DESIGN GUIDELINES FOR A MAGNETICALLY PROPELLED HOISTING SYSTEM

by

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Abstract

Magnetically propelled hoisting is a novel system for moving containers full of rock in underground mines. Current practice is to hoist these containers, called skips, to surface using a cable. In a magnetically propelled hoisting system, the cables are replaced by a tubular linear motor.

The research began with a detailed literature search on hoisting, magnetic levitation, pneumatic transport, and mining applications.

Virtual modeling, kinetic modeling, simulation, and analysis were used to formulate a number of design options. The project resulted in the construction of a testbed where future research into the concept of Magnetically-Propelled Hoisting can be continued. The testbed should enable analysis of: electrical delivery system, control system design, skip design, instrumentation configuration, speed-payload variation, multi-vehicles, and system orientation amongst other design criteria.

Some preliminary testing has indicated the following: achieved speed of 2m/s, horizontal through vertical motion of the skip, controlled motion of 2 skips, controlled acceleration, braking, and reversing of the skip.

A risk assessment shows that the hoisting system failure potential to be low and likely controllable. A 96% mechanical availability is likely. A preliminary economic assessment shows that a MagLev system can be competitive with a conventional hoisting system with similar capital and operating costs.

Several advantages over conventional hoisting were demonstrated regarding economic and mine mill integration. In addition the research has highlighted a number of potential problems with the concept that may hinder its acceptance by the mining industry.

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1 Introduction

From 1997 through to 2003 UBC's Mining Engineering Department pursued investigations into novel material handling systems. In the proposed system, the cables used to lift a skip in a conventional hoisting operation were removed. The use of a tubular linear motor was proposed as a new means to propel the skips to surface.

The project evolved from brainstorming to address how such a system might look and work. Computer generated virtual models were built from these ideas to demonstrate and illustrate the design options. The concepts from the virtual models were then tested through a parallel physical modeling and testing program. The final result has been the creation of a 14' x 7' demonstration testbed for use in future research.

The work completed at UBC and the projected application of the system is presented in this thesis which is organized as follows: Chapter 2 describes the concept of a linear motor. Chapter 3 presents a brief overview of hoisting systems in underground mining. Chapter 4 contains the virtual reality work that was undertaken to provide alternative design options. Chapter 5 gives the conceptual approach to the design work. Chapter 6 presents details on the construction of the testbed system. Chapter 7 presents the test work performed on the testbed while Chapter 8 gives an analysis of the preliminary test work. Chapter 9 presents an overview of the elements that would likely make up such a system in an operating mining situation. Chapter 10 carries out a risk assessment of potential problems that might occur with each component. Chapter 11 contains a detailed economic assessment of this approach to indicate the likelihood of its financial viability. Chapter 12 details the potential application scenarios and limitations while Chapter 13 discusses requirements for future work. Finally conclusions and recommendations are given in Chapter 14.

2 Linear Motors

Linear motors were first conceived in the mid 1800s and the first design was patented in 1890. (Gieras, 1994) Linear motors have been around for a long time, but they have found relatively few applications when compared to their cousin, the rotary motor.

2.1 Theory

A linear motor can be thought of as taking a conventional rotary motor, cutting it along its axis, and unrolling it to form a flat motor. The outer stationary portion of the rotary motor is now called the primary, while what was the rotor is now referred to as the secondary. This is illustrated in Figure 1 below:

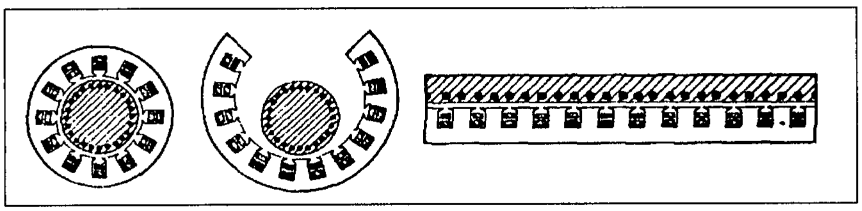

Figure 1: Unrolling of a Rotary Motor to Make it Linear. (Laithwaite, 1971)

By this analogy it can be seen that any rotary motor can be treated in the same fashion to form a linear motor. The most common applications have been synchronous, induction and reluctance motors due to their ability to operate with no electrical connections to the secondary.

2.1.1 AC vs. DC

Linear motors can be designed to operate on either AC or DC power, the same as a rotary motor. Most applications tend to operate on 3-phase AC power as this is typically the form in which electricity is delivered.

Although DC is less common, there are some reasons to consider its use. DC power can be used with permanent magnets in the secondary. The magnets give the potential for a more efficient design due to the permanent magnet's ability to produce a strong magnetic field with no energy input. DC gives easy speed control and a cheaper power supply for small motors as a variable frequency AC source is not required. The main disadvantage is that the operation of DC powered linear motors tends to be jerky, leading to potential oscillations. Power also needs to be converted to DC before it can be used.

The choice between the two power types is one of the important decisions in the linear motor design.

2.1.2 Asychronous vs. Synchronous

Linear motors are also classified as either synchronous or asynchronous. In terms of the systems described in this research, in a synchronous motor the skip will be moving at exactly the same speed as the progressing magnetic wave. In this type of system, every skip will move at exactly the same speed. This will help to prevent collisions between skips. If a skip requires more force to move than the drive is providing, then it will fall out of synchronization and stop in the tube. This will have to be dealt with in the design of an overall control system.

The most common form of asynchronous motor is an induction motor. In an induction motor the progressing magnetic wave induces a current in the skip which attracts it to the wave. To keep the skip moving the waves must continually pass the car to induce the current. In an induction motor the car will always move slower then the magnetic wave, the difference in speed depending on the force required to move the skip. In this system, skips with different weights or friction will travel at different speeds, giving the potential for collisions. In an induction drive, the skip does not have any risk of losing synchronization and stopping within the tube.

2.1.3 Applications

Although linear motors were conceived at the end of the 1800s, it wasn't until the mid-1900s that they began to find commercial application. The first high-powered linear motor was built by Westinghouse in 1946. The motor was able to accelerate a 5 tonne military plane to 185km/h in 4.2s (Body 1999). Since the 1950s, linear motors have been used in many different applications from exotic space and nuclear applications to public transit and roller coasters to weaving looms and door openers.

2.1.4 Tubular vs Flat Linear Motors

It is possible to take a flat linear motor and roll it into a tube form as shown in Figure 2 below:

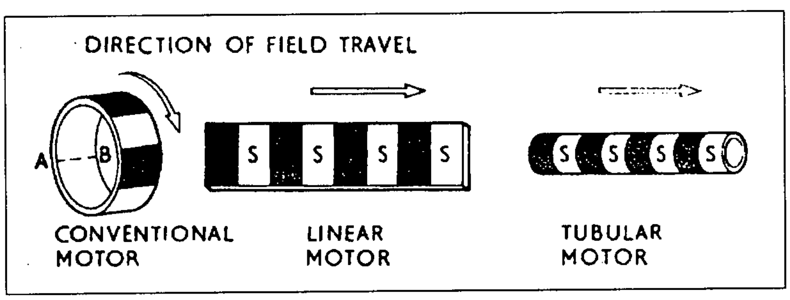

Figure 2: Forming a Tubular Motor. (Laithwaite, 1971)

A number of significant advantages in terms of efficiency and ease of construction are made to the design by rolling a linear motor into a tube.

The first advantage is that the windings in a tubular linear motor are simple spools. This makes them easy to wind and place. In a flat linear motor the windings are loops that are stacked together making them more difficult to build and much harder to repair.

When a linear motor is working there is a strong attractive force between the secondary and the primary. In a flat linear motor all of this force is in one direction. In a tubular linear motor the force is evenly distributed over 360° so the forces cancel themselves out.

The third advantage occurs from the fact that in a flat linear motor the wires within the slots are the only part of the winding doing any useful work. The wire used to connect the wires in the slots does not contribute to the force generated by the motor and is referred to as "end turns". In a tubular motor the end turns are not required as the slot now forms a ring. This saves on the amount of copper required and also makes for a more energy efficient design as there is no longer any power dissipated by the copper wire in the end turns. For these reasons a tubular motor has been designed in this research. This is shown in Figure 3 below:

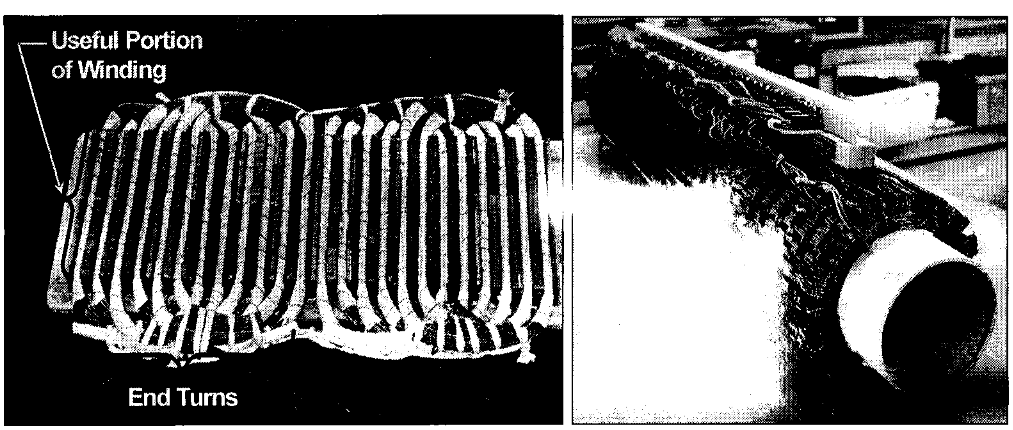

Figure 3: Flat vs. Tubular Linear Motors.

2.2 Linear Motor Propulsion Systems

Since their origin, designers have been trying to use linear motors to propel vehicles. Several systems have characteristics that might lend themselves to being applied to the mining industry. The two systems with the most potential are MagLev trains and linear motor propelled vehicles.

2.2.1 MagLev Trains

MagLev trains have been under development since the 1960s. Both Germany and Japan have been developing competing systems for the world's MagLev market. Both systems rely on magnets for zero contact levitation to reduce friction.

The Japanese system uses superconducting magnets on the car that are repelled by electromagnets in the track to lift the car by repulsion. The design has improved stability but the superconducting magnets cost millions and their cooling system costs millions more (Beaty, 2002). The track is estimated to cost $92 million US per km. (Monorail Society, 2003)

The German train uses an electromagnetic suspension system where electromagnetic attraction is responsible for levitating the car. The levitation requires constant adjustment to remain stable. This requires the use of sensors and a feedback controller.

The world's first commercial MagLev train based on the German approach is currently being commissioned in Shanghai, Peoples Republic of China, to link the Pudong airport to Shanghai's subway system, a distance of 30km. The train is designed to operate at 430 km/h making the trip in less than 8 minutes. The project cost US $1.2 billion and took 2.5 years to go from feasibility to operation. (Xinhua 2002)

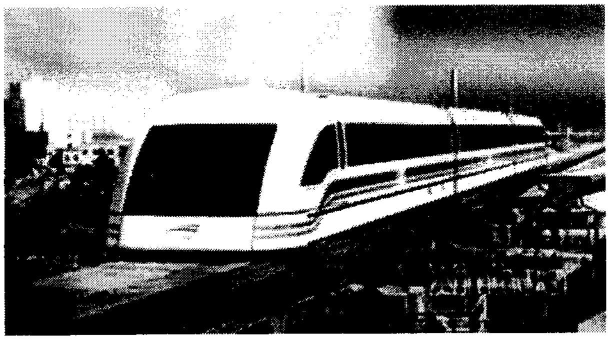

Figure 4: Shanghai MagLev Train. (Xinhua 2002)

2.2.2 Linear Motor Propelled Vehicles

The idea of light transit trains being carried by wheels on rails and propelled by linear motors was patented in 1905 by Zehden (Body, 1999) although it was decades before this became practical. Today these systems are quite common in light transit, airports, and amusement rides. Such trains are much simpler and cheaper to build than MagLev systems, but they typically have a top speed of under 100km/h and do not have the benefits of a noncontact suspension and guidance system.

One example of this technology is Vancouver's SkyTrain which has been in operation since 1985. Each car in the SkyTrain system is equipped with two bogies; each bogie has a linear motor to power the train. The system uses less energy per passenger-kilometer than any other rail system in North America. (Translink, 2002) The linear motor installed on one of the SkyTrain cars is shown in Figure 5 below:

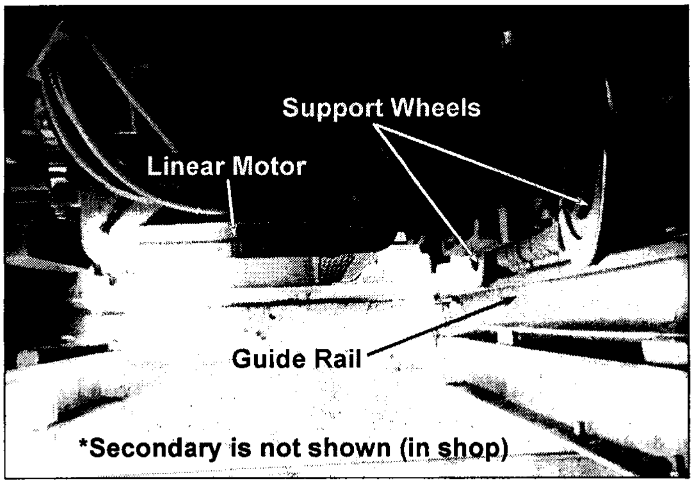

Figure 5: SkyTrain's Drive System.

3 Hoisting Systems

In mining, hoisting systems for the vertical movement of material have been around for a long time with little substantial change or innovation occurring over the years.

3.1 Conventional Hoisting

Conventional hoisting has changed little since the origins of mining. The general theory is that a bucket ("skip") is lowered into the mine on a rope, rock is put into the bucket, and then it is pulled back to surface to be dumped, as shown in Figure 6 below:

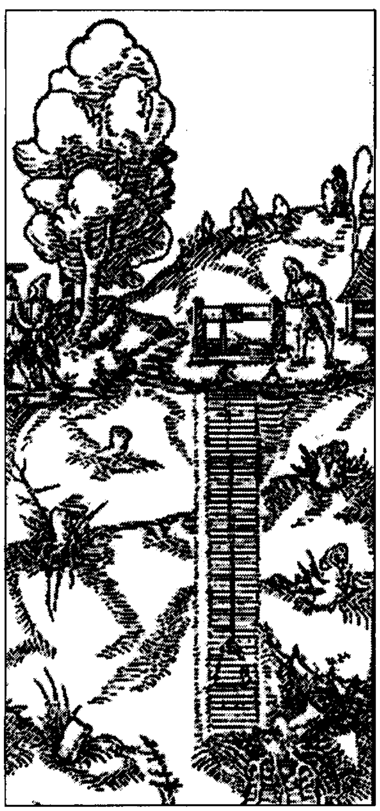

Figure 6: Historical Hoisting. (Agricola, 1550)

Modern hoisting has evolved to move higher tonnages from ever increasing depths. Currently hoists are beginning to be required to extend beyond their technical limitations. As mining depths increase beyond 3000m, the cables used to move the skip are no longer strong enough to carry their own weight, let alone the weight of the skip and rock.

Siemag has recently completed a drum hoist system able to be used to a depth of 3,000m. The drum assembly is 33m long, 11m wide, and powered by two 12,000 kW AC motors. The entire assembly weighs over 800 tonnes. A portion of this assembly is shown in Figure 7 below:

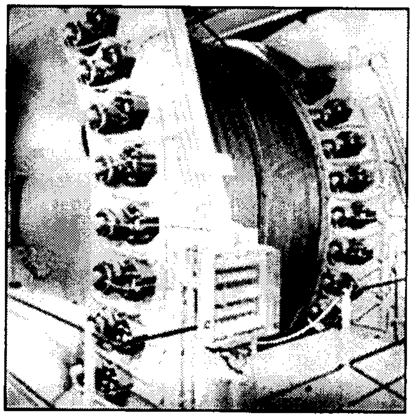

Figure 7: Seimag Hoist.

With improvements in cable manufacturing and hoist control systems, conventional hoists are unlikely to significantly improve in their ability to deal with increased depth to which they can be used. The ability of a hoist to reach added depth is only part of the problem. The other part is the time it takes for the skip to be hoisted from underground to surface. As depth increases it takes an increasing amount of time to hoist one load of rock to surface, reducing the capacity of the system. This can be overcome by adding duplicate hoisting systems but there is an obvious added cost with this approach.

3.2 Deep Mining Concept

In 1996 a deep mining research project was started in South Africa to support the mining companies who are looking at a potential 50million oz gold resource grading 13g/t located between 3000 and 5000 meters below the surface (Diering, 2000). Part of this research was directed at material handling and hoisting.

One concept that came out of this research was that of a hybrid hoist. The idea was to use a conventional hoist with a flat Linear Induction Motor (LIM) to gradually assist with the load below a depth of 2000 meters. (Cruise & Landy, 2001) From their work they were able to demonstrate that the hybrid concept, "...had tremendous promise in increasing the operating depth, the safety, and rope life of existing hoist systems. The lower capital expenditure and the fact that this technology can be retrofitted to existing shafts make the hybrid-hoist system a viable alternative to conventional hoisting techniques." (Cruise & Landy, 2001)

The hybrid concept will decrease the length of the LIM and its associated cost. As depth continues to increase, the LIM will need to become increasingly more powerful to a point where it is suspending the entire weight of the skip and payload. The LIM at this point will have to be able to propel a loaded skip.

The concept proposed in this thesis research is different from the proposed South African hybrid system in that the load is distributed within many skips that follow each other so the LIM does not require the same extent of power. A second major advantage is that with no cables attached to the skip, it is free to negotiate corners, allowing loading to occur close to the face and then to travel directly to surface and or dump. This can significantly reduce transfer points and potentially some underground equipment requirements.

Hybrid-hoists make a good first step in introducing linear motor propelled hoisting into the mining industry and gain valuable implementation and operating experience. The approach allows existing hoists to be upgraded to include a LIM.

3.3 MagLev 2000

MagLev 2000 is a forward-looking American company that is proposing a second-generation MagLev system based on the Japanese MagLev approach. They do not currently have any prototype or test models built. They are projecting systems based on technology that has not yet been developed, but historical trends indicate that the technology will become viable. As a result, they are proposing a system able to move semi-trailers on ultra high speed trains traveling at 3,200 km/h (MagLev 2000, 2001) with significantly lower capital and operating costs than the current state of the art.

MagLev 2000 has proposed a system for the mining industry called MagLev for Mining or M4M. M4M has proposed two systems for the mining industry. The first is for an open pit mine. The material would be loaded into special cars that would be propelled straight up the pit wall to its desired dump location. A second system is proposed for underground mining. In this application an inclined shaft would be developed following the ore body. The levitated cars would be loaded underground then propelled up the shaft to the surface. Theses two concepts are illustrated in Figure 8 below:

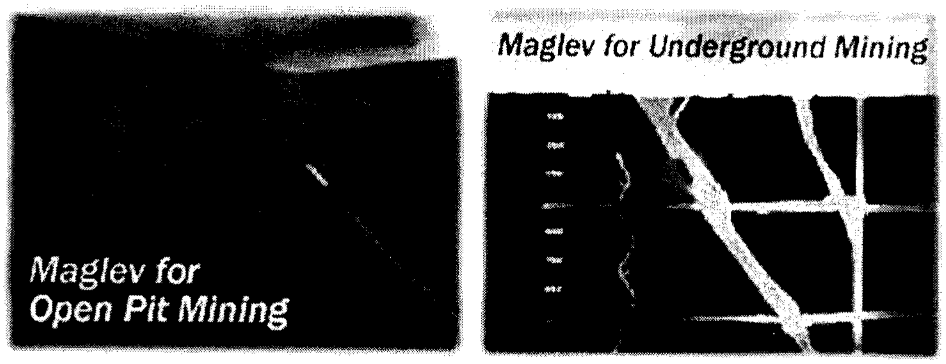

Figure 8: MagLev of Florida Concept Mining Applications. (MagLev 2000, 2001)

MagLev of Florida 2000 claims that their underground system would operate in a conduit or tube on a track with a capital cost between \$0.3M and \$0.625M US per kilometer. The entire system will be relocatable to another mine using installation robots.(Morena, 1999) If the M4M system were to be constructed it "...would greatly reduce the volume of waste rock to be excavated, and consequently the cost of the product ore. In addition, the MagLev for Mining system would greatly reduce the number of engine powered underground ore carriers, reducing both the operating cost and the pollution of the miner's air supply." (MagLev 2000, 2001)

MagLev Florida has proposed a system running 16 miles from Titusville to Port Canaveral in Florida. The projected capital cost is \$600M US with an operating cost of \$14M US. The cost of the track is projected to be under \$11.5M US per km (Morena & Haddad, 2002) or about 13% of the cost of the Japanese system. MagLev of Florida is projecting that they will be able to have this system operating by 2009.

3.4 Laurentian University Hoisting System Concepts

Researchers at Laurentian University have come up with several innovative hoisting systems in the past couple of years. The first one is a Pneumatic Capsule Pipeline System where air pressure is used to move cars inside a tube. The second system looked at using superconducting magnet technology to levitate cars and then apply a linear motor to propel the cars that transport material in a mine.

3.4.1 Pneumatic Capsule Pipeline System

In this system a fan is used to pressurize air flowing through a tube. Capsules were introduced at the bottom of the tube and were blown up the tube to the top. The system was found to be feasible. This idea, in common with the concept proposed in this thesis employs small cars traveling in tubes. (Muldowney, 2001) There is no mention in the paper of how the containers will be loaded or dumped.

3.4.2 MagLev

The system proposed by the Laurentian University researchers has looked at installing MagLev to move material vertically in an existing underground mine. The technology is based on that proposed by Florida MagLev. The work examined the application of MagLev technology at Inco's Creighton Mine located in Sudbury Ontario Canada. In this application the system was required to move 1360 tonnes from a depth of 2,040m. The system was designed with cars each having a capacity of 7 tonnes. The track is projected to cost between $0.45M and $1M US per km (Krueger, 2001) or about 1% of the cost of the Japanese system with a skip cost of $0.45M US. Both systems use the same technology, similar sized cars, the main difference being that the mining version is running vertically instead of horizontally.

The MagLev system is estimated to have a capital cost of 20% of a conventional cable hoisting system, 50% of the conventional operating cost, and consume less than 50% of the conventional energy. (Krueger, 2001)

3.4.3 USBM Magnetically-Levitated Coal Car

In the early 1990s the US Bureau of Mines (USBM) launched a project, "Magnetic Levitation Transport of Mining Products". The project focused on designing a magnetically levitated container that was suspended by an array of permanent magnets. Propulsion of the container was provided by placing the container into a specially designed tube and using air pressure to blow the container to its destination. The container was guided by sensors controlling the current to four electromagnets located on each corner of the container's base.

The USBM was successful in building a levitating base 1.19m long, 0.5m wide, and 5cm thick. The platform was able to levitate a 153kg payload and traverse a 2.4m long track with noncontact, frictionless movement. The USBM found that, "This innovative materials transport system design appears as a promising means to improve the safety and to reduce the cost of underground mining and materials handling." (Geraghty, Wright, & Lombardi, 1995) Figure 9 below shows the USBM prototype model:

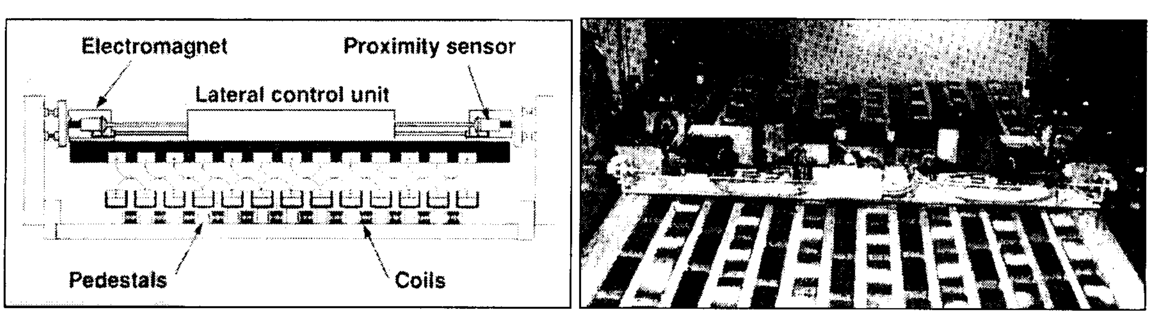

Figure 9: USBM Magnetically-Levitated Container.

Shortly after the initial stage of this project was completed the USBM was disbanded. To date the author is unaware of anyone who has followed up on this research.

3.4.4 Freight Pipelines

There are numerous research projects being conducted around the world on moving freight within pipelines. The freight is typically loaded into cars that have traditionally been propelled by either pneumatic or hydraulic means. Currently there are a number of projects investigating using linear motors to propel the cars.

Sumitomo Metals has been building pneumatic capsule pipelines since 1983 (Roop, 2000). In October 2001 they commissioned their first vertical system to move earth from an underground tunneling machine. The system is 1m in diameter, and transports $30m^3$ of material an hour a distance of 33m vertically. Sumitomo plans on applying their system to depths to 1000m deep (Mining Technology, 2001), see 10.

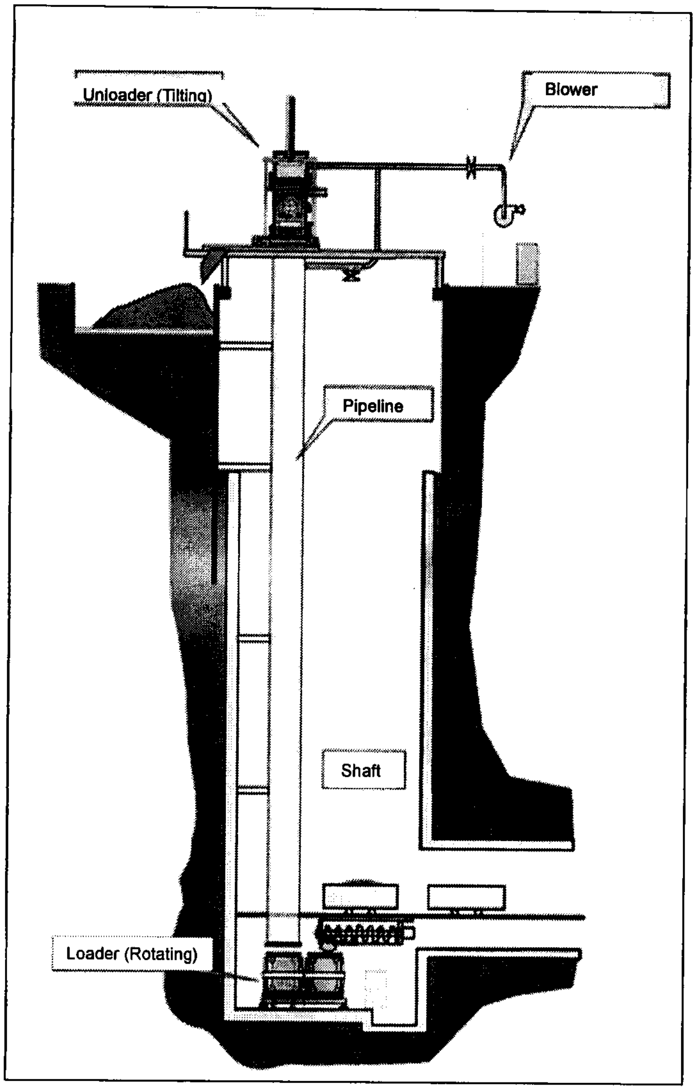

Figure 10: Sumitomo's Vertical Pneumatic Hoisting System. (Kosugi, 2001)

The department of Aerospace Engineering at the University of Minneapolis is a second group working on a freight pipeline system. The goal of their system is to move freight that is currently being trucked around the USA using an underground pipeline powered by a linear motor. The reason for using a linear motor over a pneumatic one is that, "Pneumatic pipelines suffer from short haul range limits, high noise level, and poor energy efficiency." (Zhao & Lundgren, 1997) The use of linear motors is always more efficient than that of pneumatic blowers according to these authors. (Zhao, Lundgren & Sampson, 2000)

A company called Magplane, a spin off from MIT, has constructed a demonstration project of a freight pipeline powered by a linear motor at IMC-Global a phosphate mining company in Lakeland, Florida. The demonstration line is built of 0.61m diameter pipe 275m long. The cars have a 2.4m long wheelbase and carry a 270kg payload. (Montgomery, Fairfax, & Smith, 2001)

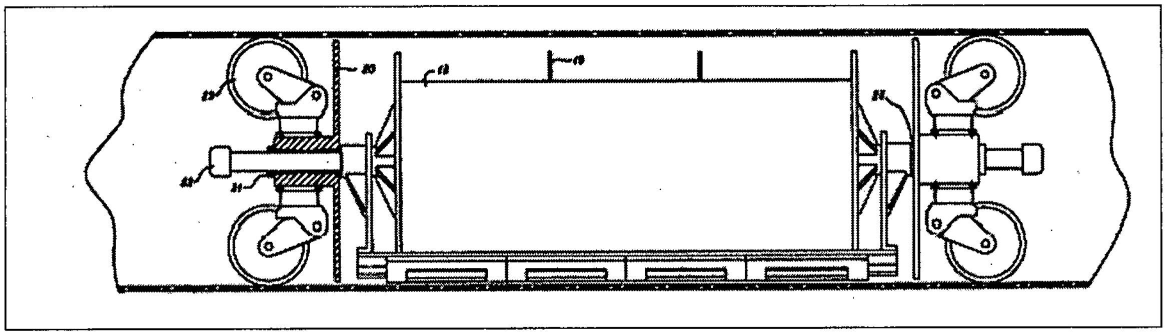

Figure 11: Long Section Showing Magplane Car Inside Tube.

Table 1 gives cost estimates for an economic study that was completed to transport 9 million tonnes per year a distance of 48 km.

Table 1: Freight Pipeline Costs (Montgomery, Fairfax, & Smith, 2001)

Capital Costs	**$M US**
Right of Way	$ 2.7
Pipeline	$ 13.2
Vehicles	$ 15.6
Magnet Assemblies	$ 7.8
Motor Windings	$ 2.3
Load / Unload Stations	$ 2.8
Power Units & Control	$ 5.8
Total	**$ 50.1**
Operating Costs	**$M / Year**
Insurance & Property Tax	$ 0.8
Power	$ 1.1
Maintenance	$ 1.8
Labour	$ 1.5
Total	**$ 5.2**

The above table shows that the projected tube cost is about $0.32M / km which is a small fraction of the cost of a MagLev system with similar capacity. Magplane found, "…electromagnetic drive systems can effectively compete with (surface) truck and rail transport, and in selected cases, with slurry pipelines and conveyor systems." (Montgomery, Fairfax, & Smith, 2001)

This application is predominantly a horizontal application and the cost of the drive components would be expected to increase significantly when applied to a vertical application.

3.5 Origins of the UBC System

The UBC magnetically propelled hoisting system originated from an observation of a semaphore on a 1949 British Ford Prefect. The semaphore is powered by a steel plunger inside a solenoid. When the solenoid was energized, the steel plunger is pulled down raising the indicator arm as shown in Figure 12 below:

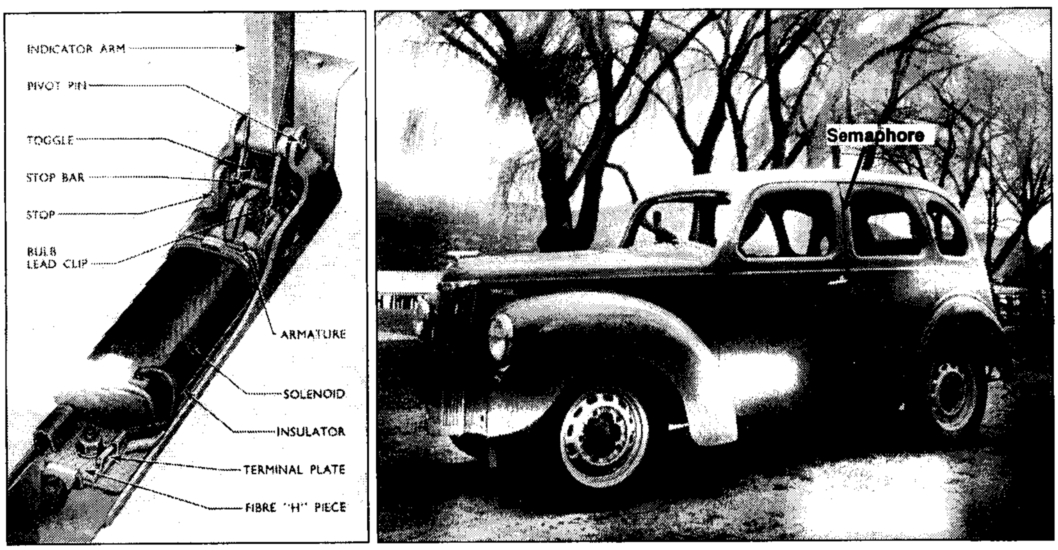

Figure 12: British Ford Prefect.

The idea was that if a series of coils were joined end to end, the steel plunger could be moved from coil to coil by controlling the energizing sequence of the coils. To make a material handling system, the plunger could be hollowed out and filled with rock, and then the rock could be moved up the series of coils or hoisted to surface. The first prototype model was built in 1998 and showed its solenoid heritage. This prototype is shown in Figure 13 below:

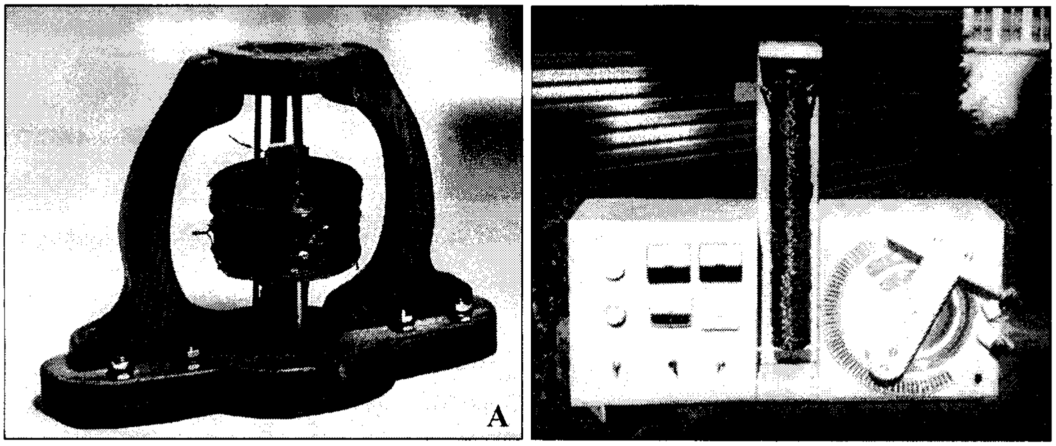

Figure 13: First Prototype Models. (A) Prototype 1, (B) Prototype 2.

The picture on the left was the first prototype model used to determine the capability of a coil. The picture on the right was the second model built to demonstrate the concept. The left side of the model housed a power supply to deliver 5 amps at 15 volts DC. The right side housed a 100-pole switch that controlled the energized portion of the central solenoid. A steel "skip" was placed in the center of the solenoid and could be observed to follow the energized portion of the solenoid. The model was able to lift about 100 grams over a distance of 60cm. The system was interesting but hardly practical, it required too much copper wire, too much energy, and the control system was too cumbersome.

4 Virtual Reality Designs

The UBC magnetically levitated hoisting system started with the question - what could you do if the skip no longer required cables for it to move? Almost immediately there were a myriad of ideas of what the system could look like or do. To facilitate communicating these ideas, computer generated virtual reality simulations were created.

The virtual reality simulations were created using Ray Dream Studio, Poser, and Bryce from Meta Creations and Carrara Studio from Eovia. These software packages enabled the creation of both still images and full motion animations. With some post processing some designs were rendered into 3D stereoscopic animations. The evolution of the concept is shown in the following sections.

4.1 Mark 1 System

The first proposed system perceived carrying material inside steel containers from an underground loading facility to the surface dumping location. With the removal of cables the ability to negotiate corners was also a component of the idea.

4.1.1 Skip Design

Material would be carried in axially loaded cylindrical steel containers. On the bottom of the skip a permanent magnetic band would be located to react with the linear motor on the tube.. The Mark 1 skip is shown in Figure 14 below:

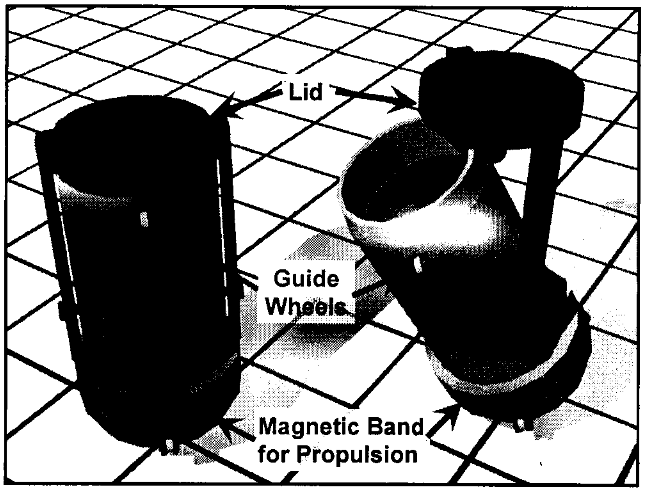

Figure 14: Mark 1 Skip Design.

4.1.2 Propulsion

The Mark 1 system was conceived as being guided by rails and propelled by a segmented coil built around the tube. It was conceived that by controlling the sequential energizing of the coil segments, the skip could be made to move. This is shown in Figure 15 below:

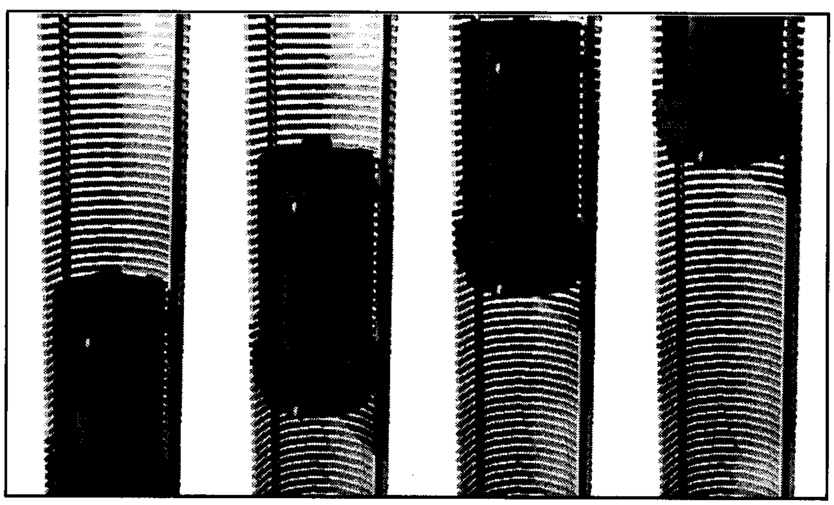

Figure 15: Mark 1 Propulsion System.

4.1.3 Loading system

The loading system was considered as a modular system that could be relocated underground to follow mining progress. Stopes would be developed in a conventional fashion. The material mucked from the stopes would go through a single crushing stage to reduce the top size enough to allow the material to be loaded into a 1 meter diameter vessel. The material would be stored in a small surge bin to adsorb the batch loads delivered by the scooptram. From the surge bin the material would be weighed for placement into a skip.

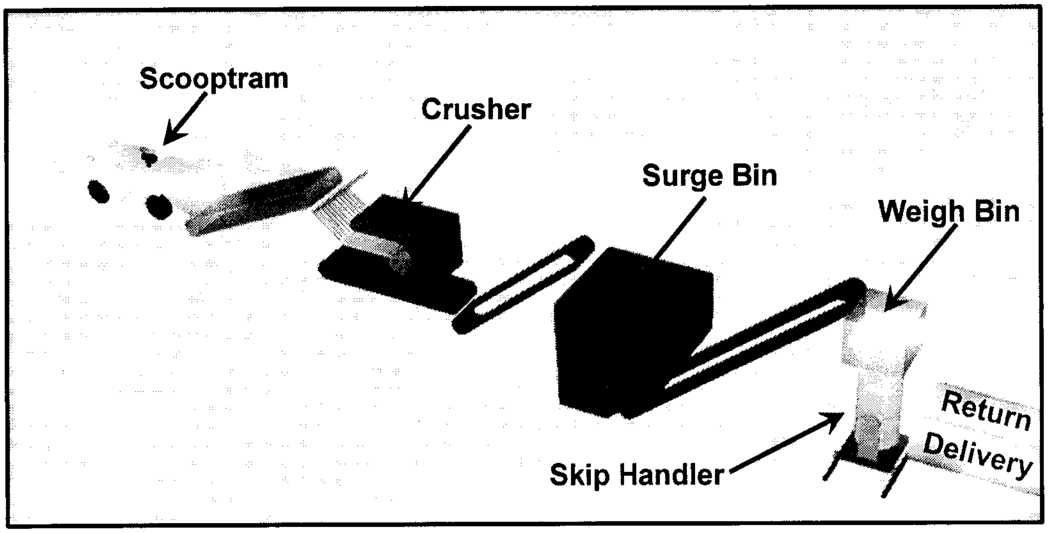

Figure 16: Mark 1 Loading System.

In this design a skip handler is responsible for manipulating the skip for loading. The handler is designed to slide to the return tube and rotate to catch the skip. It would then slide and rotate to align the skip under the weigh bin for loading. The skip handler would then close the lid of the skip and align it with the delivery tube to be propelled to surface.

4.1.4 Dumping System

To dump the skips on surface, a hydraulic lift is mounted on a movable platform above the ends of the drive tubes. When a skip reaches the end of the tube it would be grabbed by the lift and transferred to the return tube. Due to the attachment to the lid, the loaded skip would become unstable during transfer and invert, dumping its load. This concept is show in Figure 17 below:

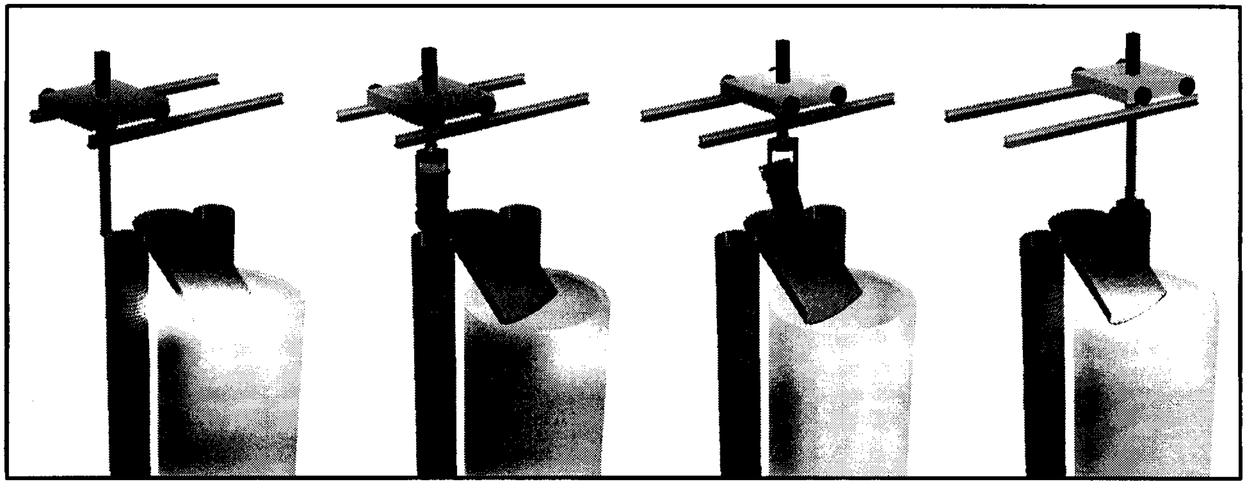

Figure 17: Mark1 Dumping Sequence.

4.2 Mark 2 System

The Mark 2 system is a complete change in application. Instead of looking to improve the material handling system of an underground mine, the design examined the movement of material in a large open pit mine. The original idea was to look at a large mountain coal mine with valley dumps as a hydroelectric operation, where instead of capturing energy from falling water, the energy would be captured from falling rock. In this case, millions of tonnes of waste rock are deposited in a valley dump 500 meters lower in elevation.

4.2.1 Car design

Crushing would likely consume more power than would be generated by the system, so the cars had to be large enough to handle most of the material without crushing. The cars are designed with a permanent magnet array intended to levitate the loaded car. Propulsion would to be provided by an electric motor spinning a generator type device in the magnetic field of the track. Power rails would have to be used to collect power on the down trip and provide power to return the car. The proposed car is shown in Figure 18 below:

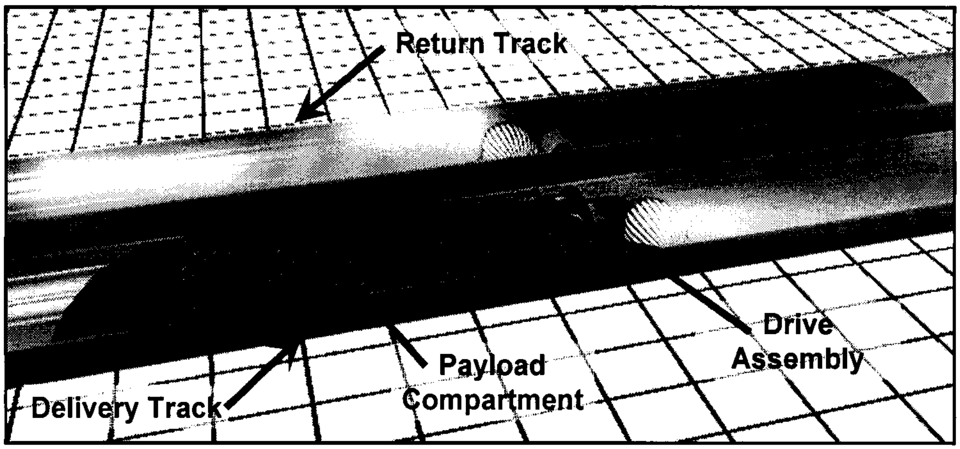

Figure 18: Mark 2 Car Design.

4.2.2 Loading Station

Since crushing is not required before loading the cars, the loading system is much simpler than in the Mark 1 case. Conventional haul trucks would dump the blasted material through a grizzly into a hopper. An armored conveyor is located at the bottom of the hopper to transfer the material to the car. The car arrives from the return track, is slid into position for loading, and then departs on the delivery track. The loading system is shown in Figure 19 below:

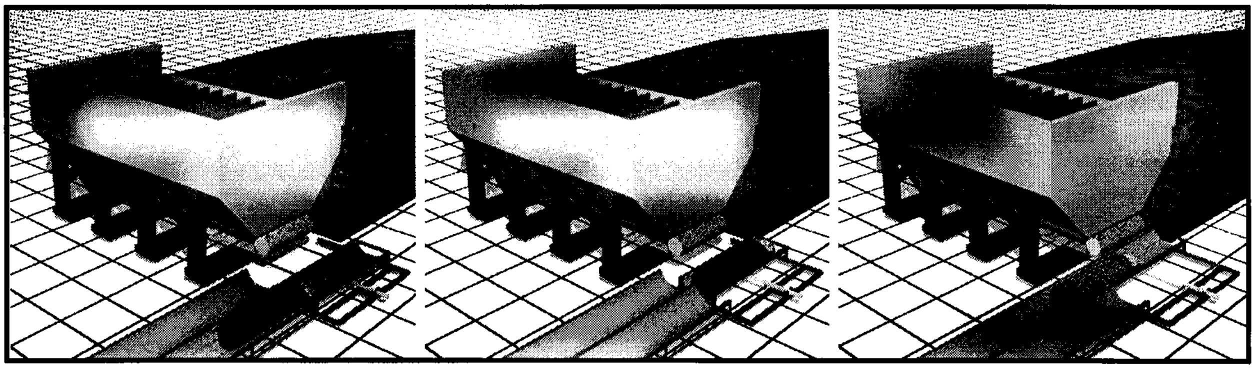

Figure 19: Mark 2 Loading System.

4.2.3 Dumping Station

A dumping system was proposed in which the cars roll over allowing the material to be dumped. The skip is then righted and aligned with the return track to travel back to the loading station. This system is shown in Figure 20 below:

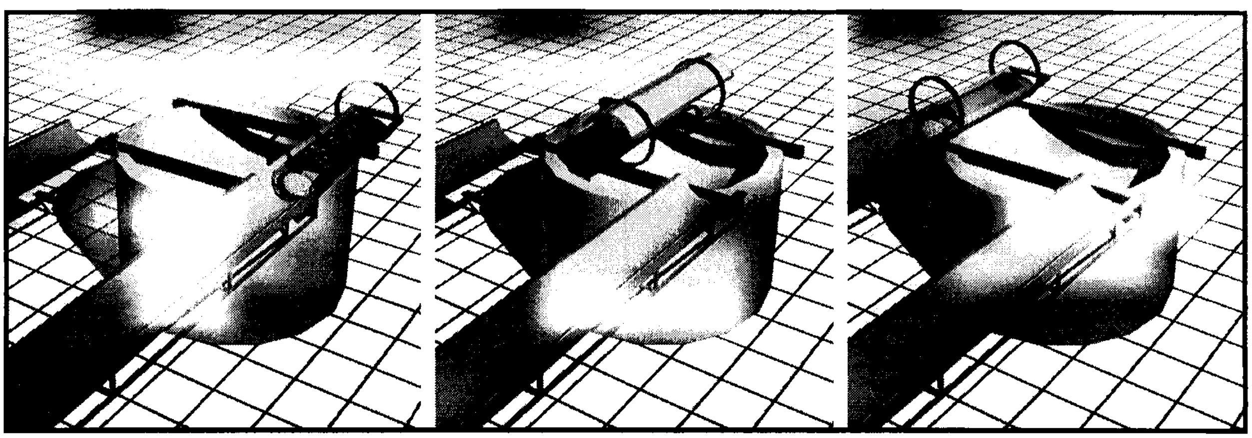

Figure 20: Mark 2 Dumping System.

4.3 Mark 3 System

The Mark 3 system returned to underground material handling. Magnetically propelled material handling systems will be more competitive in an underground mining environment where their small size and lack of emissions should provide the most benefit.

4.3.1 Skip Design

In the Mark 3 system, the permanent magnet arrays were retained from the Mark 2 design and other changes were made to the skip to simplify loading and dumping.

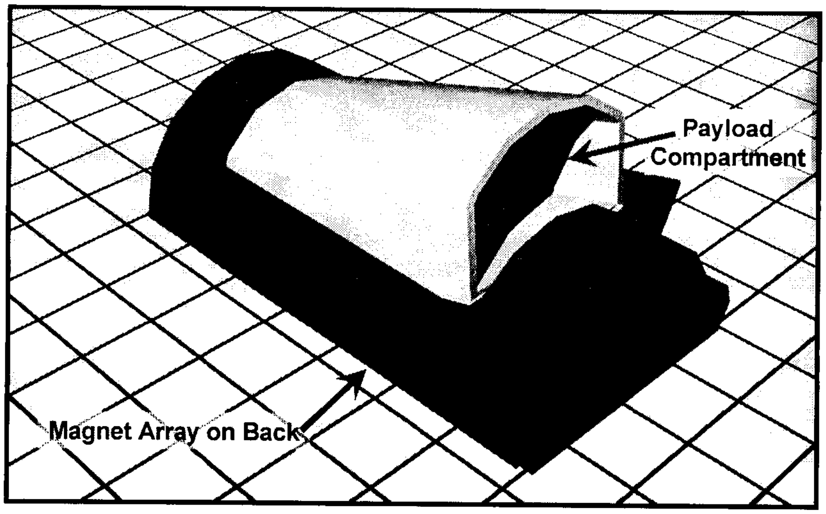

Figure 21: Mark 3 Skip Design.

4.3.2 Loading system

The front end of the Mark 3 loading system is much the same as the Mark 1 system. The system comprises a grizzly, crusher, surge bin, weigh bin, and loading chute. The main difference between the two systems is the replacement of the skip handler with a sloping curved track. As the car descends along the curved track, it is stopped and loaded. The Mark 3 loading system is shown in Figure 22 below:

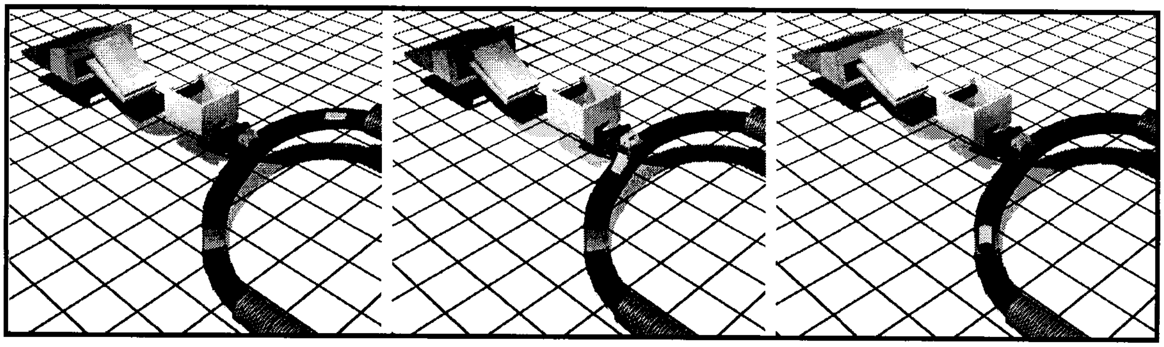

Figure 22: Mark 3 Loading System.

4.3.3 Dumping System

To dump the cars, a system is envisioned in which the car would be automatically inverted, the payload compartment opened, and the contents allowed to fall out. The skip would then be returned to the tube for its journey back to the loading station. This concept was never actually modeled as the Mark 4 design was beginning to take shape even while the Mark 3 system was being created.

4.4 Mark 4 System

The Mark 4 system was based on the Mark 3 concept. The goal of the Mark 4 system was to simplify the skip design to allow for easier loading and dumping.

4.4.1 Skip Design

While the virtual modeling was being conducted, physical models were also being built and tested. One was an automated model of the Mark 3 loading facility which is described in detail in Appendix I. During testing it was observed that the opening compartment would bind from small amounts of spillage during loading. As a result the skip in the Mark 4 design was altered to have an opening lid instead of an opening compartment. The second change was to give the skip an elliptical cross section. This was done to provide more volume to the skip, and to maintain a near flat surface for magnetic levitation. The skip design is shown in Figure 23 below:

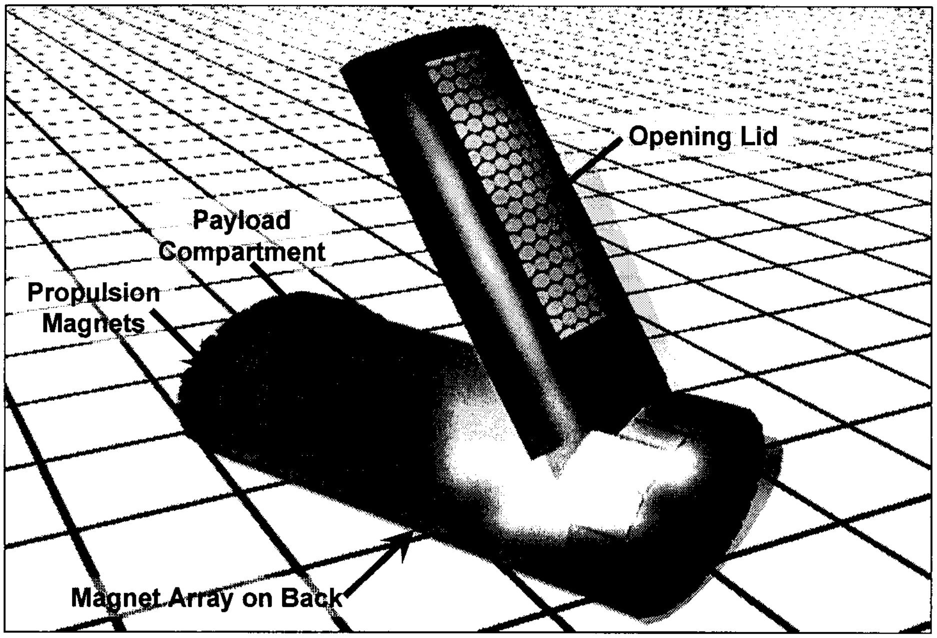

Figure 23: Mark 4 Skip Design.

4.4.2 Loading system

The main difference between the Mark 3 and the Mark 4 loading systems is how the skip is handled at the loading facility. In the Mark 4 system the skip is caught by a mechanism that opens its lid, rotates it for loading, closes the lid, and then aligns it with the delivery tube for its return trip to surface. This design is show in Figure 24 below:

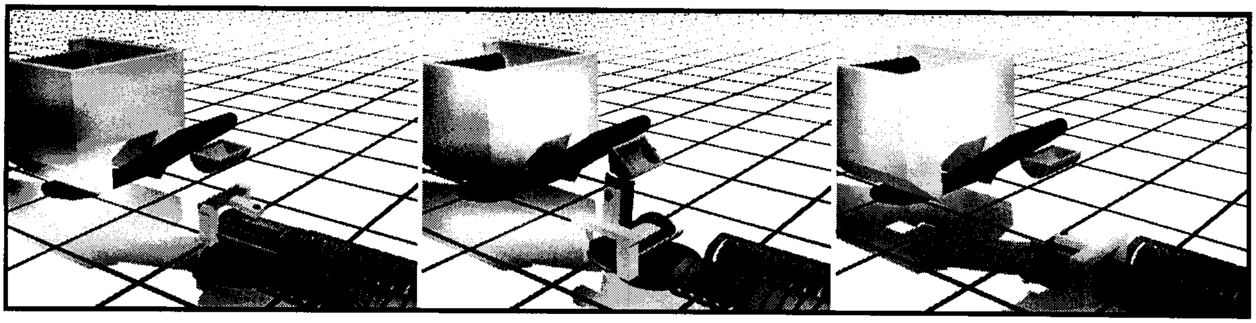

Figure 24: Mark 4 Loading System.

4.4.3 Dumping system

The initial concept for dumping the skips was to place them in a rotary dumping mechanism. The car would be inverted, its lid would open, and its contents would fall out as shown in Figure 25 below:

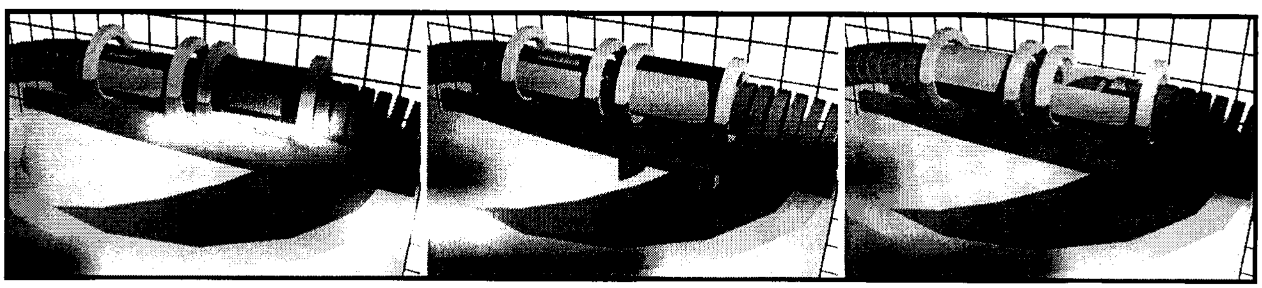

Figure 25: Mark 4 Dual Skip Rotary Dump System.

When the rotary dump system was virtually modeled it appeared to be complicated and fairly slow at dumping the skips. A new system was conceived; one which would be continuous and have the ability to handle a steady stream of skips. In this system the skips are inverted by bending the track, then the lid is opened while over a cutout portion of the track, and the contents spill out while the skip traverses the section of track. The skip is no longer required to stop so the capacity of the dumping station greatly increases. The lid would be designed to close automatically once the ore has fallen out. With this design the skip speed can be maintained throughout the dumping activity. Figure 26 below illustrates this concept.

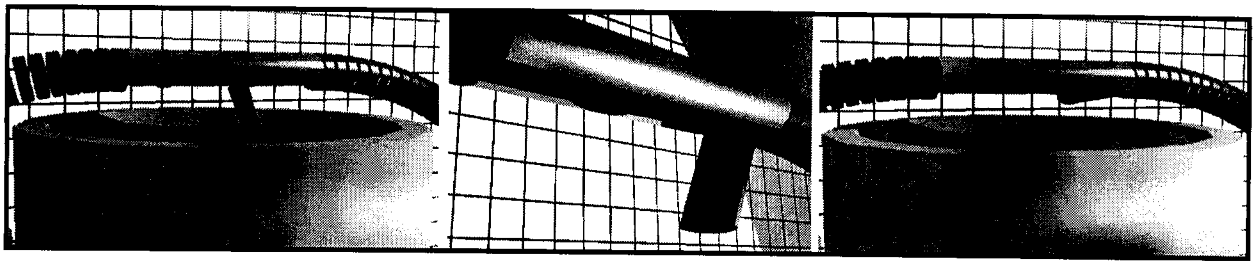

Figure 26: Mark 4 Continuous Dumping Station.

4.5 Mark 5 System

The Mark 5 virtual system was designed after completion of the Mark 3 physical model and track testing. See Appendix 1 for more information on the test work done on the Mark 3 loading facility.

One aspect of the Mark 3 loading system model was to use a permanent magnet levitation system on the track. The levitation system would be used to support and guide the skip while the linear motor propelled it. As a result of the construction and testing of the magnetic array, it was found that the permanent magnet levitation and guidance system was going to require an additional active control system to keep the skip stable in the track. As a result the idea of permanent magnets to support and guide the skip was discarded from the design and wheels and rails returned for simplicity. The design also focused on attempting to make the loading and dumping operations continuous in order to simplify and increase capacity.

The Mark 5 design was the first approach that moved away from a solenoid-based system and introduced a linear motor. Both systems are similar in operation with the linear motor having a ferrous core. The core is used to conduct the magnetic flux generated by the windings to the air gap between the skip and the tubular motor. By conducting the magnetic flux to the air gap, much more of the flux is used to move the skip. This makes the design more powerful and efficient than in the solenoid design. As a result iron components are shown between the windings.

4.5.1 Skip Design

The skip design is drastically changed from the Mark 4 system. The skip is no longer magnetically levitated, so there is no reason not to use a cylindrical cross section. This will make fabrication simpler and cheaper. Wheels were added to the skip to support and guide it while in transit. The Mark 5 skip is shown in Figure 27 below:

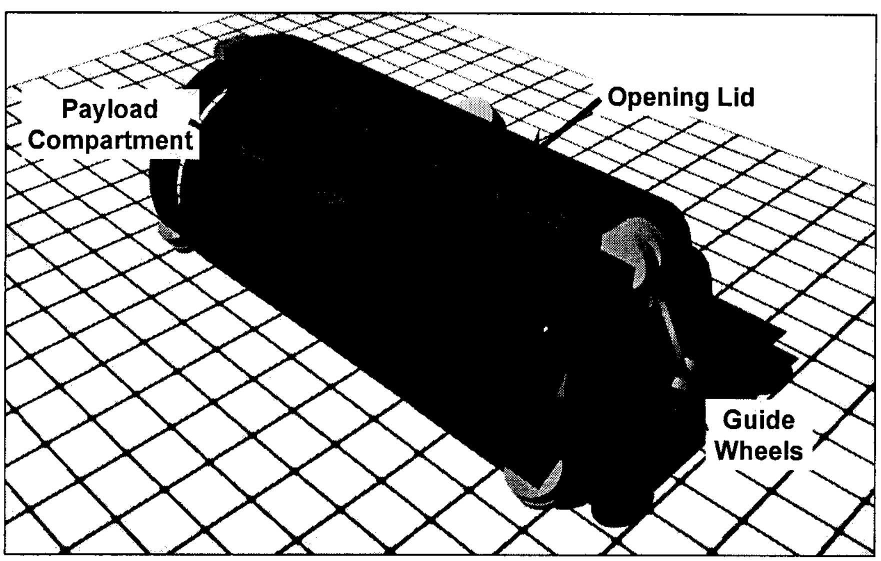

Figure 27: Mark 5 Skip Design.

4.5.2 Loading System

The Mark 5 loading system was the first design to use a continuous loading system in which the skip never stops. The track simply curves through the loading station. The lid is swung open and the car travels under the loading chute. Then the lid is closed as the skip starts its ascent to surface. The loading sequence is shown in Figure 28 below:

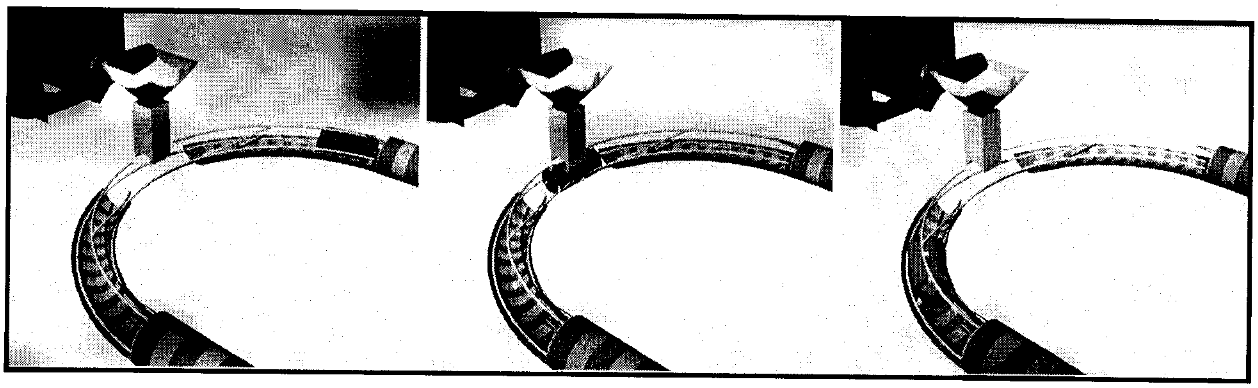

Figure 28: Mark 5 Continuous Loading System.

The looping of the track through the loading facility makes the track very wide, and it is unlikely that it would fit into a drift. This might be a possible option if two parallel drifts were being developed. Since this would be quite a limited application, a new design had to be created to make the loading system smaller.

A new design was created to allow the loading system to fit inside a single drift. A skip handler was reintroduced to enable the skips to turn a much sharper corner.

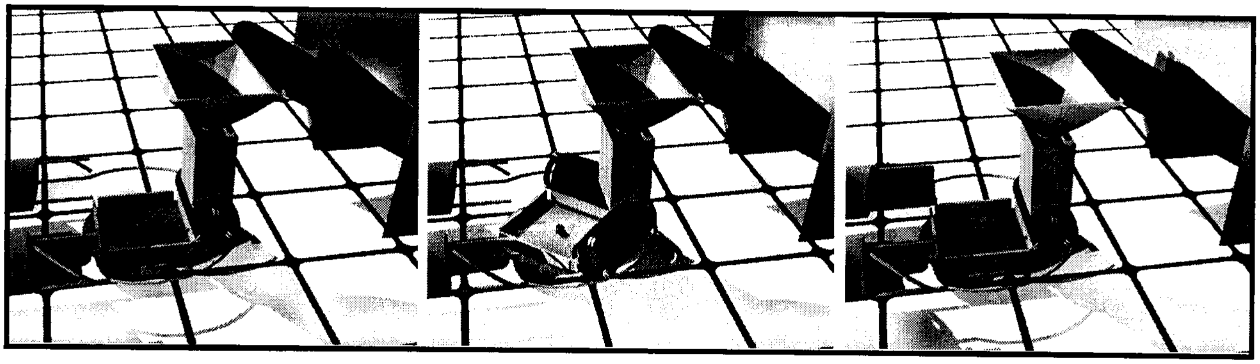

Figure 29: Mark 5 Semi-continuous Loading System.

4.5.3 Dumping System

The dumping system followed on from the Mark 3 design. The skip is inverted as it traverses across the dump point. The skip rotates in a fashion that causes the lid to open automatically and discharge its load. A reverse process is used to close the lid before the return trip back underground. The idea of multiple dump points also appeared in the Mark 5 dumping system. The idea was that by controlling when the door opened, different materials could be hoisted from different locations to different silos or stockpiles within the same hoisting facility. This can be beneficial to keep development waste separate from ore, or to separate different types or grades of ore. The Mark 5 dumping system is shown in Figure 30 below:

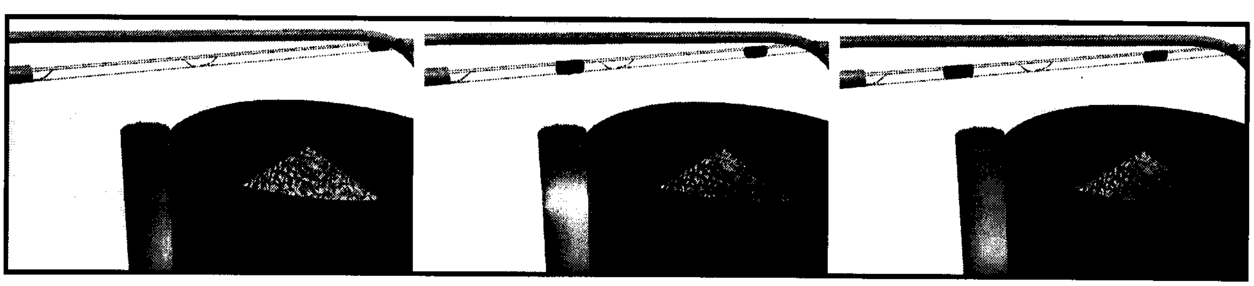

Figure 30: Mark 5 Dumping System.

5 Conceptual Approach to the Problem

The following sections will describe how the problems with specific components of the system were addressed.

5.1 Track

Initial work focused on keeping the system as simple and conventional as possible. For this reason the Mark 1 system used wheels on rails to support and guide the skip inside the linear motor. The maintenance and replacement of the rails inside a small diameter tube was a constant concern and a potential Achilles Heel to the proposed system. The work conducted by the USBM seemed to have the solution: use an array of permanent magnets to suspend the skip.

The Mark 2 and Mark 3 systems envisioned a curved array of magnets that would center the car on the track much like sliding down a water slide. A section through the Mark 3 design is shown in Figure 31 below:

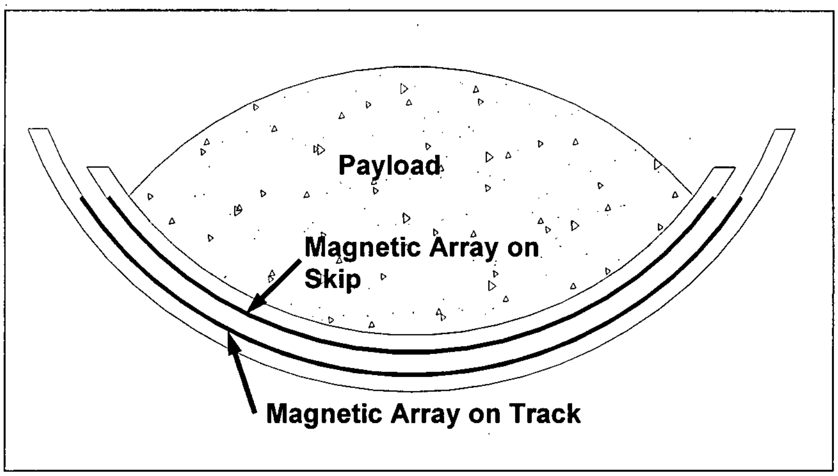

Figure 31: Section Through the Open Pit Design – Mark 3 System.

When the first models of this concept were created, it was immediately apparent that this would not be a stable arrangement. The car would require an external force to remain centered in the track. This follows from Earnshaw's theorem which states that an object can not be held in a stable equilibrium by purely magnetic forces (Geraghty, Wright, & Lombardi, 1995). Lateral instability can be overcome through the use of a mechanical or electromagnetic guidance system (Bahmanyar & Ellison, 1975). However, since the goal was to simplify the design, the magnetic arrays were rejected and rails returned to guide the skips.

The rails were a problem with respect to maintenance and added concern for skip derailment. The final design was to build the drive around a replaceable tube. The skip's wheels would ride on the inside of the tube instead of rails. The advantage of this arrangement is the potential for less frequent maintenance, and the elimination of skips derailing while in transit. This introduced the problem of no longer knowing the orientation of the skips when they entered the loading or dumping facilities. It was considered that there would be fewer problems designing a system to deal with reorientation of the skips rather than dealing with rails and derailment.

5.2 Skip

The design of the skip was heavily influenced by the design of the track. The skips were always envisioned to be a low cost disposable conveyance for the mined material. Initial designs had the skips being end-loaded making it resemble a pail. Subsequent designs switched to an axially loaded design to simplify the design and operation of the loading and dumping facilities. The axially loading designs were eventually abandoned returning to an end-opening design since this will be able to handle larger fragments and have a greater percentage of the skip filled with material. An additional consideration is when the skip is traveling vertically the door will be on the top preventing spillage.

5.3 Loading & Dumping Systems

The loading and dumping systems were under constant evolution; attempting to find a solution that would be simple, reliable, and as continuous as possible. These designs were not however developed past an initial concept.

6 Construction of the Prototype Models

To bring some reality to the design process, a series of test models were created. The following table shows the design parameters for the model:

Table 2: Model Design Parameters.

Drive Design	Value	Reason
Tube Diameter	4”	To facilitate access to interior
Section Length	~36”	To enable manual handling of section
Number of Turns	250	Experimentally determined from previous models
Width of block	½”	Availability of cheap banding for material
Width of Winding	½”	To match block width
Winding Wire	Copper, single build	Standard electrical component.
Gage of Winding	22 AWG	Lower current requirement and still strong enough to stand up to manual handle and construction.
Skip Design		
Material	Steel Tube	Existing material available
Diameter	3.7”	To allow clearance for cornering
Length	4.75”	Minimum length required to support steel sleeve.
Power (per winding)		
Voltage	0–50 VDC	Adjustable depending on required power.
Current	0-8 A	Varies depending on Voltage

6.1 Schedule

Construction of the prototype models started in September 2001 and took just under 22 months to complete. The Gantt chart shown below in Figure 32, gives the time line for the different stages of this project.

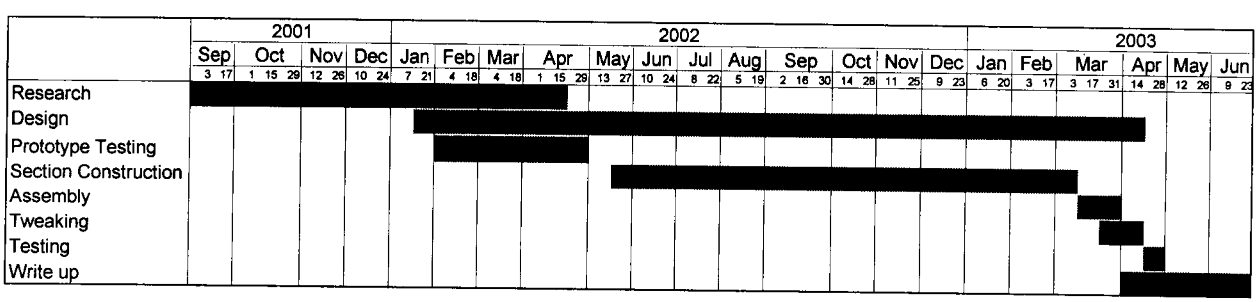

Figure 32: Construction Time Line.

6.2 Tube/Shaft

The sections that comprise both the track for the cars and provide propulsion and braking for the cars were built using standard PVC pipe. The first test section was built using 3" ID pipe but this size was discarded as it was too small to provide internal access to the section. The model was thus built using a 4" ID pipe. The increased diameter pipe allowed access but also added 33% more materials and labour to construct. Construction started by cutting 5 km of ½" wide by 0.02" thick steel banding to designed lengths to build the iron core. To accomplish this task, a machine was developed to autonomously perform the measuring and cutting of the pieces. Figure 33 below shows this machine.

Figure 33: Banding Cutting Machine.

Pieces of banding 10 and 25mm in length were alternated and glued to form blocks, see Figure 34 below:

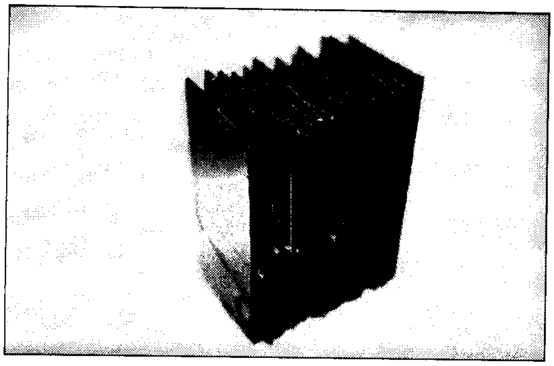

Figure 34: Finished Block.

The 10mm length was based on the height of the windings, the 25mm length was required to extend high enough above the winding to allow the blocks to be connected along the axis of the tube.

A total of 6,048 blocks were built. Initially the blocks were glued to the center tube to act as a form when winding the copper wire as shown in Figure 35 below:

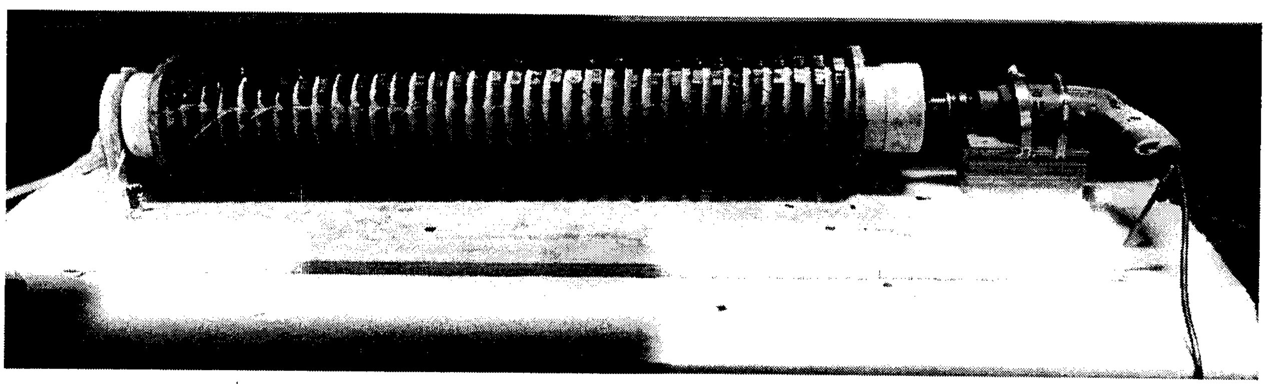

Figure 35: Center Tube with Blocks Glued into Place.

Next, 250 turns of 22 gauge copper wire were wound into the gaps between the blocks. In the straight sections the entire tube was spun to wrap around the wire. With curved sections, a different procedure was required. To wind the curved sections, a second machine was built to support the section while a moving arm wrapped the copper wire on to the section. The arrangement for winding the copper wire is shown in the following two photographs:

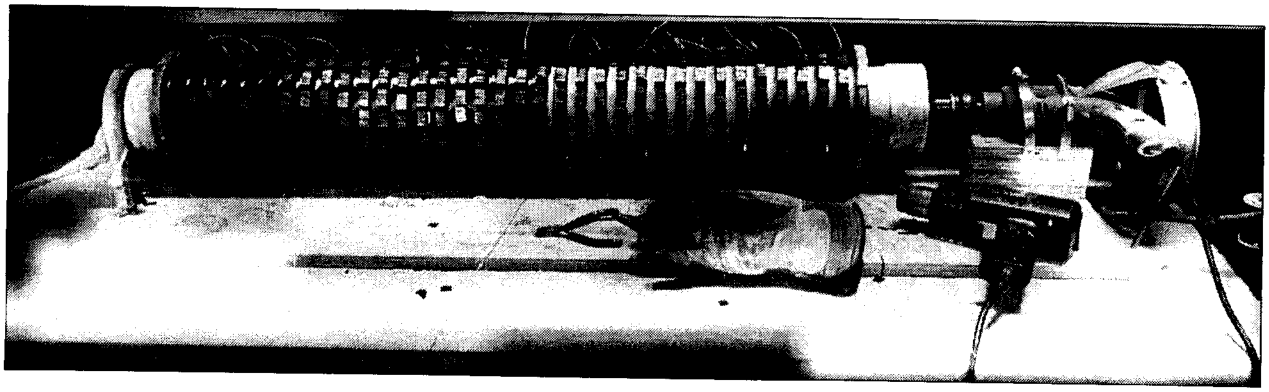

Figure 36: Winding Copper Wire onto Straight Sections.

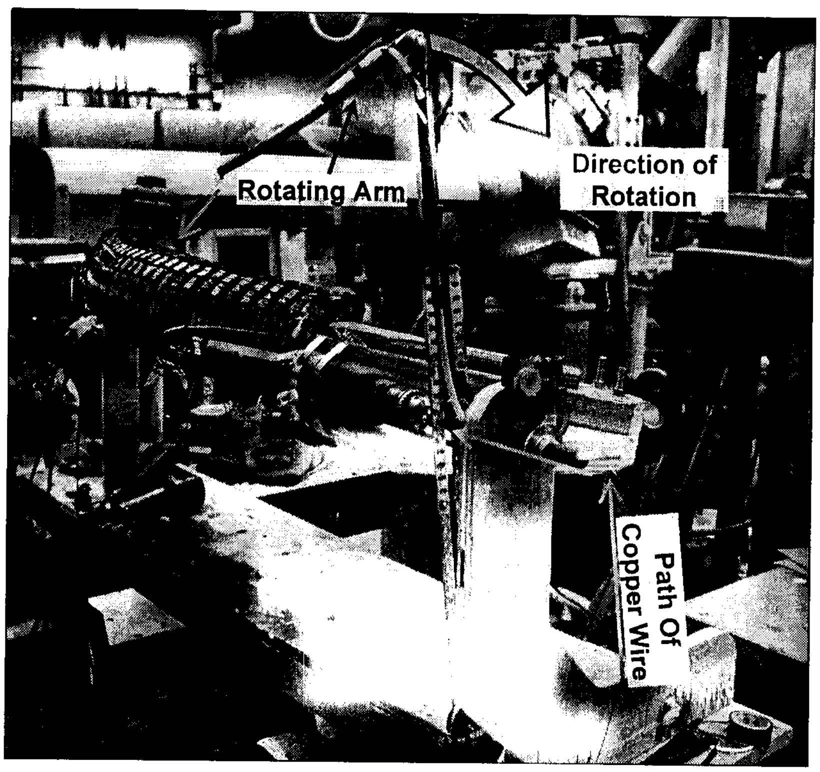

Figure 37: Winding Copper Wire onto Curved Sections.

Once the windings were in place, the blocks were connected along the axis of the tube. The straight sections used 12" pieces of banding while the curved sections required 4" pieces to follow the curve. Installing the banding on to the sections is shown in Figure 38 and Figure 39 below:

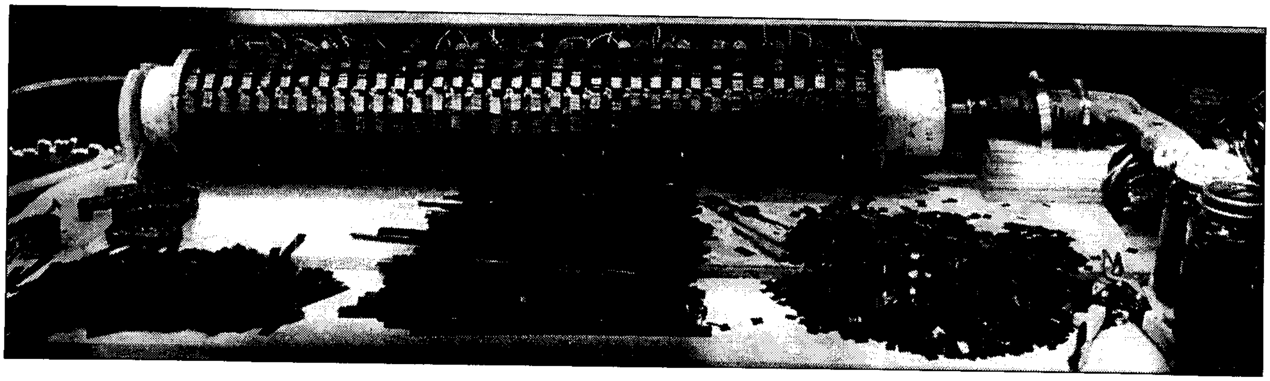

Figure 38: Installing Banding on a Straight Section of Track.

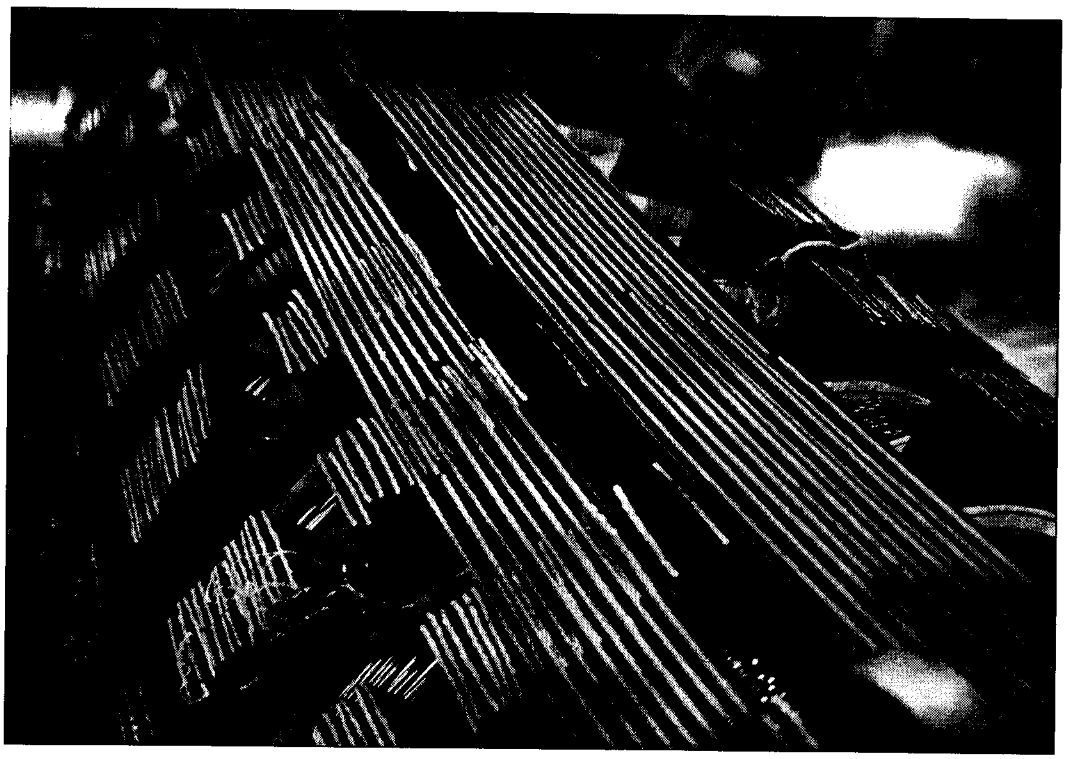

Figure 39: Installing Banding on a Curved Section of Track.

Once all of the banding was in place the entire section was saturated in varnish to bond the windings and banding together.

To control the motor, a switching signal is combined with a car position signal to produce a signal to energize the power transistor. This circuit is built onto a fiber glass board that was connected to the section once all of the steel was in place. A section with the control card in place is shown in Figure 40 below:

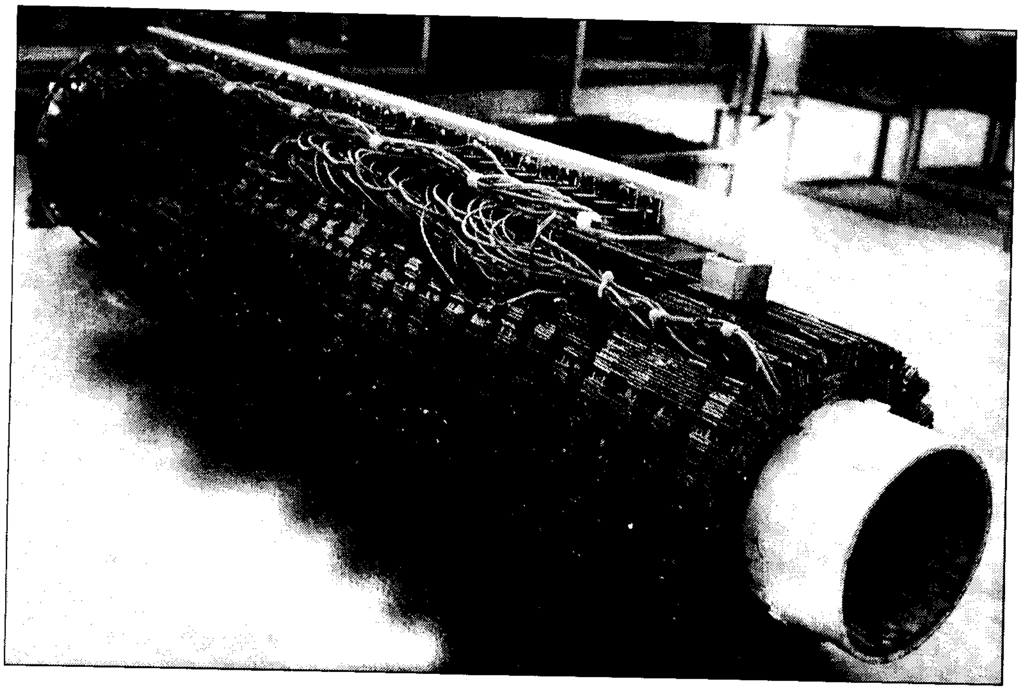

Figure 40: Straight Section with Control Card Attached.

This process was repeated until four 36" long straight sections were completed, two 12" long sliding straight sections, and eight 45° corner sections were completed. The completed sections awaiting final assembly are shown in Figure 41 below:

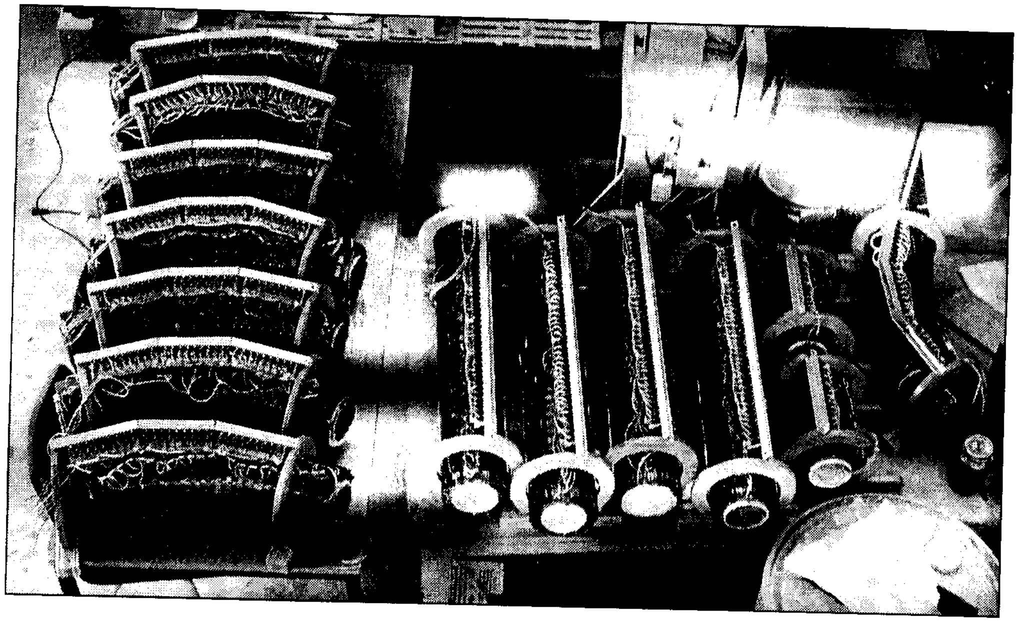

Figure 41: Sections Awaiting Final Assembly.

Once the sections were assembled, a support backing was built to allow the completed loop to be orientated in any direction. The conceptual design is shown in Figure 42 below:

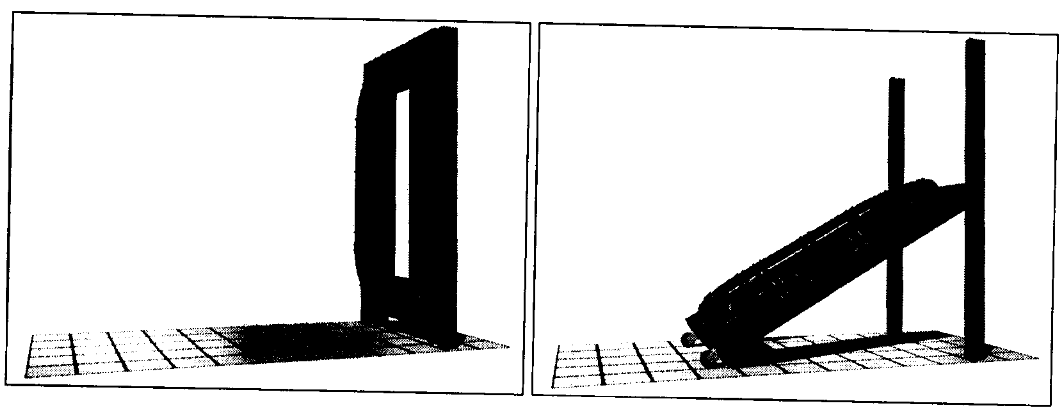

Figure 42: Frame Design.

The frame was designed to be rigid to limit the amount of flexing that would occur as the orientation changed. If the frame flexed too much, a section might be thrown out of alignment and the car would be unable to pass from section to section. To make the frame rigid, eight 2x8s run the length of the frame, with four layers of 5/8" plywood reinforcing where the support pipe passes through the 2x8s. The frame is shown in Figure 43 below:

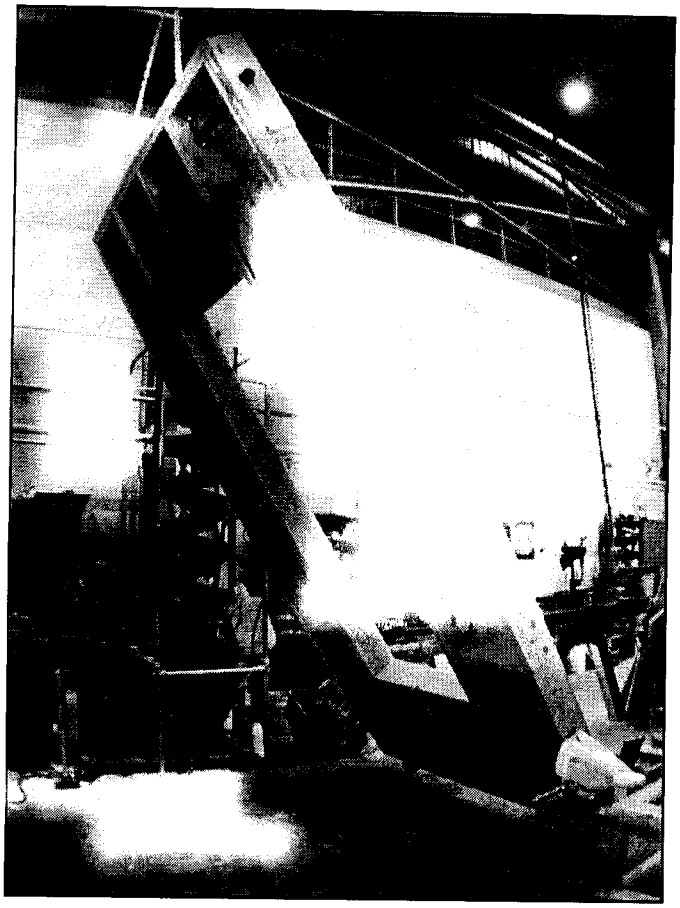

Figure 43: Constructed Frame for the MagLev Hoisting Testbed.

Once the frame was built and painted, the sections were laid out for final mounting.

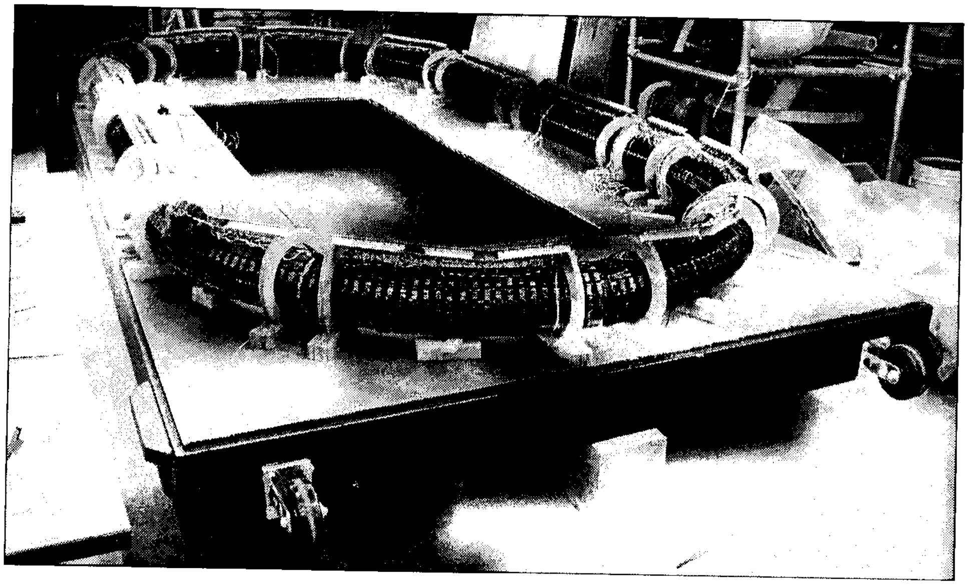

Figure 44: Sections Ready for Mounting.

With the supports bolted to the frame, the sections were shimmed into alignment. When the sections were all aligned, the gap between each section and its support was filled with a micro-fiber filler to bond and secure each section in place as shown in Figure 45 below:

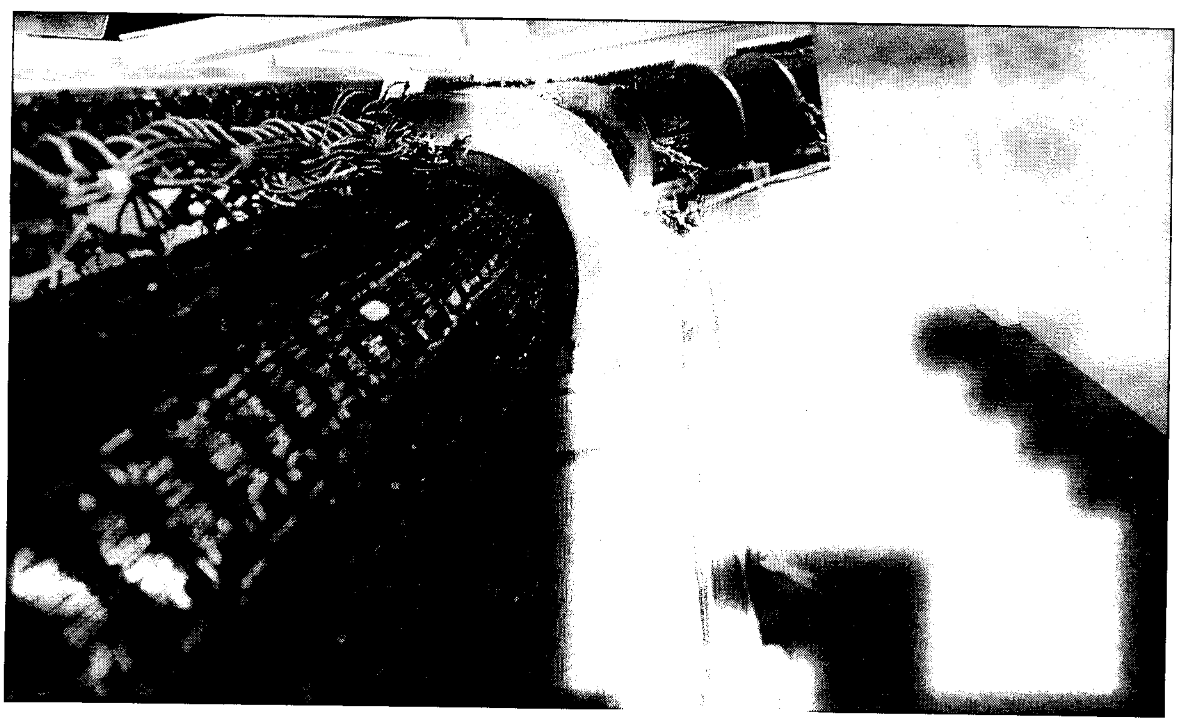

Figure 45: Mounted Sections.

With the sections mounted to the frame, the final wiring of the tube and assembly of the testbed cover were completed to finish the model as shown in Figure 46 below:

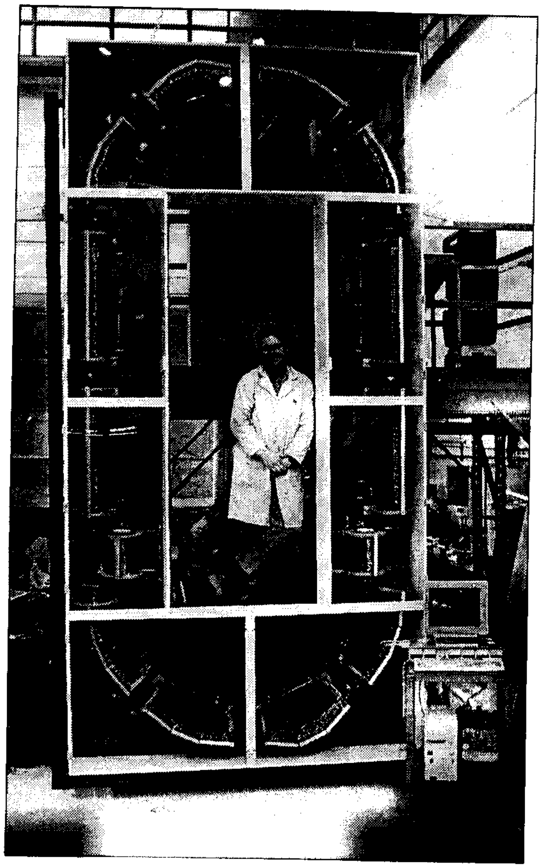

Figure 46: Final Assembled MagLev Testbed.

6.3 Skip

This testbed model was not designed to actually move any material. As a result, the skips did not have to be designed to hold any material for transportation. The skip vehicles essentially consisted of a stainless steel tube to interact with the tubular linear motor with 6 wheels attached to allow the skip to move smoothly around the track. The two skips created and used for this test model are shown in Figure 47 below:

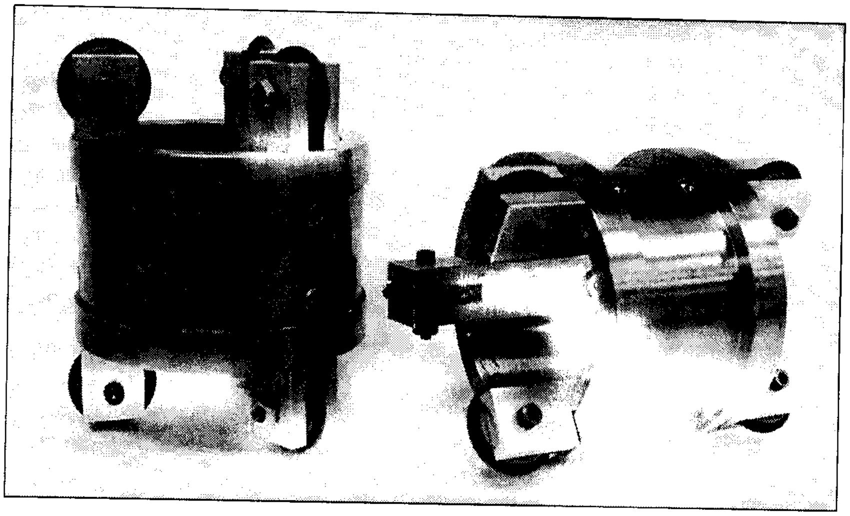

Figure 47: Test Skips.

The line drawings of the skip are included in Appendix V.

6.4 Control System

The control system consists of three separate components. The first component is an adjustable DC power source made from a modified DC welding unit. The second component is a transistor wired in series with each individual coil on the tubular motor. This allowed for a small control signal to be used to switch the coil on and off at the desired time. The control signal was generated by the third component – a computer – which switched on and off a 12V DC signal. The direction and speed of energizing the sequence of coils is controllable using this PC computer. The control system is shown in Figure 48 below:

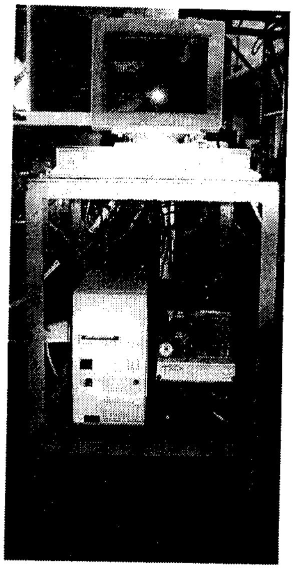

Figure 48: The MagLev Testbed Control System.

6.5 Instrumentation

The model uses two types of sensors. First infrared LEDs are located along one side of the tubular motor. Exactly across from these LEDs, infrared sensitive transistors are placed so that as a skip passes between the LED and the transistor, a detection signal is generated which can also record the speed of the skip. By examining sequential sensors, the skip acceleration can be measured.

A second set of sensors is used to monitor the temperature of the windings. A total of 10 thermisters are built into the blocks and windings in one section of the tube. Use of the sensors will be valuable to measure the rate at which the windings heat up to monitor their maximum duty cycle and to investigate the possibility of the coils over-heating. The positioning of the sensors also allow for heat flux to be measured to provide information on whether or not significant quantities of the generated heat enters the tube from the motor windings.

7 Preliminary Testbed Testwork

The test program for this project was conducted in four stages. At each interval the results were collected and the experience gained was used to modify the next stage in the design.

7.1 Small Solenoid Model

This was the first model built. The main goal of the model was to see if it was possible to exert enough electromagnetic force to hold a mass off the ground. Three parameters were investigated, the effect of the strength of a permanent magnet, the effect of the electromagnetic field strength, and the effect of the length of the energized coil. To compensate for the weight of the skip, an apparatus was built to allow a counterweight to be connected to the skip and an adjustable test load to be suspended under the skip, see Figure 49.

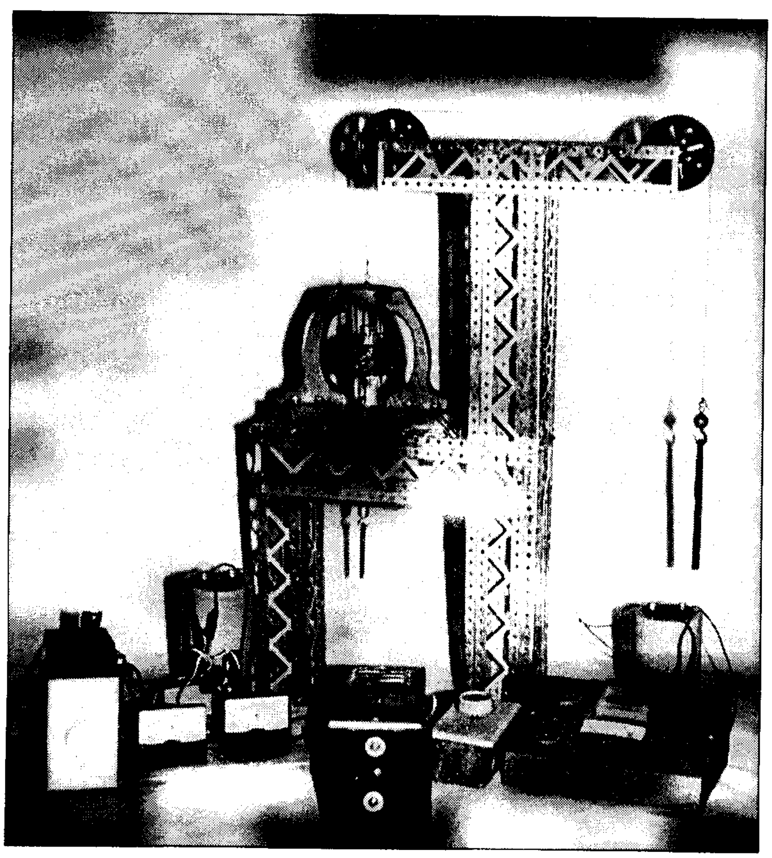

Figure 49: Small Solenoid Test Apparatus.

The skip contained an electro-magnet powered by one power source, while the windings were energized by a second power source allowing the magnetic field strength of each component to be varied independently.

A number of problems occurred during the testing of the model. First, mechanical friction inherent in the system made accurate measurement of the forces a difficult exercise. Secondly, as power was applied to the copper windings, the coil would heat up, changing its resistance causing the test conditions to drift. Third, the length of the solenoid was too short to perform a kinetic analysis of the system, and so all tests carried out consisted of stationary pull-out tests or holding capacity tests.

7.1.1 Effect of Energized Length of Winding on Holding Capacity

The first test evaluated the influence of the length of the energized portion of the coil on the holding capacity of the system. The test was run using 2.5 A of current running through the windings. As windings were added to the system, the power was adjusted to maintain 2.5 A, and the skip was loaded to the point where it would fall out of the system. The results of this test are shown in Figure 50 below:

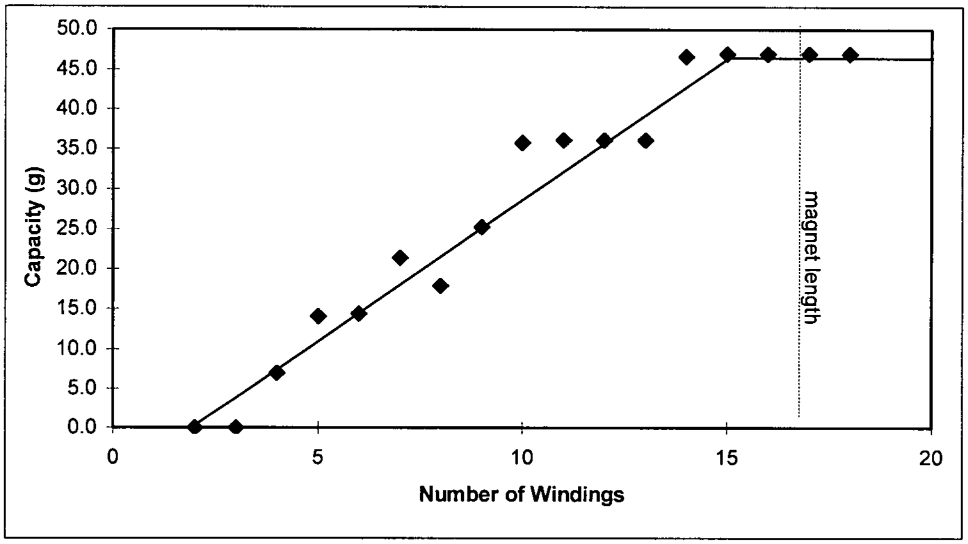

Figure 50: Skip Holding Capacity vs. Number of Windings.

The test showed significant scatter which made it difficult to accurately show the trend. The trend is shown as being linear but, it may very well be curved. Once the length of energized solenoid exceeded the length of the magnet on the skip, there was very little additional improvement in the ability of the coil to hold a higher load in the skip.

7.1.2 Effect of coil current on holding capacity

The second test investigated how the magnitude of the solenoid current affected the holding capacity of the skip. The same apparatus was used except this time 12 windings were energized with a variable current passed through the coil. The results are shown in Figure 51 below:

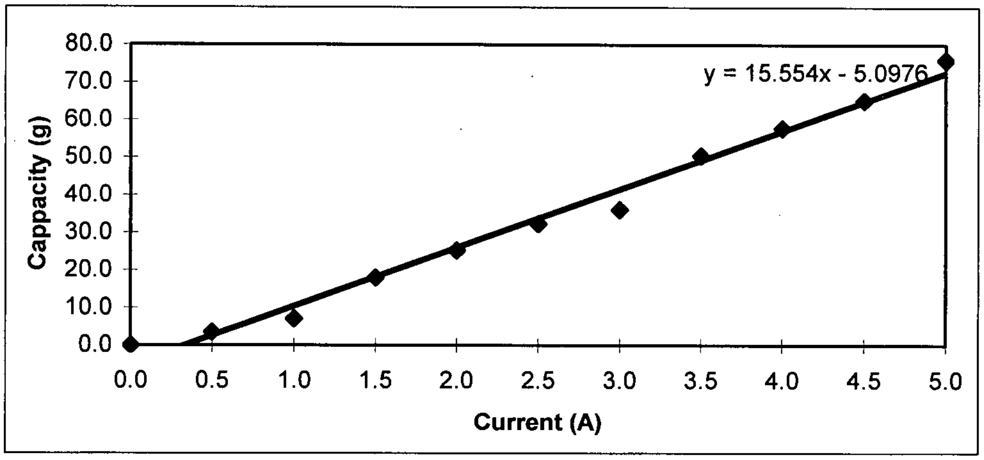

Figure 51: Skip Holding Capacity vs. Solenoid Current.

Once again a nearly linear relationship between current and holding capacity was observed. The system limitation clearly moved to the ability to prevent windings from overheating as the current increased.

7.1.3 Effect of Skip Current on Holding Capacity

The third set of tests determined how holding capacity is affected by changing the strength of the electromagnetic field in the on-board electromagnet attached to the skip by varying the current delivered to the skip magnet, see Figure 52 below:

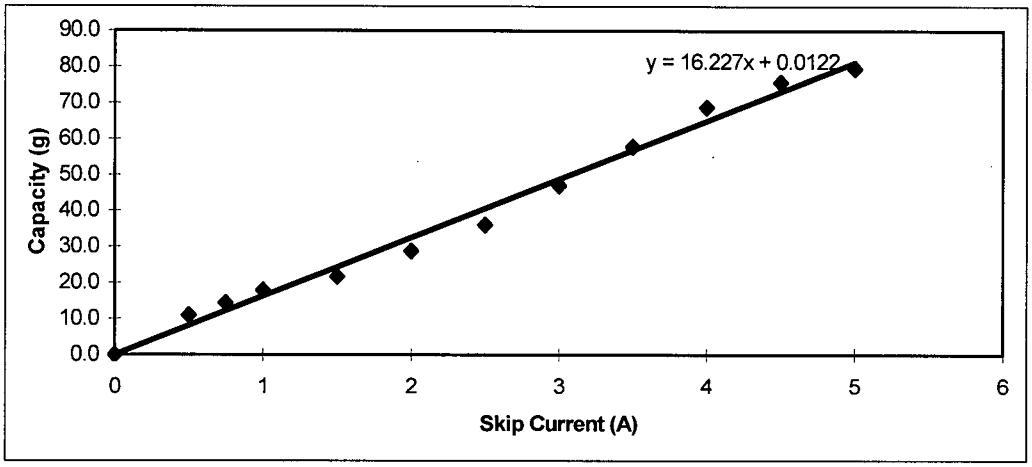

Figure 52: Skip Holding Capacity vs. Skip Current.

The results also revealed an essentially linear relationship between the skip current and the holding capacity. Again the major system limit is the ability to prevent the electromagnetic windings on the skip from overheating as the current is increased.

7.2 Solenoid Kinetic Testing

To enable kinetic testing, a second model was designed and built using the same components as the first model but with a longer solenoid and an automated switching control system as shown in Figure 53.

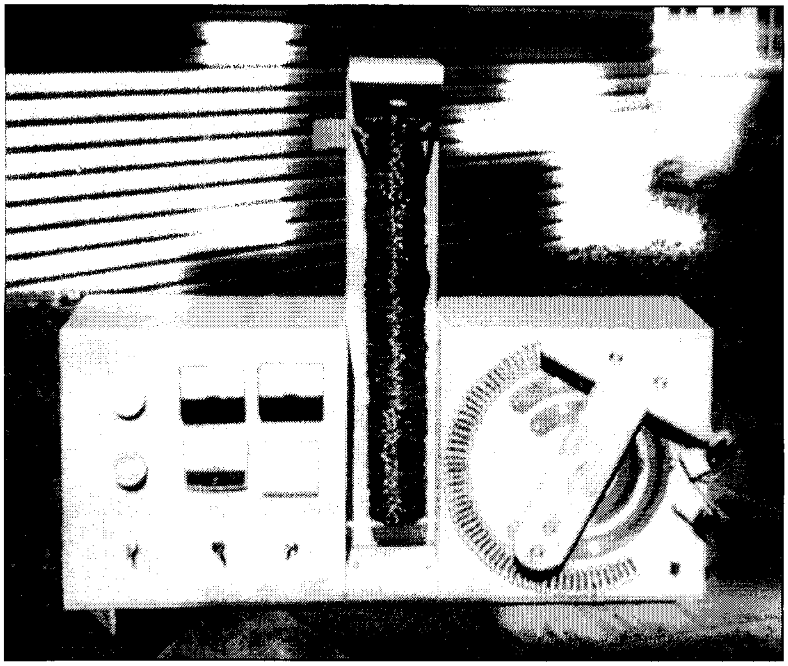

Figure 53: Second Demonstration Model.

The left box on the model housed a 15V DC power supply for the model. The center portion of the box contains a segmented, air-core solenoid that can be switched in sequence to move the skip. The right box houses a motorized 100-pole switch. When the switch rotates back and forth, the energized portion of the solenoid moves up and down causing the skip to move. The top speed for the skip was 1m/s and was limited be the switching system. The model was able to lift a 100-gram payload a distance of 60cm. The system was impractical however since it required too much copper wire, too much energy, and a rather clumsy control system.

7.3 Automated Loading Station

As design work continued on the magnetically propelled hoisting system, the question of how long it would take to load a car was a continual concern. The time to load a car will determine the capacity of a single loading station and will define the system utilization. To gain some information on how quickly a skip could be loaded, a test model was constructed.

7.3.1 Design

The model was based on the Mark 3 virtual loading design as it was the current design at the time of construction. The model was designed to achieve the parameters shown in Table 3 below:

Table 3: Design Specification

Parameter	Design Value	Tolerance*
Payload	1Kg	5%
Spillage	< 0.1%	
Load time	< 5s	
Automation	able to load without operator input	

* +/- 99% of the time

The model design is shown in Figure 54 below:

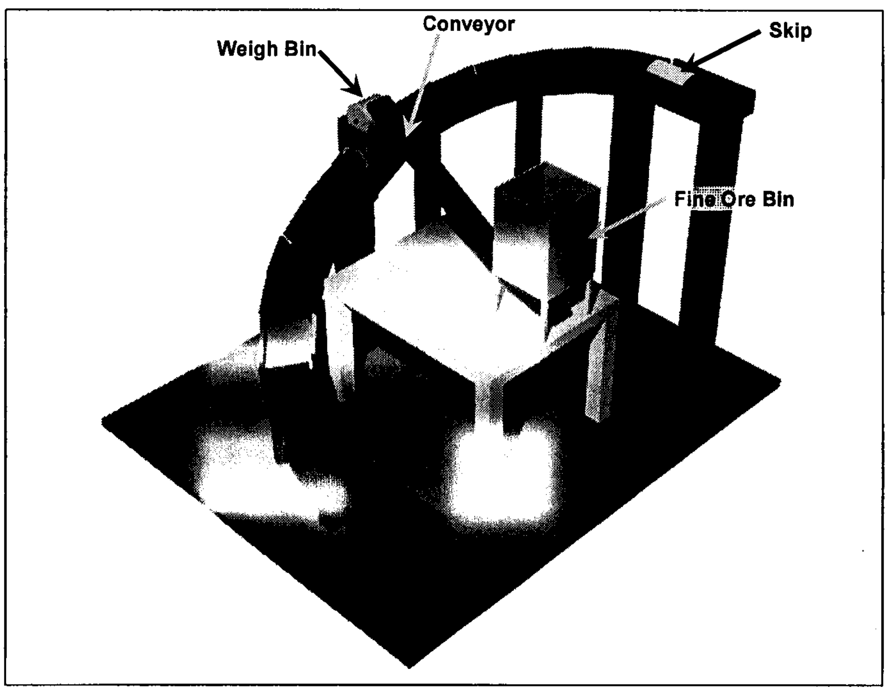

Figure 54: Model Design

7.3.2 Results

The completed model is shown in Figure 55 below:

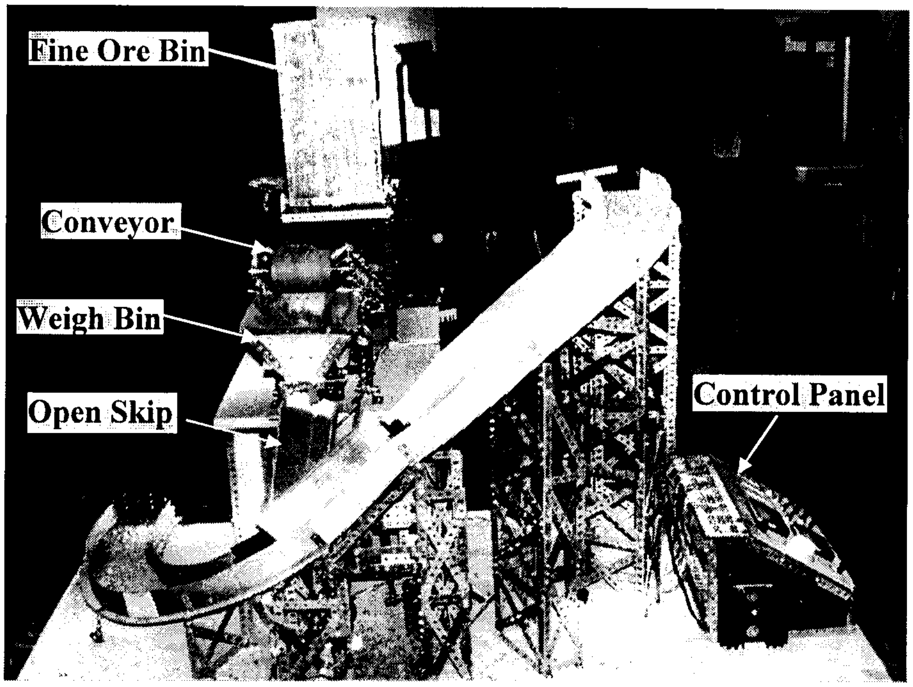

Figure 55: Completed Automated Loading Station

The results of the model testing are shown in Table 4 below:

Table 4: Automated Loading Station Performance

Parameter	Design	Actual
Payload	1 Kg	1.0 Kg
Tolerance	+/- 5%	+/- 3.5%
Spillage	< 0.1%	0.7%
Load time	< 5s	5.1s
Automation	no operator input	difficulty handling errors

The model was able to accurately weigh its designed load. It spilled too much material on the ground particularly around the tail spool of the conveyor and around the loading hopper. With some design modifications the spillage could be significantly reduced. Load time was very close to the target. The model was able to load a skip with no operator input, but it had very little ability to recognize faults. Whenever something went wrong the operator was required to step in and correct it. More details on this model are included in Appendix I.

7.4 First Linear Motor Model

The first linear motor model consisted of 6 windings built around a 3"ID PVC pipe. The motor portion was 6.5" long. The biggest test for the model was of its construction process. Success in this area gave confidence to start preparations for the construction of the bigger model. The model is shown in Figure 56 below:

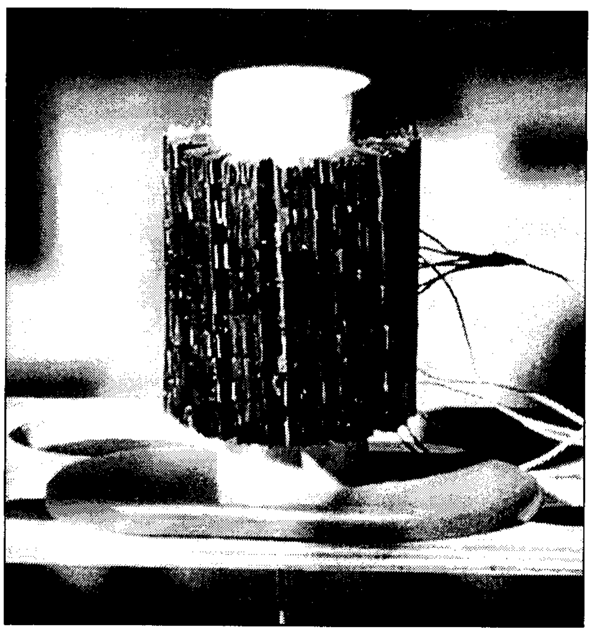

Figure 56: First Linear Motor Model.

A series of tests were performed using this model primarily testing its control system, then its ability to lift and move a suspended load.

7.4.1 Control Tests

The first stage of test work on this model was to attach a suitable control system to the model. The first plan was to use 3-phase AC power to run the system. To enable the model to run at different speeds and to enable acceleration of a skip from rest, a variable frequency AC power source was required. One was sourced from ET Power Systems for a price of 54,000 £. This is far beyond the available budget at this point in time, and so the 3-phase AC-power option was abandoned.

Since the model is to be a demonstration testbed, a control system was built to use a chopped DC power supply. The idea was to pass single-phase 110V AC power through a Variac creating an adjustable voltage supply. This power was rectified to a positive and negative DC signal and a computer was used to switch a series of transistors to form a square wave 3 Phase DC-power supply. The wiring schematic is included in Appendix VI. By using the computer, both the frequency and direction of the 3-phase power could be controlled. The control system worked in principle, but was very ineffective in moving a piece of steel inside the tube. When the tube was orientated in a horizontal mode, the steel just barely moved back and forth inside the tube.

By trial and error, it was observed that when a winding was energized, it was very difficult to pull the piece of steel out of the motor. A mechanical toggle switch was wired in series with each winding connected to a positive DC power supply. By manually turning the switches on and off in sequence, the windings could be energized one at a time with a significant improvement in operation. For testing, a Variac was used to control the power supplied to a modified DC welder that was able to supply the current to the model. For the initial test work, these mechanical switches were used for simplicity.

7.4.2 Performance Testing

To perform these tests a range of lengths of steel pipes were created. A total of seven lengths ranging from 15mm to 140mm in length were tested as shown in Figure 57 below:

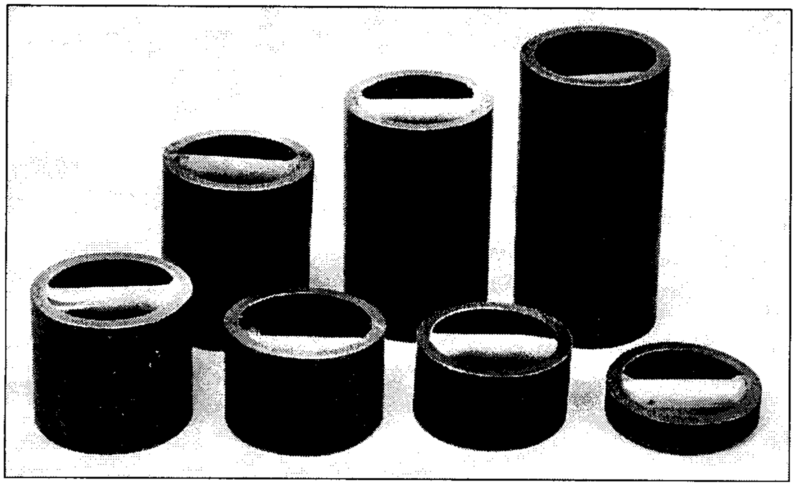

Figure 57: Testwork on Length of a Steel "Skip"

The pipe was used to simulate different lengths of skips inside the tube. The general procedure used to test the performance of the motor was as follows:

1. Attach a container to a wood dowel placed inside the test length of pipe.
2. Set desired power level.
3. Use the toggle switches to manually apply the test conditions.
4. Energize the windings.
5. Load the container until the system pulls out (a capacity test), or until the motor can no longer move the pipe upward (a lifting test).

The problem with this procedure was that while the test was run, the windings heated up resulting in a current drop. This problem was minimized by using a consistent procedure and allowing the model to cool between tests. The second problem was dealing with friction between the inner PVC surface of the motor and the outside of the pipe. During the tests the steel pipe is attracted to the inside of the motor resisting the steel pipes vertical motion. Complete results are included in Appendix VII.

The first test evaluated the effect of the pipe length on the performance of the motor. For this test, two windings were used to lift the skip, and then the skip was loaded until it pulled out of the model. The results from these tests are shown in Figure 58.

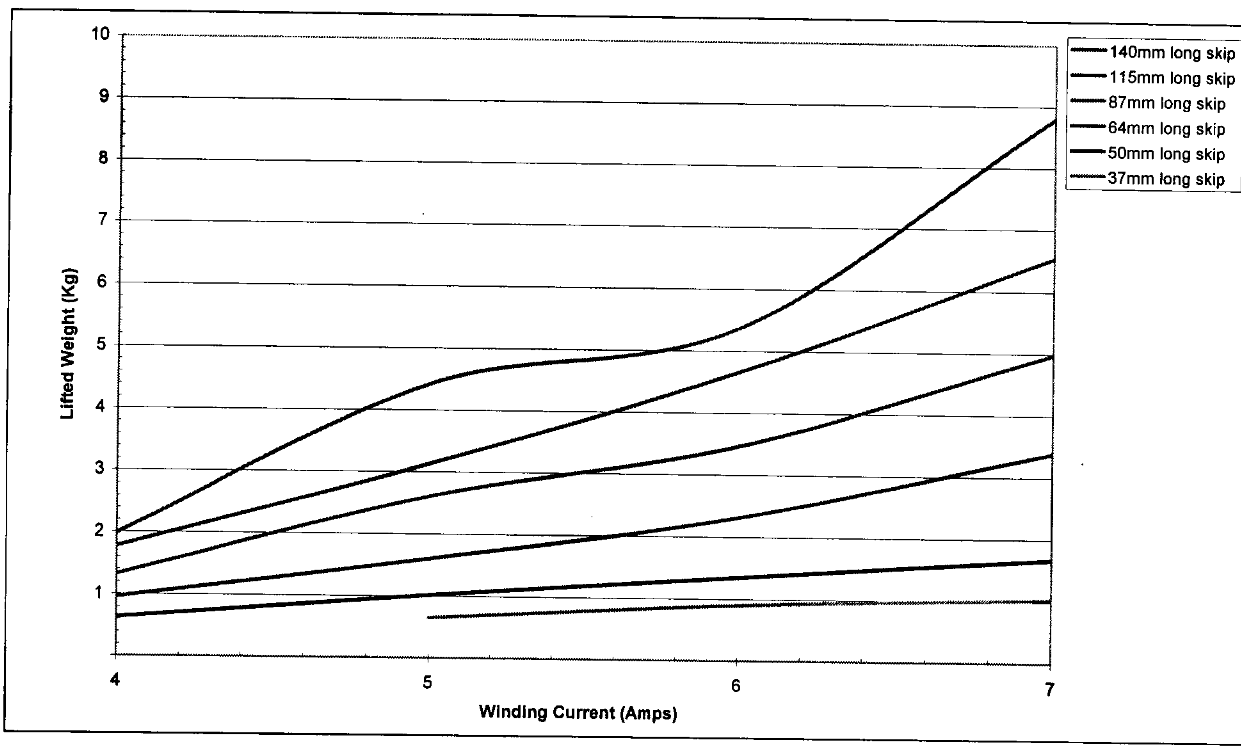

Figure 58: Effect of Skip Length and Coil Current on Skip Capacity

The results appear to be a linear relationship with the fluctuations likely due to the heating and friction problems mentioned above. On the scale of the test, the longer the test pipe the higher its capacity and the higher the current required.

The number of windings energized at any one time has a significant impact on the amount of power consumed. To study the impact of power on the capacity of the skip, a series of tests were conducted. Figure 59 shows the results for the 64mm pipe length.

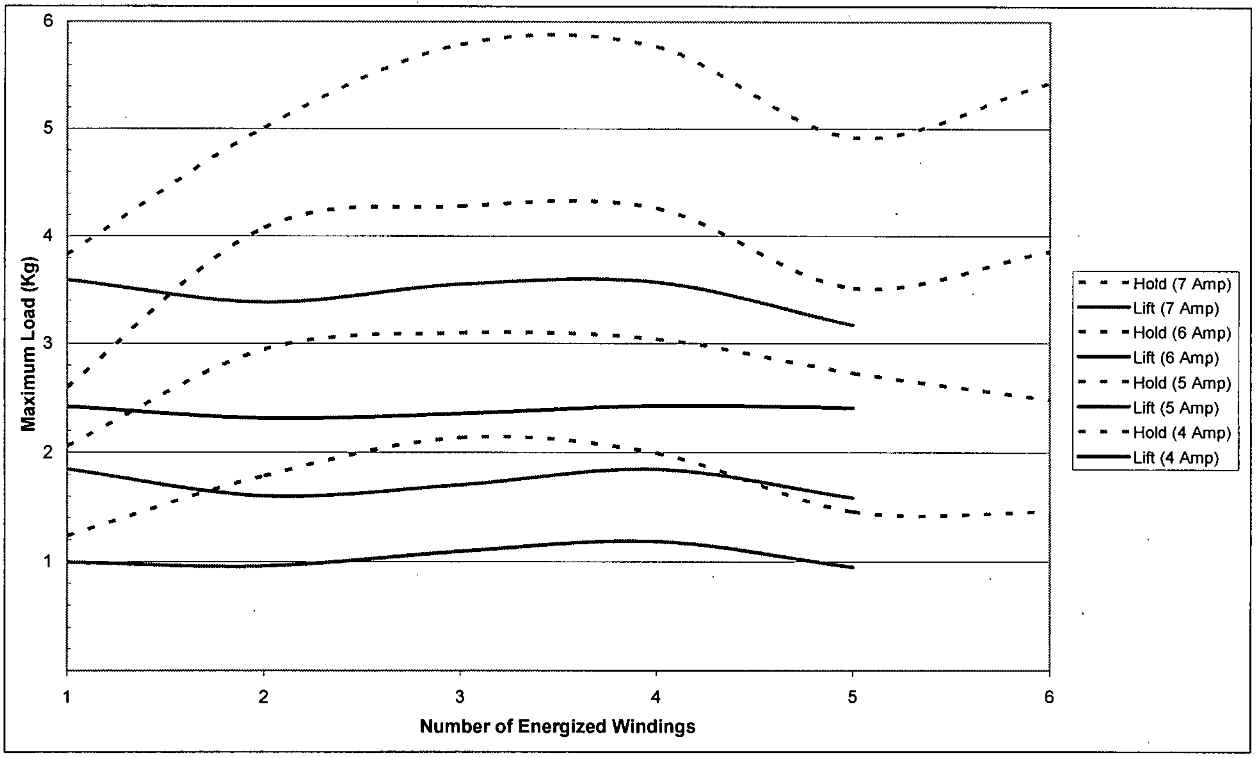

Figure 59: Effect of Number of Windings (Coils) on Skip Load (for Both Lift and Hold).

From the above figure there appears to be little correlation between the maximum skip load and the number of energized windings. There does appear to be a relationship between holding capacity and number of windings up to the pipe length (~ 3 coils). The dips observed for the five and six windings tests are attributed to rapid heating of the windings and possible "end" effects. Complete results are given in Appendix VII.

With the success of the lifting tests, focus returned to improving the control system. A new system was built using the original power supply. The Variac and rectifying diodes were replaced with the modified DC welder as the main power supply. The six signals used to switch the transistors in the previous design were now applied to generate a sequential 6-phase series of square-wave DC pulses. The control signal was employed to switch on and off the transistors wired in to each coil on the tube.

Initially conventional transistors were used with the windings. These units quickly overheated and failed. The cause was thought to be due to insufficient current being applied to the transistor gate and so Darlington transistors were tried next to avoid this problem. Darlington transistors have a second transistor built into the gate so the required current to switch the transistor is significantly reduced. A second problem related to the inability of the transistor to switch on a voltage higher than the gate voltage. Finally after many trials the problem was isolated. The transistors were placed between the power source and the load instead of between the load and ground. With this circuit configuration, the transistors functioned much better. The Darlington transistors were later replaced with MOSFETs (Metal Oxide Semiconductor Field Effect Transistors). Since they are a field effect transistor they are switched by voltage not current, which reduces the required power from the control circuitry. To prevent a coil from being energized when a car is not present, infrared emitters and sensors were placed along the tube with each coil to determine the location of the skip. This signal was combined by using a transistor with the gate controlled by the position signal and the collector connected to the main control signal as shown in Figure 60 below:

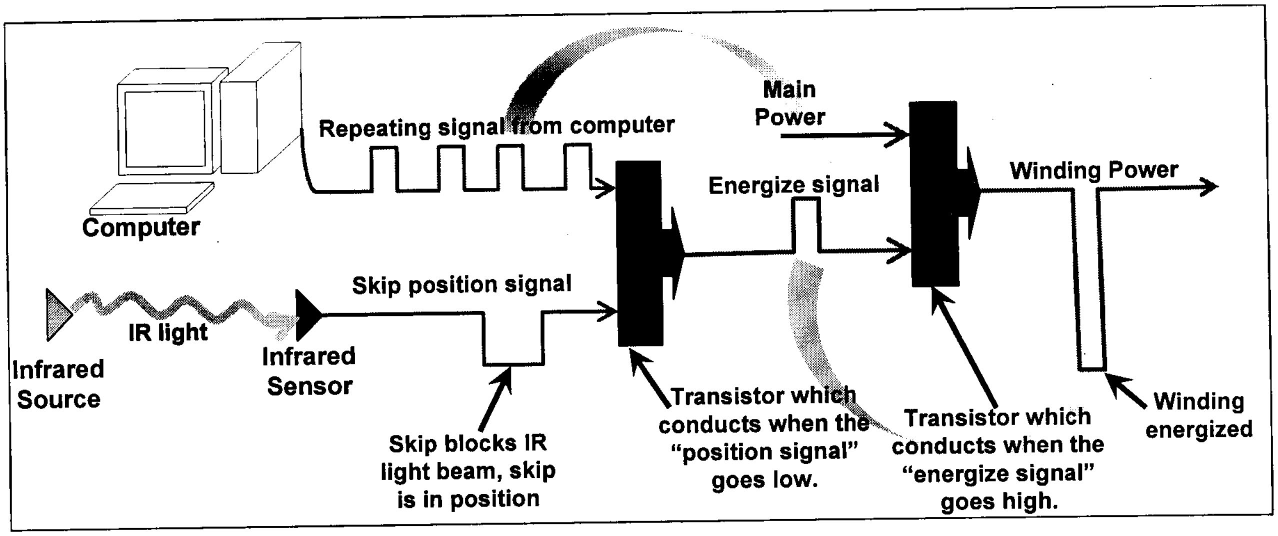

Figure 60: Control System.

The wiring schematic for the control system is included in Appendix VIII.

7.4.3 Results

From the results of these preliminary tests, the skip for the next model was designed to use two windings for redundancy. It was considered that if one winding fails for some unforeseen reason, a certain degree of force will derive from the second winding. The length of the steel portion was designed to coincide with the length of 2.5 windings. Infrared sensors were installed ahead of each winding to sense the position of the skip in the tube. The motor was also redesigned to be a 4" ID pipe instead of 3" to ease access to the interior of the motor.

7.5 Model Section Testing

With test work on the first LIM model completed, construction of the sections of the testbed model was started. As each section was completed, it was connected to its control card and tested. The first sections had many problems in getting the control system to function properly.

Transistors would sporadically explode for no apparent reason. By the time the problem was isolated, 6 sections were constructed.

The problem related to the fact that a number of windings in each section were shorting out through the steel block components. The shorts resulted from the omission of a bobbin to contain the winding on the tube during construction. The bobbin was not really needed to mechanically wind the coils onto the tube, but was actually essential to ensure that each coil remained isolated from its adjacent coils. Without a bobbin to protect the winding from the iron components, the wire came into close contact with the steel. Occasionally some of the protective varnish coating would wear off the wire during construction causing the winding to short out.

To fix this problem, all six completed sections were disassembled and rebuilt using Teflon spacers to insulate the winding from the steel components, see 61.

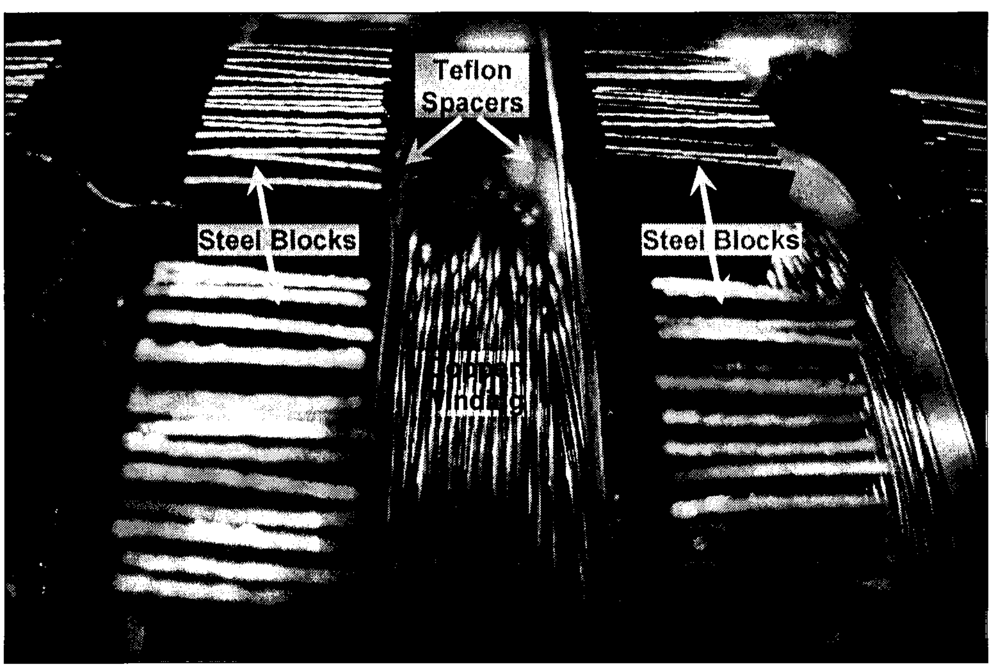

Figure 61: Teflon Spacers to Protect and Isolate the Windings.

This solved the problem with the shorting windings. The remaining problems during bench testing were due to errors in construction, e.g., hooking a winding up backwards, faulty soldered connections, faulty components, and incorrectly wiring a coil in the section. Through careful and methodical testing on the bench, each section left the bench functioning properly as per design.

7.6 Model Testing

With all sections completed and the support ready to mount the sections, one final check was performed on each section as it was being fixed to the frame. On a few sections, some faults showed up due to wires and sensors being pulled out as the sections were handled and moved into place on the frame. These were repaired and the sections were mounted onto the frame and connected to each other.

Once all of the sections were fixed in place, wires to connect the sections to each other were run and all of the terminals soldered. Sections were connected to their respective terminals one at a time with the model being retested after each section was wired into position.

Initially, considerable difficulties were experienced with the control system. The problem was traced back to diodes added to the sequencing signal. The diodes had been added to prevent a fault in one section from affecting any of the other sections. The diodes also prevented the drain resistors from grounding the control signal between energizing pulses. This problem caused random energizing of the windings. The grounding problem was rectified by adding additional drain resistors to each section effectively grounding out the signals.

Once the control signal was functioning properly, the next problem to be tackled concerned windings being energized when no car was present. This problem was attributed to the infrared sensors not functioning properly. Considerable time was spent trying to get all of these sensors operating as intended. Some sensors were successfully fixed but others could not be rendered operational. With windings being constantly energized, considerable power was being drawn from the source. These coils would heat up and eventually, their transistors would overheat and malfunction.

As more sections were hooked up, the problem escalated. Eventually the idea to get all sections working together was abandoned. To attempt to limit this overload problem, a relay was installed in the power line for each section. In this way, only the section in which the skip was located would be energized leaving the other sections to cool until required. Unfortunately the mechanical relays were unsuitable to open under load. Arcing occurred between the contacts which quickly destroyed the relay. Solid state relays would not have had this problem, but the cost of ones big enough to switch the required current was too great.

The next approach was to remove the relays and hook the main power bus back up to all the sections. This time instead of cutting power to a section, the energizing signal to each section was cut using a transistor controlled by the original mechanical relay signal. This method had some limited success. The main MOSFET transistors were sensitive enough to partially conduct from the gate voltage being applied to the isolating transistors. The performance of the

MOSFETs could be improved by tuning the circuit resistors but the problem could not be eliminated.

A final attempt to make the system work was to return to a mechanical switching system. To control the model, a 96-pole switch was built to energize 4 coils simultaneously on the model. As the arm of the switch moved, these 4 power input points would move around the loop in sequence. The switch eliminated the skip position sensors and relied on timing to energize the windings. The arm was controlled by a stepper motor enabling control of speed and direction of the arm and model. The switch is shown in Figure 62.

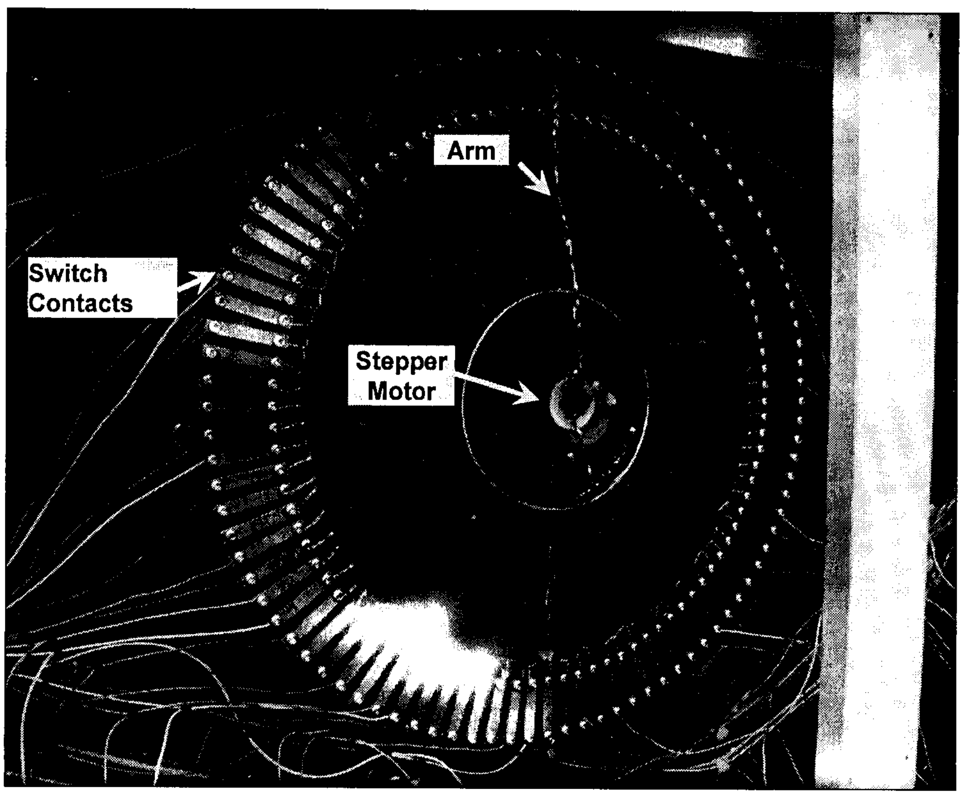

Figure 62: 96-pole Mechanical Switching System.

The mechanical switch also did not solve all of the problems, as now without any skip sensors in the circuit, the coils could not react to the skip being held up slightly (by friction or binding) and so, not following the now independent energizing sequence. Without being able to identify the position of the skip, it was not possible to control the model and so, as of to date (June 2003) the model has been unable to work as planned. At this point, time constraints prevented further work on the model.

8 Discussion of Results

Although there were problems to make the model fully functional, there was sufficient limited operating function to perform some testwork.

8.1 Speed Testwork

The model was designed as a synchronous system, so it could not function as an accelerator. A skip was placed in a section of the tube and the computer was used to speed up its switching frequency to accelerate the vehicle. Two straight 36" sections on the model were isolated to conduct the tests. As a result there was only a very short period of time in which to accelerate the skip before the end of the test section was reached. In this set up, a top speed of 1.8 m/s was reached by the end of the section.

8.2 Single Skip Control

The model demonstrated that it is possible to control a single skip moving on inclines from horizontal to vertical and around corners. The system was able to demonstrate controlled acceleration, deceleration and braking.

8.3 Multiple Skip Control

The same tests were repeated with two skips in the system. Since the model is a synchronous motor, the cars performed as designed always keeping the same distance apart during the tests.

8.4 Stopping Tests

The model was able to stop the skip in a vertical orientation for a time period of 10 seconds, after which a slow speed of 0.01 m/s had to be maintained to prevent the coils from overheating.

9 Proposed Production System

The proposed design of an actual production system had a number of objectives that guided the research process. These objectives were:

- energy efficiency
- maintainability
- safety
- longevity
- reliability
- flexibility

These elements were used to guide the evolution of the designs with the overall goal to make the system as simple as possible.

Experience and information gained from computer simulations, talking with experts, and from kinetic prototype modeling all contributed to the evolution of the design.

9.1 Shaft

The main shaft in a mine will continue to be used to provide access for many of the mine's systems such as a cage to move people and supplies, ventilation, pipes, infrastructure, and as an emergency escape route. The skip compartments in the shaft will no longer be required so the shaft could potentially be a smaller diameter allowing it to be constructed cheaper, faster, and be geotechnically more stable. The diameter might be small enough to construct using a raise-boring machine which could certainly reduce capital costs of conventional shaft-sinking.

If the purpose of the shaft is simply to hoist rock then the shaft could be developed using a raise borer with a diameter of 1.5 to 3 meters depending on the size of the system and the need to provide access to the drive components.

9.2 Drive

The drive for the proposed system is designed to propel the skips using a tubular linear synchronous motor designed to operate on standard 60 Hz 3-phase AC power. The motor is fabricated in short sections and connected underground, much the same as pipes and other utilities are extended today in a conventional mine. Each section of the drive is 4.5 meters long containing 36 copper windings in order to make the sections short enough to maneuver underground. With the windings spaced at 12.5 cm the synchronous speed of the design will be 15m/s when driven with three phase 60Hz power.

9.2.1 Materials

The drive is composed of three main components: the outer core, the electrical windings, and the inner tube.

Outer Core

The outer core must have a high magnetic permeability as it forms half of the magnetic flux path. Additionally it is used to protect the windings from damage and provide some structural strength to a section.

The choice of materials would be laminated iron or a composite powder-metal core. The composite core will have a lower permeability but much cheaper and faster to build. So for this design a powdered core is recommended.

Electrical Windings

The electrical windings are made of copper. Aluminum is another option which could potentially provide both an economic and weight savings. The potential to use aluminum should be evaluated in the future with care taken to consider fire hazard from aluminum oxidation at junction points in the circuits.

Copper wire is wound onto a bobbin to form the winding. The bobbin is necessary to protect the winding from developing short circuits during operation and ease construction. The bobbin can be formed from injection-molded thermoplastics.

Inner Tube

The inner tube is used to protect the drive from wear by the wheels of the skips, and to provide a smooth surface on which the wheels will ride.

The tube must have a low magnetic permeability and be resistant to wear. The tube could be made of plastic, fiberglass, or ceramic. Plastic will have high wear rate and high associated maintenance. Ceramic will be costly and susceptible to brittle failure. Thus fiberglass is chosen for the preliminary design. There is also the potential to examine the choice of wheel and tube materials to allow for a more economic solution while still maintaining a suitable life expectancy.

9.3 Construction and Maintenance

The construction and maintenance of the track will have a large impact on the success of this system. It is critical that the track be efficiently manufactured, but it must also be maintainable at the mine site.

9.3.1 Tube Construction

The construction will begin by making the components. The outer core is to be molded as two half-cylindrical pieces that can be bound around the inner tube and windings. The windings are pre wound on the bobbins while the inner tube is cut to length.

The first step in building a section is to set the first half of the outer core facing upwards, then the coils are set into position, followed by the fiberglass inner tube being slid into position, and finally the second half of the outer core is set into position. This process is illustrated in Figure 63 below:

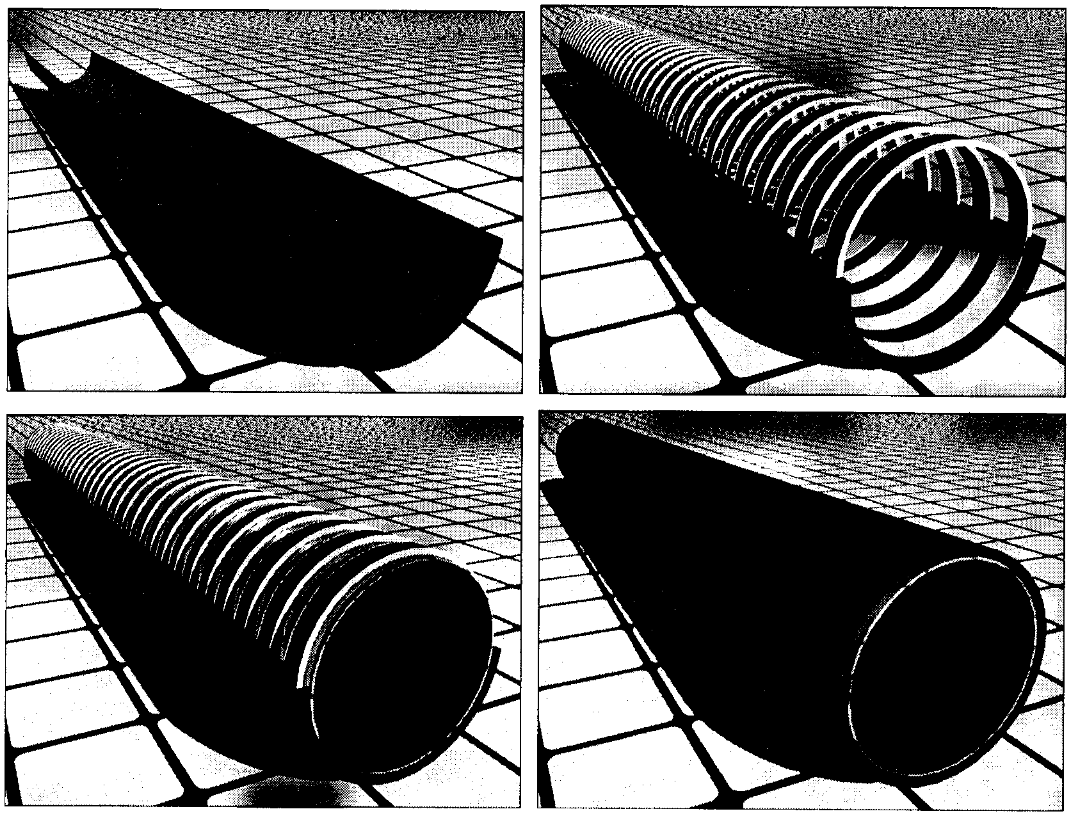

Figure 63: MagLev Hoisting Drive Construction Sequence.

The design must allow for sections of the drive to be maintained. The outer core can be split apart to allow the inner tube to be replaced or to replace a coil should a problem arise.

9.4 The Skip Vehicle

The skip is designed to be recycled at the end of its life. The only serviceable components are the guide wheels and the lid latching mechanism. At the end of its life the magnetic core, wheels, and lid components may be recovered for reuse if they are in acceptable condition. The rest of the skip will be recycled. To provide access to load the skip and to contain material during transit, a lid must be incorporated into the design. The lid will be placed on the end of the skip as opposed to its side. This should allow more efficient filling of the skip without interference from the lid and with minimum spillage.

9.4.1 Aerodynamics

As the skip passes through the tube from underground to surface it will interact with air within the tube. The skips will trap air between them as they pass along the tube in effect pumping the air in the tube. The Magplane Group instrumented their vehicle to collect air pressure information. They found that during acceleration there was a pressure pulse of 1.4kPa in front of the skip during acceleration, dropping to 0.3 kPa once a final speed of 11m/s was reached. (Montgomery, Fairfax, & Smith, 2001) These results seem to indicate that the air is being pushed in front of the skip instead of flowing around the skip.

Further research is needed into the aerodynamics of the skip. One aspect of the design is the blockage ratio of the skip. The blockage ratio is the ratio of the cross sectional area of the tube that is occupied by the skip to the total area of the tube. A large blockage ratio provides better protection on capsules in case of power failure, so that capsules will not collide with each other too hard. (Zhao & Lundgren, 1997) In effect the skips possess aerodynamic braking. This approach will result in increased air pressure within the tube and must eventually be a design consideration.

9.4.2 Materials

The skip is made of three main material components, the skip body and lid, the magnetic core, and the guide wheels. Each component has its own material constraints.

Skip Body and Lid

The skip body does not form part of the drive system, so it has does not have a magnetic permeability constraint. The main constraints on its design are durability to being loaded, its strength, cost and weight. For this design, a simple steel construction is specified. There may be an opportunity to fabricate the skip out of fiberglass and use a rubber liner to produce a cheap and lighter-weight skip.

Magnetic Core

The magnetic core has the same options as the outer core of the drive. It has the additional requirement to be light and tough. Design of this system is not far enough along to truly say if an iron core or a powder composite is best. For the design concept a powder core will be assumed.

Wheels

The composition of the guide wheels will be important from a maintenance perspective. If the wheels are too soft then they will have to be changed frequently. If they are too hard then the inner tube will wear and require frequent replacement. The wheels are currently envisioned to be a hard rubber. Their composition should be designed with the inner tube in mind to find the best material combination for the system.

9.4.3 Construction

The body of the skip is built from a section of steel tubing. A bottom is welded on one end and a lid is fabricated at the other. Two sets of three guide wheels are used to support and guide the skip inside the tube. To enable the skip to travel in any orientation a latching mechanism needs to be built into the lid as shown in Figure 64.

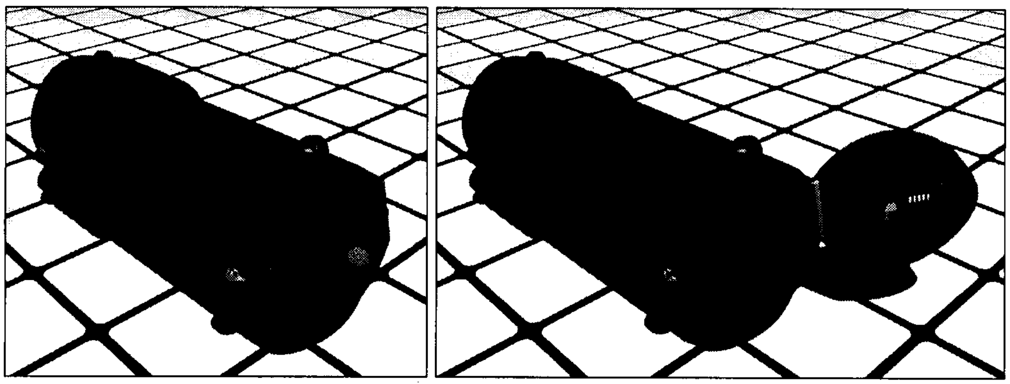

Figure 64: Skip Design Components.

9.4.4 Maintenance

Maintenance of the skips will consist of replacing wheels and repairing the latching mechanism. By having unique identification on each skip, information on hours, distance, and loads can be tracked to help with maintenance scheduling. Sensors could also be installed in the drive to measure wear and temperatures of the skip wheels which can also help in a preventative maintenance program. After a designed set of service cycles, each skip can be diverted to a repair section of the track for maintenance.

9.5 Loading Stations

The loading station is composed of four operations grouped together to get the run-of-mine (ROM) material into the skip. The first stage provides a location to which the ROM material is trammed. The second stage provides a crushing system to reduce the ROM material into small enough fragments (< 4") to be loaded into the skips. The third stage is to weigh the load for each skip. The fourth stage is to manipulate the skip from the delivery tube, to the loading point, and then to the exit tube.

The access from surface to the loading station location will have to be sized to accommodate the largest component. The largest component will likely be the crusher. The development of low profile or more compact crushers, or possibly using two smaller crushers to replace one big crusher, will enable the use of smaller drifts. Alternatively the system may be well suited to a situation where the material is already delivered in smaller fragments, like in continuous mining machines or tunnel boring equipment.

The loading station is illustrated in Figure 65 and Figure 66 below:

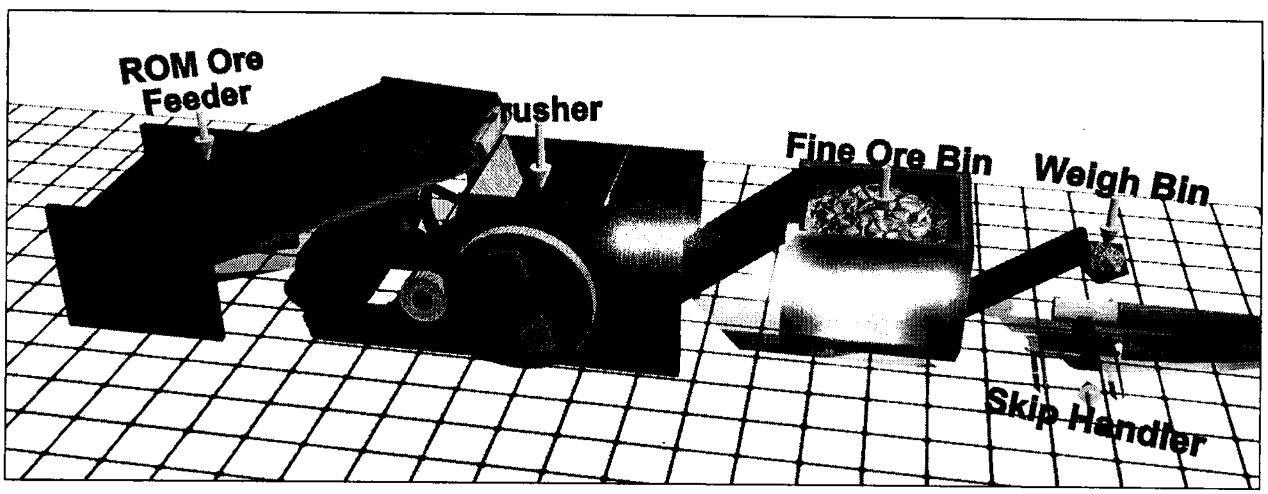

Figure 65: Proposed Loading System.

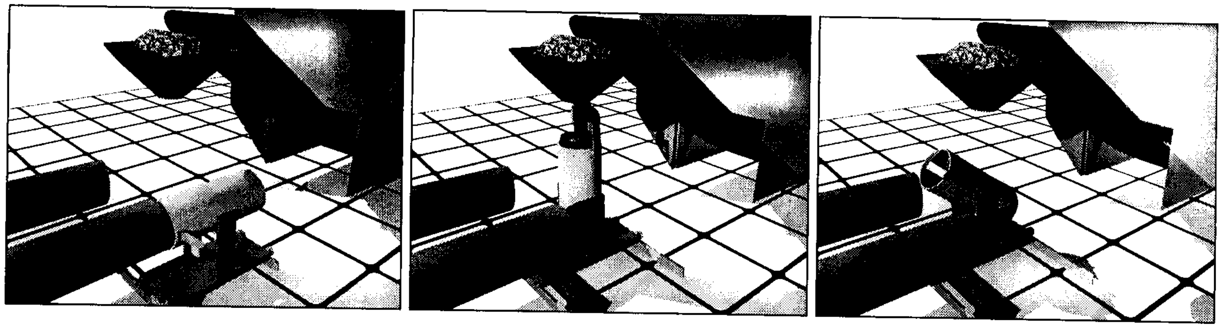

Figure 66: Proposed Loading Sequence.

9.6 Dumping Stations

At the dump point the drive tube will cross over the storage silos. Wherever it is desired to dump a skip, a hinging dump mechanism is installed. The proposed dumping system is shown in Figure 67 and Figure 68 below:

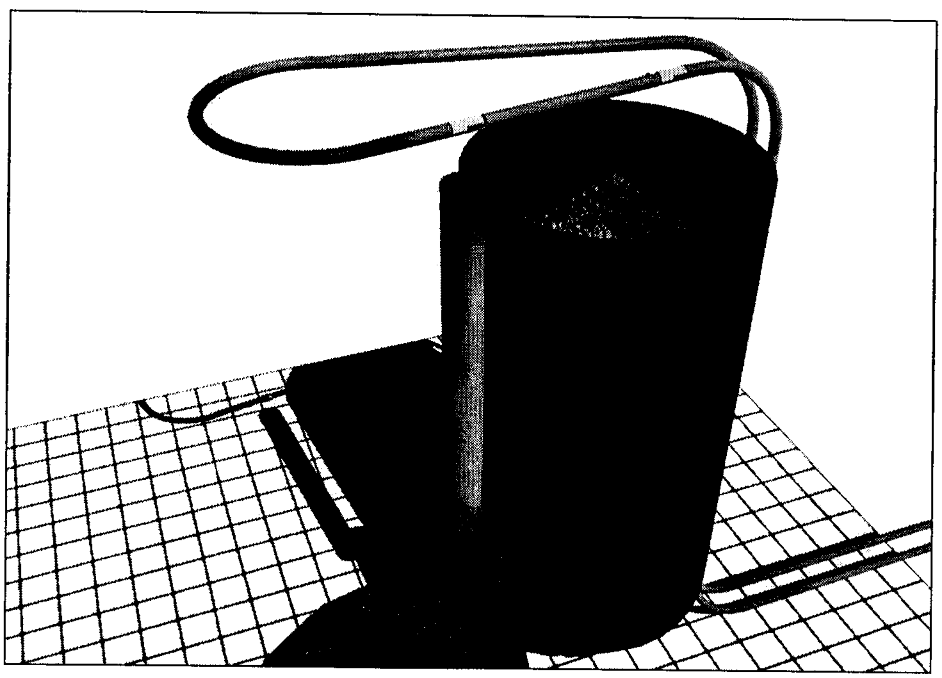

Figure 67: Proposed Dumping System.

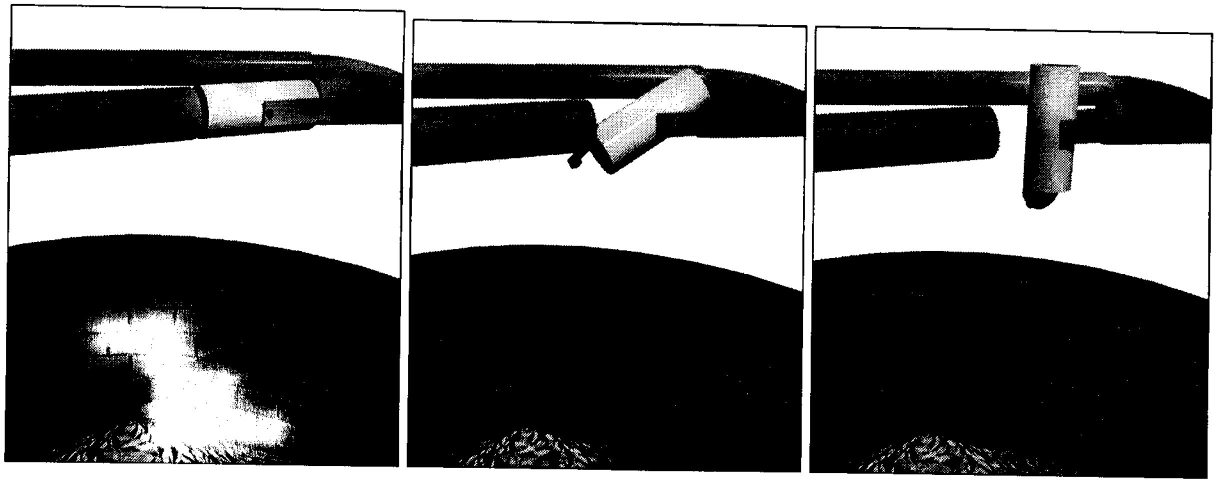

Figure 68: Proposed dumping sequence.

10 Risk Analysis

The ability to demonstrate that a magnetically propelled hoisting system has an acceptable level of risk will play an important role in convincing industry that the system deserves further development. Mining is an inherently risky business as reserves, markets, and regulations are seldom predictable. As a result companies are looking for ways to reduce their exposure to risks. For this system to be successful, the benefits of a magnetically propelled hoisting system must strongly outweigh any increased risk.

10.1 Identified Risks

To limit this risk assessment to a reasonable size, the scope of this analysis focuses on the drive system. Although the loading and dumping systems are not included in this study the implications of payload weight and spilt material have been included as these issues have important impacts on the drive system. The risk assessment started by completing a risk register (Appendix IX) then a Failure Mode Effect Analysis (Appendix X). The following risks were identified during the risk register stage of the study:

Table 5: Identified primary risks.

Number	Risk
1	Water infiltrating motor, could cause corrosion, short circuiting
2	Mine water going acidic, corroding motor
3	Motor not aligned during installation, cars jamming while in transit
4	Control system failure, winding remaining energized, overheating causes fire
5	Winding short circuit, winding over heating causing fire
6	Electrical short to outer core producing shock hazard
7	Car becoming a projectile when section removed for maintenance, possibly striking & killing worker
8	Power spike damaging control circuits, resulting in portions of system becoming inoperable.
9	Mechanical failure of structural integrity.
10	Seismic vibrations, possible deformation or breaking of joist between sections
11	Vandalism, damage to system, potential injury to personal
12	Inexperienced personnel performing inappropriate operating or maintenance practices to system.
13	Overloading system, trying to set records for bonus by tampering with weight sensors to overload cars.
14	Generation of noxious fumes, health risk to personal underground.
15	Jammed cars in tube, stopping production, cost of clearing and restarting system
16	Power failure, loss of propulsion to cars in system, possible free fall causing extreme damage and potential injury.
17	Failure of braking system, cars free fall causing damage and possible injury
18	Delays in producing sections, delay in system installation, could delay commissioning of mine
19	Ground movement throwing alignment out.

10.2 Results

Once the risks were identified, one particular outcome of the system was selected for analysis. The outcome chosen was the success of the drive system to propel the skips from an underground loading facility to a surface dumping facility then return underground for another load. Three possible outcomes can occur: the car makes it with no problems, the car encounters problems but still manages to make it, and finally the car fails to make it to surface. The applicable risks that were identified in the risk register were then combined and used as the basis for the risk flow sheet.

The partial risk flow sheet is shown in Figure 69 below:

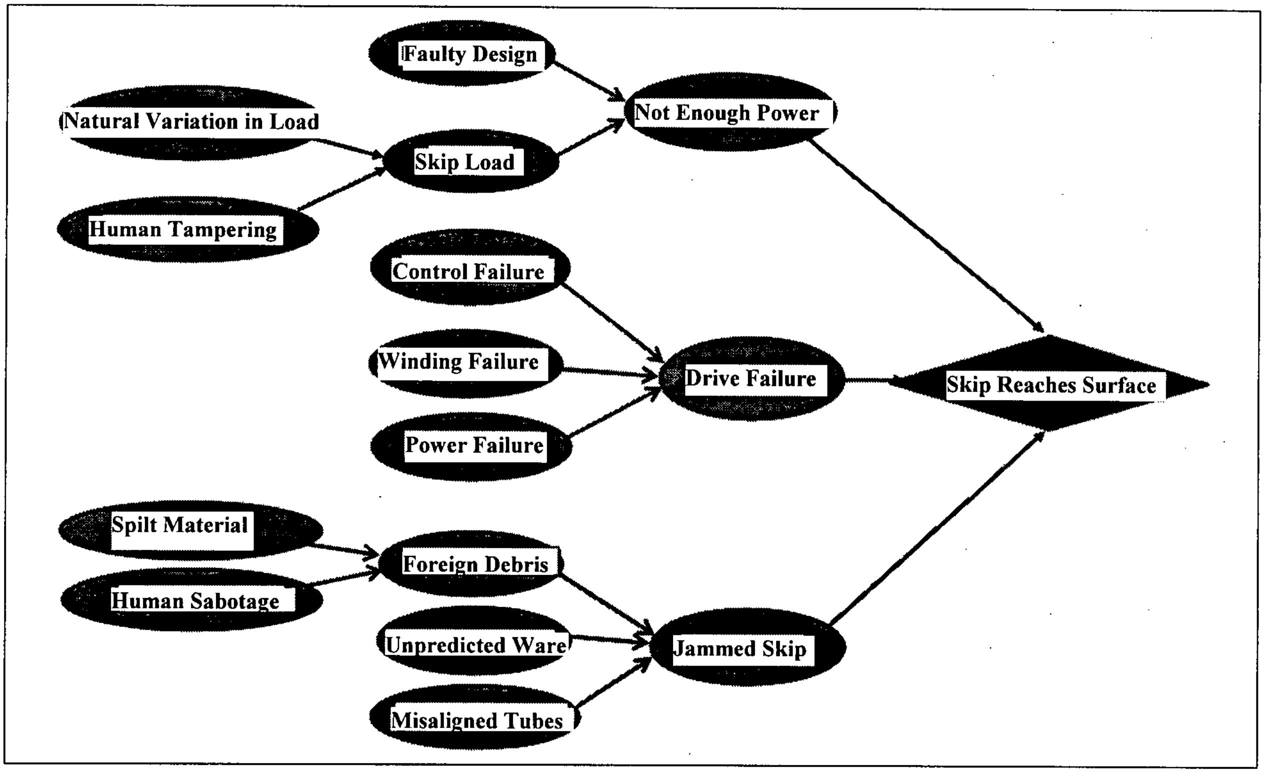

Figure 69: Partial Risk Flowsheet.

The success of the car to reach surface depends on three factors. First the drive must have enough power to move the skip. Second, the drive must function properly. Third, the car must be free to move and be propelled by the drive. These three components of the system are broken down further in the following sections.

10.2.1 Drive Power

In order for the system to function, the drive needs power. The drive power depends on the design and construction of the unit and the load required to be moved. The load will vary due to natural fluctuations in load sensors, and occasionally there may be human tampering with the system that could result in an excessively overloaded skip.

10.2.2 Drive Failure

If the drive fails to energize it does not matter how powerful it is, the system will fail. The potential of the drive to fail from a winding failure, failure of the control system, or from a power outage are included in this investigation.

10.2.3 Jamming of Cars

The cars could become jammed inside the tube from a number of factors. If foreign material is present inside the tube, as the car passes by, there is the possibility that the material will end up between the car and the tube, jamming the car. The foreign material can be present inside the tube from spillage or because of human activity or sabotage.

The guide wheels on the skip are required to keep it centered in the tube preventing the skip body from making contact with the inner side of the tube. If the materials chosen for the wheels and the tube do not perform as expected, they may prematurely fail or wear allowing contact which could potentially damage or jam the skip.

Alignment of the tube sections is important to enable skips to pass over a joint without making contact. Alignment problems may occur initially as problems arise during installation. Once the system is commissioned, alignment problems may occur as the ground moves or falls around the drive installation.

The risk flow sheet combines all these factors to determine the success of the entire system.

10.3 Quantifying Risk

In order to determine the risk associated with the entire system, the probability of occurrence of an event at the bottom level of each branch in the risk flowsheet needs to be quantified. Since this is a new and novel system, there is very little information available. When information is lacking, it must be collected by looking at similar components in other systems.

10.3.1 Faulty Design

Before a magnetically propelled hoisting system would be installed in a real application there will be numerous stages of prototyping. At the end of this process the likelihood of a faulty design will be quite low. So in an actual installation, this is more likely to account for slight variations in the construction of the drive. In the final design it is estimated that there is a 1% chance that the drive will be weaker than designed, a 10% chance that it will be stronger then designed, and an 89% chance that it will perform as designed.

10.3.2 Natural Variation in Load

The load weighed for the skip will have a natural fluctuation due to the sensors and response time of the conveyor. A model of a loading system was developed with a designed payload of 1kg. The distribution of loads derived from this model is show in Figure 70 below:

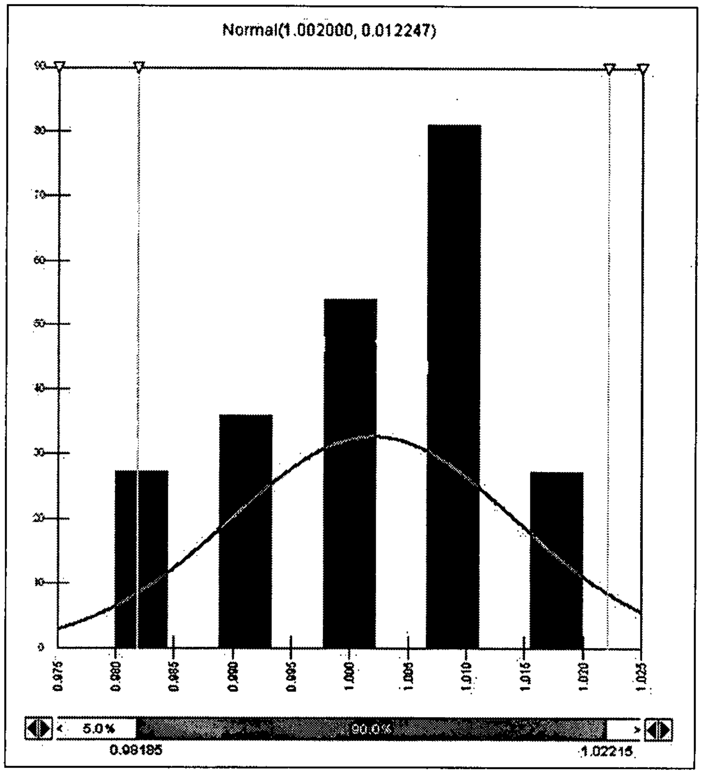

Figure 70: Modeled Load Distribution.

This distribution was used as distribution for the payload in the real system.

10.3.3 Human Tampering

Load fluctuations beyond that due to the system itself are possible because of human interference. The possibility of human intervention was assigned a frequency of 1 in 100,000 loads. When human tampering occurs the load was given a 25% chance of being 6% overloaded, a 50% chance of being 9% overloaded, and a 25% chance of 12% overloaded.

10.3.4 Control Failure

The control system is completely solid state with no moving parts to wear out. As a result the system should be extremely reliable. From information provided by Copley Motor Controls for one of their control units, a mean time before failure of one of their units is projected to be 164,000 hours of operation. If similar reliability is projected for this control system, the probability of a control failure is 1.4×10^{-6} during a typical round trip.

10.3.5 Winding Failure

The drive is composed of alternating copper windings and iron bands. The windings are pre-wound on insulating bobbins and then assembled into the iron core. In this application they closely resemble a transformer. For the analysis being conducted here, failure information for transformers will be used to represent the copper windings. According to the military reliability handbook MIL-HDBK-217A, a failure rate of 2.46 transformers per million operating hours should be expected. This works out to a 5.5×10^{-7} chance of a failure during a 13m 20s round trip.

10.3.6 Power Failure

The occurrence of a power failure while a skip is traveling vertically will quickly cause the car to slow or stop. Figure 71 was provided by Stanford University on power outage frequencies.

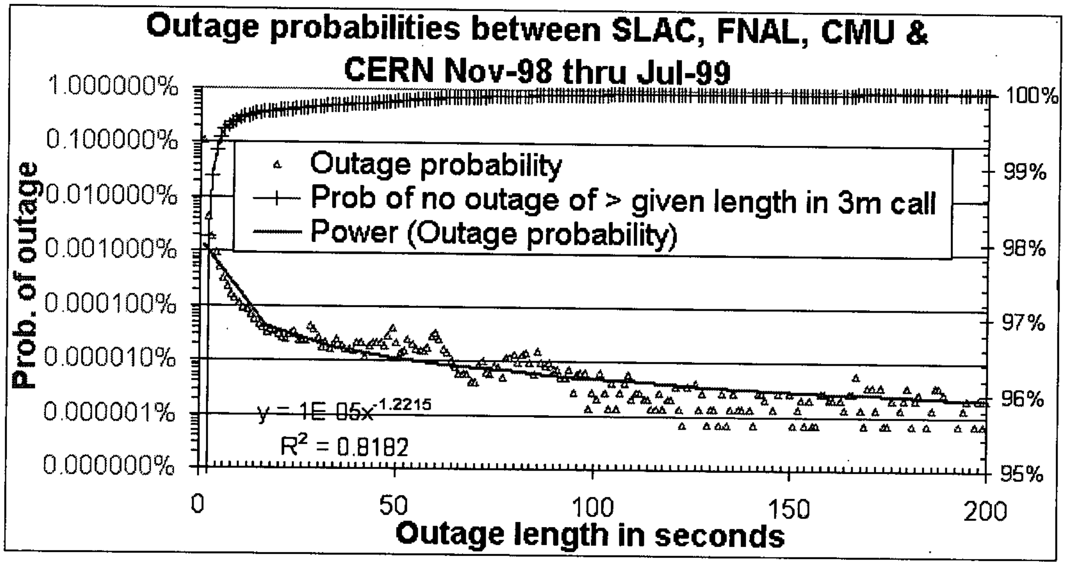

Figure 71: Power Outage Frequency. (IEPM 1999)

This data was used to represent the risk of failure in power supplied to the hoisting system.

10.3.7 Spilled Material

During the trip from the load station to the dump point a skip may spill some of its contents inside the drive tube. This spilled material may accumulate to a level where it could jam a passing car. The frequency at which this may occur is hard to determine. Reducing spillage will be an important aspect of the skip design. For this investigation it was assumed that 1 in 20,000 trips will encounter spilled material. When a skip encounters spilt material it is assumed that 10% will pass without problem, 40% will grind through, and 50% will get jammed by the material.

10.3.8 Human Sabotage and Error

There are locations in the system where people can gain access to the system. These points also provide locations where foreign objects can be thrown or dropped into the system. For this analysis, 1 in 100,000 trips will be impacted by malicious activity. When a car encounters an object there is a 5% chance that the skip will pass without a problem, a 30% chance that it is able to grind by the obstacle and continue on, and a 65% chance that it will become jammed.

10.3.9 Unpredicted Wear

After testing the prototype systems, the performance of the materials should be predictable. There will still be slight variations in materials or premature failures that will not follow the expected trends. To account for these instances 1 in 10,000 trips are predicted to encounter this problem. When a component wears in an unpredicted way there is a 90% chance that the skip will still reach surface where it can be pulled for servicing, there is a 5% chance that the car will grind its way through, and a 5% chance that the car will jam and come to a complete halt.

10.3.10 Misaligned Drive Tubes

If the drive tubes are mounted in well-supported excavations and installed with care than there should be only a small chance of the tubes becoming significantly out of alignment. Alignment problems are predicted to impact 1 in 20,000 trips. When a skip encounters an alignment problem there is a 10% chance that it will not be affected, a 50% chance that it will be able to grind its way through, and a 40% chance that it will become jammed.

10.4 Simulation

The software program Decision Tree by Palisade Decision Tools was selected to analyze the risk associated with the entire system. Unfortunately the risk flowsheet contained too many nodes for the program to solve. To overcome this problem, the flowsheet was recreated in EXCEL in a format that would allow iterative simulations to be run to determine the distributions of failures and near failures of the system. The simulation flowsheet is shown in Figure 72 below:

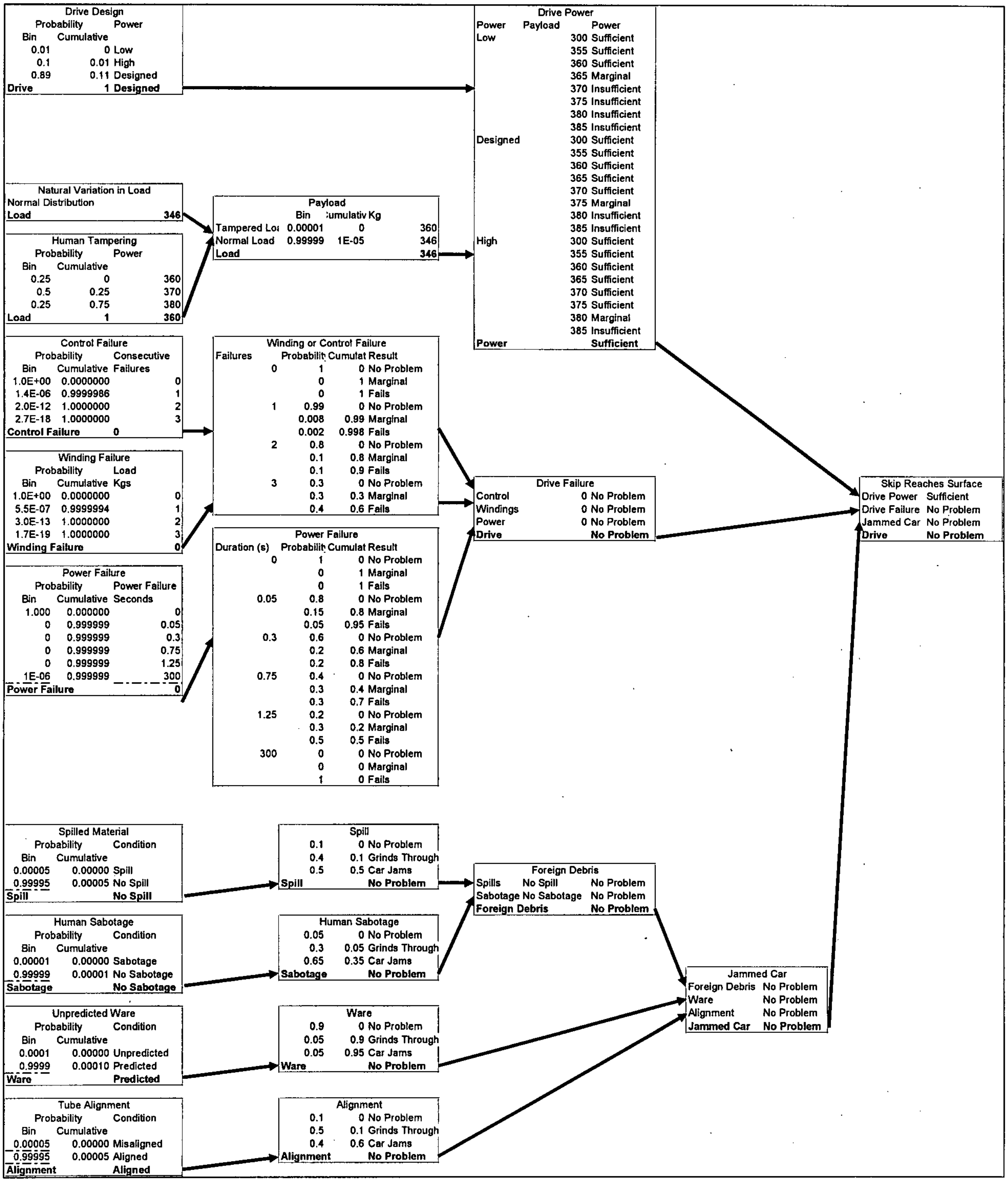

Figure 72: Risk Simulation Flow Sheet.

Values are selected from each of the boxes through random number generation. To achieve convergence, a large number of simulations need to be run. In this analysis, 2,000,000 trips were modeled which is the equivalent of 700,000 tonnes of material.

10.5 Results

The 2,000,000 iterations of the simulations was used to represent mining 700,000 tonnes of material per year. This allows the failure rate in cycles to be converted to a more meaningful time basis.

The results are shown in Table 6 below.

Table 6: Results of the Risk Analysis

	Count		Frequency	
Source	Marginal	Failure	Marginal	Failure
Power	91	2011	0.000030	0.000670
Sabotage	28	1	0.000009	0.000000
Spillage	62	40	0.000021	0.000013
Alignment	66	25	0.000022	0.000008
Material	9	6	0.000003	0.000002
Human Tampering	21	2	0.000007	0.000001
Low Drive Power	5	0	0.000002	-
Total	282	2085	0.000094	0.000695
Total Iterations	2,000,000	2,000,000		

The preceding table shows that in a year, 282 trips are expected to experience some trouble on their way to surface and 2085 trips will fail to reach surface. This is clearly not an acceptable failure rate for a material handling system. The primary reason for the very high failure rate is the occurrence of small power failures causing the cars to fall out of synchronization with the drive frequency. For this system to be attractive to the mining industry this failure frequency will have to be reduced significantly. There were not any failures of the system due to winding or control failures. This is due to the winding and control systems being highly reliable, wired in parallel, and requiring several consecutive failures to significantly affect a passing car.

After the first simulation run the magnetically propelled hoisting model was redesigned to improve its success. The power system was redesigned to incorporate a 5-minute buffer in the power supply system to prevent power failures of shorter duration from affecting the system. With the modified power profile, the results shown in Table 7 below were generated.

Table 7: Modified simulation results.

	Count		Frequency	
Source	Marginal	Failure	Marginal	Failure
Power	0	2	-	0.000001
Sabotage	21	7	0.000011	0.000004
Spillage	83	48	0.000042	0.000024
Alignment	84	20	0.000042	0.000010
Material	11	11	0.000006	0.000006
Human Tampering	0	7	-	0.000004
Low Drive Power	9	0	0.000005	-
Total	208	95	0.000104	0.000048
Total Iterations	2,000,000	2,000,000		

From the modified results a magnetically propelled hoisting system should be a fairly reliable system. Once in about every 3.5 days the system will experience a jammed car.

On average if it would take 4 hours to clear a skip and return the system to operation, the system will be available 96% of the time.

11 Economic Analysis

An economic analysis of this project was considered to be a very important for design for successful implementation of the technology into the mining industry. It was difficult to estimate accurate costs associated with such an innovative project at such a preliminary stage of development. As such, the costs reported here should be considered to be a preliminary estimate.

11.1 Capital Cost

The capital cost was estimated for implementing a magnetically propelled hoisting system by considering the following main components: skips, tubular linear motor, power supply, control system, loading facility, and dumping facility.

11.1.1 Skips

The skips are designed as a simple tubular steel construction. It is expected that the finished skips will weigh 50% of their rated payload and cost about $10/kg to build.

11.1.2 Linear Motor

The cost of building a 5kN, 0.5m diameter linear motor was estimated at $3000/m by Weisler and Rawlings. Half of this cost was scaled by the required drive force raised to the power of 0.6 which is a typical scaling factor for production units of this type. The other 50% was considered as a fixed component with respect to drive force, but dependent on tube diameter also scaled to the power of 0.6. Installation was estimated at $100/m.

11.1.3 Control System

The cost of installing a control system was estimated at $100/m. This will cover the cost to run the cabling from the power supply and hooking up the linear motor sections.

11.1.4 Power Supply

The cost of the power supply was based on the cost of the power supplied to an electric Kiruna trucking system. (ABB, 2001) A power supply of 1MW is estimated to cost $500,000 with a variable frequency source being twice that amount.

11.1.5 Loading Station

The cost of the crusher and motor was estimated by McLanahan Corporation at $276,000 (McLanahan, 2002) with the cost of bins, conveyors, and loading facility being twice that amount.

11.1.6 Dumping Station

The dumping station has not been fully designed so it is impossible to get an accurate cost for it. For this analysis $1M has been assumed to be necessary to build the dumping station.

11.2 Operating Cost

Operating and maintenance costs were broken down into the following components: electrical power, skip replacement, liner replacement, and linear motor replacement. Operating costs for the crushers were not included. It is assumed that total operating costs for crushing would be similar for both a conventional and MagLev system although the need to distribute crushers at each loading station would increase the capital and operating costs for MagLev crushing to some extent.

11.2.1 Electrical Power

Experience with Vancouver's SkyTrain has provided information on the electrical efficiency of a LIM. The SkyTrain is able to achieve 70% efficiency while propelling and a 20% efficiency to recapture energy during braking. These values were used to calculate energy consumption for the system.

11.2.2 Skip Replacement

The skips are scheduled for replacement after 150,000 loads (~1 year) with their liners being replaced every 30,000 loads (~2-3 months).

11.2.3 Linear Motor Replacement

The linear motors do not contain any moving parts, so they should not wear out. To account for flaws and unexpected failures 2% of the total motor cost is budgeted as an annual maintenance cost.

11.3 System Cost

The above cost items were incorporated into a spreadsheet to model the economics of the system. The final economic model is given in detail in Appendix XI. Using this model allowed a number of factors to be investigated.

The first scenario was the impact of skip diameter on the cost of the system. The model was run for skip diameters from 20 cm to 100 cm. The skips had a fixed length equivalent to three times their diameter. The costs were calculated to hoist 3000 tonnes per day, along a path consisting of 3000m of horizontal travel and 3000m of vertical travel. Figure 73 shows the results of the analysis.

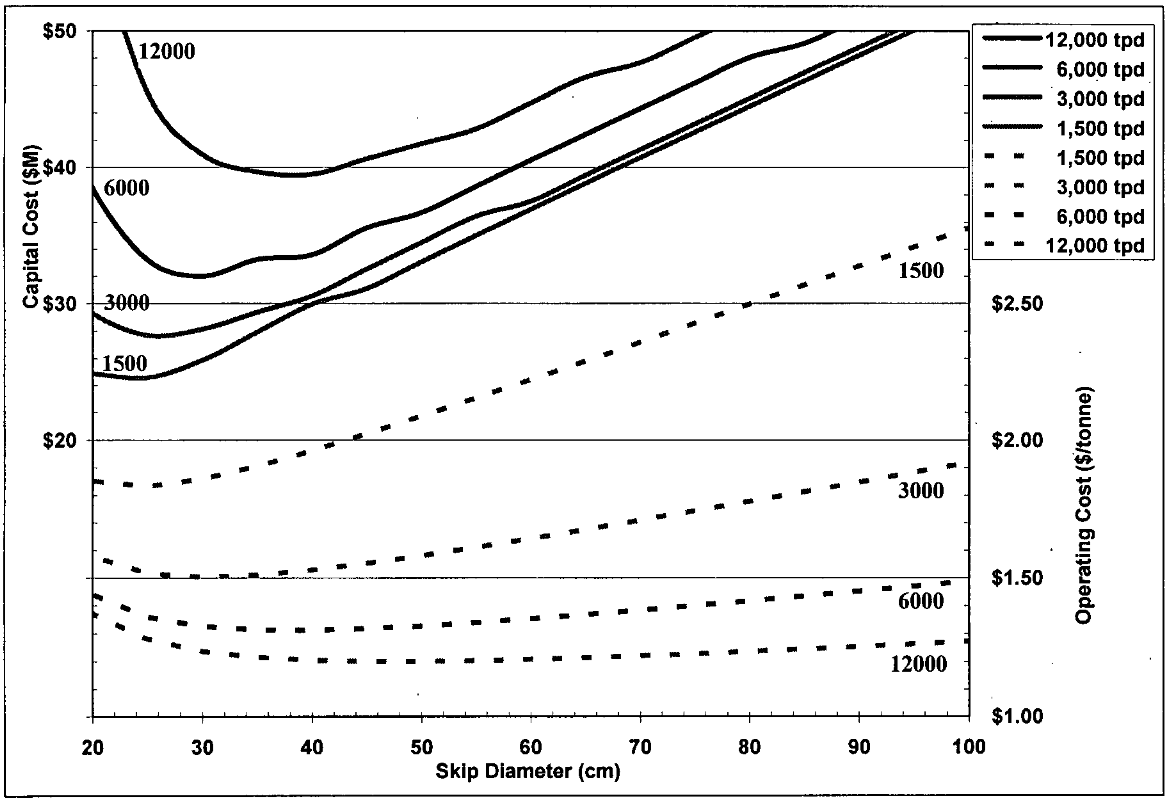

Figure 73: Effect of Skip Diameter on Capital and Operating Costs of a MagLev System.

From these data, the most economic skip size is approximately 30 cm in diameter. When the skip is smaller, the costs increase due to the large number of loading stations and individual skips required to load and move the tonnage. As the skip becomes larger, the cost escalates quickly due to the increasing cost of the drive tubes.

Although the most economic size appears to be around 30cm, it is believed that a 50cm diameter skip may be a more reasonable size from a practical sense for loading and will reduce the distributed crushing requirements which have not been accounted for here.

The next variable investigated was the length of the skip. The model was run again, this time with the skip diameter held at 50cm and the length changed from 10 cm to 400 cm. The results are shown in Figure 74.

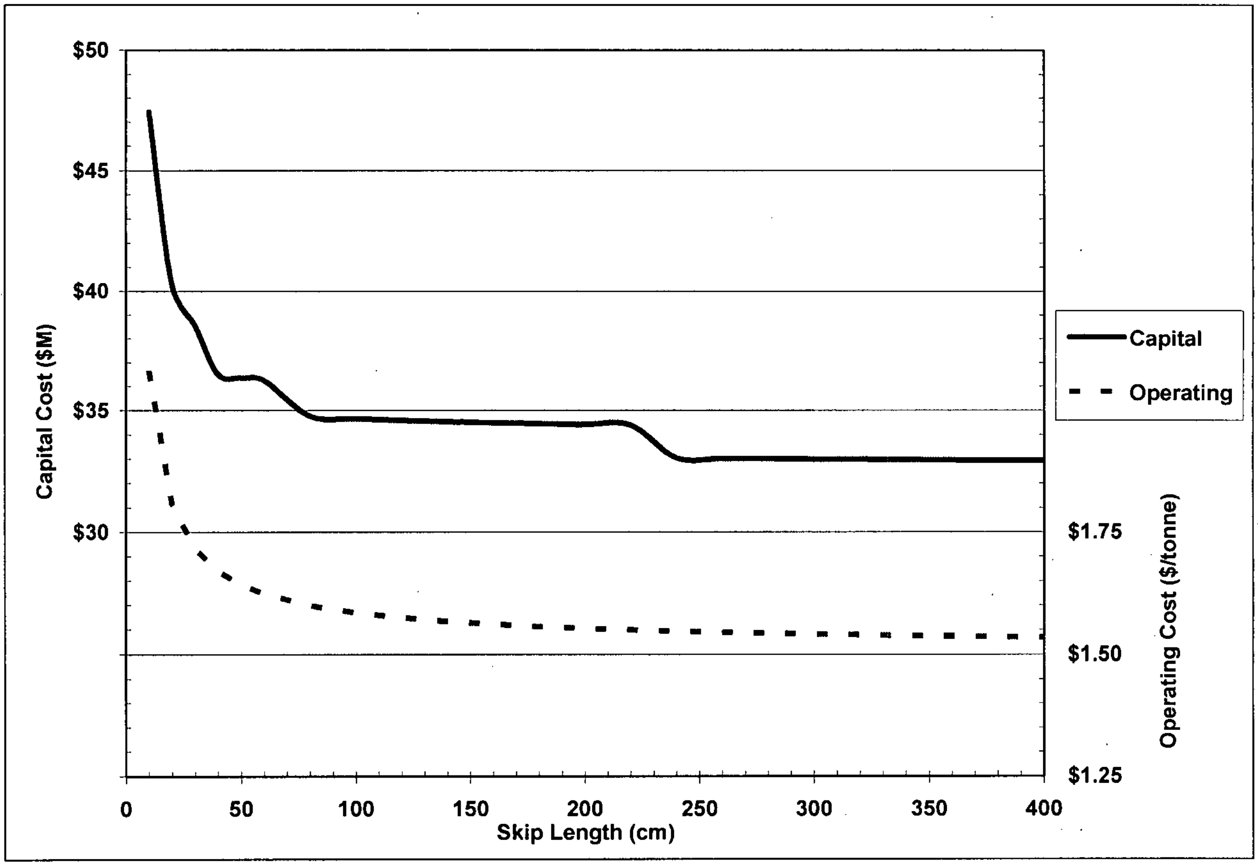

Figure 74: Effect of Skip Length on Economics.

When the skip is very short, not much material can be carried and so, a large number of skips and loading facilities are required to move the target tonnage. As the skips become longer, fewer skips are required and so, the costs decline. The bumps in the curve represent the "integer aspects" of the problem as loading stations are removed from the system using a rounding algorithm.

However, as skips increase in length they have increasing difficulty to negotiate corners and they become more difficult to handle or turn around in the confined space of an underground mine tunnel. From this study, it would appear that a skip with a length to diameter ration of 3:1 is a reasonable compromise.

With the size of skip now determined, the model was used to generate the economics of moving 3000 tpd of material vertically in an underground mine. The model was run for depths between 100m to 5,000m. The results of this analysis are shown in Figure 75.

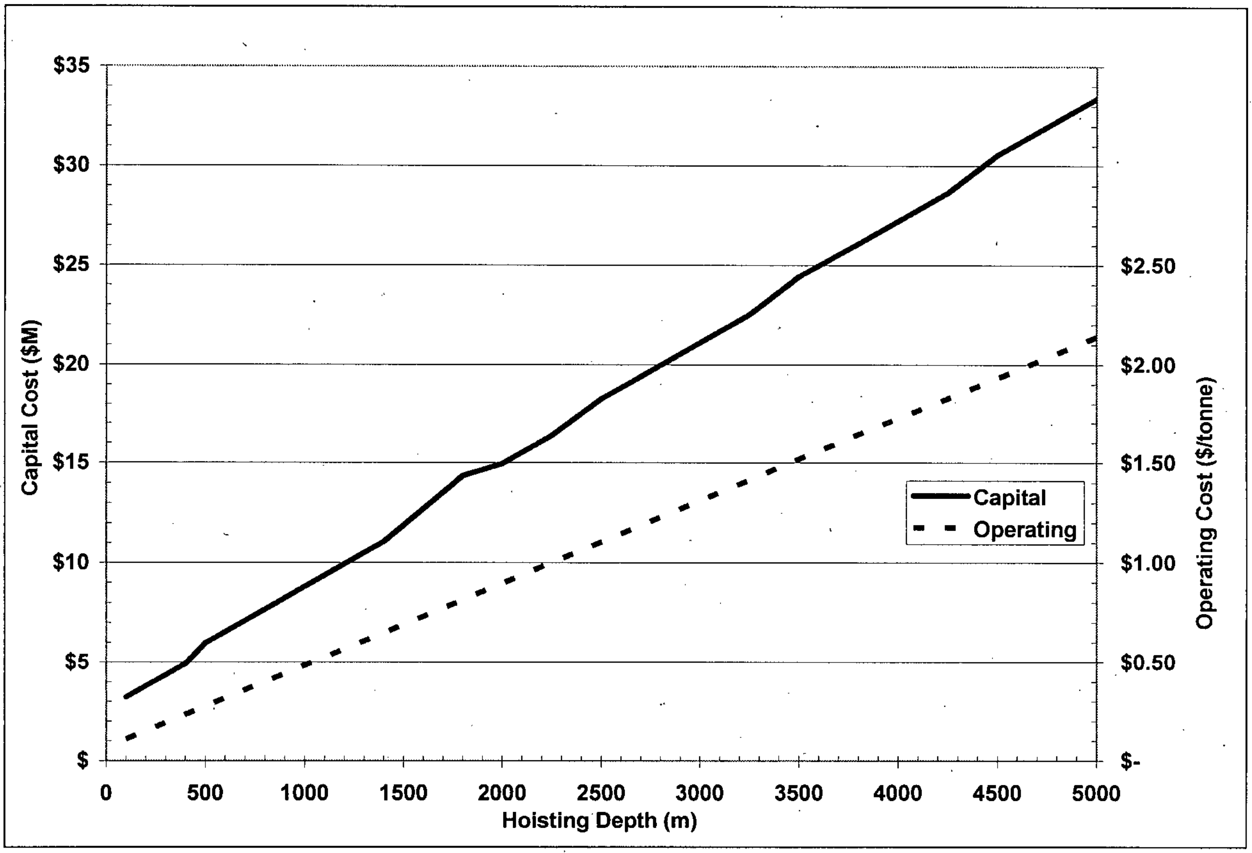

Figure 75: Effect of Hoisting Depth on the Economics of a MagLev Hoisting.

The above figure shows that both the operating and capital costs increase linearly with depth. This is a unique feature of a magnetically propelled hoisting system that is not the case with a conventional hoisting system. Since the system is semi-continuous, with the drive distributed along the entire trip; it is no longer affected exponentially by increasing depth. A deeper mine is accommodated by simply adding more sections and skips to the system. Since there is a dedicated tube to deliver the skips to surface and a second dedicated tube to return the skips to the underground loading point, both streams are separated at all times so there are no delays due to the interactions between the skips on their different directional journeys.

A similar analysis was run for a horizontal application. The model was run for level transportation over a distance from 0.1 to 30km. The results of this analysis are shown in Figure 76.

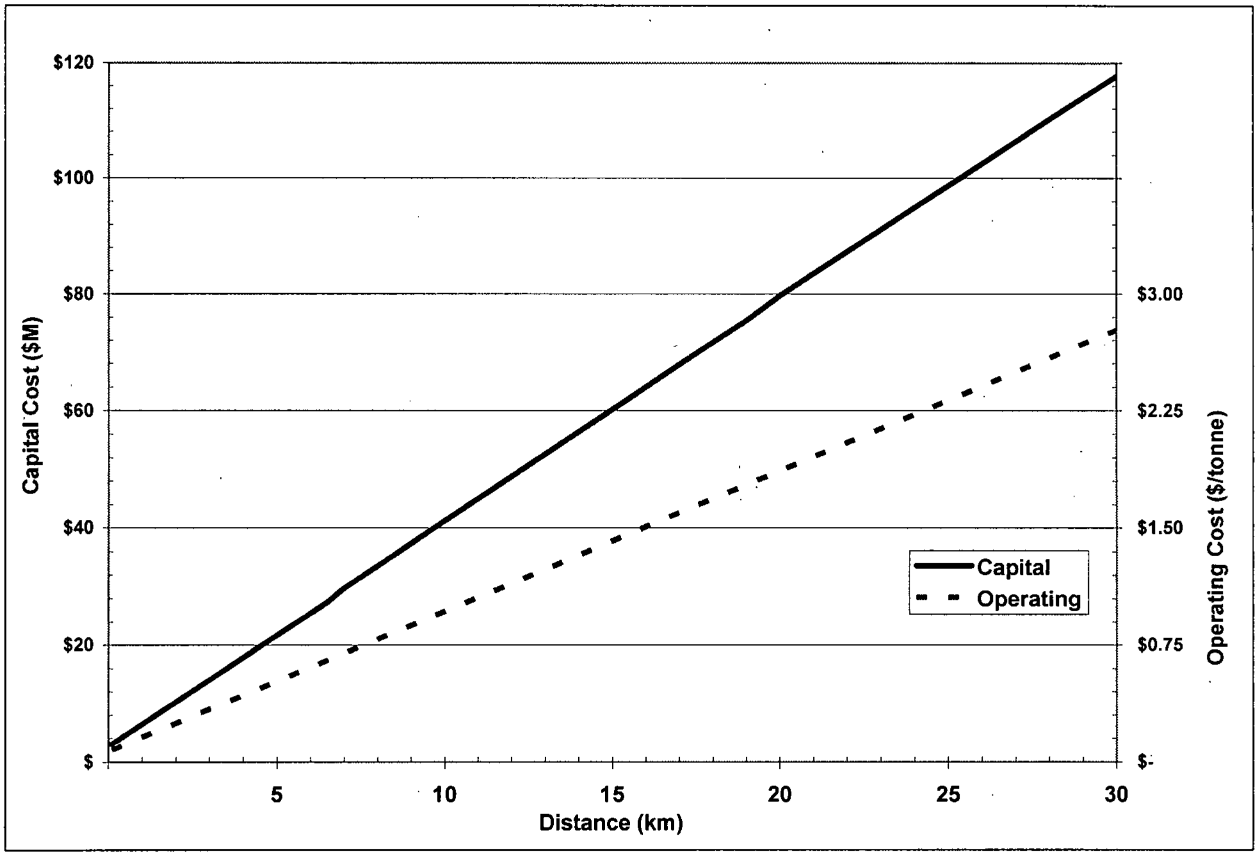

Figure 76: Effect of Horizontal Distance on the Economics of a MagLev Hoisting.

The above figure shows the same linear trend as found in the vertical case. The same reasoning applies to its linearity and this would also be exhibited by a conventional underground haulage system.

11.4 Case Study

Inco Ltd. operates an underground mine in Ontario Canada hoisting 2727 tonnes of rock from a depth of 1820 meters to surface. The hoisting system to move the skip full of rock vertically from 1,820 meters underground to surface has an estimated replacement capital cost of $14.2M and an operating cost of $1.18/tonne. (Krueger 2001) This does not include crushing, loading, or dumping facilities.

To determine if a magnetically propelled hoisting system could compete with this hoist, the economic model was run with Inco's parameters in this purely vertical application. Using a system with the following parameters:

Table 8: Proposed Production Design Parameters.

Drive Design	Value	Reason
Tube Diameter	0.52 m	To contain a 50cm diameter skip
Section Length	4.5 m	To enable handling of section
Width of iron	6.25 cm	To provide a 15m/s synchronous speed at 60hz
Width of Winding	6.25 cm	To match block width
Skip Design		
Material	Steel Tube	Lo cost material
Diameter	50 cm	Big enough to allow loading with out excessive crushing
Length	1.5m	Three times diameter
Required Number	85	Calculated
Loaded Weight	632 Kg	421 Kg of material and 212 Kg of skip
Speed	14.8 m/s	
Power (per skip)	146 kW	To move a loaded skip vertically.

The preceding parameters were then used for the economic analysis of the system. The results are shown in the following Table.

Table 9: Economic Comparison of MagLev with Conventional Hoisting

	Conventional Drum Hoist	Magnetically Propelled
Operating Cost	$1.18 / tonne	$0.73 / tonne
Capital Cost	$14.2 M	$ 13.8 M

The above table presents only part of the picture as it does not include the cost of crushing, loading and dumping facilities. The absence of these facilities is believed to favor magnetically propelled hoisting since it will require finer crushing and a more complicated loading arrangement. The analysis also does not include the secondary benefits of using a magnetically propelled hoisting system, like being able to integrate underground haulage with hoisting in the same system, to use a smaller shaft diameter, and to increase productivity. If Inco decided to double its production to 5,454 tonnes/ day, then an additional $2.1 M dollar would have to be spent on adding skips and power supplies to the system, not including the cost of additional loading stations. The increased capacity could be brought on line with little to no production delays. In a conventional hoisting system, doubling the capacity would require a whole new system or extensive modifications to the existing system. Both options will be expensive and severely impact operations while being implemented.

12 System Applications and Limitations

A magnetically propelled hoisting system has many unique characteristics that give it significant advantages over current technology used in the mining industry.

12.1 Deep Mines

When hoisting from increasing depths, conventional hoists have problems maintaining capacity. There are two factors that cause a loss in productivity. The first is that as depth increases it takes longer to lift the skip full of rock to surface, and then to lower it back down for the next load. The second factor is due to the strength of the cables. The cable at the top of the shaft not only has to carry the weight of the skip and rock, but also the weight of the entire cable connected to the skip. As depth increases, either less weight can be put into the skip or the cable needs to be stronger. Both of these options impact heavily on the operating cost of hoisting.

In a magnetically propelled hoisting system, the drive is not centralized at the top of the shaft; rather it is distributed along the entire length of the tube. This means that depth no longer affects the payload of the skip. It also allows more than one skip to be used at one time thus improving the time distribution of material arriving at surface. By adding more skips to the system, the impact of an increasing cycle time is avoided. The capacity of the system can also be increased if desired, by simply continuing to add skips until the limitation of the dumping or loading cycle time component is reached.

An additional benefit derives from the need for a very small opening in which to operate. Smaller underground openings are safer, more stable, and potentially cheaper to construct. The small portion of the shaft that is required for a magnetically propelled hoist may enable a shaft to be constructed in stages, deferring capital costs into the future and speeding up construction time.

12.2 Low Grade Deposits

Low-grade deposits are challenging to mine economically. To be profitable one must mine the deposit at a high tonnage to take advantage of the economy of scale. Traditionally these deposits are exploited using surface mining or block caving methods.

Magnetically propelled hoisting gives underground mining a key tool to continue mining low grade portions of an existing deposit or a new deeper low grade deposit. This ability derives from the movement of an order of magnitude more material through a smaller drift than a conventional system.

This benefit may enable a marginal resource to become an economic mine or in the case of an operating mine, it may allow mining to a lower cut off grade thus extending the mine life.

12.3 Integration With Continuous Mining

A magnetically propelled hoisting system is well suited to being integrated into a continuous miner. Through the use of multiple skips a nearly constant flow of material can be moved from the miner. The smaller fragment sizes generated by the miner can be directly loaded into the skips. The tubes for transporting the skips can be extended by adding sections, as the machine advances, similar in fashion to the way existing water and air lines are extended.

12.4 Integration of Underground Haulage and Hoisting

In a conventional mining sequence, it is common for the material to be handled and rehandled several times before it reaches surface: from loaders to trucks to chutes, to trains, to crushers, to bins, to skips, and finally to surface silos. It is difficult if not impossible, to keep track of all material through the entire cycle as it is mixed, spilt, and cleaned up, etc. on its way to surface.

In a magnetically propelled system, the material is loaded into the skips at a "near-face" location. On surface this same skip dumps its load. This not only avoids rehandling, spillage, and cleanup costs, but it also provides a unique opportunity to track material from the face to the mill. When a skip dumps, it is possible to know from where and when the material came. This could be very useful in isolating ores based on type, grade, or even mine-owner.

12.5 Integration of Mine with Milling

Magnetically propelled hoisting requires material to be of a small fragment size to enable loading of small skips. The small fragments may also lend themselves well to a preconcentrating step where most of the dilution would be separated from the ore. The separation may be based on density, conductivity, optical, or other differentiating property. The waste could then be sent to backfilling a stope while the ore is delivered to the mill. This will reduce material handling costs and milling costs as some of the waste is rejected much closer to the source.

12.6 Safety and Fail-Safe Systems

For the system to be acceptable to the mining industry it must remain safe and under control in all situations.

In the event of a power failure, the system must be able to keep the skips under control. If a regenerative braking system is used, it may be possible to design it to keep functioning when the main power fails. In this way, skips that are on their return trip back underground can remain safely under control.

The loaded skips being propelled to surface would coast for a second or two come to a stop and then begin to free fall back down the tube. This is clearly unacceptable. It may be possible to turn the drive windings into a regenerative braking system to slow the skips down. Alternatively, a series of one-way valves could be installed into the tube such that the skips can only move up the tube.

12.7 Important Issues to be Resolved

Being able to stop the skips in an emergency is a critical part of the design. One solution proposed in this work is to install a series of flapper valves that allow only motion in one direction. It would be necessary to evaluate whether the valves should be deployed at all times bouncing off each car as it passes, or held retracted until needed. How will it be determined when they are needed? How much shock loading can they withstand? How close together are they required? Further work is required to answer these questions.

13 Requirements for Future Research

The UBC-CERM3 magnetically propelled hoisting system is still in its infancy and there remains a significant amount of research before the system will be ready for implementation.

13.1 Shaft

The shaft in a magnetically propelled hoisting system will continue to serve the same purposes as in a conventional hoisting system: moving material, personnel, equipment, and providing ventilation and access for mine services to the underground workings. There are differences between the two systems that will require changes in the shaft design.

The compartments that contain the skips will be replaced with the two tubes for delivering and returning the skips. This will occupy a significantly smaller portion of the shaft area. The tubes must be held along their entire length to support the weight of the sections and the weight of the skips while in operation.

Ventilation requirements will also change for the mine. In a conventional hoisting system the drive motor is on surface and its waste heat is simply vented away. In a magnetically propelled system the waste heat will be released into the mine. This may require additional ventilation to keep the mine cool although the air pumping features of the system may assist in this requirement. Similarly if underground diesel equipment can be reduced by loading closer to the active mining areas, then the required quantity of ventilation may be reduced.

How a shaft design is altered to accommodate these changes has yet to be investigated.

13.2 Drive System

The drive system is in its earliest stages of development. A detailed design would be a great benefit to the project. Aspects such as synchronous or asynchronous operation, type of drive, material choice, cost, efficiency, and durability are all important aspects of the system that require further investigation.

13.3 Control System

In collaboration with the drive design, a control strategy needs to be developed. The control system must be able to accelerate, brake, and propel the skips. Simplicity and reliability will be critical aspects of the system.

13.4 Skip and Track Design

A lot of time has been spent on conceiving and designing different skip and track configurations and still only a small number of concepts have been devised. There are undoubtedly many concepts that have not been thought of just yet and many ways to improve upon the current design. Further work should be able to provide significant advantages to the existing concepts.

13.5 Loading and Dumping Stations

Loading and dumping stations require significantly more research with a goal to minimize cycle time, maintenance and operation costs and to ensure reliability and consistency. The systems will also have to respond quickly to deal with faults.

14 Conclusion and Recommendations

Conclusion

This research has established the following:

- A testbed has been constructed to enable future research into the concept of Magnetically-Propelled Hoisting Systems. The testbed design is flexible allowing one to study the system in different modes of operation from fully horizontal to fully vertical. Different types of electrical supply from DC to AC to synchronous to asynchronous operation can be applied. Different types of vehicle or skip designs can be tested, evaluating on board magnets, wheels, and effects of different shapes and sizes.
- Preliminary testwork shows that the system is at least capable of operating at a velocity of about 1.8 m/second. Once the power supply and control system is optimized it is anticipated that speeds of 12 m/second and perhaps even higher may be feasible.
- A risk assessment has been carried out which shows the failure potential to be low and likely controllable. A 96% mechanical availability is likely.
- A preliminary economic assessment shows that a MagLev system can be competitive with a conventional hoisting system with similar capital and operating costs. The additional costs of distributing the motor along the shaft are offset by the gains derived from a smaller shaft and no cable. Continuous operation from face-to-mill provide significant advantages through the integration of haulage and hoisting.
- Magnetic hoisting has a number of advantages over conventional hoisting:

  - Improved economics for certain mining applications
  - Integration of horizontal and vertical material handling into one system
  - Linear correlation between cost and depth
  - Ability to move very high tonnages through a very small opening
  - A flexible system that can respond immediately to an advancing face

Recommendations

There are a number of problems with the system that will hinder its acceptance by the mining industry. These are

- It is a new and unproven technology
- It has a lower electrical efficiency than conventional hoisting
- There is a need for new loading and dumping technology
- Its performance in adverse situations (power outages, dealing with foreign debris, component failures, etc.) require additional research and response system analysis.

- Future research must focus on how to make the power distribution work correctly. Once this is solved, detailed design test work can be accomplished to establish productivity, vehicle speed, control of multiple skips, switches for merging traffic, cycle times, etc.
- Additional studies need to examine different power delivery systems – AC vs. DC, synchronous vs. asynchronous.
- Alternative linear motor configurations could also be studied such as locating the motor on board the vehicle and using a magnetic rail as the linear component.

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Appendix I: Automated Loading Station

1.0 Introduction

As design work continued on the magnetically propelled hoisting system, the question of how long it would take to load a car was a continual concern. The time to load a car will determine the capacity of a single loading station and will define the system utilization. To gain some information on how quickly a skip could be loaded, a test model was constructed.

2.0 Model Mark 3 Design

This model is based on the design of the virtual Mark 3 loading station discussed previously although several modifications were made to simplify construction.

The skip used with this model had a rectangular cross section simply for convenience to make construction simple and cheap. This simplification does not significantly change the operation of the model as shown in Figure 1.

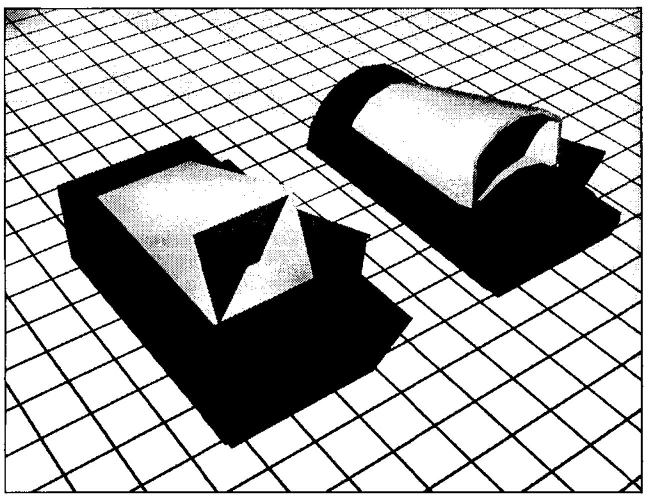


Figure 1: Square vs. Round Skip Design.

The second design change is that the bin and weight scale have been placed inside the curved track to make the system more compact and portable. The final loading model is shown in Figure 2 below:

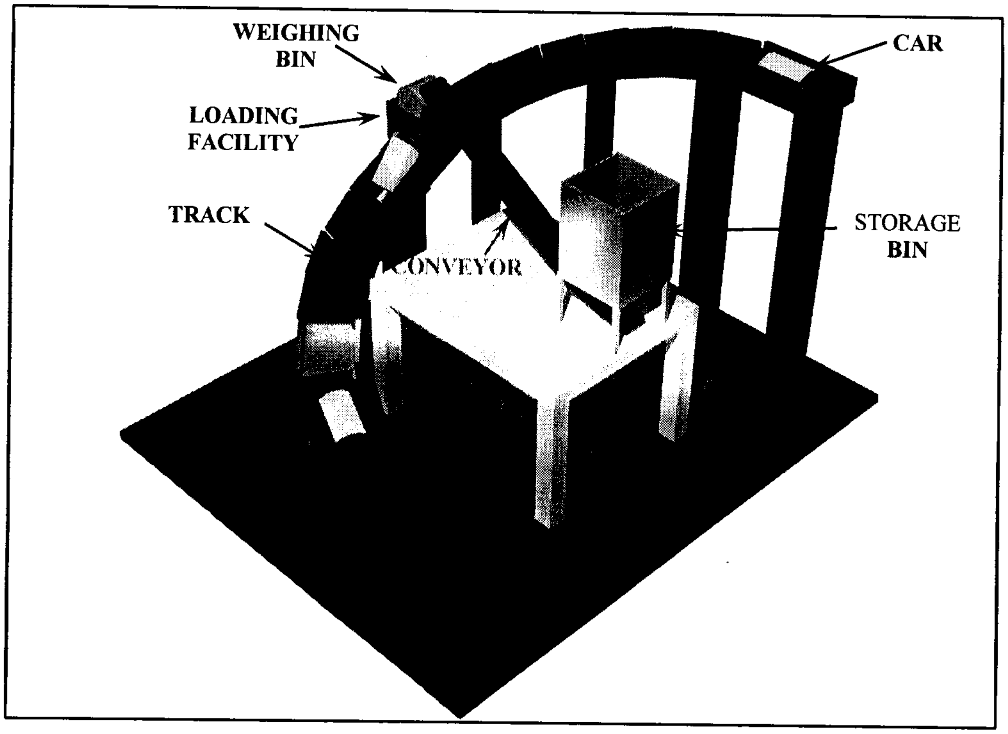


Figure 2: Loading Model Design.

Other than these changes, the model functions in much the same way as its real world counterpart. A car arrives at the top of the track; gravity propels it to the loading position where it is stopped, opened, loaded, closed, and then allowed to continue on its way.

2.1 Loadout Model Design Parameters

The model is designed to meet the parameters in the following table. These parameters control the actuator selection and system design.

Table 1: Design Specification

Parameter	Design Value	Tolerance*
Payload	1Kg	5%
Spillage	< 0.1%	
Load time	< 5s	
Automation	able to load without operator input	

* +/- 99% of the time

Justification for the selection of these parameters are as follows.

2.1.1Payload

The payload was selected to be 1kg to make the physical model small enough so it was cheap to construct and easily stored and moved. The importance of load tolerance is that the electromagnetic drive must able to hold and move the maximum load. If a load is significantly smaller than the maximum, the system will operate inefficiently. If a load exceeds the maximum, then the skip cannot be held or moved through the tube. The smaller the load deviation, than the higher the productivity and efficiency of the system.

2.1.2 Spillage
Spillage from the loading system must be cleaned up and reloaded by personnel. Spillage will be costly to cleanup and could possibly lead to additional system down time. Spillage problems are very obvious and are typically due to problems with the engineering of the design although if the control system operates ineffectively this can also produce wide swings in payload.

2.1.3 Load Time
The bottleneck in a magnetically-propelled hoisting system is the loading facility. To utilize the overall system in a mine, several loading stations will be distributed around the mine that merge into the same vertical drive system. Each loading station must be able to keep up with a scoop-tram that collects material from a nearby stope. The speed that each station can load a skip is clearly an important variable in the design.

2.1.4 Automation
In mining, labour costs compose a significant portion of the total operating costs because of high wage rates and the need for manually-operated systems. Accordingly an automated control system for this MagLev hoist is preferred.

In addition travel time from surface to the active faces can be as long as one hour leading to poor use of equipment unless many skips are used in the hoist. As more skips are used and as more loadout points are created at multiple faces, the complexity of vehicle speed and switching control at junction points becomes critical. These factors justify the use of automation. An automated system will react faster than a human operator and will be more consistent in its actions leading to faster cycle times.

2.2 Actuators
The Mark 3 load-out design requires a total of four separate actuators to perform the loading. The four actuators are the conveyor drive, the skip stopper, the skip opener, and the loading door actuator. These actuators are laid out schematically in Figure 3 below:

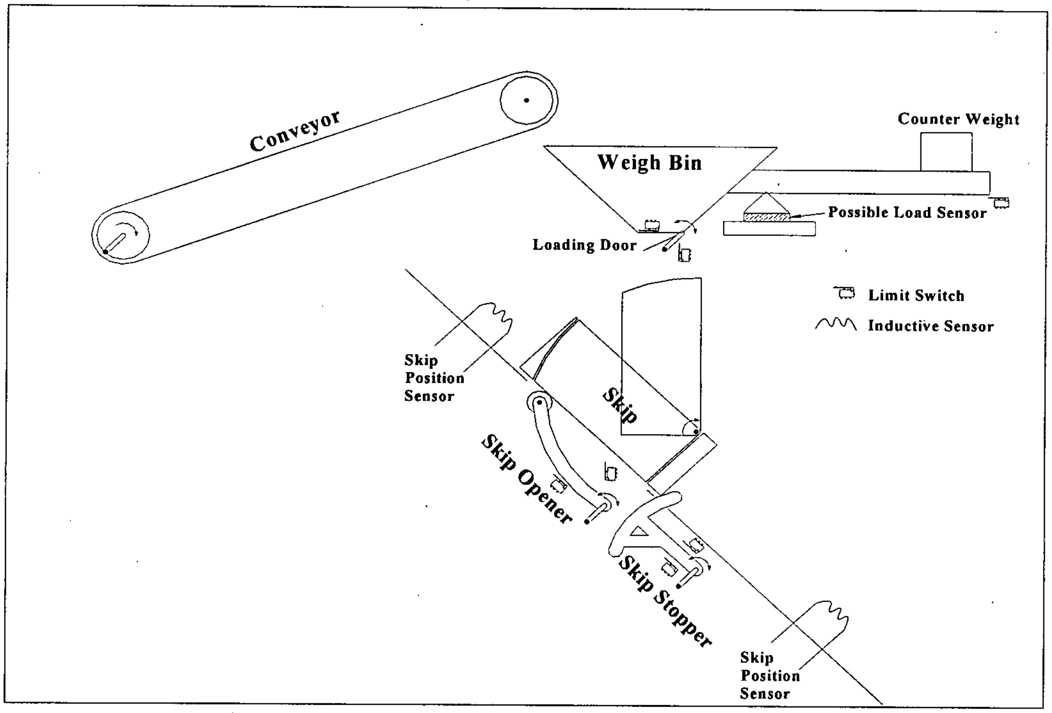


Figure 3: Loading Station Actuator Schematic.

A major constraint in the actuator design is the speed at which they must operate in order to achieve a loading cycle time below five second target. The following Gant chart in Figure 4 shows the required operating period for each actuator:

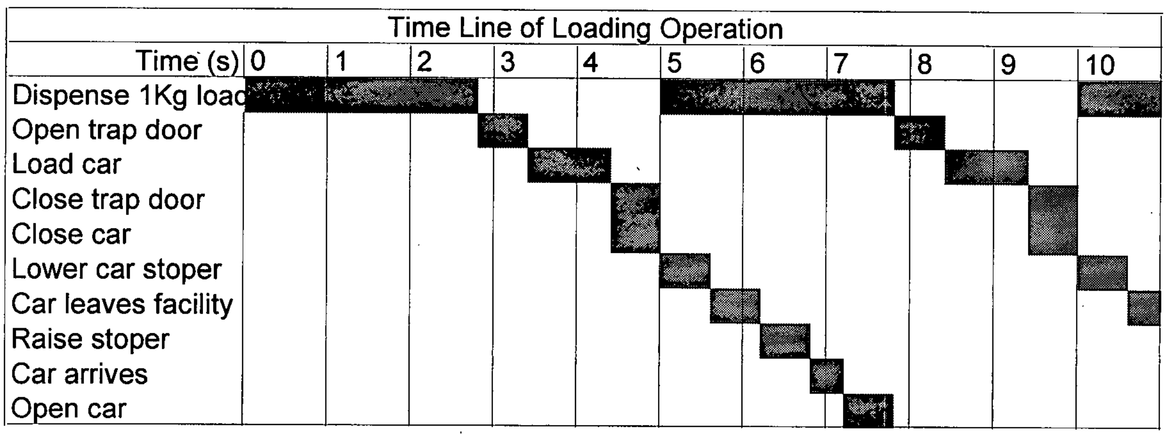


Figure 4: Design Cycle Time.

The time indicated in the above chart combined with the geometry of the facility enables the design and selection of the actuators.

2.2.1 Conveyor Drive

The purpose of the conveyor is to deliver 1 Kg of ore to the weigh bin in less than 2.8s. To achieve this, a conveyor with the following profile was designed:

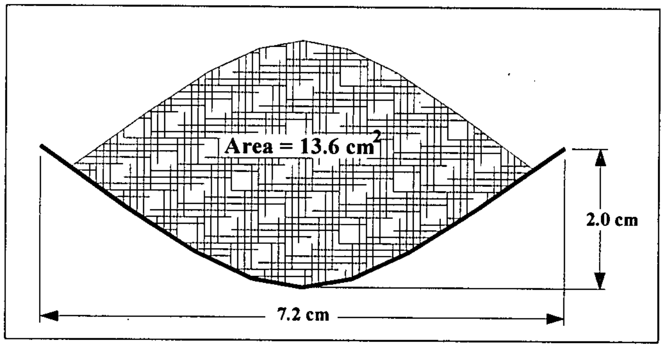


Figure 5: Conveyor Profile.

The conveyor must be driven at an average speed exceeding 9.5 cm/s to load the weigh bin in under 2.8s. This requires a drive torque of 0.327 Nm at 38 rpm. Detailed actuator designs are included as Appendix II.

To achieve this performance, the #0012 motor driving the conveyor through a 32:1 gearbox was selected which provides 45% over-design. The motor torque curves are included in Appendix III.

2.2.2 Skip Stopper

The purpose of the skip stopper is to place the skip in a pre-determined location to load the vehicle. Once the skip is loaded, the stopper is lowered to allow the skip to pass through the loading station. It is then raised back into position to stop the next skip. This will occur in less then 0.6s in both directions.

The actuator design is shown in Figure 6 below:

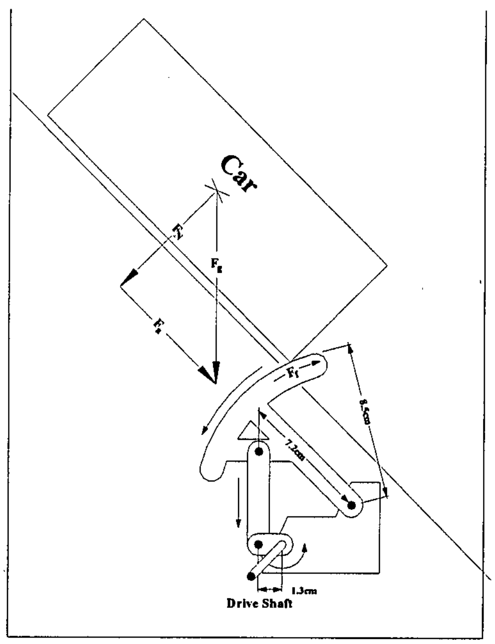


Figure 6: Skip Stopper Actuator Schematic Diagram.

This actuator requires a drive torque of 0.066 Nm at 50 rpm. A detailed design for this actuator is included in Appendix II.

To power this actuator, the #0004 motor powering the drive through a 22:1 gearbox is adequate. The motor torque curve is included in Appendix III.

2.2.3 Skip Opener

The purpose of the skip opener is to align the skip chamber with the bottom of the weigh bin. When the skip is loaded, the skip opener returns the skip chamber to its closed position. This actuator is shown schematically in Figure 7.

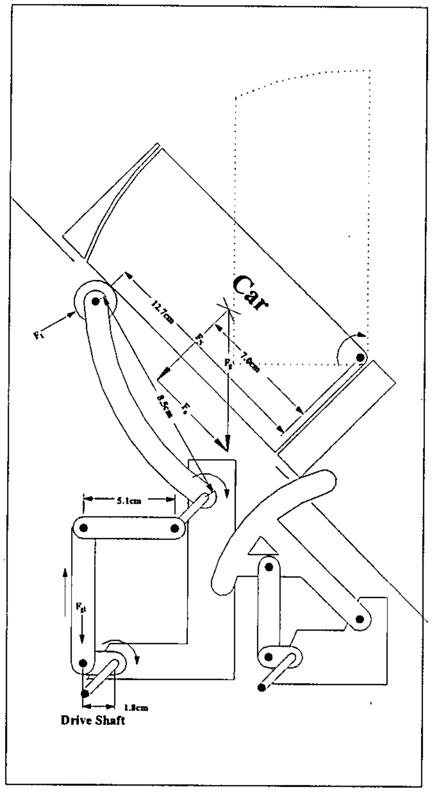


Figure 7: Skip Opener Actuator Schematic Diagram.

The skip opener requires a drive torque of 0.093 Nm at 50 rpm. The detailed design for this actuator is included in Appendix II.

To achieve this performance, the #0002 motor was selected to drive the opening mechanism using an 18:1 gearbox which provides 86% over-design. The motor torque curve is included in Appendix III.

2.2.4 Loading Door Drive

The purpose of the loading door drive is to power the loading door required to control the flow of material from the weigh bin into the skip chamber. This actuator is shown schematically in Figure 8.

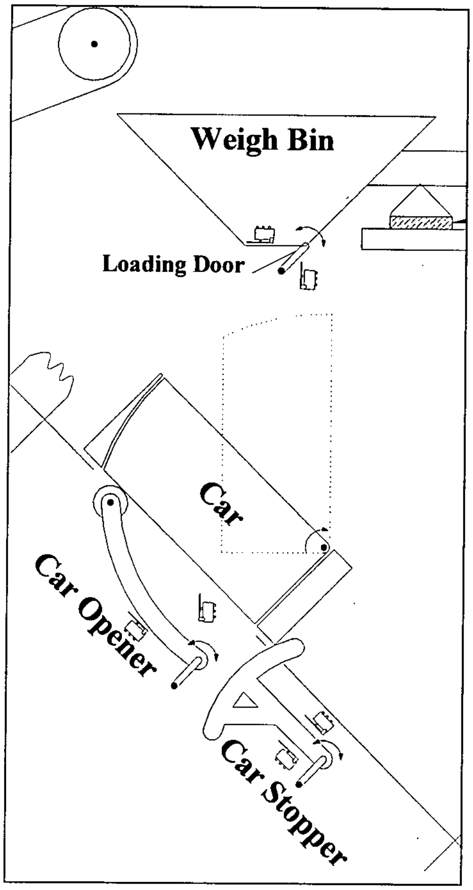


Figure 8: Loading Door Actuator Schematic Diagram.

The loading door drive requires a torque of 0.006 Nm at 800 rpm. A detailed design of this actuator is included in Appendix II.

To achieve this performance a #0005 motor was selected to drive the loading door mechanism which achieves a 32% over-design. The motor torque curve is included in Appendix III.

2.3 Sensors

To automate the loading station, a number of sensors are needed. These sensors provide the controller with three pieces of information: the position of each actuator, the weight of the weigh-bin, and the position of the skip.

2.3.1 Actuator Position

The first piece of information is the current position of each actuators. Since the actuators only have two positions, up or down, this information can be provided using limit switches. The required limit switches include:

- Skip stopper up
- Skip stopper down
- Skip opener up
- Skip opener down

- Loading door open
- Loading door closed

These limit switches are micro-switches which are simple mechanical devices. A lever is depressed which presses a button breaking the contact on the open side of the switch and closing the contact on the closed side. Micro-switches must meet the following specifications:

Table 2: Limit Switch Design Specifications.

Property	Design	S11SM25-H4
Voltage	6V	125V
Current	0.1A	1.0A
Frequency	1 Hz	> 20 Hz
Accuracy	3mm	1.5 mm

An S11SM25-H4 switch, manufactured by Honeywell, exceeds the required voltage, current, operating frequency, and accuracy required by the model and was selected for use in this model.

2.3.2 Weigh-bin Weight

The second piece of information is the weight of the weigh-bin. All the cars are identical so only one weight needs to be determined making the determination the hopper weight easy to control by counter-balancing the hopper with a position sensing limit switch. This is much cheaper and simpler arrangement than using load-cells at the mounting points of the hopper. This sensor is identical to the actuator position sensor.

2.3.3 Car Position

The third piece of information is the position of the car on the track. There are two positions that are critical: when a skip is in position to be loaded, and when a skip is clear of the loading station. To obtain this information, a permanent magnet is installed on the side of the skip with two inductive pickups located on the side of the track. Inductive sensors were chosen because of their high tolerance for dust and their resistance to vibration.

An inductive pickup is activated as the permanent magnet on the skip moves past a coil of wire. The changing magnetic flux induces a current in the coil that can be amplified and fed to the controller. The specifications for the pickup are given in the following table:

Table 3: Inductive Pickup Specifications

Air Gap	.5-8mm
Accuracy	1 cm
Frequency	5 Hz
Skip Velocity	.5-3 m/s

These specifications are achieved using a custom-built sensor consisting of a 0.5" ceramic-8 magnet on the skip and a 50-turn coil on side of the track connected to a voltage amplifier with 100,000 times amplification and a saturation voltage of 6V.

2.4 Automatic Controller

Control of the actuators can be accomplished with simple logic expressions that work well with the limit switch strategy. The logic statements are as follows:

Conveyor Drive: If "Loading Door" is "Closed" and "Weight" is "Low" then "Run"

Loading Door Open: If "Car Opener" is "Up" and "Weight" is not "High" and "Loading Door" is "not Open" then "Open"

Loading Door Closed: If "Car is loaded" and "Loading door" is "not closed" then "close"

Car Opener Down: If "Car is loaded" and "Car Opener" is "not down" then "lower"

Car Opener Up: If "Car is in" and "Car opener" is "not up" then "raise"

Car Stopper Down: If "Car is loaded" and "Car Opener" is "down" then "lower"

Car Stopper Up: If "Car is clear" and "Car opener" is "not up" then "raise" and "car is not loaded"

These rules were implemented using the CMOS logic chips controlling relays that change the position of the actuators.

The limit switches are supplied with 6V DC power so they operate in a digital fashion returning a value of either 6V or 0V. They do not require any further signal conditioning to feed the CMOS chips. The signal from the inductive pickups requires amplification to be able to trigger the controller. Once the signal is amplified, a 6V DC pulse is obtained which is suitable for input to the controller.

The wiring diagram for this control structure is included as Appendix IV.

2.5 Power Supply

The control system and the actuators are powered from independent power supplies. This allows for the actuators to have their voltage manipulated with out affecting the control system performance.

The peak current draw from the actuators occurs when the conveyor and the car opener are running. This draws a total of 3A requiring a 36W power source. An existing 60W transformer and rectifier will be able of supplying this power load.

The sensors are run through a 6V 1A power regulating chip. The maximum current draw is estimated to be 0.6A. This chip requires a 6W power source. An existing rectified 13W 7V power source was chosen to provide its power.

3.0 Model Performance

The final model is shown in the following figure:

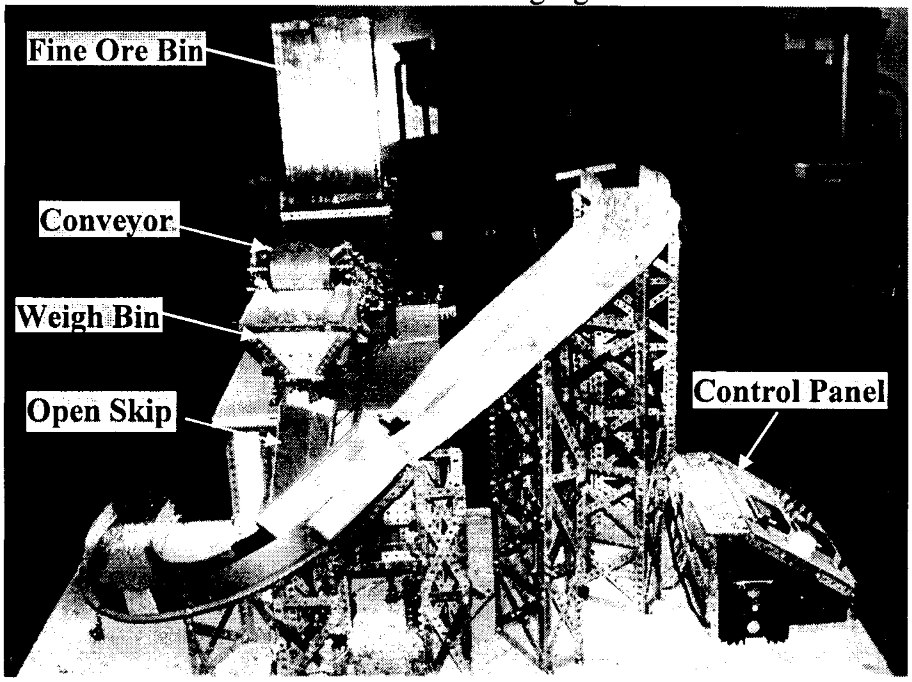


Figure 9: Final Model.

The performance of each actuator in the model is laid out in the following sections.

3.1 Conveyor Drive

The conveyor required an additional 0.3s to load the weigh bin. This was a result of the profile of the material on the conveyor being significantly smaller than designed. The loading of the hopper can be made faster by either changing the profile of the material in on the conveyor or by increasing the speed of the conveyor. Since the drive for the conveyor is significantly over designed the simplest solution would be to decrease the gear ratio of the drive to increase its speed.

3.2 Loading Door

The loading door performed better than predicted. It was able to operate 0.2s faster than designed.

3.3 Skip Opener

The drive for the skip opener was significantly over designed this lead to the actuator operating at a speed well above design. This caused the skip to be violently flung open and the control system was not capable of controlling its function. The design of this actuator was modified by increasing the gear ratio of its drive. This brought the speed of the actuator back to design specs and returned the actuator to a controllable state.

3.4 Car Stopper

The skip stopper functioned as designed.

4.0 Model Results

The only initial design change to the model was in the design of the weigh bin. The conceptual design had a rotating load hopper. In practice this was going to increase the vertical profile of the loading station. This design was rejected in favor of a stationary hopper with a trap door for loading. The main problem encountered during testing occurred when the skip was loaded, the skip's center of gravity moved forward of the front wheels causing the skip to become unstable. This problem was rectified by adding lead to the back of the skip and moving the front wheels further forward.

The ability of the model to meet the design parameters is presented in the following sections.

4.1 Payload

The design was supposed to load the cars with 1kg +/- 5% 99% of the time. To test this 25 consecutive loads were weighed. The weights of the loads are shown in the following histogram.

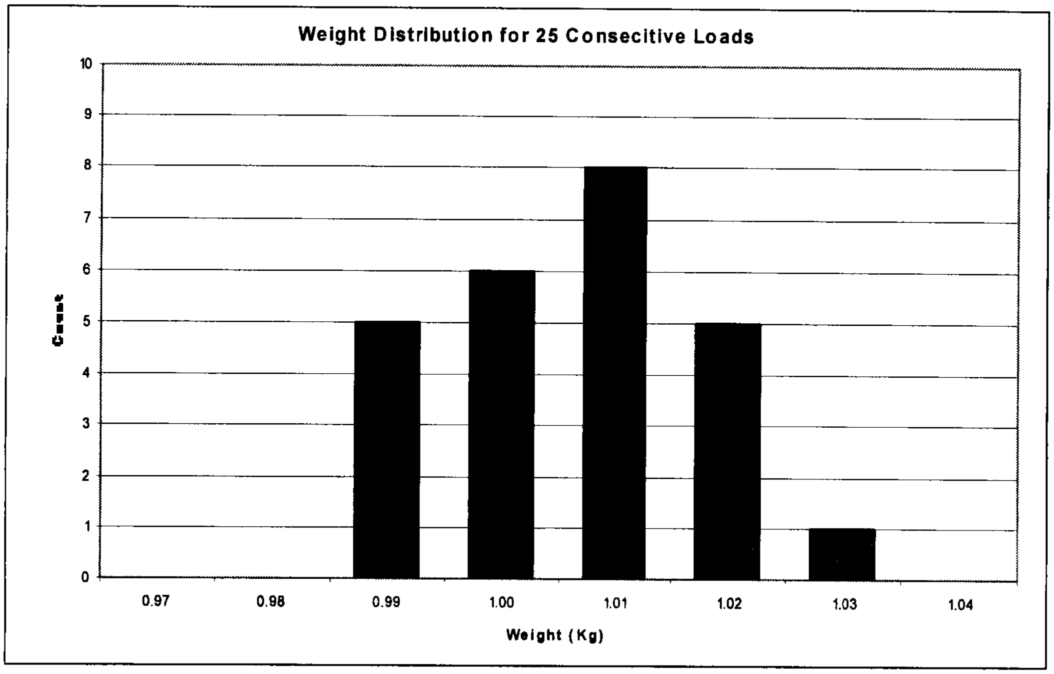


Figure 10: Load Distribution.

The loads averaged out to 1.00 kg with a standard deviation of 0.0012 kg. This works out to be 1.0 kg +/- 3.5%, 99% of the time. So this made the specification.

4.2 Spillage

The Target for spillage was to be below 0.1%. To measure spillage the model was cleaned of all previously spilt material then 21 consecutive cars were loaded then the spilt material was collected and weighed.

The spillage in three main areas; track, conveyor area, and loading area, were collected to evaluate model performance. The spillage in the three areas is shown in the following photographs.

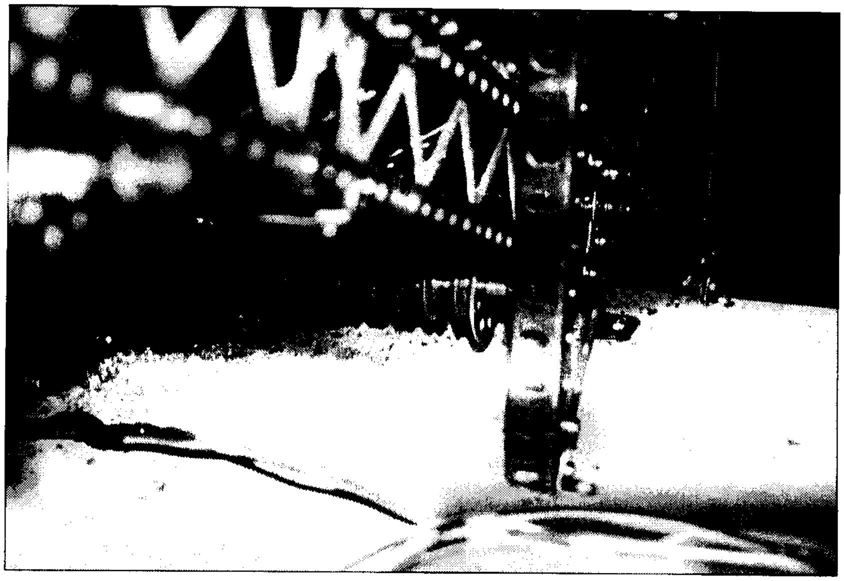


Figure 11: Spillage Around Conveyor Tail Spool.

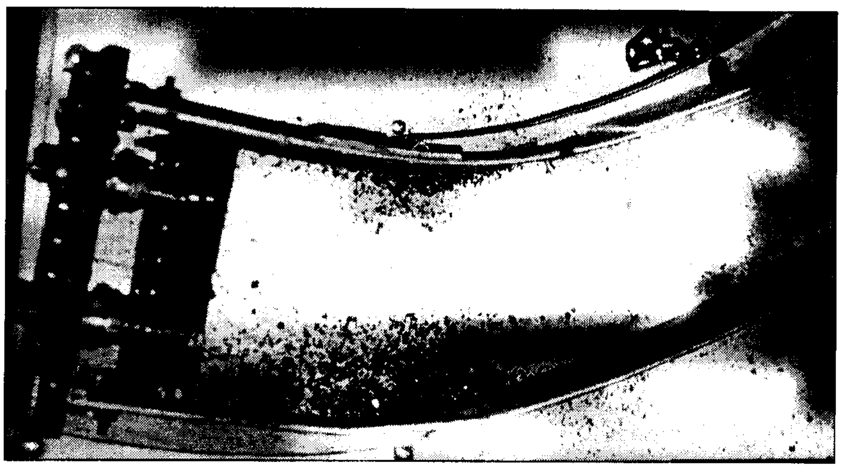


Figure 12: Spillage on Track


Figure 13: Spillage Around Loading Area

The weight of the total spillage for the 21 trials is shown in the following table:

Table 4: Spilt Material

Location	Weight (g)	%
Conveyor Tail	15.3	0.1%
Track	22.4	0.1%
Base	105.3	0.5%
Total	143.0	0.7%
Payload	1	Kg
Loads	21	#
Total Weight	21,000	g

This table shows that the target of < 0.1% spillage was not achieved. The spillage was primarily due to two sources.

The first source is from the area around the tail spool of the conveyor. This spillage could be reduced by reducing the slope of the conveyor under the chute.

The main source of spillage is from the skip loading arrangement. The main problem is that as the loading door opens, material flows off the side of the door missing the load chamber on the waiting car. This could be prevented by adding a skirt to the bottom of the weigh bin to prevent material flowing off the side of the loading door.

4.3 Loading Time

The cycle time for the loading system was timed by averaging 10 consecutive operating cycles. These times were then plotted on the design Gant chart as shown below:

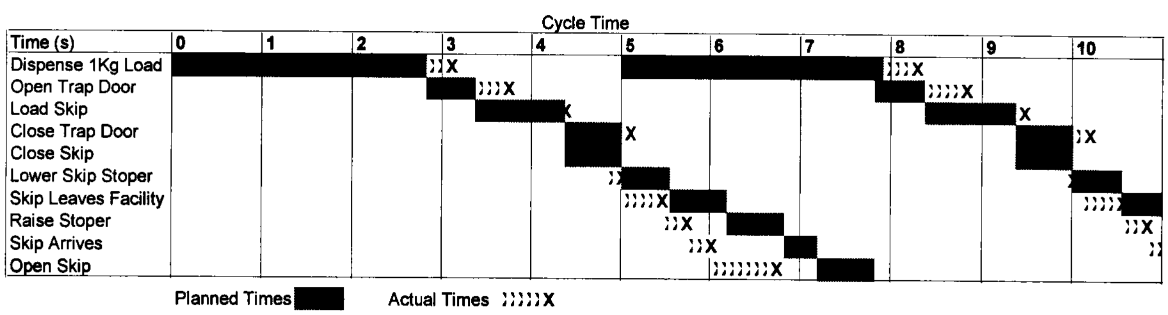


Figure 14: Actual Cycle Time.

The Gant chart shows that the cycle time is 5.1s. The critical path for the loading time is controlled by the sequence: conveyor loading weigh bin, door open, load skip, door closed. A reduction in any of these times will reduce the cycle time. The easiest time to shorten is the time taken for the conveyor to load the weigh hopper. To meet specifications the gear ratio in the conveyor drive should be reduced to increase the conveyor speed resulting in a shorter load time.

4.4 Automated Skip Loading

The controller is capable of automatically loading the skips. The controller has very limited ability to detect problems with the skip loading. This is a significant problem with this controller design.

To improve the performance of the controller the loading area needs a couple of additional sensors to indicate if the car has been successfully opened. Inductive sensors will not be able to detect the skip when it is not moving so they will not be an option. A proximity sensor would be a better option to detect a stationary metal skip.

5.0 Mark 3 Design Modifications

As a result of this model there are five significant design changes that have revised the Mark 3 design.

The first change is that with the revised skip design the track no longer needs to be steeply inclined for proper skip loading. The revised track is still inclined at 10% to simplify the propulsion of the cars through the loading station.

The second change is to the weigh bin design. The rotary design first envisioned requires too much vertical height for a semi-portable underground application. The revised design will have a fixed weigh bin with a trap door to control material flow.

The third change is to alter the profile of the conveyor under the storage bin to prevent spillage of material.

The fourth change to the design is to add a skirt to the bottom of the weigh hopper to prevent material from flowing off the side of the loading door.

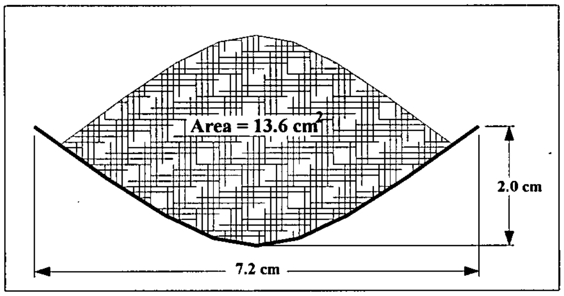
The final change is to alter the skip design to reduce the problems with the loading chamber binding in the loading car. The problem is that some small pieces of rock were able to work their way between the skip and the bin binding the joint. To prevent this, the design will be altered to change the profile of this gap making it tapered from a narrow gap at the top to a wider gap at the bottom preventing particles from hanging up in the gap.

Appendix II: Detailed Actuator Design

Conveyor design

Material Properties

Density	$4.6 t/m^3$
Swell	65%
Angle of Repose	37^O



Specifications:

Length	83.8 cm
Length of load	73.0 cm
Width	12.9 cm
Inclination	17^O
Belt width	7.3 cm
Trough Depth	2.0 cm
Max Area	$13.6 cm^2$
Rolling Res.	3%
Drive Spool R.	2.4cm

Required Drive Torque

$T_{tot} = T_F + T_{RR} + T_G + T_I$

T_{tot} = Total Torque

T_F = Torque to overcome friction in empty conveyor

T_{RR} = Torque to overcome the additional rolling resistance in the loaded conveyor

T_G = Torque to overcome the force of gravity on the conveyor's load

T_I = Torque to overcome inertia in actuator

T_F = 0.118 N*m *(measured from the drive shaft on the tail spool of empty conveyor)*

$T_{RR} = F_n * R_R * R_d$

F_n= normal component of conveyor load

R_R = rolling resistance

R_d = radius of drive spool

$F_n = \sin(90^O-17^O)*(13.6cm^2 * 73cm)*(4.6g/cm^3)*(1bcm/1.65lcm)*(.0098N/g)$
$= 25.9N$

$T_{RR} = 25.9N * 3\% * 0.024m = 0.019\ N*m$

T_G = (component of force along conveyor)(radius of drive spool)

$= \sin(17^O)*(13.6cm^2 * 73cm)*(4.6g/cm^3)*(1bcm/1.65lcm)*(.0098N/g)*(0.024m)$

$= .190N*m$

$T_{tot} = .118+.019+.190 + T_I$

$\mathbf{= 0.327 + T_I\ N*m}$

Required RPM of Drive

Weight to move = 1000g (design parameter)

Distance to travel = $1000g/[(13.6cm^2)*(4.6g/cm^3)*(1bcm/1.65lcm) = 26.4cm$

Revolutions of drive spool = $26.4cm/(\pi*2*2.4cm) = 1.75$ revolutions

Time to move material = 2.8s *(from design time line to achieve >5s loading time)*

RPM of Drive = 1.75 rev/2.8s = **38 RPM**

Drive must be able to exceed 38 RPM with an applied load of over 0.327N*m.

This will require a 3 stage reduction to generate sufficient torque.

Drive chain pinion 36 teeth
Stage 2 driver pinion 14 teeth

Required stage 2 RPM = (38RPM)*(36teeth/14teeth) = 98 RPM
Required stage 2 torque = (0.327N)/(14teeth/36teeth)/(90%eff) = 0.141N*m

Sage 2 driven pinion 57 teeth
Stage 1 driver pinion 19 teeth

Required stage 1 RPM = (98RPM)*(57teeth/19teeth) = 294 RPM
Required stage 1 torque = (0.141N)/(19teeth/57teeth)/(90%eff) = 0.052N*m

Stage 1 driven pulley 47.5 mm
Motor driver pulley 11.4 mm

Required Average Motor RPM = 294*(47.5/11.4) = **1,225 RPM**
Required Motor Torque before Inertia loads = 0.052(11.4/47.5)/(90%eff) = **0.014N*m**

Now that the actuator is designed the inertial load for the actuator can be found.

Assume that 40% of the mass of the motor is the rotor and that the rotor is equivalent to a solid cylinder

$$\text{Motor inertia} = \tfrac{1}{2}\,mR^2 = \tfrac{1}{2}\,(0.745)(40\%)(0.02\text{m})^2 = 5.96\text{x}10^{-5}\ \text{Kgm}^2$$

$$\text{Motor Drive Inertia} = J_{pulley} + J_{boss}$$
$$J_{pulley} = (\tfrac{1}{2})(0.0021\text{kg})(0.0057\text{m})^2 = 1.38 \text{ x } 10^{-6}\text{Kgm}^2$$
$$J_{boss} = (\tfrac{1}{2})(0.0028\text{kg})(0.005\text{m})^2 = 3.5 \text{ x } 10^{-8}\ \text{Kgm}^2$$
$$= 1.42\text{x}10^{-6}\ \text{Kgm}^2$$

$$\text{Stage 1 Inertia} = J_{pulley} + J_{boss} + J_{boss} + J_{washer} + J_{pinion} + J_{boss} + J_{boss} + J_{washer} + J_{shaft}$$
$$J_{pulley} = (\tfrac{1}{2})(0.0049\text{kg})(0.0238\text{m})^2 = 1.38 \text{ x } 10^{-6}\text{Kgm}^2$$
$$\sum J_{boss} = (4)(\tfrac{1}{2})(0.0028\text{kg})(0.005\text{m})^2 = 1.40 \text{ x } 10^{-7}\ \text{Kgm}^2$$
$$\sum J_{washer} = (2)(\tfrac{1}{2})(0.0006\text{kg})(0.005\text{m})^2 = 1.50 \text{ x } 10^{-8}\ \text{Kgm}^2$$
$$J_{pinion} = (\tfrac{1}{2})(0.0057\text{kg})(0.007\text{m})^2 = 1.39 \text{ x } 10^{-7}\text{Kgm}^2$$
$$J_{shaft} = (\tfrac{1}{2})(0.0089\text{kg})(0.0019\text{m})^2 = 1.61 \text{ x } 10^{-8}\text{Kgm}^2$$
$$= 1.69\text{x}10^{-6}\ \text{Kgm}^2$$

$$\text{Stage 1 Inertia brought to Motor Shaft} = (1.69\text{x}10^{-6}\ \text{Kgm}^2)(11.4/47.5) = 4.08\text{x}10^{-7}\ \text{Kgm}^2$$

Stage 2 Inertia = $J_{boss} + J_{washer} + J_{pinion1} + J_{boss} + J_{boss} + J_{washer} + J_{shaft} + J_{pinion2} + J_{boss} + J_{chain}$

$\sum J_{boss} = (4)(½)(0.0028kg)(0.005m)^2 = 1.40 \times 10^{-7}$ Kgm^2
$\sum J_{washer} = (2)(½)(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8}$ Kgm^2
$J_{pinion1} = (½)(0.0135kg)(0.0198m)^2 = 2.46 \times 10^{-6}$Kgm^2
$J_{shaft} = (½)(0.0089kg)(0.0019m)^2 = 1.61 \times 10^{-8}$Kgm^2
$J_{pinion2} = (½)(0.0036kg)(0.011m)^2 = 2.17 \times 10^{-7}$Kgm^2
$J_{chain} = (0.0063kg)(0.011m)^2 = 7.62 \times 10^{-7}$Kgm^2

$= 4.87 \times 10^{-6}$ Kgm^2

Stage 2 Inertia brought to Motor Shaft = $(4.87 \times 10^{-6}$ Kgm$^2)(11.4/47.5)(19/57)$
$= 3.89 \times 10^{-7}$ Kgm^2

Conveyor Drive Inertia = $J_{pinion} + J_{boss} + 2J_{boss} + 2J_{washer} + 6J_{h/t\ pulley} + J_{belt} + J_{load} + 2J_{boss} + 2J_{washer} + 2J_{shaft}$

$J_{pinion} = (½)(0.0294kg)(0.0245m)^2 = 8.82 \times 10^{-6}$Kgm^2
$\sum J_{boss} = (5)(½)(0.0028kg)(0.005m)^2 = 1.75 \times 10^{-7}$ Kgm^2
$\sum J_{washer} = (4)(½)(0.0006kg)(0.005m)^2 = 3.00 \times 10^{-8}$ Kgm^2
$\sum J_{h/t\ pulley} = (6)(½)(0.0464kg)(0.0245m)^2 = 8.36 \times 10^{-5}$Kgm^2
$J_{belt} = (0.4981kg)(0.00245m)^2 = 2.99 \times 10^{-6}$Kgm^2
$J_{load} = (13.6cm^2 * 73cm)*(4.6g/cm^3)*(0.0035m)^2$
$\sum J_{shaft} = (2)(½)(0.0119kg)(0.0019m)^2 = 8.58 \times 10^{-8}$Kgm^2

$= 1.52 \times 10^{-4}$Kgm^2

Drive Inertia brought to Motor Shaft =$(1.52 \times 10^{-4}$ Kgm$^2)(11.4/47.5)(19/57)(14/36)$
$= 4.71 \times 10^{-6}$ Kgm^2

Conveyor Primary Idler Inertia = $14J_{idler}$

$J_{idler} = (14)(½)(0.0241kg)(0.009m)^2 = 1.37 \times 10^{-5}$Kgm^2

Primary Idler Inertia brought to Motor Shaft
$= (1.37 \times 10^{-5}$ Kgm$^2)(11.4/47.5)(19/57)(14/36)(4.8/1.77)$
$= 1.15 \times 10^{-6}$ Kgm^2

Conveyor Return Idler Inertia = $20J_{pulley} + 20J_{boss} + 4J_{boss} + 4J_{washer} + 2J_{shaft}$

$J_{pulley} = (20)(½)(0.0060kg)(0.0116m)^2 = 8.07 \times 10^{-6}$Kgm^2
$\sum J_{boss} = (24)(½)(0.0028kg)(0.005m)^2 = 8.40 \times 10^{-7}$ Kgm^2
$\sum J_{washer} = (4)(½)(0.0006kg)(0.005m)^2 = 3.00 \times 10^{-8}$ Kgm^2
$\sum J_{shaft} = (2)(½)(0.0119kg)(0.0019m)^2 = 8.58 \times 10^{-8}$Kgm^2

$= 9.03 \times 10^{-6}$Kgm^2

Return Idler Inertia brought to Motor Shaft
$= (9.03 \times 10^{-6}$ Kgm$^2)(11.4/47.5)(19/57)(14/36)(4.8/2.32)$
$= 5.81 \times 10^{-7}$ Kgm^2

Total Inertia at Motor = 6.86×10^{-5}

Acceleration time = 0.2s
Acceleration = (1300 rpm)/(60s)/(0.2s) = 108 rev/s^2
Torque to Inertia =(108rev/s^2)(6.86×10^{-5}) = **0.0074Nm**

Required Motor Torque = 0.014Nm + 0.0074Nm = 0.0066 Nm at 1,300rpm

This size of motor fits into a gap in the existing motor selection. Since the rate of loading is the critical variable the next larger motor will be used. Motor #0012 will have no problems with this task. The use of Motor 12 provides a 45% over design to this actuator.

Loading Door Actuator Design

The loading door is designed with the motor connected with a belt drive to a worm gearing. The worm gear engages a 25-tooth pinion, which is directly coupled to the axis of the door. This arrangement will make the door self locking in the closed position.

Required torque = .006 N*m *(measured from drive shaft)*
Required drive revolutions to open = 8
Time scheduled for opening = 0.6 s *(from design time line to achieve >5s loading time)*
Required Drive RPM = 8rev/0.6s = 800RPM

Motor # 0005 appears to be able to meet these requirements.

Drive Pulley 11 mm
Motor pulley 8 mm

Designed Drive RPM = 800
Required Motor RPM = 800*(11/8) = **1,100**

Designed Drive Torque before inertia = 0.005 N*m
Required Motor Torque before inertia = 0.006(8/11)(100%/90%) = **0.0048N*m**

Actuator Inertia

Assume that 40% of the mass of the motor is the rotor and that the rotor is equivalent to a solid cylinder

$$\text{Motor inertia} = \tfrac{1}{2} mR^2 = \tfrac{1}{2}(0.083)(40\%)(0.008\text{m})^2 = 1.06\text{x}10^{-6}\ \text{Kgm}^2$$

$$\text{Stage 1 Inertia} = J_{pulley} + J_{boss} + J_{boss} + J_{washer} + J_{worm} + J_{boss} + J_{boss} + J_{washer} + J_{shaft}$$

$$J_{pulley} = (\tfrac{1}{2})(0.0060\text{kg})(0.0120\text{m})^2 = 4.32 \text{ x } 10^{-7}\text{Kgm}^2$$

$$\sum J_{boss} = (4)(\tfrac{1}{2})(0.0028\text{kg})(0.005\text{m})^2 = 1.40 \text{ x } 10^{-7}\ \text{Kgm}^2$$

$$\sum J_{washer} = (2)(\tfrac{1}{2})(0.0006\text{kg})(0.005\text{m})^2 = 1.50 \text{ x } 10^{-8}\ \text{Kgm}^2$$

$$J_{worm} = (\tfrac{1}{2})(0.0134\text{kg})(0.0065\text{m})^2 = 2.83 \text{ x } 10^{-7}\text{Kgm}^2$$

$$J_{shaft} = (\tfrac{1}{2})(0.0053\text{kg})(0.0019\text{m})^2 = 9.57 \text{ x } 10^{-9}\text{Kgm}^2$$

$$= 8.78\text{x}10^{-7}\ \text{Kgm}^2$$

$$\text{Stage 1 Inertia brought to Motor Shaft} = (8.78\text{x}10^{-7}\ \text{Kgm}^2)(0.8/2.32) = 3.03\text{x}10^{-8}\ \text{Kgm}^2$$

Stage 2 Inertia = $J_{pinion} + J_{boss} + J_{boss} + J_{washer} + J_{boss} + J_{washer} + J_{shaft1}\ J_{universal} + J_{shaft2} + J_{universal} + J_{shaft3} + J_{door}$

$J_{pinion} = (½)(0.0105kg)(0.0088m)^2 = 4.07 \times 10^{-7} Kgm^2$

$\sum J_{boss} = (3)(½)(0.0028kg)(0.005m)^2 = 1.05 \times 10^{-7} Kgm^2$

$\sum J_{washer} = (2)(½)(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8} Kgm^2$

$J_{shaft1} = (½)(0.0042kg)(0.0019m)^2 = 7.6 \times 10^{-9} Kgm^2$

$J_{shaft2} = (½)(0.0015kg)(0.0019m)^2 = 2.71 \times 10^{-9} Kgm^2$

$J_{shaft3} = (½)(0.0025kg)(0.0019m)^2 = 4.51 \times 10^{-9} Kgm^2$

$\sum J_{universal} = 2(½)(0.0139kg)(0.006m)^2 = 5.00 \times 10^{-7} Kgm^2$

$J_{door} = (0.0102kg)(0.019m)^2 = 3.68 \times 10^{-6} Kgm^2$

$= 4.72 \times 10^{-6}\ Kgm^2$

Stage 2 Inertia brought to Motor Shaft = $(4.72 \times 10^{-6}\ Kgm^2)(0.8/2.32)(1/25)$
$= 6.52 \times 10^{-8}\ Kgm^2$

Total Inertia at motor = $1.15 \times 10^{-6}\ Kgm^2$

Acceleration = (1100rpm)/(60s)/(0.1s) = 183 rev/s^2
Total torque = $(1.15 \times 10^{-6}\ Kgm^2)(183\ rev/s^2)+(0.0048Nm)$

The appropriate motor must be able to exceed 1100 RPM with a torque of 0.0050 N*m.

From the Torque curve for Motor #0005 at 1,100 RPM it will produce a torque of 0.0066N*m. This is 32% more than the minimum required torque. This motor will be able to perform the required task with some additional capacity to overcome some binding anticipated under working conditions.

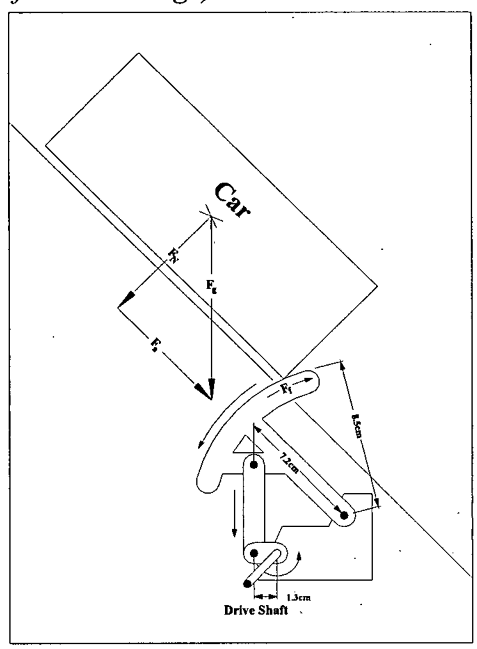
Car Stopper Actuator Design

The car stopper is designed with a third class lever operating a curved protrusion. The protrusion is raised to stop cars then lowered to allow a car to proceed.

Required torque
$T_{tot} = T_F + T_{FC}$

T_F is the torque to overcome the friction built into the system
T_{FC} is the torque to overcome the friction of the car on the stopper

T_F = 0.034N*m *(measured from the drive shaft for the linkage)*
T_{FC} can be found using the geometry of the system:



$F_g = 1.2Kg*9.80m/s^2 = 11.8N$
$F_a = 11.8N *\sin(45^o)$ (track is on a 45o <)
= 8.3N
$F_f = \tan(20^o)(8.3N)$ (assume a 20o< of fric)
= 3.0N
F_l = force applied in link
= $(3.0N)/(\sin(45^o))*(8.5cm)/(7.2cm)$
= 5.1N
T_{FC} = (5.1N)(1.3cm) = **0.066N**
Time scheduled for opening = 0.6 s
(from design time line to achieve >5s loading time)
Designed drive revolutions = 0.5
Designed Drive RPM: 0.5rev/0.6s = **50RPM**

The appropriate drive must be able to exceed 50 RPM with a torque of 0.066 N*m before inertia loading.

Motor # 0004 appears to be able to meet these requirements after a two stage speed reduction.

Drive chain pinion 36 teeth
Stage 1 driver pinion 14 teeth

Required stage 1 RPM = (50RPM)*(36teeth/14teeth) = 129 RPM
Required stage 1 torque = (0.066N)/(14teeth/36teeth)/(90%eff) = 0.0285N*m

Stage 1 driven pulley 47.5 mm
Motor 1 driver pulley 5.6 mm

Required Motor RPM = 129*(47.5/5.6) = **1,094 RPM**
Required Motor Torque before inertia loading = 0.0285(5.6/47.5)/(90%eff) = **0.0037N*m**

Actuator Inertia Calculations

Motor inertia = $\frac{1}{2}\,mR^2$
= $\frac{1}{2}\,(0.076)(40\%)(0.008m)^2$
$=9.73x10^{-7}$

Stage 1 Inertia = $J_{pulley} + J_{boss} + J_{boss} + J_{washer} + J_{pinion} + J_{boss} + J_{boss} + J_{washer} + J_{shaft} + J_{chain}$
$J_{pulley} = (\frac{1}{2})(0.0152kg)(0.0238m)^2 = 4.30 \times 10^{-6} Kgm^2$
$\sum J_{boss} = (4)(\frac{1}{2})(0.0028kg)(0.005m)^2 = 1.40 \times 10^{-7} Kgm^2$
$\sum J_{washer} = (2)(\frac{1}{2})(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8} Kgm^2$
$J_{pinion} = (\frac{1}{2})(0.0036kg)(0.0098m)^2 = 1.73 \times 10^{-7} Kgm^2$
$J_{shaft} = (\frac{1}{2})(0.0053kg)(0.0019m)^2 = 9.57 \times 10^{-9} Kgm^2$
$J_{chain} = (0.0107kg)(0.0098m)^2 = 1.03 \times 10^{-6} Kgm^2$

$= 5.66x10^{-6}\ Kgm^2$
Stage 1 Inertia brought to Motor Shaft = $(5.66x10^{-6}\ Kgm^2)(5.6/47.5)$
$= 6.68x10^{-7}\ Kgm^2$

Stage 2 Inertia = $J_{pinion} + J_{boss} + J_{boss} + J_{washer} + J_{shaft1} + J_{boss} + J_{washer} + J_{contact} + J_{boss} + 2J_{bush} + 2J_{boss} + J_{link} + J_{shaft2} + 2J_{boss} + 2J_{washer}$
$J_{pinion} = (\frac{1}{2})(0.0618kg)(0.0765m)^2 = 1.81 \times 10^{-4} Kgm^2$
$\sum J_{boss} = (6)(\frac{1}{2})(0.0028kg)(0.005m)^2 = 2.10 \times 10^{-7} Kgm^2$
$\sum J_{washer} = (2)(\frac{1}{2})(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8} Kgm^2$
$J_{contact} = (0.0028kg)(0.011m)^2 = 3.39 \times 10^{-7} Kgm^2$
$J_{shaft1} = (\frac{1}{2})(0.0119kg)(0.0019m)^2 = 2.15 \times 10^{-8} Kgm^2$
$\sum J_{bush} = (2)(\frac{1}{2})(0.0089kg)(0.0174m)^2 = 2.69 \times 10^{-6} Kgm^2$
$\sum J_{boss} = (2)(0.0028kg)(0.0128m)^2 = 9.18 \times 10^{-6} Kgm^2$
$\sum J_{washer} = (2)(0.0006kg)(0.0128m)^2 = 1.97 \times 10^{-7} Kgm^2$
$J_{shaft2} = (0.0029kg)(0.0128m)^2 = 4.75 \times 10^{-7} Kgm^2$
$J_{link} = (0.0229kg)(0.0128m)^2 = 3.75 \times 10^{-6} Kgm^2$

$= 1.89x10^{-4}\ Kgm^2$
Stage 2 Inertia brought to Motor Shaft = $(1.89x10^{-4}\ Kgm^2)(5.6/47.5)\ (14/36)$
$= 8.69x10^{-6}\ Kgm^2$

Stage 3 Inertia = $J_{arm} + J_{protrusion}$
$J_{arm} = (0.0224kg)(0.0245m)^2 = 1.34 \times 10^{-5} Kgm^2$
$J_{protrusion} = (0.0261kg)(0.0490m)^2 = 6.26 \times 10^{-5} Kgm^2$
$= 7.61x10^{-5}\ Kgm^2$

Stage 3 Inertia brought to Motor Shaft = $(7.61x10^{-5}\ Kgm^2)(5.6/47.5)\ (14/36)\ (8.5/7.2)$
$= 4.12x10^{-6}\ Kgm^2$

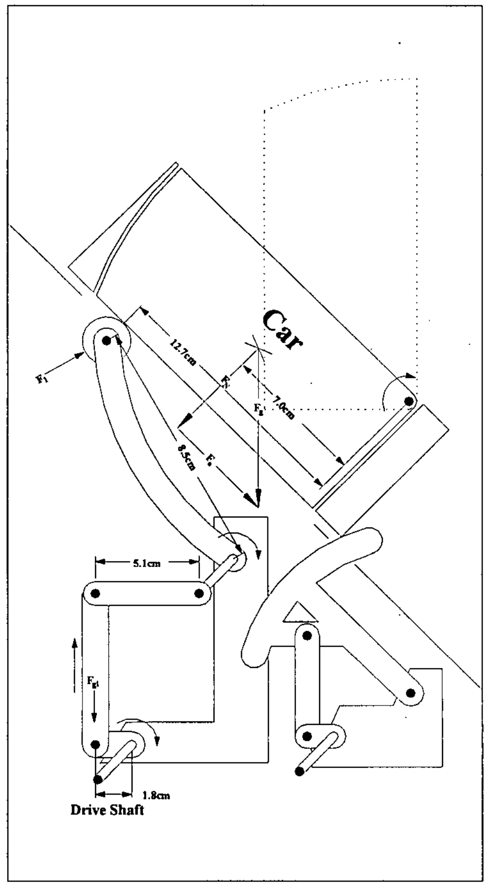
Total inertia at motor = $1.44x10^{-5}\ Kgm^2$
Acceleration = (1100 rpm)/(60s)/(0.15s) = $122 rev/s^2$

Total torque required = $(1.44x10^{-5}\ Kgm^2)(122 rev/s^2)+(0.0037Nm) = 0.005Nm$

From the Torque curve for Motor #0004 at 1,100 RPM it will produce a torque of 0.005N*m. This will just achieve the minimum required torque. This motor has the potential to be a little slow at performing the required task.

Car Opener Actuator Design

The opener is a simple third class lever. The actuator is designed to stop in TDC to lock the car open while loading. The highest torque requirements will be to open the car. The calculations for this actuator are shown below:



$T_{tot} = T_F + T_{gl} + T_c$

T_F is the torque to overcome the friction built into the system

T_{gl} is the torque to overcome the force of gravity on the linkage

T_c is the torque required to open the car

$T_F = (0.134N*m)*(1.8cm/5.1cm) = 0.0473N*m$ *(measured from the shaft of the primary linkage transferred to the drive shaft)*

$T_{gl} = (F_{gl})(1.8cm)$
$= (0.68N)\ (1.8cm)$
$= 0.0122\ N*m$

T_c

$F_N = (0.2kg)*(9.80m/s^2)*\sin(45^o) = 1.38N$

$F_{Nl} = F_N$ transposed to the location of F_l
$= (1.38N)*(7.0cm/12.7cm) = 0.764N$

$F_l = (0.764N)/(\cos(15^o)) = 0.791N$

$T_c=(0.791N)*(12.7cm)(1.8cm/5.1cm)= 0.0354N*m$

$T_{tot} = 0.0437N*m + 0.0122N*m + 0.0354N*m$
= .0913N*m

Time scheduled for opening = 0.6 s *(from design time line to achieve >5s loading time)*
Designed drive revolutions = 0.5
Designed Drive RPM = 0.5rev/0.6s = **50RPM**

The appropriate drive must be able to exceed 50 RPM with a torque of 0.091 N*m before inertia loading.

Motor # 0002 appears to be able to meet these requirements after a two stage speed reduction.

Out put driven pinion 57 teeth
Stage 1 driver pinion 19 teeth

Required stage 1 RPM = (50RPM)*(57teeth/19teeth) = 150 RPM
Required stage 1 torque = (0.091N)/(19teeth/57teeth)/(90%eff) = 0.0338N*m

Stage 1 driven pulley 35.8 mm
Motor 1 driver pulley 6.0 mm

Required Motor RPM = 150*(35.8/6.0) = **895 RPM**
Required Motor Torque before inertia loading = 0.0338(6.0/35.8)/(90%eff) = **0.0063N*m**

Motor #0002 apears to be capable of operating this actuator.

Inertia Loading

Motor inertia = $\frac{1}{2} mR^2$
$= \frac{1}{2} (0.087)(40\%)(0.008m)^2$
$=1.28 \times 10^{-6}$

Stage 1 Inertia = $J_{pulley} + J_{boss} + J_{boss} + J_{washer} + J_{pinion} + J_{boss} + J_{boss} + J_{washer} + J_{shaft}$
$J_{pulley} = (\frac{1}{2})(0.0110kg)(0.0179m)^2 = 1.76 \times 10^{-6} Kgm^2$
$\sum J_{boss} = (4)(\frac{1}{2})(0.0028kg)(0.005m)^2 = 1.40 \times 10^{-7} Kgm^2$
$\sum J_{washer} = (2)(\frac{1}{2})(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8} Kgm^2$
$J_{pinion} = (\frac{1}{2})(0.0036kg)(0.0069m)^2 = 8.56 \times 10^{-8} Kgm^2$
$J_{shaft} = (\frac{1}{2})(0.0089kg)(0.0019m)^2 = 1.61 \times 10^{-8} Kgm^2$

$= 2.02 \times 10^{-6} Kgm^2$
Stage 1 Inertia brought to Motor Shaft = $(2.02 \times 10^{-6} Kgm^2)(6/35.8)$
$= 3.38 \times 10^{-7} Kgm^2$
Stage 2 Inertia = $J_{pinion} + J_{boss} + J_{boss} + J_{washer} + J_{crank} + J_{boss} + J_{washer} + J_{link}$
$J_{pinion} = (\frac{1}{2})(0.0135kg)(0.0195m)^2 = 2.57 \times 10^{-6} Kgm^2$
$\sum J_{boss} = (4)(\frac{1}{2})(0.0028kg)(0.005m)^2 = 1.40 \times 10^{-7} Kgm^2$
$\sum J_{washer} = (2)(\frac{1}{2})(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8} Kgm^2$
$J_{crank} = (\frac{1}{2})(0.0069kg)(0.0019m)^2+(0.0024kg)(0.0228m)^2 == 1.26 \times 10^{-6} Kgm^2$
$J_{link} = (0.0694kg)(0.0228m)^2 = 3.61 \times 10^{-5} Kgm^2$

$= 4.01 \times 10^{-5} Kgm^2$
Stage 2 Inertia brought to Motor Shaft = $(4.01 \times 10^{-5} Kgm^2)(6/35.8)(19/57)$
$= 2.23 \times 10^{-6} Kgm^2$
Stage 3 Inertia = $J_{link} + 3J_{bush} + 3J_{boss} + J_{washer} + J_{shaft} + 2J_{boss} + J_{washer} + J_{link} + J_{roller}$
$J_{link} = (0.0133kg)(0.024m)^2 = 7.66 \times 10^{-6} Kgm^2$
$\sum J_{bush} = (3)(\frac{1}{2})(0.0089kg)(0.0175m)^2 = 4.09 \times 10^{-6} Kgm^2$
$\sum J_{boss} = (5)(\frac{1}{2})(0.0028kg)(0.005m)^2 = 1.75 \times 10^{-7} Kgm^2$
$\sum J_{washer} = (2)(\frac{1}{2})(0.0006kg)(0.005m)^2 = 1.50 \times 10^{-8} Kgm^2$
$J_{shaft} = (\frac{1}{2})(0.0119kg)(0.0019m)^2 = 2.15 \times 10^{-8} Kgm^2$
$J_{link} = (0.0174kg)(0.0215m)^2 = 8.04 \times 10^{-6} Kgm^2$
$J_{roller} = (0.0144kg)(0.0432m)^2 = 2.68 \times 10^{-5} Kgm^2$

$= 4.68 \times 10^{-5} Kgm^2$
Stage 3 Inertia brought to Motor Shaft=$(4.68 \times 10^{-5} Kgm^2)(6/35.8)(19/57)(1.8/5.1)$
$= 9.24 \times 10^{-7} Kgm^2$

The total inertia = $4.77 \times 10^{-6} Kgm^2$

Required acceleration = (900 rpm)/(60s)/(0.1s)
= 150 rev/s^2
Required Torque = $(4.77\times10^{-6}\ Kgm^2)(150\ rev/s^2)+(0.0063Nm)$ = **0.0070Nm**

From the Torque curve for Motor #0002 at 900 RPM it will produce a torque of 0.013N*m. This is 86% more than the minimum required torque. This motor is over designed for this application, the next smaller motor is border line so the over design will be tolerated.

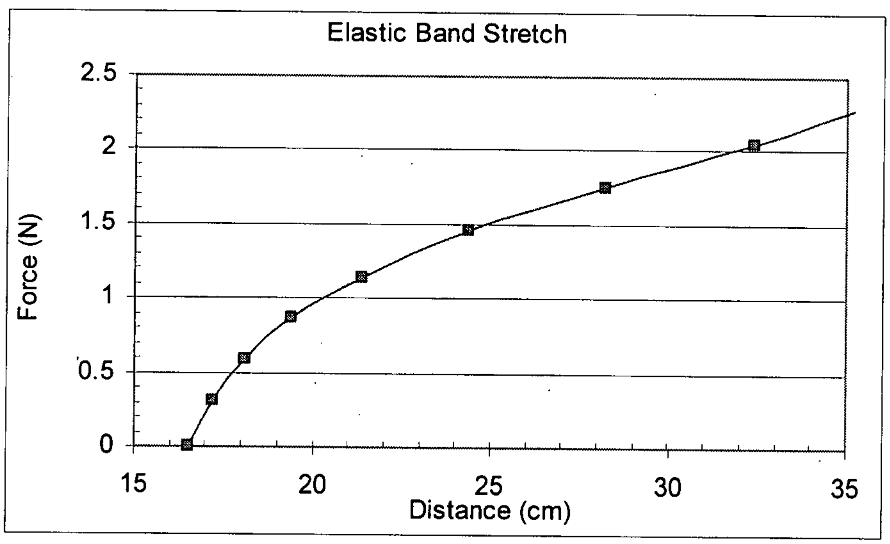
Appendix III: Loading Station, Motor Torque Curves

Motor Torque Curve Generation

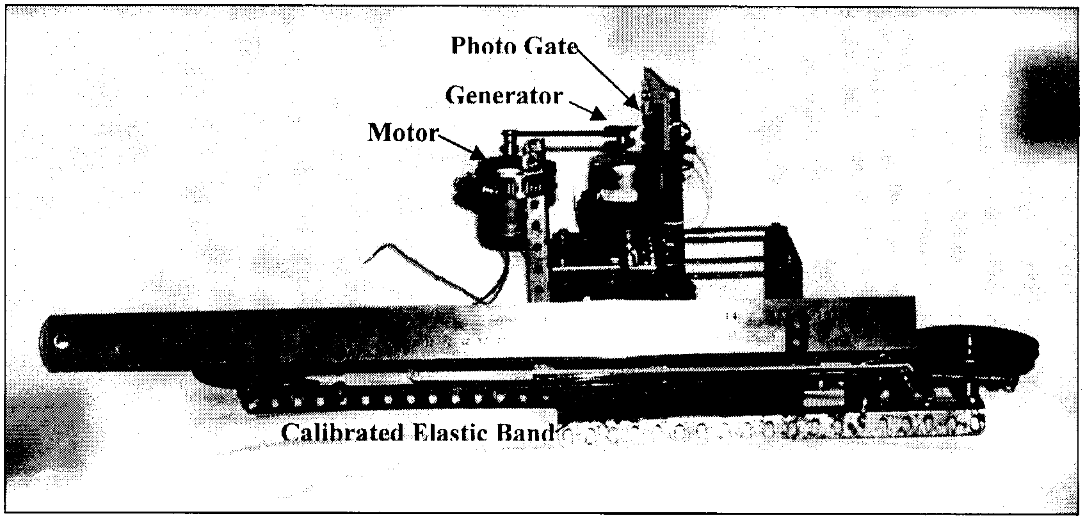
To keep with in the economic constrain for the model motors had to be chosen from a selection of existing motors. Unfortunately none of the motors have any information on them, particularly information on torque curves. To proceed with the design of the actuators the motor torque curves had to be experimentally determined.

Two mechanisms were developed to measure the generated torques and speed for the motors depending on their power.

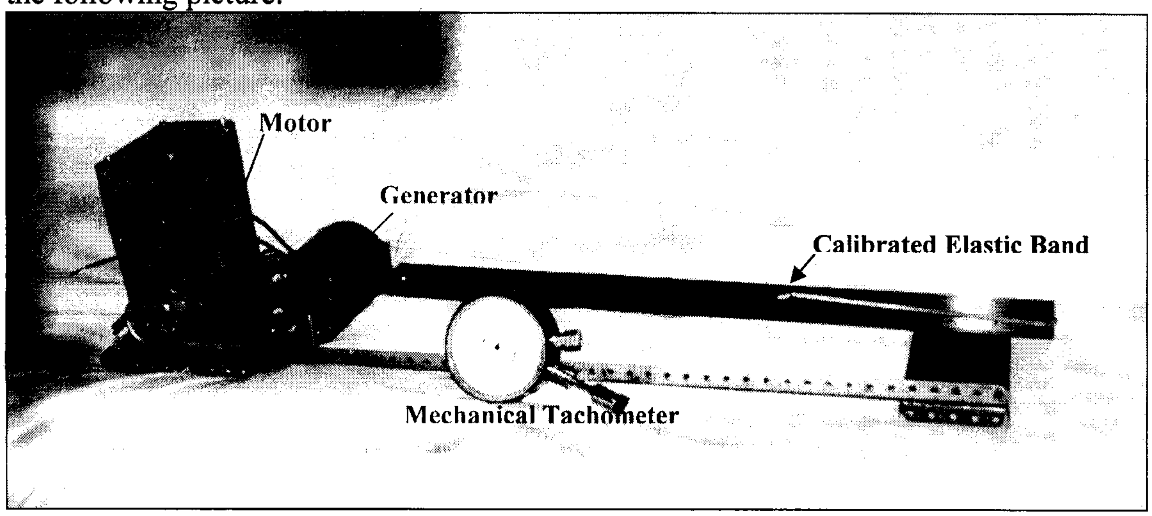
Both mechanisms consisted of the test motor driving a generator. The amount of torque could be controlled by controlling the amount of current flowing through the generator. The torque generated was then measured by measuring the force at a controlled radius from the mounting axis of the generator. The force was measured using a calibrated elastic band. As shown in the graph below:



The mechanism for the smaller motors had the generator mounted like a top with the torque being measured off of the axis of the top. The Speed was recorded using a photo gate operating a tachometer off of the generator. This mechanism looked like below:



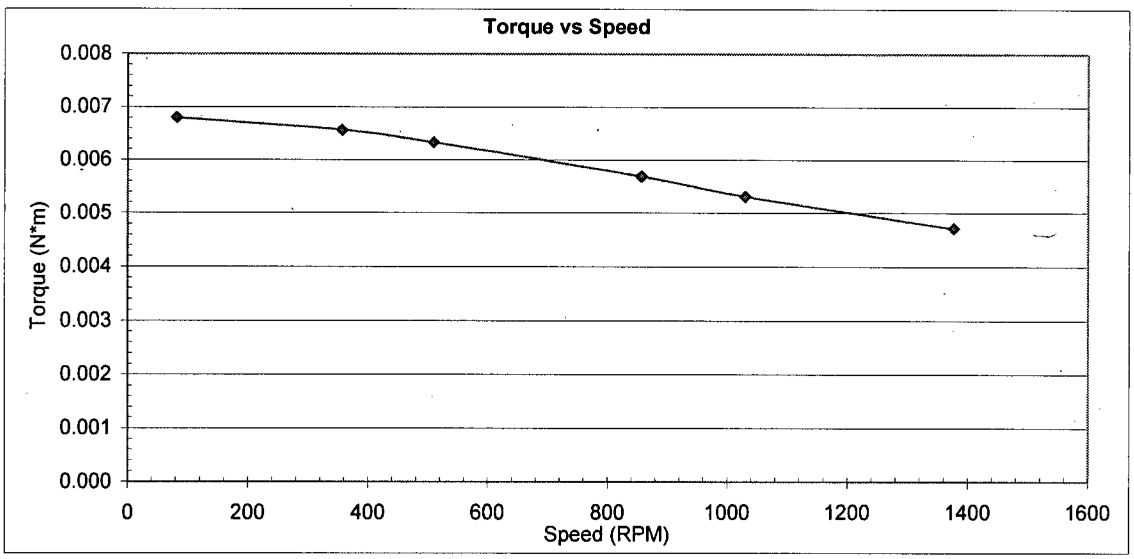
The Mechanism for measuring the torque for the larger motors was similar in principle to the first one but the torque was being measured directly from the generator housing. The speed of the motor was being directly read with a mechanical rpm gauge. This mechanism is shown in the following picture.



The generated motor torque curves are shown on the following pages.

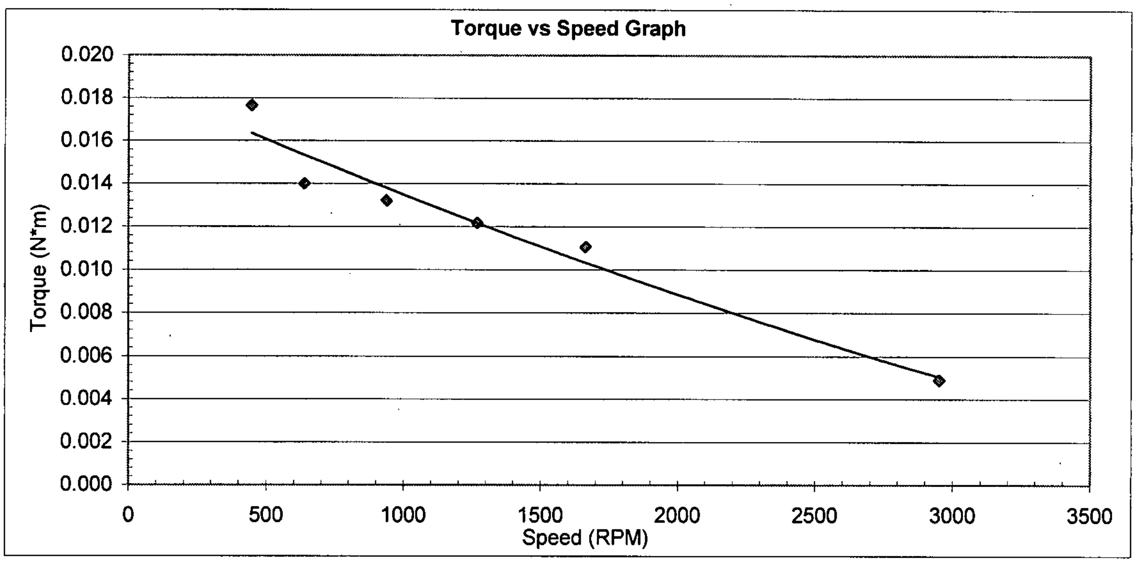
Motor #	0001
Voltage	12 V DC
Maximum Current	0.8 A
Diameter	3.26 cm
Lenfth	2.47 cm
Shaft Diameter	0.19 cm

Test Data	Recorded Data				Generator			Motor			
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.2			5.1			
2	19.4	10	1350	0.87	5.2	1350	0.004	5.1	90%	1376	0.005
3	20.2	10	1010	0.98	5.2	1010	0.005	5.1	90%	1030	0.005
4	20.8	10	840	1.04	5.2	840	0.005	5.1	90%	856	0.006
5	21.7	10	500	1.16	5.2	500	0.006	5.1	90%	510	0.006
6	22.1	10	350	1.20	5.2	350	0.006	5.1	90%	357	0.007
7	22.5	10	80	1.25	5.2	80	0.006	5.1	90%	82	0.007
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



Motor #	0002
Voltage	12 V DC
Maximum Current	1.1 A
Diameter	3.28 cm
Lenfth	2.29 cm
Shaft Diameter	0.19 cm

Test Data	Recorded Data				Generator			Motor			
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			8			
2	18	10	4630	0.56	5.1	4630	0.003	8	90%	2952	0.005
3	22.7	10	2610	1.27	5.1	2610	0.006	8	90%	1664	0.011
4	23.9	10	1990	1.40	5.1	1990	0.007	8	90%	1269	0.012
5	25.2	10	1470	1.52	5.1	1470	0.008	8	90%	937	0.013
6	26.4	10	1000	1.61	5.1	1000	0.008	8	90%	638	0.014
7	32.3	10	700	2.03	5.1	700	0.010	8	90%	446	0.018
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



Motor #	0003
Voltage	12 V DC
Maximum Current	0.5 A
Diameter	3.18 cm
Lenfth	2.68 cm
Shaft Diameter	0.19 cm

Test Data	Recorded Data				Generator				Motor		
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			8			
2	17	10	2000	0.22	5.1	2000	0.001	8	90%	1275	0.002
3	17.7	10	800	0.47	5.1	800	0.002	8	90%	510	0.004
4	19.3	10	50	0.84	5.1	50	0.004	8	90%	32	0.007
5		10		0.00							
6		10		0.00							
7		10		0.00							
8				0.00							
9				0.00							
10				0.00							
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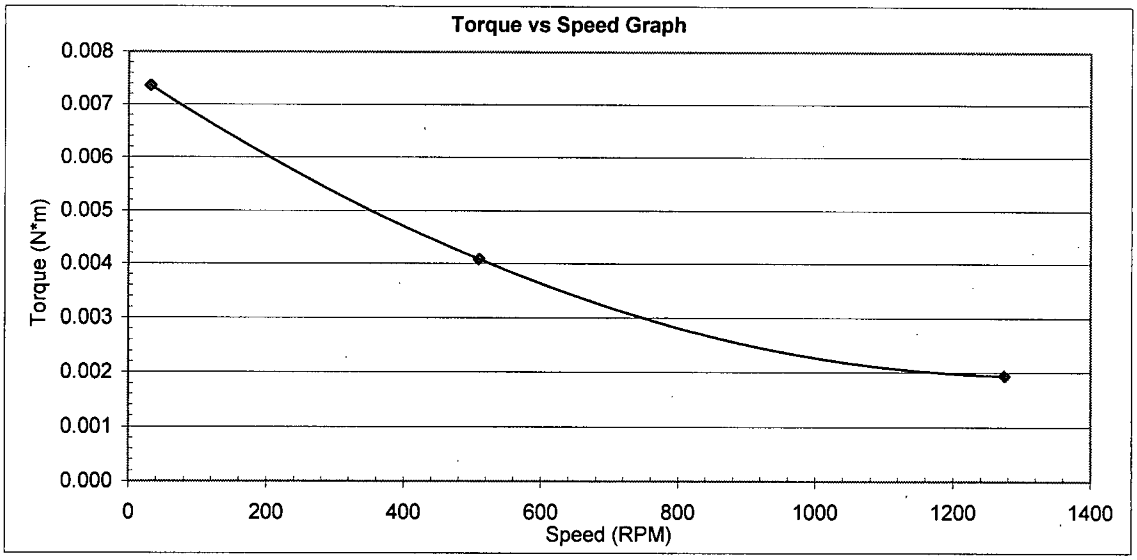
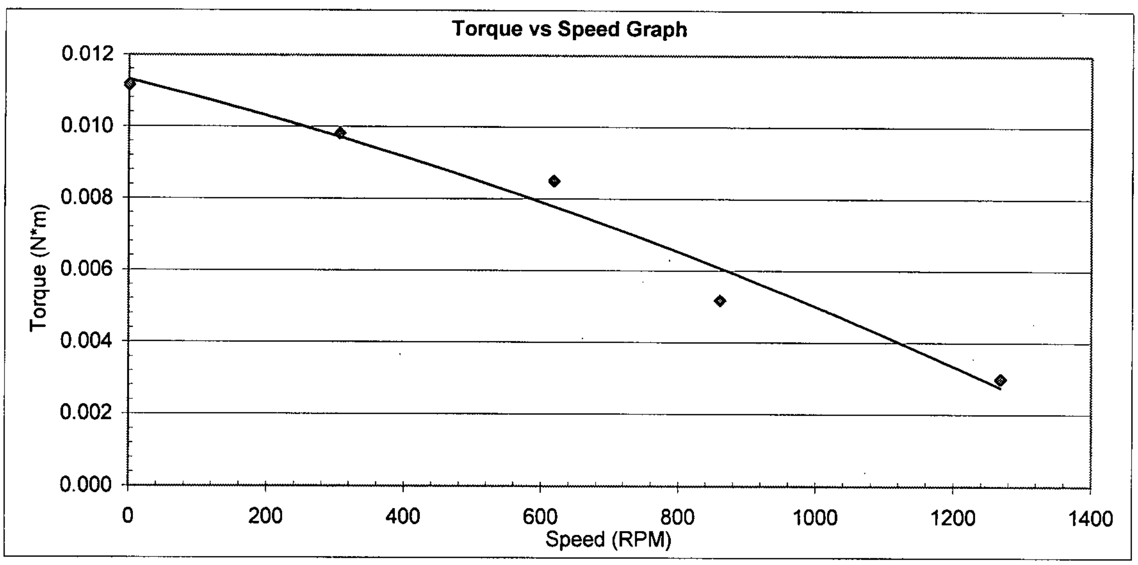


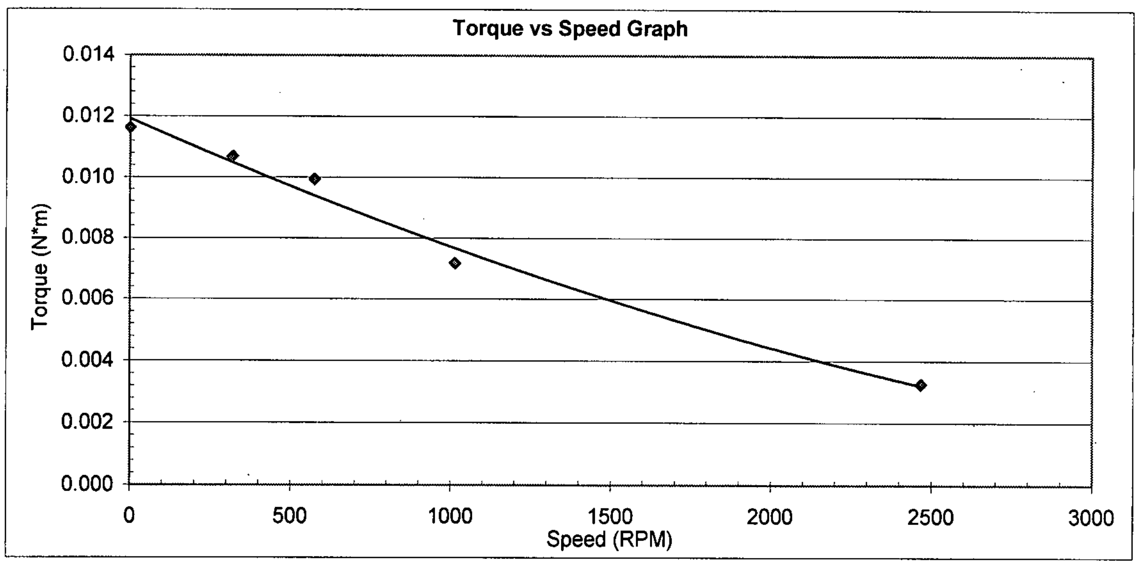
Motor #	0004
Voltage	12 V DC
Maximum Current	0.4 A
Diameter	3.39 cm
Lenfth	3 cm
Shaft Diameter	0.19 cm

Test Data	Recorded Data				Generator				Motor		
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			8			
2	17.3	10	1990	0.34	5.1	1990	0.002	8	90%	1269	0.003
3	18.1	10	1350	0.59	5.1	1350	0.003	8	90%	861	0.005
4	20.2	10	970	0.98	5.1	970	0.005	8	90%	618	0.008
5	21.4	10	480	1.13	5.1	480	0.006	8	90%	306	0.010
6	22.8	10	0	1.28	5.1	0	0.006	8	90%	0	0.011
7				0.00							
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



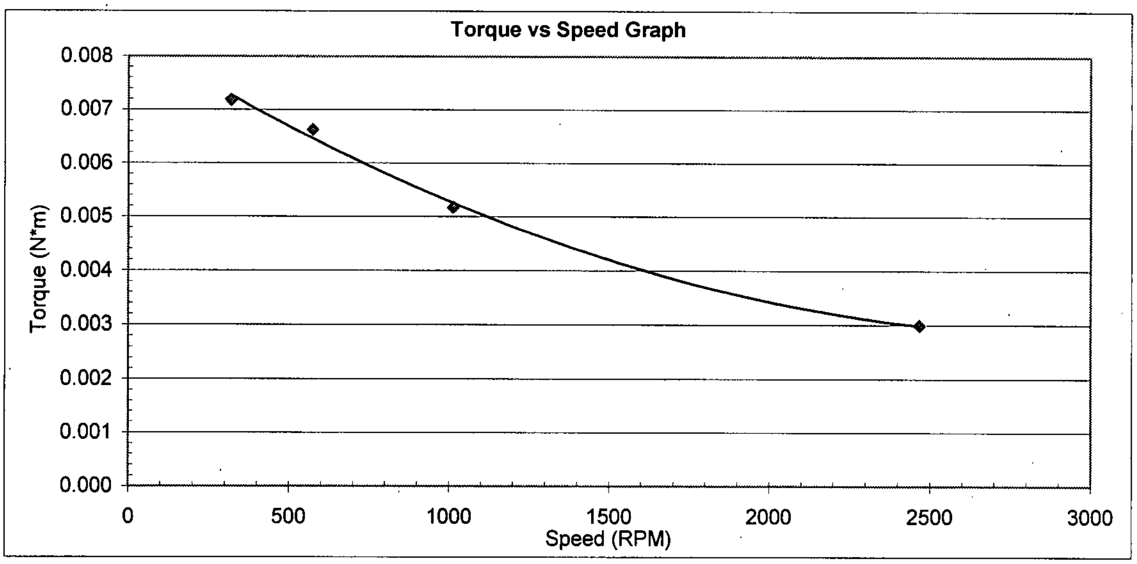
Motor #	0005
Voltage	12 V DC
Maximum Current	0.6 A
Diameter	3.27 cm
Lenfth	2.46 cm
Shaft Diameter	0.19 cm

Test Data	Recorded Data				Generator				Motor		
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			8			
2	17.4	10	3870	0.37	5.1	3870	0.002	8	90%	2467	0.003
3	19.2	10	1590	0.82	5.1	1590	0.004	8	90%	1014	0.007
4	21.5	10	900	1.14	5.1	900	0.006	8	90%	574	0.010
5	22.3	10	500	1.23	5.1	500	0.006	8	90%	319	0.011
6	23.3	10	0	1.33	5.1	0	0.007	8	90%	0	0.012
7				0.00							
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



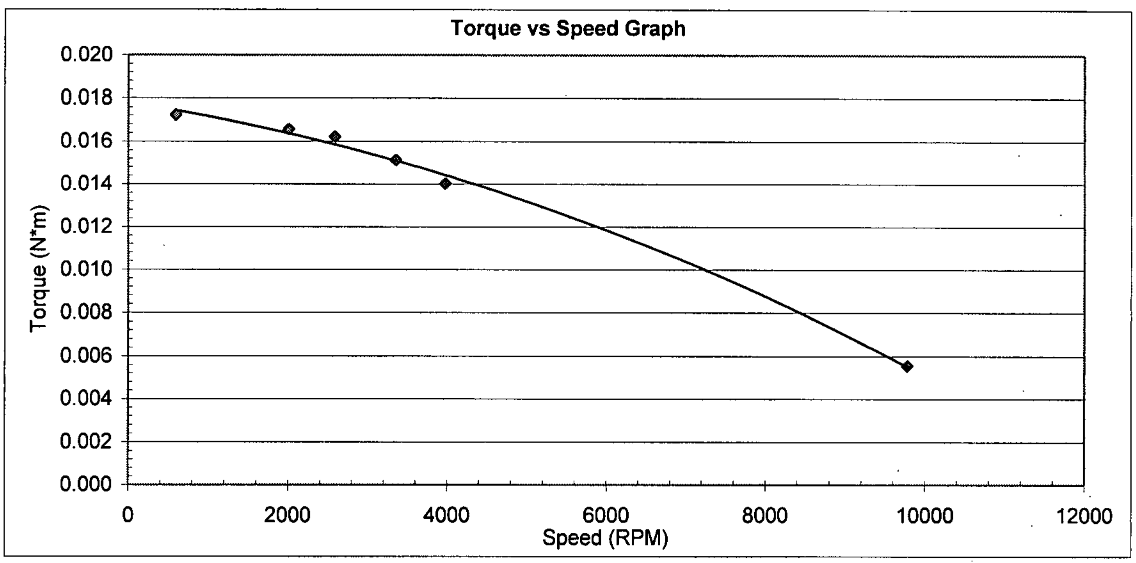
Motor #	0006
Voltage	12 V DC
Maximum Current	0.5 A
Diameter	3.4 cm
Lenfth	3.3 cm
Shaft Diameter	0.19 cm

Test Data	Recorded Data				Generator				Motor		
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			8			
2	17.3	10	3870	0.34	5.1	3870	0.002	8	90%	2467	0.003
3	18.1	10	1590	0.59	5.1	1590	0.003	8	90%	1014	0.005
4	18.9	10	900	0.76	5.1	900	0.004	8	90%	574	0.007
5	19.2	10	500	0.82	5.1	500	0.004	8	90%	319	0.007
6				0.00							
7				0.00							
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



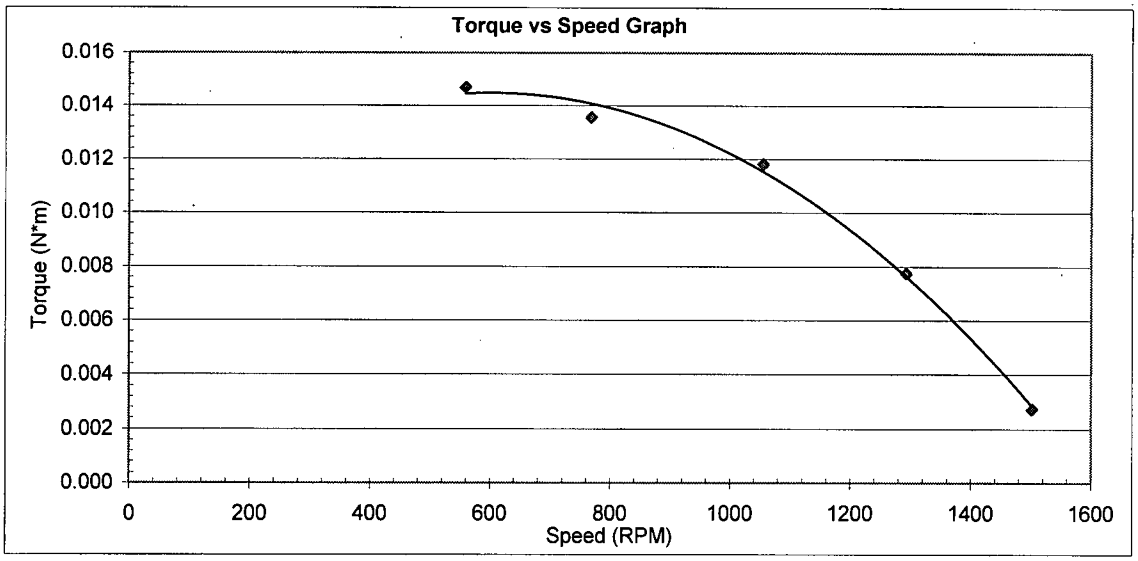
Motor #	0007
Voltage	12 V DC
Maximum Current	3.5 A
Diameter	2.83 cm
Lenfth	3.22 cm
Shaft Diameter	0.3 cm

Test Data	Recorded Data				Generator			Motor			
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			7.3			
2	18.6	10	14000	0.70	5.1	14000	0.003	7.3	90%	9781	0.006
3	28.5	10	5690	1.76	5.1	5690	0.009	7.3	90%	3975	0.014
4	30.5	10	4800	1.90	5.1	4800	0.010	7.3	90%	3353	0.015
5	32.5	10	3700	2.04	5.1	3700	0.010	7.3	90%	2585	0.016
6	33	10	2870	2.08	5.1	2870	0.010	7.3	90%	2005	0.017
7	34	10	850	2.17	5.1	850	0.011	7.3	90%	594	0.017
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



Motor #	0008
Voltage	12 V DC
Maximum Current	1.1 A
Diameter	4.03 cm
Lenfth	4.28 cm
Shaft Diameter	0.23 cm

Test Data	Recorded Data				Generator				Motor		
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			7.3			
2	17.3	10	2150	0.34	5.1	2150	0.002	7.3	90%	1502	0.003
3	20.3	10	1850	0.98	5.1	1850	0.005	7.3	90%	1292	0.008
4	24.8	10	1510	1.49	5.1	1510	0.007	7.3	90%	1055	0.012
5	27.7	10	1100	1.71	5.1	1100	0.009	7.3	90%	768	0.014
6	29.7	10	800	1.85	5.1	800	0.009	7.3	90%	559	0.015
7		10		0.00							
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							



Motor #	0009
Voltage	12 V DC
Maximum Current	1.1 A
Diameter	4.03 cm
Lenfth	4.28 cm
Shaft Diameter	0.23 cm

Test Data	Recorded Data				Generator			Motor			
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			7.3			
2	17.3	10	2150	0.34	5.1	2150	0.002	7.3	90%	1502	0.003
3	20.3	10	1850	0.98	5.1	1850	0.005	7.3	90%	1292	0.008
4	24.8	10	1510	1.49	5.1	1510	0.007	7.3	90%	1055	0.012
5	27.7	10	1100	1.71	5.1	1100	0.009	7.3	90%	768	0.014
6	29.7	10	800	1.85	5.1	800	0.009	7.3	90%	559	0.015
7		10		0.00							
8				0.00							
9				0.00							
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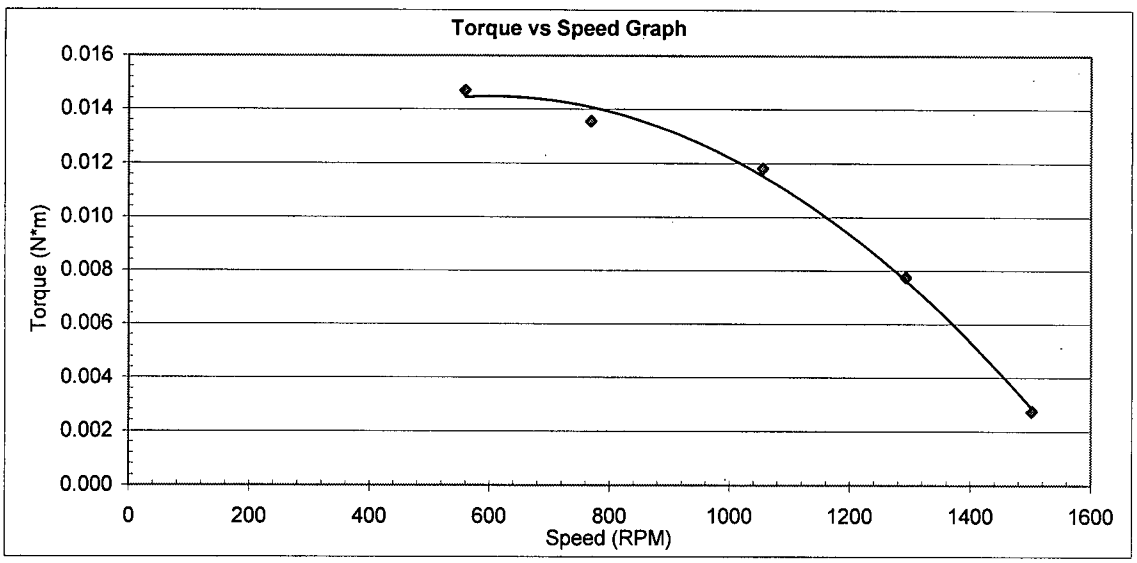

Motor #	0010
Voltage	12 V DC
Maximum Current	1.1 A
Diameter	4.03 cm
Lenfth	4.28 cm
Shaft Diameter	0.23 cm

Test Data	Recorded Data				Generator			Motor			
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			5.1			
2		10	5450	0.00	5.1	5450	0.000	5.1	90%	5450	0.000
3	17	10	2500	0.22	5.1	2500	0.001	5.1	90%	2500	0.001
4	22.5	10	0	1.25	5.1	0	0.006	5.1	90%	0	0.007
5				0.00							
6				0.00							
7				0.00							
8				0.00							
9				0.00							
10				0.00							
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13				0.00							
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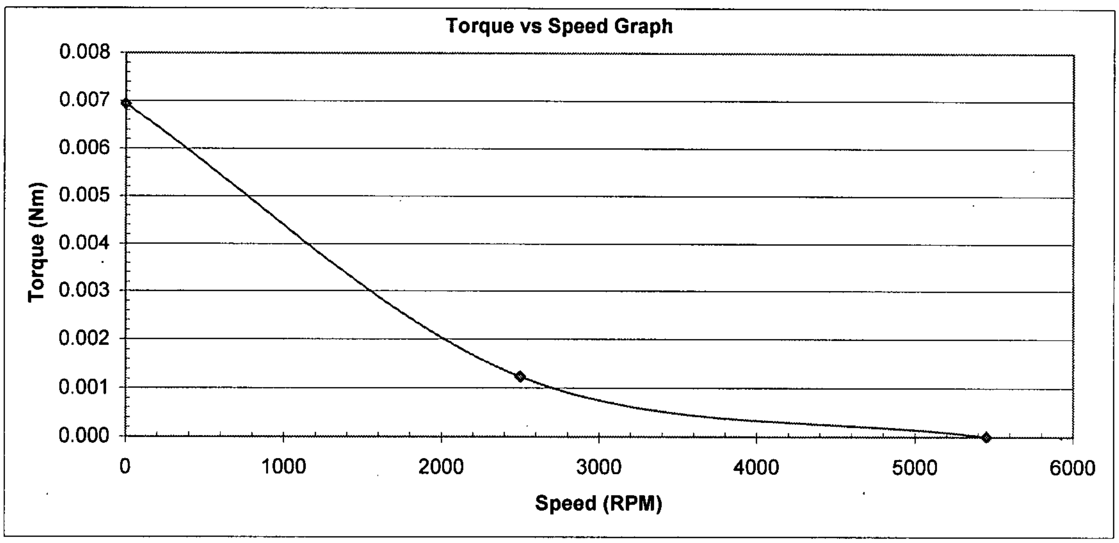
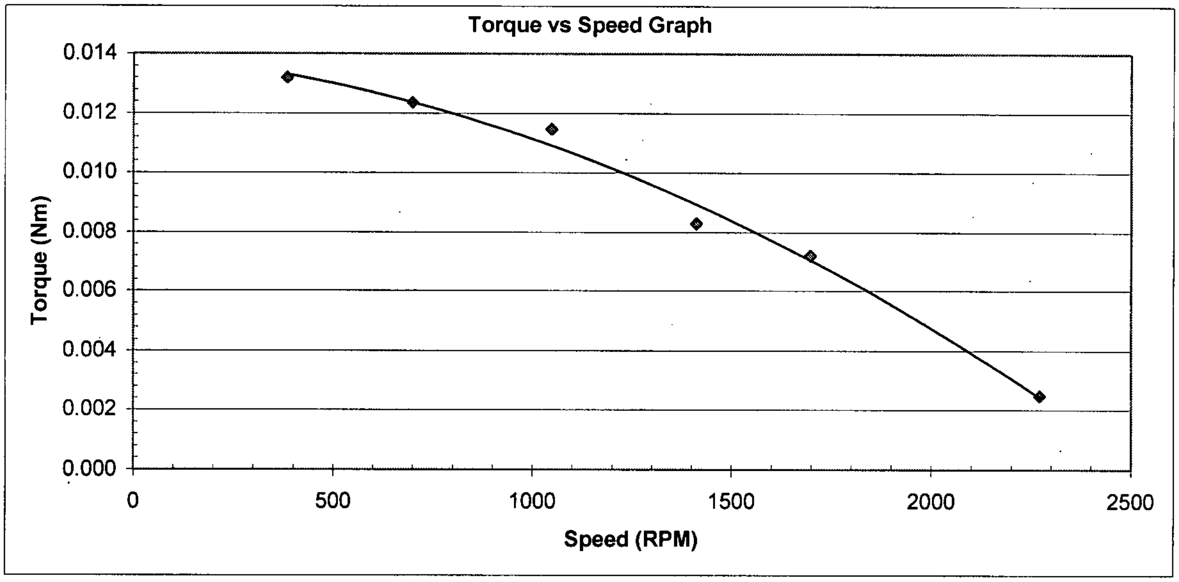
15				0.00							



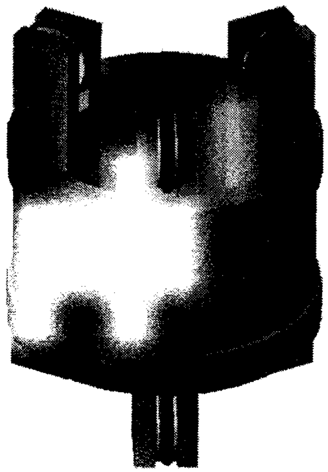
Motor #	0011
Voltage	12 V DC
Maximum Current	0.9 A
Diameter	4.03 cm
Lenfth	4.28 cm
Shaft Diameter	0.23 cm

Test Data	Recorded Data				Generator			Motor			
Trial	Distance (cm)	Pulley Size mm	RPM	Force N	Pulley Size mm	RPM	Torque (N*m)	Pulley Size mm	Drive Eff. 90%	Rpm	Torque (N*m)
1		10		0.00	5.1			7.3			
2	17.2	10	3250	0.31	5.1	3250	0.002	7.3	90%	2271	0.002
3	19.8	10	2430	0.91	5.1	2430	0.005	7.3	90%	1698	0.007
4	20.8	10	2020	1.04	5.1	2020	0.005	7.3	90%	1411	0.008
5	24.3	10	1500	1.44	5.1	1500	0.007	7.3	90%	1048	0.011
6	25.7	10	1000	1.55	5.1	1000	0.008	7.3	90%	699	0.012
7	27.1	10	550	1.66	5.1	550	0.008	7.3	90%	384	0.013
8				0.00							
9				0.00							
10				0.00							
11				0.00							
12				0.00							
13				0.00							
14				0.00							
15				0.00							

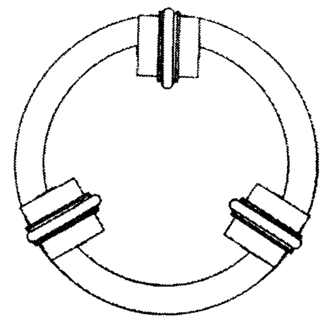


Appendix IV: Skip Design

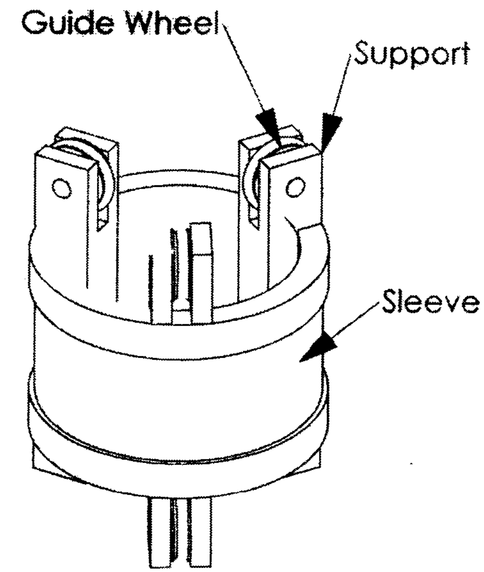
ZONE	REV.	REVISIONS DESCRIPTION	DATE	APPROVED

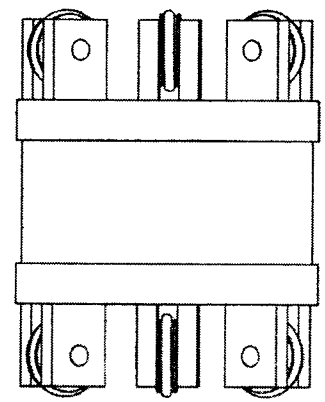


Rendered Picture of Car

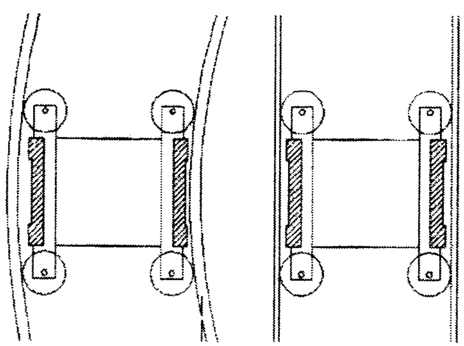


Top View





Front View



Car Negotiating Corner

PROPRIETARY AND CONFIDENTIAL
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NEXT ASSY USED ON
APPLICATION

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL --
FINISH --

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Ryan Ulansky	Oct. 03, 2002
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

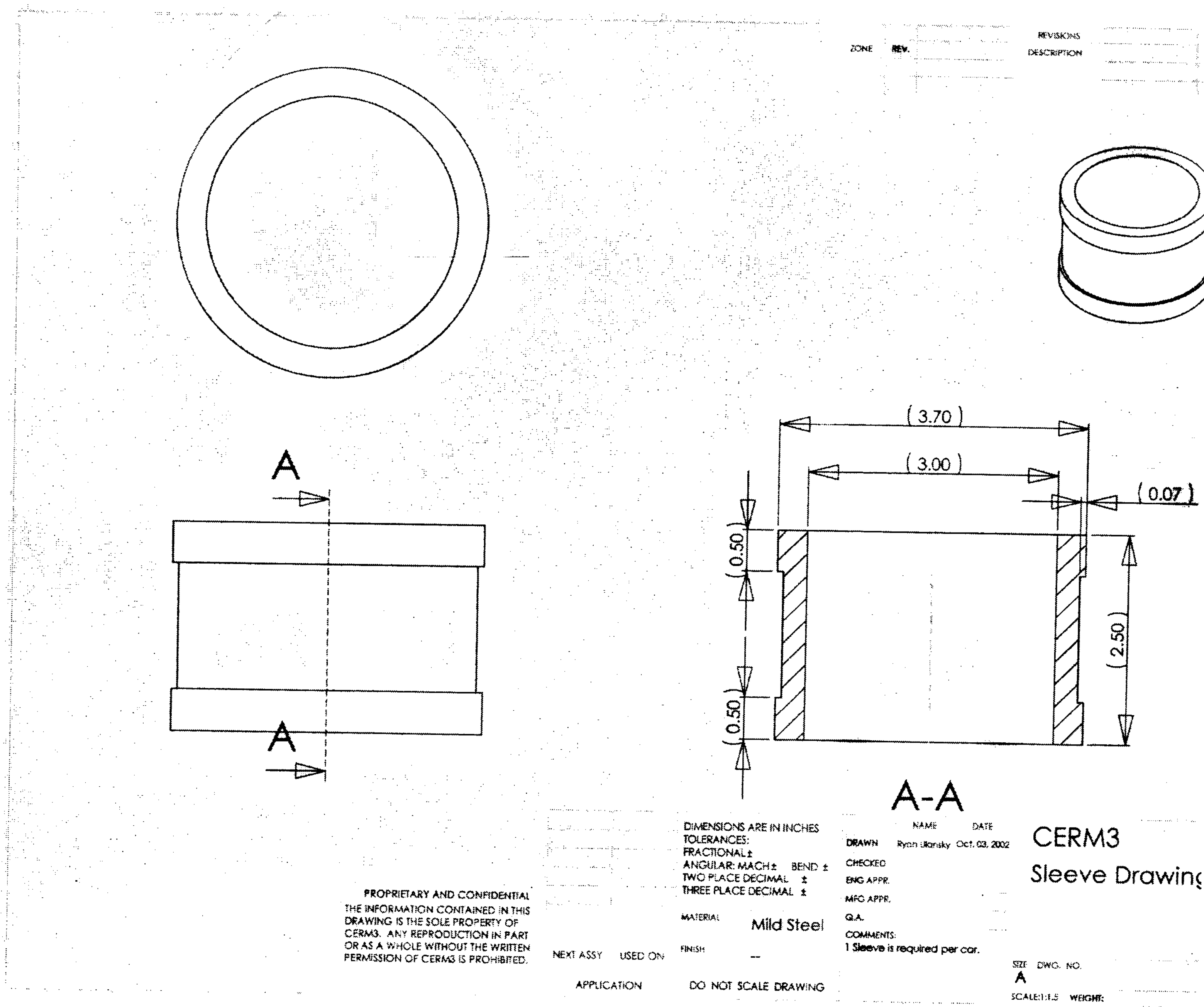
COMMENTS:
A total of 2 cars will be required.

CERM3

Assembled Car

SIZE A DWG. NO. REV. 1

SCALE:1:2 WEIGHT: SHEET 1 OF 4


ZONE REV.
REVISIONS
DESCRIPTION
A
A
(3.70)
(3.00)
(0.07)
(0.50)
(0.50)
(2.50)
A-A
PROPRIETARY AND CONFIDENTIAL
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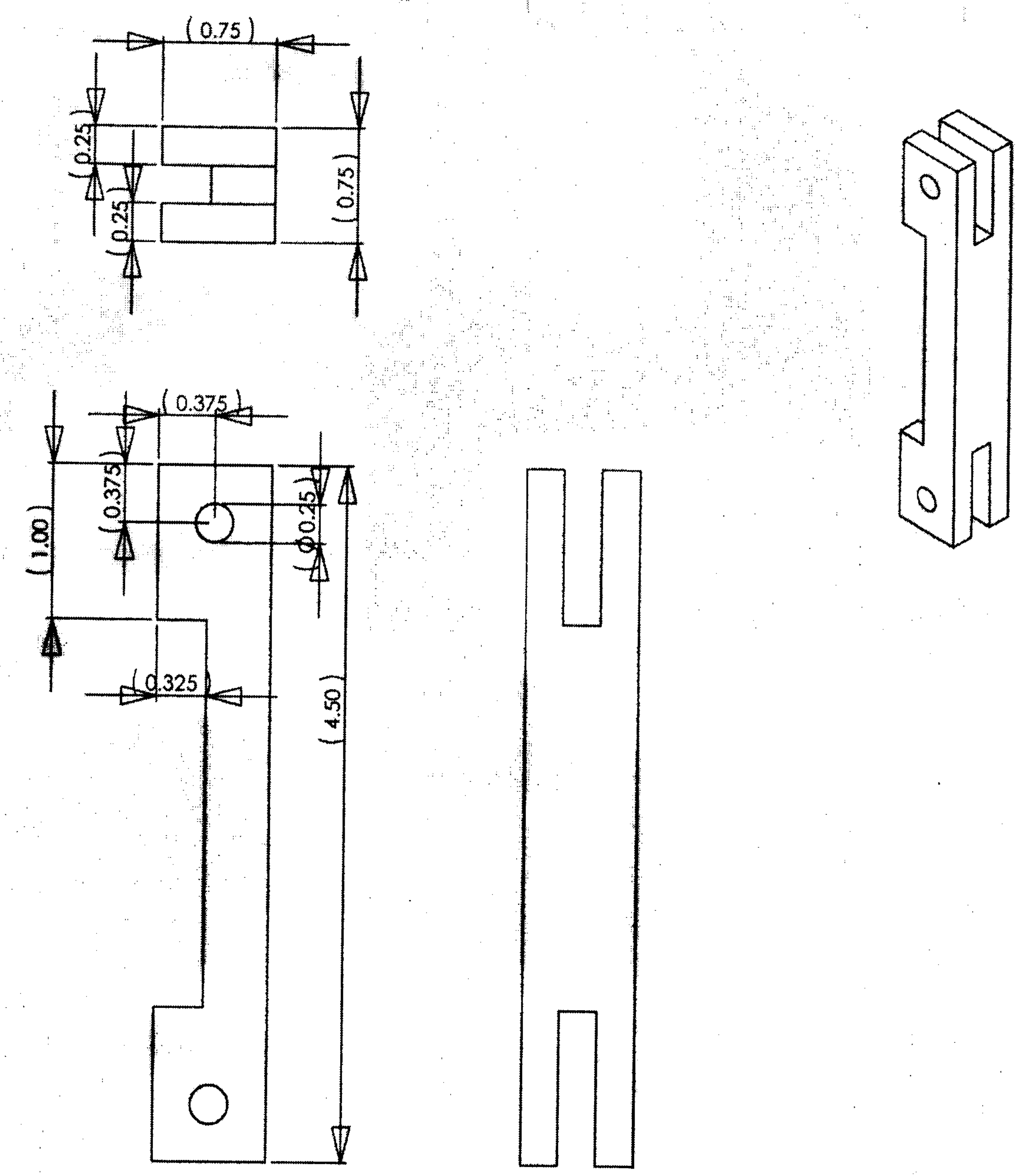
NEXT ASSY USED ON
APPLICATION
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
MATERIAL Mild Steel
FINISH --
DO NOT SCALE DRAWING
NAME DATE
DRAWN Ryan Ulansky Oct. 03, 2002
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
1 Sleeve is required per car.
CERM3
Sleeve Drawing
SIZE DWG. NO.
A
SCALE:1:1.5 WEIGHT:

ZONE	REV.	REVISIONS DESCRIPTION	DATE	APPROVED



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NEXT ASSY | USED ON
APPLICATION

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL
3/4" Aluminum Bar

FINISH
–

DO NOT SCALE DRAWING

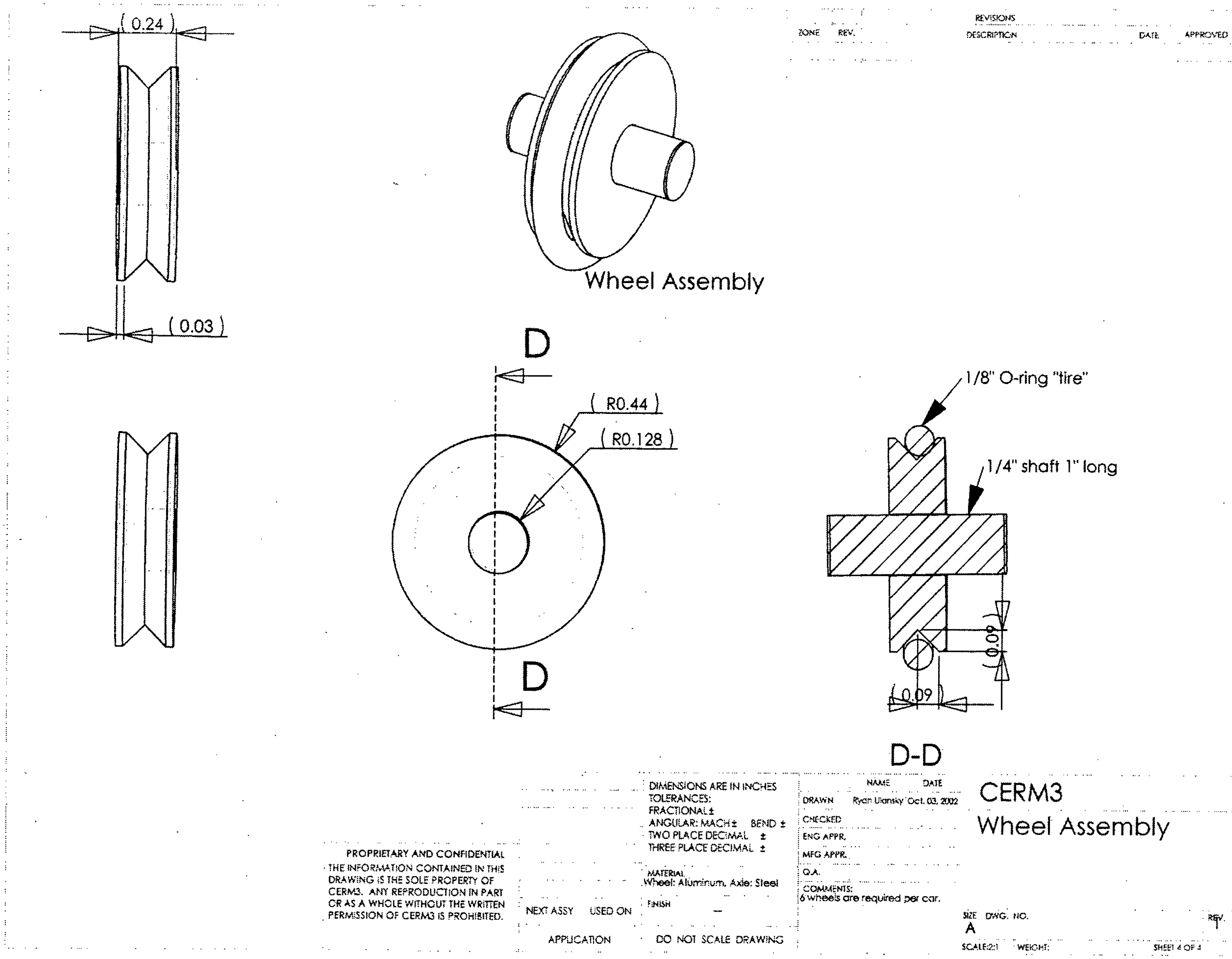
	NAME	DATE
DRAWN	Ryan Ulansky	Oct. 03, 2002
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
3 supports are required per car.

CERM3
Support Drawing

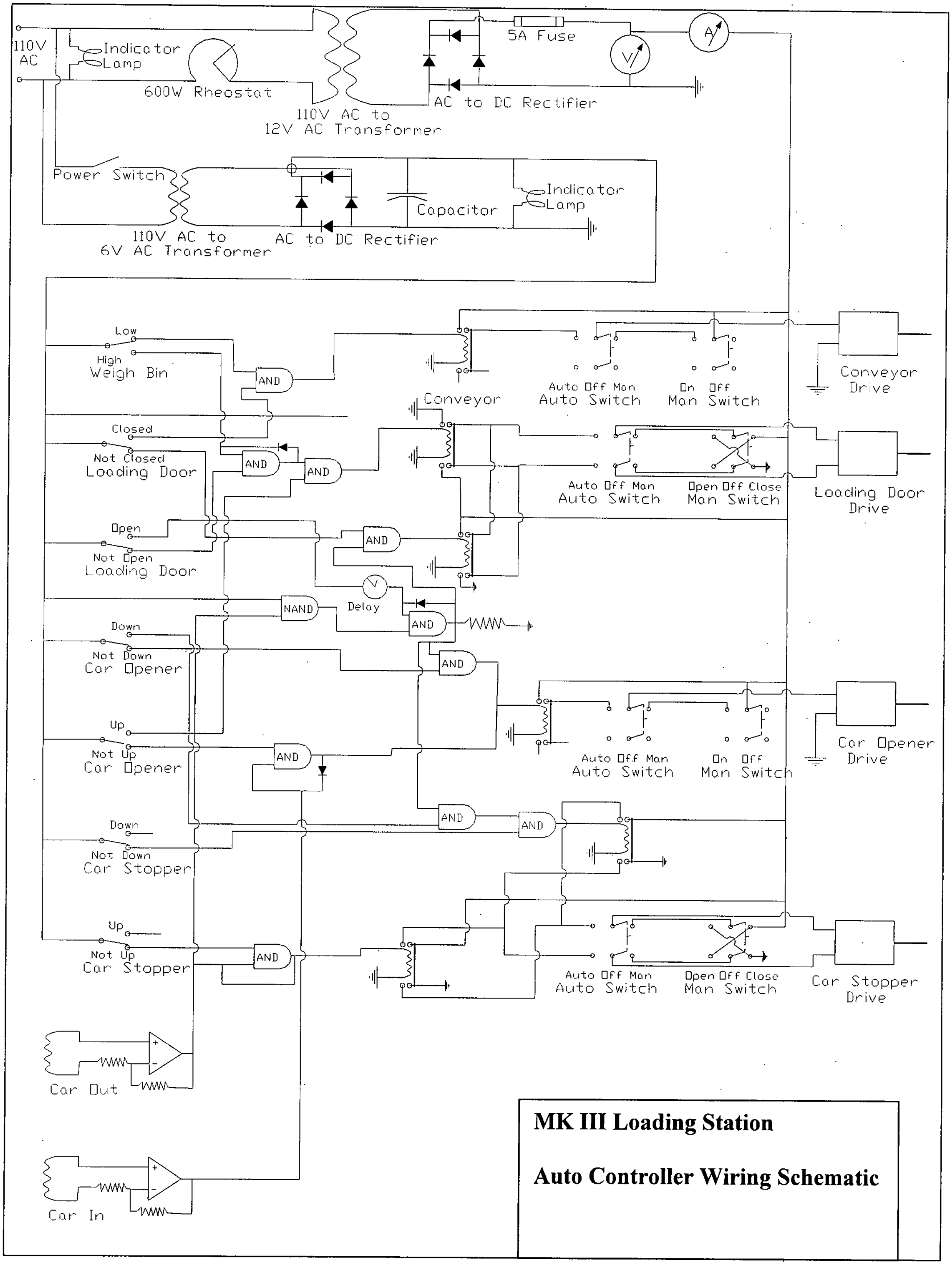
SIZE A | DWG. NO. | REV. 1

SCALE:1:1 | WEIGHT: | SHEET 3 OF 4



0.24
0.03
Wheel Assembly
D
R0.44
R0.128
D
1/8" O-ring "tire"
1/4" shaft 1" long
0.09
0.09
D-D
REVISIONS
ZONE
REV.
DESCRIPTION
DATE
APPROVED
PROPRIETARY AND CONFIDENTIAL
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF CERM3. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF CERM3 IS PROHIBITED.
NEXT ASSY
USED ON
APPLICATION
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
MATERIAL
Wheel: Aluminum, Axle: Steel
FINISH
—
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
Ryan Ulansky Oct. 03, 2002
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
6 wheels are required per car.
CERM3
Wheel Assembly
SIZE
DWG. NO.
A
REV.
SCALE:2:1
WEIGHT:
SHEET 4 OF 4

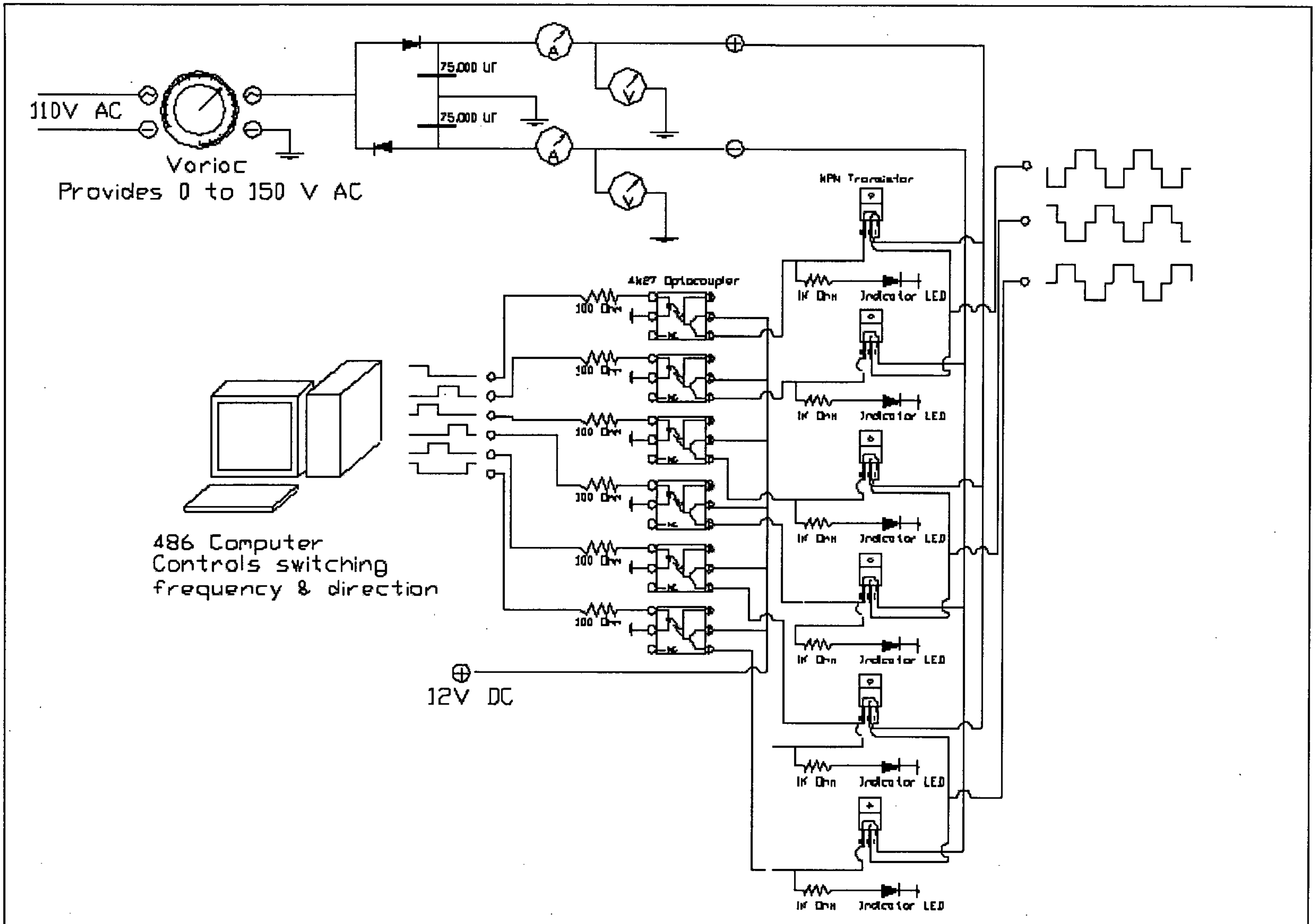
Appendix IV: Loading Station, Control Wiring Diagram



MK III Loading Station

Auto Controller Wiring Schematic

Appendix VI: Chopped 3 Phase Wiring Schematic

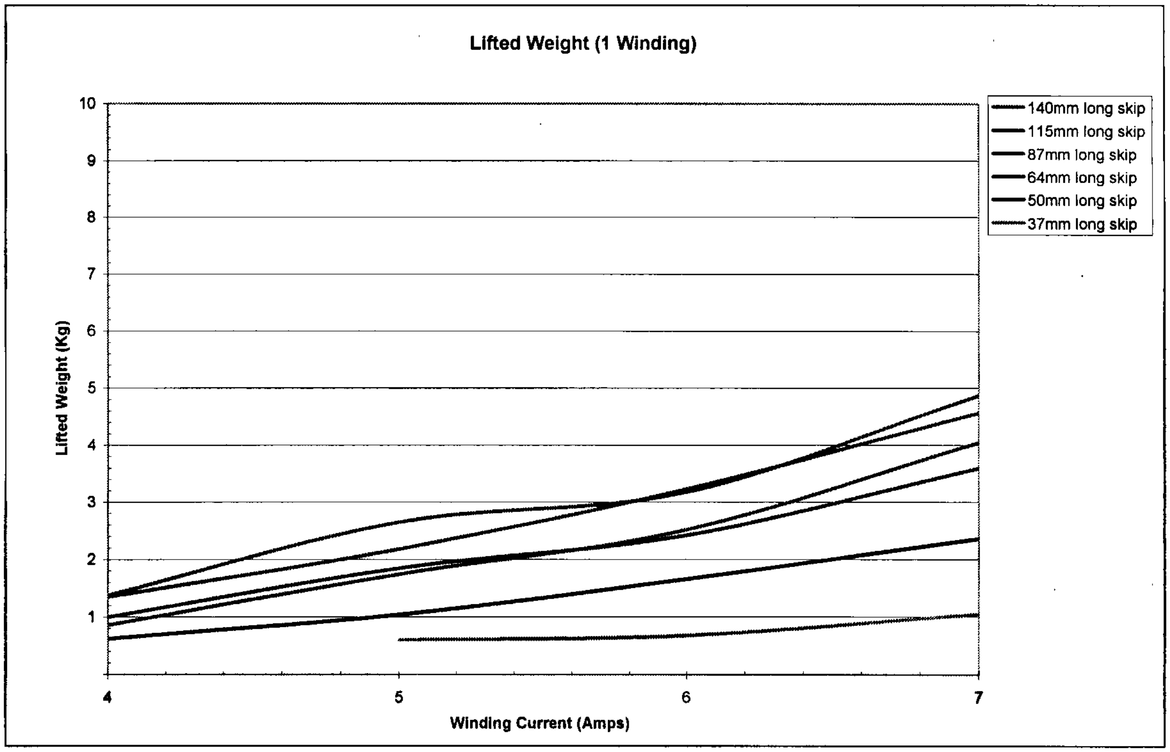


CHOPPED 3PH AC Wiring Schematic	Scale: NTS
	Date: June 14, 2003
	Drawn by: RWU
	Location: Bombom:\\Chopped AC

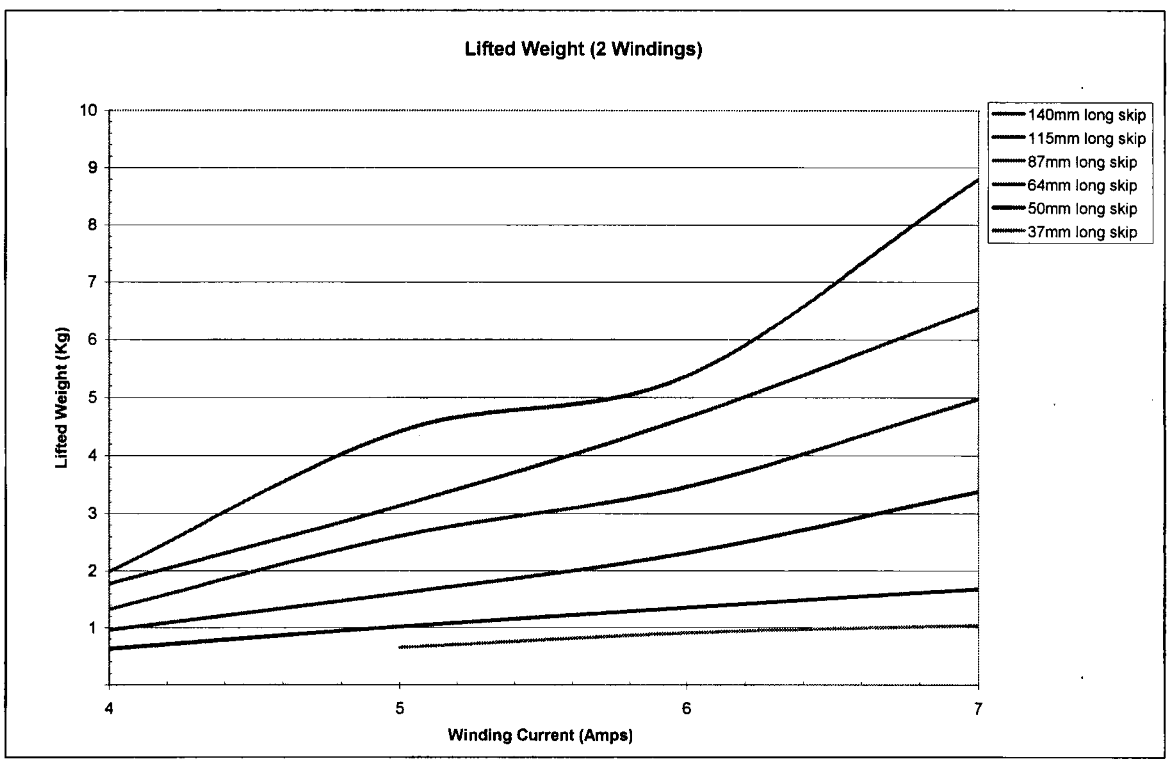
Appendix VII: First Linear Motor Test Results

Test Model Results

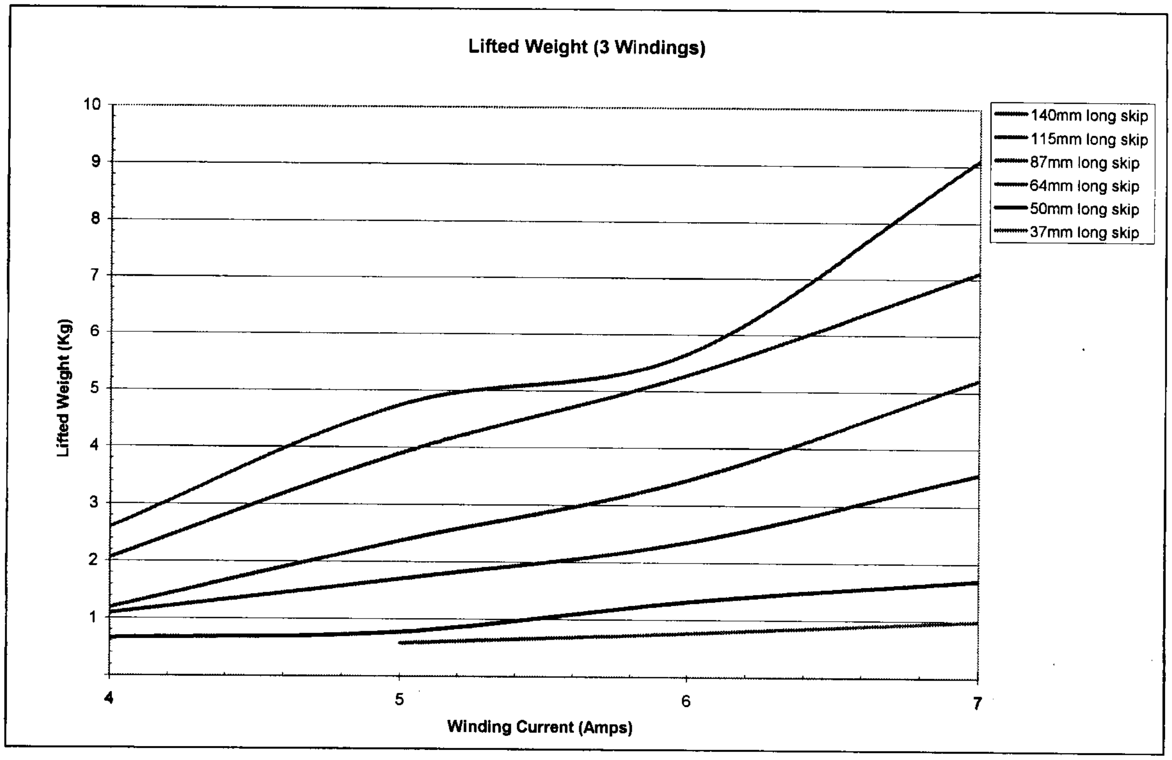
Windings	Volts	Amps	37mm Lift	37mm Hold	50mm Lift	50mm Hold	64mm Lift	64mm Hold	87mm Lift	87mm Hold	115mm Lift	115mm Hold	140mm Lift	140mm Hold
1	14	4		562	619	857	996	1235	855	1157	1350	1400	1375	1400
1	18	5	599	880	1040	1376	1850	2059	1743	2009	2181	2304	2650	2671
1	19	6	675	1120	1660	2002	2423	2596	2520	2711	3230	3314	3183	3675
1	29	7	1043	1547	2358	2584	3595	3830	4041	4123	4566	4732	4871	4945
2	14	4		488	635	1122	959	1786	1325	2493	1778	2860	1990	3090
2	18	5	656	995	1021	1844	1602	2943	2610	4174	3136	5482	4419	5469
2	19	6	906	1275	1352	2900	2314	4076	3465	6200	4658	7220	5376	8115
2	29	7	1037	1464	1674	3316	3384	5010	4975	8656	6538	10015	8805	11490
3	14	4		669	659	1059	1095	2135	1195	2833	2060	4931	2595	4745
3	18	5	578	1025	776	1844	1704	3095	2370	4544	3895	7192	4745	8776
3	19	6	768	1255	1320	2585	2357	4280	3440	6569	5286	10154	5661	11220
3	29	7	990	1579	1700	3660	3552	5790	5212	8109	7109	12696	9105	13876
4	14	4		676	621	1298	1183	1994	1236	3112	2118	4807	2615	6698
4	18	5	506	851	961	2150	1847	3039	2378	5240	3472	8324	4798	10115
4	19	6	709	1204	1396	3033	2430	4260	3437	6045	4638	11878	5910	10975
4	29	7	913	1924	1732	3720	3570	5770	4745	8980	6537	13672	8505	12957
5	14	4		580	602	926	950	1458	1125	2453	1700	4905	2739	7028
5	18	5	618	999	771	1612	1586	2730	2308	4279	3235	7884	4542	9859
5	19	6	832	1304	1109	2421	2411	3516	2676	6266	4250	10325	5624	10635
5	29	7	1150	2063	1632	3517	3167	4920	4126	9320	6300	11938	7712	14355
6	14	4		692		829		1455		2536		4705		6340
6	18	5		880		1503		2485		5105		8242		9345
6	19	6		1345		2248		3859		6750		9586		10413
6	29	7		1846		3057		5430		9158		12742		12325

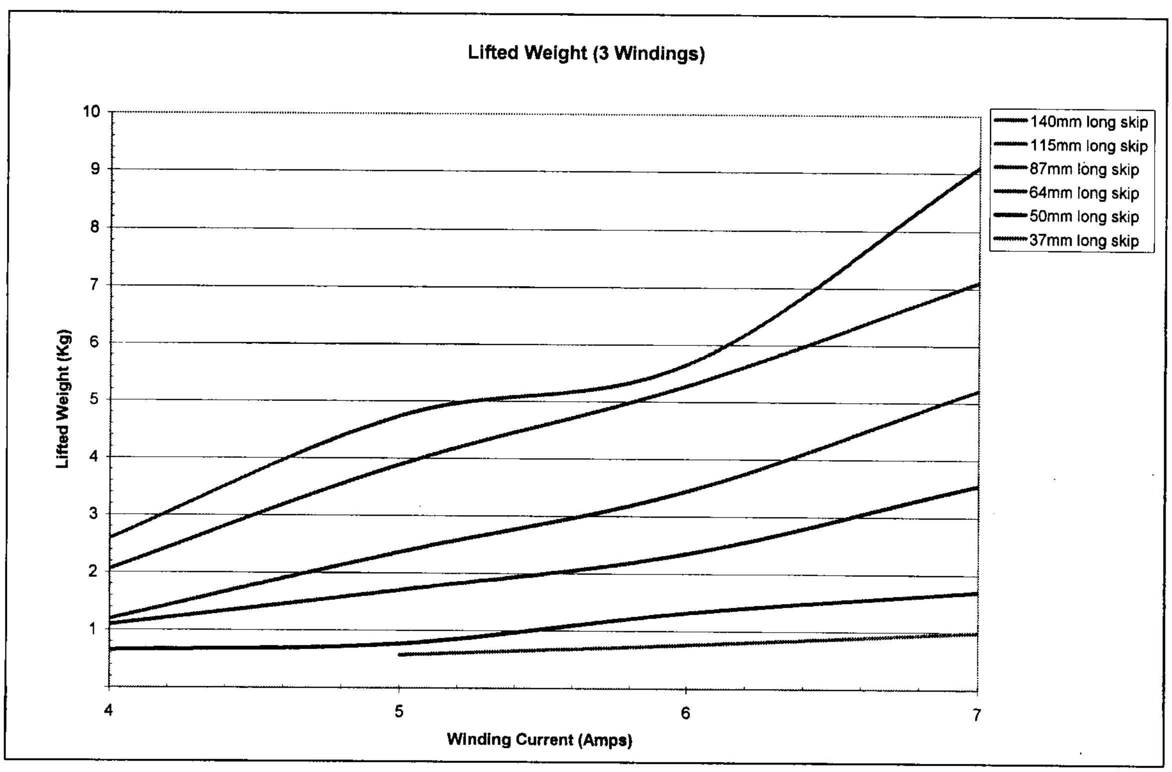


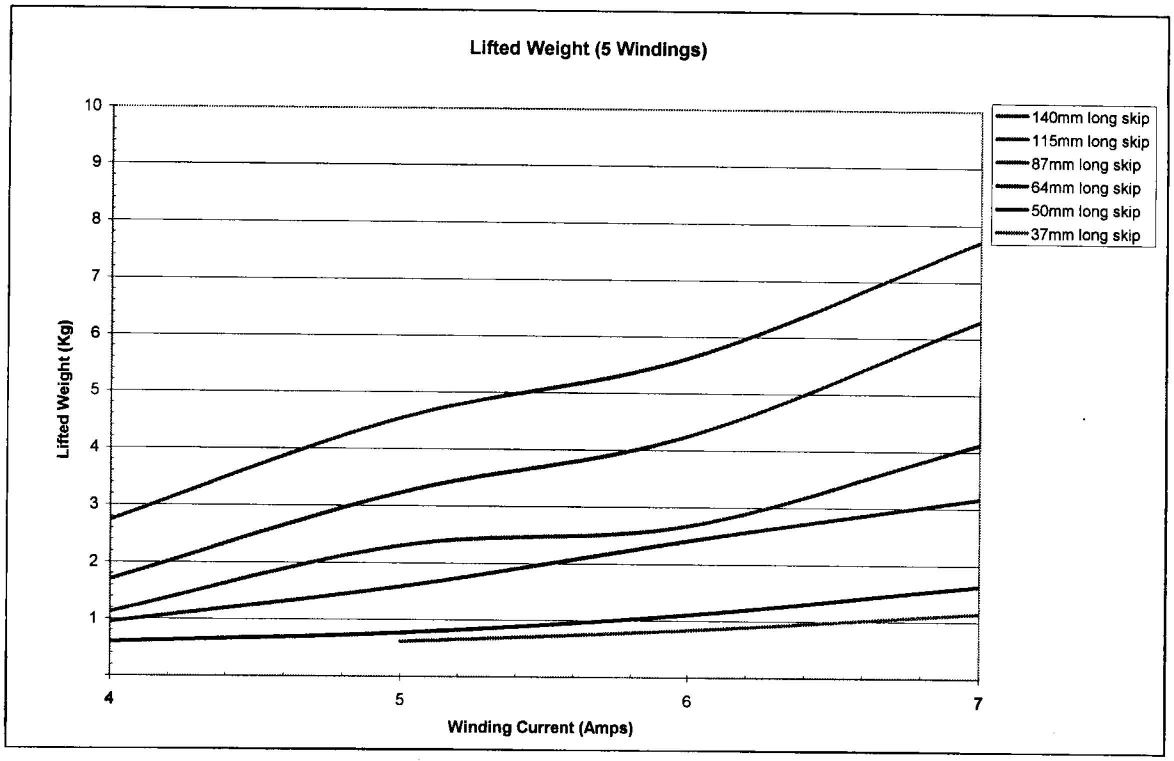
Lifted Weight (1 Winding)
Lifted Weight (Kg)
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

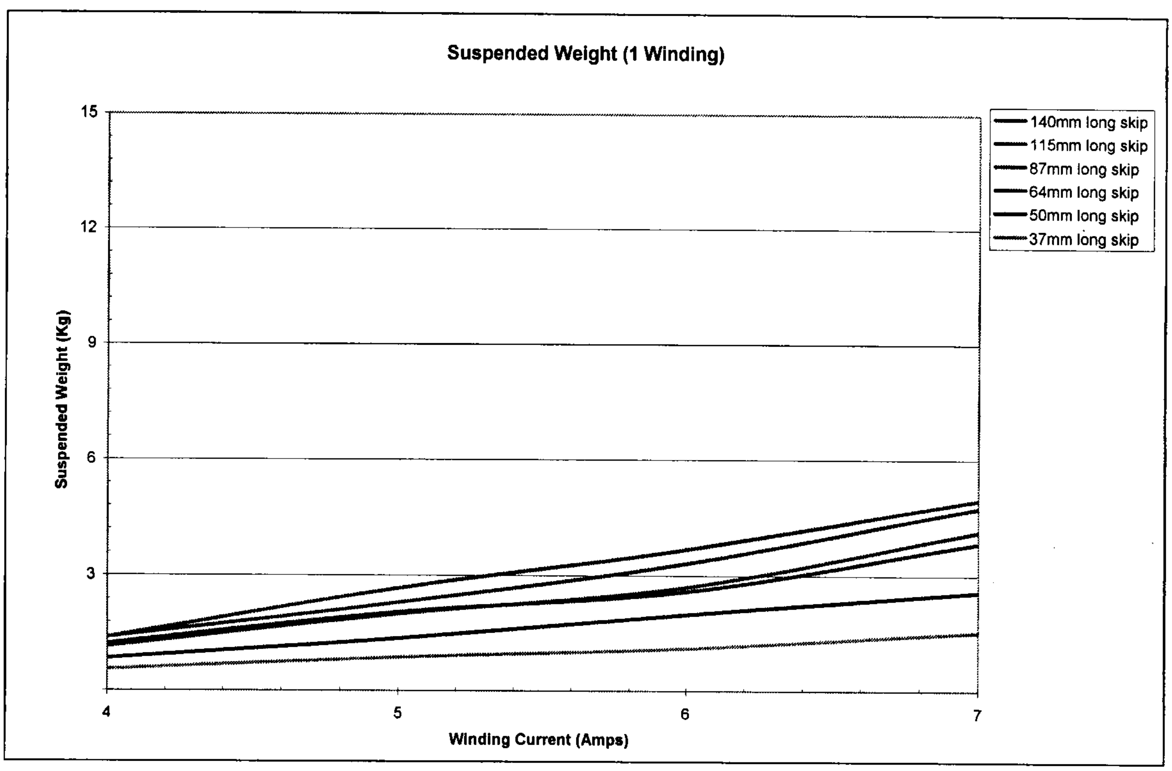


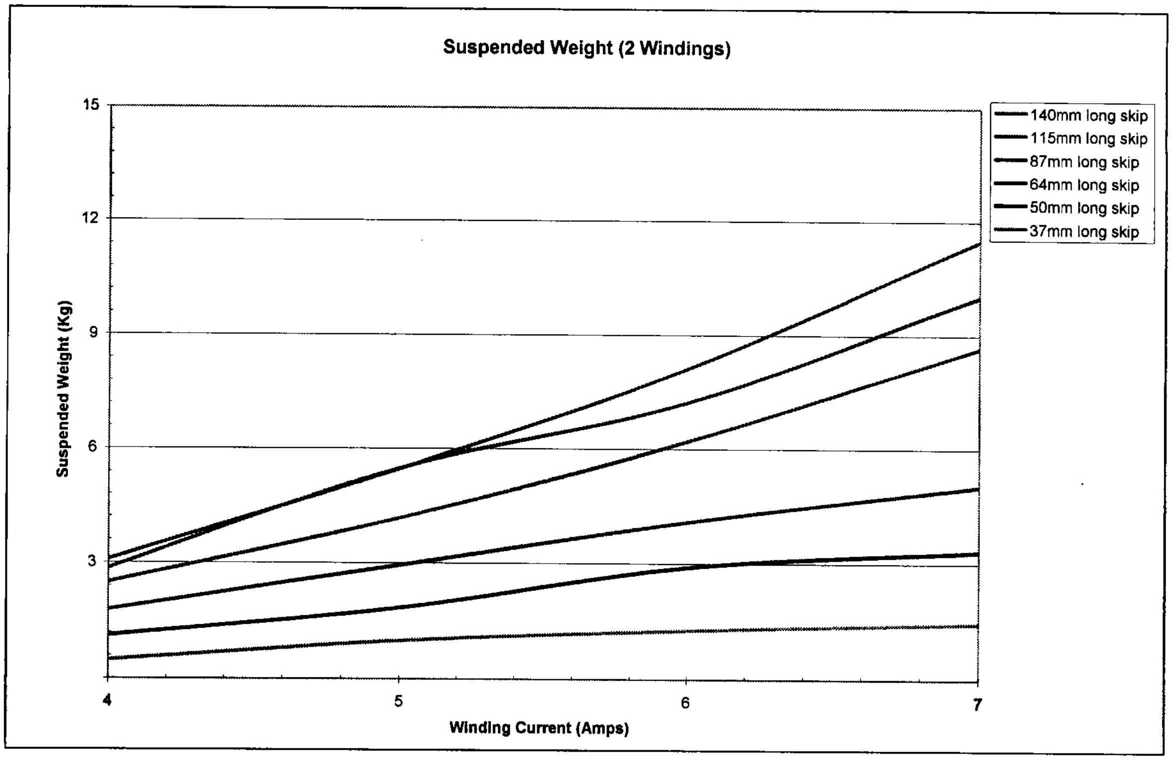
Lifted Weight (2 Windings)
Lifted Weight (Kg)
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

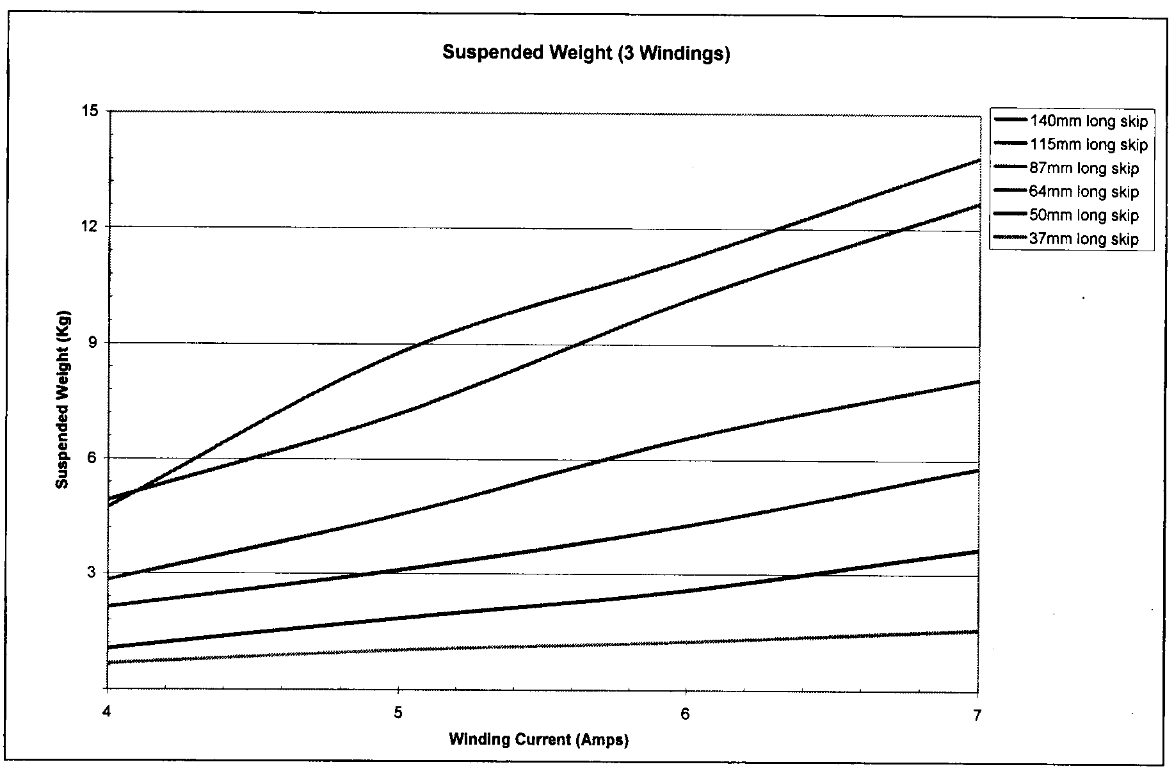

Lifted Weight (3 Windings)
Lifted Weight (Kg)
10
9
8
7
6
5
4
3
2
1
4
5
6
7
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

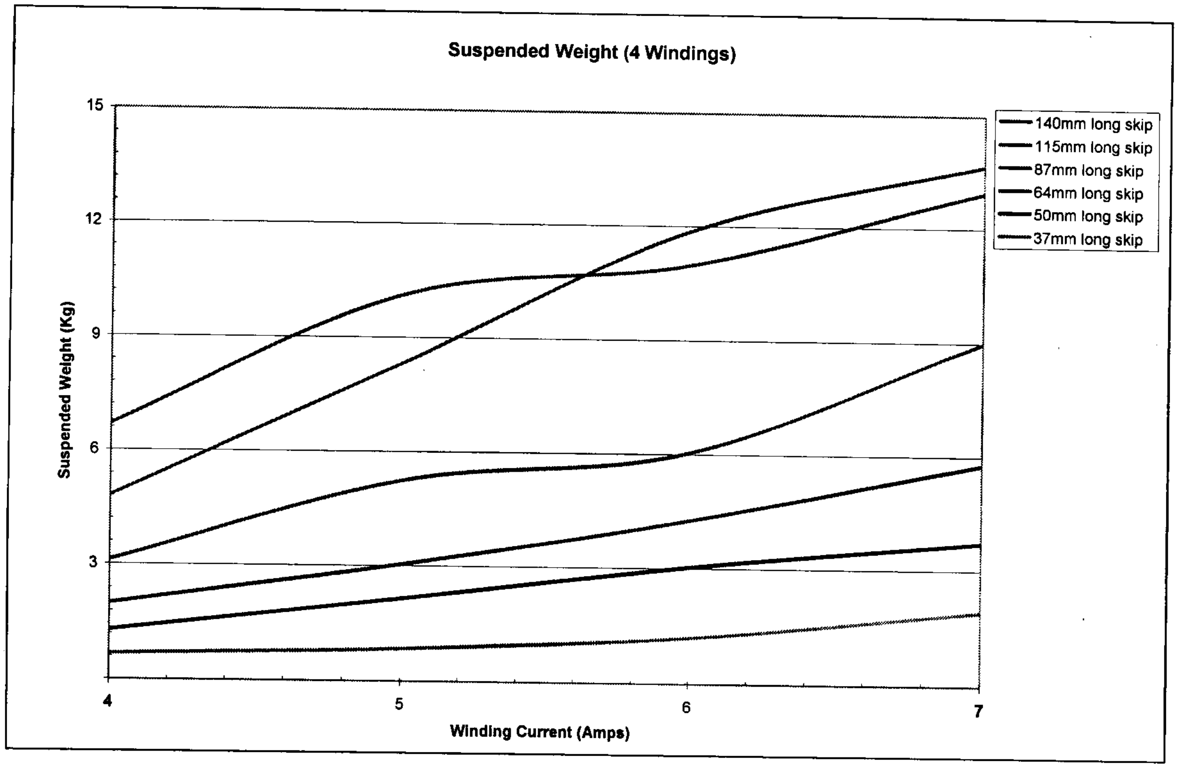

Lifted Weight (3 Windings)
Lifted Weight (Kg)
10
9
8
7
6
5
4
3
2
1
4
5
6
7
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

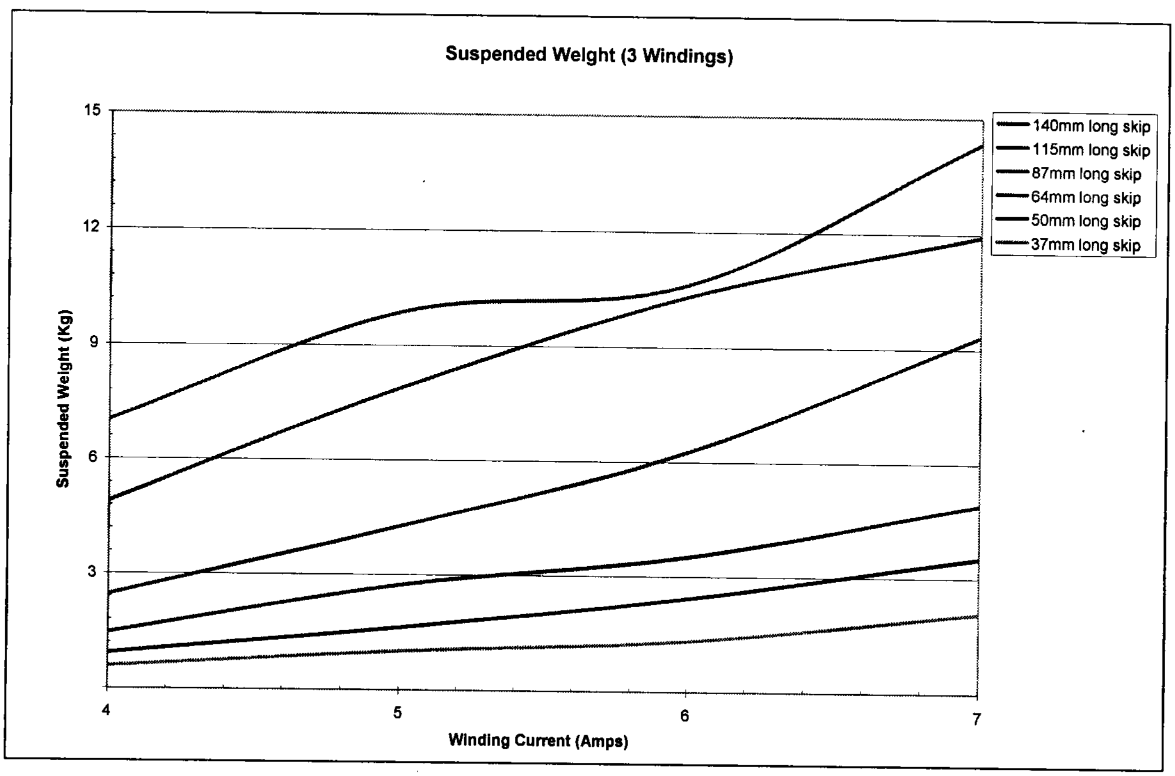

Lifted Weight (5 Windings)
Lifted Weight (Kg)
10
9
8
7
6
5
4
3
2
1
4
5
6
7
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

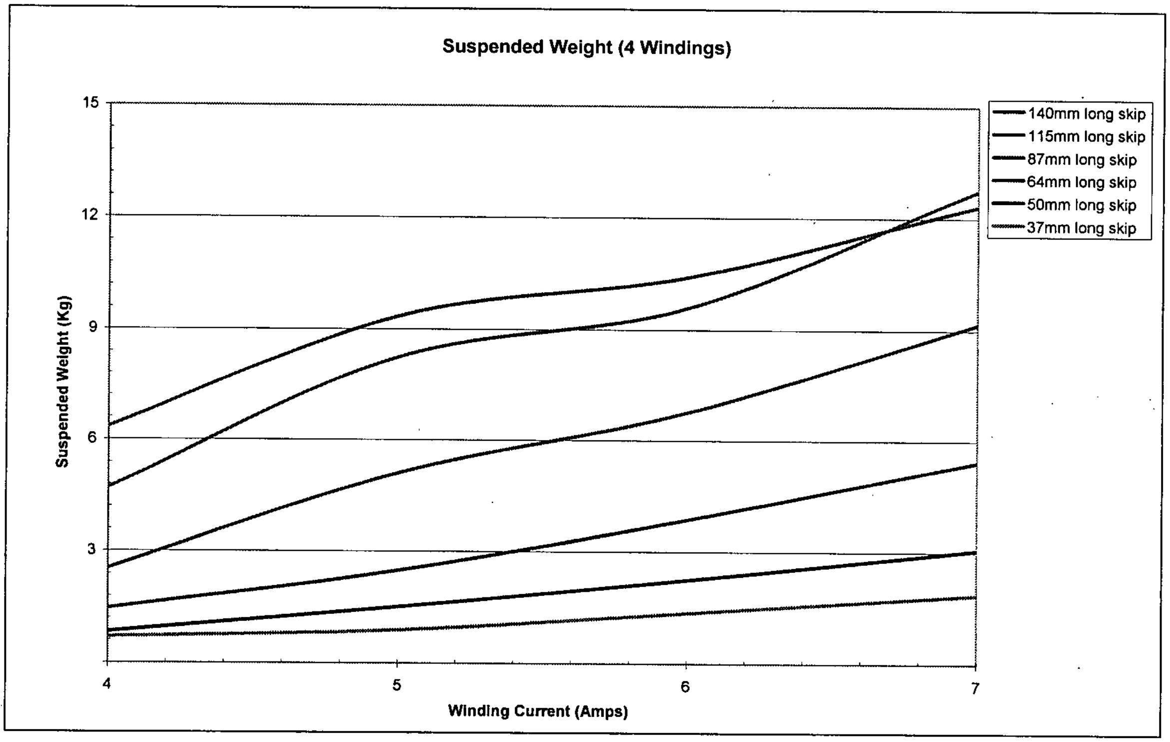

Suspended Weight (1 Winding)
Suspended Weight (Kg)
15
12
9
6
3
4
5
6
7
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

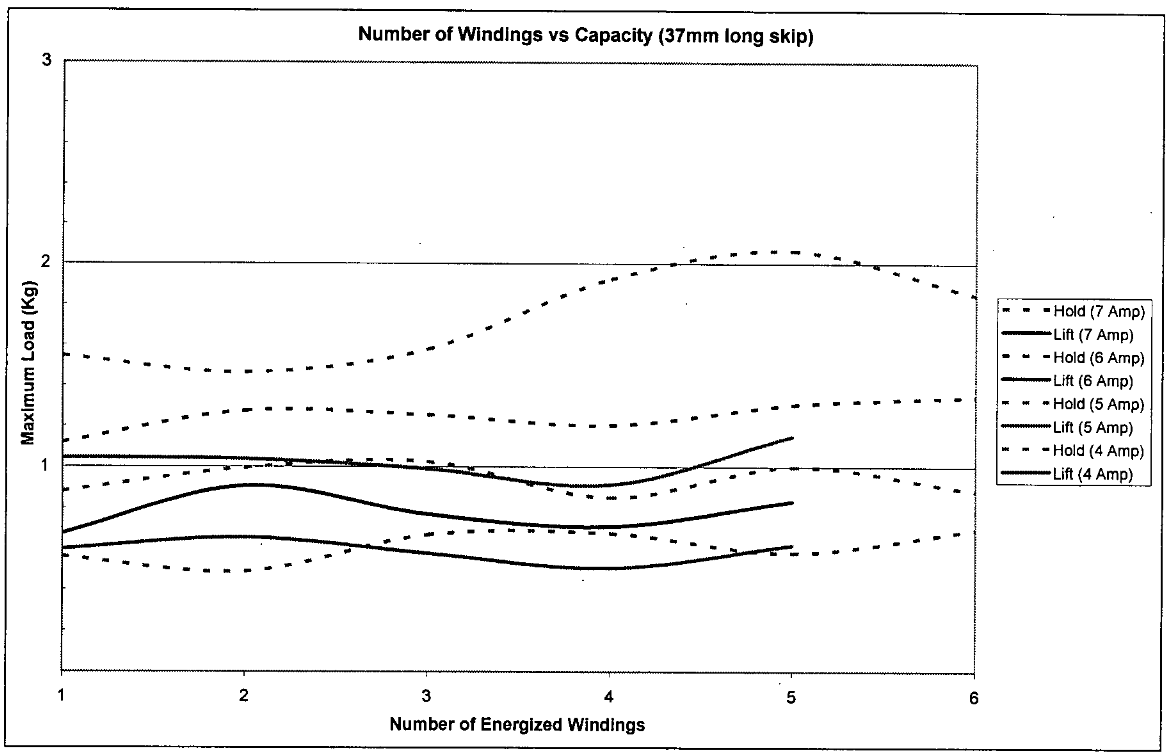

Suspended Weight (2 Windings)
15
12
9
6
3
4
5
6
7
Winding Current (Amps)
Suspended Weight (Kg)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

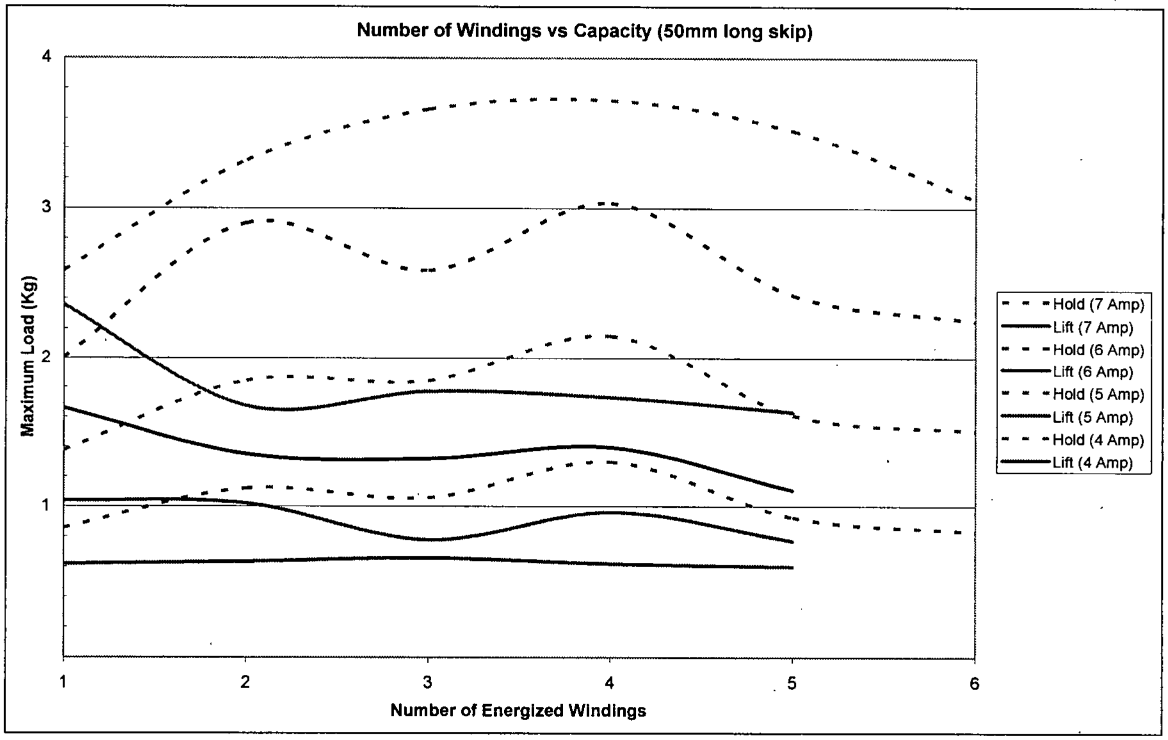

Suspended Weight (3 Windings)
15
12
9
6
3
4
5
6
7
Winding Current (Amps)
Suspended Weight (Kg)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

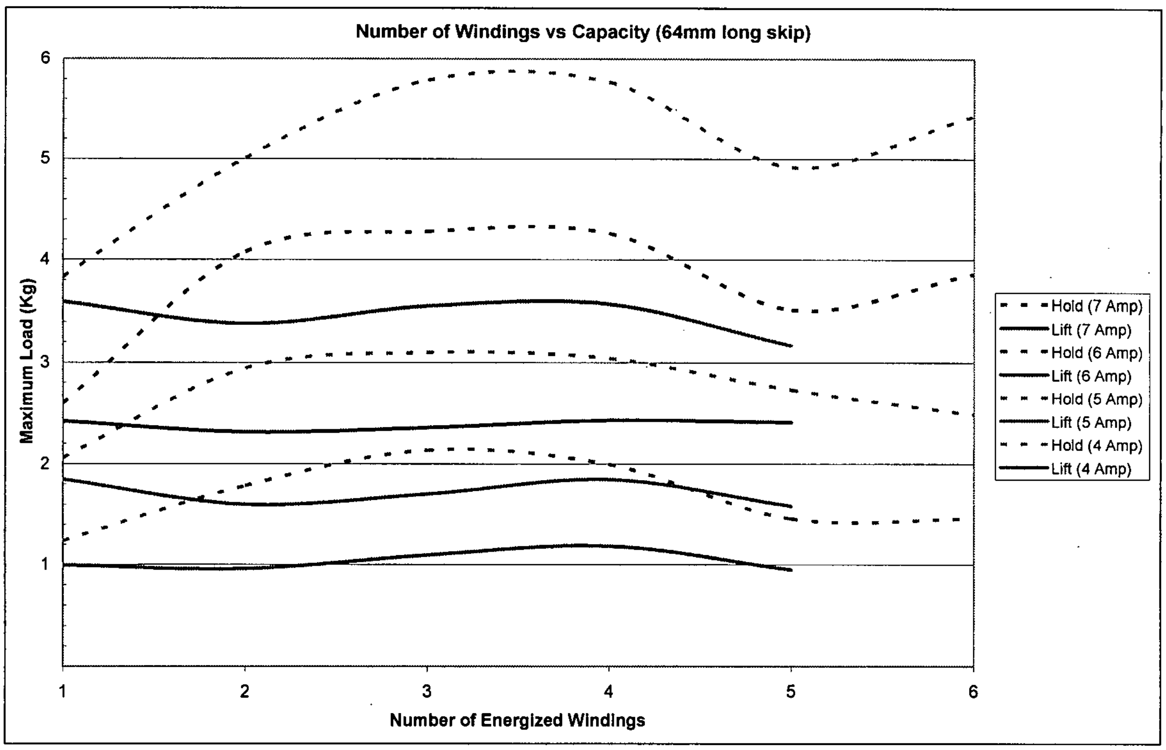

Suspended Weight (4 Windings)
15
12
9
6
3
Suspended Weight (Kg)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip
4
5
6
7
Winding Current (Amps)

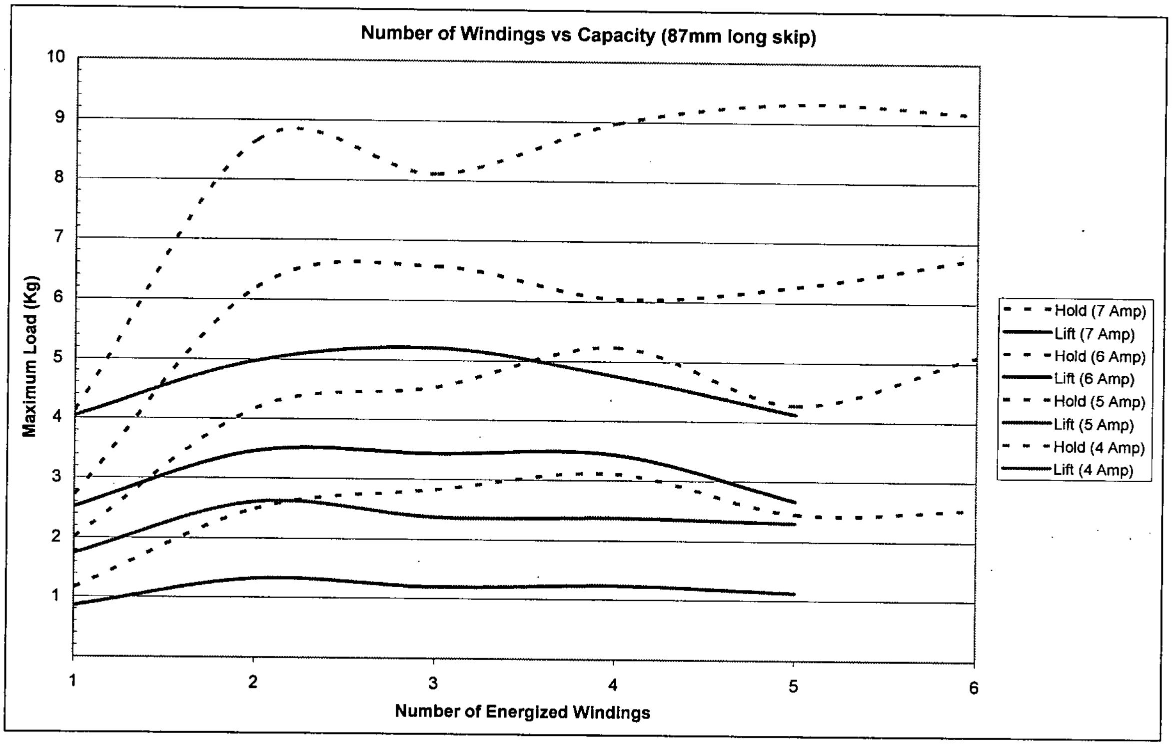

Suspended Weight (3 Windings)
15
12
9
6
3
Suspended Weight (Kg)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip
4
5
6
7
Winding Current (Amps)

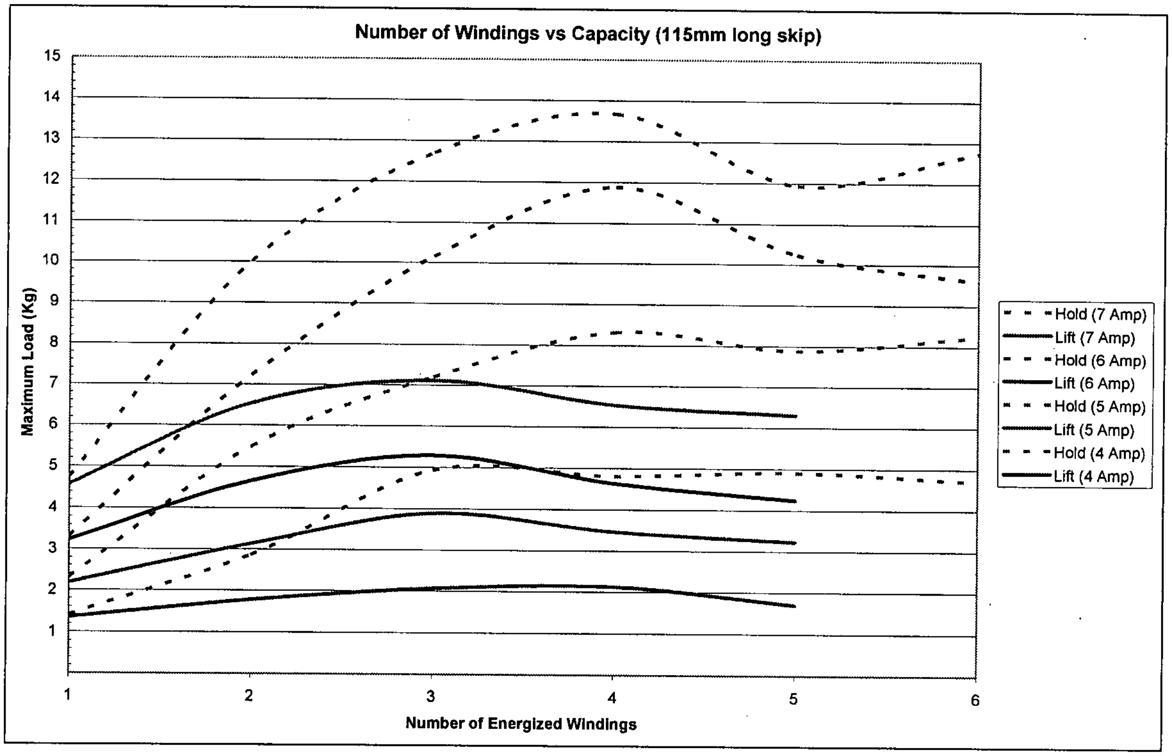

Suspended Weight (4 Windings)
Suspended Weight (Kg)
15
12
9
6
3
4
5
6
7
Winding Current (Amps)
140mm long skip
115mm long skip
87mm long skip
64mm long skip
50mm long skip
37mm long skip

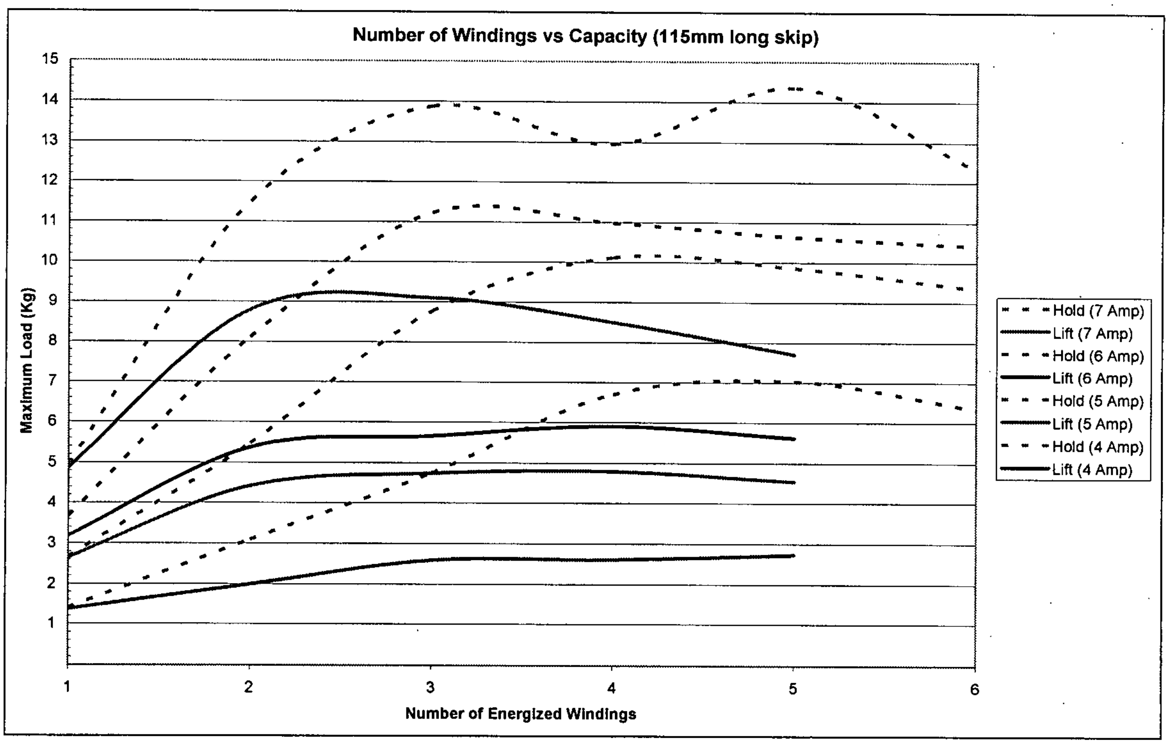

Number of Windings vs Capacity (37mm long skip)
Maximum Load (Kg)
3
2
1
1
2
3
4
5
6
Number of Energized Windings
Hold (7 Amp)
Lift (7 Amp)
Hold (6 Amp)
Lift (6 Amp)
Hold (5 Amp)
Lift (5 Amp)
Hold (4 Amp)
Lift (4 Amp)


Number of Windings vs Capacity (50mm long skip)
Maximum Load (Kg)
Number of Energized Windings
1
2
3
4
5
6
Hold (7 Amp)
Lift (7 Amp)
Hold (6 Amp)
Lift (6 Amp)
Hold (5 Amp)
Lift (5 Amp)
Hold (4 Amp)
Lift (4 Amp)


Number of Windings vs Capacity (64mm long skip)
Maximum Load (Kg)
Number of Energized Windings
1
2
3
4
5
6
Hold (7 Amp)
Lift (7 Amp)
Hold (6 Amp)
Lift (6 Amp)
Hold (5 Amp)
Lift (5 Amp)
Hold (4 Amp)
Lift (4 Amp)

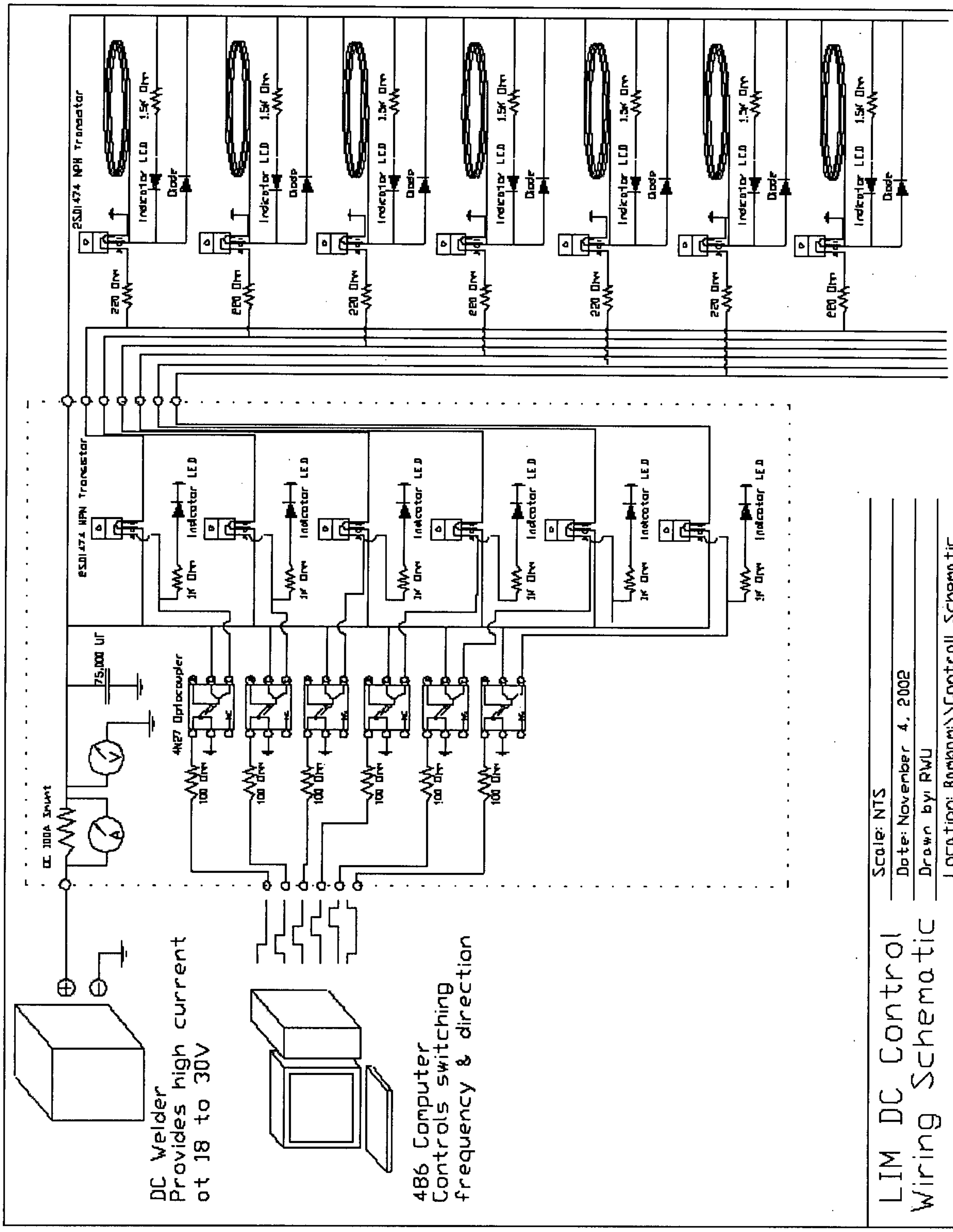

Number of Windings vs Capacity (87mm long skip)
Maximum Load (Kg)
10
9
8
7
6
5
4
3
2
1
1
2
3
4
5
6
Number of Energized Windings
Hold (7 Amp)
Lift (7 Amp)
Hold (6 Amp)
Lift (6 Amp)
Hold (5 Amp)
Lift (5 Amp)
Hold (4 Amp)
Lift (4 Amp)

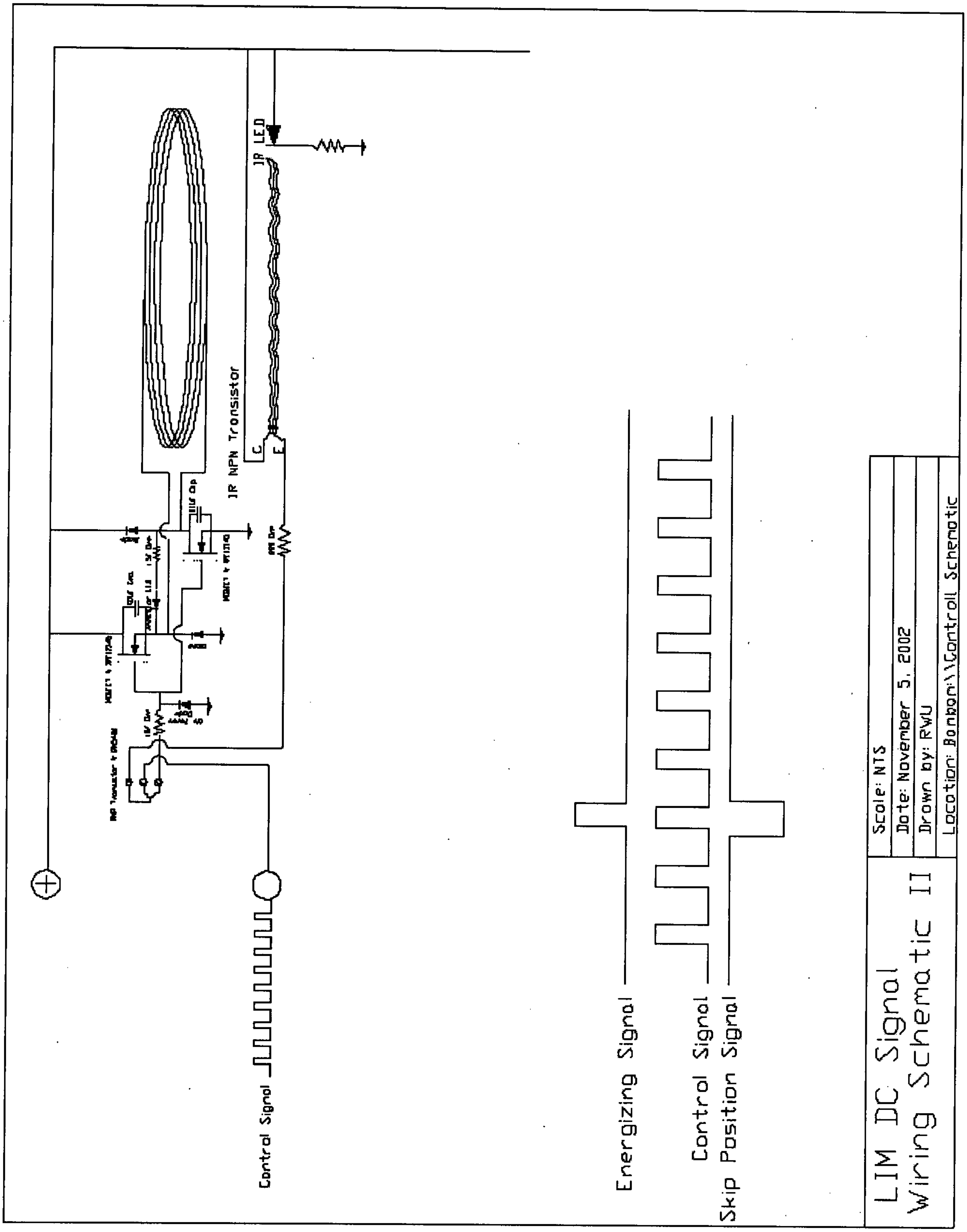

Number of Windings vs Capacity (115mm long skip)
Maximum Load (Kg)
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
1
2
3
4
5
6
Number of Energized Windings
Hold (7 Amp)
Lift (7 Amp)
Hold (6 Amp)
Lift (6 Amp)
Hold (5 Amp)
Lift (5 Amp)
Hold (4 Amp)
Lift (4 Amp)



Number of Windings vs Capacity (115mm long skip)
Maximum Load (Kg)
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
1
2
3
4
5
6
Number of Energized Windings
Hold (7 Amp)
Lift (7 Amp)
Hold (6 Amp)
Lift (6 Amp)
Hold (5 Amp)
Lift (5 Amp)
Hold (4 Amp)
Lift (4 Amp)

Appendix VIII: Control Wiring Schematics


DC Welder
Provides high current
at 18 to 30V
486 Computer
Controls switching
frequency & direction
DC 100A Shunt
75,000 uF
4N27 Optocoupler
100 Ohm
2SD1474 NPN Transistor
Indicator LED
1K Ohm
220 Ohm
1.5K Ohm
Diode
LIM DC Control
Wiring Schematic
Scale: NTS
Date: November 4, 2002
Drawn by: RWU
Location: Bomboom\\Controll Schematic


IR LED
IR NPN Transistor
C
E
Control Signal
Energizing Signal
Control Signal
Skip Position Signal
LIM DC Signal
Wiring Schematic II
Scale: NTS
Date: November 5, 2002
Drawn by: RWU
Location: Bonbon:\\Controll Schematic

Appendix IX: Risk Register

Magnetically Propelled Hoisting

Risk Register as of Feb 23, 2003

Report for: CIVL 498
Project Manager: Ryan Ulansky
Project Scope: The design, construction, operation, and maintenance of a magnetically propelled hoisting system. Primary focus is on the transportation portion of the system not the loading dumping unit operations.

Rating for likelihood and seriousness for each risk

L	Rated as Low
M	Rated as Medium
H	Rated as High
E	Rated as Extreme
NA	Not Assessed

Grade: Combined effect of likelihood / Seriousness

		Seriousness			
		Low	Med	High	Extreme
Likelihood	Low	E	D	C	A
	Med	D	C	B	A
	High	C	B	A	A

Recommended actions for grades of risk

Grade	Risk Mitigation Actions
A	Action to reduce the likelihood and seriousness to be identified and implemented as soon as project commences.
B	Action to reduce the likelihood and seriousness to be identified and appropriate actions implemented during project execution.
C	Action to reduce the likelihood and seriousness to be identified and costed for possible action if funds permit.
D	To be noted, no action needed unless grading increases over time
E	To be noted, no action needed unless grading increases over time

Change to Grade since last assessment

NEW	New Risk
⇑	Grade increasing
-	No change to grade
⇓	Grade decreasing

Risk Register – Magnetically Propelled Hoisting

Last Updated: Feb. 23, 2003

ID	Description of Risk Identify Consequences	Likelihood	Seriousness	Grade	Change	Mitigation Actions	Applicable Stage	Responsible Officer
1	Water infiltrating motor, could cause corrosion	H	H	A	NEW	Motor must be designed to operate in wet environments. Iron components must be sealed during construction.	Design & Construction	Construction manager
2	Mine water going acidic, corroding motor	NA	H	NA	NEW	Site specific, if there is the potential for acidic mine drainage, motor will have to have to be shielded from the water.	Design & Erection	Erection Manager
3	Motor not aligned during installation, cars jamming while in transit	M	H	B	NEW	Mounting system must have sufficient adjustment to allow for alignment. Erection must be inspected to insure correct alignment	Design & Erection	Erection Manager
4	Control system failure, winding remaining energized, overheating causes fire	L	M	D	NEW	No action required		
5	Winding short circuit, winding over heating causing fire	M	M	C	NEW	Thermal cut out protection should be provided to prevent fires.	Design	Engineering Manager
6	Electrical short to back iron producing shock hazard and possible death to employees	L	E	A	NEW	Outer surface must have an insulating covering to prevent shock hazard. Proper lock out procedures must be implemented when maintaining system	Design & Maintenance	Engineering & Maintenance Managers
7	Car becoming projectile when section removed for	L	E	A	NEW	Proper lockout and maintenance procedures must be developed.	Design & Maintenance	Engineering &

	maintenance, possibly striking & killing worker					Design of an impact adsorbing barrier may be required.		Maintenance Managers
8	Power spike damaging control circuits, resulting in portions of system becoming inoperable.	M	M	C	NEW	Power conditioning will be required to prevent damaging surges to circuitry. Critical control components may require zener diodes as backup protection.		
9	Mechanical failure of structural integrity.	L	H	C	NEW	Design of support system will have to include tolerance to mechanical vibrations induced by the passing cars.	Design	Design Manager
10	Seismic vibrations, possible deformation or breaking of joist between sections	M	H	B	NEW	Design will have to tolerate regular seismic activity. Blast practices may have to be altered near system.	Design & Operation	Design Manager & Blasting Engineer
11	Vandalism, damage to system, potential injury to personal	L	H	C	NEW	Access to key components must be controlled or protected. Security or alarm system may be required. Extreme care must be exercised after suspected tampering.	Operation	Hoisting Forman
12	Inexperience personal performing inappropriate operating or maintenance practices to system.	M	H	B	NEW	Extensive training will be required to educate operators on the unique aspects of system. Computerized control system should not allow operator to perform hazardous maneuvers.	Operation & maintenance	Hoisting Forman & Maintenance Manager
13	Overloading system, trying to set records for bonus by tampering with weight sensors to overload cars.	M	H	B	NEW	Education on risks involved with strict discipline for anyone caught tampering with equipment. Possible application for remote cameras.	Operation	Hoisting Forman

14	Generation of noxious fumes, health risk to personal underground.	M	H	B	NEW	Materials selection must insure that noxious fumes will not be produced by system.	Design	Design Manager
15	Jammed cars in tube, stopped production, cost of clearing and restarting system	M	H	B	NEW	Car design must insure that material will not be spilt from cars while in transit. System must be designed in such a way to allow easy access for section replacement.	Design	Design Manager
16	Power failure, loss of propulsion to cars in system, possible free fall causing extreme damage and potential injury.	M	E	A	NEW	Fail safe flapper gates will have to be installed in all sections installed on a grade. Automatically turning drive windings into regenerative braking may have some benefits to safely stopping cars.	Design	Design Manager
17	Failure of braking system, cars free fall causing damage and possible injury	L	E	A	NEW	Braking windings should be designed in a parallel fashion to prevent catastrophic failure.	Design	Design Manager
18	Delays in producing sections, delay erection, could delay commissioning of mine	H	H	A	NEW	Construction of test sections and prototypes should be used as an opportunity to test construction methods.	Construction	Construction Manager
19	Ground movement throwing alignment out.	M	H	B	NEW	Design must be tolerant of some ground movement. System must be checked before resuming hoisting after suspected ground movement.	Operation	Hoisting Forman

Appendix X: Failure Mode Effect Analysis

Component	Failure Mode	Effects	Project Stage	Likelihood	Consequences: System Function	Consequences: Component Failure	Consequences: Health & Safety	Level of Confidence	Comments
Drive Tube	Shorted Winding	Loss of Power	O	M	L	L	M	M	Winding must be insulated to prevent shocks
	Fatigued Iron Core	Loss of Mechanical Strength	O	L	M	H	L	M	Core provides structural Strength to member
	Inner Tube wear Rate	Early Replacement or Possibility of Car Jamming	O	M	L	M	N	L	Material Testing Will be Important for system success
	Faulty Design	Overheating of windings or lack of power.	D	L	H	L	N	M	Prototyping of Concept will reduce chance of failure.
	Tube Alignment	Cars may jam if tubes become unaligned.	C	M	H	N	N	M	Installation will be important.

Project Stage:
- D = Design
- C = Construction
- O = Operation

Likelihood:
- N = Not Likely
- L = Low
- M = Moderate
- H = High
- E = Extreme

Consequence:
- N = Negligible
- L = Low
- M = Moderate
- H = High
- E = Extreme

Level of Confidence:
- L = Low
- M = Moderate
- H = High

	Human Sabotage	Jamming Cars	O	H	H	L	M	M	Securing access to system will be important to prevent tampering
	Human Tampering	Overloading cars unable to traverse tube.	O	H	H	N	H	M	Securing access to system will be important to prevent tampering
	Ground Movement	Ground movement may lead to tubes becoming damaged or knocked out of alignment.	O	H	H	H	H	M	Proper ground support and monitoring will improve system reliability
Control System	Component Failure	Failure of winding to energize will result in loss of power.	O	N	L	L	N	M	Solid State Control System will be Highly Reliable
	Power Failure	Failure of winding to energize will result in loss of power.	O	N	L	L	N	M	Solid State Control System will be Highly Reliable
Car	Spillage	Wear of tube and jamming	O	H	M	M	N	M	Reducing spillage will be important in skip design
	Wheel wear	Wearing wheels reduce clearance increasing risk of car jamming	O	M	M	M	N	M	Material choice and wear monitoring will be important for system success

Project Stage:
D = Design
C = Construction
O = Operation

Likelihood:
N = Not Likely
L = Low
M = Moderate
H = High
E = Extreme

Consequence:
N = Negligible
L = Low
M = Moderate
H = High
E = Extreme

Level of Confidence:
L = Low
M = Moderate
H = High

Water Infiltration	Water shorting winding	O	H	H	N	N	M	Waterproofing winding will be important
Acidic Mine Water	Acidic attacking components	O	M	N	M	N	M	Protective coatings may be required
Body wear	Spilling material may jam cars.	O	M	M	H	N	M	Material choice and wear monitoring will be important for system success
Latch Failure	Opening of lid will lead to spilling material may jam cars.	O	M	M	H	N	M	Material choice and wear monitoring will be important for system success

Project Stage:
D = Design
C = Construction
O = Operation

Likelihood:
N = Not Likely
L = Low
M = Moderate
H = High
E = Extreme

Consequence:
N = Negligible
L = Low
M = Moderate
H = High
E = Extreme

Level of Confidence:
L = Low
M = Moderate
H = High

Appendix XI: Economic Model

		LIM on Track	
Skip Diameter	cm	50	Blue text is input data, feel free to change on any sheet
Skip Length	cm	800	Black text is calculated, don't edit these cells
Required number of Skips		69	Red Text is Capital Costs that are scaled to the power of 0.6
Skip Weight	Kg	1130	
Skip Cost	$	$ 11,300	Scaled on skip volume
Design Speed	m/s	14.8	Scaled from 12 m/s @ 25cm diameter to 20m/s @ 1m diameter
	km/h	53.1	
Maximum Gradiant	deg	90	
	%	na	
LIM Cost			
Length to surface	m	3,000	
Underground Horizontall Length	m	3,000	
Diameter (inside)	cm	52	
Duty	%	4.5%	
Design Force (up loaded)	N	36,699	
	N/cm	46	
Design Force (down empty)	N	(12,078)	
	N/cm	15	
Design Force (horizontal loaded)	N	482	
	N/cm	1	
Design Force (horizontal empty)	N	238	
	N/cm	0	
Installed Lim Cost			
LIM cost (up Loaded)	$/m	$ 3,060	
LIM cost (down empty)	$/m	$ 2,367	
LIM cost (Horizontal Loaded)	$/m	$ 1,742	
LIM cost (Horizontal Loaded)	$/m	$ 1,705	
Operating Efeciency	%	70%	
Regenerating Efficiency	%	20%	
Switching Costs	$/m	$100	
Total Required Power	kW	12,973	
Power per power supply	kW	1,000	
Cost of a Variable Frequency Power Supply	$/unit	$1,000,000	Used to accelerate skips at loading points (1/loading point)
Cost of a Fixed Frequency Power Supply	$/unit	$500,000	

		LIM on Track
Vertical Depth	m	3,000
Horizontall	m	3,000
Capacity	tonnes/day	12,000
Available hours per day	hrs	18
System Productivity	tonnes/hr	667
Operating Days per Year	days	355
Annual Capacity	tonnes/year	4,260,000
Ore Density	tonnes/m^3	3
Ore Swell (loaded)	%	60%
Loose Density	tonnes/m^3	1.9
Skip Diameter	cm	50
Skip Length	cm	800
Skip Volume	cm^3	1,506,707
Fill Factor	%	80%
Load Volume	cm^3	1,205,366
Skip Payload	Kg	2260
Skip Weight	Kg	1130
LIM Weight	Kg	n/a
Loaded Weight	Kg	3390
Slope of incline to surface	deg	90
Length to Surface	m	3,000
Underground horizontall distance	m	3,000
Return Distance	m	12,000
Design Speed	m/s	14.75
Acceleration & loading times	s	10
Cycle time	min	13.7
Productivity per Skip	Kg/hr/skip	9,879
Required Operating Skips	#	67
Skip Availability	%	98%
Skip Fleet Size	#	69
Time Between Skips	s	12.0
Utilization	%	4.5%
Loading Time	s	11.6
Required Loading Stations	#	2

		LIM on Track			
		Loaded		Empty	
		Level	to Surface	Level	to Surface
Grade	deg	0	90	0	-90
Skip Diameter	cm	50	50	50	50
Skip Length	cm	800	800	800	800
Total Skip Weight	kg	3,390	3,390	1,130	1,130
Average Speed	m/s	15	15	15	15
Normal Force	N	33,257	0	11,086	0
Rolling Resistance (1%)	N	333	0	111	0
Gravitational Force	N	-	33,257	-	(11,086)
Drag	N	106	106	106	106
Normal Operating Force	**N**	**438**	**33,363**	**217**	**(10,980)**
Overdesign Factor	%	10%	10%	10%	10%
Overdesign Force	N	44	3,336	22	(1,098)
Design Force	**N**	**482**	**36,699**	**238**	**(12,078)**

		LIM on Track	
Skip Weight	Kg	1130	
Skip Payload	Kg	2260	
Total Weight	Kg	3390	
Operating Force (up loaded)	N	36,699	
Operating Force (down empty)	N	(12,078)	
Operating Force (horizontal loaded)	N	968	
Operating Force (horizontal empty)	N	968	
Distance (up loaded)	m	3000	
Distance (down empty)	m	3000	
Distance (horizontal loaded)	m	3000	
Distance (horizontal empty)	m	3000	
Work (up loaded)	kJ	110,097	work = force * distance
Work (down empty)	kJ	(36,233)	
Work (horizontal loaded)	kJ	2,903	
Work (horizontal empty)	kJ	2,903	
Efficiency (up loaded)	%	70%	
Efficiency (down empty)	%	20%	
Efficiency (horizontal loaded)	%	70%	
Efficiency (horizontal empty)	%	70%	
Energy Consumed (up loaded)	kJ	157,281	
Energy Consumed (down empty)	kJ	(7,247)	
Energy Consumed (horizontal loaded)	kJ	4,148	
Energy Consumed (horizontal empty)	kJ	4,148	
Total	**kJ**	**158,330**	
Energy consumed per tonne of ore	kJ/kg	70	
Tonnes moved per day	tonnes	12,000	
Total energy consumed per day	MJ	840,670	
Time period that energy is consumed per day	hours	18	
Average Required Power	kW	12,973	
Power Consumed per day	**kW Hrs**	**233,519**	

		LIM on Track	
Capital Costs			
Number of Skips	#	69	
Cost / Skip	$	$ 11,300	
Total Skip Cost	**$**	**$ 779,721**	
Length of LIM (up loaded)	m	3,000	
Length of LIM (down empty)	m	3,000	
Length of LIM (horizontal loaded)	m	3,000	
Length of LIM (horizontal empty)	m	3,000	
Cost of LIM (up loaded)	$/m	$ 3,060	
Cost of LIM (down empty)	$/m	$ 2,367	
Cost of LIM (horizontal loaded)	$/m	$ 1,742	
Cost of LIM (horizontal empty)	$/m	$ 1,705	
Cost of LIM (up loaded)	$	$ 9,180,506	
Cost of LIM (down empty)	$	$ 7,100,795	
Cost of LIM (horizontal loaded)	$	$ 5,224,826	
Cost of LIM (horizontal empty)	$	$ 5,115,318	
Total	**$**	**$ 26,621,446**	
Length of Rail Required	m	$ 36,000	
Ceramic Rail Cost	$/m	$ -	
Total	**$**	**$0**	
Total Length of LIM	m	12,000	
Switching Cost	$/m	$100	
Total	**$**	**$1,200,000**	
Total Power Required	kW	12,973	
Power per power supply	kW	1,000	
Required number of power supplies	#	14	
Number of Variable Frequency Supplies	#	2	Used to accelerate skips at loading facilities
Cost of Variable Frequency Supplies	$	$ 1,000,000	
Total		**$ 2,000,000**	
Number of Fixed Frequency Supplies	#	13	
Cost of Fixed Frequency Supplies	$	$500,000	
Total		**$ 6,500,000**	
Loading Station Costs			
Number of Loading Stations	#	2	
Crusher, 24" x 48" roll, 100 hp	$	$ 276,000	
Bins & conveyors & Misc	$	$ 552,000	
Total		**$ 1,656,000**	
Dumping Station Costs			
Number of Dumping Stations	#	1	
Cost per Station		$ 1,000,000	
Total		**$ 1,000,000**	
Grand Total		**$ 39,757,166**	

			LIM on Track	
Annual power consumption	MWh		82,899	
Power Cost	$/kWh	$	0.05	
Annual Power Cost	**$/year**	**$**	**4,144,969**	
Car Life	cycles		150,000	
Replacement Cost	$	$	11,300	
Total Cycles per Year	#		1,884,905	
Cars replaced per year	#		13	
Total Annual Cost	**$**		**142,000**	
Liner Life	cycles		30,000	
Replacement Cost	$	$	1,425	Calculated on Linear area raised to the 0.6 power
Replacements per year	#		50	
Total Annual Cost	**$**	**$**	**71,645**	
Lim Capital Cost	$	$	26,621,446	
% Repaired/replaced per year	%		2%	
Total Annual Cost		**$**	**532,429**	
Total Operating Cost			**4,891,042**	
Operating Cost per tonne			**1.15**	