most three-phase motors. Most motors use the squirrel cage rotor discussed above. An alternate design, called the wound rotor, is used when variable speed is required. In this case, the rotor has the same number of poles as the stator and the windings are made of wire, connected to slip rings on the shaft. Carbon brushes connect the slip rings to an external controller such as a variable resistor that allows changing the motor's slip rate. In certain high-power variable speed wound-rotor drives, the slip-frequency energy is captured, rectified and returned to the power supply through an inverter.

Compared to squirrel cage rotors, wound rotor motors are expensive and require maintenance of the slip rings and brushes, but they were the standard form for variable speed control before the advent of compact power electronic devices. Transistorised inverters with variable frequency drive can now be used for speed control and wound rotor motors are becoming less common. Transistorized inverter drives also allow the more-efficient three-phase motors to be used when only single-phase mains current is available.

Several methods of starting a polyphase motor are used. Where the large inrush current and high starting torque can be permitted, the motor can be started across the line, by applying full line voltage to the terminals. Where it is necessary to limit the starting inrush current, reduced voltage starting using either series inductors or any number of other devices. A technique sometimes used is wye-delta starting, where the motor coils are initially connected in wye for acceleration of the load, then switched to delta when the load is up to speed. Transistorised drives can directly vary the applied voltage as required by the starting characteristics of the motor and load.

As in the single-phase motor, through electromagnetic induction, the rotating magnetic field induces a current in the conductors in the rotor, which in turn sets up a counterbalancing magnetic field that causes the rotor to turn in the direction the field is rotating. Also, the rotor must always rotate slower than the rotating magnetic field produced by the polyphase electrical supply; otherwise, no counterbalancing field will be produced in the rotor. This type of motor is excellent for use in traction cases such as locomotives, making it an excellent choice to consider for this project.

The speed of the AC motor is determined primarily by the frequency of the AC supply and the number of poles in the stator winding, according to the relation:

RPM = $2 * F * 60/p$, where

RPM = (Synchronous) Revolutions per minute,

F = AC power frequency, and

p = Number of poles, usually an even number but always a multiple of the number of phases.

The torque is a function of the amount of slip, or difference in rotation, between the rotor and stator fields. Standard motors have between 2-3% slip, special motors may have up to 7% slip, and a class of motors known as *torque motors* are rated to operate at 100% slip (0 RPM/full stall).

The torque is determined by the amount of slip, or difference in rotation, between the rotor and stator fields.

As with single-phase motors, another type of three-phase motor is the synchronous motor. If the rotor coils of a three-phase motor are fed a separate field current to create a continuous magnetic field the result is a called a synchronous motor because the rotor will rotate in synchronism with the rotating magnetic field produced by the polyphase electrical supply.

Nowadays, synchronous motors are frequently driven by transistorized variable-frequency drives. This greatly eases the problem of starting the massive rotor of a large synchronous motor. They may also be started as induction motors using a squirrel-cage winding that shares the common rotor: once the motor reaches synchronous speed, no current is induced in the squirrel-cage winding so it has little effect on the synchronous operation of the motor. Synchronous motors are occasionally used as traction motors.

Induction motors are the workhorses of industry and motors up to about 500 kW in output are produced in highly standardized frame sizes, making them nearly completely interchangeable between manufacturers. [9]

5.1.3 Electric DC Motors

The classic DC motor has a rotating armature in the form of an electromagnet with two poles. A rotary switch called a commutator reverses the direction of the electric current twice every cycle, to flow through the armature so that the poles of the electromagnet push and pull against the permanent magnets on the outside of the motor. As the poles of the armature electromagnet pass the poles of the permanent magnets, the commutator reverses the polarity of the armature electromagnet. During that instant of switching polarity, inertia keeps the classical motor going in the proper direction. This process is shown below, in figures 5.2 through 5.4. [9]

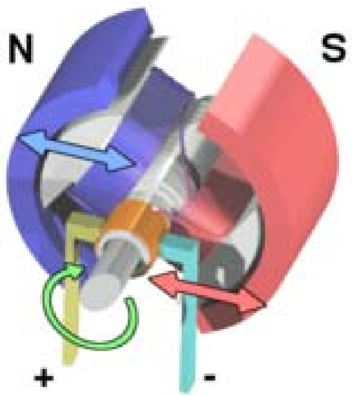


Figure 5.2: DC Motor Stage 1 [9]

When the coil is powered, a magnetic field is generated around the armature. The left side of the armature is pushed away from the left magnet and drawn toward the right, causing rotation.

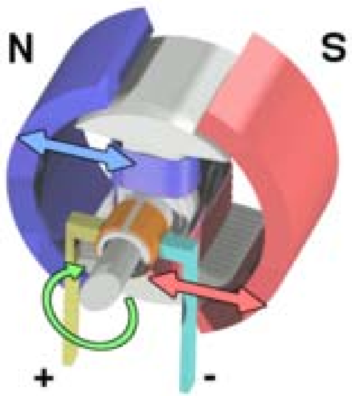


Figure 5.3: DC Motor Stage 2 [9]

The armature continues to rotate, as shown above in figure 5.3.

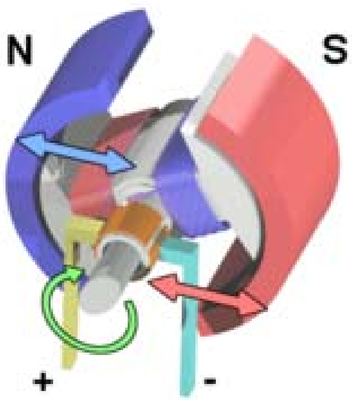


Figure 5.4: DC Motor Stage 3 [9]

When the armature becomes horizontally aligned, the commutator reverses the direction of current through the coil, reversing the magnetic field. The process then repeats itself.

DC motor speed generally depends on a combination of the voltage and current flowing in the motor coils and the motor load or braking torque. The speed of the motor is proportional to the voltage, and the torque is proportional to the current. Altering the voltage or current flow by using taps in the motor windings or by having a variable voltage supply typically controls the speed.

As this type of motor can develop quite high torque at low speed it is often used in traction applications such as locomotives.

However, there are a number of limitations in the classic design, many due to the need for brushes to rub against the commutator. The rubbing creates friction, and the higher the speed; the harder the brushes have to press to maintain good contact. Not only does this friction make the motor noisy, but it also creates an upper limit on the speed and causes the brushes eventually to wear out and to require replacement. The imperfect electric contact also causes electrical noise in the attached circuit. These problems vanish when you turn the motor inside out, putting the permanent magnets on the inside and the coils on the outside thus designing out the need for brushes in a brushless design. However such designs need electronic circuits to control the switching of the electromagnets.

Industrial applications use DC motors because the speed-torque relationship can be varied to almost any useful form -- for both DC motor and regeneration applications in either direction of rotation. Continuous operation of DC motors is commonly available over a speed range of 8:1. Infinite range for short durations or reduced load is also common.

DC motors are often applied where they momentarily deliver three or more times the rated torque. In emergency situations, DC motors can supply over five times rated torque without stalling.

DC motors feature a speed, which can be controlled smoothly down to zero, immediately followed by acceleration in the opposite direction -- without power circuit switching. DC motors respond quickly to changes in control signals due to the high ratio of torque to inertia.

Wound-field DC motors are usually classified by shunt-wound, series-wound, and compound-wound. In addition to these, permanent magnet and brush-less DC motors are also available, normally as fractional-horsepower DC motors. DC motors may be further classified for intermittent or continuous duty. Continuous-duty DC motors can run without an off period.

When choosing a DC motor for a given application, several factors need to be considered, such as speed range, speed variation with torque, and reversing. Peak torque and heating need to also be taken into account.

If a large speed range is required, the base speed of the motor must be proportionately lower and the motor size must be larger. If the speed range is much over 3:1, armature voltage control should be considered for at least part of the range. Very wide dynamic speed range can be obtained with armature voltage control. However, below about 60% of base speed, the motor should be de-rated or used for only short periods.

Applications requiring constant speed at all torque demands should use a shunt-wound DC motor. If speed-change with load must be minimized, a DC motor regulator, such as one employing feedback from a tachometer, must be used.

When the DC motor speed must decrease as the load increases, compound or series-wound DC motors may be used.

Reversing the motor affects power supply and control, and may affect the brush adjustment if the DC motor cannot be stopped for switching before reverse operation. In this case, compound and stabilizing DC motor windings should not be used, and a suitable armature-voltage control system should supply power to the DC motor.

Direct current motors are seldom used on drives that run continuously at one speed and load. Motor size needed may be determined by either the peak torque requirement or heating.

The load at which damaging commutation begins limits the peak torque that a DC motor delivers. DC motor brush and commutator damage depends on sparking severity and duration. Therefore, the peak torque depends on the duration and frequency of occurrence of the overload. The maximum current that the power supply can deliver often limits DC motor peak torque.

DC motor temperature is a function of ventilation and electrical/mechanical losses in the machine. Some DC motors feature losses, such as core, shunt-field, and brush-friction losses, which are independent of load, but vary with speed and excitation.

The best method to predict a given DC motor's operating temperature is to use thermal capability curves available from the DC motor manufacturer. vIf curves are not available, DC motor temperature can be estimated by the power-loss method. This method requires a total losses versus load curve or an efficiency curve.

For each portion of the duty cycle, power loss is obtained and multiplied by the duration of that portion of the cycle. The summation of these products divided by the total cycle time gives the DC motor's average power loss. The ratio of this value to the power loss at the motor rating is multiplied by the DC motor's rated temperature rise to give the approximate temperature rise of the DC motor when operated on that duty cycle. [9]

5.1.4 Hydraulic Motors

Hydraulic motors are powered by pressurized hydraulic fluid and transfer rotational kinetic energy to mechanical devices. Hydraulic motors, when powered by a mechanical source, can rotate in reverse direction and act as a pump. Operating specifications and features are the most important parameters to consider when searching for hydraulic motors.

The most important operating specification to consider when researching hydraulic motors is the motor type. Choices for motor type include axial piston, radial piston, internal gear, external gear, and vane. An axial piston motor uses an axially-mounted piston to generate mechanical energy. High pressure flow into the motor forces the piston to move in the chamber, generating output torque. A radial piston hydraulic motor uses pistons mounted radially about a central axis to generate energy. An alternate-form radial piston motor uses multiple interconnected pistons, usually in a star pattern, to generate energy. Oil supply enters the piston chambers, moving each individual piston and generating torque. Multiple pistons increase the displacement per revolution through the motor, increasing the output torque. An internal gear motor uses internal gears to produce mechanical energy. Pressurised fluid turns the internal gears, producing output torque. An external gear motor uses externally-mounted gears to produce mechanical energy. Pressurized fluid forces the external gears to turn, producing output torque. A vane motor uses a vane to generate mechanical energy. Pressurized fluid strikes the blades in the vane, causing it to rotate and produce output torque.

Additional operating specifications to consider for hydraulic motors include operating torque, operating pressure, operating speed, operating temperature, power, maximum fluid flow, maximum fluid viscosity, displacement per revolution, and motor weight. The operating torque is the torque the motor is capable of delivering. Operating torque depends directly on the pressure of the working fluid delivered to the motor. The operating pressure is the pressure of the working fluid delivered to the hydraulic motor. Working fluid is pressurized by an outside source before it is delivered to the motor. Working pressure affects operating torque, speed, flow and horsepower of the motor. The operating speed is the speed at which the hydraulic motors' moving parts rotate. Operating speed is expressed in revolutions per minute, or similar terms. The operating temperature is the fluid temperature range the motor can accommodate. Minimum and maximum operating temperatures are dependent on motor internal component materials, and can vary greatly between products. The power the motor is capable of delivering is dependent on the pressure and flow of the fluid through the motor. The maximum volumetric flow through the motor is expressed in terms of gallons per minute, or similar units. The maximum fluid viscosity the motor can accommodate is a measure of the fluid's resistance to shear, and is measured in centipoise. Centipoise is a common metric unit of dynamic viscosity equal to 0.01 poise or 1 millipascal second. The dynamic viscosity of water at 20 degrees C is about 1 centipoise. The correct unit is cP, but cPs and cPo are sometimes used. The fluid volume displaced per revolution of the motor is measured in cubic centimetres (cc) per revolution, or similar units. The weight of the motor is measured in kilograms or similar units.

Additional features to consider when searching for hydraulic motors include mounting in any position, rated for continuous duty, and quiet operation. [16]

However, although hydraulic loading is a possibility for the proposed system, based on the information above a hydraulic motor will not be considered. The inherent safety risks associated with hydraulic motors, as well as the suitability of the motors to larger loading applications, is enough to discount the hydraulic motor from being a feasible choice for this project.

5.1.5 Motor Decision Matrix

Using the same process outlined in section 4.6, a decision matrix was constructed for the purpose of choosing an electric motor to power the design.

The list of criteria chosen for the motor to satisfy, and the weightings assigned to these criteria, are shown below in table 5.1.

Criterion	Weight
Initial Cost	3
Ongoing Cost	3
Availability	3
Low Maintenance	2
Installation	1
Noise	1
Ability to deliver Power and Torque	3
Size	1
Reliability	3

Table 5.1: Criterion and Weighting for Electric Motor Decision Matrix

As shown in table 5.1, the criteria considered most important for the choice of an electric motor include initial cost, ongoing cost, availability, reliability, and the ability of the motor to deliver the power and torque required. Low maintenance was considered of medium importance, with ease of installation, noise, and size all considered of low importance.

The scores for each of the three alternatives are shown below in table 5.2. Again, the scores range from 1 as the lowest to 5 as the highest possible.

Criterion	Weight	AC Single Phase	AC Three Phase	DC
Initial Cost	3	4	4	3
Ongoing Cost	3	4	4	4
Availability	3	5	5	5
Low Maintenance	2	4	4	1
Installation	1	4	4	4
Noise	1	4	4	1
Ability to deliver Power and Torque	3	2	4	5
Size	1	3	3	3
Reliability	3	3	3	3
Total Rating		**73**	**79**	**70**

Table 5.2: Decision Matrix for Electric Motor

5.1.6 Motor Decision Matrix Scoring Explanation

The initial cost of the AC single-phase motor involves only the purchase of the motor, as it will run off mains power. The three-phase AC motor requires a three-phase power source that can cost up to $20000 to install. However, the facility that will house the testing system should have three-phase power, as does the majority of the industrial world. Therefore, it is reasonable to assume a three-phase supply should be available, meaning both the single-phase motor and three-phase motor achieve the same score for the initial cost. The DC motor will require a DC power source, which would not be readily available in most places. Therefore, the initial cost of installing a DC motor would also include the cost of installing a reliable DC power supply. For this reason, the DC motor receives a slightly lower score than the two AC motors.

Ongoing costs such as maintenance and operating costs are similar for both the single phase and three phase AC motors. This is due to the overall similarity of the two motors. The ongoing costs are also comparable for a DC motor. The fact the DC motor will require replacement parts due to the nature of the motor is taken care of with the maintenance criteria.

The availability of the motors is important for this project, but as many companies specifically stock many types of electric motors in a wide range of horsepower ratings, the availability of all three motors is quite high.

Both the single phase and three phase AC motors will require relatively low maintenance. The DC motor will require brush replacements because of the nature of the motor. The brushes will wear out over time because they rub against the commutator. For this reason, the DC motor scored extremely low in this category compared to the single and three phase AC motors.

Ignoring the installation of the power supply, all three motors will require approximately the same amount of expertise to install. Again, the score given to the motors in this category was the same for all motors.

Whereas the single-phase and three-phase AC motor will have comparable noise levels, the DC motor will be significantly louder. This, again, is because of the nature of the motor. The friction created by the brushes rubbing against the commutator is responsible for most of the noise created by the DC motor.

When compared with the three-phase AC motor, the single phase will have to work much harder to deliver the same amount of power and torque, which will shorten the life of the motor. However, the DC motor is able to develop high torque at low speed, which is why it is primarily used in traction applications. They can also momentarily deliver three or more times their rated torque, and in emergency situations can deliver up to five times their rated torque without stalling. For the reasons listed above, the single phase motor was given a low score of 1, the three phase motor was scored at 4, and the DC motor received the highest possible score of 5.

While the size of the motor must be taken into consideration, the three types of motor chosen to be in the decision matrix are all of comparable size. However, at the concept stage, other variations such as the hydraulic motor were considered but were discounted because of the bulkiness and awkwardness of the set-up. For this reason only the category of size was left in the decision matrix.

While DC motors are generally more reliable than the AC motors, all the motors have been considered to have the same reliability characteristics because of the possibility of the DC motor failing due to the fact the brushes are continually wearing against the commutator. Also, the faster the motor runs, the harder the brushes must press to maintain contact. This will lead to the motor wearing out more quickly.

5.2 Rails and Wheels

The rails design chosen in Chapter 4 requires the carriage to run on a pair of rails parallel to the test track. There are many different variations of rail and wheel to consider when choosing the final components for this design.

5.2.1 Rails

There are several good configurations of rails that will provide the carriage with the direction it needs. The two main ways of assembling the rails involve either mounting the rails on the test track or keeping it completely separate as shown in below in figure 5.5.

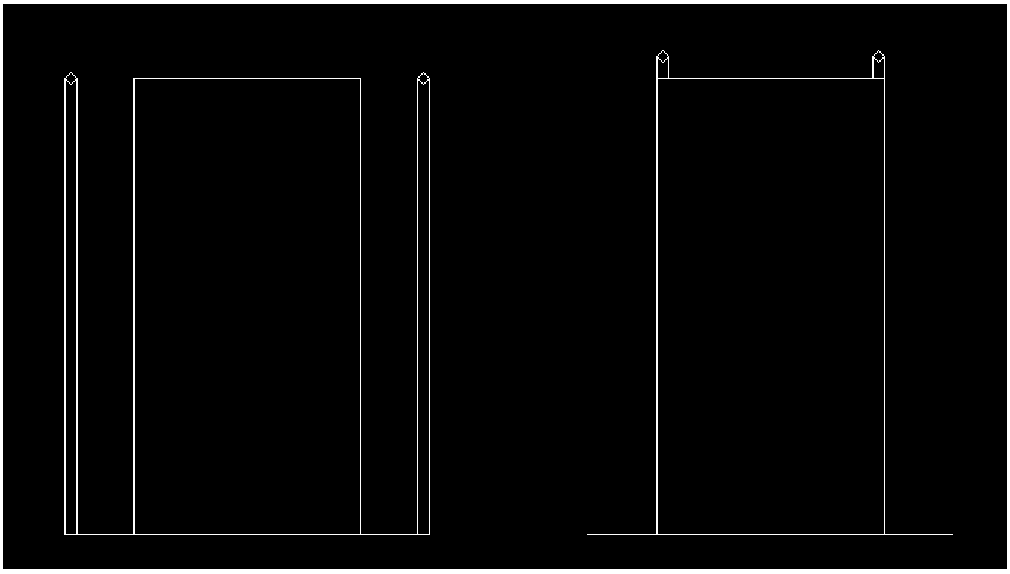


Figure 5.5: Rail Mounting Possibilities

The main difference between the two possibilities shown above is the extra material needed and unnecessary bulkiness of having the rails mounted separate from the test track. Because the design on the left offers no real advantages over mounting the rails on the actual test track, it has been decided that the design on the right is the best option for the testing facility.

Obviously the rails will be fixed to the test track, but they cannot be fixed to the test bed. So, to allow the test bed to be removed easily for examination of the test pavement, the rails will need to span the gap created by the test bed.

There are three main types of rail that could be used in this situation. The first is a regular circular section bar and the second is a normal RHS bar. However, the RHS can be orientated so it has a flat upper surface, or can be rotated through 45° to create a pointed upper surface.

5.2.2 Wheels

There are caster wheels available to suit the rails described in section 5.2.1. The three types of wheel to suit their respective rail are shown below.

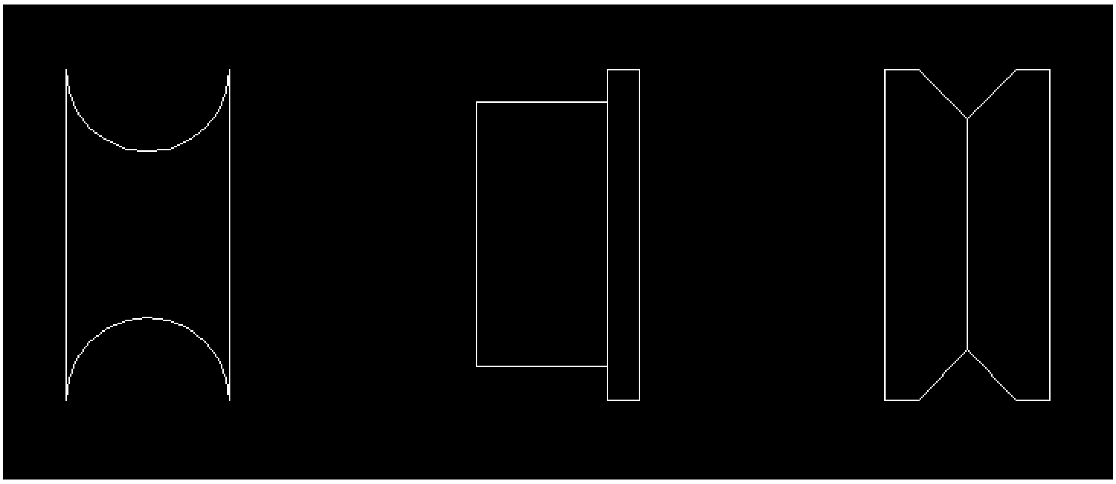


Figure 5.6: Wheel Types to suit Rails

The wheel on the left of figure 5.6 is obviously the suit the rail with a circular cross section. The wheel in the centre is known as a single flange wheel and is suited to the flat RHS rail. The last wheel is a V-groove wheel and is suited to the RHS rail rotated 45°.

Many industries are discovering that flanged wheels running on tracks give them an affordable alternative to running the loads across the ground. This permits higher capacities, easier rolling, floor protection and controlled flow all at the same time. Most flanged wheels are cast iron and furnished with either plain bore, straight roller bearings, or precision tapered roller bearings.

Running V-grooved wheels on inverted angle iron track is a popular and low-cost method of taking loads off the floor. A relief groove in the V assures proper weight distribution and makes the wheel and track self-cleaning, while a partial flat-tread permits rolling the wheels directly on floors as well. Cast iron and forged steel wheels are available with plain bore or straight roller bearings while some models also offer precision tapered roller bearings for optimum capacity and performance. However, proper adjustment of tapered bearings precludes side-to-side wheel play. [18]

All of these wheels are readily available at any hardware shop, but, as listed above, there are some advantages the V-groove wheel offers that the other alternatives cannot match. The V-groove wheel is the only wheel to effectively allow the carriage to roll along both the rail and also flat ground. This is useful when the carriage must be moved to a different location for a variety of purposes such as maintenance or relocation of the test track.

The circular section steel tends to be slightly more expensive than RHS steel, so for this reason it would appear that the V-groove wheel would be the best choice for this design. However, a decision matrix was constructed to make sure the correct choice is made.

5.2.3 Rails and Wheels Decision Matrix

The list of criteria chosen for the rails and wheels to satisfy, and the weightings assigned to these criteria, are shown below in table 5.3.

Criterion	**Weight**
Cost	3
Availability	3
Performance	2
Life	2

Table 5.3: Criterion and Weighting for Rails and Wheels Decision Matrix

As shown in table 5.3, the criteria considered most important for the choice of the rails and wheels were the cost and availability. The performance and the life of the wheels were considered of slightly less importance.

The scores for each of the three alternatives are shown below in table 5.4. Again, the scores range from 1 as the lowest to 5 as the highest possible.

Criterion	**Weight**	**V-Groove**	**Circular Groove**	**Single Flange**
Cost	3	4	3	5
Availability	3	4	4	4
Performance	2	5	5	3
Life	2	4	4	4
Total Rating		**42**	**39**	**41**

Table 5.4: Decision Matrix for Rails and Wheels

5.2.4 Rails and Wheels Decision Matrix Scoring Explanation

Further research was conducted into the prices of the three different types of wheels. The circular grooved wheel was slightly more expensive than the other two, giving it an average score of 3. The single flange wheel represented the best value, receiving a perfect score of 5. The V-groove wheel was between these two choices with a score of 4.

All three types of caster wheel are available at most hardware shops. However, they are not available at smaller hardware shops or must be ordered in. For this reason, all three alternatives received the same score, but the score was only a 4 rather than a perfect 5.

The performance of the V-groove wheel and the circular groove wheel are comparable. Both offer excellent precision as well as a very small possibility of the wheel mounting the track, resulting in a derailment. The single flange wheel, however, does not offer the same precision as the V-groove and circular groove, and for this reason scored lower in this category.

The life of the wheels are important as they are an important part of the carriage and need to withstand significant fatigue loading. Most of the wheels available are manufactured from cast iron, which has a high fatigue limit. The wheels can be manufactured from other materials at the consumer’s request, but these materials, such as timber and plastic, generally have lower fatigue limits than cast iron. Therefore, all three alternatives received a score of 4 for this category.

It can now be seen from table 5.4 that a V-groove wheel, coupled with an inverted angle iron track or a RHS track rotated through 45° would be the best choice for this pavement testing facility.

5.3 Lifting Mechanism

Another method to consider in the design of this pavement testing facility is the lifting mechanism used. The nature of the carriage with the electric motor on top driving the loaded wheel through either a belt or chain drive reduces the methods of lifting the loaded wheel above the test track.

5.3.1 Pivot Concept

One method is shown below in figure 5.7. The entire motor and wheel assembly is fixed as a rigid body and pivoted about the hinge point O. Removing the elbow bend from the wheel support and replacing with just the upright could simplify this design further. However, the design has been sketched this way so comparisons between all suggested designs can be reached more easily.

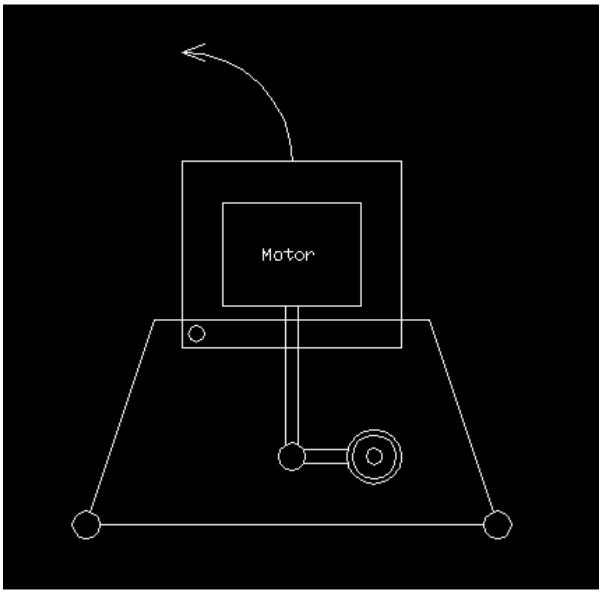


Figure 5.7: Pivot Concept for Lifting Mechanism

However, this design would require a large moment be produced every time the carriage reaches the end of a test run. Providing the necessary force to pivot the motor and wheel assembly would provide an inconvenience and is quite a simple design. With the technology available today, there are more obtainable solutions to this problem.

5.3.2 Actuator

An example of using the better technology available is through the possible use of a linear actuator. Linear actuators are easily interfaced with a computer and can provide force in both directions. This means that as well as being an effective lifting mechanism; the linear actuator could also provide the load needed for the accelerated pavement testing. An example of the design is shown below in figure 5.8.

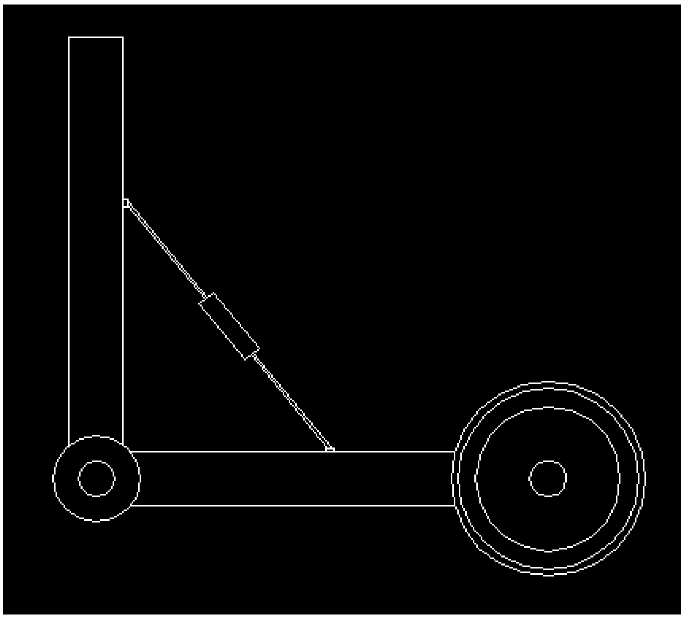


Figure 5.8: Linear Actuator Concept for Lifting Mechanism

Installing the linear actuator in the position shown above in figure 5.8 will allow the drive wheel to pivot easily about the hinge point in the lower left corner of the drawing. Also, the drive mechanism can easily run down the length of the upright bar, and then transfer power out along the horizontal bar to the drive wheel. As mentioned earlier, a belt or chain drive would be most effective at completing this task.

The orientation of the actuator can also vary slightly from the position shown. It would be just as effective to attach one end of the actuator to the bottom side of the plate that holds the motor, and attach the other end to the horizontal bar near the drive wheel. This configuration is shown below in figure 5.9.

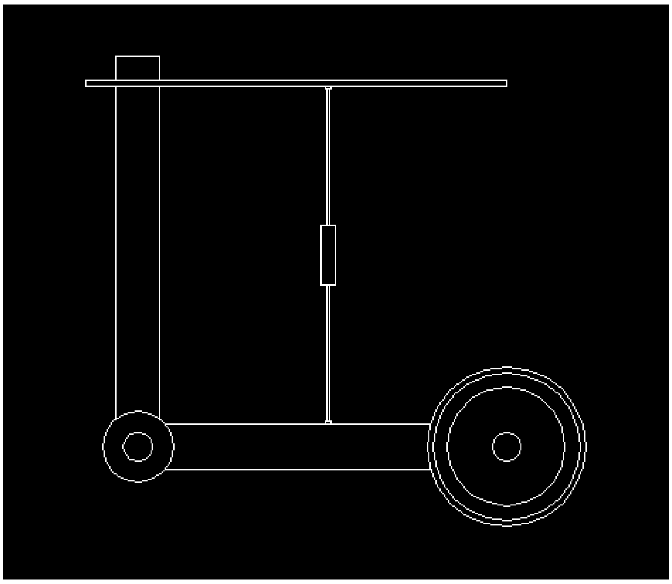


Figure 5.9: Different Configuration of Linear Actuator

The linear actuator has been chosen as both the lifting mechanism and loading mechanism for this project. The ease associated with interfacing the actuator with a computer was a great advantage. Also, applying a large force to produce the moment required to pivot the motor and wheel assembly every cycle proved a large disadvantage for the pivot concept.

5.4 Return System

As the loaded wheel can only drive the carriage in one direction, a system is needed to return the carriage to the start point for its next cycle. Various methods were considered, but the best ideas all involved the use of a second motor. Although this would make the design more expensive, it was the most effective way of returning the carriage to the start point.

5.4.1 Second Motor on Carriage

The first idea considered was to mount the second motor on the carriage itself. The mounting position was not considered very important, but the most likely place to install it was either above the primary motor on a separate shelf, or in the large space underneath the primary motor. In this case, the purpose of the secondary motor was to power any number of the caster wheels in contact with the steel rails. At a predetermined point, the lifting mechanism would lift the drive wheel clear of the test track, and the secondary motor would engage. This would allow the carriage to drive back along the rails to its start point. The process would reverse, with the secondary motor switching off and the lifting mechanism lowering the wheel back onto the track.

This process would require a great deal of control, as well as a large amount of experimentation to determine the required switch on and switch off points. There are several devices that can control the electric motors when the carriage is in a specified position such as light curtains and limit switches. These are discussed in the next section.

Ultimately, mounting the second motor on the carriage proved an unsatisfactory concept due to the increase weight of the carriage. By placing the second motor on the carriage, the power, and the cost, of both the primary and secondary motor would need to be increased. This increase is required for the carriage to obtain the same velocity in the same distance with an increased mass. It was concluded that a secondary motor driving a separate return system would be the most feasible idea.

5.4.2 Belt Drive Return System

The first system to be considered with a separate secondary motor was a belt drive return system. This system is shown below in figure 5.10.

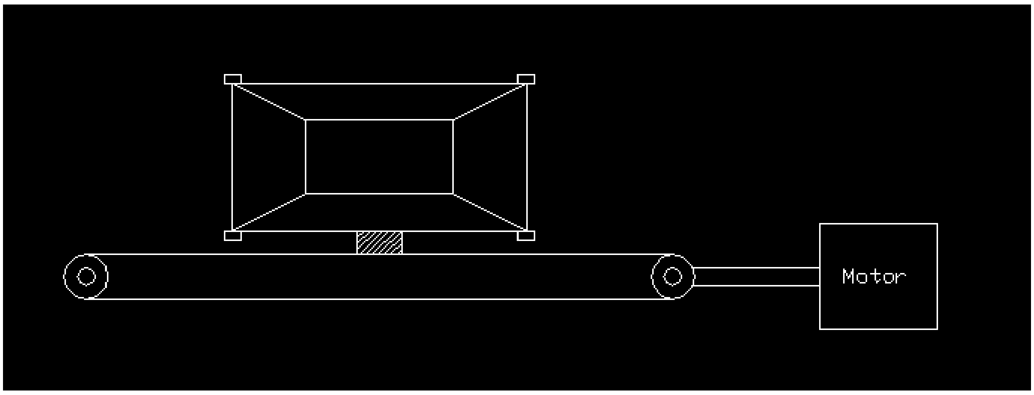


Figure 5.10: Belt Drive Return System

As shown in figure 5.10, the secondary motor will power a belt drive system that is slightly longer than the test track. The carriage will be fixed the belt by way of a metal plate. During its experimental run, the secondary motor will allow the belt drive to freely rotate with the carriage. Once the carriage reaches a predetermined point, the lifting mechanism will lift the drive wheel clear of the track, and the secondary motor will engage. Once the carriage is rolled back into position, the secondary motor will shut off and the lifting mechanism will lower the drive wheel back into place for the next cycle to begin.

However, the overall bulkiness of the belt-drive system and the inherent dangers presented by it are definite disadvantages. Also, with such a large belt drive, purchasing costs would be quite high, as would maintenance time and costs.

So, while presenting a feasible solution to the problem of a return system, the belt drive does have some disadvantages. These are remedied by replacing the belt drive with a simple steel cable.

5.4.3 Cable Return System

While in practice the same as the belt drive return system, the cable return system does offer some advantages over that system. The basic principle of the system is shown below in figure 5.11.

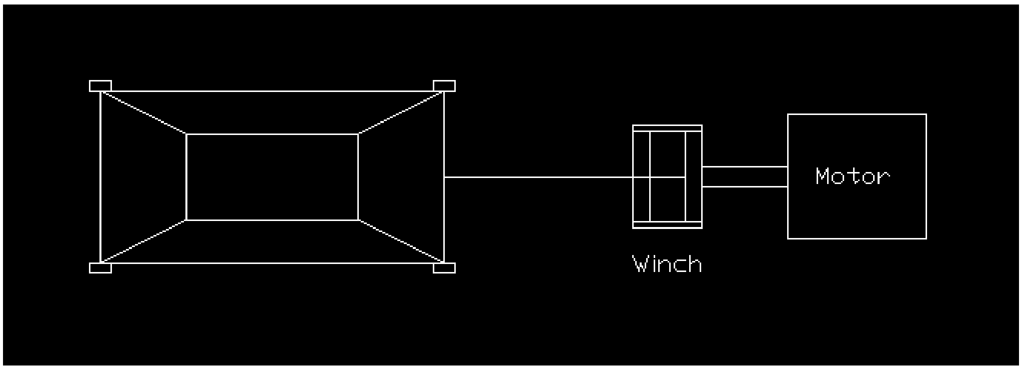


Figure 5.11: Cable Return System

As shown in figure 5.11, the secondary motor will power a winch. A steel cable is attached to the carriage at one end, and attached to the winch at the opposite end. While the carriage is moving along the track during an experimental run, the secondary motor will allow the winch to spin freely, allowing the cable to roll out. Upon reaching a predetermined point, the lifting mechanism will lift the drive wheel clear of the test track and the secondary motor will engage. This will force the winch to retract the cable, pulling the carriage back along the length of the test track via the rails. Once the second predetermined point is reached, the secondary motor will stop and the lifting mechanism will lower the wheel back onto the test track.

The cable return system is much safer than the belt drive system because it is a much more compact design. It also involves commonly available components such as the winch, and can perform the same functions as the belt drive return system. The maintenance of this system would also be simpler, and the system is more reliable as there is less that can go wrong.

For the reasons listed above, the cable return system has been chosen as the return system for the final design. The cable return system does, however, require a fairly precise degree of control over the carriage. Two methods of this are investigated below.

5.5 Methods of Control

The two methods of control investigated for this project were the use of light screens and limit switches. This control is needed to ensure the lifting mechanism and secondary motor both engage and disengage at the appropriate times.

5.5.1 Light Screens

A light screen, or light curtain, is normally used on assembly lines to ensure the safety of machinery operators. A light screen consists of two uprights with a grid between them. A typical light screen is shown below in figure 5.12.

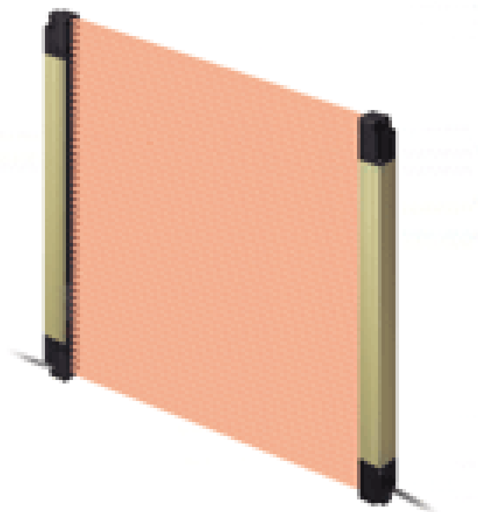


Figure 5.12: Light Screen [1]

The grid is usually made of infrared beams that cross from one side to the other. If one of the beams is broken, the light screen can cut power to any device it is connected to, such as a conveyor belt or a lathe.

As stated earlier, the main function of a light screen is to prevent injury to machinery operators. Many of the newest light screens, when broken, can detect whether the item is human or mechanical. This is useful when on a production line where the screen will be broken by many objects on the conveyor belt, but the power will not be cut unless a human passes through the screen. However, the use of a light screen for this project would involve the screen activating the lifting mechanism and the secondary motor when the carriage passes through it. Some reprogramming would be necessary so that the screen recognises the carriage as the device it is monitoring.

While a light screen is not ideal for monitoring the position of the test carriage, it is perfect as a safety device for this project. If the carriage and test track are isolated in an enclosure with a single entry point, a light screen could be setup to monitor for human entry. If the screen detects a human entering while testing is in progress, the screen could cut power to the primary motor to avoid any injury to the person. [1]

5.5.2 Limit Switches

A limit switch is an electro-mechanical device that consists of an actuator mechanically linked to a set of contacts. When an object comes into contact with the actuator, the device operates the contacts to make or break an electrical connection. Limit switches are used in a variety of applications and environments because of their ruggedness, simple visible operation, ease of installation and reliability of operation.

The simplest limit switch is just a switch in series with a motor, as shown in figure 5.13. The switch has closed contacts, meaning the switch is on when the button is not pressed. When the switch button is pressed, the switch turns off. The switch is positioned so that when the mechanism reaches its end of travel, it pushes the switch to the off position.

The problem is that when the switch is off, the motor cannot be reversed. The mechanism must be reset by hand. [28]

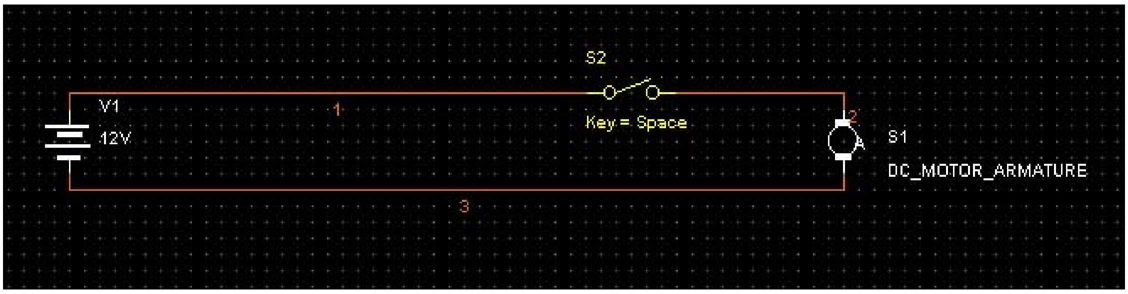


Figure 5.13: Simple Limit Switch Diagram [28]

Reversing the direction of current flow can reverse motors like the Lego motors. With the circuit in figure 5.13, reversing the power does nothing because the switch is open. This problem can be fixed by adding a diode. A diode is an electronic component that allows current to flow in only one direction. The symbol for a diode is an arrow with a perpendicular line across the tip. Current flows in the direction of the arrow. Figure 5.14 shows a diode in parallel with the switch. When the switch is closed, the diode has no effect. When the switch is open, the motor can run in one direction but not the other. [28]

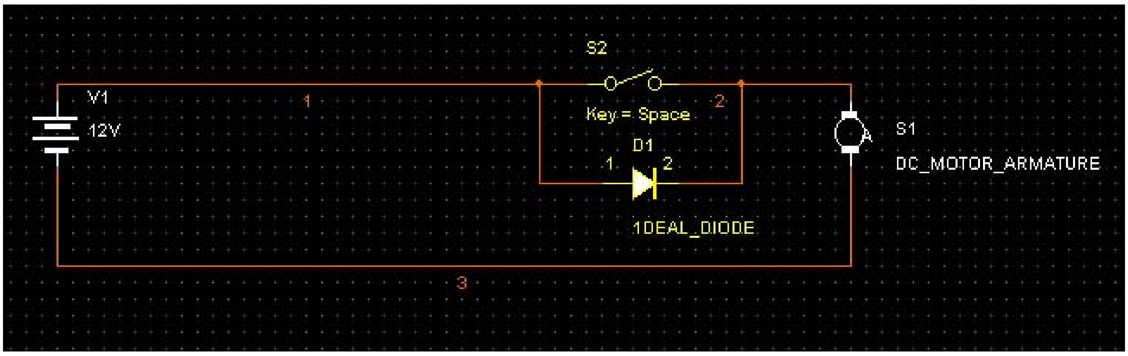


Figure 5.14: Limit Switch Diagram with Diode [28]

To stop a mechanism at both ends of travel, add a second switch and diode, as shown in figure 5.15. Place a switch at each end of the mechanism, with diode direction chosen so the motor may be reversed when the switch stops it. [28]

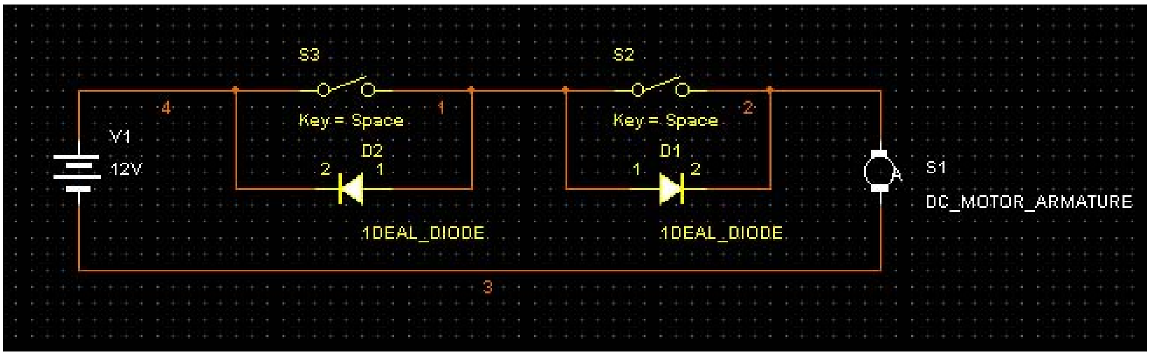


Figure 5.15: Limit Switch Diagram with Two Switches and Diodes [28]

The position of a mechanism can be set fairly well by simply running the motor for a set amount of time. If a mechanism is cycled back and forth many times, small errors from each cycle will accumulate to create an error that grows large. Error can be prevented from accumulating by using a limit switch to reset the error on each cycle. Instead of moving the mechanism back for a set amount of time, apply reverse power for a longer time, and let it be stopped by a limit switch. This way it will always return to the same starting position. [28]

5.5.3 Control Selection

From the information given above, it has been decided to place a limit switch towards the start and end of the test track. The switch at the end of the track will engage the lifting mechanism and the secondary motor. Once the carriage triggers the limit switch near the start of the test track, the power will be cut to the secondary motor and the lifting mechanism will lower the wheel back onto the track.

The exact positions of the limit switches will only be found through experimentation so that the optimum cycle time for the system can be found.

Chapter 6

Component Specification

The types of components needed to build this project have all been selected. The next step is to specify sizes and powers for all components. This involves, for example, calculating the power and torque required from the motor or the reduction ratio of the gearbox. Once the calculations are complete, the correct parts can be selected from any number of catalogues available.

6.1 Wheel Specification

Before the power and torque of the motor can be calculated, the driving wheel must be specified. As stated in section 3.6.1, the Sydney Pavement Testing Facility uses a 220mm steel rim pneumatic wheel. However, the same rig only reaches a maximum speed of 7.2km/h. As the design for this project needs a top speed of approximately 10km/h, a slightly larger wheel must be specified. This is because the load bearing capacity of the wheel decreases at higher speeds, and the ability of a standard 220mm diameter steel rim to handle a load of 100kg at 10km/h would be marginal at best.

Using an online catalogue obtained from Fallshaw Wheels and Castors, a steel centred wheel measuring 250mm by 70mm has been selected. The online catalogue used to specify this wheel is included in this project as Appendix B. This particular wheel has a load bearing capacity of 140kg at 8km/h and 110 at 15km/h. The model of this wheel is 300X4DMD-SB20.

The wheel has deep groove ball bearings with an expected life of 5 years. It is guaranteed wobble-free for 3 years, and has a 2mm thick steel rim. [13]

6.2 Belt Drive Reduction Ratio

Now the driving wheel has been specified, the belt drive reduction ratio can be calculated. The following assumptions have been made:

- Outside Diameter of Wheel = 250mm
- Velocity across Test Section = 10km/h
- A 4-pole motor will be chosen because it is a common choice as a three-phase electric motor, and therefore will rotate at 1500 revolutions per minute (rpm)

The circumference of the wheel was the first value to be calculated, as shown below.

$$C = \pi D$$
$$C = \pi \times 250mm$$
$$C = 785.4mm$$
$$C = 0.7854m$$

The velocity of the loaded wheel was then converted to m/s.

$$V = 10km/h$$
$$V = (10km/h) \times (1000m) \div (3600s)$$
$$V = 2.778m/s$$

The number of revolutions per minute (R_w) of the loaded wheel was then calculated.

$$R_s = V \div D$$
$$R_s = (2.778m/s) \div (0.7854m)$$
$$R_s = 3.537rps$$

$$R_w = R_s \times 60$$
$$R_w = 212.2rpm$$

The reduction ratio required is simply the ratio of the rpm of the motor to the rpm of the loaded wheel. This simple calculation is shown below.

$$R_r = R_m \div R_w$$
$$R_r = 1500 \div 212.2$$
$$R_r = 7.07$$

As an exact reduction ratio of 7.07 is difficult to obtain, a reduction ratio of 7 has been chosen. This will result in a new top speed of 10.1km/h, which is extremely close to the initial value of 10km/h. In fact, the new top speed will slightly improve the cycle time for the device.

6.3 Electric Motor Specification

Once both the wheel and belt drive reduction ratio have been specified, the electric motor can now be specified. However, the torque and power of the motor have to be calculated. The following assumptions have been made:

- The coefficient of friction between the wheel and the surface of the track, μ, is 0.8
- The mass of the carriage is approximately 50kg
- The run-on and run-off lengths of the test track are 1.25m

The maximum acceleration possible is found as follows:

$$F = \mu F_N$$
$$F = 0.8 \times 50kg \times 9.81m/s^2$$
$$F = 392.4N$$

$$F = ma_{\max}$$
$$392.4N = 50kg \times a_{\max}$$
$$a_{\max} = 7.848m/s^2$$

The acceleration the carriage requires to achieve a velocity of 10km/h in a distance of 1.25m is:

$$V_f{}^2 = V_i^2 + 2ad$$
$$2.778^2 = 0^2 + 2 \times a \times 1.25$$
$$a = 3.087m/s^2$$

The required torque at the wheel is then calculated to be:

$$F = ma$$
$$F = 50 \times 3.087$$
$$F = 154.35N$$

$$T_{wheel} = Fd$$
$$T_{wheel} = 154.35N \times (0.250m \div 2)$$
$$T_{wheel} = 19.3Nm$$

Now, the required torque of the motor is calculated using the belt drive reduction ratio. The required torque is found to be:

$$T_{motor} = T_{wheel} \div R_r$$
$$T_{motor} = 19.3Nm \div 7$$
$$T_{motor} = 2.76Nm$$

So, the motor must be able to deliver at least 2.76Nm of torque.

Using an online catalogue obtained from Baldor Pty Ltd, a 0.75kW High Efficient Metric Motor has been chosen for this design. A section of the catalogue used has been included in this project as Appendix C. This particular motor is a three-phase, 4 pole, AC motor that can apply a torque of 5Nm at full load and will be relatively small and inexpensive. The model number of this motor is MM3546.

The efficiency of this motor has been calculated to be approximately 82.1%. If a larger expenditure were allowed, a better choice would be the 0.75kW Super-E Premium Efficient Metric Motor. This motor can only develop 4.9Nm of torque at full load, but has an efficiency of 84.3%. This would result in lower ongoing costs because of lower electricity bills due to the increased efficiency of the motor. [3]

The secondary motor to drive the return system should also be the 0.75kW High Efficient Metric Motor specified above. This is to ensure that the return system can perform to the level that is needed to return the carriage in the shortest possible time.

6.4 Rail Specification

The rails for this design will need to span the gap of the test bed without deflecting too far. This is the main factor used to decide the size of the rails as the same size V-groove wheel will run on any square hollow section (SHS) rail. Choosing the smallest rail that will not deflect excessively will also minimise the cost of the rail.

The test bed measures 1200mm long and for the purposes of this project, a deflection of 1mm will be considered excessive.

Using an online catalogue obtained from One Steel, values of self-weight are given for different sizes of beams. This catalogue is included in this project as Appendix D. This self-weight will be treated as a uniformly distributed load spanning the 1200mm gap created by the test bed.

The formula for deflection of a beam under a uniformly distributed load is $\delta = {5(wL^4)}/{384(EI)}$. [7]. The smallest, and most inexpensive, rail offered in the Onesteel catalogue is the 20mm by 20mm. The steel section has a mass per metre of 0.873kg/m and a Z_n value of $0.474 \times 10^3 mm^3$. The deflection of this beam is calculated as follows:

$$Z_n = {I_n}/{y}$$

$$y = 20mm \times \sin 45^\circ$$
$$y = 14.14mm$$

$$0.474 \times 10^3 = {I_n}/{14.14}$$
$$I_n = 6.702 \times 10^3 mm^4$$

$$\delta = {5(wL^4)}/{384(EI)}$$
$$\delta = {5 \times 0.873 \times 9.81 \div 1000 \times 1200^4}/{384 \times 200000 \times 6.702 \times 10^3}$$
$$\delta = 0.1725mm$$

So the 20mm by 20mm rail deflects by 0.1725mm under its own self-weight. However, under the worst-case scenario when the 50kg carriage is in the centre of the gap, the carriage can be assumed to be a point load of 25kg on each rail. The formula for deflection in this case is $\delta = {(PL^3)}/{48(EI)}$. [7]. The deflection of the rail in this case is calculated below.

$$\delta = {(PL^3)} \big/ {48(EI)}$$

$$\delta = {25 \times 9.81 \times 1200^3} \big/ {48 \times 200000 \times 6.702 \times 10^3}$$

$$\delta = 6.5868mm$$

The deflection of the rail due to the carriage moving over it is about 6.5868mm. The total deflection of the rail can be assumed to be the sum of these two deflections, giving 6.7593mm. This is an unacceptable deflection, so a larger rail size must be estimated.

Using the same process as above to find the deflections, and assuming a new rail size of 50mm by 50mm with a thickness of 2.5mm, the new deflections are given below.

$\delta_{sw} = 0.0265mm$ and $\delta_c = 0.2453mm$

This gives a total deflection of 0.2718mm. While this value is quite lower than 1mm, 50mm by 50mm steel is very common and should be quite cheap. If a different, less common, size was chosen, while smaller than the 50mm by 50mm, it may be more expensive due to its relative obscurity.

Therefore, the rail size chosen for this project is 50mm by 50mm with a thickness of 2.5mm. This steel section should be readily available from One Steel. [21]

6.5 Belt Drive Specification

It has been decided to use V-belts as the drive system for the design. V-belts are normally used to drive industrial machinery and multiple V-belts are often used as the drive system when a single V-belt has insufficient capacity. The belt drive layout for the proposed pavement testing system is shown below in figure 6.1.

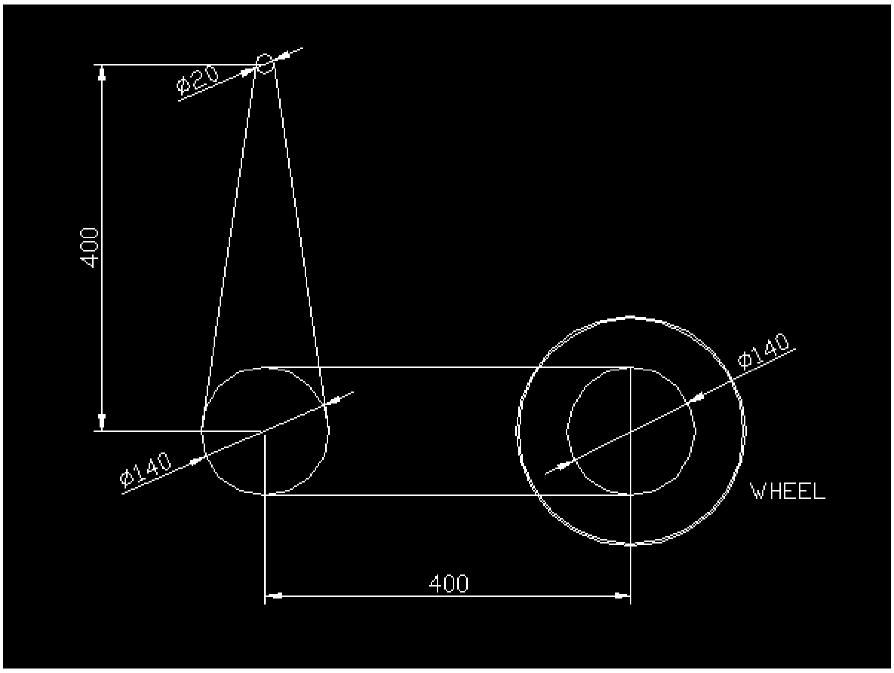


Figure 6.1: Belt Drive Diagram

The requirements of the belt drive have been calculated below to determine the specifications needed.

Step 1

$$T_{peak} = T \times Factor$$
$$T_{peak} = 2.76 \times 1.5$$
$$T_{peak} = 4.14Nm$$

Step 2

From figure 6.2 below, it can be seen with a torque of 4.14Nm and a rotational speed of 1500rpm, a 3mm GT2 timing belt is required.

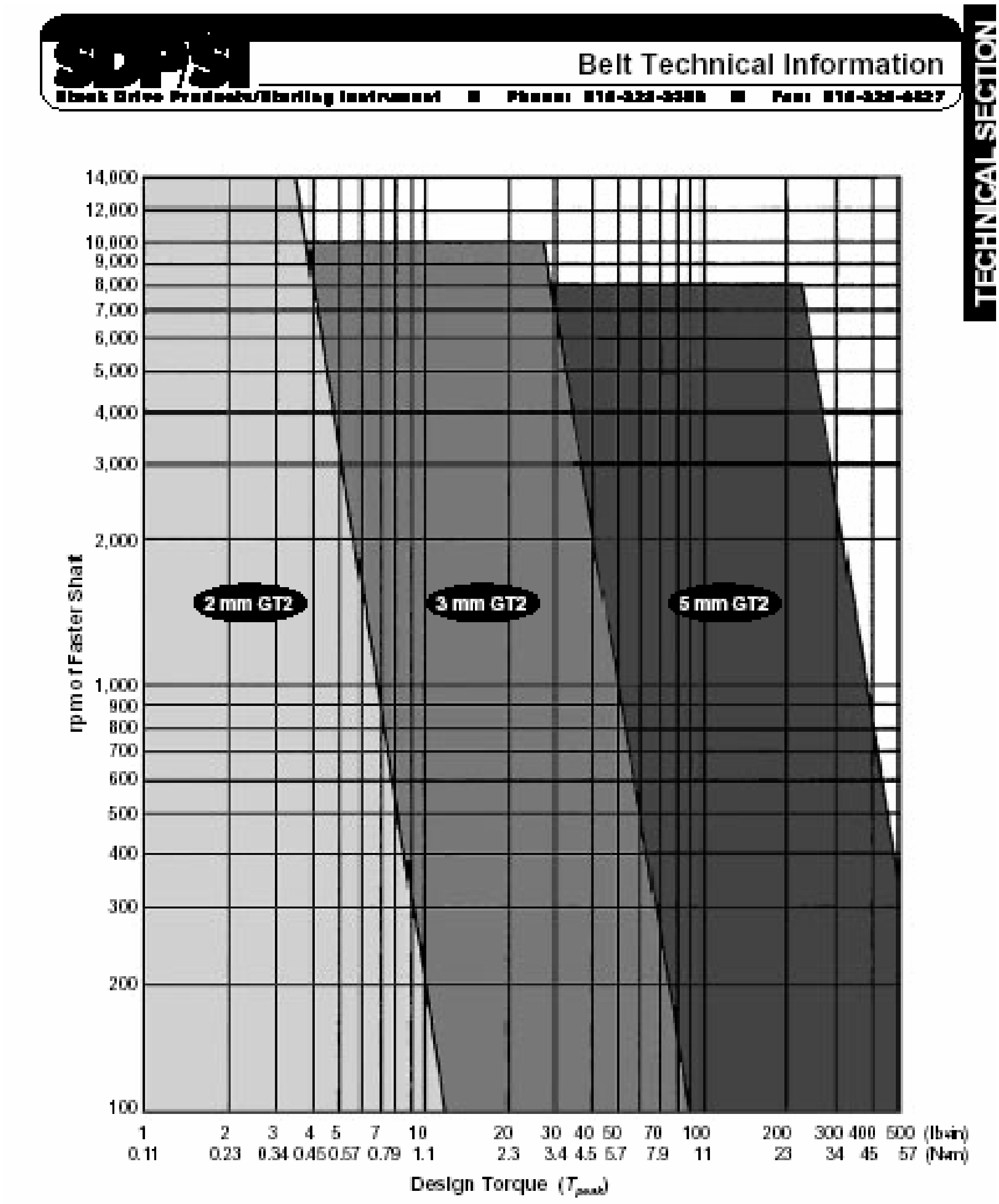


Figure 6.2: GT2 Belt Selection Guide

A timing belt has been chosen because it has several advantages over a flat belt or V-belt. Since the drive is by means of teeth rather than friction, there is no slippage and the driving and driven shafts remain synchronised. The toothed drive, having tension-carrying cords with minimum stretch, permits installation with minimal initial tension.

Timing belts are slightly more expensive than flat or V-belts, but this extra expense is well worth it because of the advantages the timing belt offers.

Step 3

$$V(m/s) = 52.4 \times 10^{-6} \times PD \times R_m$$
$$V(m/s) = 52.4 \times 10^{-6} \times 20 \times 1500$$
$$V(m/s) = 1.572 m/s$$

Step 4

Values obtained in the catalogue available at http://www.sdp-si.com suggest a belt width of 35mm. To simplify the belt drive this belt width should be used between the 20mm and 140mm pulley and also between the two 140mm pulleys. Therefore, the belt drive for this system should be a 35mm wide 3mm GT2 timing belt available from the above website.

6.6 V-groove Wheel Specification

Specifying the V-groove wheels for this design is a simple task. An online catalogue from Rapid Stock has been used to select the wheels for this project. The catalogue has been included in this dissertation as Appendix E.

There is a distinct shortage of Australian companies that offer a range of sizes of V-groove wheels online, so an American company has been used to source the availability of certain sizes. However, this does involve conversion from imperial units to metric.

Therefore, the wheels chosen for this project have been converted to metric units. The wheels are 100mm in diameter and are 37.5mm wide. They have a straight roller bearing and have a load capacity of about 300kg, meaning they are more than strong enough for this project. [22]

6.7 Miscellaneous Specifications

6.7.1 Limit Switches

From the limit switch catalogue shown in Appendix F, the limit switch with part number 802T has been chosen for this project. This particular model is quick to install and is operated by a lever. For the purposes of this project two limit switches will be required, one for each end of the test track. [1]

6.7.2 Actuator

From the actuator catalogue shown in Appendix F, the actuator chosen for this project has the model number CAT(R/L)33…x4/E220C. This actuator is capable of applying loads of 1000N, or approximately 100kg and is driven by a small AC motor. The actuator can move at a velocity of 20mm/s, which is ample for the requirement of this project. [24]

6.7.3 Carriage Framework

The framework of the carriage shall be built from 25mm by 25mm SHS displayed in the One Steel catalogue in Appendix D. This steel will be sufficiently strong to withstand the approximate point load produced by the motor being placed in the centre of the steel plate at the top of the carriage. This steel should be inexpensive due to the commonness of this particular size. [21]

All joints will be 4mm fillet welds as they are sufficient to carry the loads of the carriage and will be fast and inexpensive to produce.

6.7.4 Safety Enclosure

The facility shall be constructed inside an enclosure to ensure safety to operators and bystanders. The enclosure should protect people from all moving parts and should also protect them if any debris is thrown from the design.

The enclosure should a least be constructed from a fine mesh, although a solid enclosure wall such as glass is preferred. However, to keep costs down, it is satisfactory to construct the enclosure from meshed steel.

Multiple entry points to the enclosure are allowable, but a light sensor must protect each entry point. The light sensor will immediately cut power to the design as soon as it detects an object passing into the enclosure.

6.8 Final Design

The final design shall be constructed of the components listed below in table 6.1.

Component	Size	Model Number
Driving Wheel	250mm x 70mm	300X4DMD-SB20
Three Phase Electric Motors	0.75kW	MM3546
Parallel Rails	50mm x 50mm	
Timing Belt Drive	35mm x 3mm GT2	
Wheels	100mm x 37.5mm	
Limit Switches		802T
Actuator	1000N capable	CAT(R/L)33…x4/E220C
Framework	25mm x 25mm	
Connections	4mm Fillet Welds	

Table 6.1: Final Design Specifications

Chapter 7

Conclusions and Further Work

7.1 Conclusion

The pavement testing system designed in this project should be capable of producing accurate results.

In-depth research was conducted into the areas of both destructive and nondestructive testing. Current equipment from both of these fields were also researched and analysed for advantages and disadvantages.

A similar test rig at the Sydney University Pavement Testing Facility provided the basis for one concept design, while another concept design was developed from an overhead gantry crane. A decision matrix was used to choose between these two alternatives, with the overhead gantry crane, or rails, concept proving to be the better choice.

While some components were already decided upon, such as an electric motor, further research was needed to ensure the best decisions were made.

After all the large task of researching the types of components was completed, exact components were specified. An example of this is specifying a 250mm by 70mm driving wheel and a 0.75kW three-phase electric motor.

7.2 Further Work

The next step in the design of this pavement testing system is construction. However, both space and money is needed before the facility can be built.

Once built, experimental testing would finalise areas of the design such as placement of the limit switches to allow the secondary motor and lifting mechanism to activate and deactivate at the precise times.

Also, once built it would be obvious if any design changes were needed. It is hard to predict how some components will interface with each other without actually building the system, so further work may be required if trouble is encountered.

This system has been designed to allow investigation of the deformation of road surfaces. The system specified in this project will be one of the safest and most accurate traffic simulators developed, and once built, will provide meaningful results that can be used to develop better roads.

References

[1]

AB [Homepage of AB] [Online]. Available: http://www.ab.com [Accessed 3 October 2005].

[2]

AEG Electric Motors n.d., *Back to Basics: Single Phase Motor Theory* [Online]. Available: http://www.engineeringtalk.com/news/aeg/aeg129.html [Accessed 6 September 2005].

[3]

Baldor [Homepage of Baldor Motors and Drives] [Online]. Available: http://www.baldor.com.au [Accessed 12 September 2005].

[4]

Bandara, N. & Briggs, R. C. n.d., *Nondestructive Testing of Pavement Structures* [Online]. Available: http://www.asnt.org/publications/materialseval/basics/jul04basics/jul04basics.htm [Accessed 23 March 2005].

[5]

Caster City [Homepage of Caster City] [Online]. Available: http://www.castercity.com [Accessed 22 September 2005].

[6]

Darling, M. 1997, *Heavy Vehicle Simulator Condenses Pavement Testing Times* [Online]. Available http://www.crrel.usace.army.mil/news/news-archives/hvs/monthly.htm [Accessed 17 March 2005].

[7]

Durack, J. M. 2005, *Structural Design 1 USQ Study Book*, University of Southern Queensland, Toowoomba.

[8]

Dynatest [Homepage of Dynatest] [Online]. Available: http://www.dynatest.com/hardware/fwd_hwd.htmDynatest [Accessed 17 March 2005].

[9]

E.D. Informatics [Homepage of E.D. Informatics] [Online]. Available: http://www.edinformatics.com/inventions_inventors/electric_motors.htm [Accessed 6 April 2005].

[10]

Electric Motors [Homepage of Electric Motors] [Online]. Available http://www.electricmotors.machinedesign.com [Accessed 6 September 2005].

[11]

Erbe, R. & Reichard, J. n.d., *Drive Technology Alternatives* [Online]. Available: http://www.electricity-today.com/et/sept99/drives.html [Accessed 6 September 2005].

[12]

Fairbank Highway [Homepage of Turner Fairbank Highway Research Centre] [Online]. Available: http://www.fhwa.dot.gov/pavement/utwweb/facilit.cfm [Accessed 17 March 2005].

[13]

Fallshaw [Homepage of Fallshaw Wheels and Castors] [Online]. Available: http://www.fallshaw.com.au [Accessed 4 September 2005].

[14]

Garg, N. 2002, *Comparison between Falling Weight Deflectometer and Static Deflection Measurements on Flexible Pavements at the National Airport Pavement Test Facility (NAPTF)* [Online]. Available: http://www.airporttech.tc.faa.gov [Accessed 23 March 2005].

[15]

Gautrans [Homepage of Gautrans] [Online]. Available: http://www.gautrans-hvs.co.za/background.htm [Accessed 17 March 2005].

[16]

Global Spec [Homepage of Global Spec] [Online]. Available: http://fluid-power.globalspec.com/LearnMore/Fluid_Power_Components/Hydraulic_Equipment_Components/Hydraulic_Motors [Accessed 15 September 2005].

[17]

Grogg, M. G. & Hall, J. W. 2004, *Measuring Pavement Deflection at 55 mph* [Online]. Available: http://www.tfhrc.gov/pubrds/04jan/04.htm [Accessed 21 March 2005].

[18]

Hamilton Caster [Homepage of Hamilton Caster] [Online]. Available: http://www.hamiltoncaster.com [Accessed 22 September 2005].

[19]

Juvinall, R. C. & Marshek, K. M. 2000, *Fundamentals of Machine Component Design*, 3rd edn, R. R. Donnelley & Sons, North America.

[20]

Mastrad [Homepage of Mastrad] [Online]. Available: http://www.mastrad.com/track.htm [Accessed 12 September 2005].

[21]

One Steel [Homepage of One Steel] [Online]. Available: http://www.onesteel.com.au [Accessed 7 October 2005].

[22]

RWM [Homepage of Rapid Stock] [Online]. Available: http://www.rwm.com [Accessed 22 September 2005].

[23]

Sargand, S. n.d., *ORITE Accelerated Pavement Load Facility* [Online]. Available: http://webce.ent.ohiou.edu/orite/APLF.html [Accessed 17 March 2005].

[24]

SKF [Homepage of SKF] [Online]. Available http://www.skf.com [Accessed 22 September 2005]

[25]

SUNS [Homepage of SUNS International] [Online]. Available: http://www.suns-usa.com [Accessed 3 October 2005].

[26]

Sydney University Pavement Testing Facility [Homepage of the Sydney University Pavement Testing Facility] [Online]. Available: http://www.civil.usyd.edu.au/people/prathapa/Htmldocs/Draft1.htm [Accessed 15 March 2005].

[27]

Team Hybrid [Homepage of Team Hybrid] [Online]. Available: http://www.teamhybrid.co.uk/pages/viper_hubspec2.htm [Accessed 7 September 2005].

[28]

Techno-Stuff [Homepage of Techno-Stuff] [Online]. Available: http://www.techno-stuff.com/lswitch.htm [Accessed 3 October 2005].

[29]

Woodweb [Homepage of Woodweb] [Online]. Available: http://www.woodweb.com/knowledge_base/Threephase_motors_101.html [Accessed 7 September 2005].

APPENDIX A

Project Specification

University Of Southern Queensland
FACULTY OF ENGINEERING AND SURVEYING

ENG 4111/4112 Research Project
PROJECT SPECIFICATION

FOR: **RYAN DOMROW**

TOPIC: TESTING SYSTEM TO INVESTIGATE DEFORMATION OF ROAD SURFACES

SUPERVISOR: Dr. Selvan Pather

ENROLMENT: ENG 4111 – S1, D, 2005;
ENG 4112 – S2, D, 2005

PROJECT AIM: This aim of this project is to investigate the design of a system to simulate the deformation of road surfaces on a reduced scale.

PROGRAMME: Issue B, 27th October 2005

1. Background research on systems that include similar motion of the test rig to develop ideas on design.

2. Research on Australian Standards to obtain both the appropriate loading of the rig and also a suitable size scale.

3. Conduct feasibility studies on a variety of possible system designs to obtain the optimum system.

4. Detailed design of all components of chosen test system.

5. Outline further work to be undertaken.

As time permits:

6. Construct the test rig and run a series of tests to investigate the deformation of road surfaces.

AGREED:

Student Signature: ____________________ Date: _____________

Supervisors Signature: ____________________ Date: _____________

APPENDIX B

Excerpt from Fallshaw Catalogue for Pneumatic Wheels

pneumatic wheels
STEEL AND PLASTIC CENTRED

Choose from steel or plastic centres with these quality features:

- Deep groove ball bearings
  - normal life 5 years (many imports use pressed cone bearings normal life 1 year)
- 4 ply and 6 ply
  (on HWY tread) for longer, tougher life (many imports use 2 ply)
- High engineering standards
  for wobble-free wheels backed by a 3 year warranty
- Bushes & bearings available in 20mm, 1", 3/4" & 5/8", to suit your application*

*Standard bore sizes shown, others available by negotiation.

STEEL CENTRED

Max. temp. 55°C

- 1.8 - 2mm thick steel - many imports use 1.2mm

^ = Cone Bearing
< = Ball Bearing

tread patterns	diameter x width	KPa/ psi	bearing IDx length	capacity kg 8km/hr	capacity kg 15km/hr	model no.
	220x54	210 30	20x80	120	100	^250x4IND-SB20
	220x54	210 30	20x80	120	100	^250x4LUG-SB20
	250x70	210 30	20x80	140	110	^300 X4DMD-SB20
	265x70	210 30	20x80	180	160	^350 X4DMD-SB20
	275x65	210 30	1"X80	140	120	^250 X6IND-SB10
	320x80	210 30	1"X80	200	180	^350 X6KNO-SB10
	400x100	210 30	1"X72	360	340	<400 X8HWY-SQ10
	400x100	210 30	1"X80	220	200	^400 X8KNO-SB10
	400x100	210 30	1"X80	220	200	^400 X8DMD-SB10
	330x125	210 30	1"X80	130	100	^500 X6GRA-SB10
	420x165	210 30	1"X150	250	200	^650 X8GRA-SB10

PLASTIC CENTRED

Max. temp. 45°C

- Thick sections, larger radii, multi-web design for strength (stronger than most imported steel centres)
- Corrosion resistant plastic with UV fade protection

* = Plain Bearing ^ = Cone Bearing > = Roller Bearing
< = Ball Bearing

tread patterns	diameter x width	KPa/ psi	bearing IDx length	capacity kg 8km/hr	capacity kg 15km/hr	model no.
	200x50	210 30	20x80 12x80	75 75	- -	>200 X50RIB-PR20 <200 X50RIB-PQ12
	220x54	210 30	20x80	100	80	^250 X4IND-PWB20 * 250X4IND-PWA20
	220x54	210 30	20x80	100	80	^250 X4LUG-PWB20 *250 X4LUG-PWA20
	250x70	210 30	20x80	120	90	^300 X4DMD-PWB20 *300 X4DMD-PWA20
	275x60	210 30	20X80	110	90	^250 X6IND-PWB20 *250 X6IND-PWA20
	320x80	210 30	20X80	140	110	^350 X6KNO-PWB20 *350 X6KNO-PWA20
	400x100	210 30	1"X107 1"X105	200 200	180 180	^400 X8HWY-PWB10 *400 X8HWY-PWA10
	400x100	210 30	1"X107 1"X105	200 200	180 180	^400 X8KNO-PWB10 *400 X8KNO-PWA10
	400x100	210 30	1"X107 1"X105	200 200	180 180	^400 X8DMD-PWB10 *400 X8DMD-PWA10

WARNING - Never exceed recommended capacities or speeds

Figure B.1: Excerpt from Fallshaw Catalogue for Pneumatic Wheels [13]

APPENDIX C

Excerpt from Baldor Pty Ltd Catalogue for Electric Motors

Performance... Baldor 50 Hertz High Efficient Metric Motors provide true industrial performance.

IP54
Performance Data
High Efficient Metric Motors – Meet or Exceed EFF2

kW	*Catalog Number	Frame Size	F.L. R.P.M.	Nominal Eff. @ 1/2	Nominal Eff. @ 3/4	Nominal Eff. @ F.L.	Power Factor P.F. @ 1/2	Power Factor P.F. @ 3/4	Power Factor P.F. @ F.L.	Current 380V -65*	Current 400V -58*	Current 415V -57*	Torque FLT N-M	Torque BDT N-M	Torque LRT N-M	Rotor Inertia (kg m^2)
0.75	MM3545	D80	2850	73.8	77.9	79.1	63.0	75.0	82.0	1.87	1.66	1.86	2.50	10.2	9.50	0.0026
0.75	MM3546	D80	1440	79.0	81.8	82.1	54.0	67.0	76.0	1.70	1.70	1.70	5.00	16.0	13.0	0.0110
0.75	MM3556	D90	950	71.7	78.5	78.2	42.0	54.0	64.0	2.10	2.20	2.20	7.50	26.0	19.0	0.0230
1.1	MM3550	D80	2850	79.5	82.2	82.8	69.0	80.0	86.0	2.34	2.26	3.06	4.30	16.8	14.7	0.0110
1.1	MM3554	D90	1425	81.6	83.9	84.1	64.0	76.0	83.0	2.40	2.30	2.30	7.50	23.0	15.0	0.0230
1.1	MM3607	D90	950	78.1	80.8	80.5	51.0	65.0	74.0	2.80	2.30	2.70	11.5	32.0	24.0	0.0330
1.5	MM3555	D90	2850	80.0	83.0	83.8	71.0	81.0	87.0	3.06	2.97	2.92	4.30	16.8	14.7	0.0110
1.5	MM3558	D90	1425	82.0	84.1	84.0	65.0	77.0	83.0	3.20	3.10	3.10	9.90	30.0	19.0	0.0260
1.5	MM3614	D100	950	78.9	82.8	83.8	38.0	50.0	59.0	3.40	3.50	3.60	14.6	46.0	28.0	0.0390
2.2	MM3559	D90	2850	82.8	84.8	84.8	75.0	85.0	89.0	4.46	4.28	4.19	7.50	25.7	19.0	0.0130
2.2	MM3611	D100	1450	84.6	86.7	87.0	55.0	68.0	76.0	4.80	4.90	5.0	14.6	49.0	31.0	0.0310
2.2	MM3704	D112	950	81.4	84.2	84.7	45.0	57.0	66.0	5.30	5.80	5.90	22.0	60.0	34.0	0.0620
3.7	MM3613	D100	2850	84.7	86.8	86.8	81.0	89.0	92.0	7.10	6.80	6.70	12.3	46.0	34.0	0.0210
3.7	MM3616	D112	1450	87.3	88.5	88.2	61.0	74.0	81.0	7.80	7.50	7.70	24.5	74.0	48.0	0.0470
3.7	MM3708	D132	950	85.2	86.9	86.9	44.0	56.0	65.0	9.30	9.60	9.60	37.0	94.0	45.0	0.1400
5.5	MM3709	D132	2900	87.5	89.0	88.9	82.0	89.0	91.0	10.4	10.0	9.70	18.4	56.0	35.0	0.0400
5.5	MM3710	D132	1450	88.8	89.9	89.7	60.0	72.0	80.0	11.5	11.4	11.5	36.5	104.0	61.0	0.1200
5.5	MM2276	D132	970	86.7	87.9	87.5	44.0	57.0	65.0	13.9	14.0	14.4	56.0	130.0	64.0	0.1900
7.5	MM3711	D132	2900	89.2	90.2	90.0	84.0	90.0	92.0	13.6	13.0	12.8	24.5	78.0	44.0	0.0610
7.5	MM3714	D132	1450	89.0	90.1	90.0	59.0	72.0	79.0	15.3	15.2	15.4	48.5	146.0	84.0	0.1500
7.5	MM2332	D160	970	87.8	89.5	89.6	56.0	68.0	74.0	16.7	16.3	16.1	73.0	160.0	88.0	0.5300
11	MM2394	D160	2900	90.1	90.8	90.4	83.0	90.0	92.0	20.4	19.4	18.9	36.9	119.0	69.0	0.0610
11	MM2333	D160	1450	89.8	90.8	90.8	68.0	79.0	84.0	21.8	21.2	20.9	72.8	278.0	136.0	0.2400
11	MM4100	D160	970	88.3	89.8	89.9	56.0	67.0	74.0	25.0	24.5	24.3	109.0	230.0	124.0	0.6700
15	MM4106	D160	2900	90.9	91.3	90.6	85.0	90.0	92.0	27.2	25.8	25.0	48.8	133.0	89.0	0.1500
15	MM2334	D160	1450	90.5	91.4	91.2	70.0	80.0	85.0	28.7	27.8	27.3	97.0	230.0	166.0	0.2900
15	MM4102	D180	970	89.7	90.8	90.2	65.0	76.0	81.0	30.6	29.6	29.2	146.0	351.0	245.0	0.6700
18.5	MM4118	D160	2925	91.5	91.8	91.1	85.0	90.0	92.0	33.7	32.0	31.0	60.9	164.0	89.0	0.1800
18.5	MM4103	D180	1450	91.0	92.1	92.1	69.0	79.0	83.0	36.3	35.0	34.4	121.0	305.0	226.0	0.5500
18.5	MM4111	D200	970	91.0	91.9	91.8	69.0	79.0	83.0	36.9	35.6	34.8	182.0	408.0	275.0	1.3000
22	MM4108	D180	2925	92.4	91.5	91.4	82.0	88.0	90.0	41.0	39.2	38.2	73.0	182.0	119.0	0.2500
22	MM4104	D180	1450	90.8	92.0	92.2	66.0	77.0	82.0	44.0	42.9	42.4	145.0	382.0	285.0	0.6200
22	MM4117	D200	970	91.1	92.3	92.3	63.0	77.0	82.0	44.3	42.9	42.2	218.0	510.0	360.0	1.5000
30	MM4109	D200	2925	92.5	91.7	91.8	83.0	89.0	91.0	53.9	51.4	49.9	98.7	277.0	121.0	0.4500
30	MM4110	D200	1450	92.3	93.1	93.1	73.0	82.0	86.0	56.0	54.0	53.0	193.0	516.0	272.0	0.5900
30	MM4308	D225	970	92.2	93.0	92.9	69.0	80.0	84.0	57.2	55.2	54.0	290.0	386.0	430.0	2.0000
37	MM4114	D200	2925	91.7	92.5	92.4	86.0	91.0	92.0	66.6	63.3	61.2	121.0	300.0	145.0	0.5400
37	MM4115	D225	1450	92.1	93.1	93.2	76.2	84.0	87.0	69.0	66.0	65.0	241.0	625.0	300.0	1.2000
37	MM4312	D250	970	92.6	93.2	93.0	74.0	83.0	86.0	70.0	67.0	66.0	360.0	500.0	530.0	1.5000
45	MM4310	D225	2950	91.6	92.8	93.0	85.0	91.0	93.0	79.0	75.0	73.0	144.0	425.0	168.0	0.5400
45	MM4314	D225	1470	92.7	93.8	93.8	77.0	85.0	89.0	82.0	79.0	77.0	289.0	746.0	350.0	1.5000
45	MM4403	D250	970	92.8	93.8	93.8	72.0	81.0	86.0	84.0	81.0	79.0	432.0	1190.0	730.0	4.3000

NOTE: * When placing an order please specify the catalog number along with the voltage number of the motor. E– MM3550-65. (See page 15)
Motors rated .75 - 7.2 kW are steel band construction, with the exception of MM2276 and MM2332.

12

Figure C.1: Excerpt from Baldor Pty Ltd Catalogue for Electric Motors (p12) [3]

IP54
Performance Data
High Efficient Metric Motors

kW	*Catalog Number	Frame Size	F. L. R.P.M.	Nominal Eff. @ 1/2	Nominal Eff. @ 3/4	Nominal Eff. @ F.L.	Power Factor P.F.@ 1/2	Power Factor P.F. @ 3/4	Power Factor P.F. @ F.L.	Current 380V -65°	Current 400V -58°	Current 415V -57°	Torque FLT N-M	Torque POT N-M	Torque LRT N-M	Rotor Inertia (kg m^2)
55	MM4313	D250	2958	91.6	92.9	93.2	87.0	88.8	91.0	99.0	95.0	93.0	180.0	615.8	231.0	1.2008
55	MM4316	D250	1458	92.7	93.9	94.2	74.0	82.8	86.0	103.0	100.0	90.0	380.0	978.8	480.0	3.0008
55	MM4404	D280	978	92.5	93.5	93.7	74.0	82.8	86.0	104.0	100.0	98.0	540.0	1468.8	755.0	5.2008
75	MM4402	D250	2958	92.8	93.7	93.7	85.0	98.8	92.0	131.0	125.0	121.0	240.0	765.8	380.0	1.4008
75	MM4408	D250	1458	93.2	94.2	94.5	74.0	82.8	86.0	137.0	133.0	131.0	480.0	1278.8	570.0	3.8008
75	MM4409	D280	978	93.3	94.2	94.3	74.0	83.8	86.0	180.0	133.0	130.0	721.0	1898.8	970.0	6.3008
90	MM4412	D280	2958	93.4	94.3	94.5	84.0	98.8	92.0	157.0	150.0	145.0	289.0	888.8	390.0	2.7008
90	MM4418	D280	1458	93.6	94.4	94.5	76.0	84.8	88.0	169.0	163.0	180.0	680.0	1678.8	880.0	4.0008
90	MM4411	D315	978	93.7	94.4	94.4	76.0	84.8	87.0	171.0	164.0	180.0	982.0	2348.8	1380.0	7.4008
110	MM4413	D280	2958	94.1	94.8	94.8	87.0	91.8	92.0	194.0	185.0	179.0	380.0	1808.8	380.0	2.4008
110	MM4406	D280	1458	94.1	94.8	94.8	77.0	85.8	88.0	201.0	194.0	190.0	720.0	2898.8	1180.0	4.8008
110	MM44156	D315	978	93.9	94.6	94.6	74.0	83.8	87.0	205.0	197.0	193.0	1080.0	3808.8	1780.0	8.8008
130	MM4414	D315	2958	94.4	94.9	94.7	88.0	92.8	93.0	226.0	214.0	207.0	420.0	1158.8	470.0	2.8008
130	MM4409	D315	1458	94.4	95.0	95.8	79.0	86.8	89.0	233.0	224.0	219.0	840.0	2338.8	1265.0	5.3008
130	MM4428T	D315	978	94.1	94.8	94.7	75.0	84.8	87.0	239.0	230.0	225.0	1280.0	3318.8	1920.0	9.8008
150	E[illegible]4416	D315	2958	94.6	95.1	95.1	87.0	91.8	93.0	256.0	244.0	237.0	480.0	1468.8	615.0	2.9008
150	MM4407	D315	1458	94.5	95.1	95.1	78.0	86.8	89.0	267.0	256.0	250.0	980.0	2738.8	1480.0	5.8008
150	MM44286	D315	978	94.1	94.8	94.8	74.0	83.8	86.0	275.0	264.0	259.0	1437.0	3938.8	2340.0	10.7008
185	MM44252	D315	2958	95.0	95.4	95.3	88.0	92.8	93.0	318.0	304.0	294.0	680.0	1778.8	880.0	3.3008
185	MM4408	D315	1458	95.0	95.5	95.4	87.0	87.8	90.0	330.0	315.0	307.0	1280.0	3228.8	1815.0	7.2008
185	MM44256	D315	978	94.4	95.0	94.8	75.0	84.8	87.0	330.0	327.0	320.0	1883.0	4888.8	3110.0	12.9008

NOTE: * When placing an order please specify the catalog number along with the voltage number of the motor. E- MM3550-65. (See page 15)

Baldor 50 Hertz Premium Efficient Metric Motors designed to conserve energy over extended time periods.

IP55
Performance Data
Super-E® Premium Efficient Metric Motors – Meet or Exceed

kW	*Catalog Number	Frame Size	F. L. R.P.M.	Nominal Eff. @ 1/2	Nominal Eff. @ 3/4	Nominal Eff. @ F.L.	Power Factor P.F.@ 1/2	Power Factor P.F. @ 3/4	Power Factor P.F. @ F.L.	Current 380V -65°	Current 400V -58°	Current 415V -57°	Torque FLT N-M	Torque POT N-M	Torque LRT N-M	Rotor Inertia (kg m^2)
0.75	EMM3545**	D80	2850	76.8	80.2	81.1	68.0	79.0	85.0	1.59	1.56	1.56	2.5	6.40	6.10	0.0051
0.75	EMM3546**	D80	1450	81.3	83.9	84.3	52.0	68.0	75.0	1.88	1.71	1.78	4.9	15.6	11.0	0.0180
0.75	EMM3582	D90	950	73.9	78.2	79.5	36.0	48.0	58.0	2.2	2.4	2.5	7.4	21.6	14.0	0.0230
1.1	EMM3583	D90	2850	78.3	82.6	83.5	68.0	79.0	85.0	2.32	2.28	2.29	3.7	14.9	10.6	0.0092
1.1	EMM3584	D90	1450	84.3	85.7	85.4	59.0	72.0	80.0	2.39	2.38	2.41	7.4	23.6	14.9	0.0230
1.1	EMM3587	D90	950	77.9	81.2	81.8	38.0	50.0	60.0	3.2	3.3	3.4	11.1	33.0	20.0	0.0330
1.5	EMM3586	D90	2850	83.7	85.9	86.1	75.0	84.0	89.0	2.92	2.82	2.77	4.9	17.4	12.0	0.0120
1.5	EMM3587	D90	1450	85.4	85.6	86.2	59.0	72.0	80.0	3.15	3.13	3.18	9.9	30.3	20.9	0.0290
1.5	EMM3854	D100	970	81.2	84.3	85.2	40.0	52.0	61.0	4.0	4.1	4.3	14.6	44.0	26.0	0.0390
2.2	EMM3850	D90	2850	84.8	85.4	86.3	73.0	83.0	88.0	4.37	4.25	4.22	7.4	25.0	17.0	0.0130
2.2**	EMM3611**	D100	1450	85.6	87.6	88.0	49.0	63.0	72.0	5.01	5.08	5.24	14.3	58.4	43.6	0.0360
2.2	EMM3764	D112	950	83.1	85.4	85.7	43.0	55.0	64.0	5.8	5.9	6.0	22.0	61.0	37.0	0.0520
3.7	EMM3853	D112	2850	83.7	88.8	88.5	75.0	85.0	89.0	6.97	6.82	6.64	12.3	48.0	31.0	0.0210
3.7	EMM3855	D112	1460	88.3	89.4	89.1	56.0	72.0	80.0	7.87	7.80	7.88	24.0	92.0	32.0	0.0510
3.7	EMM3958	D132	960	86.4	88.0	87.9	42.0	55.0	64.0	9.5	9.7	9.9	37.0	94.0	45.0	0.1400
5.5	EMM3959	D132	2900	90.0	90.7	90.3	81.0	88.0	91.0	10.3	9.9	9.6	18.4	59.0	33.0	0.0420
5.5	EMM3730	D132	1450	89.4	90.5	90.5	61.0	73.0	80.0	11.3	11.2	14.6	36.0	105.0	56.0	0.1300
5.5	EMM2276	D132	960	87.7	88.9	88.5	43.0	55.0	64.0	14.0	14.2	14.6	55.5	132.0	64.0	0.1800

NOTE: *When placing an order please specify the catalog number along with the voltage number of the motor. IE- MM3550-65. (See page 15)
** Steel band

13

Figure C.2: Excerpt from Baldor Pty Ltd Catalogue for Electric Motors (p13) [3]

APPENDIX D

Excerpt from Onesteel Catalogue for Square Hollow Section Steel Properties

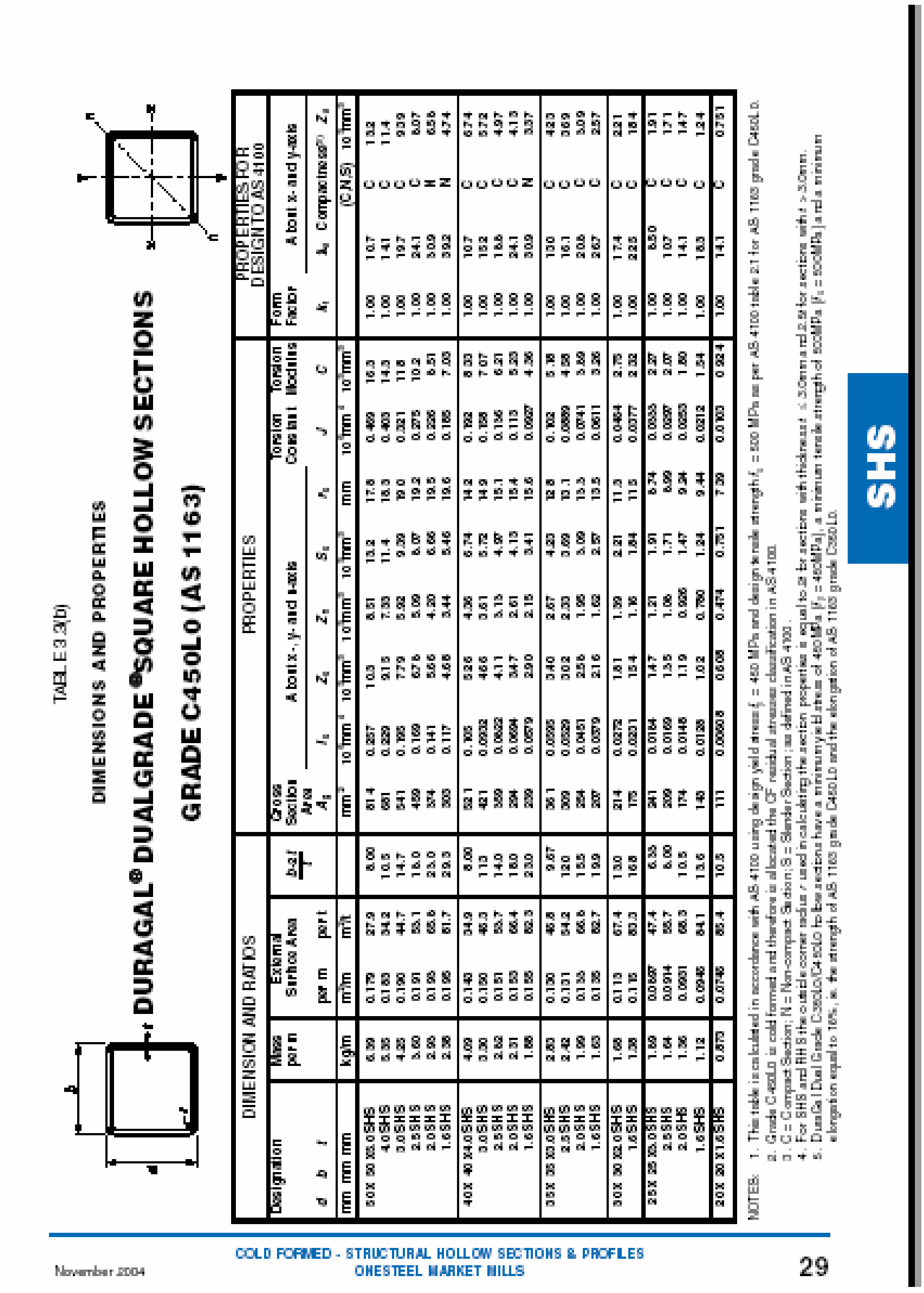


TABLE 3.3(b)

DIMENSIONS AND PROPERTIES

DURAGAL® DUALGRADE® SQUARE HOLLOW SECTIONS

GRADE C450L0 (AS 1163)

Designation $d \times b \times t$	Mass per m	External Surface Area per m	External Surface Area per t	$\frac{b-2t}{t}$	Gross Section Area A_g	I_x	Z_x	Z_n	S_x	r_x	Torsion Constant J	Torsion Modulus C	Form Factor k_f	λ_e	Compactness (C,N,S)	Z_e
mm mm mm	kg/m	m²/m	m²/t		mm²	10^6 mm⁴	10^3 mm³	10^3 mm³	10^3 mm³	mm	10^6 mm⁴	10^3 mm³				10^3 mm³
50 X 50 X 5.0 SHS	6.39	0.179	27.9	8.00	814	0.257	10.3	8.51	13.2	17.8	0.469	16.3	1.00	10.7	C	13.2
4.0 SHS	5.35	0.183	34.2	10.5	681	0.229	9.15	7.33	11.4	18.3	0.403	14.3	1.00	14.1	C	11.4
3.0 SHS	4.25	0.190	44.7	14.7	541	0.195	7.79	5.92	9.39	19.0	0.321	11.8	1.00	19.7	C	9.39
2.5 SHS	3.60	0.191	53.1	18.0	459	0.169	6.76	5.09	8.07	19.2	0.275	10.2	1.00	24.1	C	8.07
2.0 SHS	2.93	0.193	65.6	23.0	374	0.141	5.66	4.20	6.66	19.5	0.226	8.51	1.00	30.9	N	6.56
1.6 SHS	2.38	0.195	81.7	29.3	303	0.117	4.68	3.44	5.46	19.6	0.185	7.03	1.00	39.2	N	4.74
40 X 40 X 4.0 SHS	4.09	0.143	34.9	8.00	521	0.105	5.26	4.36	6.74	14.2	0.192	8.33	1.00	10.7	C	6.74
3.0 SHS	3.30	0.150	45.3	11.3	421	0.0932	4.66	3.61	5.72	14.9	0.158	7.07	1.00	15.2	C	5.72
2.5 SHS	2.82	0.151	53.7	14.0	359	0.0822	4.11	3.15	4.97	15.1	0.136	6.21	1.00	18.8	C	4.97
2.0 SHS	2.31	0.153	66.4	18.0	294	0.0694	3.47	2.61	4.13	15.4	0.113	5.23	1.00	24.1	C	4.13
1.6 SHS	1.88	0.155	82.3	23.0	239	0.0579	2.90	2.15	3.41	15.6	0.0927	4.36	1.00	30.9	N	3.37
35 X 35 X 3.0 SHS	2.83	0.130	45.8	9.67	361	0.0595	3.40	2.67	4.23	12.8	0.102	5.18	1.00	13.0	C	4.23
2.5 SHS	2.42	0.131	54.2	12.0	309	0.0529	3.02	2.33	3.69	13.1	0.0889	4.58	1.00	16.1	C	3.69
2.0 SHS	1.99	0.133	66.6	15.5	254	0.0451	2.56	1.95	3.09	13.3	0.0741	3.89	1.00	20.8	C	3.09
1.6 SHS	1.63	0.135	82.7	19.9	207	0.0379	2.16	1.62	2.57	13.5	0.0611	3.26	1.00	26.7	C	2.57
30 X 30 X 2.0 SHS	1.68	0.113	67.4	13.0	214	0.0272	1.81	1.39	2.21	11.3	0.0464	2.75	1.00	17.4	C	2.21
1.6 SHS	1.38	0.115	83.3	16.8	175	0.0231	1.54	1.16	1.84	11.5	0.0377	2.32	1.00	22.5	C	1.84
25 X 25 X 3.0 SHS	1.89	0.0897	47.4	6.33	241	0.0184	1.47	1.21	1.91	8.74	0.0333	2.27	1.00	8.50	C	1.91
2.5 SHS	1.64	0.0914	55.7	8.00	209	0.0169	1.35	1.06	1.71	8.99	0.0297	2.07	1.00	10.7	C	1.71
2.0 SHS	1.36	0.0931	68.3	10.5	174	0.0148	1.19	0.926	1.47	9.24	0.0253	1.80	1.00	14.1	C	1.47
1.6 SHS	1.12	0.0946	84.1	13.6	143	0.0128	1.02	0.780	1.24	9.44	0.0212	1.54	1.00	18.3	C	1.24
20 X 20 X 1.6 SHS	0.873	0.0746	85.4	10.5	111	0.00608	0.608	0.474	0.751	7.39	0.0103	0.924	1.00	14.1	C	0.751

NOTES:
1. This table is calculated in accordance with AS 4100 using design yield stress f_y = 450 MPa and design tensile strength f_u = 500 MPa as per AS 4100 table 2.1 for AS 1163 grade C450L0.
2. Grade C450L0 is cold formed and therefore is allocated the CF residual stresses classification in AS 4100.
3. C = Compact Section; N = Non-compact Section; S = Slender Section; as defined in AS 4100.
4. For SHS and RHS the outside corner radius r used in calculating the section properties is equal to 2t for sections with thickness t ≤ 3.0mm and 2.5t for sections with t > 3.0mm.
5. DuraGal Dual Grade C350L0/C450L0 hollow sections have a minimum yield stress of 450 MPa (F_y = 450MPa), a minimum tensile strength of 500MPa (F_u = 500MPa) and a minimum elongation equal to 16%, ie. the strength of AS 1163 grade C450L0 and the elongation of AS 1163 grade C350L0.

SHS

November 2004

COLD FORMED - STRUCTURAL HOLLOW SECTIONS & PROFILES
ONESTEEL MARKET MILLS

29

Figure D.1: Excerpt from One Steel Catalogue for Square Hollow Section Steel Properties [21]

APPENDIX E

Rapid Smart Catalogue for V-grooved Wheels

RAPIDTRACK

VF and VI Wheels

V-Groove Wheels

V-Groove wheels are high strength cast iron (VI) castings or forged steel (VF) forgings machined with a 90° groove for operation on an inverted angle iron track. They are used to control the flow of a load, i.e., into an oven, between machines, over long distances or where "production line" sequence is required. Care must be made to assure the inverted angle iron track is parallel to ensure smooth and constant tracking. In some cases it may be advisable to run inverted angle iron on one side with V-Groove wheels and flat bar stock steel on the other side with a flat faced cast iron or forged steel wheel.

VFT-1040-20

UP TO 15,000 POUND CAPACITY

FEATURES

Wheel Bearing - Choice of straight roller bearing or tapered bearing

Temperature Range - -50° to +800°F

Keyways - Can be installed on most sizes

Hub Lengths - Can be cut or lengthened with spacers

Bores - Various sizes available on some wheels

Lubrication Fittings - Standard on 2-1/2" and wider wheels

SPECIFICATIONS

Wheel Diameter	Wheel Width	lbs Capacity	Axle Diameter	Straight Roller Bearing	Tapered Roller Bearing	Width of Groove	Depth of Groove	Hub Length	Approx Weight
4"	1-1/2"	700	1/2" 5/8" 3/4"	VIR0415-08 VIR0415-10 VIR0415-12		7/8"	7/16"	1-7/8"	3.4
4"	2"	800	1/2" 5/8" 3/4"	VIR0420-08 VIR0420-10 VIR0420-12		7/8"	7/16"	2-7/16"	4.5
5"	2"	800	1/2" 5/8" 3/4"	VIR0520-08 VIR0520-10 VIR0520-12		7/8"	7/16"	2-7/16"	6.3
6"	2"	1000	1/2" 5/8" 3/4"	VIR0620-08 VIR0620-10 VIR0620-12	VITO620-08 VITO620-12	7/8"	7/16"	2-7/16"	7.2
6"	2-1/2"	1500	3/4" 1" 1" 1-1/4"	VIR0625-12-YY VIR0625-16-SI-YY VIR0625-16-YY VIR0625-20-YY	VITO625-12-YY VITO625-16-YY	7/8"	7/16"	3"	11.2
		2500	3/4" 1" 1" 1-1/4"	VIR0625-12 VIR0625-16-SI VIR0625-16 VIR0625-20	VITO625-12 VITO625-16	7/8"	7/16"	3-1/2"	11.2
6"	3"	3500	3/4" 1" 1" 1-1/4"	VIR0630-12 VIR0630-16 VIR0630-16-SI VIR0630-20	VITO630-12 VITO630-16	7/8"	7/16"	3-1/2"	20.0
		7000	3/4" 1" 1" 1-1/4" 1-1/2"	VFR0630-12 VFR0630-16 VFR0630-16-SI VFR0630-20 VFR0630-24	VFTO630-12 VFTO630-16 VFTO630-20 VFTO630-24	1-3/8"	11/16"	3-1/2"	30.0
		10,000	1-1/2"		VFTO630-20-HD	1-3/8"	11/16"	3-1/2"	30.0
8"	2"	1050	1/2" 5/8" 3/4"	VIR0820-08 VIR0820-10 VIR0820-12	VITO820-08 VITO820-12	7/8"	7/16"	2-7/16"	13.2
8"	2-1/2"	3000	3/4" 1" 1" 1-1/4"	VIR0825-12 VIR0825-16-SI VIR0825-16 VIR0825-20	VITO825-12 VITO825-16	7/8"	7/16"	3-1/2"	15.8
8"	3"	5000	3/4" 1" 1" 1-1/4"	VIR0830-12 VIR0830-16-SI VIR0830-16 VIR0830-20	VITO830-12 VITO830-16	7/8"	7/16"	3-1/2"	23.5
		6000	3/4" 1" 1-1/4" 1-1/4" 1-1/2"	VFR0830-12 VFR0830-16-SI VFR0830-20 VFR0830-20-SI VFR0830-24	VFTO830-12 VFTO830-16 VFTO830-20	7/8"	7/16"	3-1/2"	30.0
8"	4"	10,000	1-1/4" 1-1/2" 2"	VFR0840-20-SI VFR0840-24-SI VFR0840-32		1-3/4"	7/8"	4-1/2"	30.0
		15,000	1-1/4" 1-1/2"		VFTO840-20 VFTO840-24	1-3/4"	7/8"	4-1/2"	30.0
10"	3"	5000	3/4" 1" 1" 1-1/4"	VIR1030-12 VIR1030-16-SI VIR1030-16 VIR1030-20	VIT1030-12 VIT1030-16 VIT1030-20	7/8"	7/16"	3-1/2"	21.0
		6000	3/4" 1" 1" 1-1/4"	VFR1030-12 VFR1030-16 VFR1030-16-SI VFR1030-20	VFT1030-12 VFT1030-16 VFT1030-20	1-3/8"	11/16"	3-1/2"	31.0
10"	4"	10,000	1-1/4" 1-1/2" 2"	VFR1040-20-SI VFR1040-24-SI VFR1040-32		1-3/4"	7/8"	4-1/2"	38.0
		15,000	1-1/4" 1-1/2"		VFT1040-20 VFT1040-24	1-3/4"	7/8"	4-1/2"	38.0
12"	3"	6000	3/4" 1" 1" 1-1/4"	VIR1230-12 VIR1230-16 VIR1230-16-SI VIR1230-20	VIT1230-12 VIT1230-16 VIT1230-20	7/8"	7/16"	3-1/2"	39.0

OPTIONS

WS	Wheel Seals
ZF	Zerk Fittings on 2" wide and smaller wheels

V-Groove Wheels
Forged Steel Wheels

Technology in Motion™

TABLE OF CONTENTS

RWM Product Catalog

60

Figure E.1: Excerpt from Rapid Smart Catalogue for V-Groove Wheels [22]

APPENDIX F

Limit Switch and Actuator Catalogues

Limit Switch Selection Guide

	802T Plug-In	802T Safety Plug-In	802M Factory Sealed	802MC Corrosion Resistant
Description/ Features	■ Plug-in construction provides quick and easy installation ■ Non front-mounting design with equal-length mounting screws make it easy to apply ■ Simple mode-change mechanism in head for easy setup ■ 0° to 230° F (-18° to 110° C) ■ Optional -40° to 230° F (-40° to 110° C)	■ For use in control reliable and safety applications per ISO 14119 ■ NEMA style mounting dimensions ■ Rugged metal construction for harsh industrial environments ■ Plug-in style for quick and easy installation ■ 0° to 250° F (-18° to 110° C)	■ Compact, pre-wired switch is factory sealed to meet the requirements of demanding applications, wet or dry ■ 32° to 180° F (0° to 80° C)	■ Compact, pre-wired switch is factory sealed to meet the requirements of demanding applications, wet or dry ■ 32° to 180° F (0° to 80° C)
Actuators	■ Lever, maintained, low operating force, top and side push (with or without rollers), cat whisker, wobble stick, neutral position	■ Lever, top push roller, side push vertical roller, side push horizontal roller	■ Lever, maintained, top and side push (with or without rollers)	■ Lever
Enclosure Classification①	■ NEMA 1, 4, & 13 (6P – select side rotary models)	■ NEMA 4, 12, 13, 6P, IP67	■ NEMA 1, 4, 6P & 13, IP67 ■ Exceeds requirements for NEMA 6P	■ NEMA 1, 4, 4X, 6P & 13, IP67 ■ Exceeds requirements for NEMA 6P
Additional Info	■ C115 catalog page 5-6 ■ www.ab.com/catalogs	■ C115 catalog page 5-35 ■ www.ab.com/catalogs	■ C115 catalog page 5-32 ■ www.ab.com/catalogs	■ C115 catalog page 5-32 ■ www.ab.com/catalogs

	802B Compact	802B Precision	802B Small Precision	440P-C International Small Plastic	440P-M International Large Metal
Description/ Features	■ Compact metal body is pre-wired to maintain enclosure seals. Industry standard mounting for ease of installation ■ 14° to 158° F (-10° to 70° C)	■ Precision style limit switch uses industry standard mounting. Low trip and reset points for more precise sensing ■ 14° to 176° F (-10° to 80° C)	■ Small Precision limit switches are metal bodied for use in industrial applications. 17 different styles available for solving multiple applications ■ 14° to 176° F (-10° to 80° C)	■ Conforms to EN50047 (22mm) ■ Glass reinforced thermoplastic housing ■ Most feature direct opening contacts designed to meet IEC 947. Available in snap-acting, slow make/ break with 2 or 3 pole contact arrangement ■ Heads can be rotated in 90 degree increments for flexible mounting ■ -13° to 176° F (-25° to 80° C)	■ Conforms to EN50041 (30mm x 60mm) ■ Cast aluminum housing ■ Most feature direct opening contacts designed to meet IEC 947. Available in snap-acting, slow make/ break with 2, 3 or 4 pole contact arrangement ■ Heads can be rotated in 90 degree increments for flexible mounting ■ -13° to 176° F (-25° to 80° C)
Actuators	■ Rotary arm, center rotary arm, wobble stick, top push, top push lever, top push roller, top push cross roller	■ Top push, top push roller, top push cross roller, roller lever, one-way roller lever	■ Top push, top push roller, top push cross roller, hinge lever, short hinge lever, roller lever, short roller lever, one-way roller lever, short one-way roller lever	■ Roller plunger, dome plunger, hinge lever, short lever, offset hinge, adjustable lever, large rubber roller	■ Roller plunger, dome plunger, short lever, adjustable lever, rod lever, spring rod, telescope arm
Enclosure Classification①	■ NEMA 1, 3, 4, 6, 12, 13 and IP67	■ Non-Sealed: NEMA 1 and IP50 ■ Sealed: NEMA 1,3,4 and IP65	■ NEMA 1,3,4,6,13 and IP67	■ IP66	■ IP66
Additional Info	■ C115 catalog page 5-46 ■ www.ab.com/catalogs	■ C115 catalog page 5-22 ■ www.ab.com/catalogs	■ C115 catalog page 5-56 ■ www.ab.com/catalogs	■ C115 catalog page 5-60 ■ www.ab.com/catalogs	■ C115 catalog page 5-65 ■ www.ab.com/catalogs

① Enclosures are designed and tested to the listed NEMA ratings (Ref. NEMA standard ICS 1-110)

43

Figure F.1: Excerpt from AB Catalogue for Limit Switches [1]

Performance

Performance CAT 33 with DC-motors

Actuator	Dyn. Load (N)	Speed (mm/s)	Current cons. (A)
CAT(R/L) 33...x1/C12C	3000	13-10	18
CAT(R/L) 33...x2/C12C	2000	24-20	18
CAT(R/L) 33...x4/C12C	1000	48-38	18
CAT(R/L) 33...x1/D12C	2400	11-7	16
CAT(R/L) 33...x2/D12C	1600	21-15	16
CAT(R/L) 33...x4/D12C	800	39-21	16
CAT(R/L) 33...x1/C24C, D24C	3000	13-10	9
CAT(R/L) 33...x2/C24C, D24C	2000	26-20	9
CAT(R/L) 33...x4/C24C, D24C	1000	48-35	9
CAT(R/L) 33...x1/C24CW, D24CW	3000	7-5	5
CAT(R/L) 33...x2/C24CW, D24CW	2000	13-8	5
CAT(R/L) 33...x4/C24CW, D24CW	1000	26-19	5
Performance CAT 33 with AC-motors			
CAT(R/L) 33...x1/E220C 50Hz (6μF), E380C 50Hz	3000	5	
CAT(R/L) 33...x2/E220C 50Hz (6μF), E380C 50Hz	2000	10	
CAT(R/L) 33...x4/E220C 50Hz (6μF), E380C 50Hz	1000	20	
CAT(R/L) 33...x1/E110C 60Hz (25μF)	2400	6	
CAT(R/L) 33...x2/E110C 60Hz (25μF)	1600	12	
CAT(R/L) 33...x4/E110C 60Hz (25μF)	800	24	

Max. Static load for actuator CAT 33: 4000 N.

Performance CAT 33H with DC-motors

Actuator	Dyn. Load (N)	Speed (mm/s)	Current cons. (A)
CAT(R/L) 33H...x1/C12C, D12C	1000	50-38	18
CAT(R/L) 33H...x2/C12C, D12C	600	100-80	18
CAT(R/L) 33H...x4/C12C, D12C	400	174-150	18
CAT(R/L) 33H...x1/C24C, D24C	1200	56-36	9
CAT(R/L) 33H...x2/C24C, D24C	900	113-79	9
CAT(R/L) 33H...x4/C24C, D24C	500	174-140	9
CAT(R/L) 33H...x1/C24CW, D24CW	1200	27-17	5
CAT(R/L) 33H...x2/C24CW, D24CW	800	60-35	5
CAT(R/L) 33H...x4/C24CW, D24CW	500	100-69	5
Performance CAT 33H with AC-motors			
CAT(R/L) 33H...x1/E220C (6μF)	1200	25-20	
CAT(R/L) 33H...x2/E220C (6μF)	900	50-37	
CAT(R/L) 33H...x4/E220C (6μF)	600	100-90	

Max. Static load for actuator CAT 33H: 3000 N.

Performance CAT 32B with DC-motors

Actuator	Dyn. Load (N)	Speed (mm/s)	Current cons. (A)
CAT(R/L) 32B...x1/C12C	4000	17-12	18
CAT(R/L) 32B...x2/C12C	2500	32-25	18
CAT(R/L) 32B...x4/C12C	1500	63-48	18
CAT(R/L) 32B...x1/D12C	3000	17-11	16
CAT(R/L) 32B...x2/D12C	2000	34-19	16
CAT(R/L) 32B...x4/D12C	1000	67-43	16
CAT(R/L) 32B...x1/C24C, D24C	4000	17-13	9
CAT(R/L) 32B...x2/C24C, D24C	2500	33-24	9
CAT(R/L) 32B...x4/C24C, D24C	1500	65-50	9
CAT(R/L) 32B...x1/C24CW, D24CW	4000	9-5	5
CAT(R/L) 32B...x2/C24CW, D24CW	2500	18-10	5
CAT(R/L) 32B...x4/C24CW, D24CW	1500	34-24	5
Performance CAT 32B with AC-motors			
CAT(R/L) 32B...x1/E220C 50Hz (6μF), E380C 50Hz	3500	6.5	
CAT(R/L) 32B...x2/E220C 50Hz (6μF), E380C 50Hz	2100	13	
CAT(R/L) 32B...x4/E220C 50Hz (6μF), E380C 50Hz	1300	26	
CAT(R/L) 32B...x1/E110C 60Hz (25μF)	3500	8	
CAT(R/L) 32B...x2/E110C 60Hz (25μF)	2100	16	
CAT(R/L) 32B...x4/E110C 60Hz (25μF)	1300	32	

Max. Static load for actuator CAT 32B: 5400 N.

Figure F.2: Excerpt from SKF Catalogue for Actuators [24]