

California AHMCT Research Center  
University of California at Davis  
California Department of Transportation

**DEVELOPMENT OF A  
PROTOTYPE  
AUTOMATED CONE MACHINE  
AND  
A HIGH CAPACITY STORAGE SYSTEM**

Michael B. Cline  
Cornelis J. Belltawn  
John B. Mcleod  
Wilderich A. White  
Steven A. Velinsky

AHMCT Research Report  
UCD-ARR-99-06-30-07

Final Report of Contract  
RTA 65A0019

June 30,1999

This report has been prepared in cooperation with the State of California, Business and Transportation Agency, Department of Transportation and is based on work supported by Contract Number RTA 65A0019 through the Advanced Highway Maintenance and Construction Technology Research Center at the University of California at Davis.



## ABSTRACT

The Advanced Highway Maintenance and Construction Technology (AHMCT) Research Center has been developing robotic equipment and machinery for highway maintenance and construction operations. It is a cooperative venture between the University of California at Davis and the California Department of Transportation (Caltrans). The research and development projects have the goal of increasing safety and efficiency of roadwork operations through the appropriate application of automation solutions. This report describes the continuing development of automated equipment for deploying and retrieving traffic cones.

In this latest phase of the project, the center has created a fully functional automated cone machine (ACM) prototype which is being used in tests and demonstrations on the highways of California. Operators using the ACM can place and retrieve cones without any set up and control the machine from within the confines of the cab. The machine can easily be run by a single operator and is very compatible with the process of closing a lane. As part of this development, second generation components of two critical components, the retrieval arm and the stowage system were accomplished and are described.

Concurrent with development of the ACM prototype, a means of adding to the cone carrying capacity has been defined and a demonstration unit was produced. This concept can be readily integrated into the ACM and will allow it to carry a large numbers of cones for the less common long lane closures requiring 250 or more cones. These machines are an unambiguous demonstration of the successful application of automation in a very demanding environment.



## EXECUTIVE SUMMARY

The Advanced Highway Maintenance and Construction Technology (AHMCT) Research Center has been developing robotic equipment and machinery for highway maintenance and construction operations. This report describes the continuing development of automated equipment for deploying and retrieving traffic cones.

The need for mechanizing the cone handling process is well established throughout the world where automobiles are integral to society. Rising standards of living lead to higher standards of worker safety and tasks such as traffic cone handling require serious attempts to remove the human from unnecessary exposure to danger. Handling the traffic cones is physically demanding and very unsafe. Since accidents do occur and the work must be accomplished, developing methods by which tasks can be achieved from within the relative safety of a vehicle is an obvious solution. This requires the application of mechanization and automation technology in order to give personnel a modicum of protection comparable to what the traveling public receives. By effective use of efficient and safe machines, the danger to crews and the public will be reduced.

There have been several significant attempts at developing methods to assist in the cone laying process. None of them have succeeded in the market and cannot meet the needs of Caltrans.

Initially the cone handling process appears to be ideally suited to mechanization. Never the less, it is very difficult to compete with the capabilities of the human cone handler in the current Caltrans cone truck. Cones are often damaged, knocked over, and coated with grit and tar. In the heat they are very gummy and flexible while at colder temperatures they are hard and almost brittle. An automated machine has to be very well designed to deal reliably with these variables. Due to the different cone laying situations and the inevitability of equipment failures, the ability to enable a manual operating mode is considered critical. The closure has to be put out before the work can begin. Dealing with a large bulky machine that is mal-functioning will not be acceptable to the crew.

By working directly with Caltrans crews, the real world requirements were made clear and effected the selection of an ACM concept that clearly would meet the needs of the crews. By creating the working prototype that is described, the intent has been to readily demonstrate its effectiveness to persons who would want to use the machine and those that might commercialize it.

With the single layer of cones, the ACM prototype machine is able to handle the vast majority of maintenance cone closures. Given the success of this concept, the addition of a means to extend the capacity of the machine was a natural development.

The multistack system layout is characterized by horizontally oriented cone stocks, which are stored in multiple, vertical layers. The system configuration is consistent with current methods for storing cones on the ACM and manually operated cone trucks. A forklift unit design was chosen to raise and lower the cone stacks within the storage framework. Successful integration

and operation of the entire system can be mostly attributed to the simplicity of the forklift design. It effectively handles cone stacks and supports the reconfigured main conveyor.

In this latest phase of the project, the center has created a fully functional automated cone machine (ACM) prototype which is being used in tests and demonstrations on the highways of California. Operators using the ACM can place and retrieve cones without any set up and control the machine from within the confines of the cab. The machine can easily be run by a single operator and is very compatible with the process of closing a lane. The center has also produced a demonstration unit that enables the ACM to handle a very large numbers of cones for the less common long lane closures. This machine is an unambiguous demonstration of the successful application of automation in a very demanding environment.

The Automated Cone Machine design is a very viable design and should be used as the basis for a cone machine that can be marketed. Testing and demonstrations of the ACM can continue to be done by AHMCT, Caltrans and others to refine the design. The prototype is being used to identify weaknesses in mechanisms that will greatly assist in development of a robust commercial unit. This development effort is of high value to Caltrans who could manufacture the machine in-house if necessary. It is expected that commercialization of the machine will occur due to the ubiquitous use of the traffic cone and the success of this machine.

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## DISCLAIMER / DISCLOSURE

The research reported herein was performed as part of the Advanced Highway Maintenance and Construction Technology (AHMCT) Program, within the Department of Mechanical and Aeronautical Engineering at the University of California, Davis and the New Technology and Research Program of the California Department of Transportation. It is evolutionary and voluntary. It is a cooperative venture of local, state and federal governments and universities.

The contents of this report reflect the view of the author(s) who is (are) responsible for the facts and accuracy of the data presented herein. The contents do not necessarily reflect the official views of the STATE OF CALIFORNIA or the FEDERAL HIGHWAY ADMINISTRATION and the UNIVERSITY OF CALIFORNIA. This report does not constitute a standard, specification, or regulation.



# CHAPTER 1

## INTRODUCTION

Maintaining the California State highway system is a difficult and ongoing task. The demands of highway maintenance are continually increasing as automotive travel grows. Recent highway statistics show that California motorists drove over 459 billion km (278 billion miles) in 1997, which reflects a 5 percent increase in annual travel since 1994 (Table VM-2M, 1997 Highway Statistics, US Department of Transportation). The California highway infrastructure must remain well-kept to support commercial and personal travel, yet the large volume of traffic poses a great obstacle to achieving this task. In addition to the inconvenience of work amidst heavy traffic, the human risks associated with road work are very high. Strict traffic control regulations are needed to ensure a safe and isolated work zone for maintenance operations. In response to this need and concern, the California Department of Transportation (Caltrans) has developed guidelines and procedures to enforce traffic control.

There are many types of regularly performed operations to build, repair and maintain highways, most of which require traffic control zones in the form of lane closures, road closures, detours or other means of redirecting traffic. Before maintenance work can begin, a temporary boundary of portable traffic markers is created to separate the work area from the open road. Caltrans uses dedicated cone trucks to assist in placing the markers. Though traffic control zones improve the safety and working conditions of the maintenance crews, the procedure for preparing the closure and deploying traffic markers is hazardous. Cone truck operators are exposed to high speed traffic and road debris, which is often kicked up by passing vehicles. To improve the nature of this procedure, an Automated Cone Machine (ACM) has been developed to replace the high risk manual process. The fully operational ACM vehicle is the result of research and development efforts at the Advanced Highway Maintenance and Construction Technology Center (AHMCT) at the University of California – Davis (UC-Davis).

The AHMCT Center represents a unique partnership between UC-Davis and Caltrans for the purpose of developing technology for highway maintenance and construction. Research efforts at the center explore methods of automation to improve operation capabilities and efficiency. The objective of the design work and development for each AHMCT project is to improve safety and effectiveness of Caltrans' maintenance operations.

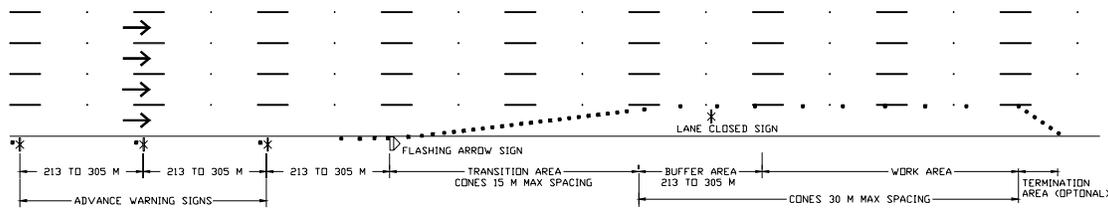
The ACM has become a trademark project at the AHMCT Center because of its importance in supporting all types of highway maintenance and construction operations. A previous report, Development of an Automated Cone Placement and Retrieval Machine, describes the conceptual development of the ACM. This report describes the most recent work in the development of the ACM. Included are the design of second generation components that have been incorporated into the prototype, and the development of a demonstration unit for high volume storage known as the multistacking unit. Caltrans has expressed great interest and satisfaction with the ACM. As the result of the success of the design and the continuing interest in this type of machine by Caltrans and other potential users, serious investigations into its commercial viability are in progress.

The remainder of this chapter describes the typical cone laying process and describes cone handling designs that have been attempted by others. The next chapter describes the ACM as built. Development of the multistack unit is described in the third chapter, while the fourth and fifth chapters describe the second generation retrieval system and the stowage system. Concluding statements are in the final chapter.

## 1.1 Caltrans Operations and Existing Procedures

### 1.1.1 Lane Closure Procedures

For almost all road maintenance performed, a lane closure is first required to allow delineation of the work area and provide the maintenance crew some safety and separation from traffic. Many road work activities are of short duration and use a typical highway cone as a separation barrier. The typical placement of these cones is done in conformance with Chapter 5 of the 1990 Caltrans “Manual of Traffic Control”, and is shown graphically in Figure 1.1.



*Figure 1.1 Typical Cone Placement Configuration*

The procedure calls for an Advanced Warning Area, to allow for driver reaction and provide time for lane correction. The Advance Warning Area is created by the placement of four warning signs before the coned section. These signs are placed from 213m to 305m (700 to 1000 ft) apart. The first three signs warn the public of road construction and specify which lane(s) are closed. The procedure calls for a single cone to be placed next to the base of each of these three signs. The fourth specified sign is a flashing arrow sign which indicates the merge direction. This sign is preceded by the placement of four traffic cones placed at 15m (50ft) intervals.

The Transition Area is created by the placement of cones, spaced at a maximum of 15m (50ft) apart, in a taper configuration to slowly close the lane. It starts at the flashing sign and stops when the far edge of the lane(s) to be closed is reached. The total allowable minimum length of the transition is 305m (1000ft) per lane closed. Furthermore, if more than one lane is closed, additional flashing arrow signs are required for each additional lane closed.

After the Transition Area and prior to the Work Area a Buffer Area is created. This is an additional safety area where no work is to be performed and allows a vehicle to safely come to a complete stop, prior to entering the Work Area, in case of an unintentional or emergency closed lane entrance. This Transition Area is a minimum of 213m to a maximum of 305m (700 to 1000ft) long, and is created by cones spaced at a maximum of 30m (100ft) apart.

The extension of the buffer area creates the actual Work Area, and is started with the placement of a “Lane Closed” sign. For the entire length of the delineated Work Area the cones are also spaced a maximum of 30m (100ft) apart. Additional “Lane Closed” signs are required in the work area at 610m (2000ft) intervals.

Finally an optional Termination Area is created, which consists of a small tapered area at the end of the Work Area. If used, this section of the lane closure will aid redirection of traffic flow back into the previously closed lane. The cones in this section are placed at a maximum spacing of 15m (50ft).

### 1.1.2 Typical Cone Truck

Caltrans has developed its own specialized vehicle for almost all types of cone placement operations. This vehicle is based on a domestic one-ton pickup truck and usually has a single rear axle with dual wheels on each side. Current truck models used have a gross vehicle rating of 6804 kg (15000 lbs), a wheel base of 4.11 m (162 in) and a total length of approximately 6.27 m (247 in).

The Caltrans cone body bed is mounted as shown in Figure 1.2. This cone body consists of a conveyer belt on which cones are stored, two seating areas, and additional storage compartment adjacent to the conveyer belt. The current cost of a cone body retrofit is in the range of \$12,000.

The seating areas are outfitted with reversible seats that normally face rearward. These seating areas are directly behind the cab and are located close to the ground to allow the worker to place the cones on the road.

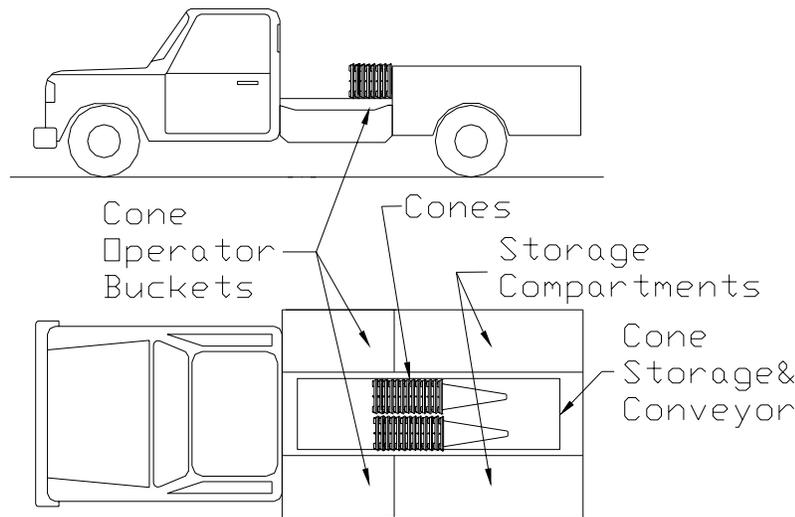


Figure 1.2 Truck Modified for Cone Operations

The cone storage conveyer belt has a capacity of 80 cones in two rows of 40 each stacked in the longitudinal direction shown. A payload of 80 cones is typically enough to close a lane for a length of 1.8km (1.1mi). The conveyer brings the stacks of cones forward to and backward away from the cone operator and is controlled by foot switches in the seating areas.

The storage compartments allow convenient storage of the required road signs, and miscellaneous equipment such as tools and sandbags. If a long lane closure is required, additional cones are stacked on top of the first layer or placed in the storage compartments.

### 1.1.3 Cone Placement and Retrieval Operations

There are two main operations of the cone truck. One is the placement of the cones to create the lane closure. The other is the retrieval of the cones after the completion of the road work. Both placement and retrieval operations require two workers. The driver monitors the traffic, creates the alignment of the cones, and coordinates the operation in conjunction with the shadow truck. The second worker, the cone operator, is positioned in a bucket seat of the cone body and handles the cones during both placement and retrieval using the foot switches to operate the conveyer.

During cone placement, the ACM is driven forward inside the lane being closed, with the truck adjacent to the edge of the lane. The cone handler sits in the appropriate bucket seat so that s/he is located at the edge of the lane closure and facing rearward. The cone operator picks up each cone from the conveyer and sets it on the road directly adjacent to the bucket seat. The spacing of the lane markers assists the cone operator in judging the distance between cones. The speed of this placement operation is usually limited by the cone operator's ability to keep up with cone handling and to prevent the cone from toppling over immediately after placement on the road. Normally cone placement is performed at 18 km/h (10 mph).

The retrieval is usually done in reverse order of cone placement. By backing up, the cone truck is always within the coned off section. The cone truck does not drive against the direction of traffic. The cone operator reaches out and manually picks up each cone to place it back on the conveyer.

The signs are picked up last, after all the cones have been retrieved. Since the retrieval vehicle still faces the direction of traffic, the driver is easily able to merge back into traffic during the departure from the maintenance site.

While operating, the cone truck is almost always followed by a shadow vehicle which protects the cone truck in case a stray vehicle enters the closed lane. This shadow vehicle is equipped with a large energy-absorbing crash attenuator which is usually mounted at the rear. The shadow vehicle is positioned between the cone truck and oncoming traffic.

### 1.1.4 Inherent Shortcomings of the Cone Laying Procedure

The methods currently used offer limited protection to the cone operator. Although the cone bed's bucket seat area is reinforced and provides some protection to the cone operator, it still allows significant exposure. Never the less, the Caltrans design is a tremendous improvement

over the practice of sitting on the tail gate of a truck, which is still done by some private contractors.

On a typical freeway, cone workers are exposed to a large speed differential between their cone truck and the passing traffic. Their only protection is the use of the shadow vehicle which is limited to the longitudinal traffic flow direction. The side of the cone truck is constantly exposed to high-speed passing vehicles. This exposure poses the largest danger to the cone operator. His upper torso is unprotected from any road debris kicked up by passing vehicles. Any vehicles traveling close to the lane closure can strike a traffic cone and make it a hazardous flying object. Even more threatening is the possibility of injury to the cone operator's extended arm. Besides the immediate dangers due to the location of cone placement, cone operators must constantly exert themselves during the repetitive motion of cone operation which often leads to injury. There is no question that automation of this process is essential.

### 1.2 Caltrans Traffic Cones

Traffic cones come in a wide variety of shapes and sizes and are made out of various materials. Even when a traffic cone is specified by height, they vary in the angle of the conical section, total weight and width of the base. When an operation uses cones purchased in different batches, there are potential difficulties in stacking and handling. Caltrans specifies the use of a heavier 4.5 kg (10 lbs) cone that is 71 cm (28 in) tall. These cones are molded from poly-vinyl chloride (PVC) plastic and industry appears to have a de facto standard that minimizes some of the potential stacking problems. The base is the heaviest part, thus keeping the center of gravity low. The conical section is thin walled and the outer shell is colored a highly visible orange. Since the specified cone is heavier than most cones, it resists being blown over in windy conditions or by a close-passing vehicle. The center of gravity is approximately 11.5 cm (4.5 in) from the bottom of the base.

The material properties of these PVC cones vary. Temperature induced changes in material properties are significant and California encounters a wide variety of temperatures, ranging from desert heat to sub-zero high altitude winter conditions. The cones become extremely pliable when hot, such that they can be easily flattened, yet they are brittle and rigid when cold. The cones' properties also vary due to different levels of service endured. Some have asphalt and tar coated surfaces while others have twists, indentations and abrasion from being impacted or run over by highway traffic. New cones are still coated with a mold release compound used in manufacturing which makes the new cones extremely slippery until this compound is worn off. Typically these properties and cone conditions present limited problems for manual operations, but these conditions are problematic for any kind of automated system.

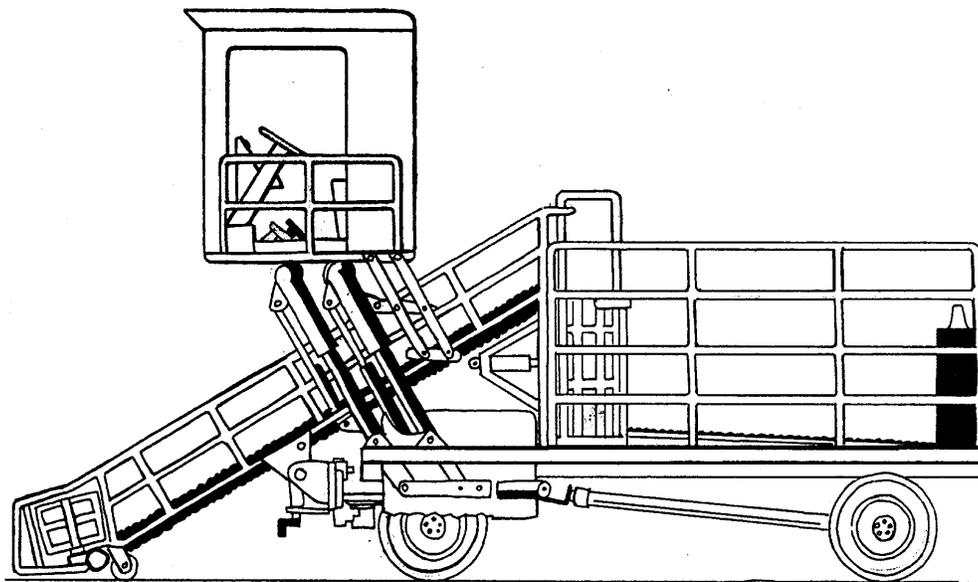
### 1.3 Existing Automated Cone Machines

Prior to the development of the ACM, the AHMCT center researched and investigated the existence of other machines created for the retrieval and placement of traffic cones (Tseng et al, 1997). The mechanisms found were the Traffic Cone Retriever, the French Baliseur Machine, and the Addco Cone Wheel Dispenser and Collector. The Baliseur and the Cone Wheel were commercially available but have not been successful. Caltrans was unable to use these

machines. Continuing patent research conducted during the recent phases of the project has identified some additional concepts. All the concepts are described below.

### 1.3.1 Traffic Cone Retriever

The Traffic Cone Retriever, shown in Figure 1.3, was the first automated cone machine to receive a patent, number 3,750,900, issued in 1973. This machine is rather large and is described operating with a forward speed at up to 56 km/h (35 mph) into a standing cone in order to retrieve it. After the initial contact, the cone enters the revolving paddles which force the cone onto a conveyor that brings it towards the rear of the vehicle. The cone is then stacked vertically in the storage area. This area holds up to 2000 traffic cones.



*Figure 1.3 Traffic Cone Retriever (Fig. 1 of Patent #3,750,900)*

The problem with the Traffic Cone Retriever is its inability to dispense cones. Furthermore, it seems rather bulky and large and is most likely unable to drive to the maintenance site using public roads since it fails today's vehicle safety standards. Other issues never addressed with this device are the retrieval of cones that are knocked over and oriented various directions. This system appears to never have been commercialized.

### 1.3.2 Cone Wheel

The original Cone Wheel Dispenser and Collector, designed and manufactured by the ADDCO company, and shown in Figure 1.4, was issued U.S. patent number 5,054,648, in October of 1991. This wheel is claimed to be capable of retrieving and placing cones at speeds up to 40 km/h (25 mph). However, this operation is not fully automated and still requires a cone operator in the truck bed. To retrieve a cone, the driver maneuvers the truck so that the cone wheel rolls over the cone, which will cause the conical section of the cone to become wedged

between two disks. These two disks together make up the large cone wheel which is approximately 1.2m (4ft) in diameter. The wedged cone is brought up to the top during the rotation of the cone wheel. At the top the cone is stripped of the wheel by a metal bar and left for the cone operator to manually handle and store. The cone placement operation is similar but in opposite sequence of the retrieval. The cone operator will manually set a cone upside down in the top of the wheel. As the wheel rotates, the cone is firmly placed upon the road surface and then stripped loose of the wheel by a metal bar.

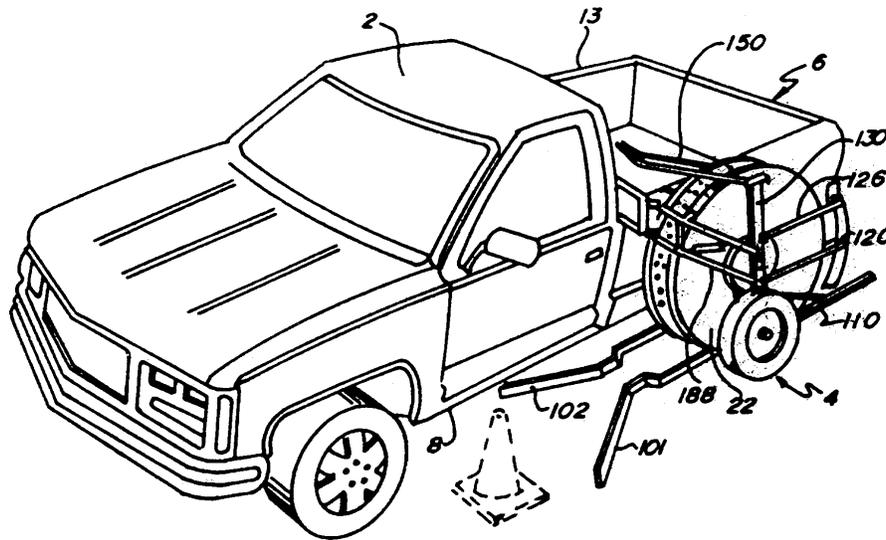


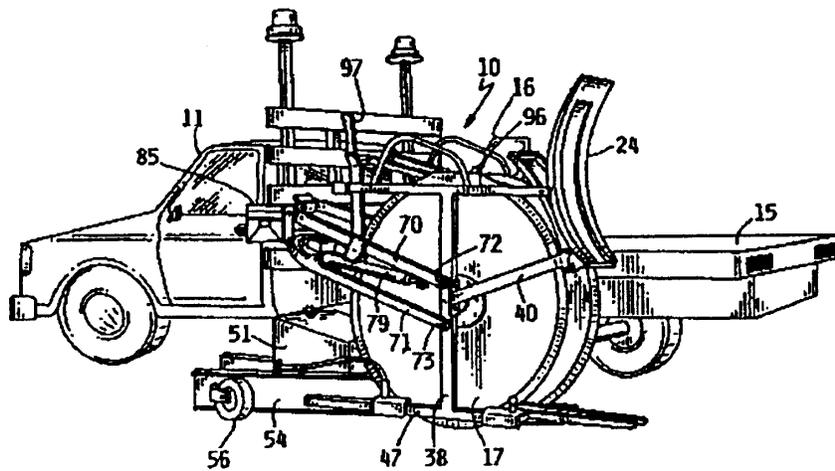
Figure 1.4 Cone Wheel (Fig. 1 of Patent #5,054,648)

Although this system is used in several states and other countries it still has certain significant shortcomings. One problem with this system is the setup of the cone wheel and various attachments. The entire wheel is stored in the truck bed during transport to the maintenance site and requires significant manual force to deploy the wheel to the side of the truck. The operational shortcomings are its inability to retrieve cones while traveling in reverse, or cones that are tipped over with the point facing away from the cone wheel.

The updated patent on the Cone Wheel, number 5,213,464 issued in 1993, discusses changes in operation and covers some of the above described shortcomings. The updated Cone Wheel shown in Figure 1.5 has a slightly slower operational speed of 32 km/h (20 mph). The main difference in operation is the way in which the cones are grabbed, in that the newer version will squeeze the base of the cones between the two disks of the wheel rather than the conical section. This new retrieval operation causes significant bending and deflection of the cone base during the squeeze time in the wheel. The inside of the wheel disks have scoop-shaped guides to engage and confine the cone being squeezed. The cone is lifted upwards during the wheel's rotation. During this upward rotation the squeeze on the base is loosened to allow the cone to rotate so that the tip of the cone points down. Similar to the old model, the cones arriving at the top are stripped from the wheel by a stripper bar and are left to be manually handled and stored. The cone placement operation with the updated version of the Cone Wheel is very similar. The

only significant change is the addition of a loading magazine at the top to aid the cone operator with the loading of cones into the wheel.

One significant overall change is the addition of hydraulic cylinders which can either lower the wheel from the truck bed at the beginning of work or lift the wheel onto the truck bed at the conclusion of work, thus eliminating the strenuous physical work previously required. The redesigned wheel also facilitates cone retrieval in the reverse direction after the entire wheel has been completely disconnected and remounted in reverse. If wheel mounting brackets are installed on both sides of the truck the wheel is versatile enough to allow mounting on either side. These combinations allow for cone operation on both sides of the truck and in either forward or reverse direction, but require a significant amount of changeover. The only unclear aspect of this newer version is whether the cones are always properly oriented for retrieval by the cone guides or need manual intervention when the cone is tipped over and pointing toward the cone wheel.



*Figure 1.5 Cone Wheel II (Fig. 1 from Patent #5,213,464)*

Although significant improvements were made to the Cone Wheel it still has several undesirable characteristics. First, a cone operator is still required in the truck bed, and s/he is left open to the associated dangers. The switch from cone placement to retrieval or visa-versa requires manual changes. The cone stabilizers need to be installed or removed, and the scoop-shaped guides on the inside of the wheel disks need to be tilted up or down. Also, in order to operate the truck in the reverse direction, the whole wheel assembly needs to be disconnected and remounted in reverse. This apparatus also causes undue stress to the cone base during the retrieval operation and could wear out the cones prematurely. Manual intervention is probably still needed to grab some tipped over cones.

Overall, the Cone Wheel operation is not an improvement over Caltrans' current cone truck. The cone operator is still outside the confines and safety of the truck cab and has to manually handle the cones. During the setup of the wheel, usually done directly adjacent to traffic, the

cone workers are outside the truck and highly exposed. Marketing of this machine has not been successful and it is apparently no longer manufactured.

### 1.3.3 The Baliseur Cone Picker

This Baliseur Cone Picker, made by the French company SEP in 1986, is shown in Figures 1.6, & 1.7 and holds U.S. patent number 4,597,706. This machine is described as retrieving all cones, standing up or tipped over, at a speed of 18 km/h (11 mph). Just prior to retrieval, the cones are tipped and oriented so that the bottom of the cone faces the retriever. They are then lifted up the chain link conveyer with a rod that inserts into the bottom opening of the cone. Once the cones reach the top of the conveyor, they are stored in one of the 10 revolving vertical cylinders, which can hold 24 cones each. In the placement operation, each cone is dispensed from the bottom of these cylinders and is then stabilized and guided to its desired lateral position on the road by flexible bristles.

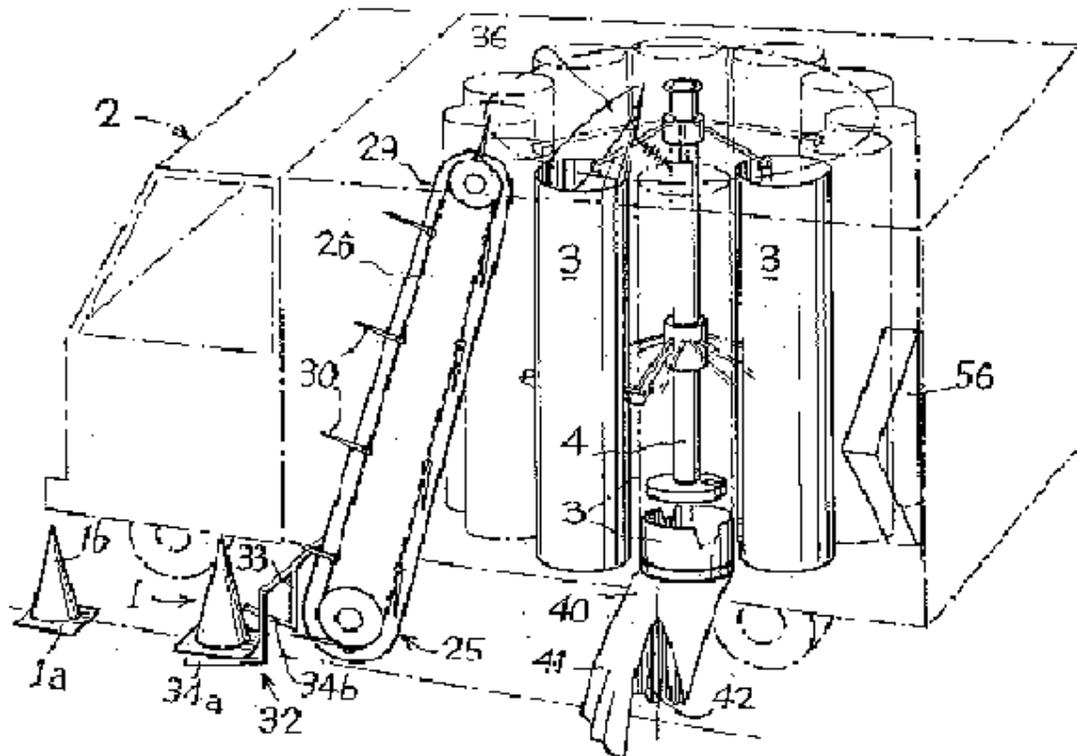


Figure 1.6 Baliseur Machine (Figure 1, US Patent 4,597,706)



Figure 1.7 Baliseur Cone Machine (from SEP brochure)

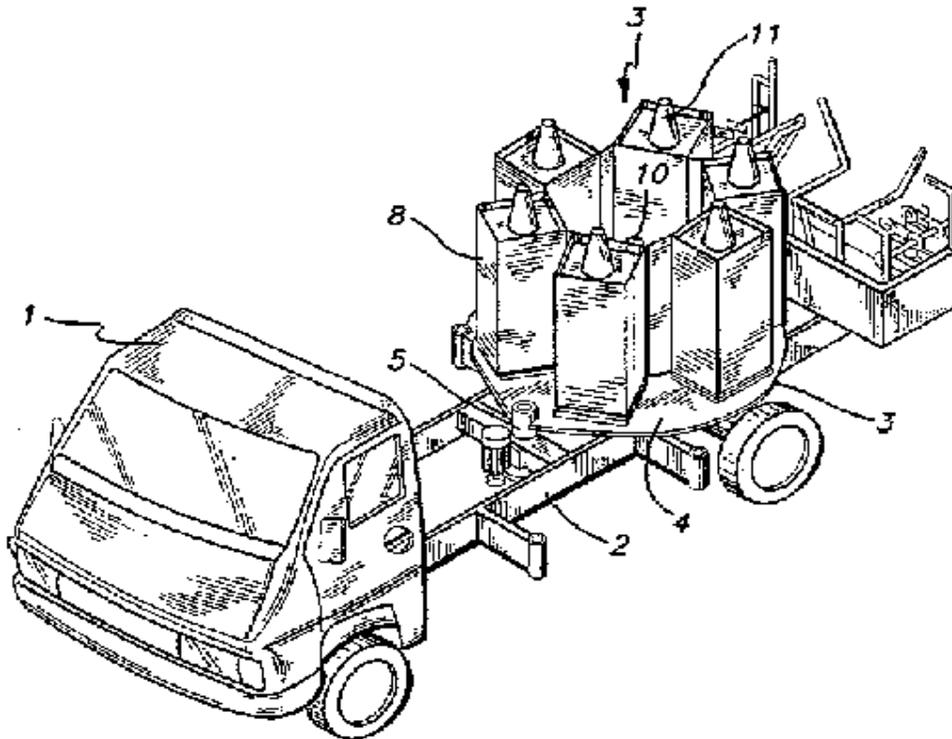


Figure 1.8 Baltic Ingenierie Cone Vehicle (Figure 1, U.S. Patent 5,525,021)

Another variation of this machine is shown in Figure 1.8 as patented by Baltic Ingenierie. These machines have the advantages of full automation but require a unique cone design, are very bulky and expensive, and cannot pick cones in both directions. A machine of this type is not compatible with typical cone laying operations that require a nimble easy to maneuver machine that can pick cones in both directions including those that are knocked over. This machine is not acceptable for Caltrans operations.

### 1.3.4 Toyota Cone Machine

A more recent system has been developed by Toyota (Figure 1.9). It uses a conveyor unit similar in layout and function to the main conveyor on the Caltrans cone truck. Short, vertical cone stacks are stored side to side along the entire length of the conveyor. The cones are stacked at the back end of the vehicle by mechanisms that retrieve the cones from the road. In discussions with parties interested in cone machines for use in Japan, this machine and the Addco machine have also not been successful in the Japanese market. It has similar characteristics to the Baliseur machine and would not meet Caltrans' needs either.

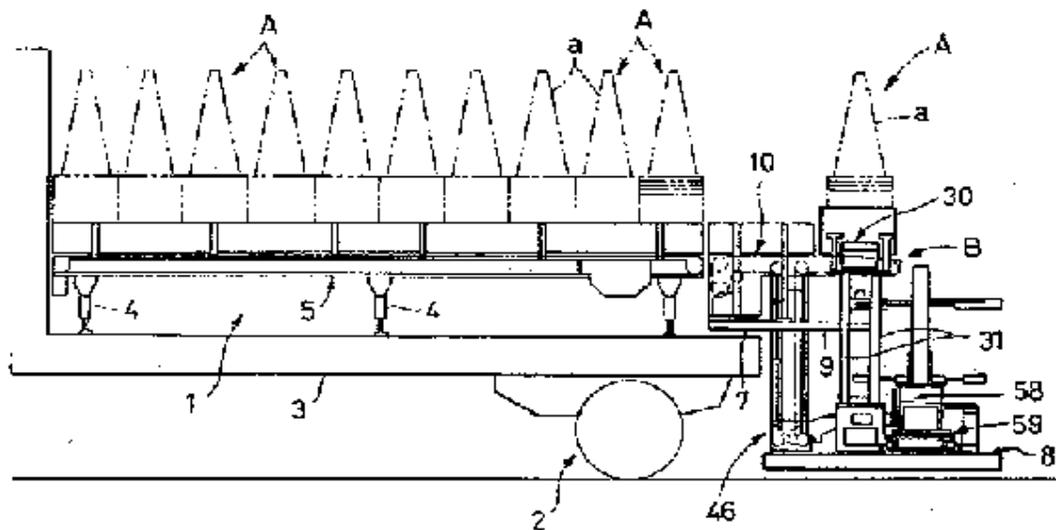


Figure 1.9 Toyota Cone Machine (Figure 1, U.S. Patent 5,244,334)

## 1.4 The AHMCT Automated Cone Machine Solution

### 1.4.1 Challenges to Automation in the Lane Closure Operation

Obviously there is a need for automation of the cone laying process as several serious attempts have been made to meet this need. The AHMCT ACM design is the first design that meets the needs of the personnel that are responsible for typical lane closures.

This typical lane closure operation is a relatively quick process spanning less than 20 minutes and using less than 80 cones. Various warning signs, trailered flashing signs and other

components of the closure have to be supported. Personnel have to get out onto the pavement to deal with these aspects of the closure and machinery cannot interfere with these activities.

Closures are required in a variety of difficult traffic situations and the actual placement of the cones is not the most important aspect of the operation. Moving traffic over, especially on a heavily traveled metropolitan road, is a very serious task that requires strategy, coordination and concentration. The driver is extremely involved in controlling this aspect of the activity and does not want to be dealing with a machine that limits him in any way. Machinery that handles the cones has to be very easy to use.

Although handling the cones is physically demanding and unsafe, it is very repetitive and simple. Initially the cone handling process appears to be ideally suited to mechanization. Never the less, it is very difficult to compete with capabilities of the human cone handler in the current Caltrans cone truck. Cones are often damaged, knocked over, and coated with grit and tar. In the heat they are very gummy and flexible while at colder temperatures they are hard and almost brittle. An automated machine has to be very well designed to deal reliably with these variables. The ability to enable a manual operating mode is very important since hardware failures are inevitable.

Automated machinery cannot compromise the current operations. The machine has to conform to all the typical design criteria such as reliability, durability, maintainability, and low cost. The workers must remain within the confines of the cab to limit their exposure and work site set up of the machine should not be required. The machine should be capable of operation by the driver alone and through a simple operator interface. The ability to retrofit existing Caltrans cone trucks is potentially a cost advantage. The machine has to be very reliable and operate in the dusty, wet, cold and hot conditions of California and the world. It must be capable of operating on an unimproved shoulder and function on typical road surfaces. When traveling to and from the work site the vehicle must be a fully functional truck with a minimum of features that interfere with driving across medians or in construction sites. It has to carry at least eighty regular traffic cones and be compatible with a means to increase this capacity to 200 cones or more. If the system breaks down, it should be possible to handle the cones manually. The machine should be able to dispense cones from either side of the truck while driving forward. It must retrieve cones while traveling forward or backward and from either side of the truck without any manual set up. It must also be able to retrieve any cone that has been knocked over without manual intervention. It should operate at speeds comparable to manual operations, about 18 km/h (10 mph).

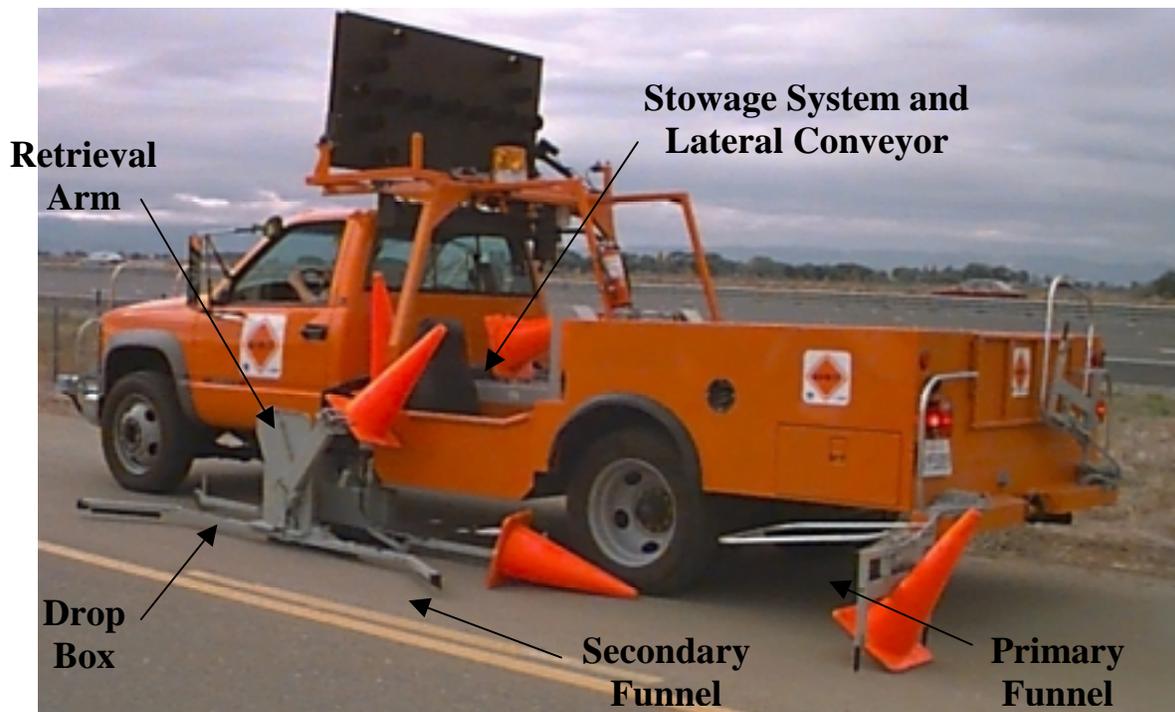
#### 1.4.2 The Real World Solution

The ACM developed at the AHMCT center was designed to meet the real world challenges described above. A fully functional ACM prototype is operating and is being used to demonstrate the best solution to the cone laying challenge. This machine and a modification to add very large quantities of cones are described in the following chapters.

## CHAPTER 2

### FIRST INTEGRATED AUTOMATED CONE MACHINE

Continuing development of the automated cone machine (ACM) has progressed to the design and assembly of the first integrated prototype cone machine at the AHMCT center. This machine is based on the previous work which resulted in a test bed automated cone machine on which first generation components were installed. The test bed supported road testing that enabled development of these components, but it could not adequately support testing by crews on the highways. In this latest phase, second generation components were integrated on a new 'road worthy' truck and the resulting machine is what is known as the AHMCT automated cone machine. It is a fully functional system capable of supporting regular demonstrations and road tests by Caltrans crews and other potential users. This unit does not include the multistacking capability and therefore carries a single layer of cones. The following sections describe the machine as built and it is shown in the following figure.



*Figure 2.1 The AHMCT Automated Cone Machine*

#### 2.1 General Description

The ACM was intended to be compatible with Caltrans cone laying operations and, as a result, it has been designed around the Caltrans cone body truck which is an effective machine for manual cone laying operations. Caltrans has in prior years installed their cone body bed onto trucks with a GVW of about 4500kg (10000 lb), but is now using vehicles with a GVW of 6800kg (15000 lb). This has been required because a fully loaded cone truck can exceed the weight limit of the smaller truck. The vehicle used for the ACM is a 1996 GMC HD 3500.

A slightly modified cone body bed was fabricated and mounted onto the truck frame. In order to facilitate the placement of the automated machinery the cone bed is moved back from the cab 43 cm (17 in), as shown in Figure 2.2. The main conveyor is shortened to open the space between the buckets. Several other changes are incorporated to accommodate the shifted bed. Clearance for the leaf spring shackles is required and infringes into the bucket area.



*Figure 2.2 Cone Truck with Rearward Moved Cone Body*

The ACM has nine major subsystems which are presented in the order encountered by a cone traveling the path from the conveyor storage to the road. Then, the additional subsystems encountered only during the retrieval from the road are presented, and this chapter concludes with the control and power systems

The ACM achieves placement and retrieval of cones through the coordinated operation of its four primary subsystem units: the stowage system, the lateral conveyor, the drop box assembly and the primary and secondary funnels. Each subsystem is identified in Figure 2.1. The stowage system is located directly in front of the main conveyor between the buckets. It consists of two kinematically linked gripper arms, one dedicated to each cone stack. The grippers insert and remove cones, one at a time, to and from the stacks. The arms operate out of phase, moving back and forth between the front end of the cone stacks and the lateral conveyor. During drop off mode, the arms translate horizontally to pull a cone from the stack then follow a path of curvature to rotate the cone into a vertical, upright orientation before placing it on the lateral conveyor. The reverse sequence of cone manipulation is performed during pick up mode.

The lateral conveyor mounts inside the gap between the truck cab and the buckets and it spans the width of the cone body. The lateral conveyor consists of a series of lightweight belts and pulleys, which convey one cone at a time. Cones are placed onto the lateral conveyor by the stowage system gripper arms in an upright orientation and shuffled laterally to the left or the right, depending upon which side cones are being deployed. Likewise, the lateral conveyor returns cones to the middle of the cone body to be stacked by the stowage system during cone retrieval.

The drop box assembly is a retractable unit that deploys outward from the lateral conveyor during ACM operation. It is stowed behind the cab during normal driving operation of the

vehicle. The drop box assists in stabilizing cones during drop off mode. Once the lateral conveyor moves a cone out toward the side of the cone body, the cone falls through the drop box. The drop box helps preserve the upright orientation of the cone as it is placed on the road. The drop box also houses and supports the retrieval arm, which picks up cones off the road. The retrieval arm is activated during pick up mode only. The arm has a hand, which grasps the base of the cone and rotates it up and onto the lateral conveyor.

The primary and secondary funnels are also used exclusively during cone retrieval. The secondary funnel is mechanically linked to the drop box. It opens when the drop box is deployed and retracts when the drop box is stowed. The primary funnel is independently actuated. Together, the funnels orient each cone to be picked up. It is certain the cone will be properly positioned after passing through the two funnels. The cone must be tipped over with the base end facing the drop box in order for the retrieval arm hand to grasp the cone and lift it from the road.

The ACM is highly versatile for use in traffic control operations. Right and left drop boxes make cone deployment possible from both sides of the vehicle. Four sets of funnels, on the front and back of each side of the ACM, allow for cone retrieval in the forward or reverse direction on either side of the truck. Overall, the ACM demonstrates a high level of successful operation as a result of extensive fine tuning, testing and operation. It effectively achieves the same results as the manual operation procedure, while increasing safety by removing the worker from the bucket of the cone body.

Although there is great potential for using the ACM to support highway maintenance, the low cone capacity of the cone body was identified as a significant shortcoming. For many special maintenance operations neither the ACM, nor the standard cone truck, hold enough cones. For this reason, serious consideration was given to increasing the cone load capacity of the trucks. It was believed that an automated cone storage system would offer an increase in cone capacity that would expand the ACM capabilities and broaden its potential for traffic control operations.

### 2.1.1 Main Conveyor Belt System

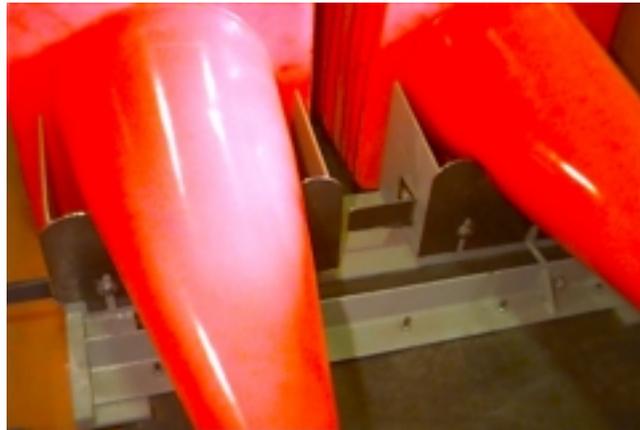
As on the Caltrans cone body, the ACM is equipped with a longitudinal conveyor belt running down the center of the bed. It supports the horizontal stacks of cones and in manual operations, the belt is activated by the operator and moves the stowed cones within his reach. On the ACM, this belt assembly is shortened to accommodate the stowage system and is a component of the main conveyor belt system. The cones are stowed in two adjacent stacks on top of the conveyor belt.

Four modifications to the manual conveyor belt system are added. First, the length of the belt was shortened to allow the cone to be rotated from an upright standing position to the horizontal storage position or vice versa. Three components are added, the cone support fixture, the photoelectric sensor mounts, and the lateral cone guides. A view of this system from the rear of the truck is shown in Figure 2.3.



*Figure 2.3 Main Conveyor Belt System*

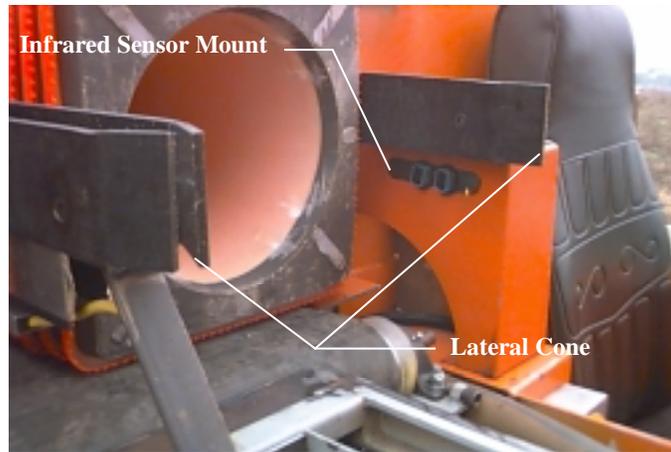
The cone support fixture is mounted on top of the belt and is shown in Figure 2.4. The purpose of the cone support fixture is to keep all the cones lined up in the longitudinal direction and keep the cone base plane perpendicular to the belt surface. This is accomplished by holding the first cone in each stack in the correct alignment. This alignment is required for proper interfacing with the stowage system.



*Figure 2.4 Cone Support Fixture*

Both of the photoelectric sensor mounts are located at the forward end of the main conveyor belt system as shown in Figure 2.5. The photoelectric sensors monitor the position of the first cone in each stack and are used to coordinate the transfer of the cones between the main conveyor and the stowage system. During retrieval, the cone stacks are moved one cone base thickness back to allow a cone to be added to the stack. During deployment the cones are moved forward to the end of the main conveyor belt system so that the stowage system can remove the cones.

Since both cone stacks are on the same belt, each operation mode alternates between the two stacks.



*Figure 2.5 Infrared Sensor Mounts and Lateral Cone Guides*

The lateral cone guides align the cones laterally on the main conveyor belt system to facilitate proper interfacing with the stowage system. These guides are composed of black Ultra High Molecular Weight (UHMW) Polyethylene formed to guide the cones to the proper lateral position and are mounted near the centerline of the cone as shown in Figure 2.5.

The main conveyor belt comes equipped with its own electro-hydraulic power system and is shown in Figure 2.6. This system uses a 12 Volt, Direct Current (DC) motor that is directly coupled to a hydraulic power unit with an attached switching manifold. The manifold directs fluid to a hydraulic motor that drives the rear roller of the main conveyor belt. This is part of the standard cone body configuration and, since the loads on the system were not increased, no changes to the system were required. This power system is left intact to allow the standard manual belt operation in case the automated system is off and the operators are using the manual operating mode. Additional electrical circuitry is added to allow ACM control of this unit.



*Figure 2.6 Main Conveyor Belt Power System*

## 2.2 Stowage System

Centrally located between the two cone operator buckets is the stowage system as shown in Figure 2.7. The stowage system provides a cone transport link between the lateral conveyor belt system and the main conveyor belt system. This requires the stowage system to rotate a cone 90° since the cone is in an upright position on the lateral conveyor belt and is stored in a horizontal position on the main conveyor belt system.



*Figure 2.7 Stowage System*

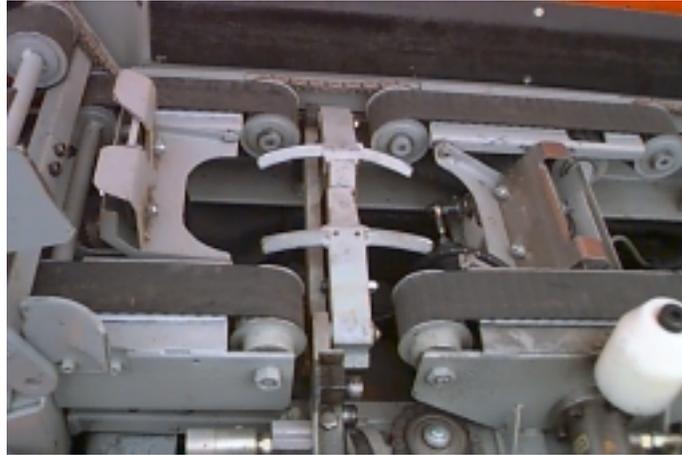
During cone transport by the stowage system the cone is firmly grabbed on the inside of the conical section with a set of expanding grippers. These grippers are pivot-mounted on an arm and linked by a double acting hydraulic cylinder. This cylinder's action opens and closes the grippers as is shown in Figure 2.8.



*Figure 2.8 Open and Closed Grippers*

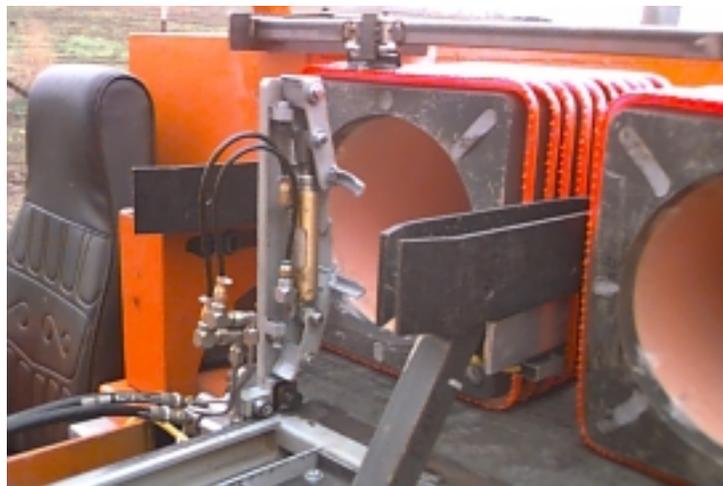
The whole gripper arm assembly is pivoted on a roller assembly. This roller assembly moves linearly along a track to facilitate movement between the main conveyor belt and the lateral conveyor belt. Mounted above the track is a contoured surface that controls the pivot motion of

the gripper arm assembly. This surface forces the gripper arm assembly to rotate to a horizontal position at the lateral conveyor belt and to a vertical position at the main conveyor belt system. When the gripper arm assembly is at the lateral conveyor belt, it is located below the lateral conveyor belt surfaces so that the cone can be moved over the top of the grippers as shown in Figure 2.9. At this point, the grippers can either grip the cone from underneath as during the retrieval mode or release a cone that has just been placed on the lateral conveyor belt as during the dispatch mode.



*Figure 2.9 Gripper Arm Assembly at Lateral Conveyor Belt*

At the other end of the track, the gripper arm assembly is vertical at the bottom of the cone stack on the main conveyor belt system as shown in Figure 2.10. The gripper arm assembly at this location can again either grip a cone to be dispatched or release a cone that has been placed in the stack.



*Figure 2.10 Gripper Assembly at Main Conveyor Belt System*

Since there are two different stacks of cones on the main conveyor belt system, a gripper arm assembly with its track was manufactured for each stack. As previously described the cone stacks must be accessed alternately. The stowage system was designed so that this alternating access motion is accomplished with the power of a single hydraulic vane motor. The gripper arm assemblies are linked via a single chain so that when one arm is at the lateral conveyor belt the other is located at the main conveyor belt system. To exchange the gripper arm positions, the hydraulic fluid flow to the vane motor is reversed and the grippers will be moved to the other end of their track. This combination of components allows for efficient and convenient cone operation between the two systems. A cone moving on the stowage system while in transition from upright to horizontal position is shown in Figure 2.11.



*Figure 2.11 Stowage System in Operation*

### 2.3 Lateral Conveyor Belt

The gap created by moving the cone body back (as previously shown in Figure 2.2) is occupied by the lateral conveyor belt system. This system spans the entire width of the truck and is responsible for the lateral motion of cones. The lateral conveyor belt interfaces with the stowage system in the middle of its length and terminates at the drop boxes on both ends. This is shown in Figure 2.12.



*Figure 2.12 Lateral Belt System*

The lateral conveyor belt is comprised of a total of ten notched groove belts, each 5 cm (2 in) in width. The belts are spaced so that they will contact the frontal and rear edges, usually the feet, of a cone placed on the system. The frontal and rear tracks always move at the same rate and each equally support the weight of the cone. Any cone moving on the lateral conveyor belt is guided on both sides. The frontal guide spans the entire length of the lateral conveyor belt, while the rear guide is interrupted to allow for interfacing with the stowage system.

The lateral conveyor belt operates in both modes of cone operation. During the dispatch mode, the cone is placed on the belts by the stowage system. After the stowage system gripper releases the cone, the lateral conveyor belt then moves the cone to the drop box at the operating side of the truck. During the retrieval mode the cone slides from the retrieval arm to the lateral conveyor belt which has its belts in motion to receive the moving cone. The cone is transported and then positioned on top of one of the two gripper arm assemblies of the stowage system. To stop the cone over the gripper, the lateral conveyor belt has gates with switches that sense the cone's lateral position. These gates retract beneath the lateral conveyor belt and position the cone over alternating gripper assemblies as required.

Also part of the lateral conveyor belt, there are two rotating sections, called 'wings', one located at each end of the lateral conveyor belt. The wing sections are lifted up or positioned down depending on the operating mode. During the cone dispatch mode the wing sections decline at a 30° angle to bring the cone as close to the ground as possible just prior to being positioned into a drop box. In the retrieval mode, the wing section is raised up to ensure correct placement of the retrieved cone onto the lateral conveyor belt. A small hydraulic cylinder positions the wing section. Figure 2.13 shows a cone at the transition to the wing section.



*Figure 2.13 Cone Positioned on Wing Section*

The lateral conveyor belt is powered by a single hydraulic rotary motor which rotates one main shaft of the lateral conveyor belt. Since all the belts are notched and roll over matching notched pulleys, with the front and rear pulleys connected by shafts, the entire set of ten belts rotates in synch and at the same speed. The speed of a traversing cone on the lateral conveyor is set at approximately 0.6 m/s (2.0 ft/s).

#### 2.4 Drop Boxes

Located at both ends of the lateral conveyor belt is a drop box system, which is shown in Figure 2.14. The drop box system only operates during the cone dispatch mode, and also serves as a mounting base for the retrieval arm and secondary funnel system. The drop box system receives the cone from the lateral conveyor belt and guides the cone as it drops to the ground. Several guides are used to stabilize the cone as it contacts the road. The drop box with stabilizing guides is shown in Figure 2.14. Once the cone leaves the stabilizing guides of the drop box system, it has completed its journey from the main conveyor belt system to the road.



*Figure 2.14 Drop Box System (Left Side)*

When not in use, the drop box system is automatically stowed by retracting the system within the confinement of the ACM body as shown in Figure 2.15. The retraction of the box includes all the attached subsystem components. Each drop box system is mounted on a track system that allows the drop box system to be lowered 25.4 cm (10.0 in) and moved out laterally 49.0 cm (19.3 in) from the stowed position. The position of the drop box system is controlled by the operator from within the cab.

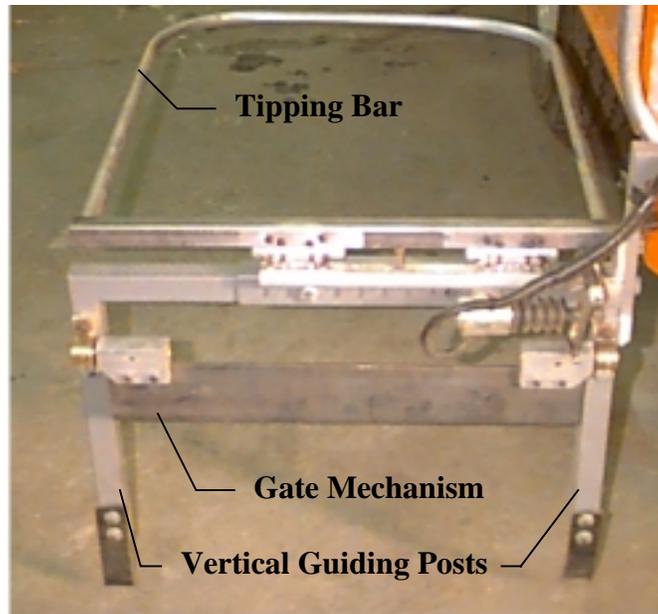


*Figure 2.15 Stowed Drop Box System (Right Side)*

## 2.5 Primary Funnel System

The primary funnel system is only used during the retrieval mode of operation and is the first subsystem encountered by a cone being retrieved from the road. The primary funnel system reorients the cones so that the cone enters base first as it approaches the secondary funnel. The ACM has a primary funnel system mounted all four corners of the truck. The four funnels are necessary for cone retrieval from either side of the ACM and while driving forward or in reverse. Each primary funnel system is comprised of three main components, the gate mechanism, the vertical guides, and a tipping bar.

The components are used to place the cones in the base first position. The gate mechanism is a metal plate that is able to freely rotate along its longitudinal top edge and has a locking device that can hold it in the vertical position. It is activated when necessary to either raise a cone that is pointed tip first to the ACM or tip over a standing cone. The locking device is activated by a switch in the truck cab. The vertical guides consist of two bars that rotate cones as necessary to achieve the base first orientation. The tipping bar is used in conjunction with the gate to flip a cone over to the base first orientation. The left rear primary funnel system and its main components are shown and labeled in Figure 2.16.



*Figure 2.16 Primary Funnel System*

Since the primary funnel system is only used during the retrieval mode, the ACM must be able to retract the primary funnel system when not in use, and deploy it when needed for retrieval. This function is accomplished by activating the hydraulic vane motor to which each of the primary funnel systems is mounted. The tipping bar is designed so that it folds down during the primary funnel system retraction. It automatically unfolds to the open position when the primary funnel system is deployed. Figure 2.17 shows a retracted primary funnel system.



*Figure 2.17 Retracted Primary Funnel System*

## 2.6 Secondary Funnel System

One of the subsystems attached to each drop box is the secondary funnel system which only operates during the retrieval mode. The secondary funnel receives the cone from the primary funnel system and aligns it with the retrieval arm which then picks it up. As a secondary function, its structure also supports some of the cone stabilizing guides used in the cone deployment process. On the ACM a secondary funnel is oriented in both the forward and aft directions on both sides of the truck. The secondary funnel retracts automatically under the ACM when the drop boxes are retracted. This component was redesigned along with the retrieval arm and is a second generation component that is described in greater detail in Chapter 4.

## 2.7 Retrieval Arm

The second subsystem mounted to each drop box is the retrieval arm which also operates only during the retrieval mode. This system receives the cone from the secondary funnel, raises the cone vertically and releases it onto the lateral conveyor belt. This component is obviously a critical component of the retrieval process. The arm was redesigned as a second generation component and is described in Chapter 4. The arm is able to quickly rotate between the forward and reverse directions to pick up a cone. This second generation design has made the ACM an extremely versatile cone retrieval machine. As a result of this design, all set up has been eliminated.

## 2.8 Automated Control System

The automated control system is made up of various sensors, actuators, solenoids, coordinated by a commercially available micro controller. The model use is the ZWorld Co. Little Giant C-Programmable Miniature Controller. This controller is based on a 16 bit Z180 microprocessor and is mounted in a metal enclosure located behind the seat in the truck cab. Sensors are incorporated throughout the subsystems. Besides the infrared sensors on the main conveyor belt system and the gate switches on the lateral conveyor belt that were previously described, sensors exist on other subsystems. The lateral conveyor belt has a sensor on each wing section to indicate if a cone has been dropped off into the drop box. The retrieval arms have sensors to indicate if a cone has arrived and is ready for retrieval. Potentiometer are used to determine the position of the arm. In all, the control system controls the operation and timing of six hydraulic cylinders, one hydraulic motor, seven hydraulic vane actuators, 5 solenoids, and three DC motors, one of which in turn operates the main conveyor belt system hydraulics.

The desired operating mode is defined by the operator in the truck cab. Figure 2.18 shows the various operator control interfaces. The micro controller is located behind the touch-pad keyboard shown in the figure. This panel is mounted behind the driver's seat and is normally not accessed except when trouble shooting. During normal operations the operator uses only a pendant with the four switches shown at the left of the keypad and the switch that operates the primary funnel gate. Allowing for safe operation and emergency situations, panic stop buttons were incorporated into the ACM. One button is present in each bucket of the cone bed and one is located in the cab.

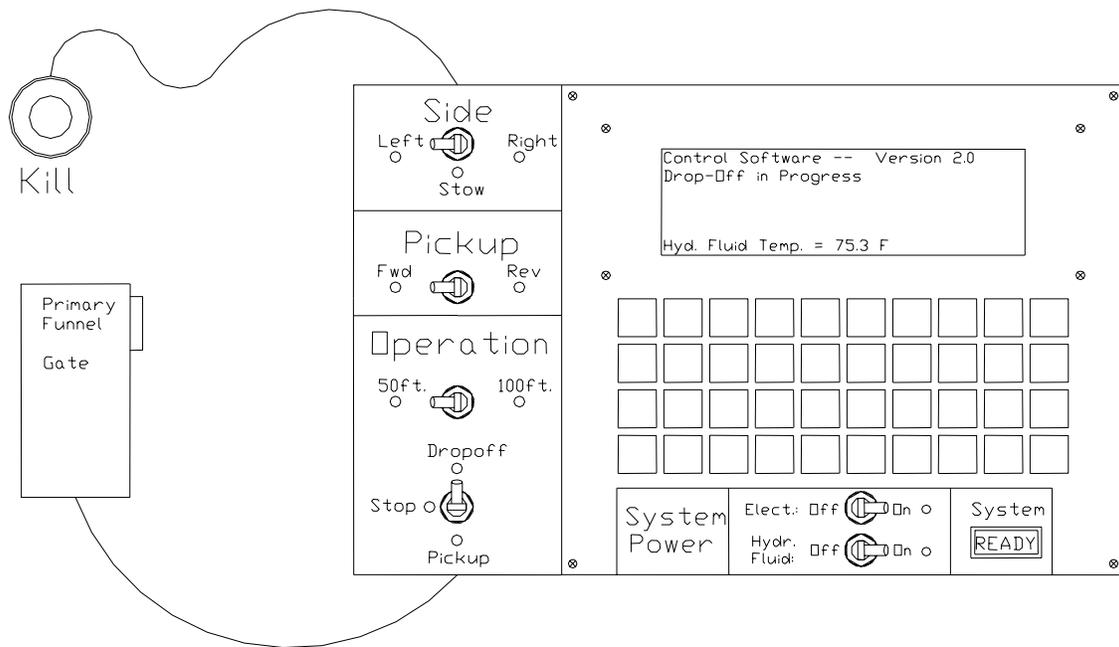


Figure 2.18 Control Interface for Operator

## 2.9 Power Systems

Two sources of power are utilized by the ACM. Some systems require electrical power while most motion systems utilize hydraulic fluid power.

Electrical power is provided by the truck's standard electrical system which is comprised of a 12 Volt DC battery and an alternator driven by the truck's engine. The subsystems that require electrical power include the main conveyor belt system motor, the drop box system drive motor, and the control system with all its associated switches and solenoids. Miscellaneous systems such as the standard sign board mounted on top of the ACM also require 12V DC.

The hydraulic power system uses a variable displacement rotary vane pump using fluid from a ten gallon reservoir located behind and above the cab. Six cylinders, one rotary motor and seven vane actuators are powered by this system. The pump, is driven by the engine's crank shaft via a pulley and two belts and is shown in Figure 2.19.

The fluid tank is mounted below the truck's sign board and above the rear of the truck cab. This location was chosen to allow for easier cooling. A heat exchanger with cooling fan was also mounted below the sign board. The fluid reservoir and heat exchanger with cooling fan are shown in Figure 2.20.



*Figure 2.19 Hydraulic Pump*



*Figure 2.20 Hydraulic Fluid Reservoir and Heat Exchanger*

## 2.10 Summary

In this Chapter the basic operation of the ACM has been provided by a description of each subsystem. The journey of cones from the main conveyor belt system to the road and back is now achievable by using the totally automated ACM. The following chapter will discuss the testing of the retrieval arm and Secondary Funnel of the test bed ACM. The weaknesses and improvements needed in each of these systems will be outlined.

It is easy to see that the typical lane closure procedure is very hazardous, with a high level of exposure to traffic and road hazards. As a result, many precautions are taken to reduce the duration and extent to which the workers and cone truck are exposed. As previously mentioned, the ACM assists in traffic control operation by automating the cone handling process and reducing worker risks. It performs cone deployment and retrieval tasks with a high level of accuracy and efficiency. The machinery performs the repetitive motion of placing and collecting cones, possibly preventing human injury. In addition, the ACM offers a way to remove the workers from the cone body, which frees them to assist with other aspects of the maintenance operation. There is clearly a need for automating the delineation procedure. Soon, the ACM

technology will replace the current, manual method of performing lane closures for traffic control.

Although there is great potential for using the ACM to support highway maintenance, the low cone capacity of the cone body was identified as a significant shortcoming. For many special maintenance operations neither the ACM, nor the standard cone truck, hold enough cones. For this reason, serious consideration was given to increasing the cone load capacity of the trucks. It was believed that an automated cone storage system would offer an increase in cone capacity that would expand the ACM capabilities and broaden its potential for traffic control operations. The first step towards developing the system was searching for existing, related technology.

## CHAPTER 3 DEVELOPMENT OF A HIGH CAPACITY STORAGE UNIT

### 3.1 Selection of a Storage System Design Concept

Selecting a successful design concept for storing cones was a critical step in developing a complete and functional system. The concept selection process began by identifying specifications related to the project scope and the system design. The specifications were divided into a set of requirements and aims, which guided preliminary concept development. Through brainstorming efforts, a number of system design ideas were conceived. Several promising concepts were expanded in detail to gain insight into the long-term potential for development. Methodical comparison of the design concepts, using a trade-off analysis, assisted in identifying a final concept. The design criteria, the concepts considered, and the method of comparison are described and presented in this section.

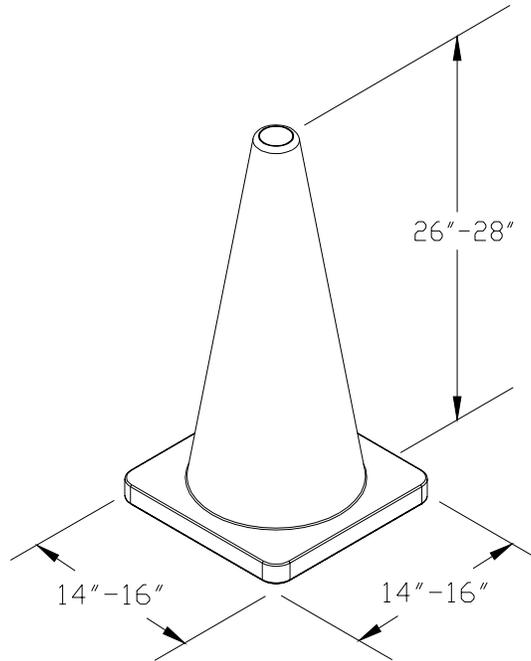
#### 3.1.1 Project Requirements and System Criteria

Project specifications were developed from the project proposal and standards adopted during development of the ACM. From the collection of specifications, the requirements were differentiated from the goals and aims. The project requirements placed constraints on the system design to ensure compatibility with maintenance operations and machinery. The project goals and aims were reduced to a set of criteria, which were used to produce and evaluate the best system design.

##### 3.1.1.1 Cone Related Requirements

Cone related factors naturally had the strongest influence on concept development. Therefore, it was important to specify standards for handling and storing cones in order to properly guide and direct the conceptual design efforts. In particular, careful consideration was given to cone shape, cone size, cone storage capacity and cone storage configuration.

The traffic cones used by Caltrans for maintenance operations were composed of a plastic inverted hollow conical section mounted to a 3.2 cm (1.25 inches) thick rubber base pad. This basic cone shape was accepted as the standard for the storage system. The overall supply of traffic cones, however, contained notable size variations due to the vast number of cone suppliers. As shown in Figure 3.1, the range of the base pad dimensions varied between 14 inches square and 16 inches square and the height of the cones ranged from 26 inches to 28 inches. The difference in height proved to be a minor problem as the cones were stored in stacks. Different cone heights resulted only in slight changes to the overall length of the stacks. However, variation in the size of the base introduced more significant problems related to manipulating the cones individually and in stacks. Regardless, it was critical for the storage system to handle all sizes of cones in this range. Due to the diverse cone sizes, it was also required that each truck carried a uniform supply of cones. This ensured that the cones would stack properly and prevented additional cone-handling problems.



*Figure 3.1 Cone Shape and Acceptable Sizes*

A minimum increase in cone carrying capacity of the vehicle was necessary to justify the storage system development project. So, it was decided that a minimum of 160 cones were to be stored by the system. This effectively doubled the previous capacity of the maintenance vehicle cone body, increasing the number of stacks to four. A tripled cone capacity of six cone stacks, or 240 cones, was even more desirable.

A horizontal stack orientation was preferred, similar to the current configuration of storing cones on the cone body. Maintaining the horizontal configuration ensured system compatibility with maintenance operation methods. The horizontal stack orientation simplified the system design by minimizing the amount of stack handling required to store the cones. The stacks were also more stable and easy to secure in a horizontal position. Changing the stack orientation ventured beyond the project scope, potentially required significant changes to the manual and automated methods of retrieving cones and significantly increased the complexity of the storage system.

### 3.1.1.2 Vehicle Specifications

Much effort was made to design the storage system around the vehicle constraints and to minimize the number of vehicle specifications. The Caltrans cone body was the platform chosen for implementing the storage system on the maintenance vehicle. In addition to the cone body load, the vehicle weight capacity needed to exceed the weight of the storage system and a full load of cones. The conservatively predicted weight of the system components and six stacks of 40 cones was 15,680 N (3500 lb). All remaining vehicle related considerations were accounted for during the detail design phase of system development.

### 3.1.1.3 Safety Requirements

Though a broad and general requirement, the overall safety of the storage system was essential. The system was intended for low risk automated operation. No additional hazards were to be created that might endanger the workers or the vehicle during maintenance procedures. In the event of system failure, cone placement and retrieval was to continue uninterrupted, without jeopardizing the operation. The cone stacks inside the storage system framework were also required to be secure at all times.

### 3.1.1.4 System Integration Specifications

Many constraints were placed on the system design to ensure a compatible fit with the vehicle. However, maintenance vehicle modifications were inevitable prior to storage system integration. Modifications were limited to features of the cone body and the main conveyor. Cone body modifications were not desirable in order to preserve the original design and function. A modular system design was preferred to minimize modifications to the vehicle and simplify system assembly and integration. Developing the system as a modular unit increased the appeal for large-scale system integration into the existing fleet of maintenance vehicles.

While capable of independent operation, the storage system was required to be functionally compatible with the ACM. The critical interface between the storage system and the ACM stowage system was at the front end of the main conveyor. Preserving the spatial layout between the main conveyor of the storage system and the stowage system ensured integration compatibility with the ACM.

### 3.1.1.5 System Design Specifications

The system design specifications highlighted important physical factors that were considered in developing the design concepts. The size of the system was constrained from extending outside the cone body perimeter. Height limitations were not specified other than prohibiting visual obstruction of the vehicle overhead traffic signal. The weight of the system was preferred to be low. System weights were approximated from the combined weights of the major components of each concept. Minimizing the weight of the system selected using optimization studies was postponed for the detail design phase. A symmetric system configuration was preferred to evenly distribute the cone load on the vehicle and prevent weight imbalance problems.

The power requirements of the storage system were to be designed within the limitations of the vehicle power systems. The electrical power supplied to the system by the vehicle delivered 12 Volts and up to 60 Amps. The standard electro-hydraulic unit built into the cone body, which supplied just under 750 Watts (1 horsepower), was to be extended to operate the system but potentially required a larger fluid reservoir. The ACM was outfitted with a hydraulic pump, which operated at 6.9 kPa (1000 psi) maximum working pressure and a flow rate of 2 GPM to output 872 Watts (1.17 horsepower).

A low overall mechanical complexity was desired for the system, which included simple components and ease of assembly. A minimal number of actuators were preferred to simplify

and reduce cone stack handling. Another objective was to minimize maintenance demands of the system components.

#### 3.1.1.6 System Operation Specifications

The system controls were greatly influenced by the mechanical complexity of the system and the number of actuators. Therefore, breaking up the system operation into simple, executable actions and decreasing the number of sensors required was desirable to reduce the control complexity. The storage system control would be achieved using a Z-World micro-controller. The Z-World was originally selected for ACM development and its capabilities were easily extended to control the storage system. The system was also required to have a manual control mode. Minimizing the number of exposed moving parts was preferred to prevent potential interference during system operation. Another important consideration was reducing the operating time of the system to prevent disruption of the cone placement and retrieval operation.

#### 3.1.1.7 Cost

Minimizing the costs associated with the development was important, but not at the expense of the system quality. Affordable testing of the preliminary design concept was a priority. Strong consideration was also given to savings gained by using standard parts and readily available components. Fabrication and custom machine work were identified as the most costly aspects of development, however the expenses were difficult to predict without further developing the concept details. Thus, during the preliminary concept development phase, the total development costs of each concept were approximated using the information available.

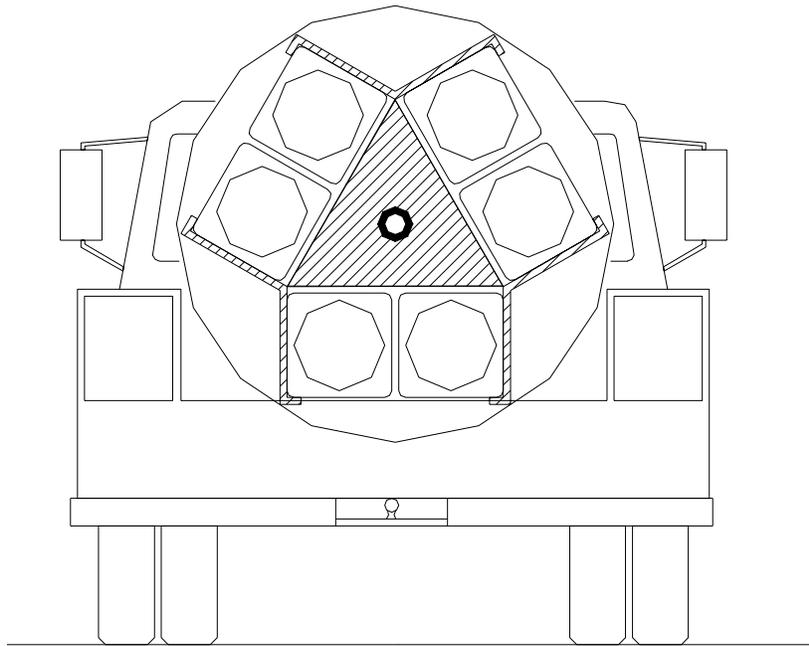
### 3.1.2 Design Concepts

After developing a detailed list of specifications, significant time was spent researching and exploring potential system design options that would most effectively meet the design criteria. Three solid concepts were expanded in greater detail: the Revolving Drum, the Hinged Rack and the Vertical Lift. Each was named after its function for storing cones and characterized and evaluated by identifying key strengths and weaknesses.

#### 3.1.2.1 Revolving Drum

The revolving drum concept, shown in Figure 3.2, used a cylindrical shaped drum to store cone stacks. The rotation axis of the drum was oriented along the length of the truck bed and centered above the main conveyor belt. The drum container design included several slots, each large enough for two cone stacks. By rotating the drum, the slots were positioned directly above the main conveyor to store or dispense cones. During cone retrieval, the main conveyor inserted cones into the slotted openings. Then, features inside the drum clamped and secured the stacks. Actuation of the drum was achieved using a minimum of one rotary motor and several linear actuators, which were used to raise the drum. Raising the drum was necessary to avoid interference with the main conveyor while the drum rotated.

The revolving drum design minimized manipulation of the cone stacks, which was advantageous. It also offered high cone storage capabilities. Expanding the drum design potentially enables it to carry up to ten cone stacks, which amounted to between 300 and 400 cones. However, increasing the cone carrying capacity of the drum was to likely exceed the weight limitations of some maintenance trucks and required a larger vehicle. The necessary cone body modifications included decreasing the storage bin size depending on the drum diameter. However, by carefully designing the drum support features, the main conveyor required little redesign work. Another disadvantage of the design was the overall size and excess drum volume. The void space in the center region of the drum was inevitable due to the spatial layout configuration of the cones. The overall mechanical complexity was moderate and controlling the drum rotation posed potential problems due to weight imbalances created by the cone stacks.



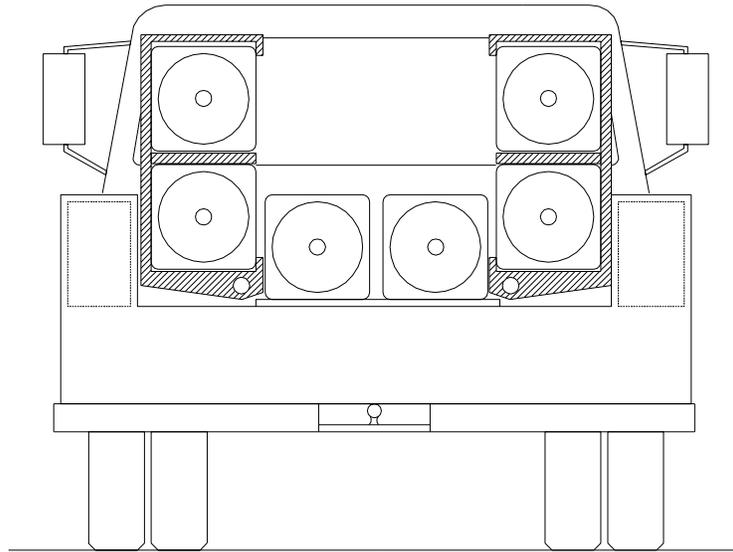
*Figure 3.2 Revolving Drum Concept*

### 3.1.2.2 Hinged Rack

The hinged rack concept consisted of two, symmetric cone storage racks, one on each side of the main conveyor, as illustrated in Figure 3.3. Each rack consisted of a double stack support framework, rotary motion actuators and additional features to secure the cones. The racks rotated between a stored position above the cone body storage bins and a deployed position above the main conveyor. Like the revolving drum design, the main conveyor fed cones in and out of the deployed racks during cone retrieval and placement operations, respectively.

The hinged rack concept offers a solid option for tripling the cone capacity of the maintenance vehicle by storing six stacks, or approximately 240 cones. However, configuration limitations prevented the design from being expanded to store more than six stacks. The low

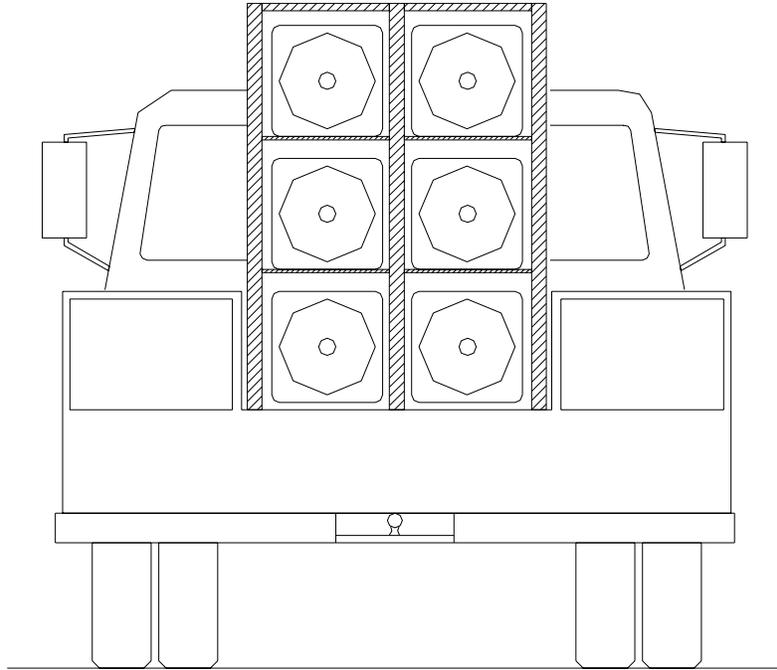
mechanical complexity simplified the detail design and control development. Moderate cone body modifications were required for system integration. The main conveyor configuration likely needed changes and, depending upon the location of the rack hinge point, portions of the storage bins also required modification. Weight imbalance posed a potential problem with the design. With one rack full of cones and the other empty, an unsymmetrical loading condition was created, which amounted in a 3585 N (800 lb) load difference across the width of the vehicle. For short operations weight imbalance raised little concern. However, it was undesirable for this condition to exist for extended periods of time, especially while traveling at highway speeds.



*Figure 3.3 Hinged Rack Configuration*

### 3.1.2.3 Vertical Lift

The vertical lift design, depicted in Figure 3.4, stored cone stacks in an array configuration directly above the main conveyor. An actuated vertical lift was required to raise and lower stacks between an overhead storage framework and the main conveyor during cone storage and retrieval, respectively. The main conveyor was used to build and properly position the cone stacks on the lift prior to storing them. The lift then raised and positioned the cone stacks in the framework where securing features were used to constrain and support the cones. During system operation, the storage framework was progressively filled with cone stacks one level at a time, beginning with the top level and proceeding down. The stacks were retrieved in the reverse order.



*Figure 3.4 Vertical Lift Concept*

The vertical lift design increased the cone load of the vehicle to between four and eight cone stacks, a total of 160 to 320 cones. Height and weight constraints limited a larger cone capacity. Implementing the vertical lift required minimal modifications to the storage bins but a significant redesign of the main conveyor. The lifting mechanism introduced a potentially high level of mechanical complexity into the system. Physical constraints of the system also complicated the method of securing the stacks. The linear motion of the lift, and the potential for coupling the cone retention features, reduced the control complexity. However, the time required to operate the system was evidently longer compared to the other concepts. Though the weight of the framework and lift were a concern, weight imbalance problems were eliminated by storing the cones along the middle of the cone body above the main conveyor.

### 3.1.3 Concept Comparison

Concept comparison began after finalizing the preliminary concept design details. A decision matrix was used to identify the most qualified concept. The system criteria were weighted according to importance, relative to each other. The concepts were then rated on a scale of 1 to 5 based upon the extent to which they satisfied the design selection criteria. To be considered, each concept was required to satisfy all the project requirements. To improve the reliability of the trade-off analysis results, a reference concept method was chosen to assist in the rating assignment process. The reference concept was given average ratings (3) for each of the selection criteria. Each of the other concepts was then rated relative to the reference concept. If a concept rated better than the reference, a rating of 4 was assigned. A concept rated worse than the reference was given a rating of 2. Ratings of 1 or 5 were reserved for concepts severely worse or significantly better than the reference concept, respectively. The weighted scores of

each concept were determined by computing the product of each criterion weight factor and the respective concept ratings. Finally, a rank was assigned to each concept based on the total scores. The highest score received the top rank, while the lowest score received the lowest rank.

This process offered many benefits beyond identifying a final design concept. The decision matrix concisely highlighted and organized the major design parameters. It also provided a method for evaluating the concepts using a number of subjectively developed criteria. This process documented the conceptual development phase for future reference and justified concept selection.

The trade-off analysis results ranked the vertical lift highest of the three concepts considered, as summarized in Table 2.1. Compared to the hinged rack concept (reference), the vertical lift scores differed slightly for a number of the selection criteria. The key factor, which identified the vertical lift as the top scoring concept, was its higher than average ratings for several of the most important design criteria. The deciding design criteria include cone storage capacity, operational hazards and required actuation modes.

The hinged rack storage capacity limited the number of stored cone stacks to six. The vertical lift design offered expanded storage capabilities of up to eight cone stacks and received a higher rating as a result. The operational hazards of the vertical lift rated high because the required storage framework enclosed a majority of the moving parts, unlike the other concepts. The number of actuation modes predicted for the vertical lift was lower than for the others, as the potential for linking the cone securing devices seemed possible. The load distribution and system placement also proved more desirable than for the hinged rack design.

The vertical lift was given lower than average ratings for several criteria, most significant being mechanical complexity. The mechanical complexity of the vertical lift was expected to be higher, especially considering the details of the lift design. An evaluation of mechanical complexity also led to the belief that the cost would be higher to develop the vertical lift concept. The weight and size of the vertical lift rated lower than the rotary arm, however, these criteria were given low weight factors. Finally, the vertical lift run time duration was also identified as below average according to the standard set by the hinge rack concept.

Table 3.1 Design Concept Scoring Matrix

SELECTION CRITERIA	WEIGHT FACTOR	CONCEPTS					
		A (reference)		B		C	
		HINGED RACK		REVOLVING DRUM		VERTICAL LIFT	
		Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
<b>Requirements</b>							
Minimum 160 cones stored	-	yes		yes		yes	
Handles range of cone sizes	-	yes		yes		yes	
Maintains horiz. cone orientation	-	yes		yes		yes	
ACM compatible	-	yes		yes		yes	
Compatible power system	-	yes		yes		yes	
<b>Cones</b>							
Cone storage capacity	8	3	24	4	32	4	32
<b>System Implementation</b>							
Minimal cone body modifications	4	3	12	4	16	3	12
Modular	3	3	9	3	9	3	9
<b>Safety</b>							
Operational hazards	10	3	30	3	30	4	40
Securely stored cones	10	3	30	3	30	3	30
Visibility	4	3	12	2	8	2	8
<b>System Design</b>							
Overall weight	6	3	18	2	12	2	12
Load distribution	4	3	12	4	16	5	20
Size	4	3	12	2	8	2	8
Placement	4	3	12	4	16	4	16
Required actuation modes	6	3	18	3	18	4	24
Mechanical complexity	7	3	21	2	14	2	14
<b>System Operation</b>							
Controls complexity	4	3	12	3	12	3	12
Run time duration	2	3	6	3	6	2	4
<b>Cost</b>							
	6	3	18	2	12	2	12
<b>TOTALS</b>			246		239		253
<b>RANK</b>			2		3		1

### 3.1.4 Summary

The process followed during the preliminary phase of conceptual development was critical to starting the project correctly. Identifying and defining the project requirements and system criteria helped focus the preliminary brainstorming phase and direct the concept trade-off analysis. Comparing and rating the concepts for each design criteria proved to be an excellent method of judging the designs. While the comparison results proposed the most qualified

system, the final selection required consideration of other non-quantifiable factors. All things considered, the vertical lift concept was selected for further development. Conceptual brainstorming and preliminary design of the primary features of the vertical lift system constituted the next phase in the development process.

### 3.2 Conceptual Design and Preliminary Development

After selecting the vertical lift as the fundamental design concept for the multistack system, conceptual development was continued for the primary system components. A concept selection approach was used, similar to the procedure previously followed. Specific design criteria were identified, concepts were generated and tested, and trade-off analyses were completed. Thorough consideration was given to component interface, layout constraints and the system operating procedure. Fundamental details were considered and preliminary calculations were performed to verify critical aspects of the designs, but the primary objective of conceptual design was to lay the basic design framework for future expansion. The component designs considered include the lift, the retention features, the redesigned main conveyor and aspects of the structural framework. This section describes the process of transforming the vertical lift system concept into a concrete conceptual design.

#### 3.2.1 The Vertical Lift

The vertical lift was identified as the primary functional unit of the multistack system. The fundamental purpose of the lift, to support, raise and lower the cone stacks, clearly influenced every other aspect of the system design. The lift also required the longest lead time for detail design work and fabrication. Therefore, conceptual design of the lift was addressed first. Several design concepts were developed and compared based upon specific design criteria and functional testing.

##### 3.2.1.1 Selection Criteria and Considerations

The lift was required to securely raise, lower and support the cone stacks. Beyond the one functional responsibility, a number of criteria were used to evaluate the designs. Some of the lift selection criteria were similar to those considered in selecting the multistack system concept, such as size, weight, design simplicity, actuation requirements, cone body modifications, and cost. Other specific criteria were also established, which included design robustness, commercial availability of components, redesigned conveyor configuration, ease of implementation and assembly and expandability. All the criteria combined were used to rate the lift designs.

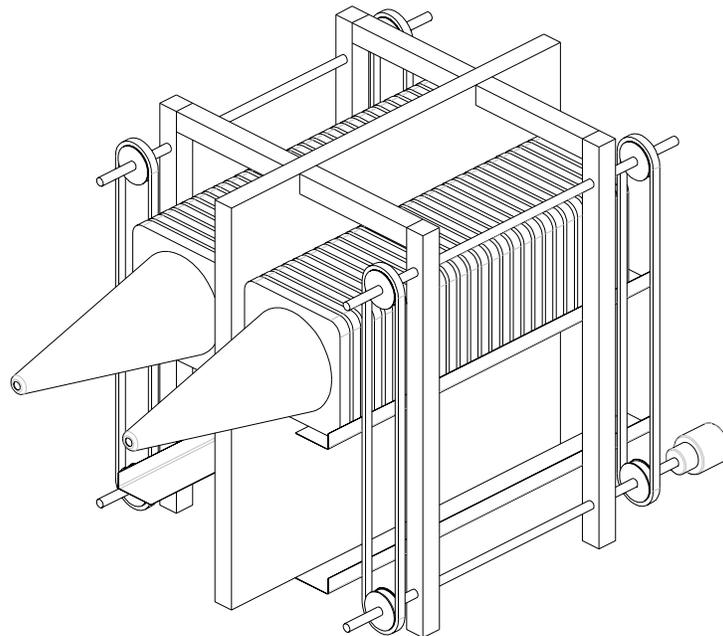
##### 3.2.1.2 Lift Design Concepts

Significant time was spent brainstorming and exploring potential vertical lift ideas, a number of which stemmed from commercially available lift designs or material handling equipment. In general, the concepts were categorized into two types: dedicated and multifunctional lift configurations. All lift concepts which functioned only to raise and lower the cone stacks were considered dedicated lifts. The multifunctional lift concepts were also responsible for raising

and lowering the stacks. In addition, they functioned as the primary means of securing the stored stacks. A multifunctional lift completely achieved the functional requirements of the multistack system, while a retention system was needed to store the cone stacks with a dedicated lift configuration. Redesign options for the main conveyor were evaluated and finalized simultaneously with each lift concept, due to the close interface between the two units. After generating a number of ideas, four concepts were chosen for further consideration and comparison, namely the vertical conveyor lift, the scissor lift, the multistage platform lift and the forklift.

### 3.2.1.3 Vertical Conveyor Lift with Slip Wall

The vertical conveyor lift concept, shown in Figure 3.5, originated from several ideas related to material handling equipment and conveyor drives. Vertically oriented conveyor chains, located laterally outward from each of the stacks, were responsible for raising the cones. In the place of chains between each pair of cone stacks, a vertical slip wall was added to stabilize the middle-facing surface of the stacks. Stack support members attached to custom links on the chains to brace the underside of the cones, which were raised and lowered by activating the chain up and down. The stack support members bore the weight of the cones and the medial side surface of each stack slid up and down against the slip wall. The slip wall contained outer sheets of low friction plastic to offer a low resistance slip surface for the stacks. The two sets of chains were driven by parallel motors. The redesigned main conveyor required a single belt, wide enough to support both stacks, yet narrow enough for the stack support members to contact and lift the bottom outer edge of the cones. The vertical conveyor lift design included stack support members for each stack, which secured the cones in storage, and characterized the lift as multifunctional.



*Figure 3.5 Vertical Conveyor with Slip Wall*

There were several advantages to the vertical conveyor lift concept. The multifunctional lift configuration eliminated the need for independent stack retention features. The slip wall loosened the coordination requirements for lifting the stack, which simplified the motor drive control. Many of the design components were commercially available, either as custom ordered equipment or off the shelf parts. The system design was relatively simple, low in weight and medium in size. System fabrication was predicted to be in the medium price range, with expensive cone body modifications offset by lower prices for commercial components.

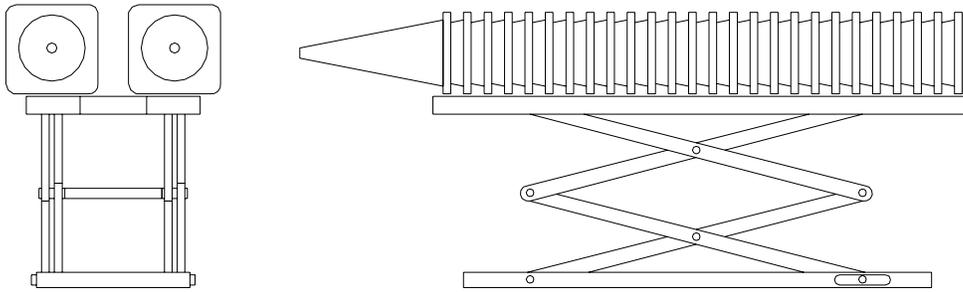
There were also disadvantages of the vertical conveyor lift design. The means of raising and lowering the cones by actuating the chains was questionably robust. The slip wall and stack support member configuration also raised concerns about adequately securing the stacks. While the main conveyor configuration remained the same, narrower rollers and a new conveyor support framework were needed. Potentially, significant cone body modifications were also required to assemble and operate the vertical chains.

#### 3.2.1.4 Scissor Lift

The scissor lift concept resembled typical commercial scissor lift designs, as illustrated in Figure 3.6. It would function as a dedicated lift to raise and lower stacks in the storage framework. The primary design features included a custom lift platform, lift actuators, scissor linkages and lateral support features. The platform was required to support both stacks and interface with the retention devices to ensure proper transfer of the cones inside the storage framework. The scissor linkages and actuators mounted underneath the base position of the platform and the cone body bed. The simplest configuration for the main conveyor involved replacing the existing belt with two smaller belts, one belt for each stack, and mounting the conveyor components to the lift platform.

Some of the advantages of the scissor lift design included a secure means of elevating the cone stack and a highly robust design. The lift design offered a very compact configuration, which conveniently stored underneath the middle of the cone body bed. Some of the scissor lift components were available through commercial vendors, which made the design easily expandable for larger storage systems.

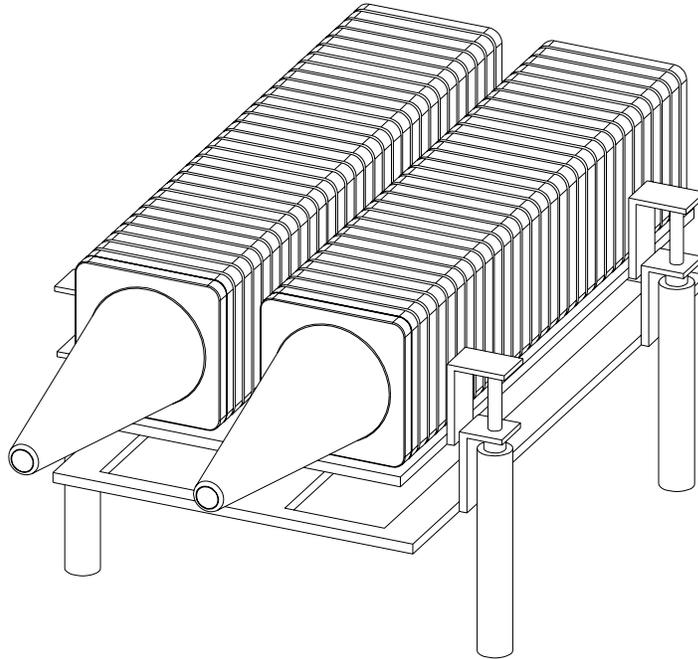
The scissor lift design disadvantages were mostly related to integration issues. Potentially, significant cone body modifications would be necessary to mount the actuators and scissor mechanism below the bed level. The redesigned main conveyor was functionally equivalent to the existing configuration, but mounting to the lift platform was not preferred. Complexities associated with lift assembly and control were also predicted. Cost estimates fell in the medium range, assuming many of the lift components would be purchased commercially.



*Figure 3.6 Scissor Lift Concept*

### 3.2.1.5 Multistage Platform Lift

The multistage platform concept was characterized as a multifunctional lift design and used separate cone stack support platforms to raise and lower each pair of cone stacks. The platforms were actuated by telescopic cylinders, as shown in Figure 3.7. When empty, the platforms would stack below the bed of the cone body, inlaid with the main conveyor. Each platform would mount to a corresponding stage of the telescopic cylinders. As cone stacks became full, the cylinders would be activated one stage at a time. Progressively, the stages would be lifted and the cone stacks stored. To alleviate side loading of the cylinders, linear guides would be assembled to constrain the lateral motion of the platforms. The redesigned conveyor would be split into two separate belts, one for each cone stack. The width of each belt was required to be several centimeters narrower than the stack width for the platforms to properly support the cones.



*Figure 3.7 Multistage Platform Lift*

The multistage platform concept had many positive attributes. The overall and individual component designs were simple and the platforms and cylinders could be made highly robust. A moderate system weight was estimated. The design configuration was compact in size and the redesigned main conveyor configuration was acceptable. A stack retention system was not needed with the multistage platform lift.

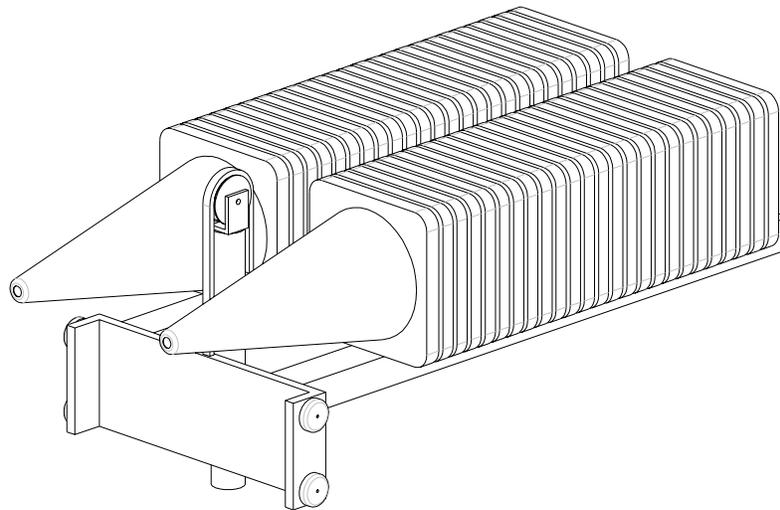
There were also shortcomings related to the concept design. The multistage design potentially required significant modifications to the cone body framework and storage bins. The main conveyor redesign also needed a completely new layout configuration to be compatible with the platform design. Integration of the system posed challenges related to placement and assembly of the cylinders on the truck, which would be difficult with the tight tolerances required to ensure proper lift operation. The large number of cylinders, which required coordinated actuation, complicated system control. The projected costs associated with development were moderate, primarily due to customized components

#### 3.2.1.6 Forklift

The forklift design concept was based upon the fundamental design of industrial forklifts. The components of the forklift included forks, a carriage, masts and a cylinder assembly. The concept is depicted in Figure 3.8 without the masts. The cylinder would mount to the truck frame toward the back end of the cone body. The carriage would be placed below the cone stacks and adjacent to the cylinder. A leaf chain would couple the carriage to the cylinder piston. Activation of the cylinder raised and lowered the carriage. The channel masts were to be mounted on both sides of the carriage. Roller bearings attached to the sides of the carriage would travel inside the channels, which constrained the motion of the carriage to the vertical

plane. The forks would be rigidly attached to the carriage and cantilevered forward to support the entire length of the cone stacks. A split belt design was identified as the best main conveyor configuration. Two narrow, parallel belts could be used to support and convey each stack. Interference conflicts between the conveyor belts and the carriage led to a conveyor design which mounted to the forklift unit.

There were a number of positive aspects about the forklift concept. The design was simple, yet robust. The means of actuation was also simple, requiring only one cylinder to achieve vertical motion of the forks. The forklift design also lent itself to many options for using commercially available parts and equipment. Since the forklift functioned as a dedicated vertical lift, the design was easily expanded for larger systems.



*Figure 3.8 Forklift Concept*

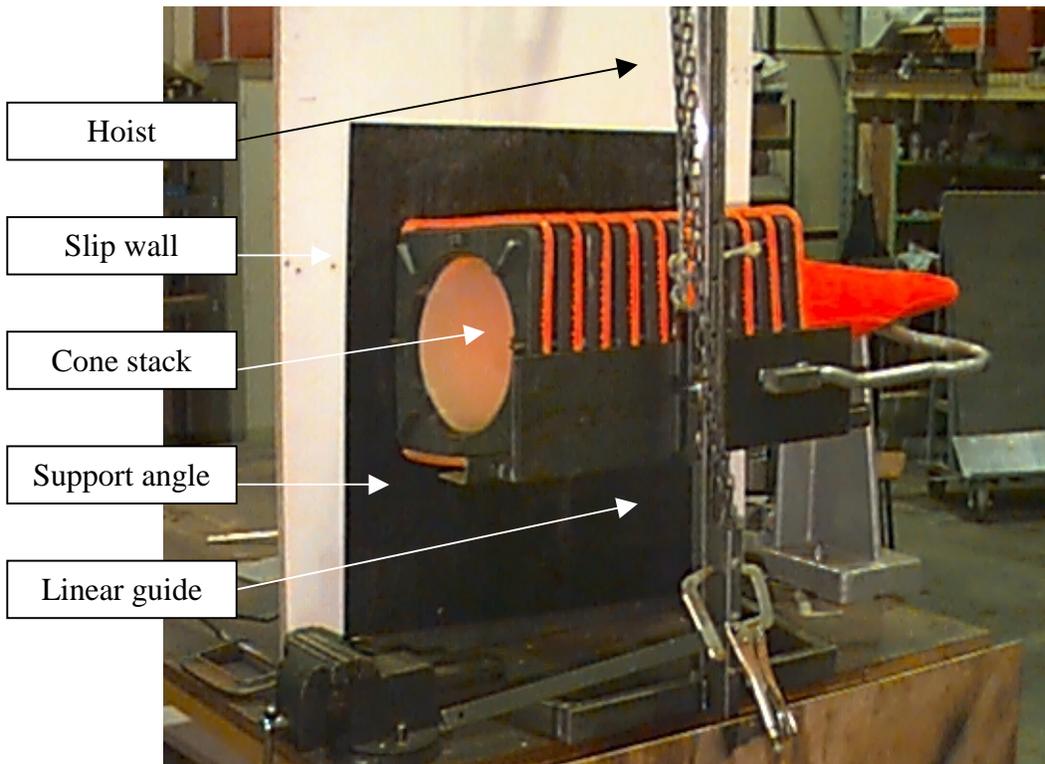
Negative aspects of the forklift design included high weight due to hefty components, specifically the large size of the masts and cylinder. An independent retention system was also needed to secure the stacks in the structural framework.

#### 3.2.1.7 Functional Testing of the Lift Concepts

Preliminary testing of the vertical lift was performed simultaneously with concept brainstorming. Aspects of the concept designs raised questions and concerns about functional operation and performance. In light of the lift design's influence on the remaining system components, it was important to identify, evaluate and correct potential design problems and shortcomings. A single test set up was designed to help understand the critical aspects of lifting the stacks and offer insight into better lift design concepts.

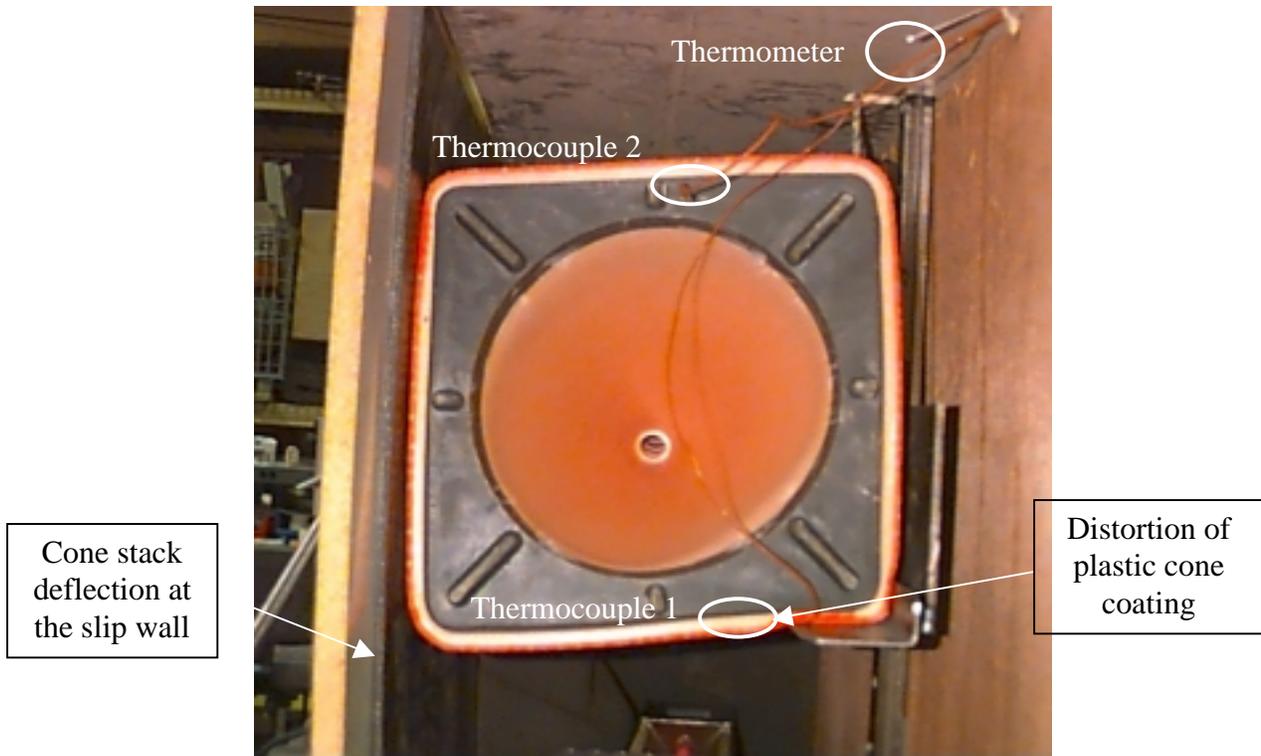
### 3.2.1.8 Test Description and Objectives

The test configuration resembled a small scale model of the vertical conveyor concept. The assembly consisted of readily available parts, which were functionally equivalent to the components of the slip wall design. The set up included a short stack of 10 cones, a rigid, ultra high molecular weight plastic (UHMW) plated wall, an angle iron beam, a linear guide, a manual hoist and a support framework, as shown assembled in Figure 3.9. One side of the cone stack rested against the vertical wall and the bottom corner of the opposite side was supported by the angle iron. The angle iron fastened to the linear guide bearing block, which constrained rotation in all directions. The guide also prevented the angle from moving laterally. The hoist attached to the angle, adjacent the bearing block. The slip wall and linear guide were rigidly secured to the steel support framework and an enclosure was constructed around the assembly.



*Figure 3.9 Vertical Lift Test Assembly*

An infrared heater was placed below the cone stack and directed upward to heat the enclosure, as shown with the rest of the assembly in Figure 3.9. Two thermocouples and a thermometer gauge were used to collect temperature measurements during testing. One thermocouple was placed in contact with the bottom surface of the cones to measure the approximate surface temperature of the cone stack directly above the heater. The other thermocouple was inserted into a hole in the base of a cone, which provided a rough measurement of the internal temperature of the cones. The thermometer was set above the cone stack to measure the air temperature at the top of the enclosure. The measurement device locations are shown in Figure 3.10.



*Figure 3.10 Temperature Measurement Device Locations and Heating Effects*

The test objectives were twofold. First, functional aspects of the slip wall design were examined, giving specific attention to the support of the cone stack by the slip wall and angle. Second, the test was designed to explore the effects of heating on the cone stack. During the summer, when road temperatures often exceed 120 degrees F, the plastic properties of cones are altered. The conical section loses rigidity and the cone surface becomes very sticky. By producing similar temperature conditions, it was possible to observe how heating the stack effects the functional operation of the slip wall design.

### 3.2.1.9 Testing Procedure, Observations and Results

Preliminary measurements revealed that the heater created a large temperature distribution inside the enclosure. The bottom surface of the stack heated very quickly. The temperature above the cone stack gradually increased, but remained considerably lower than the temperature below the stack. The internal cone temperature elevated slowest, closely trailing the temperature above the stack. After identifying the heating tendency, a cyclic heating procedure was followed to gradually raise the internal cone temperature without driving the bottom area temperature excessively high. To begin, the heater was turned on. Once the cone surface temperature reached a maximum of 185 degrees F, the heater was then turned off. The enclosure was cooled until the temperatures equilibrated. The process was repeated five times. After each cycle, the internal cone temperature gradually increased, reaching a peak temperature of 125 degrees F prior to testing. Figure 3.11 graphically displays the peak temperature measurements collected during each heating cycle.

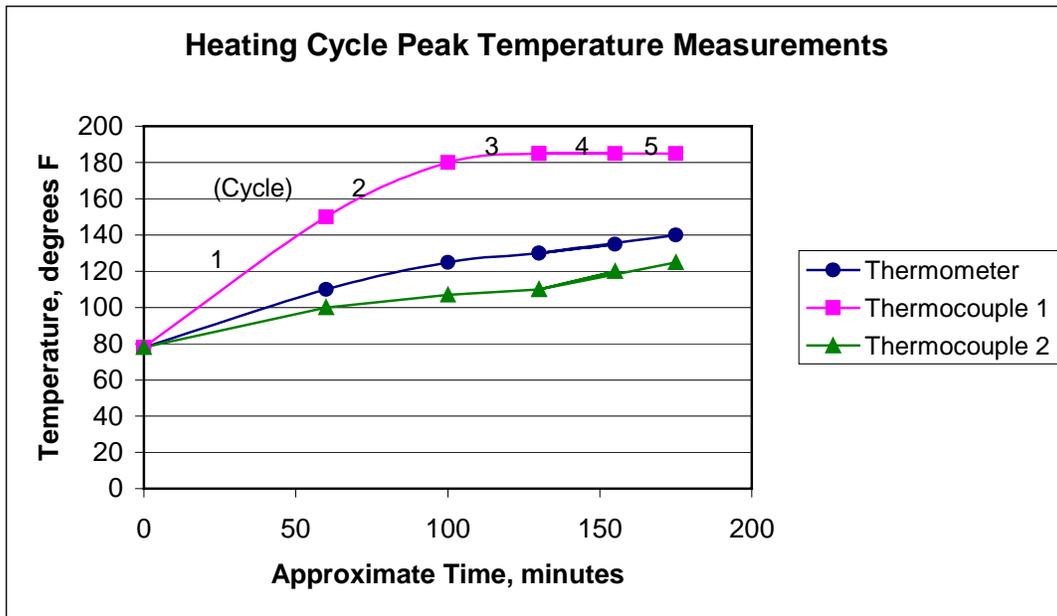


Figure 3.11 Progressive Heating Measurements Recorded during Lift Testing

Upon reaching the peak internal cone temperature, the set up was tested by raising and lowering the stack. The stack experienced large deflections, up to 6.3 mm (2.5 inches), at the wall contact as shown in Figure 3.10. The plastic coating on the surface of the cone base was noticeably soft and slightly sticky. As a result, there was resistance in sliding the stack up the wall, which caused the cones to tend to roll off the angle iron. The supported edge of the cones also showed significant deformation, also shown in Figure 3.10. The UHMW sheet experienced significant thermal expansion as large bulges appeared in several areas on the wall. However, the distortion in the plastic had little noticeable affect on raising the stack and probably helped minimize the stack deflection at the wall. Overall, higher temperatures hindered the lift operation and led to inadequate support of the cone stack.

The lift assembly functioned normally at room temperature. The stack experienced some resistance due to friction at the wall support, however, the cones deflected significantly less. Several times, when the stack was lifted, the cones would deflect and begin to roll and compress between the wall and the angle iron, as observed during high temperature testing. To prevent this condition, a more secure method was needed to support the cones.

### 3.2.1.10 Test Conclusions

Several specific conclusions were made regarding the vertical conveyor design concept. It successfully lifted the cones, but failed to properly support the stack. The support angle alone was not sufficient to secure the bottom surface of the stack due to the high compliance of the plastic outer coating on the cone. The lack of rigidity in the cone base also contributed to large deflections at the vertical wall. The UHMW plating on the wall created a sufficient, low friction

surface for sliding the stack. However, the sticky nature of the cone surface at high temperatures led to unacceptable deflections. Fundamentally, the slip wall design concept achieved stack storage, but additional cone support features were needed and reliability was questioned.

The test results also led to additional brainstorming of other lift designs. The slip wall design represented a multifunctional lift configuration that supported the cone stack on both sides and from underneath along one edge. Better options for supporting the stack were identified and organized based upon the lift classification types. Additional layout configurations were considered for other potential dedicated lift designs, specifically the forklift and scissor lift, which required stack retention features. Interface arrangements between the lifts and the main conveyor were also explored. Implicit conclusions were made from the slip wall testing that helped incorporate desirable design characteristics into other vertical lift concepts. Because the slip wall design provided fundamental insight into some of the other vertical lift designs, no further testing was performed.

#### 3.2.1.11 Trade-off Analysis and Concept Comparison

A relative weighting method was used to independently rate the lift concepts on a scale from 1 to 10 in relation to each of the design criteria. Weight factors were subjectively assigned to the individual criteria and weighted scores were computed for each concept by computing the product of the criteria weight factors and concept ratings. The sum of each concept's weighted scores determined the ranking order.

From the trade-off analysis results, the forklift concept was identified as the most qualified vertical lift design. Though it received lower ratings for the conveyor configuration, cone body modifications and weight, the advantages of the system, such as design simplicity, ease of assembly, robustness, simple means of actuation and commercially available components, compensated for the weaknesses. Aside from the decision matrix comparison, the forklift design offered an accessible and packaged configuration, which easily mounted to the back end of the truck. The expandable forklift design offered great flexibility for a number of different storage system sizes. The scissor lift concept ranked second and the multistage concept ranked third. Both received overall lower ratings than the forklift.

Table 3.2 Vertical Lift Concept Decision Matrix

SELECTION CRITERIA	WEIGHT FACTOR	CONCEPTS							
		VERTICAL CHAIN CONVEYOR		SCISSOR LIFT		MULTISTAGE PLATFORM LIFT		FORKLIFT	
		Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
<b>Lift Design</b>									
Secures stacks	10	2	20	4	40	4	40	4	40
Simple	8	3	24	3	24	3	24	4	32
Size	4	3	12	3	12	3	12	2	8
Weight	4	4	16	3	12	3	12	2	8
Robust	8	2	16	5	40	4	32	5	40
Means of actuation	8	2	16	4	32	2	16	5	40
Commercially available parts	6	4	24	3	18	3	18	3	18
<b>Lift Integration</b>									
Ease of Integration	8	2	16	2	16	2	16	3	24
Ease of Assembly	6	3	18	2	12	3	18	4	24
Minimal cone body modifications	8	2	16	2	16	2	16	3	24
Main conveyor configuration	8	4	32	2	16	3	24	2	16
Expandability	6	3	18	4	24	2	12	5	30
<b>Cost</b>	6	3	18	3	18	2	12	3	18
<b>TOTALS</b>			246		280		252		322
<b>RANK</b>			4		2		3		1

### 3.2.2 The Cone Stack Retention Subsystem

Selection of the forklift concept necessitated an independent stack retention subsystem to support cones in the storage framework. The subsystem requirements included complete support of the cone stacks and coordinated operation with the forklift during stack storage and retrieval. The retention device and actuator designs were identified as aspects of the subsystem requiring conceptual development. A similar selection process was used, which involved establishing design criteria, developing and defining the design concepts and comparing the designs to identify the concept best suited for further development.

### 3.2.3 Retention Devices

#### 3.2.3.1 Selection Criteria and Considerations

Two specific requirements were made of the retention device concepts. As specified in the project scope, all cone stacks with width dimensions between 35.5 cm and 40.6 cm (14.0 and 16.0 inches) must be secured by the devices. The retention devices were also required to retract outside the 40.6 cm (16.0 inch) stack envelope width to avoid obstructing the forklift operation. Furthermore, retraction of the middle devices was limited to within the 6.3 cm (2.5 inch) gap between each pair of cone stacks. Several, more general criteria identified other important

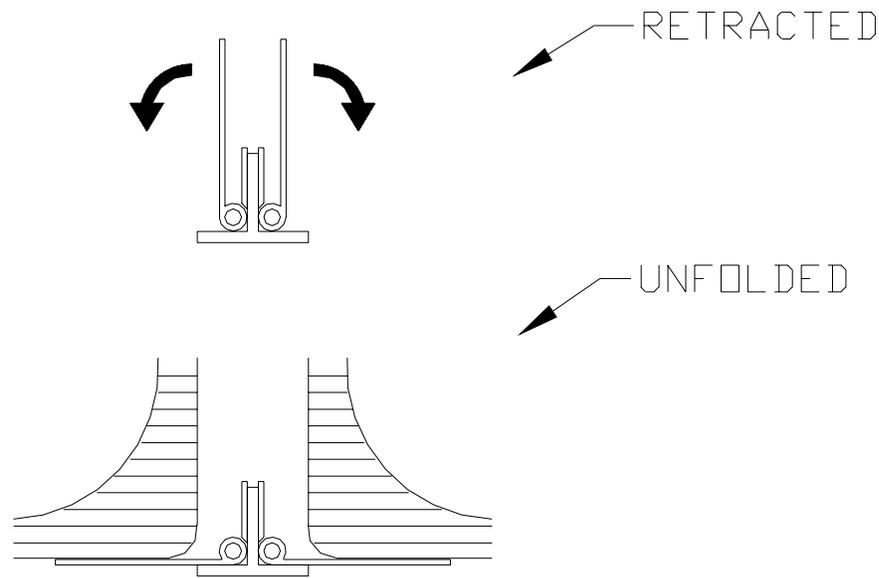
factors of the retention device design. It was desirable to minimize the number of actuators required to operate the devices. A robust, yet optimized design was needed to properly support the cone stacks and minimize component weight. A simple design was also desired for ease of component assembly and integration. Consideration was also given to the required number of devices and parts, commercial availability of components, cost, and implications on the structural framework design.

### 3.2.3.2 Design Concepts

Three different retention device concepts were considered. Fundamentally though, the concepts were similar as the fork design restricted the number of options for retaining the stacks. The fork beams supported the center section of the bottom surface of the cone stack, leaving the bottom outer edges of the stack exposed. Therefore, the open surface area was the target for designing the retention support device. Additional features were also needed to secure the stack laterally. The individual retention device designs were the primary focus of conceptual development, however, serious consideration was also given to the complete assembly of devices, which required coordinated actuation.

### 3.2.3.3 Piano Hinge Concept

The piano hinge design concept consisted of a number of unequal leaf piano (continuous) hinges. Each hinge would span the length of the cone stack and be supported by an angle beam. One leaf of the hinge would fasten to the vertical, upward directed web of the angle. The other leaf would be free to rotate between the horizontal and vertical web of the angle. The horizontal web was intended to act as a mechanical stop and provided support and stability to the outstretched leaf under the load of the cone stack. Several means of actuating the free leaf were conceived and determined feasible. Overall, eight hinges were necessary for this design; one pair for each stored cone stack.

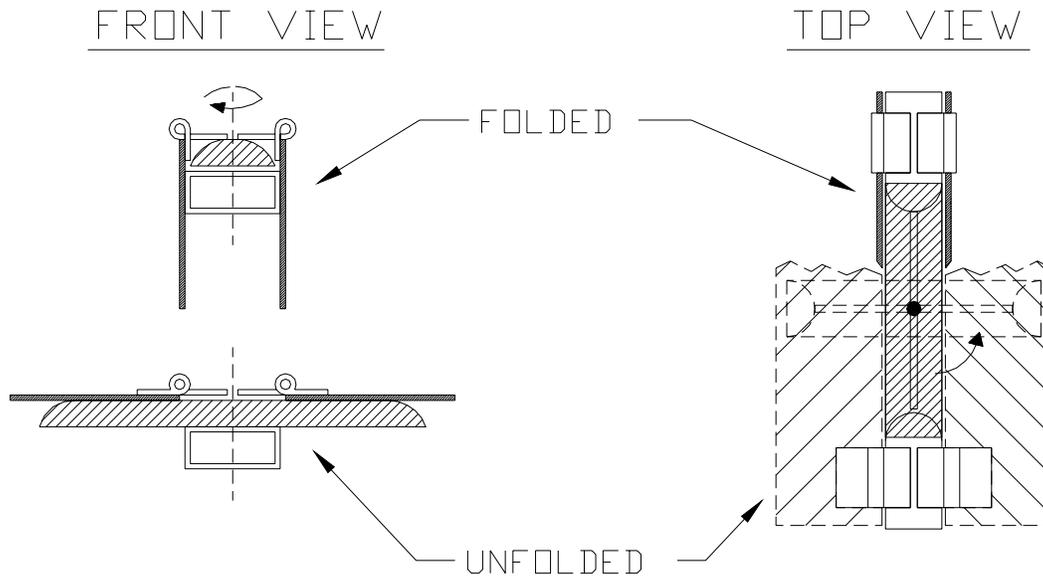


*Figure 3.12 Piano Hinge Concept*

Several aspects of the continuous hinge concept made it very desirable. The design was simple, consisting only of one primary component per device, apart from the actuation parts. Likewise, the hinges were easy to assemble. Custom specified hinge features afforded a design to accommodate all cone sizes and it was confirmed that the size required was commercially available. Depending upon the hinge specifications, though, high cost was unavoidable for special fabrication orders. The compact nature of the hinge was ideal for retraction within the small gap between the stacks and several options for hinge retraction were available. Preliminary calculations indicated the free leaf to be adequately robust to support the cones. The only potential disadvantages of the piano hinge concept were moderately high weight and questionable use of the hinges for a load bearing application unlike its typical utilization.

#### 3.2.3.4 Cam Follower Concept

This design concept consisted of rotary actuated cams and a follower plate to support the cone stack along its length. The plate would attach to an anchored hinge joint along one side to provide a rotational degree of freedom. A series of cams would be positioned at several locations along the length of the plate underneath the hinged connection. Each cam would be made of a narrow, oblong block with a top surface curvature to function as the cam contact surface for the follower plate. When rotated about a vertical axis, the curved surface of the cams would engage the plate, raise it to a horizontal orientation and support it under the load of the cone stack. A simple rectangular cross tube within the storage framework would be sufficient to support the cam follower components. A total of eight plates and eight corresponding hinged connections would be required apart from the cams and actuation components.



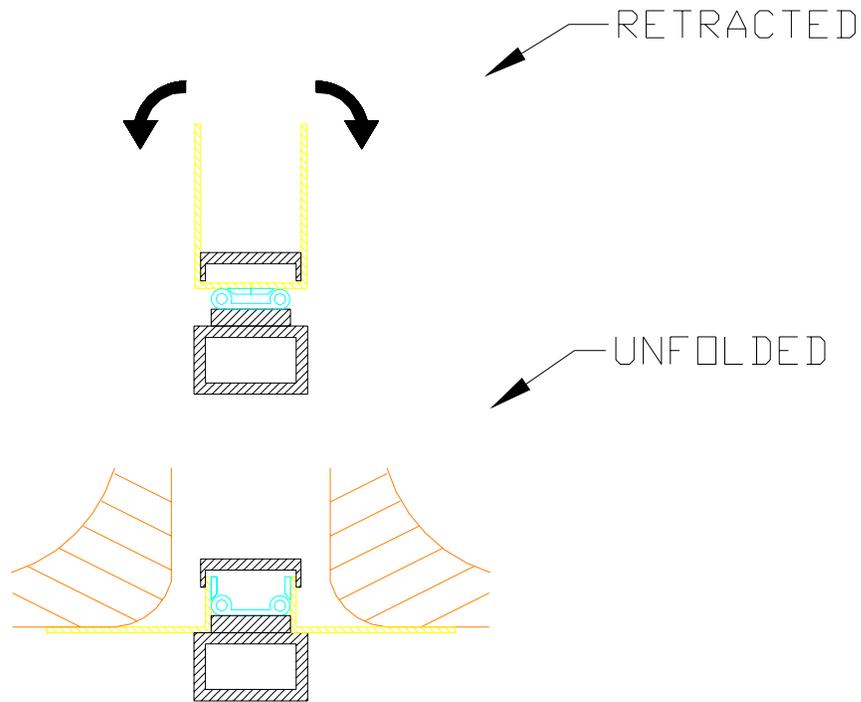
*Figure 3.13 Cam Follower Concept*

The cam follower concept was promising, as it offered a reduction in the height of the structural framework. Because the follower plates retracted downward, unlike the piano hinges, the vertical gap between stored cone stacks could be reduced considerably. The design was also simple, lightweight and would be relatively inexpensive to fabricate, apart from the custom designed cams. There was also some concern about concept, though. The robustness of the plates and hinged connections were questionable, due to the tight retraction envelope constraints. Integration of the cam actuation components created potential complexities. A number of actuators might be required, which created challenges associated with layout design and subsystem assembly. Furthermore, the high complexity and large quantity of custom cam blocks was a big disadvantage of the concept.

### 3.2.3.5 Retractable Angle

This concept design used retractable angle iron bars to support the cones along the length of the stack. The angle would mount to an anchored hinge and rotate between a vertical and horizontal orientation. Structural members would mechanically constrain the angle in the lowered and retracted positions and support it under the load of the cone stack. Custom machined angles would be required between the stacks to ensure both angles effectively retracted without interference. A number of angle rotation concepts were available for rotating the angles up and down.

The main strength of the retractable angle concept was the robust components, which provided a secure method of storing the stacks. Like the continuous hinge concept, similar means of angle rotation could be used, which would greatly simplify actuation of the entire angle array.



*Figure 3.14 Retractable Angle Concept*

The greatest weakness of the design was integration and assembly onto the structural framework. Very tight tolerances would be required to assemble the angles and place the hinged joints so that the angle properly engages the mechanical constraint surfaces. The predicted difficulty in assembly was also a serious consideration. The retractable angle design complicated the structural framework by requiring two cross member supports for each retractable angle assembly. Weight was predicted to be high and the cost moderate.

### 3.2.3.6 Concept Comparison and Trade Off Analysis

Using the design criteria as a standard to subjectively evaluate each concept, as shown in Table 3-3, the continuous hinge concept was identified and selected as the most qualified design. Overall, the simplicity of the continuous hinge design set it apart from the other concepts. It offered a prepackaged solution to securing the stack, which was believed to be invaluable in quickly and effectively developing the system prototype. These advantages clearly outweighed the disadvantage of higher cost and a slightly higher weight. The options for hinge actuation were another strength of the design, as discussed in the next section.

Table 3.3 Stack Retention Device Decision Matrix

SELECTION CRITERIA	WEIGHT FACTOR	CONCEPTS					
		CONTINUOUS HINGE		CAM FOLLOWER		RETRACTABLE ANGLE	
		Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
<b>Requirements</b>							
Secures all cone sizes	10	YES	-	YES	-	YES	-
Retracts within 6.3 cm (2.5 inch) gap	10	YES	-	YES	-	YES	-
<b>Device Design</b>							
Simple	8	5	40	4	32	3	24
Weight	6	2	12	4	24	3	18
Robust	8	4	32	3	24	4	32
Means of actuation	8	4	32	2	16	4	32
Commercially available parts	6	5	30	4	24	3	18
<b>Device Integration</b>							
Impact on structural framework	4	3	12	3	12	2	8
Ease of Assembly	6	5	30	2	12	3	18
<b>Cost</b>	6	2	12	3	18	3	18
<b>TOTALS</b>			200		162		168
<b>RANK</b>			1		3		2

### 3.2.4 Retention Device Actuator Units

Developing an effective and reliable actuator configuration design was equally important to the retention device selection process. The actuation concepts were considered separately to ensure thorough and complete development of the retention system components before proceeding with fabrication of all eight sets of devices. The primary objective in designing the actuation components was to achieve complete rotation of the hinges in a simple manner and to minimize the number of actuators required. There were a couple of ways to rotate the hinges and it was important to carefully evaluate and compare each idea. Functional demonstration of the design was also critical to offer insight into the dynamic effects and potential problems that are easily overlooked in designing systems with moving parts. This also afforded the opportunity to suggest and implement modifications before proceeding with fabrication of the actuation devices. Therefore, instead of following a qualitative selection process, as used for the retention device designs, the hinge actuation concepts were prototyped and tested.

#### 3.2.4.1 Actuation Criteria and Considerations

Each hinge actuator unit was required to rotate the free leaf of the hinge outside of the cone stack envelope. Components associated with actuation also needed to remain outside of the vertical path of the cone stack and the forks. Because of the large quantity of identical components, great efforts were made to use commercially available parts. It was also important to simplify the design to minimize the number of parts, optimize weight and reduce costs. Finally, it was desirable to minimize the number of actuators.

### 3.2.4.2 Design Concepts

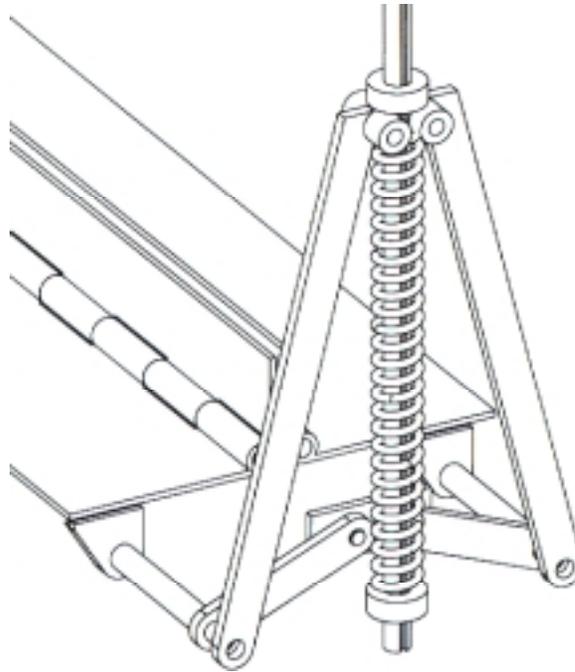
Two hinge rotation design concepts were tested and compared: a cable actuation design and a retraction mechanism. An emphasis was placed on designing the individual hinge rotation unit, which would become the fundamental building block for all eight sets. Larger assembly layout configurations were then considered to explore options for linking the rotation devices and minimizing the number of actuators required.

### 3.2.4.3 Cable Actuation Design

The cable actuation design made use of a thin cable to transfer the linear motion of a cylinder into rotational actuation of the hinge. The cable attached to the free leaf of the hinge, wrapped around a fixed pulley block and connected to a cylinder at the front end of the system framework. The cable functioned to retract the hinge only. Therefore, a torsional spring was mounted to the hinge for the purpose of unfolding the free leaf upon retraction of the cylinder. Several other components were required as part of the cable actuation design assembly, including springs to tension the cable and absorb the excess stroke of the cylinder, pulleys to redirect the cable, and clamps. To achieve full retraction of the hinge under the tight spatial constraints, the pulley block was designed with a fixed, pulley-like wedge for the cable to slide over. A teflon coated cable and teflon pulley wedge were selected to minimize the friction resistance created during actuation of the cable. Complete actuation of all the hinges was possible using two cylinders, one per storage level. The four cables at each level joined together at the front end of the structural framework and attached to the cylinder piston to achieve simultaneous hinge actuation.

### 3.2.4.4 Retraction Mechanism

The retraction mechanism concept consisted of a backward driven slider crank mechanism for rotating the hinge. The mechanism was designed to actuate the free leaf of the hinge up and down from the front end of the structural framework. Like the cable actuation design, the mechanism converted linear actuation from a cylinder, directly to a slider and indirectly into rotational motion of the hinge. A complete set of mechanism parts was required for each hinge except for the middle pairs, which shared several common components. The retraction mechanism design lent itself to an interconnected configuration that required only a single cylinder to actuate all the hinges. To achieve this result, a spring component was added as part of each linkage to introduce some compliance into the mechanisms. A connecting framework was also designed to link the mechanisms to the cylinder.



*Figure 3.15 Retraction Mechanism Design Concept*

#### 3.2.4.5 Testing Results and Conclusions

Single hinge actuation units were fabricated and tested for each rotation design concept. The primary objectives were to understand the dynamic motion and interaction of the components and observe the effects of repeated actuation. Both devices were actuated manually, which was sufficient to evaluate the functional performance of each design and simplified the set up.

The cable design configuration effectively actuated the hinge. The free leaf was easily retracted by pulling the cable and successfully unfolded by the torsional spring. The pulley block design proved small enough to fold the hinge outside the cone stack envelope. Actuation of the set up was smooth and relatively effortless. Though the design successfully achieved hinge rotation, there were issues which required some attention. First, the torsional spring used for demonstration and testing proved far too rigid, which led to an increased cable tension during hinge retraction. It was necessary to select an optimum spring, which would effectively open the hinge and offer the least amount of resistance to the cable actuation. Second, the teflon cable coating quickly frayed during repetitive hinge actuation. This effect was a significant concern and likely resulted from the contact conditions between the cable and the pulley wedge. The cable rubbed against the blunt edge at the mouth of the wedge groove, which probably initiated damage to the surface of the coating. In addition, the friction force generated between the cable coating and the pulley wedge surface also contributed to the wear as a result of repetitive actuation of the cable.

There were alternative solutions to avoid fraying of the cable coating, most of which introduced new concerns, though. A rotating pulley, with increased diameter, would be the best solution to reduce wear. However, increasing the size of the pulley block assembly would

prevent complete retraction the free leaf. Using a plain steel cable was another possible solution. The teflon pulley wedge could then be replaced with a steel wedge. The disadvantages to this modification were increased friction between the cable and the pulley wedge and a smaller, long term potential for fraying of the steel cable.

In considering the layout options for coupling the cables to a single actuator, the configuration grew increasingly complex and required more accessories to properly join and route the cables. A simpler solution was to increase the number of actuators on each level of the system by dedicating one actuator for the middle pair of hinges and another for each outer hinge. Altogether, six actuators would be needed.

The retraction mechanism also successfully achieved hinge rotation, both up and down. Testing proved that the hinge was rigid enough to rotate the free leaf from one end. The free leaf was completely withdrawn from the path of the cone stack, which satisfied the primary intention of the mechanism design. Actuating the mechanism also verified the predicted dynamic response. Upon initial actuation, the spring deflected, applying a force to the slider block. The spring force resulted in a reaction between the linkage bar and the free leaf, which caused hinge rotation once the spring force exceeded the resisting reaction. Reverse actuation of the mechanism easily unfolded the hinge leaf, which resolved contrary initial predictions.



*Figure 3.16 Binding of Mechanism Slider*

After successfully demonstrating hinge rotation, the free leaf was clamped open to test the spring in the mechanism. Ideally, the spring would compress and absorb the stroke of the cylinder while the slider and linkage, being rigidly attached to the constrained leaf, would remain fixed. Upon repetitive actuation of the vertical rod, two potential problems were identified. First, the hinge leaf deflected significantly as the spring compressed. Even under a full load of cones, the leaf would likely experience some deflection at the end nearest the mechanism and, therefore needed stiffening. Second, a binding problem was observed between the slider bushing and the actuated rod. Compression of the spring, leading to high tensile forces in the mechanism link, created a substantial normal force between the bushing and the rod. The large friction

reaction forced the bushing out of the slider block. To eliminate the problem, it was necessary to insert a flanged bushing between the slider block and the spring or design a fastener to secure the bushing to the slider block. A connecting framework was designed to utilize a single actuator to activate all six retraction mechanisms. Successful testing of the mechanism validated assumptions made in designing the framework.

#### 3.2.4.6 Comparison and Selection

Testing provided great insight into the design and operation of both actuation concepts. The results were qualitatively compared to help select the best means for actuating the hinges. Several common design characteristics and functional similarities were identified between the two designs. Both the cable actuation concept and the retraction mechanism successfully actuated the hinge outside the specified cone stack envelope. A large number of off the shelf parts could be used for both designs and a low overall weight was estimated for each concept. The predicted development costs and overall costs were also comparable.

The two concepts were also dissimilar in a number of ways. The retraction mechanism clearly demonstrated a more robust design than the cable actuator. By effectively actuating the hinge from one end, the retraction mechanism design was less restricted by spatial constraints. The layout of the cable actuation components, however, was highly constrained, which limited design robustness and raised serious concerns about the durability and longevity of the subsystem. The retraction mechanism design required only one actuator. The cable concept required at least two actuators, potentially as many as six, which complicated the subsystem control, increased the power demands of the system and raised the overall cost.

After comparing the actuation concepts against the design criteria and evaluating the test results, the retraction mechanism design was the logical choice for actuating the hinges. Overall, it offered a solid and robust design with simplified control and ease of integration at a minimal cost.

#### 3.2.5 Summary

During the conceptual design phase, two critical subsystem designs were selected based upon design comparison analyses and preliminary testing. The forklift was identified as the most qualified vertical lift concept for use in storing and retrieving cone stacks within the system. The piano hinge and retraction mechanism were selected as the best suited designs for the cone stack retention subsystem. The main conveyor conceptual redesign was performed alongside the conceptual development of the vertical lift concepts. The new design amounted to a split belt configuration, which would be integrated as part of the forklift unit. Aspects of the storage structure were also confirmed in conjunction with the conceptual design of the retention device concepts. Overall, the conceptual design phase solidified the design of the primary components within the storage system. The detail design work was the final step remaining in developing a complete, initial design of the multistack system. The next section contains thorough documentation on the detail design work of the entire multistack system and elaborates upon the designs introduced.

### 3.3 Detail Design

The multistack system detail design work was performed following selection and establishment of the fundamental design concepts. Specification and optimization of numerous components was required to ensure successful function and assembly of the system. A somewhat iterative approach was taken in performing the detail design. Since a majority of the component designs were interdependent, it was necessary to make some assumptions early on to begin the process. Over time, as the details were filled out, many designs were revisited and modified as necessary.

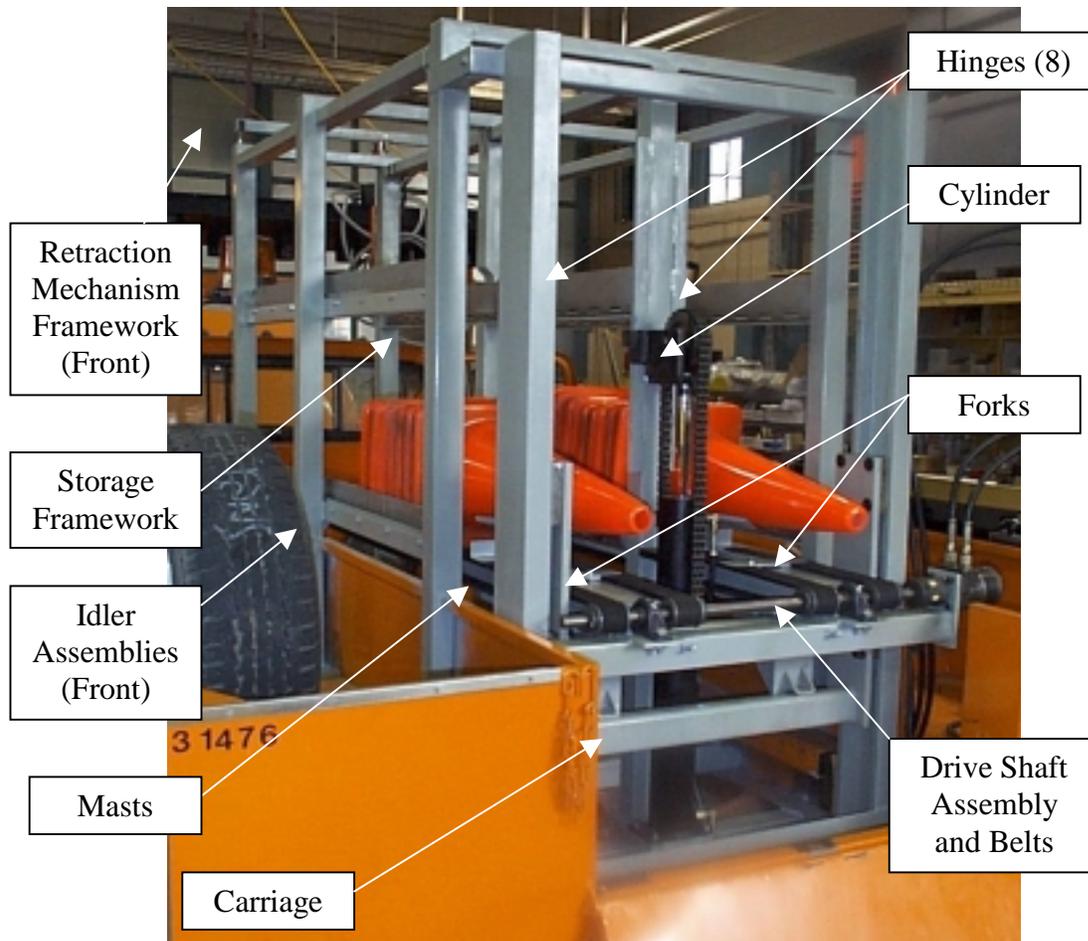
This section presents the outcome of the detail design phase with some explanation regarding the development process. The contents do not necessarily represent the order in which the design was accomplished or completed. In general, this account of the multistack design follows a methodical approach by discussing critical layout dimensions, component design details and functional purposes of the system subassemblies.

#### 3.3.1 System Overview and Component Identification

Before considering the details of the engineering design, it is beneficial to identify the primary components of the system, as seen in Figure 3.18, and to provide an overview of system construction. Extensive efforts have been made to simplify component interfaces and designs for ease of assembly. System assembly is presented here in a broad sense by the order in which the components are described.

The storage framework is a structural weldment to which many other components are attached. Along with a few other structural members, which are added to the cone body to assist in supporting the multistack components, it is constructed onto the vehicle first.

The retention system is made of three sets of components: the retention hinges, the retraction mechanisms and the actuated mechanism framework. The eight continuous hinges mount to the cross members of the storage framework and the actuated mechanism frame mounts to the front end of the storage structure. The mechanisms assemble onto the actuated framework and connect to the front end of the hinges.



*Figure 3.17 Complete Multistack System Assembly*

The forklift consists of three separate components: the lift cylinder, the carriage with supporting masts, and the forks. With the exception of the forks, all the components are placed rearward of the storage framework. The lift cylinder and mast weldment mount to the truck frame and attach to the storage framework. The carriage connects to the lift cylinder and assemblies inside the mast weldment. The forks mount to the carriage.

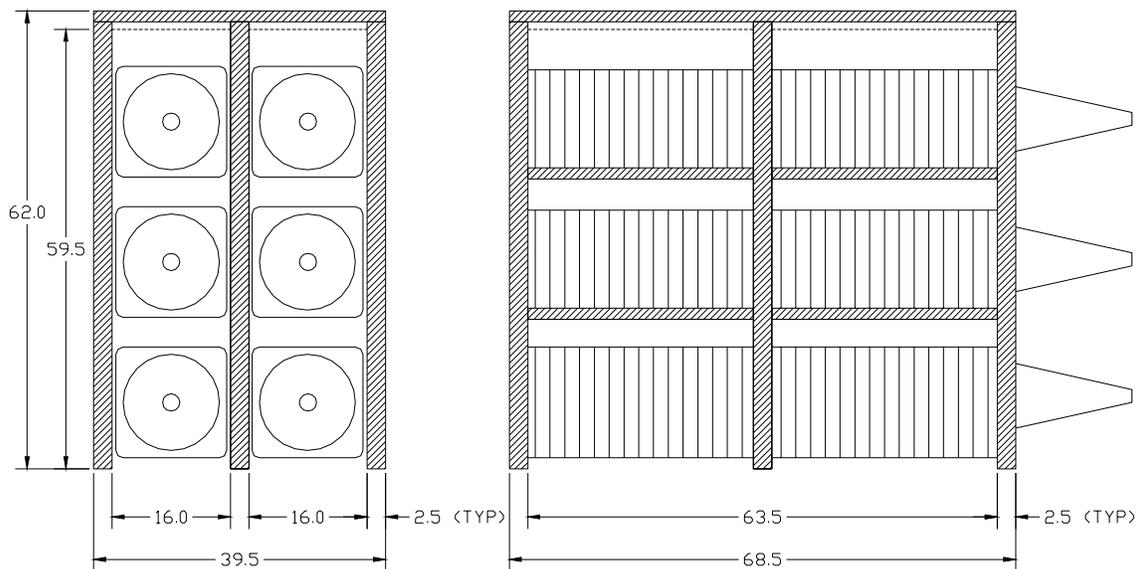
The main conveyor attaches to the forklift unit. An idler assembly fastens to the front of each fork and the drive shaft assembly mounts to the rear end of both forks. Two timing belts are assembled and wrapped around the top and bottom of the fork along its length.

### 3.3.2 Structural Framework

#### 3.3.2.1 Layout

The storage framework layout was created around the cone stack storage envelopes, which measured 40.6 cm (16 in) wide, 174.0 cm (68.5 in) long and 151.1 cm (59.5 in) tall. See Figure 3.19. The envelope dimensions were determined based upon a stack of 40 cones, having 40.6 cm

(16 in) square base dimensions and stacked three layers high. The stacks were horizontally oriented and positioned side by side with a 6.35 cm (2.5 in) gap in between. Longitudinal cross members within the framework measured 161.3 cm (63.5 in) long. They were supported at both ends and at the midpoint by vertical columns. The bottom layer of cross members was placed 45.7 cm (18 in) above the bed of the cone body, a nominal distance of 5.08 cm (2.0 in) greater than the stack. The added clearance prevented interference between the cones and lower retraction mechanisms. The top layer was positioned 54.6 cm (21.5 in) above the bottom members. An additional clearance of 14.0 cm (5.5 in) was required to raise the retrieved stacks high enough to retract the hinges without contacting the stacks stored above. The top of the frame was an additional 54.6 cm (21.5 in) above the top layer of cross members. Two plates were welded to the rear, middle upright member to function as a guide for the forklift cylinder piston. The top surface of the plates were positioned 11.4 cm (4.5 in) below the top of the cone stack envelope and the back surface of the plates were positioned flush with the rear surface of the storage framework



*Figure 3.18 Storage Framework and Cone Stack Envelope Layout Dimensions*

### 3.3.2.2 Detail Design

Several objectives and requirements were considered in designing the storage framework. Adequate support of the cone stacks was a must, as was unobstructed access for the cones and forklift unit within the stack envelope areas. Careful design of the framework members was important to optimize the overall size and weight. The frame also needed a high level of rigidity to which other components could be secured.

The cross member design was key to supporting the retention hinges. Several different cross sections were considered, but the most suitable option was an angle. With the angle oriented in an upright “L” shape, one leaf of the hinge could easily be fixed to the vertical leg of the angle.

The horizontal leg could then be used to support the free leaf under the cone stack load. The angle section was ideal to achieve 90-degree hinge rotation and it presented a low weight solution to support the cones. The width of the horizontal leg was specified as 3.18 cm (1.25 in), which was the maximum allowable width for the middle angles to maintain the 6.35 cm (2.5 in) gap width specified between stacks. A wall thickness of .305 cm (.120 in) was chosen to minimize weight and optimize the space available to mount and retract the hinges.

The angle cross members were supported at the ends and the midpoint, which cut the length of the unsupported section in half and significantly reduced the maximum deflection. Assuming a distributed load of 700.5 N/m (4 lb/in), it was determined that the maximum vertical deflection was .0064 cm (.0025 in) at positions one quarter and three quarters along the overall length of the member. The maximum angle of twist of the angle bars occurred at the same locations. It was computed to be 6.5 degrees due to the torsion created by the eccentric load of the cone stack weight. While the deflection was minimal, the maximum twist in the angle was larger than desired. Increasing the angle thickness was not recommended, therefore, it was proposed that a stiffener be added underneath each angle to reduce the twist to under 3 degrees.

Overall, eight identical angle members were required: four outer angles and four middle angles. The middle angles were welded back to back in pairs. A total of nine upright supports were used in the storage framework: three to support the front end of the angles, three to support the back end of the angles and three to support the middle. The members selected were tubes of 2.5 in square sections. The width was chosen primarily to match the width of the middle angles. It also provided a larger area for mounting the structure to the cone body and truck frame. The indeterminate reactions within the framework are assumed to be small judging from the magnitude of the reactions at the welded angle joints. Under assumed worst case loading conditions, the bending stresses in the vertical members amounted to 10.6 MPa (1.54 ksi), which gave a safety factor of 19.5. Buckling was not an issue given the low vertical loading. Therefore, the storage framework was designed sufficiently robust.

### 3.3.3 Retention System

The retention system consisted of eight continuous hinges, eight retraction mechanisms, a mechanism framework and an actuator. The retention system was responsible for supporting and securing the cone stacks in storage. Therefore, it was critical that the components be designed for a high level of robustness and operational reliability.

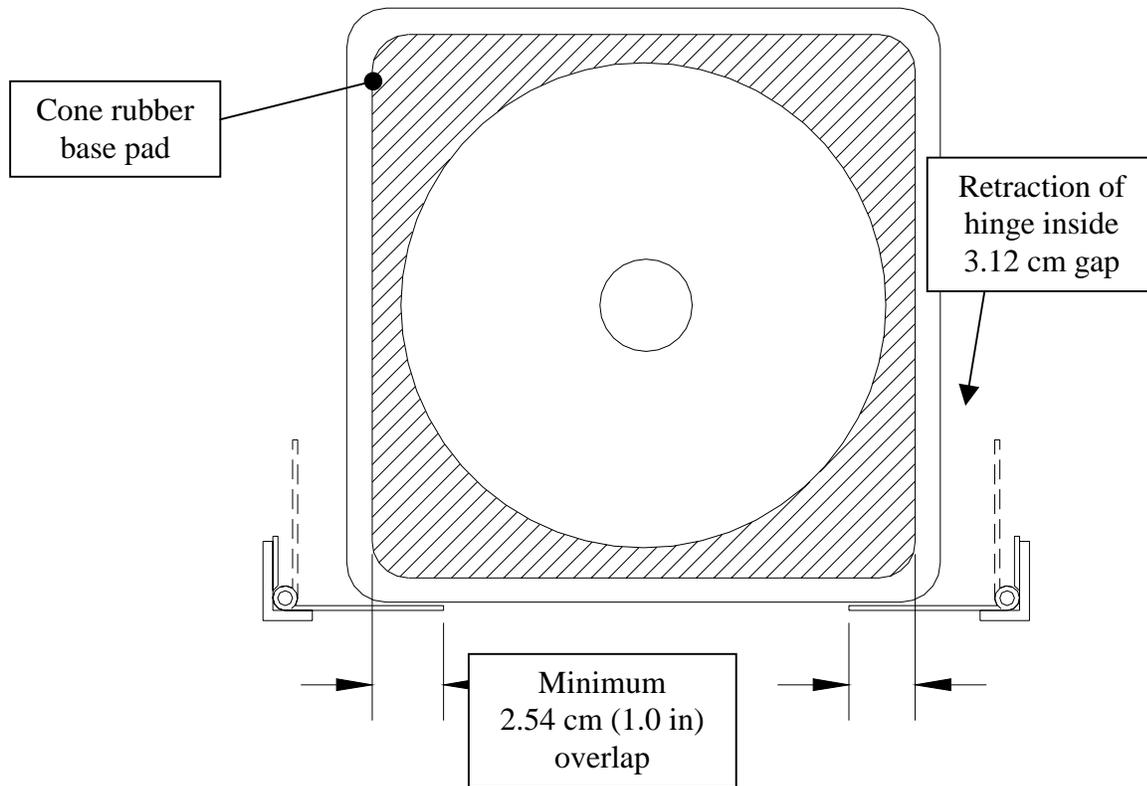


Figure 3.19 Minimum stack support and hinge retraction requirements

### 3.3.4 Continuous Hinges

#### 3.3.4.1 Design Specifications

The continuous hinges needed to support the full cone stack load with a maximum deflection under .159 cm (.063 in). The weight of each stack was split between two parallel hinges and distributed along the length of the free leaves. It was required that each pair of hinges accommodate cones with width dimensions ranging from 36.8 cm (14.5 in) to 40.6 cm (16.0 in). A 2.54 cm (1 in) deep minimum overlap along the length of the hinge, as illustrated in Figure 3.20, was required between the rigid, rubber base pad of the cones and the supporting hinge leaf to ensure the stacks were adequately secured. The width between each open pair of hinges needed to be large enough for the fork beams to pass between unobstructed. The hinges were also required to completely retract to within a 3.2 cm (1.25 in) gap allotted in the storage framework to prevent interference with the cones during forklift operation.

#### 3.3.4.2 Layout Design

The front end of the hinges mounted .318 cm (.125 in) rearward of the front, structural uprights. The fixed leaf of each hinge was securely fastened to the vertical leg of the angle cross members. The hinge leaves were shimmed off the angle iron legs using 20 gauge sheet metal to eliminate resistive contact during hinge rotation.

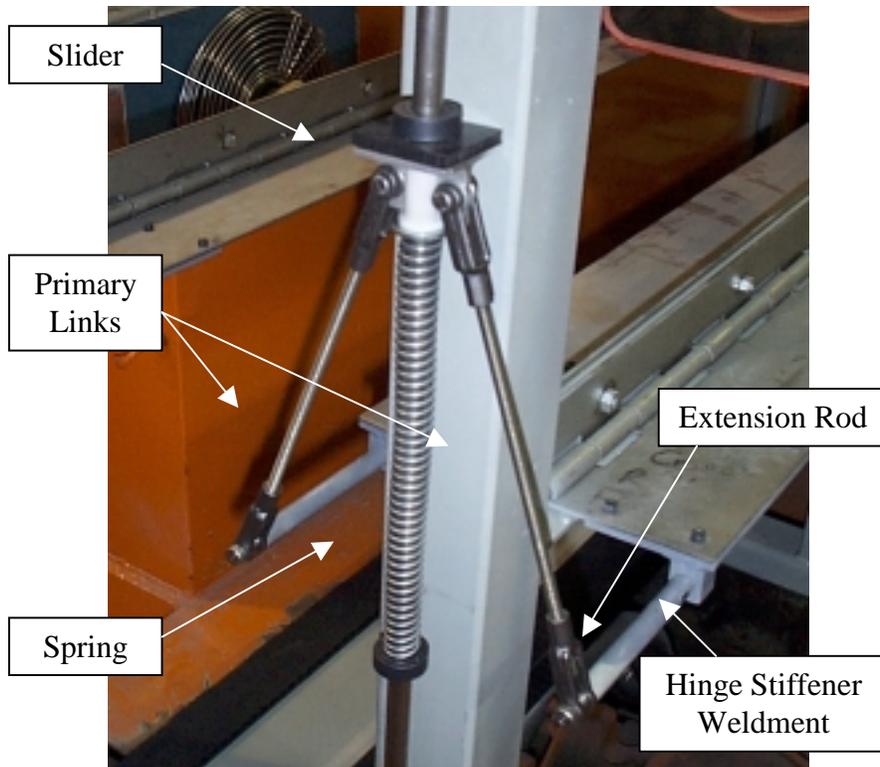
### 3.3.5 Hinge Retraction Mechanism

#### 3.3.5.1 Specifications

The primary functional requirement of the retraction mechanism was to rotate the free hinge leaf 90 degrees between a horizontal orientation and a vertical, upright orientation. The layout of the mechanism components was limited by the storage framework and restricted from protruding into the cone stack clearance envelope. Therefore, each mechanism mounted to the front surface of the storage framework, outside the cone stack and fork envelopes. The front idler assembly, which attached to the front end of the forks and measured 25.4 cm (10 in) wide, prevented the mechanism width from exceeding more than 7.62 cm (3.0 in) on either side of the vertical, structural members. For the mechanism array design it was necessary to limit the height of each mechanism to no more than the height of each storage level. Finally, the mechanisms were required to absorb the actuation stroke of the cylinder when the hinges were loaded with cones.

#### 3.3.5.2 Mechanism Components

The primary components of the mechanism included a slider block, a connector link, an extension rod, and the free leaf of one hinge. The slider block housed a bushing, which freely slid on a vertical rod. The connector link consisted of two yokes and a threaded rod. The top end of the connector link pinned to the slider block and the bottom end attached to the extension rod using a shoulder bolt connection. The extension rod rigidly mounted to the free leaf of the hinge as part of a stiffener plate weldment. The free leaf constituted the final mechanism link. Its motion was constrained by the fixed pin joint of the hinge. Several other parts were required for proper mechanism operation, including two collars and a compression spring, which were assembled onto the vertical rod. One collar clamped directly above the slider block. The other clamped beneath the spring. The top of the spring was placed in contact with the bottom of the slider block and the bottom of the spring rested on top of the lower collar. Eight sets of components were used to completely actuate all eight hinges. Only three vertical rods and one actuator were needed. The view in Figure 3.21 identifies the mechanism components.



*Figure 3.20 Middle, Double Retraction Mechanism*

### 3.3.5.3 Layout

The mechanisms mounted to the front of the structural framework. Front end actuation of the hinges was the only option because of obstructions created by the forklift unit at the rear. Mounting outside of the framework was also necessary to configure the mechanisms for actuation by one cylinder.

A majority of the components were assembled onto the vertical rods of the mechanism framework, including the slider blocks and springs. The rod centerlines were offset 2.22 cm (.875 in) forward of the front surface of the storage frame. The remaining mechanism components were assembled in the same plane. The hinge extension rod extended rearward from the mechanism plane to the hinge leaf.

Placement of the rod onto the hinge was critical to the overall mechanism layout and operation. A spacer block was used to offset the rod centerline 2.22 cm (.875 in) from the bottom of the free leaf. This allowed the hinge to rotate completely without the rod contacting the structural tube. The spacer block and rearward end of the rod were required to mount within 2.54 cm (1.0 in) from the front end of the hinge to avoid interference problems with the cone stack during lift operation. The rod centerline was placed 1.59 cm (.625 in) radially inward from the outer edge of the free leaf to prevent interference with the front idler assembly. The remaining layout parameters were driven according to the detail design performed for the mechanism components, which is described in the next section.

#### 3.3.5.4 Detail Design

The mechanism design closely resembled a reverse driven slider crank linkage. The vertical rods were actuated up and down, transmitting motion through the spring to the slider block, which resulted in hinge rotation. Though the spring component added some functional complexity, the input motion of the slider was prescribed, resulting in the desired output motion of the driven crank (free leaf). The biggest challenge associated with developing the design was incorporating the required multifunctional characteristics of the mechanism. The mechanism needed to rotate the free hinge leaf, yet also absorb the actuation stroke when the hinges were supporting cones. To achieve dual functionality, a compression spring was included in the design to interface between the mechanism slider and the actuated vertical rod. The spring was designed for an optimum stiffness to effectively actuate the slider and yet compress with minimum resistive force. Selecting the spring characteristics drove most of the mechanism detail design and helped solidify the mechanism parameters such as the actuator stroke length, the mechanism link length and the slider geometry.

The actuator stroke was most influenced by the mechanism kinematics. The computed change in height of the extension rod during complete hinge retraction was 11.51 cm (4.53 in). In addition, an even larger stroke length was needed to account for compression of the spring, which resulted from actuation of the hinge. Therefore, a 15.24 cm (6.0 in) stroke was specified. Under normal mechanism operation, the spring would compress approximately 3.81 cm (1.5 in). The internal spring force created by the compression of the spring needed to exceed the final force required to complete hinge retraction.

When the hinges were open and supporting cones, the springs would be required to absorb the 15.24 (6.0 in) actuator stroke during mechanism actuation. To achieve this result, compression springs having lengths between 10-12 in. were chosen. The inside diameter of the spring was required to be greater than 1.27 cm (0.5 in) to fit over the vertical rods.

The final step in selection of the springs was obtaining the stiffness constant. The mechanism height was first designed to perform the static force analysis throughout the actuation range. With the bottom of the spring positioned evenly below the hinge pin joint and assuming a spring length of 28 cm (11 in), the primary mechanism link was computed to be 29.2 cm (11.5 in) long. After finalizing the mechanism geometry, the relationship between the spring force and the hinge rotation was derived. Figure 3.22 shows the curve describing the relationship. The final spring force (hinge angle = 90 degrees) was the critical value in selecting the spring constant. Assuming a spring deflection of 3.81 cm (1.5 in) at full stroke to be 1751 N/m (10 lb/in). The actual spring constants of the springs selected were adjusted to account for friction in the pin joints. The middle springs were specified with spring constant values of twice the computed stiffness since they functioned for two mechanisms simultaneously.

Detail design of the mechanism components was optimized for the most extreme loading conditions. Maximum forces were generated when the mechanism was actuated, the hinges were loaded and the spring completely compressed. A maximum axial, tensile force of 350 N (78.7 lb) was created within the main link, resulting in a normal stress of 10.6 MPa (1.53 ksi) in the threaded rod. A resulting end load of 350 N (78.7 lb) was applied to the cantilevered extension rod from the primary link. To minimize the deflection to under .076 cm (.03 inches), a rod

diameter of 1.27 cm (.5 in) was used. Overall, low stress conditions existed in the mechanism components.

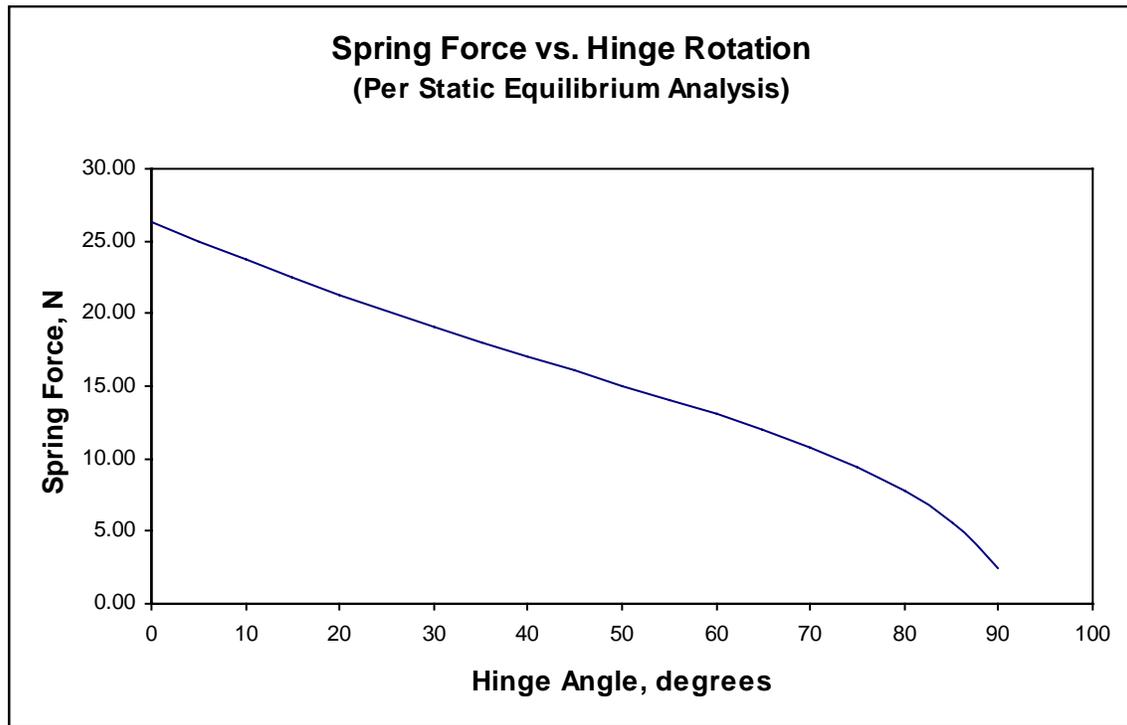


Figure 3.21 Force vs. Rotation Relationship of the Retraction Mechanism

### 3.3.6 Mechanism Framework and Actuator

A framework was needed to join the mechanisms to the actuator. The mechanism framework consisted of three vertical rods and a top cross bar. The layout closely mirrored the front end layout of the storage framework. The components transferred the motion of the cylinder stroke to the mechanisms. Figure 3.23 shows the framework configuration

#### 3.3.6.1 Layout

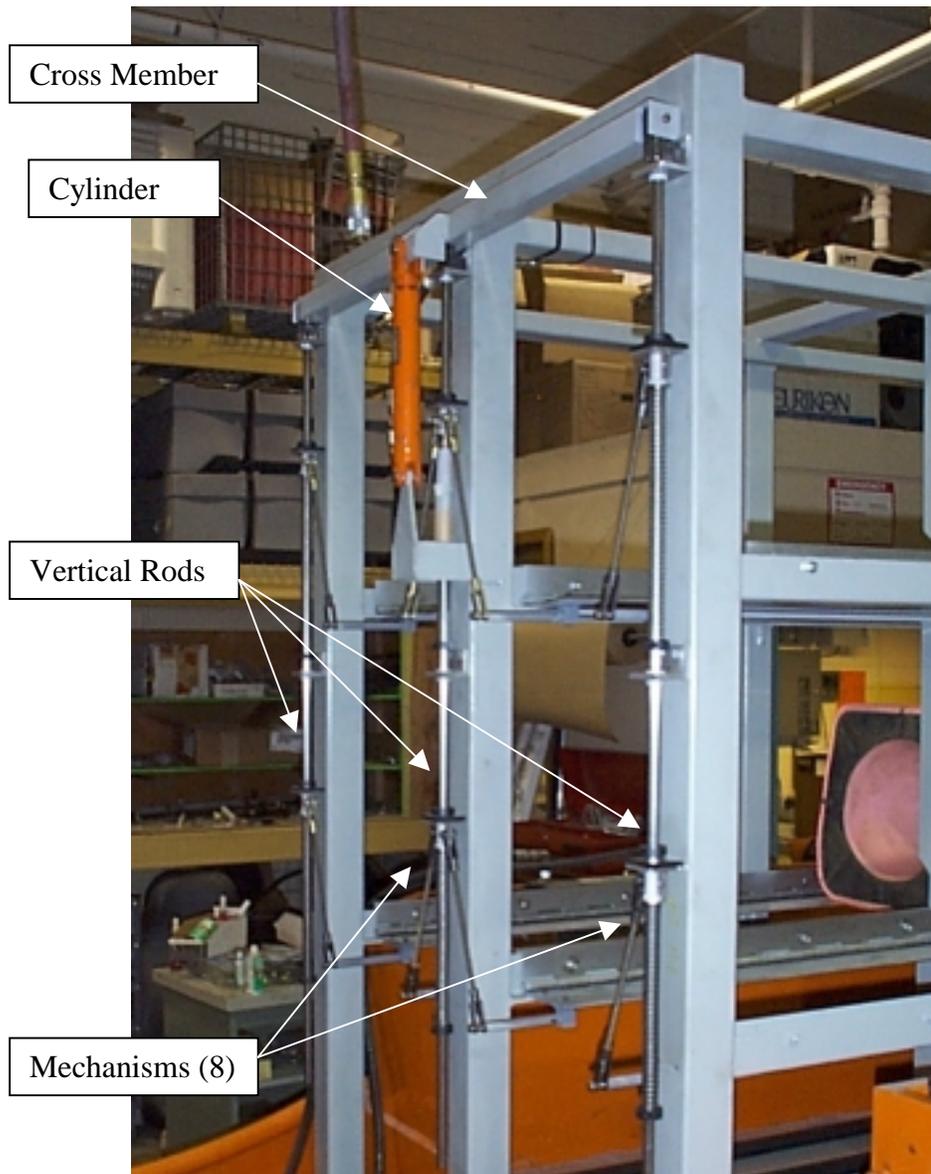
The vertical rods were set in bushing blocks which mounted to the front end of the storage framework. The blocks laterally centered the rods forward of the respective vertical tubes. The rod centerlines were offset 3.81 cm (1.5 in) forward of the front surface of the storage framework. The bushing blocks were mounted 15.24 cm (6.0 in) above the top of each slider to prevent interference during hinge actuation.

#### 3.3.6.2 Detail Design

The mechanism framework was intended to achieve simultaneous actuation of the eight retraction mechanisms. The actuation cylinder drove the framework up and down, compressing the mechanism springs and thus activating the hinges. The mechanism framework components

were designed for compatibility with the retraction mechanism components and loading conditions.

The vertical rod design was performed alongside the mechanism design. Many factors played a part in determining the rod diameter. It needed to be small to minimize the bushing hole in the slider so that the hinge pin and the joint between the slider and the primary link could be vertically aligned. The rod diameter also needed to be sized compatibly with the inner diameter of the mechanism spring. Finally, a large rod cross section was desired to minimize axial stress and deflection under the maximum loading conditions. To fulfill these requirements, a rod diameter of 1.27 cm (.5 in) was selected. The remaining clamps and bushings, in both the slider and the fixed alignment blocks, were selected accordingly. A plate was welded to the top of each vertical rod. The plates were used to connect the rods to the top cross member.



*Figure 3.22 Retraction Mechanisms, Framework and Cylinder*

The cross member consisted of a straight tube with end caps on both sides. Holes and openings were added for attaching the vertical rods and actuation cylinder. Functionally, the cross member linked the vertical rods and the cylinder. Therefore, it was important that it have a high stiffness and rigidity. Given the maximum axial load in the rods, the member was designed for a maximum deflection of .076 cm (.03 in). A 3.81 cm (1.5 in) square tube met the deflection specification.

The actuator cylinder mounted to the front, middle upright of the storage structure, forward of the mechanism framework. The clevis mount underneath the cylinder body pinned to a boss weldment. The weldment was anchored to the storage framework between the top and bottom, middle mechanisms. The piston head then pinned to the cross member. The cylinder bore diameter required to actuate all eight mechanisms at the maximum loading condition was computed to be 2.0 cm (.78 in). A 2.54 cm (1.0 in) diameter bore was selected for availability and cost reasons to ensure a safety factor greater than 2.

### 3.3.7 Forklift Unit

The forklift unit design was broken up into sections related to the three primary components, which are the lift cylinder, the forks and the carriage with masts. The forklift unit was the critical piece responsible for inserting and retrieving cones into the storage framework. Therefore, reliable functionality was key. Commercial forklift component designs were used as a resource in developing the details of the multistack forklift unit and producing an effective design.

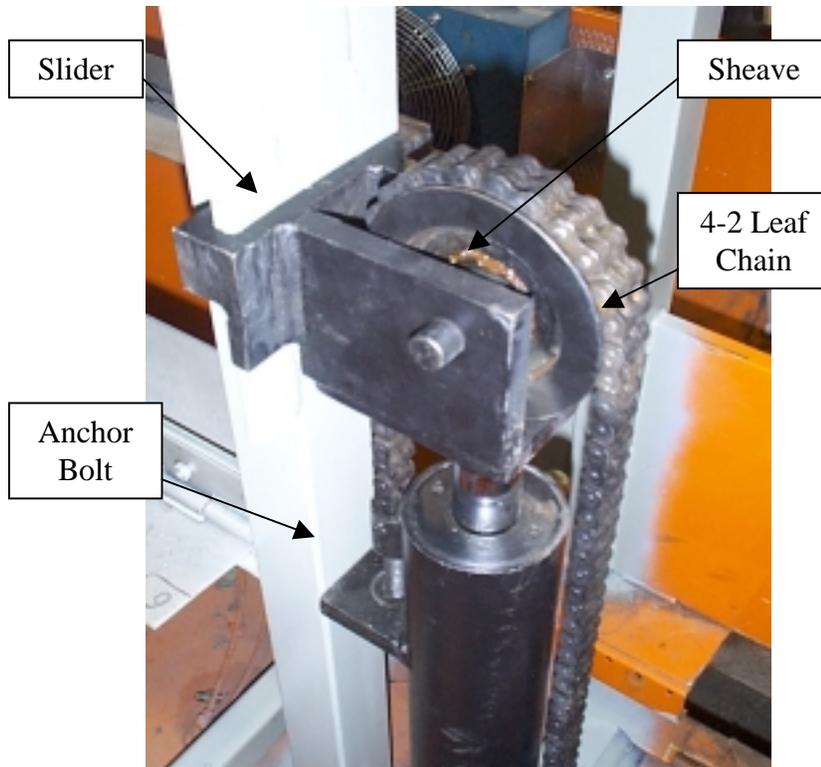
### 3.3.8 Lift Cylinder

#### 3.3.8.1 Design Specifications

The cylinder was required to raise two full size cone stacks to the top layer of the storage framework, which amounted to a lift height of 111.8 cm (44.0 in). The forks, the carriage and the cones weighed approximately 6005 N (1350 lbs). A cylinder design envelope was specified based on the layout of the cone stacks and other forklift features. The maximum lateral width of the envelope was limited to 15 cm (6 in) to prevent interference with the cone stacks at the rear end of the system. A longitudinal depth of 20 cm (8 in) was required to maintain a tight layout pattern between the cylinder, the carriage, and the structural framework.

#### 3.3.8.2 Layout

The cylinder body was centered behind the back middle upright of the support framework. The cylinder base plate mounted to an anchor plate that was structurally supported by an I-beam. The I-beam was seated in the truck frame channels. The axis of the cylinder sheave measured 65.72 cm (25.88 in) above the base height of the main conveyor. A groove in the slider that was mounted to the piston head fit over guide bars on both sides of the upright support member. The vertical centerline of the cylinder bore was positioned 10.5 cm (4.13 in) behind the back surface of the storage framework. The cylinder features are pointed out in Figure 3.24.



*Figure 3.23 Cylinder head components*

### 3.3.8.3 Detail Design

The key features and attributes of the cylinder, which required careful design and specification, were the bore diameter, the stroke, the sheave diameter, the chain type and size, the chain configuration, and the slider design. The first decision made was to use a single chain configuration. The single sheave was the only option to meet the maximum envelope width specification. However, it required a larger longitudinal depth and stronger chain to bear the entire weight of the loaded forklift. The sheave was centered above the piston and the chain passed over the cylinder from the front to the rear. Anchor bolts at both ends of the chain were used to secure the forward and rearward ends of the chain to the cylinder body and carriage.

The bore diameter was calculated based on the maximum loading conditions imposed on the cylinder. The weight of the loaded forklift was transferred to the leaf chain (at the connection with the carriage) as an axial, tensile load. Tension in the front and rear sections of the chain resulted in a compressive load applied to the cylinder. The magnitude of the load amounted to twice the tension in the chain, or 12,010 N (2700 lbs). Designing for a safety factor of 2, the cylinder diameter was computed to be 6.78 cm (2.67 in), assuming a hydraulic working pressure of 6.9 Mpa (1000 psi). A 7.62 cm (3 in) diameter bore was specified to be conservative and because of higher availability.

For the tension load in the chain, a 4-2 heavy leaf series was selected with a 1.27 cm (.5 in) pitch. The chain was rated for a maximum load capacity of 66.7 kN (15,000 lbs), giving a safety

factor of 11. The 4-2 leaf chain was also selected for its wide profile of nearly 2.5 cm (1.0 in) which prevented the chain from twisting near the contact points with the sheave. Compatible anchor bolts were selected for use with the chain. The chain length, including anchor bolts at both ends, was approximately 89 cm (35 in). The exact chain length depended on the anchor bolt length and cylinder dimensions.

After selecting the chain and confirming the cylinder diameter, the sheave diameter was specified to be 8.9 cm (4.5 in). This diameter ensured a minimum clearance of .64 cm (.25 in) between the vertically hanging chain and the outer wall of the cylinder body. A 2.54 cm (1.0 in) contact surface width was prescribed to match the leaf chain dimensions. Flanges were also specified to prevent the chain from derailing.

The slider design was functionally simple but had many features. A horizontal hole was designed to support the sheave pin. A vertical hole in the bottom of the slider fit onto the piston rod. The slider was then secured down with a retaining ring. The primary functional feature of the slider was a groove on the front end. The groove fit loosely over the two guide bars on the back of the storage framework and fully constrained the piston head to move vertically. The overall slider dimensions were kept within the cylinder envelope except for the groove feature, which protruded partly into the framework. The slider was included as part of the cylinder not only to constrain the top end of the piston, but also as a safety device to improve the forklift operation.

The stroke length of the cylinder was determined based upon the maximum vertical lift height of the cone stack which was, 111.8 cm (44.0 in). The chain coupling between the cylinder and carriage resulted in a two to one lift ratio. Therefore the stroke was required to be 55.9 cm (22.0 in) or greater. Accounting for deflections in the forklift unit, the total stroke length was specified as 58.4 cm (23.0 in). Proper specification of the sheave base height was essential to achieve full actuation of the carriage without interference between the carriage and cylinder sheave. Therefore, the cylinder was positioned vertically to offset the sheave 65.72 cm (25.88 in) above the main conveyor. This layout placed the sheave centerline 7.62 cm (3.0 in) higher than the carriage at full stroke length.

The cylinder was mounted to a support weldment consisting of an I-beam and a plate. The I-beam was seated inside the truck frame channels and placed forward of the cylinder to avoid interference with the forks. Therefore, a rigid plate was welded to the top of the I-beam and cantilevered rearward underneath the cylinder. The I-beam and plate were designed for several criteria. First, the top surface of the plate was specified to be 19.37 cm (7.63 in) below the cone body bed to achieve the proper base height for mounting the cylinder. Second, the weldment was designed for a specific stiffness to minimize the deflection of the plate at the point of cylinder support to .15 cm (.06 in). The results led to an I-beam 10.2 cm (4.0 in) deep, with 7.10 cm (2.80 in) flange width and .828 cm (.326 in) wall thickness. The plate was 1.90 cm (.75 in) thick, 10.2 cm (4 in) wide and 33.0 cm (13 in) long.

### 3.3.9 Forks

#### 3.3.9.1 Specifications

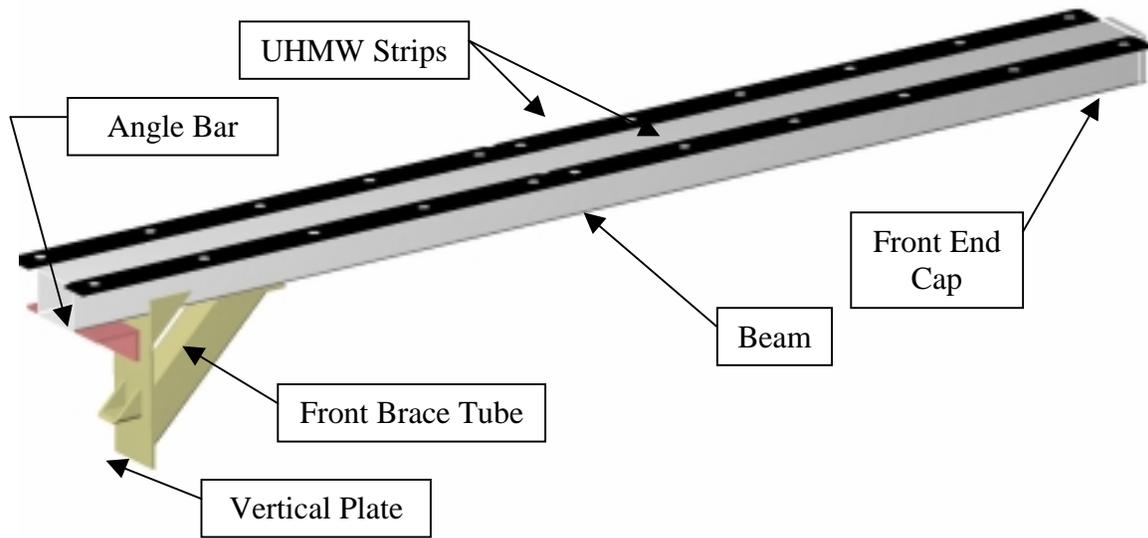
The weight of the horizontally oriented cones resulted in a distributed load of 1400 N/m (8 lb/in). The greatest loading of the forks was defined by the maximum stack length, 57.5 cm (62.0 in), and the front end location of the stack, which was 7.62 cm (3.0 in) rearward of the front idler axis. Under such conditions, the maximum allowable end deflection for the fork beam was 1.27 cm (.5 in). To ensure adequate clearance between the open retention hinges, the width of each fork was limited to less than 22.9 cm (9.0 in). A shallow beam depth was also desired to maximize the clearance underneath the forks at the cone body bed level. The fork design required a means to securely mount to the carriage. The forks also needed to support the redesigned main conveyor, which consisted of front and rear pulley assemblies, the drive shaft and motor, and four belts. In general, it was desirable to minimize the weight of the forks to reduce the load applied to the cylinder and optimize the forklift weight.

#### 3.3.9.2 Layout

The critical fork dimensions were established relative to the theoretical locations of the carriage and the front idler of the conveyor. The front edge of the carriage was positioned 190.5 cm (75 in) behind the front idlers and the main conveyor drive shaft was centered 209.5 cm (82.5 in) rearward of the front idlers. From these dimensions, the length of the fork beams were designed 198.5 cm (78.13 in) long. This provided options for mounting the conveyor assemblies to the front and back ends of the tube. The attachment between the carriage and forks was then calculated to be 184.5 cm (72.63 in) from the front end of the beams.

#### 3.3.9.3 Detail Design

The beam section, the means of attachment to the carriage and the features needed to support the conveyor demanded careful attention and design. It was necessary to identify the means of attaching the forks to the carriage first because the cantilevered beam length was needed before optimizing the fork section. The carriage set underneath the cone stack for clearance and functional reasons. Therefore, fork-supporting features were needed below the beam for mounting to the carriage. In developing the design, there were layout restrictions based on the location of the forklift cylinder supports and the truck and cone body frame components. The solution proposed in light of the design constraints was to use rigid braces that attach to the front and rear surfaces of the carriage. The brace to the front of the carriage was a triangular, gusset-like feature, designed to provide vertical support for the entire weight of the loaded fork. The rear brace was an angle bar that functioned to resist the bending moment of the load and physically secure the forks.



*Figure 3.24 Fork Assembly*

The triangular brace consisted of a thin vertical plate, a front tube and a rear horizontal plate. The vertical plate was welded to the bottom of the fork and designed to butt against the front of the carriage. The tube was made of heavy 5.08 cm (2 in) by 7.62 cm (3 in) tube. Oriented at a 45 degree angle, it was inserted and welded between the front surface of the vertical plate and bottom surface of the fork beam. The horizontal plate was positioned to seat on top of the lower carriage cross member. It was also welded to the vertical plate and reinforced with two gussets. The weight of the forks was transferred through the tube to the lower half of the carriage. To reduce bending in the fork beam at the tube support, a stiffener plate was added to the bottom of the beam. The layout relationship between the components is reflected in Figure 3.25.

The brace design supported the fork weight through areas of surface contact between the brace and carriage. The connection offered no opposing moment to prevent beam rotation, though. Therefore, an angle bar was welded to the bottom of the fork beam in a position of contact with the back surface of the top carriage member. Addition of the angle resulted in a horizontal support reaction from the carriage that opposed the moment from the cantilevered loading. An angle was chosen over other possible members to load the welds in shear.

After developing the general concept for the carriage interface features, the design details were hashed out based on the maximum loading conditions. The carriage support reactions were computed based on the maximum loading condition of the forks. Then the brace and angle were considered independently. Two assumptions were made in performing the analysis. First, only reaction forces, not moments, were created between the carriage and forks. Second, loading of the welded joints between the fork beam and the triangular brace and between the fork beam and the angle was purely in shear.

The horizontal reaction between the angle and the carriage was computed to be 14,250 N (3203 lb). This created an acceptable shear stress of 20.9 MPa (3.0 ksi) within the angle weld.

The horizontal and vertical reactions between the bottom member of the carriage and the brace were 14,250 N (3203 lb) and 2562 N (576 lb), respectively. After resolving the reactions in the triangular brace by making some simplified loading assumptions, the state of stress in the vertical plate and angled tube were computed to be 20.9 MPa (3.0 ksi) and 84.6 MPa (12.3 ksi), respectively. For the maximum loading condition, the vertical deflection of the tube amounted to less than .025 cm (.01 in). This proved adequate for creating a truly rigid support and had a minimal effect on the end deflection of the fork.

In designing the fork beam, two objectives were followed: maintaining an end deflection below 1.27 cm (.5 in) and limiting the overall fork width to less than 22.9 cm (9.0 in). The beam width was influenced by the conveyor design. Using small diameter pulleys the conveyor belt depth could be less than 6.35 cm (2.5 in). For this configuration layout, it would be difficult to wrap the belt around a beam, given the small clearance between the top and bottom of the belt. Therefore, the beam was required to fit between the belts. Assuming a nominal fork width of 20.3 cm (8.0 in), and given that the belt width was 5.08 cm (2.0 in) each, the beam could be 10.16 cm (4 in) wide. A rectangular tube section was chosen for its optimum area properties as compared to other beam sections.

A cantilevered beam model was developed for the maximum loading conditions of the fork. Using the model, the beam section was selected and optimized. Several viable beam depth and wall thickness combinations were computed, which satisfied the maximum allowable end deflection. The best solution was a 7.62 cm (3.0 in) depth with a .305 cm (.120 in) wall thickness. A maximum deflection of .65 cm (.256 in) was computed assuming a fixed support at the front edge of the brace tube. Under the most extreme loading, the maximum bending stress in the beam peaked at 69.9 MPa (10.1 ksi), giving a safety factor of 3.1.

To support the top half of the conveyor belts, a 20.3 cm (8.0 in) wide strip of 14 gauge sheet metal was welded to the top of the beam. The sheet upheld the belts under the load of the cones. UHMW strips were riveted to the top of the sheet along the path of the belts to create a low friction, sliding surface for optimum conveyor operation.

The final features included in the fork design were mounting holes for the conveyor assemblies. The front end of the tubes were capped and tapped with a hole pattern to which the idler assembly mounted. At the back end of the fork, holes were drilled in the top and bottom walls for mounting the drive shaft assembly. The hole placement was called out to obtain the specified pulley centerline spacing for the belt length selected.

### 3.3.10 Carriage and Masts

#### 3.3.10.1 Specifications

The primary requirement of the carriage and mast design was to rigidly support the fixed ends of the forks. Compliance within the structures was not permitted. Deflection of the carriage could result in no more than .160 cm (.063 in) of end deflection in the fork beam. Additionally, the components were responsible for attaining purely vertical motion of the forklift by constraining movement in the horizontal plane.

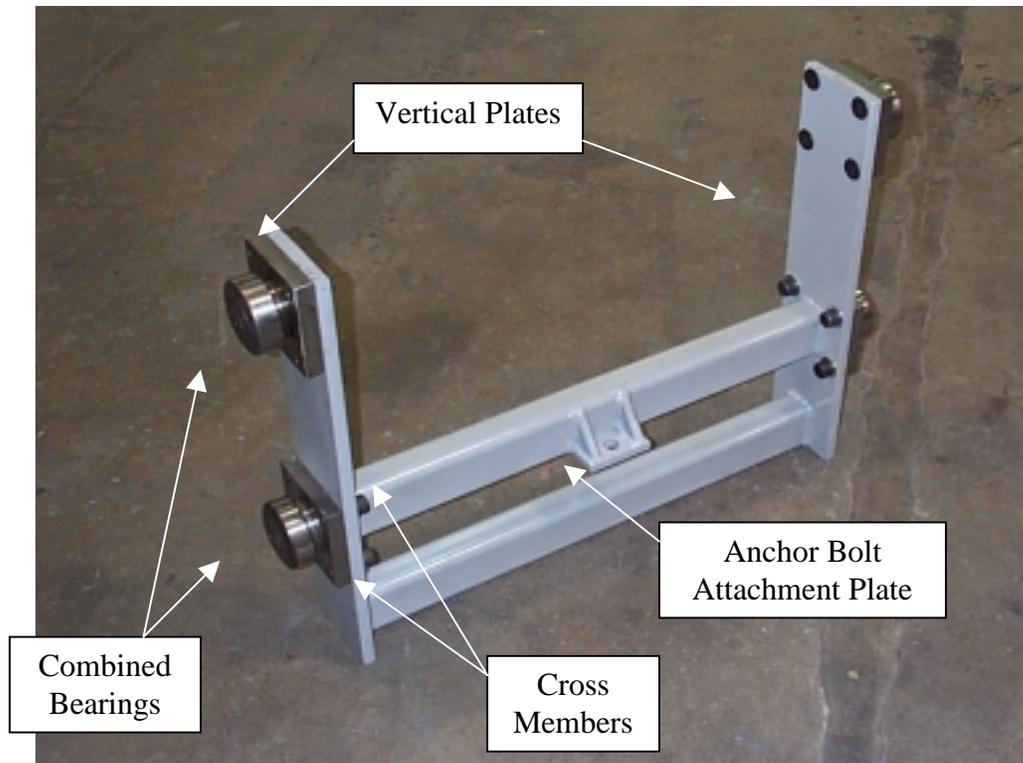
### 3.3.10.2 Layout

The front surface of the carriage cross members were offset 190.5 cm (75.0 in) to the rear of the main conveyor idlers. The maximum allowable vertical clearance for the carriage design underneath the fork beams was 25.4 cm (10.0 in). The entire depth was used in designing the carriage to provide maximum support to the forks. Given the height of each cross member, 7.62 cm (3.0 in), the members were spaced 10.16 cm (4.0 in) apart vertically. The mast channel length was specified as 167.6 cm (66.0 in). This dimension was determined from the travel height of the carriage and the vertical spacing of the top and bottom bearings on the carriage.

### 3.3.10.3 Detail Design

The carriage design paralleled the detail design and development of the forks. The general form and layout of the carriage was established first, based upon interface requirements with the forks. Then, the maximum loading conditions were computed to optimize the design of each member.

The assembled carriage is depicted in Figure 3.26. Two cross members in the carriage design configuration were used to support the base of the forks. The bottom member alone sustained the weight of the forks and cone stacks. Then, both the top and bottom member functioned together, creating a counter-couple to oppose the moment created by the loaded, cantilevered beams. The cross members were fixed at both ends to vertical plates. On the outer surface of the vertical plates, four combined bearings were mounted. The bearing fit inside the open face of the mast channels and functioned as the means for achieving vertical motion of the carriage.



*Figure 3.25 Assembled Carriage*

The assembly layout was developed based upon a number of factors including the fork beam spacing, the rear position of the cone stack, the forklift cylinder location, the truck frame layout and the location of other miscellaneous structural members. The only possible configuration for the carriage was to place it below or level with the forks. Designing the carriage above the forks was not an option because the space was occupied by the cone stacks. After identifying the vertical location of the carriage relative to the forks, the most influential layout related decision was determining the spacing between the mast channels. The remainder of the fork and mast design followed naturally.

Ideally, the distance between the mast channels was set equal to the nominal width across the two cone stacks. Unfortunately, this layout was not feasible because the inner width of the truck chassis was narrower than the outer width of the bearings on the carriage frame, resulting in an interference. Thus, the next best solution was to mount the combined bearings above the chassis. The vertical plates were lengthened to extend higher and the bearings were mounted above the cross members. In making this change, it was necessary to adjust the carriage width. The middle facing surfaces of the vertical plates were spaced slightly wider than the tapered end of the cone stack. From this dimension, the cross member length was determined to be 70.5 cm (27.75 in).

The corresponding axial load limit for the combined bearings was 5190 N (1167 lb). The maximum radial loading condition was computed based on an asymmetric loading of the forks. Assuming a full cone stack on one fork beam and no cones on the other, an axial load of 1824 N

(410 lb) would be applied to two of the bearings. The safety factor for this worst case, eccentric loading of the forklift was 2.8.

The mast channels were selected of a profile compatible with the combined bearings. The height of the masts was specified to be 167.6 cm (66.0 in), as mentioned in the layout discussion. The masts were spaced 86.7 cm (34.14 in) apart, from inner wall to inner wall, as determined from the width of the carriage assembly. Rigid members were welded to the top and bottom of the channels to maintain spacing. These members were then mounted to the truck frame and cone storage frames to secure the masts.

### 3.3.11 Main Conveyor

A split main conveyor configuration was adopted with the forklift concept. Four thin belts replaced a single, wide belt. The two pairs of belts each conveyed a single stack of cones. The belts, the front idler assembly and the drive shaft assembly all mounted to the forks and thus traveled up and down with the forklift unit. The design work performed for the main conveyor involved identifying and calculating functional requirements and selecting appropriate components to meet the performance specifications.

### 3.3.12 Belts and Pulleys

#### 3.3.12.1 Specifications

The conveyed load per belt amounted to 700 N/m (4.0 lb/in) based on the weight and distribution of the cones. The maximum conveying distance was conservatively given to be 157.5 cm (62.0 in), the maximum length of the stacks.

#### 3.3.12.2 Details of Selection

Timing belts and pulleys were chosen primarily to achieve a means of conveying the cones without slip. The toothed interface and the steel cables offered a solid and reliable solution for incorporating the downsized components of the redesigned conveyor. The belt and pulley sizes were determined using several selection guides. The conveyor demands were computed assuming worst case conditions and the results were all consistent. One approach is discussed here to offer insight into the selection procedure.

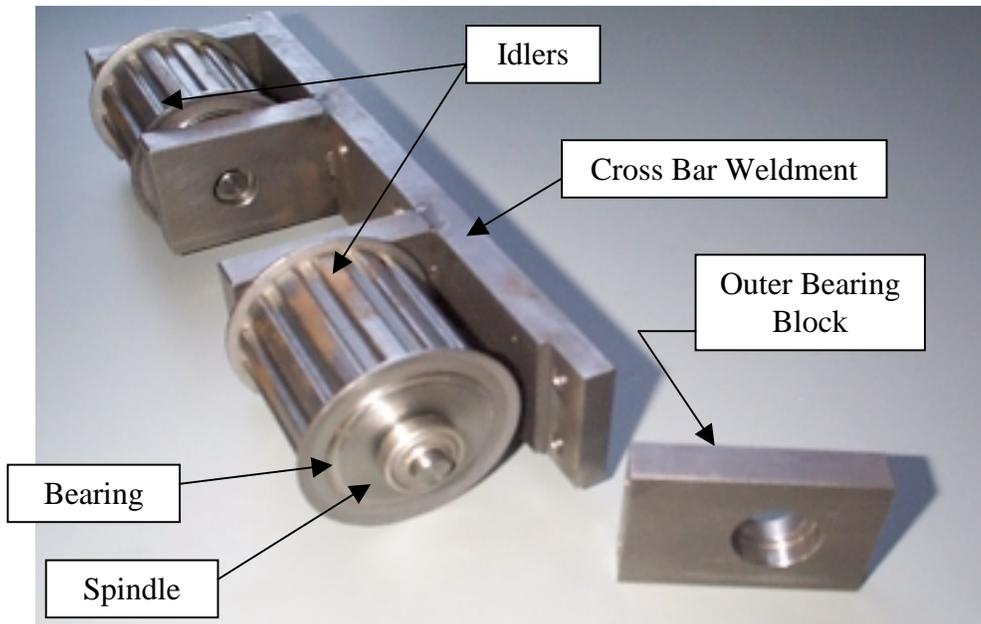
First, the maximum tension created in each belt was computed to be 790.1 N (177.6 lb). The contributing factors and conditions included friction between the belt and the guide strips and inertial acceleration of the cones and pulleys. The conveyed load and length were assumed equal to the weight and length of a full stack of cones. The belt pitch was then selected based upon the maximum tension. Both light, .953 cm (.375 in), and heavy, 1.27 cm (.50 in), pitch belts were viable options. However, only the heavy pitch belt was available in the long lengths required. Although a 2.85 cm (1.12 in) wide belt was sufficient to withstand the induced tension, a width of 5.08 cm (2.0 in) was chosen to increase the contact area between the belt and the cones. The overall belt length selected based upon the fork and carriage layout was determined to be 431.8 cm (170.0 in).

The smallest available pulley was sufficient for operation at the maximum belt loads. Therefore, it was chosen for this reason and to maximize clearance between the fork and ACM machinery. The pulley pitch diameter measured 5.66 cm (2.228 in), there were 14 teeth, and side flanges were specified to keep the belt from walking. A 1.91 cm (.75 in) idler bore diameter was called for and the drive pulley bores were specified for use with a q-d bushing, as described.

### 3.3.13 Front Idler Assembly

#### 3.3.13.1 Components

The front idler assemblies each contained a pair of idlers, two free spinning spindles, four bearings and bearing blocks as shown in Figure 3.27. The idlers and bearings were press fit onto the spindles, which could then be easily sandwiched between the outer, removable bearing blocks and those welded to the cross plate.



*Figure 3.26 Front Idler Assembly*

#### 3.3.13.2 Layout

Placement of the front idlers was kept consistent with the old main conveyor layout and served as a reference for many components of the system. The centerline was nominally set 84.8 cm (33.38 in) rearward from the front of the cone body. The idler spin axis was offset 3.49 cm (1.375 in) forward and 1.90 cm (.747 in) below the top, front edge of the fork weldment to achieve correct vertical alignment with the belts and ensure maximum tooth engagement. The spindle spacing was designed to position the idlers in line with the belt guide strips. The depth of the weldment and outer bearing blocks was limited to under (2.13 in) to ensure adequate clearance for the belt above and below.

### 3.3.13.3 Detail Design

The most critical aspects associated with the idler assembly design were optimizing the spindle design and selecting appropriate bearings. This will be the focus of the following detail design discussion. The remaining component designs evolved around the specified layout dimensions described. Upon pre-tensioning the belts, the idler assembly was firmly pressed against the fork face. The resulting compression forces and stresses generated within the weldment and outer bearing blocks were minimal. The attachment bolts also experienced minimal loads of magnitude less than the idler assembly weight. Therefore, the bolts functioned primarily to locate the assembly in the proper orientation.

The spindle design was developed based upon the forces induced during conveyor operation and for compatible fit with the idler and bearing bore diameters. During conveyor operation, tight side and slack side tension conditions were created in the belt, resulting in a radial load applied to the spindle. The maximum force under the tight side tension condition amounted to 2447 N (550 lb). The resulting bearing reactions were then computed to be 1224 N (275 lb), half the idler-induced radial force, assuming symmetric loading.

Self-aligning bearings were selected to support the spindles in the idler assemblies. They were available in small inner race to outer race diameter ratios, which was important for the shallow depth of the assembly. They were also rated for high radial loads compared to other single row bearing types. The self-aligning characteristic was important as well to withstand the misalignments in the assembly. The free play in each bearing also simplified belt assembly over the front idlers. The specific bearing contained an inner race bore of .953 cm (.375 in) and an outer race diameter of 1.91 cm (.75 in). The radial loading was rated for 4980 N (1120 lb), which gave a conservative, safety margin of 2.

## 3.3.14 Drive Shaft Assembly

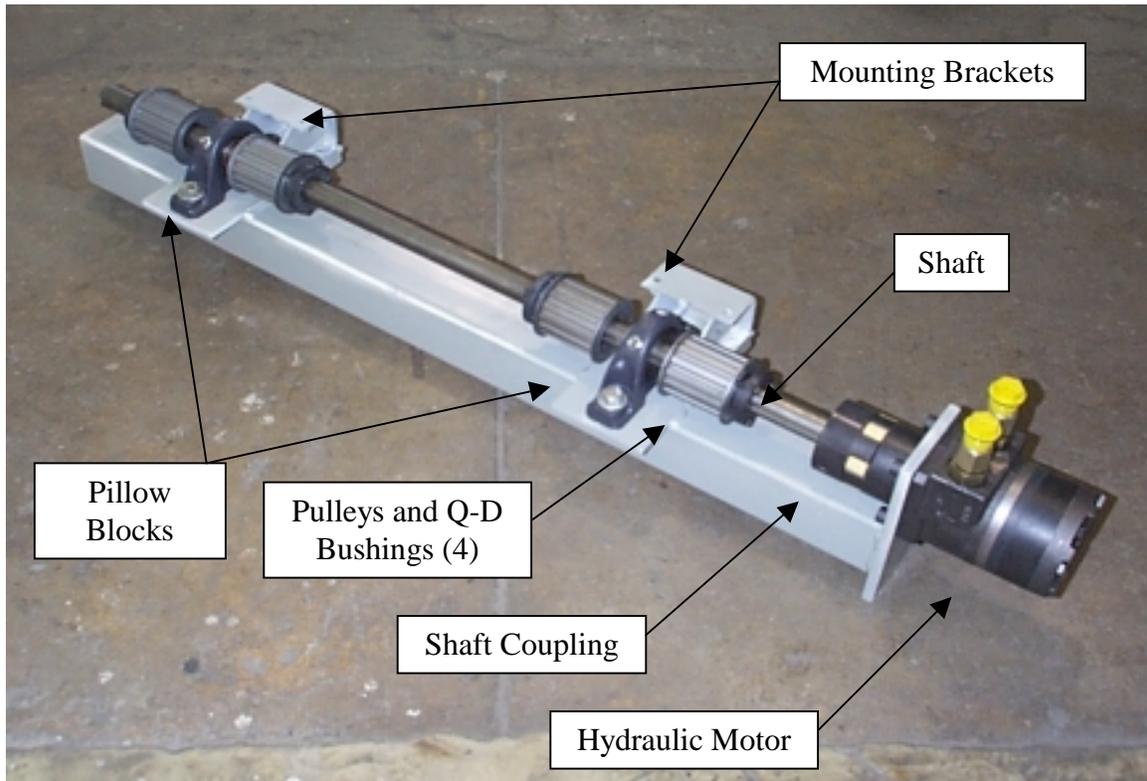
### 3.3.14.1 Specifications

A common shaft was to be designed to drive all four belts. The operating angular velocity of the shaft needed to be 50 rpm to achieve a belt speed of 15.24 cm/s (6in/s), which was consistent with the old conveyor operation. A minimum motor torque of 90.4 N-m (800 lb-in) was required to convey the two stack maximum load. The hydraulic power supply specifications for the conveyor drive were identical to the overall system requirements, which included a working pressure of 6.9 MPa (1000 psi) and a minimum volumetric flow rate of 7.6 L/min (2 gal/min).

### 3.3.14.2 Components

The drive shaft assembly consisted of a support weldment and the necessary drive components required to operate the conveyor. A 6.35 cm (2.5 in) square tube functioned as the weldment backbone. The motor mount plate was welded to one end of the tube. The pillow block mount plates and support brackets were welded to the top of the tube. Four pulleys and four respective q-d bushings fit onto a long drive shaft. The shaft was secured by two pillow blocks. The shaft and drive motor were assembled to the weldment, aligned and joined with a

spider shaft coupling. Figure 3.28 identifies the component locations within the drive shaft assembly.



*Figure 3.27 Drive Shaft Assembly*

### 3.3.14.3 Layout

The drive shaft assembly mounted to the rear end of the fork tubes. Attachment brackets were inserted inside the tubes and positioned to attain a nominal center to center distance of 207 cm (81.5 in) between the drive shaft and the front idlers. Two shaft-supporting pillow blocks mounted behind the brackets. They were spaced 47.0 cm (18.5 in) apart and aligned with the centerline of the fork tubes. Both pairs of pulleys were fixed onto the drive shaft in line with the guide strips and the front idlers. Such placement resulted in a symmetric layout about the pillow blocks. As with the idler spindles, the drive axis was also required to be 1.90 cm (.747 in) below the top surface of the fork beam.

### 3.3.14.4 Detail Design

The drive shaft assembly took shape around the nominal layout of the forks. The symmetry of the forklift configuration was utilized to create a favorable design. Each pillow block was centered behind a fork beam, equidistant between each pulley pair. This layout minimized the shear and moment loading applied to the shaft and evenly distributed the shaft supporting reactions between the two pillow blocks. The maximum radial reaction between the shaft and pulleys was equal to the maximum tight side belt tension plus the maximum slack side belt

tension. Therefore, the shaft force created by each belt amounted to 1459 N (328 lb) and resulted in a maximum radial reaction of 2918 N (656 lb) at each pillow block. The belts also applied a resisting torque of 22.5 N-m (200 lb-in) against the shaft due to the difference in tight side and slack side tensions.

The drive shaft design was optimized based on the applied loading conditions. Given the maximum bending moment of 100 N-m (890 in-lb) and the maximum torque loading of 90 N-m (800 lb-in), a shaft diameter of 2.54 cm (1.0 in) was required for a safety factor of 3. The shaft was designed 83.8 cm (33.0 in) long to span the width between the outermost pulleys and couple to the drive motor at one end.

Q-d bushings were selected for use with the four drive pulleys to provide a tight fit onto the drive shaft. Set screws in the pillow blocks were then used to secure the shaft and prevent lateral shifting. The pulleys were specified with the q-d bushings for use on a 2.54 cm (1.0 in) diameter shaft, as were the pillow blocks. The pillow block bearings were rated for a maximum static load of 6950 N (1560 lb), which gave a safety margin of 2.4 times the maximum radial reaction force.

Selection of the drive shaft motor was based upon the hydraulic power supply, the desired conveyor speed, and the maximum torque required to operate the conveyor. The hydraulic motor chosen was designed to operate at a speed of 48 RPM and supply an output torque of 110 N-m (975 lb-in) under the specified hydraulic inputs. The spider shaft coupling chosen was rated for a torque of 90 N-m (800 lb-in), the maximum resisting torque created during conveyor operation.

The drive shaft weldment design was created to satisfy the specified layout requirements of the shaft and pulleys. The support brackets were dimensioned to position the pillow block bore at the specified height below the top surface of the fork tube. The bracket and pillow block mount plates were also set narrow enough to be inserted inside the tube opening. This offered the assembly some adjustability in and out of the tube and was required to wrap the belts over the pulleys. The brackets fastened to the top and bottom walls of the fork tube in a location that positioned the drive shaft 207 cm (81.5 in) rearward of the front pulleys. The shear stress created in the bracket bolts due to tension in the belts was minimal.

Both brackets and plates were seated on top of the cross tube. The tube provided torsion and bending rigidity to the assembly. It was intended to eliminate excess stress in the shaft created by misalignment or unequal loading of the fork beams. The vertical, motor mount plate was strongly joined to the end of the tube with an all-around weld. A mating hole pattern was created in the plate, to which the motor flange was firmly bolted. Stresses within the vertical plate caused by the motor torque output were computed well below the yield strength of the material.

### 3.3.15 Summary

The detail design was a process of solidifying and optimizing the preliminary design concepts of the multistack system. This section discussed important aspects associated with the design of the multistack subsystem units and subassemblies. Detailed layout and assembly information was established and individual component designs were presented in light of the functional requirements and specifications. The result of the detail design efforts was a complete

system design that was fully documented and supported with engineering analysis. The next step in the development process was to build and test the system. In doing so, a thorough design evaluation could be completed and appropriate improvements could be made to complete the development of a fully operational and functional system.

### 3.4 Prototype Operation and Preliminary Testing

A prototype system was fabricated and assembled in accordance with the multistack detail design. The objective of prototype development was to test system operation, inspect the system design and evaluate overall function and performance. The prototype system was observed during repeated operation and pertinent measurements were taken to quantitatively compare aspects of the physical response to the analytical predictions. This section describes the system operation sequence, discusses observations of preliminary system operation, and presents the data collected to characterize system performance.

#### 3.4.1 General System Operation

##### 3.4.1.1 Operating Sequence

The multistack operating sequence is best described as a highly linear, sequential procedure. While rapid execution is physically limited, system control is greatly simplified. The operation procedure documents the means and requirements for handling cone stacks within the storage system. Overall, system operation is divided into two modes: stack storage and stack retrieval.

##### 3.4.1.2 Cone Stack Storage

The system enters the storage mode during ACM cone pick up. The main conveyor assists in stacking cones during retrieval from the road and once the stacks reach maximum size, it aligns the front of the stack approximately 7.6 cm (3 in) behind the axis of the front conveyor pulleys. The forklift is then powered to raise the stacks into the storage structure. Upon activation of the forklift unit, the retraction mechanisms are actuated to retract the hinges and clear the vertical envelope for the cones. The stacks are lifted to the highest unoccupied layer in the storage structure and continue above the retracted hinges. The retraction mechanisms are then reverse actuated to unfold the hinges. The forklift then slowly lowers. The forks pass between the deployed hinge leaves, transferring the stacks from the forks to the hinges, and then return to the truck bed level. Figure 3.29 demonstrates the system storage sequence. Storage mode operation continues throughout cone pick up until all storage levels are filled, including the main conveyor.

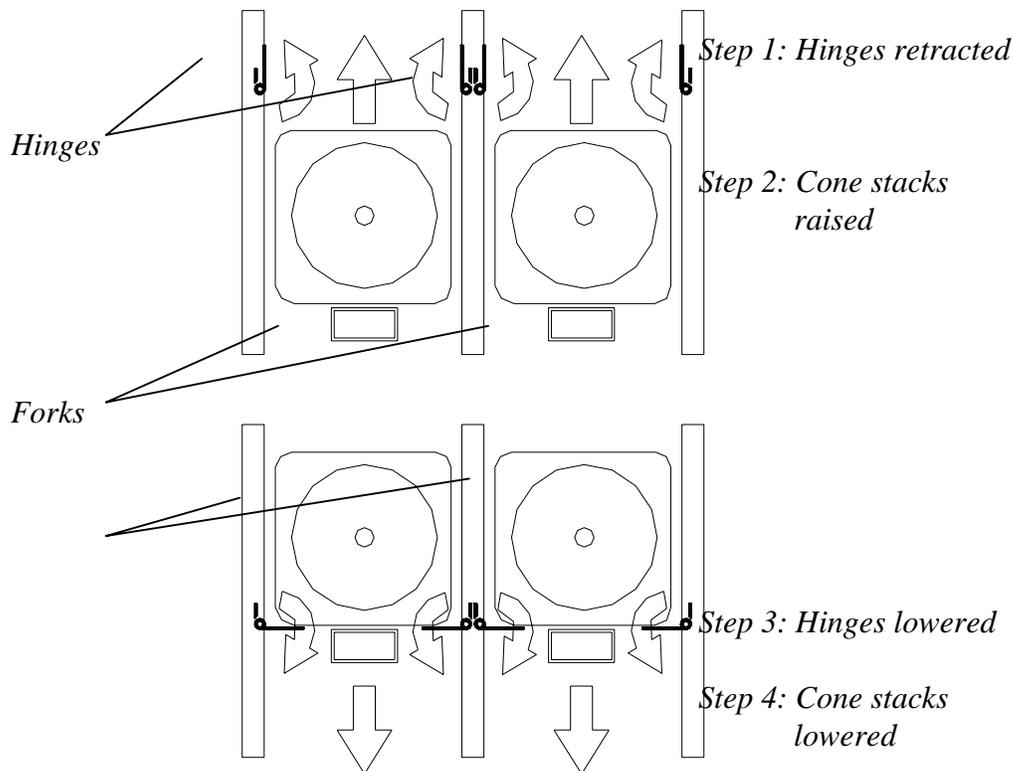


Figure 3.28 Stack Storage Sequence

### 3.4.1.3 Cone Stack Retrieval

Stack retrieval mode is invoked during ACM cone drop off operations. When the supply of cones on the main conveyor is depleted, cones must be retrieved from storage for road placement. The conveyer first repositions the saddles to the rearmost position to support the cone stacks. The forklift is then raised to the lowest level containing cones. The forks pass slowly between the pairs of hinges to contact and lift the stacks up 12.7 cm (5 in). The retraction mechanism retracts the hinges. The forklift returns to the bed level of the cone body with two full stacks and positions them for deployment. The stack retrieval process executes throughout cone drop off until the stored cone supply is depleted. The retrieval sequence follows the reverse procedure shown in Figure 3.29.

### 3.4.1.4 Qualitative Results

Qualitative evaluation of the prototype system involved operating the multistack system as a whole to store and retrieve cone stacks and closely observing critical aspects of the system. Prototype operation was performed through manual control of a hydraulic power pump. Two full stacks of 36 cm (14 in) square base cones were used to fully demonstrate the forklift and retention subsystem operation one level at a time.

Overall, preliminary operation of the prototype multistack system was very successful. The cone stacks were raised and lowered by the forklift unit as intended. The forklift motion was

smooth, with little end vibration. A small end deflection was evident by a slight downward slant at the end of the forks. Without the lateral support features to center the stacks on the forks and hinges, the cones tended to shift laterally. When this occurred, there was an increased likelihood of contact and interference with the storage framework members and the hinges. This observation confirmed the need to implement lateral supports in the design. The sequence of transferring the cone stacks from the forks to the hinges was also clean and smooth. The forks easily passed between the unfolded hinges, and the stacks were firmly positioned.

The main conveyor configuration proved very robust in conveying the cone stacks. The double belt layout easily supported and moved the cones. The saddle was designed with superior adjustability to support the back end of the stack but required some stiffening at the adjustable joints. The design for clamping the saddle to the belt also worked well, and the wide base design provided adequate stability to prevent strain on the belts. The belts were tensioned tightly with sag in the bottom segment, so a tensioner was not required. The only problem with the new conveyor design involved the front end pulley assembly. During forklift operation, contact was observed between the cross plate of the assembly and the connector rod of the mechanism. The interference problem was primarily attributed to the warped beam assembly. However, the easiest solution was removing excess material off the sides of the front conveyor assembly cross plate.

The hinges adequately supported the stacks, but the free leaves showed noticeable deflection. During system assembly, the hinges were mounted to the angle bars and shimmed for two reasons: to prevent rubbing during rotation, and to ensure horizontal constraint of the hinge free leaf when supporting the stack. Slight variations in the vertical placement of the hinges led to uneven load distribution between pairs of hinges. It was concluded that the free leaves required stiffening and that appropriate fixtures be used to consistently place the hinges. Another issue became apparent upon assembling and operating the continuous hinges. The ease of hinge rotation varied between hinges and was attributed to variations and imperfections in the originally supplied product. To loosen the stiffness of some hinges, the knuckles were slightly opened to reduce the friction created between the leaf knuckles and the pin. This was discovered early on and accounted for in the design and selection of components for the retraction mechanism.

The retraction mechanism functioned effectively to retract and open the hinges. Upon actuation of all six mechanisms, the hinges were retracted completely with the exception of the bottom middle set. In place of the originally selected spring, which was improperly sized, a substitute spring was used. The substitute spring failed to retract the hinges completely. The retraction problem was easily solved by selecting another spring. The fixed bushing blocks of the mechanism array framework presented no binding problems. Instead, actuation of the hinges was rapid and smooth. The pinned joints at both ends of the top cross bar proved effective during mechanism actuation. The cross member at the top of the mechanism frame rotated slightly to the left and right throughout the actuation stroke due to unbalanced forces. This result was expected, however, and the cross member leveled off at full stroke.

One problem encountered during actuation of the mechanism framework cross member was binding of the top cross member. Eccentric loading of the framework resulted in a twisting moment that forced contact between the cross member and the storage framework. This

condition resulted in significant resistance during mechanism actuation and required design modifications to more rigidly attach the cylinder to the cross member. Constrained actuation of the upper mechanisms was also achieved successfully. The hinges remained fixed under the load of full cone stacks and the respective springs absorbed the stroke of the cylinder. The weight of the cone stack and the added stiffener plate on the hinge resulted in a rigid configuration, which led to negligible localized deflection of the hinge at the mechanism attachment point.

### 3.4.2 System Functional Measurements

It was useful to measure aspects of system performance to better characterize and understand the component responses during operation. It was also important to verify the design work and identify sources of discrepancy between the experimental data and the design analysis. The measurements presented here represent a preliminary collection of data related to response characteristics and performance of the forklift and the retraction mechanism.

#### 3.4.2.1 Retraction Mechanism

Similar to the forklift, the retraction mechanism was another important component to closely test. Measurements were taken of individual mechanisms and of the mechanism array as a whole. In both cases, a force gauge was used to measure the static equilibrium force required to actuate the mechanism throughout hinge retraction, a scale was used to measure the spring length and rod stroke, and a protractor template was used to measure the hinge rotation angle. An overhead hoist was also set up in place of the cylinder to manually actuate the vertical rod, which kinematically transferred the actuation stroke as input to the mechanism. Measurements were taken of the retracted angle, the spring length and the static force measured by the gauge at 1.3 cm (.5 in) increments throughout the total stroke of 15.2 cm (6 in). Figure 3.30 illustrates the set up and identifies the measured parameters. The results discussed and presented here are measurements of specific mechanisms, which accurately represent a majority of the data collected.

#### 3.4.2.2 Unconstrained Individual Mechanism Operation

Under normal actuation, the single mechanism measurement results varied depending on the spring characteristics and friction in the hinge joints. Stiffness in the hinge joints was accounted for during detail design by selecting appropriate springs to compensate for any added resistance. Therefore, the mechanisms all functioned similarly in controlling the hinges. However, there were differences in the actuation force required to rotate each free hinge leaf. The measurement data presented in this section was collected from mechanisms attached to hinges, which rotate freely without joint resistance.

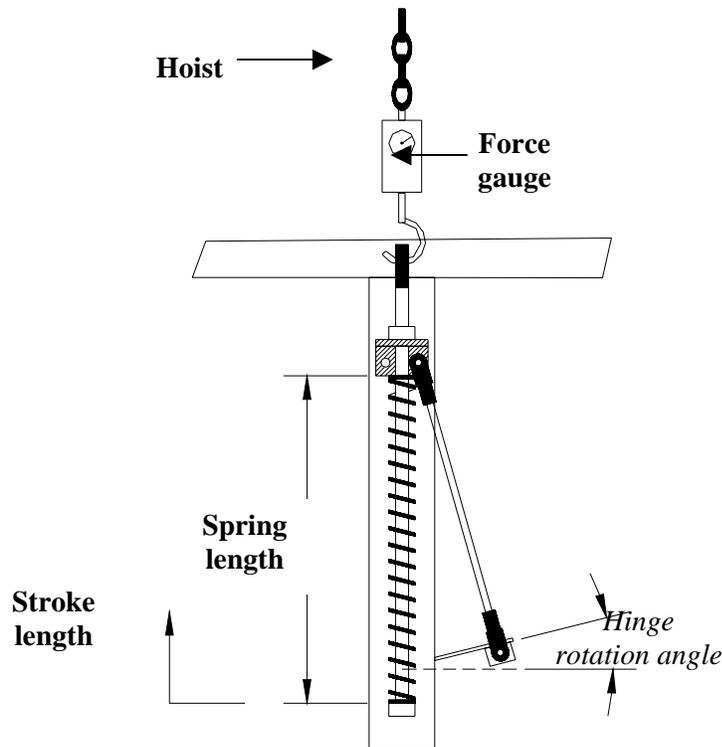


Figure 3.29 Retraction Mechanism Measurement Set-up

An important consideration was the relationship between the angle of leaf rotation and the force required to maintain static equilibrium throughout the actuation range. The force initially spikes above 55 N (12.3 lb) until the angle of rotation reaches 10 degrees. After passing the peak reaction point, the equilibrium force gradually decreases throughout the remainder of leaf rotation, reaching a minimum of about 37 N (8.3 lb). Following complete retraction of the leaf, the force spikes as the spring compresses throughout the remaining actuation stroke

Analytically, the mechanism was challenging to model because of the spring component. The spring translates vertically and deflects, and the spring force and rotation angle computation must be adjusted accordingly throughout actuation. A program was developed using an iterative loop structure to calculate the spring force required to hold the hinge leaf in equilibrium during retraction. Despite the small difference in magnitude between the experimental and theoretical static equilibrium forces, the model generated closely resembles the physical response, which validates the original mechanism design work.

### 3.4.3 Summary

Operation and testing of the prototype multistack system proved successful. The forklift and retention subsystem designs were demonstrated to effectively handle and store stacks of cones within the storage framework. Measurements were taken to characterize aspects of the forklift and retraction mechanism. The forklift measurement data revealed the fork deflections to be larger than predicted due to compliance in the carriage and a loose fit between the combined

bearings and the channel profile. The fork beam response to the cone load proved to be consistent with the theoretical model. Extensive time and effort would be required to more effectively test and measure the entire forklift response for complete comparison with an analytical model. It is recommended that additional testing be performed for future system development. The retraction mechanism data proved much more consistent with the analytical expectations. While some inconsistencies were observed between the experimental and theoretical data, the results revealed a response consistent with the intentions of the design. Based on the testing results presented, there are several recommendations proposed for modification to the existing multistack system design.

### 3.5 Conclusions and Recommendations

This report documents and describes the progression of the multistack cone storage system from selection of a preliminary concept to operational prototype. The process approach and engineering design work were equally important to the development of a successful product. This section highlights key aspects and accomplishments of the project. Recommendations and suggestions are also presented for consideration during future development of the multistack system based on conclusions made from prototype design, testing and operation.

#### 3.5.1 Project Summary and Results

Motivation for the project originated from the need to increase the cone storage capacity of the maintenance trucks. The system was intended to extend the capabilities of the ACM and improve the safety conditions of traffic control operations. Numerous requirements and specifications were established to define the project scope and guide multistack development efforts. System concepts were generated and the best design was selected using comparison and trade off analyses. The detail design work was completed to optimize system characteristics. Finally, a prototype was fabricated, assembled and tested on the cone body platform.

The multistack system layout is characterized by horizontally oriented cone stocks, which are stored in multiple, vertical layers. The system configuration is consistent with current methods for storing cones on the ACM and manually operated cone trucks. A forklift unit design was chosen to raise and lower the cone stacks within the storage framework. Successful integration and operation of the entire system can be mostly attributed to the simplicity of the forklift design. It effectively handles cone stacks and supports the reconfigured main conveyor. The retention subsystem was another significant aspect of the multistack system. It secures and supports the cone stacks in the storage framework. The simple design and operation of the retention hinges and retraction mechanism has also proven to be highly successful, though some redesign may be necessary to increase the hinge stiffness in order to robustly support the cone stacks.

Overall, the multistack system design fulfills the needs and desires of a cone storage system for maintenance vehicles. Preliminary testing and operation of the prototype system successfully demonstrates storage of a tripled cone load. Compared with current methods, it improves the efficiency of extensive traffic control operations and offers increased safety conditions to maintenance workers exposed to the ever-present hazards associated with highway maintenance

work. Continued development and testing will be performed to refine the system design and ensure successful integration onto highway maintenance vehicles.

### 3.5.2 Design Recommendations

After reviewing the design, and analyzing system measurements and operation testing results, several modifications have been proposed for the system design.

#### 3.5.2.1 Optimized Fork Beam Profile

The fork beam was designed to meet specified end deflections. For the system prototype, a standard size beam with constant section was selected for ease of fabrication and to minimize costs. However, to fully optimize the beam weight, strength and rigidity, a custom profile should be designed using a tapered or stepped section design. The beam could also be designed for ease of mounting to the carriage and the section depth could be reduced to create added clearance above the cone body structural members. The beam width must remain constant to support the conveyor belts and end assemblies as presently configured.

#### 3.5.2.2 Optimized Carriage Component Design

As concluded from the system measurement results, compliance in the carriage structure and carriage supporting members significantly influenced the fork end deflection. The carriage weldment consists of highly rigid members, however, there is an inherent looseness in the fit between the combined bearings and channel profile. To minimize the effect on the fork deflection, the carriage could be redesigned to support the forks at an upward angle such that the desired cone load would act as a self-correcting response to return the fork to the horizontal orientation. The weight of the carriage components can also be reduced by selecting a smaller combined bearing and channel section without compromising structural support. The prototype bearings and channel were selected early in the design stage due to their long lead time. Upon finalizing the calculations, it was discovered that a smaller bearing (Winkel, 98AP2) would be sufficient for the induced loading, along with the corresponding channel profile.

#### 3.5.2.3 Stiffened Retention Hinges

Observed in the retention hinges was a noticeable deflection upon loading. The hinges are required to be robust and it is recommended that the load bearing leaves be reinforced or that a stiffer hinge be used. The hinges offer a simple, preassembled solution to supporting the cone stacks in the storage framework. However, the hinges have negative characteristics such as compliance in the pin joint and inconsistencies associated with formation and manufacturing of the hinge assembly. Therefore, consideration should be given to developing a custom designed assembly, equivalent to the hinges in function and form, but exhibiting higher stiffness, strength, quality and geometric consistency.

#### 3.5.2.4 Simplified, Modular Storage Framework

The primary recommendation for improving the storage structure design is to enhance the modular characteristic of the framework. This modification would involve splitting the frame into a standard, upper section and a customized base section. This would save time and cost and eliminate unnecessary work required to redesign the entire framework for maintenance trucks of different makes and chassis. The upper weldment section remains the same, while the base section is customized according to the truck frame. The storage framework also needs to include lateral support features to secure the stored stacks.

#### 3.5.2.5 Additional System Testing

It is highly recommended that testing continue to more extensively validate the multistack system design and operation. Forklift vibration testing would be useful to understand the response induced by simultaneous operation of the system and the cone truck. A maximum loading test would be appropriate to ensure proper stack support once the retention hinges have been stiffened. System inspection should be performed following prolonged testing and vehicle operation to identify aspects of the design that may be susceptible to wear and failure. More extensive operation will also be required to observe and fine tune the coordinated operation between the multistack system and the ACM machinery.

### 3.5.3 System Integration Considerations

The multistack system is designed for use with the ACM or as an independent system for storing cones on standard cone trucks. The multistack system is also expandable in size. One possible configuration is a double stack structure which could easily be developed by making minor modifications to the current triple layer design. Serious consideration has also been given to a longer double stack system, which would store a triple cone load by lengthening the cone body and storage framework. There are a number of options for integrating the multistack system as originally intended or by customizing the design configuration to fit the needs of highway maintenance operations.

## CHAPTER 4

### DEVELOPMENT OF THE SECOND GENERATION RETRIEVAL ARM AND SECONDARY FUNNEL

This chapter discusses the development of the second generation retrieval arm and secondary funnel systems. These components are integrated into the drop box assembly and are used to capture the cones during the retrieval process. The second generation design has performed extremely well and been critical to the acceptance of the ACM. In addition to the improved reliability, the increased functionality and compact packaging has been instrumental in demonstrations to prospective users and commercializers of the ACM.

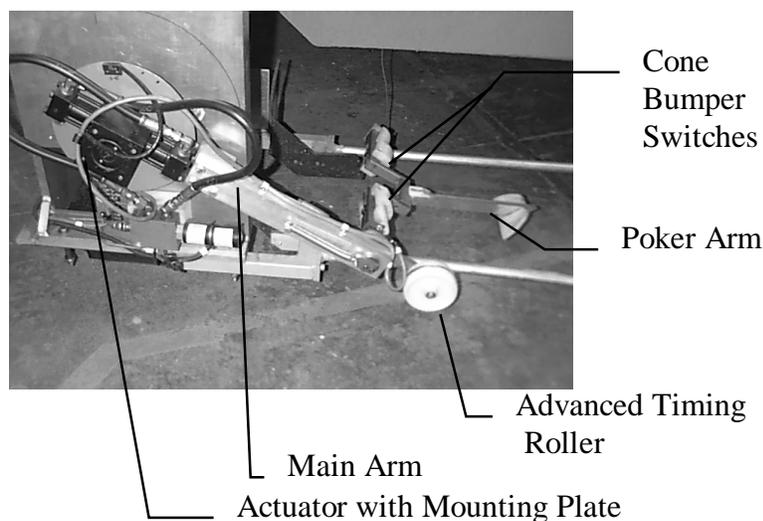
Section one describes the testbed ACM stowage components requiring redesign. Sections two and three describe the concept and detail designs of the new stowage system while section four describes testing and additional refinements. Section five describes the results of the redesign efforts and identifies additional issues that would be important for commercialization of the system.

#### 4.1 Testing and Modifications to the First Generation Components

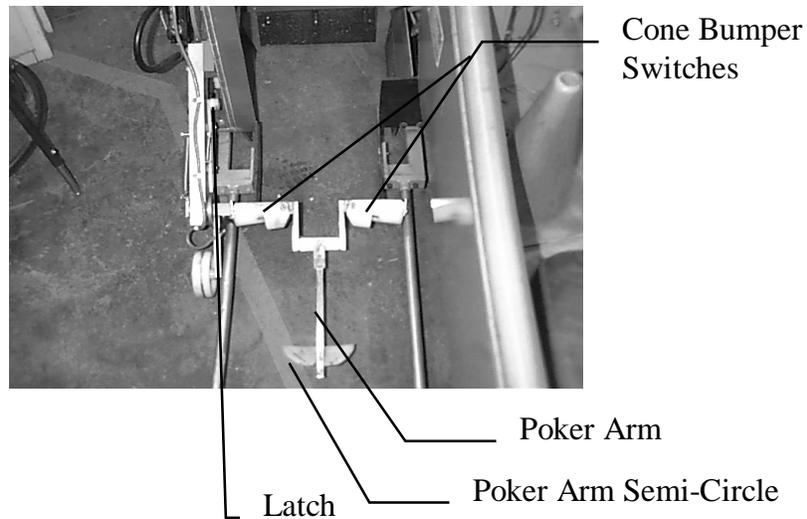
The completed testbed ACM lacked several of the original design features due to time and resource constraints. The testbed ACM was only capable of deploying and retrieving cones from the left side of the vehicle in the reverse direction. In spite of these handicaps, the ACM served well as an informational testbed to gain insight and experience in automated cone handling.

##### 4.1.1 Testbed ACM Retrieval Arm System

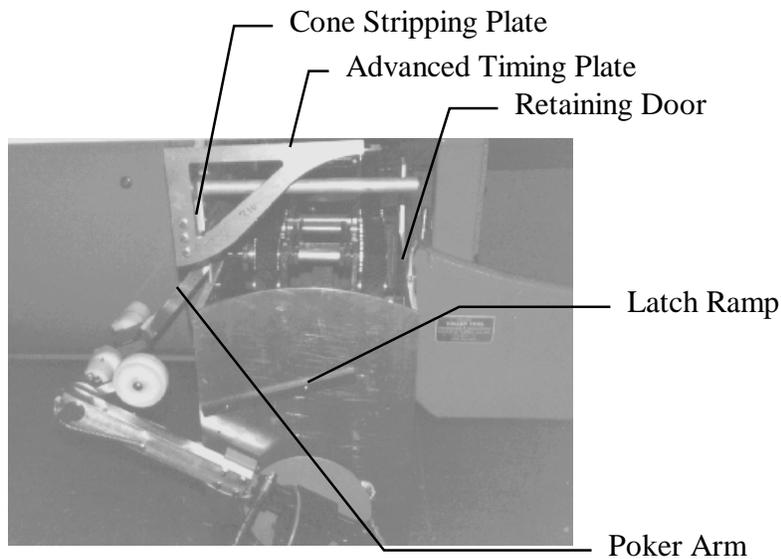
The testbed retrieval arm system is shown in Figures 4.1, 4.2 and 4.3 so that its main components are clear to the reader without the need to refer to Tseng et al, 1996.



*Figure 4.1 Retrieval Arm System, Side View*



*Figure 4.2 Retrieval Arm System, Top View*



*Figure 4.3 Retrieval Arm System in Retracted Position*

The retrieval arm performs a series of simple operations. While the arm is in the down position, the cone's open base is guided onto the poker arm by the secondary funnel. The impact of a cone against the two cone bumper switches activates them. Once both switches are continuously activated, the rack and pinion actuator rotates the arm upwards with the cone being lifted up on the poker arm. As the main arm approaches the vertical position, the latch contacts the latch ramp, unlocking the poker arm assembly and allowing its rotation around its pin connection at the main arm. At the same time the advanced timing roller contacts the advanced timing plate, which controls the rotation of the poker arm assembly. As the main arm and the

poker arm assemblies continue to rotate, the cone base comes into contact with the cone stripping plate, at which point the poker arm retracts out of the cone. Immediately after the poker arm is retracted out of the cone, it is left to fall freely onto the lateral conveyor belt (LCB) system. At this point, the retrieval arm is positioned in the retracted position as shown in Figure 4.3.

The retrieval arm is able to retrieve cones in the forward direction in the same way as described above. This retrieval direction requires the symmetrical components to be remounted in a mirror image of the configuration shown. These symmetrical remounted components include the latch ramp, advanced timing plate, actuator mounting plate, poker arm assembly, cone stripper plate, and retaining door. Testing in the forward retrieving mode was not done with the testbed ACM.

#### 4.1.2 Retrieval Arm System Testing

The retrieval arm was fine tuned and had its guides positioned correctly prior to all testing. Also, maintenance or lubrication required, especially on the cone bumper switches, was meticulously performed prior to each testing session. These tasks were performed to ensure the best possible operation of the subsystems.

Most testing was video taped to enable a frame by frame analysis of failure occurrences if necessary. This facilitated the correct identification of the individual failures and the failure categories.

The testing of the retrieval arm system consisted of two basic operations. One was the operation of the subsystem in the laboratory in manual operation mode. This method eliminated most of the environmental conditions and any interface errors with the other subsystems. The other test method consisted of fully automated functioning on a section of a public roadway. This testing method tested the subsystem's interface with other subsystems and the effect of varying environmental conditions. These conditions include road crown and cross slope, irregularities in the pavement, varying operational speed, and varying weather conditions. In the test series described, a total of 52 retrieval attempts were performed in the laboratory setting and 118 retrieval attempts were performed, in numerous different testing sessions, during the open road tests.

#### 4.1.3 Retrieval Arm System Testing Results

Seven basic failure modes were identified during the testing and these are described below.

**Failure Mode 1:** When a cone randomly rotates on the poker arm about the central axis of its conical section, it is still lifted upward but when it makes contact with the stripper plate, the rotated cone base is tilted beneath the stripper plate. This action prevents the poker arm from fully retracting out of the cone resulting in a jammed retrieval arm, which requires manual intervention.

**Failure Mode 2:** Activation of the bumper switches is not consistent. Since the cone bumper switches' activation directions are not aligned along a horizontal plane, additional friction is encountered during activation. The sliding force of the cone against the road surface is often not

enough to engage the switches. As a result, any built up road grime on the inner bumper switch surfaces will prevent the switches from properly activating. The switches must be regularly cleaned and lubricated.

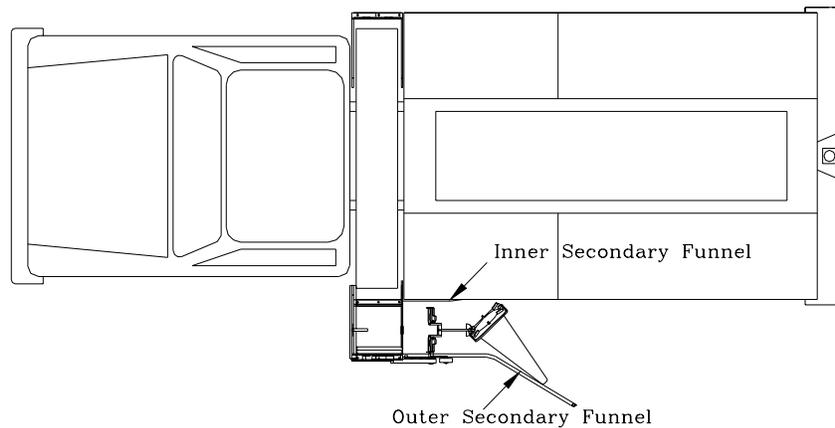
Failure Mode 3: The cone does not transfer from the retrieval arm to the lateral conveyor belt consistently. As a cone is stripped off the poker arms, it falls freely onto the lateral conveyor belt. This free-fall often results in an improperly positioned cone which does not seat correctly. This problem varies in degree of severity since often the motion of the lateral conveyor belt will correct the cone position; its motion moves the cone and forces it between the guides. This process often results in a retrieval delay and, at other times, manual intervention is required to free the cone, which is usually necessary for a cone that lands with an edge of its base resting on the outer edge lip of the Drop Box.

Failure Mode 4: Cycle time of the retrieval arm is too long. Cone retrieval failure occurs when the arm is not back in the receiving position when the next cone arrives. This failure mode is a combination of the time delay due to the large degree of rotation required by the retrieval arm and delays due to the first three failure modes.

Failure Mode 5: This mode is a result of improper interfacing with the retaining door. A cone that has slightly shifted its position on the poker arm will often come in contact with the retaining door which will shift the cone further out of position. Another improper interface with the retaining door occurs when the stripped free-falling cone is incorrectly guided onto the lateral conveyor belt by the retaining door. This failure mode usually results in the need for manual intervention.

Failure Mode 6: In this mode the cone simply falls off the poker arm. During the time between the cone bumper switch activation and the initiation of the upward motion of the arm, the cone can bounce around against the cone bumper switches as a result of the irregular abrasion of the cone base edge rubbing against the road. If the cone is not seated against the switches when the retrieval arm starts its upward motion, the cone will slide and fall off the poker arm. The driver is then required to reposition the truck to reattempt the retrieval causing significant delays.

Failure Mode 7: As the cone is guided onto the retrieval arm by the secondary funnel, it sometimes jams between the two. This occurs when the cone axis is parallel to the outer bend of the secondary funnel resulting in a cone with its base wedged between the poker arm tip and the poker arm semi-circle guide plate (see Figure 4.4). If this failure mode occurs, the operator can move the truck momentarily forward to clear up the wedged base, and reattempt the retrieval.



*Figure 4.4 Cone Wedged on Poker Arm (Failure Mode 7)*

The results of 118 retrieval attempts during full automation are summarized by the percentage of each failure mode occurrence in Table 4.1.

*Table 4.1 Fully Automated Retrieval, Road Testing Results*

Fully Automated Road Testing (118 Attempts)		
Failure Mode	Failure Description	Failure Percentage
1	Mode 1 - Cone Rotated on Poker Arm	0.85%
2	Mode 2 - Cone Bumper Switches not Activated	2.54%
3	Mode 3 - Improperly Positioned Cone on LCB	5.08%
4	Mode 4 - Retrieval Cycle Time too Slow	0.85%
5	Mode 5 - Retaining Door Interference	0.00%
6	Mode 6 - Cone Slid or Fallen off Poker Arm	1.69%
7	Mode 7 - Cone Wedged on Poker Arm	7.63%

The fully automated road testing had a total failure rate of 18.64%, leaving only a 81.36% successful retrieval rate. This is a low success rate and possible improvements were considered as described below.

The cone bumper switches are high maintenance items and require constant lubrication. They were usually lubricated prior to any testing and as a result, the percentages for failure mode 2 are artificially low. However, the mode 2 failure could be eliminated if the cone bumper switches were redesigned to reduce the friction and switch activation forces.

Failure mode 3 could also be significantly reduced or eliminated if more guides were provided to the free-falling cone. This is, however, a packaging challenge since limited room is available.

Failure mode 7 should really be considered a secondary funnel system failure, and can be remedied by either lengthening the secondary funnel or by shortening the poker arm. If the poker arm is shortened, failure mode 6 increases. Therefore, the best remedy for the mode 7 failure is to lengthen the secondary funnel system.

If the failure modes 2, 3, and 7 were resolved with the suggestions provided, the failure mode 4 occurrence would most likely be reduced or possibly be eliminated. Therefore if the modes 2, 3, 4, and 7 were eliminated, then the successful retrieval rate would increase to 97.46%. This is a significant improvement but depends on the success of the suggested solutions and additional failure modes could be expected. The remaining 2.54% failure would prove to be very difficult or impossible to eliminate, and a success rate approaching 100% is required to provide the maximum protection to the cone workers and engender acceptance of the ACM concept.

Table 4.2 shows the failure occurrences for 52 retrieval attempts during laboratory testing.

*Table 4.2 Laboratory Retrieval Testing Results*

Laboratory Testing (52 Attempts)		
Failure Mode	Failure Description	Failure Percentage
1	Mode 1 - Cone Rotated on Poker Arm	0.00%
2	Mode 2 - Cone Bumper Switches not Activated	N.A.
3	Mode 3 - Improperly Positioned Cone on LCB	11.54%
4	Mode 4 - Retrieval Cycle Time too Slow	N.A.
5	Mode 5 - Retaining Door Interference	3.85%
6	Mode 6 - Cone Slid or Fallen off Poker Arm	1.92%
7	Mode 7 - Cone Wedged on Poker Arm	N.A.

Several failure modes are not applicable in the laboratory retrieval testing. The mode 1 failure did not occur since the surface in the laboratory is constant and has no slope. Also the dynamic motion of a moving truck, which sometimes induces the cone to roll on the poker arm, was not present in the stationary lab setting. Failure modes 4 and 7 were not present since the cones were manually fed to the retrieval system and not evenly spaced on the road as in the road testing.

The failure mode 3 was more prevalent during the laboratory testing since the self-correcting effects of the lateral conveyor belt were not in action. The difference between the failure mode 3 occurrences during the two testing types provides the total percentage of improper landing positions corrected by the LCB, which is 6.46%. The solution of additional guides for this failure described above should still eliminate this higher laboratory failure percentage, although the higher failure rate will never be encountered during normal operation since the LCB System will always be in operation.

The fact that the failure mode 5 did not occur in the road testing but only on the laboratory, is somewhat of an anomaly. The only difference that could account for the discrepancy is the dynamics of the moving truck. The additional guides that would be needed to correct mode 3 failures could also prevent most of the mode 5 failures.

Mode 6 failure occurrence rates are very similar between the two testing types. The slightly lower percentage in the road testing setting could be the result of the dynamics of the truck motion. In road testing the truck is typically moving toward the cone so that when a cone does lose contact with the poker arm assembly, even while rotating upwards, the truck's motion brings them back together.

Mode 7 failure was not seen in the laboratory testing since the slope of the laboratory floor is nearly level. In order to test the effect of road camber, the cones were manually rotated to simulate the condition. A cone rolled on the poker arm over 3° (5% slope) consistently caused jams at the stripper plate. Therefore, if the testbed ACM's operating plane and the retrieval plane slopes differ by more than 5% cones cannot be picked up.

#### 4.1.4 Evaluation of Testbed ACM Retrieval Arm System

From the testing results it can be clearly seen that improvements to the first generation retrieval arm system are needed in order to support the first generation integrated prototype ACM. Besides the improvements already listed above, improvements in manufacturability, aesthetics, robustness and reliability are necessary. The system is very sensitive to modifications or impacts and requires constant fine tuning. Since the cones themselves vary significantly in both dimension and physical properties reliability of the system would be limited. When the retrieval arm is in the retracted position, it prevents the cone operator from opening the truck door, which would be a hazard in an emergency situation. If the desired retrieval direction is frequently changed, the required efforts to reconfigure the retrieval arm would use an excessive amount of time. Every time the retrieval arm is mounted, the components must be adjusted to provide the best operation. When the ACM system encounters road surface slopes of over 5%, it has no way of correcting and successfully retrieving the cones. In the testbed configuration, if

the ACM were to be used on a construction site with rough roads, the dynamics of the truck would result in retrieval failures.

After a thorough evaluation of the test results and solutions to problems identified, it was determined that another design iteration of the retrieval arm was required. Since a reliable retrieval process was critical to acceptance of the ACM concept, the redesign of the retrieval system had a high priority.

#### 4.1.5 Testbed ACM Secondary Funnel System

The secondary funnel is a simple system, whose function is to guide the cone to the retrieval arm system. It is an integral part of the cone retrieval process and is designed around the retrieval arm. The secondary funnel receives the cone after it has passed through the primary funnel system. If the cone is correctly passed from the primary funnel, the secondary funnel receives the cone between its outer and inner funnel and simply guides it to the retrieval arm system. The layout of the first generation secondary funnel is illustrated in Figure 4.4.

#### 4.1.6 Secondary Funnel System Testing

The testing of the secondary funnel was done simultaneously with the retrieval arm testing. Laboratory testing is not applicable to the secondary funnel system since there is no truck motion relative to a stationary cone. Therefore, only the automated road testing performed was considered for analyzing its function and failures.

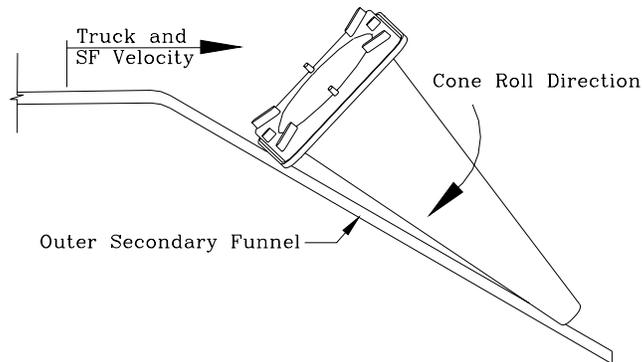
Since the secondary funnel system is a relatively simple system only a few failure modes exist. One retrieval failure that could occur is driver error, which would not be considered a failure. This type of error occurs if the driver of the truck takes sharp erratic turns during the retrieval and rotates the truck away from the cone resulting in its arrival at the secondary funnel outside the normal passage lane boundary of a cone traveling from the primary funnel. This retrieval failure is dependent on only the driver's skill and experience level with the ACM and was not encountered during the 118 retrieval attempts.

#### 4.1.7 Secondary Funnel System Testing Results

There were only two basic failure modes identified during the secondary funnel system testing. The first failure mode is the same as failure mode 7 of retrieval arm system. In the above discussion, this failure was assigned to the secondary funnel system. This failure occurs when the cone is passed to the retrieval arm from the secondary funnel with the cone's central axis parallel to the outer bend of the secondary funnel. When this occurs the cone wedges its base between the poker arm tip and the poker arm semi-circle guide. This failure was previously illustrated in Figure 4.4, and occurred in 7.63% of the retrieval attempts.

The second failure mode also occurs when the cones meet the outer bend of the secondary funnel but is only seen during faster retrieval speeds above the operating speed limit. The failure mode consists of a cone rolling over the top of the outer bend. This scenario is the result of the impact of the outer bend with the cone below its center of gravity. This failure is illustrated in

Figure 4.5. Since this failure mode was known prior to the road testing, the operational speed during the testing was kept below the functional limit.



*Figure 4.5 Cone Rolling Over Secondary Funnel*

#### 4.1.8 Evaluation of Testbed ACM Secondary Funnel System

Overall the secondary funnel functions very well. The main improvement needed to prevent retrieval failure is the lengthening of the outer secondary funnel. This will prevent the cone from wedging its base between the poker arm tip and the poker arm Semi-circle. Care must however be taken not to intrude into the operational space of the primary funnel and to prevent the funnel from contacting the rear tires and axle while either deployed or retracted. If faster cone retrieval is desired the critical speed of second failure mode must be raised. In order to increase the critical speed, the outer secondary funnel should be raised, effectively lowering the moment arm length that causes the cone to rotate and roll over the outer secondary funnel. However, if the secondary funnel is raised too much, it will create another problem by allowing the cone tip and conical section to be wedged beneath the funnel. If this occurs, stiff nylon bristles could be mounted vertically along the edge of the funnel to guide the cone tip. An additional enhancement required is a mechanism to deploy and retract the funnel as the drop box is extended and retracted.

### 4.2 Conceptual and Detail Design of the Second Generation Retrieval Arm

This section focuses on the development and design of second generation of the retrieval arm. The major design factors will be discussed in this chapter. The design of the retrieval arm subsystem was done first since it is a critical subsystem. Then, the secondary funnel was designed to meet its own general requirements and the interfacing requirements with the retrieval arm.

#### 4.2.1 Retrieval Arm Concept Requirements

As a result of testing and design reviews, the following requirements were considered important. As requested by the Caltrans workers, the width of the system should be kept to a

minimum to facilitate operation in the narrow confines often encountered on bridge decks and to minimize protrusion into traffic lanes. The system must function rapidly to reduce the retrieval cycle time to between 3 and 4 seconds. It must effectively handle cones at any ambient temperature encountered, as well as handle cones that are coated with the mold release compound or with road tar, oil, or dirt. The retrieval arm should not trap an occupant inside the truck cab. To the extent possible, the cone should be positively grasped during the entire retrieval process to prevent uncontrolled motion. The cone sensing switches also require operational improvements and must be sealed to prevent water and road dirt from interfering with normal operation. If possible, the retrieval arm system should allow bi-directional (forward and backward) retrieval without requiring extensive manual change over. Other desired functionalities include an operational speed of at least 10 mph and the possibility of handling other standard cone configurations with a minimum of adjustments.

All these requirements lead to a retrieval system that has some sort of positive grip on the cone during retrieval. This requires an arm with a higher degree of mechanical sophistication and is likely to be heavier.

Since previous testing was performed, the cone behavior was fairly well known and established, and many concepts were generated, discussed and evaluated. From this process, five main concepts emerged. The five final concepts are the Telescopic Arm, the Partial Linear Slide Arm, the Telescopic Arm with Table, the Arm with Sliding Table, and the Arm with Pull-in System. All these concepts still operate on the cones that are oriented with the bottom openings facing the retrieval arm system. All the concept are arms that rotate about a central point on the drop box and are driven via a sprocket and chain. This allows the rotational actuator to be somewhat remotely mounted and supply the torque to the arm by the chain-driven sprockets. This configuration is preferable to minimize the overall width and facilitate the mounting of a potentiometer to monitor the arm's position.

#### 4.2.1.1 Telescopic Arm Concept

The Telescopic Arm concept still has a poker arm that inserts into the bottom opening of the cone. The telescopic section retracts after the arm, with a cone on the poker arm, is rotated up to the vertical position. This action sets the retrieved cone onto the lateral conveyor belt (LCB). The cone is centered on the drop-off table by either grippers, flap contact as the arm retracts, or fingers that spread inside the cone. The chosen mechanism will be activated by and during the retraction of the telescoping arm but is attached to the drop box. This limits the weight of the arm. The poker arm folds out from under the cone by rotating down during the lateral motion of the cone on the LCB. This requires the poker arm to have a rotational degree of freedom with some sort of passive return spring action. The linear hydraulic actuator is mounted inside the arm assembly.

This concept requires a LCB drop-off table. A linear hydraulic actuator that rotates with the arm is also required, which creates minor hydraulic complications in that it requires rotational hydraulic fittings. Also, the telescoping system is a rather heavy mechanism. Another questionable aspect is whether the folding poker arm would create a problem for the laterally moving cone. The resistance created by the poker arm folding could prevent the cone from moving laterally, and the cone could be misaligned by the poker arm.

#### 4.2.1.2 Partial Linear Slide Arm Concept

This concept still has a poker arm and has a partial length linear slide on the retrieval arm. This system has the linear hydraulic actuator mounted on the side of the drop box to prevent the complications of rotating the actuator and to reduce the arm's weight. As the arm rotates to the near vertical position, a latch unlocks the telescopic locking mechanism of the arm, and then, the actuator engages with the linear slide when the arm reaches its vertical position. The actuator then retracts the upper section of the arm on which the cone is resting. This retraction action also provides some limited rotation to the upper arm section to assist in the retraction of the poker arm. This retraction action positions the cone onto the LCB drop-off table. The cone is centered on the drop-off table by a mechanism activated by the retraction action. This mechanism could be either a set of grippers, flaps moving as the arm retracts, or a set of fingers that spreads inside the cone. The poker arm is partially retracted by the rotation of the upper section of the arm, but still folds out from under the cone by the lateral motion of the cone on the LCB.

This concept has a lighter weight arm and no rotating hydraulic actuators. The poker arm still needs to partially fold down and rotate out of the cone. This could still create a problem for the laterally moving cone. Also, this concept still requires a drop-off table.

#### 4.2.1.3 Telescopic Arm with Table Concept

This concept is very similar to the Telescopic Arm concept except that it has the LCB wing actuator table attached to the arm. In this concept, the cone on the poker arm and resting against the wing actuator table is rotated up to the vertical position during retrieval. At the vertical position the telescoping arm retracts and causes the actuator table to come into contact with the LCB system. The contact points are friction pulleys which cause the belts on the actuator table to rotate, thus laterally moving the cone. The poker arm folds under the cone by the lateral motion of the cone. The cone is centered by some sort of gripper or otherwise moved inward on the LCB and centered by guides.

The retraction actuator in this concept could be mounted on the side of the drop box to limit the weight of the arm which is already weighted down by the wing actuator table attached. This also simplifies the hydraulic connections by leaving them non-rotating. The poker arm could be partially retracted by the telescoping action. This would simplify and limit the folding out of the cone's hollow conical section required by the poker arm. The main disadvantage in this concept, besides from the extra weight, is wear on the contact pulleys and the possibility of the somewhat fragile actuator table coming in contact with the road surface.

#### 4.2.1.4 Arm with Sliding Table Concept

In this concept the arm has a slide table attached. This arm also has a clamping device that engages on the cone's base at the vertical edge. When the arm rotates to the vertical position, the arm releases the clamp and allows the cone to slide onto the LCB. This clamp most likely will be actuated hydraulically, necessitating that a hydraulic cylinder be mounted inside the arm. This actuator probably is single acting and is returned by using springs. This requires only one complicated rotating hydraulic fitting.

Although this concept will have the additional weight of an attached table, the weight could be limited by designing a simple but effective table slide. This concept does not require any poker arm, but requires rotating hydraulics. The major benefit of this concept is that it has a positive grip on the cone during retrieval, and it does not require a drop-off table.

#### 4.2.1.5 Arm with LCB Pull-in System Concept

This concept is similar to the Telescopic Arm with Table concept. However, in contrast, this concept does not telescopically retract, and it has a very simple table with no rotating belts attached to limit the arm's weight. The arm with the cone on the poker arm will rotate up to the vertical position. At this location the arm's table height matches the height of the LCB. To move the cone inward off the table onto the LCB, a pull-in device attached to the LCB is activated. This pull-in device inserts some sort of hook into the open bottom of the cone which then pulls and moves the cone inward with the motion of the lateral belts.

This concept is one of the simplest RA systems, but certainly complicates the LCB system. The poker arm folds under and out of the cone's conical section during the cone's lateral motion of the LCB which could be problematic. The arm itself is relatively light in weight and has no extra hydraulics except for the rotational actuator, which all the RA Systems have in common.

### 4.2.2 Retrieval Arm Concept Selection

In order to select the superior concept to be further developed, a trade-off table was created and used. This table is divided into several sections. The first section lists all the mandatory capabilities that each concept must possess. However, since all of the concepts have passed the previous screening, these criteria have been met. The next section lists the optional capabilities, which shows the concept's flexibility and usefulness. The final section lists all the design considerations. This section includes factors such as durability, safety, complexity and most certainly includes cost.

*Table 4.3 Trade off Table for Retrieval Arm Concepts*

<b>Considerations</b>	<b>Weighting Factor</b>	<b>Telescopic Arm</b>	<b>Partial Linear Slide Arm</b>	<b>Telescopic Arm with Table</b>	<b>Arm with Sliding Table</b>	<b>Arm with LCB Pull in System</b>
<b>Mandatory Capabilities</b>						
Operate on both left and right sides		yes	yes	yes	yes	yes
Pick up moving forward and backward		yes	yes	yes	yes	yes
Operate up to 10 Mph		yes	yes	yes	yes	yes
Handle the standard Caltrans cones		yes	yes	yes	yes	yes
Install on current drop box		yes	yes	yes	yes	yes
No interference for manual operation		yes	yes	yes	yes	yes
<b>Optional Capabilities</b>						
Complexity of retrieval	4	2	2	0	5	5

table/landing						
Able to stabilize cone on dispatch	3	0	0	0	3	3
Eject defective cones	2	0	0	0	0	0
Quick conversion for change in direction	3	5	5	5	5	5
Weight and size of arm	4	2	2	2	4	5
Modifications required of other subsystems	3	4	4	4	5	1
<b>Design Considerations</b>						
Width of design	5	4	4	4	5	5
Positive grip on cone	4	2	2	2	5	2
Operate in variable weather conditions	5	4	4	4	5	4
Safety of mechanism	5	3	3	3	3	3
Minimum number of actuators	4	2	2	2	2	5
Safety of traffic	5	4	4	4	4	3
Setup upon arrival	3	5	5	5	5	5
Aesthetics	3	3	2	2	3	3
Estimated cost	3	2	1	3	3	3
Minimum sensors and control	4	2	2	2	5	1
Durability & robustness	5	3	2	3	3	2
Minimum operators view obstructions	2	3	3	2	2	3
Compatible with various cones	3	4	4	3	3	1
Operate in various road conditions	4	3	3	3	3	3
Scrubbing of cone	3	3	3	3	4	3
Compatible with existing stowage mechanism	5	4	4	3	5	1
	Totals	246	235	228	314	248

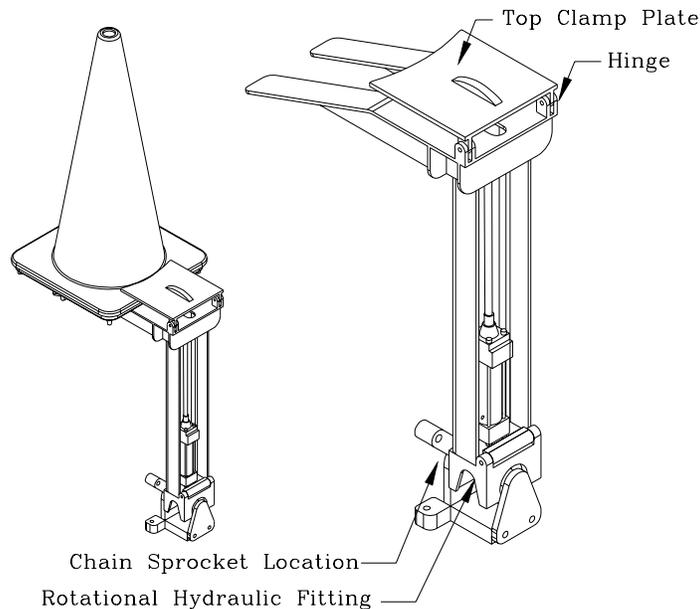
The total scores in the trade-off table clearly indicate the Arm with Sliding Table as the superior design to be further developed and tested.

#### 4.2.3 Retrieval Arm Design

To further develop the Arm with Sliding Table concept, preliminary component and assembly drawings were generated. The concept drawing allowed the fabrication shop personnel to create a test fixture which provided a way of quickly testing questionable aspects of the design and quantifying some operational parameters of the concept.

Figure 4.6 shows the first assembly drawings for the Arm with Sliding Table concept. The assembly on the left shows the concept with a cone in the gripper, while the assembly on the right is a larger scale version without the cone. The gripper opens by simply having the

hydraulic actuator extend and move the rod, connected to the Top Clamp Plate, up thus forcing the plate to rotate about its two stationary hinge points, to the open position. The main shaft cover is purposely not shown in the arm assembly drawing to show additional detail. The hydraulic actuator is mounted as low as possible to minimize the moment of inertia of the assembly. Additional space in the bottom rotational area is strategically created to allow room for a rotational hydraulic fitting and a chain sprocket. This concept also has a break away mechanism with the hinge located at the bottom of the arm shaft.



*Figure 4.6 Arm with Sliding Table Concept, Assembly Drawing*

The point of rotation of this arm is determined by the geometry of the component layout. The area on which the cone is clamped needs to be co-planer with the plane of a cone bottom that is lying down ready for its retrieval. The cone clamping area also needs to come in contact with the cone base about the edge's midpoint. Measuring along the arm's length, from the center of the top plate, would pinpoint the arm rotational axis. Determining the total length of the arm is an iterative process which is determined by the geometry discussed above, the vertical drop distance created by the sliding ramp, and the height of the LCB. The optimum resulting arm length is quite long. This length causes interfacing difficulties with the secondary funnels since the cone needs to be ready for retrieval much earlier and further away from the drop box. This long arm also requires extra torque for upward rotation. A shorter arm creates a LCB interface dilemma as the cone cannot reach the height required to properly slide onto the LCB.

The solution to this dilemma was a modified LCB system. Since the drop height from the LCB onto the ground during a cone dispatch was preferably minimal, the LCB subsystem was modified to include an actuated outer section. During the cone dispatch this section of the belt inclined down at a 30° angle, thus dropping the cone much lower to the ground. This same section can be rotated upwards to meet the desired interfacing height of the retrieval arm.

#### 4.2.3.1 Retrieval Arm Clamp Test Fixture

In order to determine some preliminary operating parameters for the retrieval arm system, a clamp test fixture was built (see Figure 4.7). The main purpose was to estimate the amount of force required on the cone base to ensure a good positive hold on the cone during retrieval. The force required would dictate the hydraulic clamping cylinder size. Also, the amount of wear and the deflection of the cone base during this applied force needed to be established. The ideal sliding ramp needed to be determined and tested. This testing included determining the behavior of a cone traveling the path down the sliding ramp and then on to the LCB. All the tests included wet weather simulation.

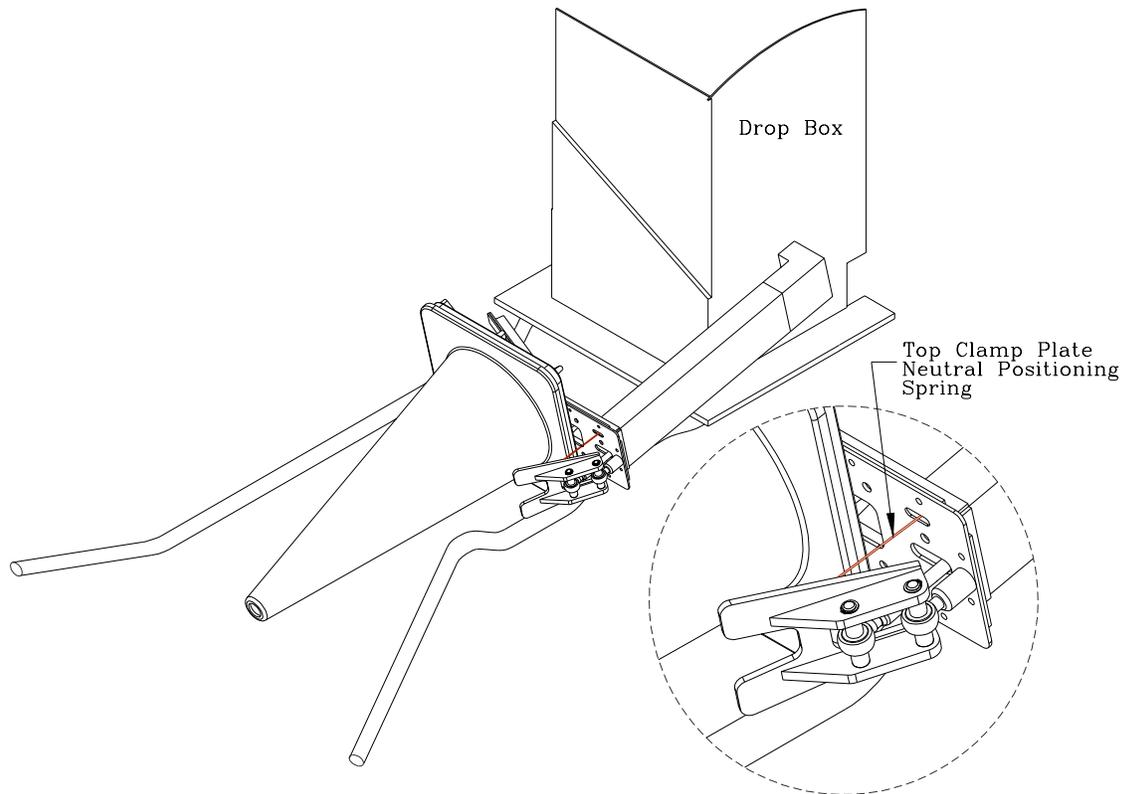


*Figure 4.7 Clamp Testing Fixture*

The clamp test fixture and a bar with a spring scale attached, was used to determine the amount of clamping force at the cone base. The initial considerations were based on the maximum initial forces encountered by the cone during fast upward arm rotation. These calculations provided a threshold level to which an additional 50% was added as a safety factor. This force was applied to the center of the cone base perpendicular to the clamped cone base edge. Other tests included just simply pulling on the cone base to see if a good hold was achieved.

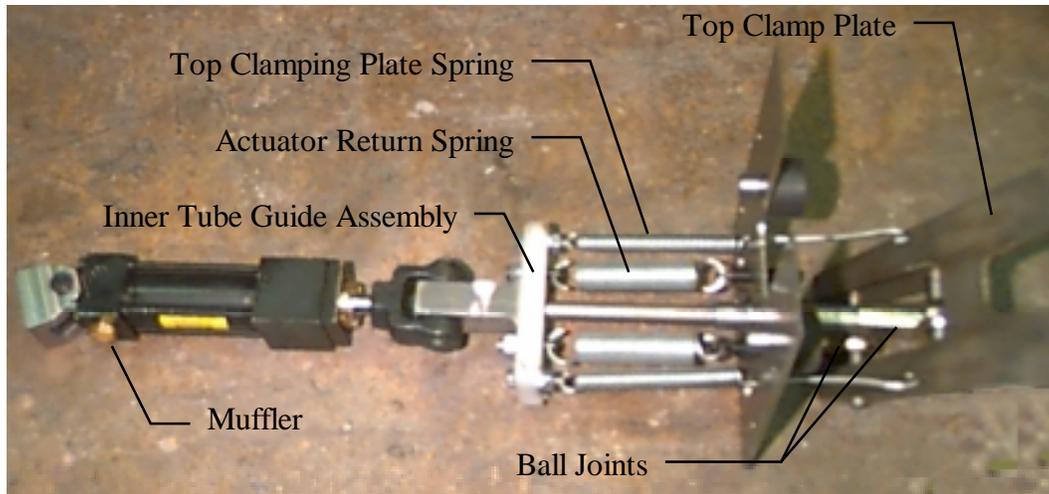
#### 4.2.3.2 Retrieval Arm Design Details and Further Testing

Knowing the design parameters and operation limits allowed the creation of the first set of detail drawings. The detail assembly drawing for this arm is shown in Figure 4.8. The figure shows a cone, guided by the secondary funnel to the arm, that is ready for retrieval. The arm is shown attached to a conceptual drop box, however, the rotational actuator and the chain and sprocket configuration are not shown. The figure also shows a close up of the clamping mechanism.



*Figure 4.8 Retrieval Arm, Detail Drawing*

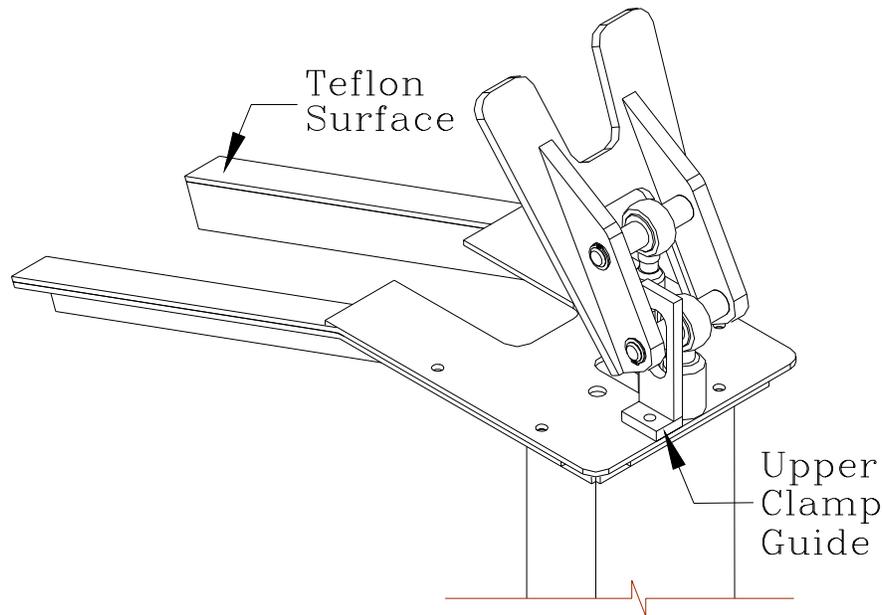
A thin walled rectangular tube was specified for the arm shaft due to its higher torsional rigidity compared to an open channel with a cover. A single acting 2.5 cm (1 inch) diameter hydraulic cylinder was chosen to power the clamping mechanism. This actuator has a muffler installed in the open port to prevent dirt and moisture from entering the cylinder. The inner tube guide assembly, which guides the motion inside the tube and has spring anchors mounted to the top surface, is mounted on top of the cylinder. The top clamp plate is mounted on two ball joints to provide it with a roll degree of freedom. The top clamp plate maintains a nominal neutral position using two springs which easily deflect during clamping. The system as a whole is shielded from water and the specified cone sensing switch is water proof and a commercially available item. The arm utilizes only one cone position sensing switch which has a long stainless tap extending into the cone's arrival path during the retrieval. The final geometry layout specified a 56 cm (22 inch) total arm length and a sliding table inclination of 20°. This arm was fabricated and assembled on the testbed ACM for further testing prior to committing this design to the ACM. As the prototype arm was assembled an inner arm shaft assembly picture was taken (see Figure 4.9), which shows features of the original configuration.



*Figure 4.9 Retrieval Arm, Inner Shaft Assembly*

The prototype arm mounted on the testbed functioned remarkably well during the laboratory testing. The arm held the cone securely for up to five or six, back and forth, maximum rotational acceleration cycles, a demanding duty cycle never encountered during normal retrieval operations. The only questionable aspect of the design was the top clamp plate neutral positioning springs' durability and functionality. These springs were the only aspect modified in the arm's transition to the final design.

The final arm only needed one final modification. This consisted of removing the top clamp plate springs and designing an upper clamp guide. The guide designed allows the top clamp plate to rotate about its roll axis only when the clamp is in the clamping position. When the plate rotates upward, its roll axis is restricted. In the full open position the plate is in a neutral fully open position. Figure 4.10 provides a closer look at the clamping mechanism and clearly shows the new upper clamp guide.



*Figure 4.10 Clamping Mechanism with Upper Clamp Guide*

The final retrieval arm design produced a unique and fully functional system that possesses all the desired features needed to effectively retrieve a cone. This retrieval arm system has a secure positive grip on the cone except for the quick transition to the lateral conveyor. Slightly longer than the first generation arm, it allows for additional cone stabilizing guides on the bottom of the drop box. Due to the elimination of the poker arm, the effective length of the secondary funnel is less.

#### 4.3 Conceptual and Detail Design of the Second Generation Secondary Funnel

Once the retrieval arm was designed, the requirements for the secondary funnel design were defined. The secondary funnel receives the cone from the primary funnel and guides it to the retrieval arm in such a manner that successful retrieval is ensured. Since the second generation arm has no poker arm, the new funnel orients the cone in the correct retrieval position closer to the drop box. A longitudinally aligned cone tip orientation is no longer critical since it is not necessary for retrieval with the new arm. The above differences result in a shorter overall secondary funnel.

Only a few operational improvements were needed to the first generation secondary funnel system. The funnel is required to retract automatically. The retraction should be powered by the drop box system retraction and occur simultaneously. The funnel should be higher to prevent the cone from rolling over the outer funnel bend section. The outer funnel should allow operating clearance for the top clamp plate of the retrieval arm. The plate in the open position should be above the outer secondary funnel so that it will not hinder the entrance of a cone. The

last desired improvement is a modular design of the outer funnels to minimize the need for spare parts.

Numerous concepts for the secondary funnel were generated and considered. The three final concepts are the Body Pin Retracting Concept, Cable Retracting Concept, and the Folding Retraction Concept. All these concepts are functionally equivalent but differ in the retraction mechanism. The retraction is only needed in the outer funnel, and the inner funnels are fixed to the drop box.

#### 4.3.1.1 Body Pin Retracting Concept

In this concept, the outer funnel has a hinge near the middle of its section just beyond the funnel bend. This hinge has an upward pointing pin mounted on a short moment arm that is connected to the outer bend section. During the drop box retraction, the pin comes in contact with the underside of the cone body. The contact point at the cone body is a horse shoe shaped receiver to help align the contact with the upward pointing pin. As the drop box continues to retract, the pin is restricted from moving inward any further. This restriction activates the hinge and effectively turns in and retracts the outer bend section of the outer funnel. This hinge has a stiff torsional spring which normally keeps the outer bend funnel section in the deployed position.

#### 4.3.1.2 Cable Retracting Concept

This concept is similar to the Body Pin Retracting concept and has a similar hinge. The main difference is the way the hinge is activated. This concept has a cable attached which runs inside the stationary outer funnel section. At the drop box, this cable is routed so that it can be connected to the cone truck frame. The cable length is adjusted so that the cable is pulled tight when the drop box is fully deployed. The cable can have some spring tension and compliance to keep it from breaking during accidental over extension and to always keep the cable taut. The hinge on the outer funnel section has a stiff torsional spring that automatically retracts the outer funnel bend section when the cable is not taut.

#### 4.3.1.3 Folding Retraction Concept

In this concept the entire outer funnel will be retracted inward about a rotation point beneath the drop box. The outer funnel will be one continuous section of tube without any hinges. Each funnel is modular and inserts into a rectangular steel tube sleeve section beneath the drop box. These sleeve sections each have their own rotation point location beneath the drop box. Since they rotate about their own point, these sleeves are connected by slotted connecting links so that if one funnel rotates inward or outward, it will force the same movement on the other funnel. The forward funnel sleeve has the first bar of a three bar linkage attached. The third link is attached to the truck frame. This configuration will force the movement of the forward sleeve section during the drop box retraction and deployment. Again, since the sleeve sections are connected, both sleeves retract or deploy simultaneously. This whole retraction mechanism operates whether or not an outer funnel is inserted into a sleeve section.

### 4.3.2 Secondary Funnel Concept Selection

The concepts were evaluated by utilizing the trade-off-table 4.4. Based on the scoring and design reviews, the Folding Retraction Concept was developed and tested.

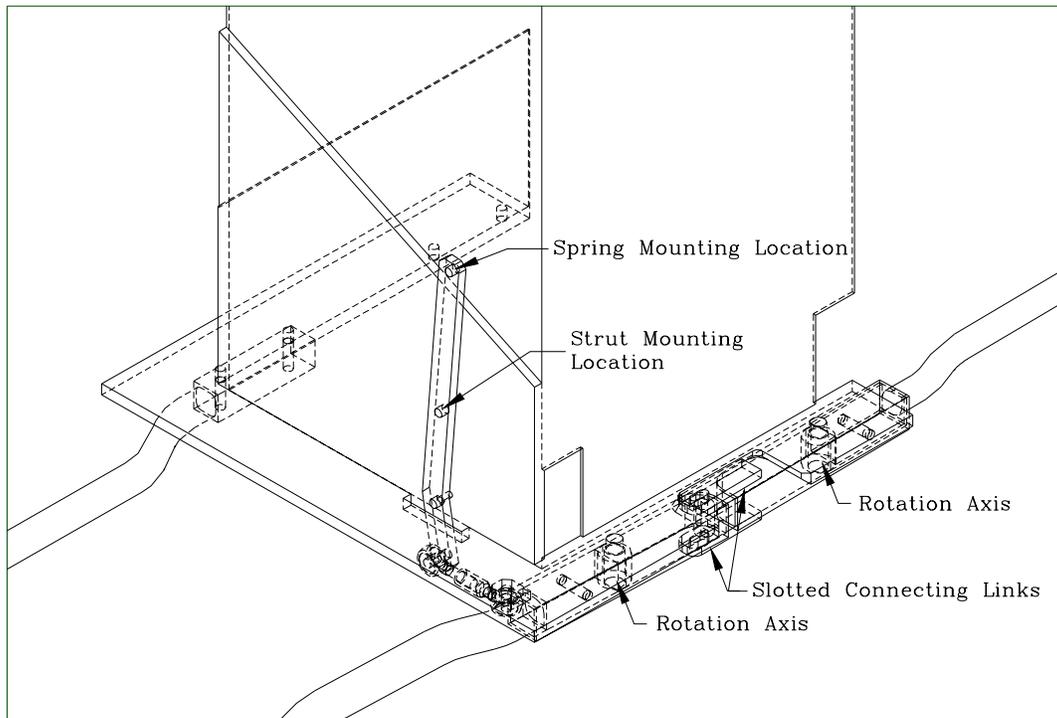
*Table 4.4 Trade off Table for Secondary Funnel Concepts*

<b>Considerations</b>	<b>Weighting Factor</b>	<b>Body Pin Retracting</b>	<b>Cable Retracting</b>	<b>Folding Retraction</b>
<b>Mandatory Capabilities</b>				
Operate on both left and right sides of truck		yes	yes	yes
Guide cones moving forward and backward		yes	yes	yes
Operate up to 10 Mph		yes	yes	yes
Guide the standard Caltrans cones		yes	yes	yes
Install on ACM Drop Box		yes	yes	yes
Automatically retract with Drop Box motion		yes	yes	yes
No interference with Top Clamp Plate operation		yes	yes	yes
Proper interface with RA System		yes	yes	yes
No interference during manual operation		yes	yes	yes
<b>Optional Capabilities</b>				
Modular outer funnel	3	3	0	5
Eject defective cones	2	0	0	0
Quick conversion for change in direction	3	4	3	5
Ease of disassembly	4	4	3	5
Ease of replacing damaged funnel	4	2	1	5
Weight and size of retraction mechanism	4	4	5	3
Modifications required of other subsystems	3	5	5	5
<b>Design Considerations</b>				
Width of design	5	4	4	4
Operate in variable weather conditions	5	4	4	4
Safety of mechanism	5	3	4	4
Safety of traffic	5	4	4	4
Setup upon arrival	3	5	5	5
Aesthetics	3	2	3	4
Utilization of commercially available parts and stock metal	4	3	4	4
Estimated cost	3	4	4	3

No sensors and control	3	5	5	5
Durability & robustness	5	4	2	5
Dependability of Retraction	5	3	3	5
Compatible with various cones	3	4	4	4
Operate during various road conditions	4	5	5	5
Compatible with existing Primary Funnel	5	5	5	5
	Totals	294	285	331

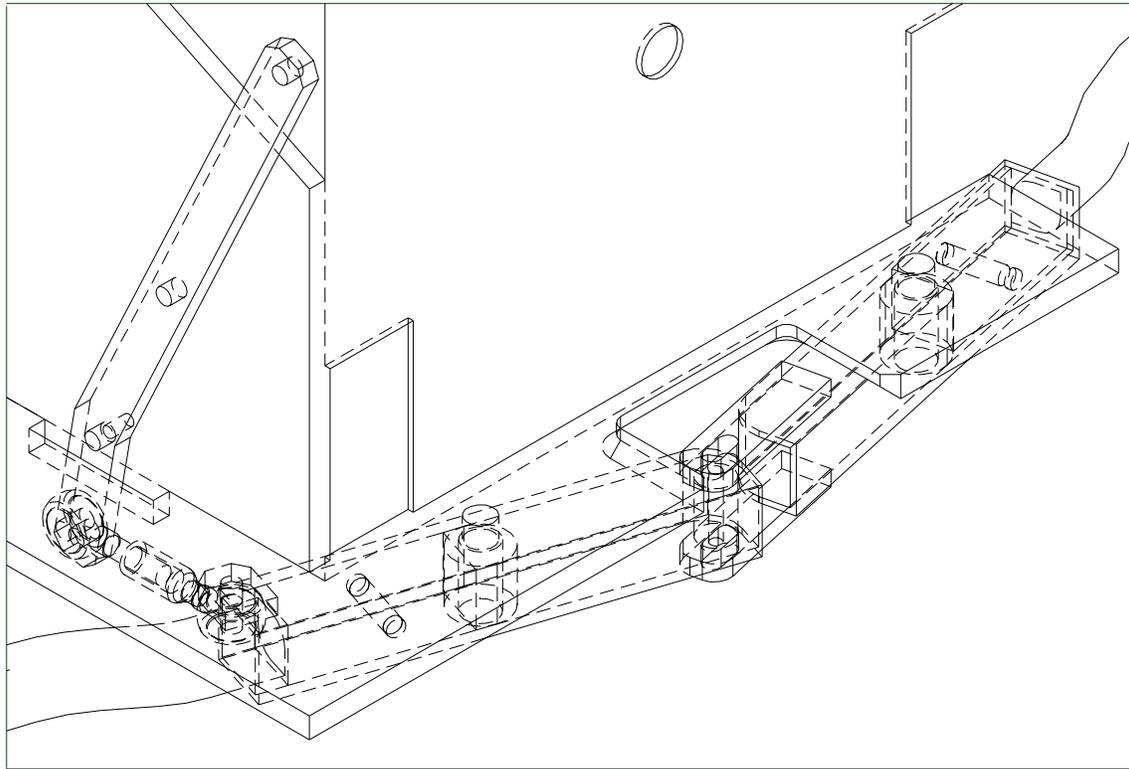
### 4.3.3 Secondary Funnel Design

The concept was further developed and detail drawings were generated. The design utilizes maximum amount of commercially available items and standard stock metal configurations. The sleeves are stock 3.8 cm (1.5 inch) rectangular tubing with a 0.32 cm (0.13 inch) wall thickness. The funnel sections are 2.5 cm (1.0 inch) diameter electrical conduit which are pressed into and welded to a 3.2 cm (1.25 inch) rectangular tubing which inserts into the sleeve weldments. Since the three bar linkage is orthogonal to each bar respectively, the lower (first) link is comprised of two ball joints to allow some relative angular displacement due to the rotational motion of the sleeve weldment. The two ball joints also allow length adjustments, since they are connected with male and female threads. The final link is connected to the truck frame and is a strut type cylinder that breaches the 55.2 cm (21.7 inch) traveled by the drop box. The activation distance required to deploy or retract the sleeves is only a fraction of the drop box distance traveled. The outer funnel has a dip at the retrieval arm top clamp plate location to allow the plate to freely move over the secondary funnel, while still keeping the rest of the funnel high enough to ensure proper functioning. An isometric assembly drawing of the deployed secondary funnel system is shown in Figure 4.11.



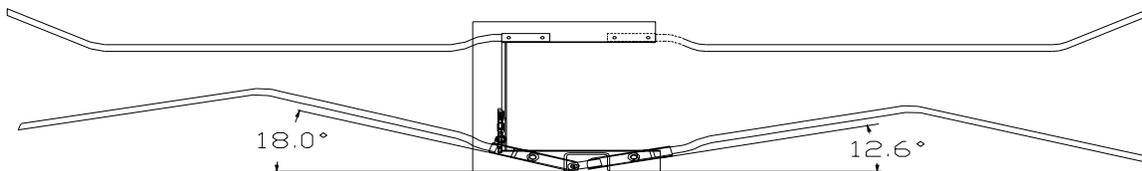
*Figure 4.11 Deployed Secondary Funnel System*

The two sleeve weldments, into which the funnel sections insert, are held to the drop box by 1.91 cm (0.75 inch) diameter stainless steel shoulder bolts. The funnel sections are held in the sleeve weldment by a clevis pin. The two sleeve weldments are connected by the slotted connecting links which have shoulder bolts inserted in the slots. None of these fasteners are shown in the assembly drawing. In the deployed position the rear sleeve upper slotted connecting link rests against the drop box and prevents the system from over-deployment. A closer assembly view of the funnel system in the retracted position is shown in Figure 4.12.



*Figure 4.12 Retracted Secondary Funnel System*

Since the truck cab is narrower than the cone body bed, the front outer funnel section needs more angular retraction than the rear outer funnel. The sleeves were designed so that this angular variance is accomplished by unequal sleeve length. The main difference in sleeve weldment length is due to the slotted connecting links which are permanently mounted to the rear sleeve weldments making them longer. The top view of a drop box and secondary funnel system is shown in Figure 4.13, which also shows the angular displacement of each funnel in the retracted position.



*Figure 4.13 Retracted Right Side Secondary Funnel System, Top View*

At the end tip of the outer funnel section, no downward pointing drag devices were specified at the design stage. It was unknown at this point how a forward pointing funnel would behave on the open road, especially when it needs to move or deflect over bumps. Either non-sparking wire, a semi-circular UHMW Polyethylene shape, or a caster could be a consideration for the funnel ends.

#### 4.4 Assembly and Testing of the Second Generation Components

The focus of this section is the testing of the second generation designs developed. Initially each subsystem was tested in manual mode to make sure that the correct operation and motion occurred. During this manual testing, the need for some preliminary modifications became apparent which were then performed prior to full automation mode testing. The testing of the retrieval arm and secondary funnel systems was done simultaneously since the two subsystems interface so closely.

##### 4.4.1 Retrieval Arm System

After fabrication the retrieval arm had its operation manually tested in both a laboratory setting and on the road. The arm was built to the design specifications and the arm's as-built top portion is shown in Figure 4.14. The cone sensing switch and the bumper stops are labeled in this figure. The bumper stops were specified in the design to stop the upper clamp so that the hydraulic cylinder is prevented from bottoming out when accidentally actuated for the case when no cone is present.



*Figure 4.14 Retrieval Arm, As Built*

The rotational and break away mechanisms at the bottom of the arm are shown in Figure 4.15. In the figure, the cover for the chain housing has been removed to show extra detail, including the chain and the hydraulic fluid line traveling up the center of the housing. Also shown is the cone sensing switch wire coming out of the bottom of the rotational mechanism. The retrieval arm system, in operation, is shown in Figure 4.16.

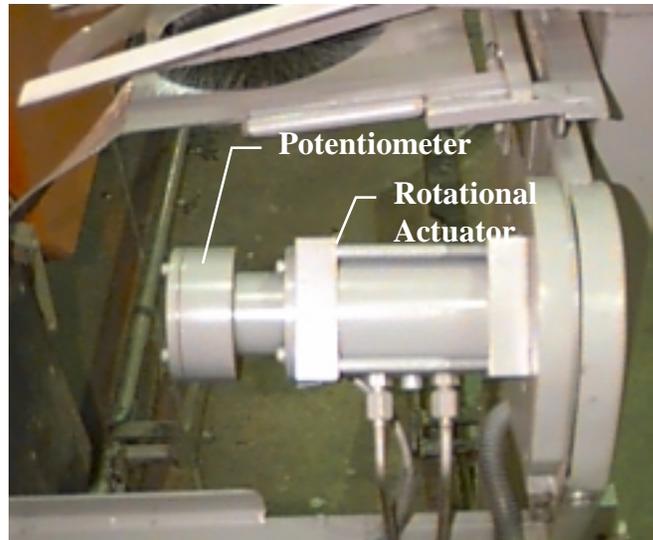


*Figure 4.15 Retrieval Arm, Rotational and Break Away Mechanisms*



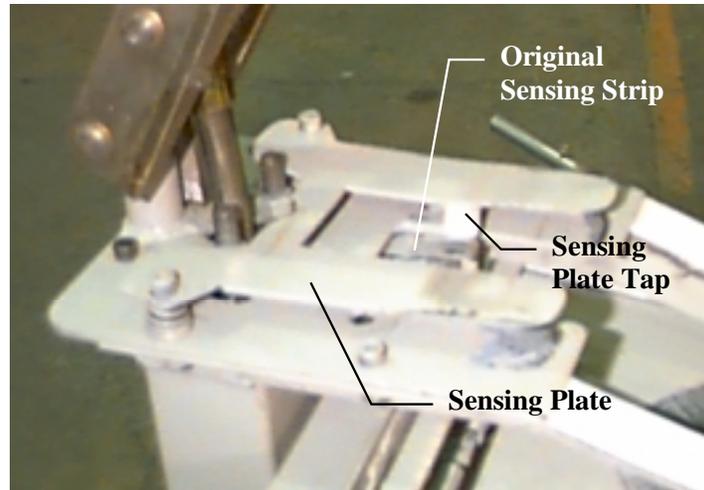
*Figure 4.16 Retrieval Arm, In Operation*

The rotational actuator is mounted in the front-top corner of the drop box with the potentiometer attached to its back surface, as shown in Figure 4.17.



*Figure 4.17 Rotational Actuator with Potentiometer*

Early in the manual testing, it was revealed that the cone sensing switch was not sufficiently rugged and the stainless steel sensing strip was sometimes bent or deflected. Another problem with the switch occurred when the cone bottom plane would not perfectly meet the top cover plate plane of the RA due to relative road height variations. This sporadic occurrence resulted in the prevention or the delay of the activation of the switch. The solution was a sensing plate mounted above of the cone sensing switch. The sensing plate was mounted with a spring compliant hinge which would allow the plate to rotate downwards when contacted by a cone and thus activate the switch. The spring would also return the sensing plate to its original position when not contacted. The original switch is activated by the sensing plate tap pointing towards and positioned directly above the original sensing strip. The sensing strip was cut shorter to accommodate the sensing plate. This modification would allow activation of the switch even when the cone was not perfectly aligned with the top of the retrieval arm, and would also protect the sensing strip from permanent deflection. Another benefit of this solution is that it maintained the original switch which is waterproof and cost-effective. The only drawback of this solution was that the bumper stops had to be removed. The cone sensing switch along with the shortened original sensing strip, the sensing plate tap, and the sensing plate are shown in Figure 4.18.



*Figure 4.18 Cone Sensing Plate*

Besides some of the minor complications and corrections discussed above, the retrieval arm system tested very well during the manual testing and was further tested in the automated mode.

#### 4.4.2 Secondary Funnel System

The secondary funnel system was fabricated and had its operation manually tested in both a laboratory setting and on the road. The system was built to specification and the underside of the drop box with the folding mechanism attached, in the stowed position, is shown in Figure 4.19.



*Figure 4.19 Stowed Secondary Funnel Folding Mechanism*

The second bar link of the secondary funnel folding mechanism consists of ball joints and is shown in Figure 4.20.



*Figure 4.20 Ball Joint Link of Secondary Funnel Folding Mechanism*

The strut link that was originally designed and manufactured for the third and final link is shown in Figure 4.21.



*Figure 4.21 Secondary Funnel Folding Mechanism Strut Link*

Due to deviations in the drop box location, the clearance in the area of the strut link was reduced. As a result truck cab consistently came in contact with the strut link and a link was designed. The new link is mounted on the side of the drop box and deploys the funnel by spring action. During retraction of the drop box, this link comes in contact with the truck frame. As the drop box continues to retract, the rod is forced through its holder and compresses the spring, thus retracting the funnel system. This new link is shown in the deployed position in Figure 4.22 and in the retracted position in Figure 4.23.



*Figure 4.22 Final Strut Link in Deployed Position, Left Side*



*Figure 4.23 Secondary Funnel in Retracted Position, Right Side*

Another result of the changes in drop box location is that the previously required dip in the outer funnel for the upper clamp motion is no longer needed. This greatly simplifies the outer secondary funnel. During testing, it was discovered that the outer funnel could be even shorter than originally designed.

An aspect of the funnel system that was not yet determined in the design phase was the best way to keep the ends of the outer secondary funnels from dragging on the road. The solution implemented during the manual testing was the mounting of a caster at the funnel ends. The casters would normally not come in contact with the road surface, and only touch the surface when elevation differences were encountered.

The changes described resulted in outer secondary funnels that deviated somewhat from the original design. These changes were made during the manual testing of the funnels and the final outer secondary funnel configuration is shown in Figure 4.24.



*Figure 4.24 Final Outer Secondary Funnel Configuration, Left Front*

#### 4.4.3 Testing Results and Final Modifications

The secondary funnel and folding mechanism functioned very well during the manual testing. The outer funnel configuration received some modification during the manual testing of the SF, but the final outer secondary funnel configuration functions in the same manner as the previous funnels. The strut link of the folding mechanism received a major modification but created an overall improvement to the system.

After concluding the manual testing the automated controls were implemented. During these changes the machine was continuously tested to ensure that all the systems were operating correctly and interfaced properly with each other. A significant amount of tuning and adjustments were required but the retrieval arm and secondary funnels continued to function well.

The only potentially serious retrieval problem that may occur is due to road height variations. With severe variations the edge of the cone base could be either gripped too high or too low. As a result, when the arm releases the retrieved cone to slide onto the lateral conveyor belt, it would not be properly aligned and would be prevented from moving towards the stowage system. To solve this problem, guides that align the cone were added but, since room for these guides is limited, the solution will not work when the road variations are extreme.

#### 4.5 Conclusions and Recommendations

In the continuing testing and demonstrations of the retrieval arm and secondary funnel only minor further modifications to the design have been made. The mechanical functions of the second generation designs have been very robust. Currently the only outstanding issue is the potential for the system to function incorrectly on road surfaces that deviate significantly from level. If the gripper grasps a cone more than a few centimeters off center, the captured cone will be misaligned when it moves to the lateral conveyor and can jam. This problem has been seen

sporadically, but much less often than anticipated. Several remedies exist but the simplest has been to modify the guides that realign the cone. Since the problem occurs at deviations of about 5 cm (2 in), the condition may occur when transitioning from a shoulder onto the main road.

The retrieval arm redesign has been critical to the acceptance of the automated cone machine concept. Its reliable action and ability to access the forward or rear directions quickly make the cone retrieval process much more flexible. The efforts to integrate a clean automated stowage of the arm and funnel mechanisms with the drop box has made the first generation automated prototype a very functional system.

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## CHAPTER 5

### DEVELOPMENT OF THE SECOND GENERATION STOWAGE SYSTEM

This chapter discusses the development of the second generation stowage system which moves cones between the lateral conveyor system (LCS) and the main conveyor belt. Compared to the other mechanical systems on the ACM, this mechanism is the most complex and it experiences large repetitive loads that make it subject to rapid wear in the outdoor environment. The resulting design has performed well and successfully supported beta testing of the first generation integrated prototype ACM. Section one describes the testbed ACM stowage components requiring redesign. Sections two and three describe the concept and detail designs of the new stowage system while section four describes testing and additional refinements. Section five describes the results of the redesign efforts and identifies additional issues that would be important for commercialization of the system.

#### 5.1 Testing and Modifications to the First Generation Cone Stowage System

The completed testbed ACM lacked several of the original design features due to time and resource constraints. The ACM was only capable of deploying and retrieving cones from the left side of the vehicle in the forward and reverse directions respectively. Although installed, the right side components were never functional, and only one outer gate was permanently mounted on the right side surface of the LCS. In spite of these handicaps, the ACM served admirably as an informational testbed to gain insight and experience in automated cone handling.

##### 5.1.1 System Testing

During the time span of six months, the ACM was extensively tested and modified to produce a functional unit. Testing occurred in summer and winter weather conditions on several different road surfaces and the resulting information was used to make system modifications to improve performance. Initially, the existing stowage system did not meet performance expectations in laboratory testing. A maximum reliability rate of eighty to ninety percent was achieved, while road testing further reduced this percentage. The system needed several modifications to improve consistency and improve successful cone deployment or retrieval performance. Stowage system modifications were divided into three major problem areas: trolley arm, stowage drive belt, and main conveyor.

##### 5.1.2 Trolley Arm Modifications

Initially, the trolley arm was the most critical of the three problem areas. During transition from vertical to horizontal or vice versa, the traffic cones would undergo high accelerations and decelerations that would cause the gripper shoes to lose retention of the cone. Once the system lost grip on the cone, the stowage mechanism would invariably jam and create very high stresses in the system components. This retention problem mainly appeared during the cone retrieval operation where the cones transitioned from vertical to horizontal, prior to stowage in the main stack. However, the problem would also sometimes occur in the deployment operation. The gripper shoes simply could not provide enough moment resistance to offset the abruptness of this transition due to poor placement and low friction. The lateral opening action of the grippers did

not provide a very large lever arm to resist the bending moment caused by the distance between the cone's center of gravity and the engagement location of the grippers. Also, the composition of the traffic cones resulted in a very low coefficient of friction. The varying expansion rates of the cones with temperature also greatly exacerbated both of these effects. Since time constraints did not allow for a complete redesign, a quick and temporary correction was necessary that did not involve major component changes. As a temporary solution to this problem, large machine screws were inserted tips outward into the gripper shoes to bite into the cones and provide additional friction (see Figure 5.1).



*Figure 5.1 Addition of Machine Screws to Gripper Shoes*

An additional problem with the trolley arm was the lack of a hard stop for the condition in which the arm rotated forward to a horizontal position. When a cone was retained in the grippers, the cone hitting the top of the LCS provided a flexible stop for the trolley arm. However, when a cone was not present on the trolley arm, the arm bearings themselves provided the hard stop. Over time, the repeated damage caused the bearings to develop excessive play which in turn resulted in increased friction for the stowage drive belt system. A further concern was the variance in gripper engagement depth inside the cone due to the lack of a set positioning device. To correct these problems, an adjustable hard stop consisting of a steel bar and a rubber bumper bonded to a stud was added (see Figure 5.2).



*Figure 5.2 Trolley Arm Adjustable Hard Stop*

### 5.1.3 Stowage Belt Tensioning Modifications

The second problem manifested itself as belt slippage over the drive pulleys as the system was subjected to the heavy lifting loads induced by the traffic cone transitioning from a vertical to a horizontal orientation. The required force to lift and rotate the cones was twice the anticipated value. This indicated that the amount of friction or stiction in the mechanism was higher than the conceptual values due to loose tolerances in the machined parts and the excessive play generated in the bearings. A spring mechanism was added to assist the belt in providing additional force during cone transition (see Figure 5.3).

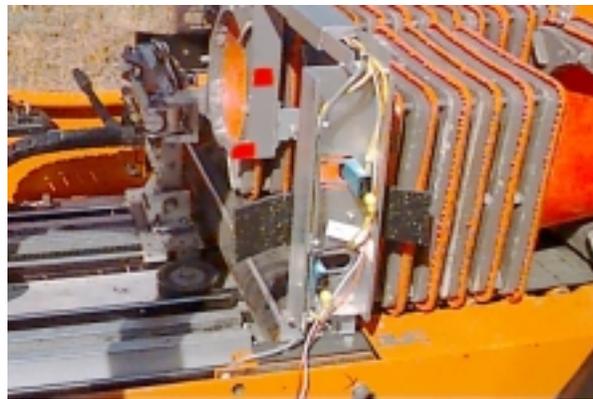


*Figure 5.3 Spring Assist Mechanism*

Belt slippage over the drive pulleys would cause damage to the teeth on the belt and would perpetuate the slippage. To decrease this effect two spring tensioning devices were added. The first device constituted several short overlapping pieces of shaped spring steel clamped between the belt and the right stowage trolley. Additional testing determined that this still did not provide enough force to eliminate belt slippage and another tensioning device was added. The second tensioner, located on the left trolley, consisted of a spring-loaded lever that rotated around a shoulder bolt to provide the additional amount of belt take-up and tensioning required.

#### 5.1.4 Main Conveyor Cone Retention Modifications

The last problem area for the testbed stowage system was the interface between the stowage system and the main conveyor cone stack. The first cone in either side of the stack had a tendency to slide forward and off the main conveyor's forward roller and jam the stowage system trolleys. The problem was caused by the design of the trolley systems that placed the cones just forward of the roller crown so that any type of vibration or truck motion would provoke slippage onto the stowage system. A simple weldment, containing a steel bar and two mounting tabs, was added just forward of the conveyer crown to function as a cone retention device (see Figure 5.4). The bar placement was designed to keep the cones on the conveyor without overly impeding gripper retention of the cones during deployment. Additional modifications, in the form of material removal by grinding, were performed on the rear segments of the stowage trolleys to help alleviate the problem by moving the cone placement point on the main conveyor rearwards by 0.06 m (0.2 ft).



*Figure 5.4 Cone Retention Device*

#### 5.1.5 Evaluation of Modifications

The modifications to the trolley arms, stowage belt system, and the main conveyor dramatically improved the performance of the stowage system. However, none of these modifications could be considered complete solutions to these problems. Addition of the machine screws to the gripper shoes had the obvious detriment of ruining the cones over time, but did provide a simple solution to the retention problem. After installation, the two stowage system tensioning devices decreased the slippage to an acceptable amount, but did not eliminate the problem. The drive belt would still occasionally slip and cause itself damage, especially when operating in wet conditions. Finally, the cone retention device worked very well, but harsh vehicle motions could still cause cone loss. Additionally, the modified system did not meet the cycle times of 1 to 1.5 seconds required to deploy cones at 16km/hr (10 mph). The best recorded cycle times averaged at 3.4 seconds for a set of five cycles. Proper solutions to these problems would require the complete redesign of the stowage system.

## 5.2 Conceptual Design and Development of the Cone Stowage System

In this section, the development of the second-generation stowage system is covered. This development includes the creation of multiple design concepts for the gripper assembly, transfer mechanism, and drive system. After the generation of new design parameters, the final designs are chosen and integrated into a complete system.

### 5.2.1 Conceptual Design Requirements

Based upon the testing and modification of the existing testbed stowage system, it was decided that a second-generation system was required to correct the deficiencies. This iteration of the stowage system would be designed for a new cone body vehicle that would become the first integrated prototype for the ACM. New parameters for the second generation stowage system included increasing cone retention during rotational motion, decreasing force required for cone transit, simplifying mechanism complexity, and increasing cone retrieval and deployment speeds. A further design consideration was the diminished working envelope from the original system. The new truck frame was located 0.05 m (0.17 ft) higher in relation to the cone body, requiring a similar reduction in the stowage system depth. The second generation stowage system was also required to interface to redesigns of the LCS and main conveyor designs using the existing interface schemes. Combining these new parameters with the original testbed parameters produced a complete set of design criteria for the conceptual design of the second generation stowage system.

Conceptual design of the stowage system began with a subdivision of stowage system operational tasks into three separate subsystems: gripper assembly, transfer mechanism, and drive system. Each subsystem was evaluated independently of the other two subsystems and system interface issues were ignored by selection of concepts with similar interfaces. Several brainstorming sessions were performed by the ACM group for each of these subsystems to produce a multitude of conceptual ideas. These concepts were then culled to produce a final set of feasible solutions that included three gripper concepts, two transfer mechanism concepts, and two drive concepts.

### 5.2.2 Gripper Concepts

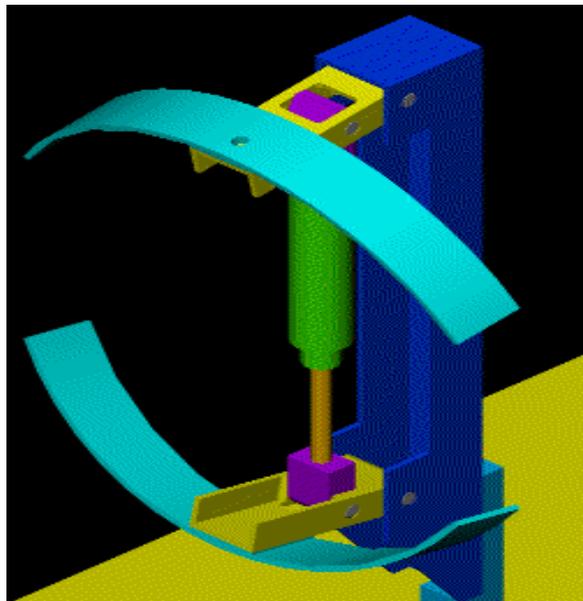
The grippers are the only stowage system component that form the physical link or interface to retain the traffic cone during operation. The three gripper concepts consist of two internal system concepts and one external system concept. The two internal concepts are the modified testbed and the quad system concepts.

#### 5.2.2.1 Modified Testbed Concept

The first gripper system concept is a redesigned version of the original testbed gripper system (see Figure 5.5). Instead of the original T-shaped upper arm configuration, the grippers are rotated 90° to form an I-shaped upper arm. Rotating the grippers into this longitudinal or vertical orientation provides a larger resistance to the cone tipping off of the grippers during rotational motion. This increased moment resistance translates into less required surface area for the

gripper shoes due to the decrease in the necessary amount of friction at the gripper shoe to cone interface. The modified testbed concept also places the double-acting hydraulic piston over the gripper arm linkage pivots to minimize depth. Both the decrease of gripper shoe size and the relocation of the hydraulic linear actuator produce an extremely compact system.

Operation of the modified testbed concept is identical to the original system except for the gripper reorientation. For retrieval, after the cone is properly positioned by the LCS and the gate mechanisms, one port of the hydraulic piston is energized and the grippers open into the cone base. The piston is preset to the pressure required to generate the necessary force at the gripper shoe to cone interface. The gripper arms continue to open and expand the cone base until the arms hit a set of adjustable hard stops that limit excessive cone deformation. Cone deployment is similar to retrieval once the cone is properly positioned by the main conveyor and photoeye array. After transit by the transfer mechanism, the hydraulic pressure to the piston ports is reversed and the gripper arms release the cone base and retract. The cone is placed on the LCS or main conveyor stack depending upon the type of operation.

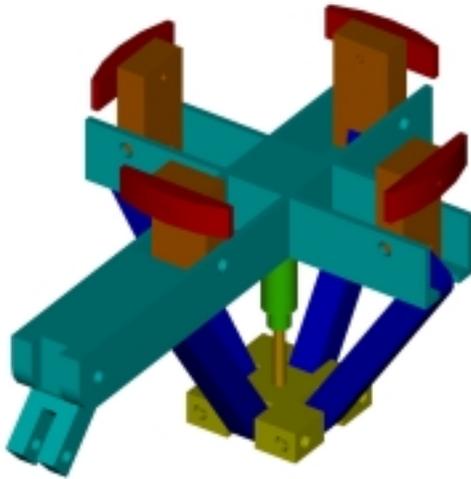


*Figure 5.5 Modified Testbed Gripper Concept*

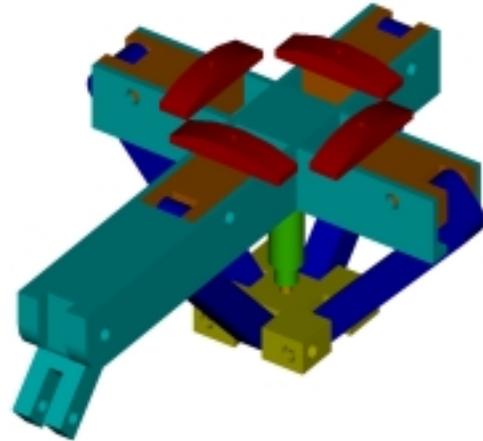
#### 5.2.2.2 Quad Concept

The second gripper concept is a combination of the original testbed and the first gripper concept by combining a lateral or horizontal set of grippers with another set of longitudinal or vertical grippers (see Figure 5.6). Both sets of grippers are actuated by a double-acting piston that connects to the gripper arms by individual linkages. These linkages are L-shaped with the shorter leg connecting to the other side of the gripper arm pivots from the shoes. One end of the piston is rigidly fixed to the upper arm assembly while the rod end is free to move to simplify hydraulic connections. This concept allows the use of very small gripper shoes while still maintaining excellent cone retention. Furthermore, this arrangement of four evenly spaced grippers causes uniform expansion and deformation of the cone base.

Due to the quad gripper arrangement, operation of this concept differs slightly from the modified testbed concept. Proper location of the cones is still ensured by either the LCS and gates or the main conveyor and photoeye array depending upon the type of operation. After cone positioning, the hydraulic piston is extended and forces the linkages to pull open the grippers to grasp the cone base. The transfer system then rotates the trolley upper arm to vertical and transfers the trolley assembly. Upon completion of deployment or retrieval, the piston is retracted and forces the grippers to rotate shut and release the cone onto the LCS or main conveyor.



*Figure 5.6 Quad Gripper Concept - Engaged*

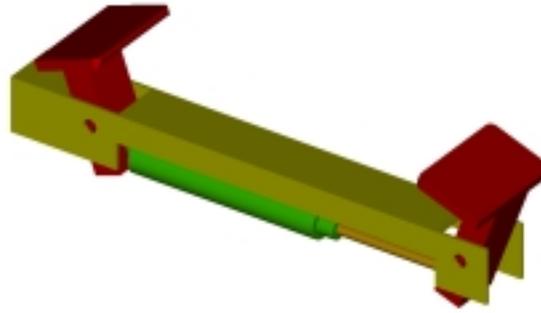


*Figure 5.7 Quad Gripper Concept - Released*

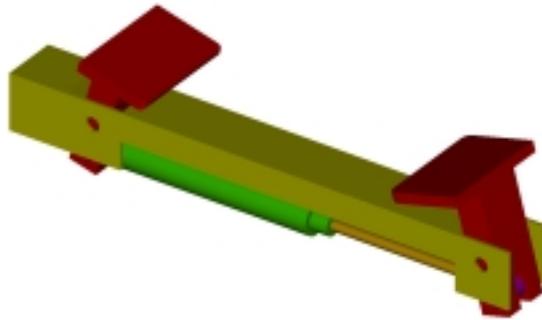
### 5.2.2.3 External Concept

The last gripper system is radically different from the other systems by operating on the outer surface of the cone base instead of expanding the interior and can apply more clamping force than the other concepts (see Figure 5.8). This system is assembled from a machined tube, two gripper arms, and a double-acting hydraulic piston. Similar to the modified testbed concept, the external grippers are oriented longitudinally or vertically and the trolley upper arm is I-shaped. This orientation is not necessary to resist the tipping moment, but keeps the cone from deforming due to inertia during the transfer motion. The piston is required to be positioned behind the gripper arm pivots to not interfere with cone retention. This system is able to handle variations in cone base size and is unaffected by varying cone stiffness due to temperature. Also, the system envelope is very compact.

The external system concept retrieval and deployment operations begin with proper location of the cones by the LCS and gates or the main conveyor and photoeye array. Once placement is accomplished, the hydraulic piston is extended, causing the gripper arms to rotate over the cone base and clamp the base firmly against the flat surface of the trolley upper arm. The system is now prepared for transit. During the end of the transit cycle, the piston is retracted and the gripper arms rotate away from the cone base surface to provide clearance for the cone to disengage onto the LCS or the main conveyor.



*Figure 5.8 External Gripper Concept - Released*



*Figure 5.9 External Gripper Concept - Engaged*

### 5.2.3 Transfer Mechanism Concepts

The transfer mechanism is the stowage system component that rotates and transfers the entire trolley assembly to either deploy or retrieve cones between the LCS and main conveyor. The two transfer mechanism concepts combine rotary and linear motion into compact systems and are the simple track system and the actuator system.

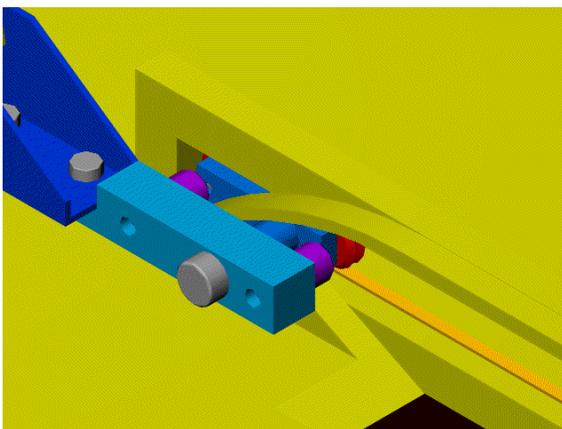
#### 5.2.3.1 Simple Track System Concept

The first transfer concept utilizes a roller and track system to perform both rotary and linear motion requirements (see Figure 5.10). The two components of the transfer concept are the trolley and track assemblies. The trolley assembly consists of the upper arm, lower arm, and cart subassemblies. The upper arm subassembly provides a mounting frame for the gripper system and connects to the lower arm that contains the arm pivot bearings and two cam rollers located on opposite sides of the pivot point. These combined subassemblies connect to the cart subassembly that interfaces to the track assembly. The cart subassembly includes the arm pivot mount and three vertically mounted v-wheels that ride on a matching set of rails connected to the

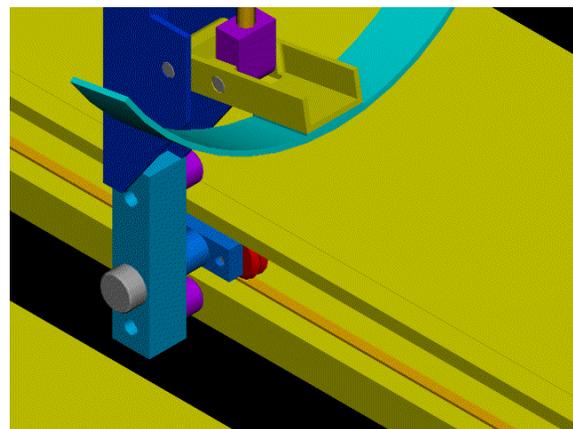
track frame. The rails, track frame, upper track, and lower track form the completed track assembly. The upper and lower tracks provide contact surfaces for the cam followers to perform the rotational motions for deployment and retrieval. The upper track runs the entire length of the stowage system and combines a short circular section at the front of the track frame with a flat section that constitutes the remainder of the track. The lower track is a short sloping section at the bottom front of the track frame. The entire trolley and track assemblies form a compact and space efficient unit. Linear movement of the trolley assembly is accomplished by a single rotary actuator connected to the drive assembly.

The operational sequences for the simple track concept vary for retrieval and deployment operations. The retrieval operation begins with retention of the cone on the LCS by the gripper system. At this stage of the cycle, the trolley assembly is located at the front of the track assembly. Once the cone is captured by the grippers, the cart subassembly is transferred rearwards by the drive assembly and forces the upper cam follower to roll on the upper track. Since the first section of this track is circular, the arm subassemblies are forced to rotate upwards until the cam follower reaches the second section of the upper track. Once at this section of flat track, the upper cam follower forces the arm subassemblies to remain vertical for the remainder of the trolley transfer. The lower cam follower and lower track are not used during retrieval operations. At the end of the trolley transfer, the cone is released by the grippers and deposited on the main conveyor.

The deployment operation begins with the trolley assembly positioned at the rear of the track assembly with the arm subassemblies in the vertical position. After the cone is retained from the main conveyor by the gripper system, the trolley assembly transfers forward as the upper cam follower rides on the second section of the upper track and maintains the vertical orientation of the arm subassemblies. Once the trolley assembly reaches the location of the lower track, the lower cam follower on the lower arm assembly is forced to lag behind the cart subassembly. This forces the arm subassemblies to rotate until the center of gravity of the cone is over the upper and lower arm pivot point. Once the cone center of the gravity passes over the pivot point, the lower cam follower loses contact with the lower track as the upper cam follower contacts the circular section of the upper track. Gravity forces the arm subassemblies to complete the rotation to a horizontal position at the LCS where the cone is released by the grippers.



*Figure 5.10 Simple Track Transfer Concept*

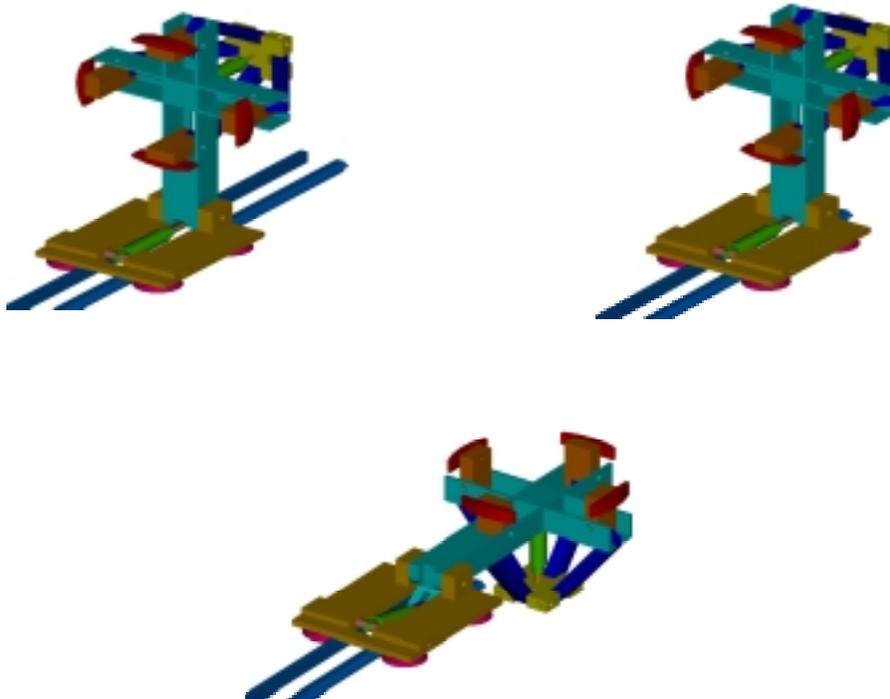


*Figure 5.11 Simple Track Transfer Concept -Rotated Gripper*

### 5.2.3.2 Actuator System Concept

The final transfer mechanism concept involves the use of a double-acting hydraulic linear actuator to facilitate the rotary motion of the trolley assembly, while a rotary actuator connected to the drive assembly controls the linear motion along the track assembly (see Figure 5.12). The trolley assembly contains only the arm and cart subassemblies. The gripper system connects to the arm subassembly which consists of a simple set of bearings and an offset linkage point for the hydraulic piston rod end. The linkage is located opposite the grippers in relation to the bearing pivot. The arm subassembly then connects to a set of block bearings mounted on the cart subassembly, while the other end of the piston connects to a fixed linkage attachment on the same subassembly. A set of v-wheels are mounted horizontally to the underside of the cart subassembly and ride on a pair of guide tracks attached to a mounting plate. The guide tracks and mounting plate form the track assembly. Sensors and hard stops located on the cart assembly are required for accurate positioning of the arm subassembly.

Cone retrieval for the actuator concept is a two-step operation that begins with cone retention by the gripper system. The first step is extension of the hydraulic piston that applies force to the offset linkage and rotates the arm subassembly into a vertical configuration. Next, the rotary actuator and drive assembly, where the cone is placed into the main stack and released, transfers the trolley rearwards to the main conveyor. Deployment reverses the two-step process to place and release cones on the LCS. The actuator concept decouples the linear and rotary motions into separate actions that shorten the length of the working envelope to allow for a longer main conveyor. This translates into more cone storage in the stack.



*Figure 5.12 Actuator System Concept*

### 5.2.3.3 Drive Concepts

The drive system connects a single hydraulic rotary actuator to the trolley assemblies of the stowage system. Two concepts of belt and chain connections are considered.

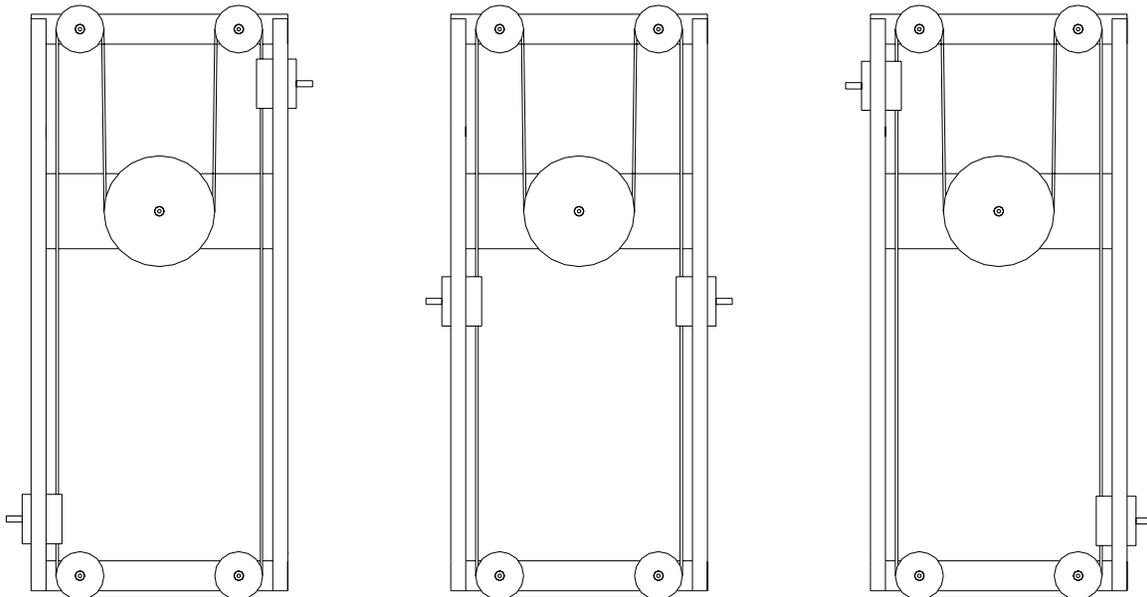
### 5.2.3.4 Belt Concept

The belt connection concept employs a toothed serpentine-style belt to provide power transfer from the rotary actuator to the two parallel trolley assemblies (see Figure 5.13). The belt is powered by a drive sprocket attached to the keyed rotary actuator shaft and is positioned beside the track assemblies with four smooth idler pulleys. Belt tensioning is accomplished by moving the actuator mounting location. The trolley assemblies are individually attached to the belt using machined clamps that engage the belt teeth.

As the actuator rotates counterclockwise, the left trolley is driven forward while the right trolley is driven rearwards. Motion continues until the trolleys reach hard stops at the end and beginning of the track assemblies. Clockwise rotation of the actuator causes opposite motion in the trolley assemblies. This opposite motion ensures that one of the trolley assemblies is always ready to either deploy or retrieve cones as required.

### 5.2.3.5 Chain Concept

The chain connection concept is identical in construction and operation to the belt connection concept except for the substitution of a roller chain drive element instead of the serpentine toothed belt element.



*Figure 5.13 Belt and Chain Drive System Concepts*

## 5.2.4 Evaluation of Stowage System Concepts

After creation of multiple concept solutions for each of the three operation tasks, the selection of an optimal concept for each task was required. To facilitate these selections, individual sets of general and weighted criteria were generated for each task. The weighting factors varied from one to three based upon importance of the criteria to completion of the task while the concept rankings varied from zero to two based upon satisfaction of the criteria. Once the selection of three optimal concepts was finished, an integrated concept for the stowage system could be produced.

### 5.2.4.1 Evaluation of Gripper Concepts

The critical or primary detailed design parameters for the gripper system concepts are a simplification of interface and mechanism complexity, an increase in cone retention performance, a reduction in working envelope, and an increase in safety features. The concept rankings for these primary and other secondary and tertiary criteria are illustrated in Table 5.1 for the modified testbed, quad, and external gripper concepts.

The modified testbed gripper concept offers the simplest mechanism, the smallest working envelope, and the least interface modifications of the three concepts. The mechanism is a robust and modular design that requires a simple modification to the central conveyor of the LCS while presenting a large envelope for the central gate system. No main conveyor modifications are required. However, this concept design provides the least tolerance in cone variation and the lowest cone retention force.

While the quad gripper concept presents minimal changes to the main conveyor interface and the simplest hydraulic connections, this system is too bulky and requires major redesign of the LCS. The large dynamic envelope provides little space for the gate mechanisms and can potentially interfere with the truck frame. Furthermore, the mechanism durability is questionable due to the number and complexity of the linkages.

The external gripper concept is the best concept for providing maximum cone retention and tolerance to cone variations. Additionally, this concept offers a visually clean appearance and excellent flexibility. Unfortunately, this concept requires a complete redesign of the main conveyor and moderate modifications to the LCS. These redesigns will add extensive time to the design cycle for the ACM and are unfeasible.

Table 5.1 Ranking System Table for Conceptual Gripper Designs

DESIGN	Weight	Concept 1	Concept	Concept 3
<b>A. GENERAL REQUIREMENTS</b>				
1. Does not intrude in		Yes	Yes	Yes
2. Easily retrofitted		Yes	Yes	Yes
3. Field serviceable		Yes	Yes	Yes
4. Handles climate		Yes	Yes	Yes
5. Limited access to		Yes	Yes	Yes
6. No patent infringement		Yes	Yes	Yes
7. Option for manual cone		Yes	Yes	Yes
8. Works with standard		Yes	Yes	Yes
<b>B. DETAILED REQUIREMENTS</b>				
1. Aesthetics	2	1	0	2
2. Anticipated time	2	2	1	2
3. Handles variable	3	0	1	2
4. Minimal alterations to	3	2	1	0
5. Minimal alterations to	2	2	2	0
6. Minimal hydraulic	2	1	2	1
7. Minimum sensors	1	2	2	2
8. Minimal setup	1	2	2	2
9. Minimum working	3	2	0	1
10. Minimum linear	2	2	2	2
11. Positive cone retention	3	0	1	2
12. Simplified mechanism	3	2	0	1
13. System flexibility	2	1	1	2
14. System modularity	1	2	0	1
15. Worker safety	3	1	1	1
<b>TOTAL</b>		45	32	44

#### 5.2.4.2 Evaluation of Transfer Mechanism Concepts

For the two transfer mechanism concepts, the primary detailed requirements are minimum static and dynamic working envelopes, minimum interface redesign for LCS, fewest hydraulic connections, simplified mechanism design, and worker safety. Depth is the most critical of the working envelope dimensions due to the raised frame on the new truck. Important secondary requirements are failure frequency, minimum modifications to main conveyor interface, and debris tolerance. Some tertiary considerations are cone storage capacity and system modularity. These criteria and concept rankings are displayed in Table 5.2 for the simple track and actuator transfer concepts.

The simple track system supplies a very compact and elegantly simple solution that requires no hydraulic connections and provides a smooth and low force transfer mechanism. This concept is extremely modular and allows changing the gripper subassembly or drive assembly without serious effort. While no interface changes are necessary for the main conveyor, some minor modifications are required for the LCS interface. The two major caveats about this concept are questionable durability of the cam follower and v-wheel components of the lower arm and cart subassemblies and decreased cone storage capacity. The first problem of component durability can be alleviated by careful detailed design and selection of final components. However, the second problem cannot be corrected. Although the overall working envelope is smaller than the original stowage system, the envelope length has increased to provide a longer transition area for the rotary motion. This increase in system length dramatically smoothes the motion and decreases the required force but causes a corresponding length decrease in the main conveyor.

The main attributes of the actuator transfer mechanism concept are robustness and extreme flexibility. By decoupling the rotational and linear motions of the stowage system, this concept requires less overall force input and can transition the cone from vertical to horizontal without any linear motion. This translates into a decreased length of 0.3 m (1 ft) from the working envelope of the simple track concept. Increasing the length of the main conveyor by this amount adds additional storage capacity for twelve cones in the two stacks. However, this system still requires a larger system envelope to accommodate the larger trolley assemblies caused by the linear actuator. These larger trolley assemblies also necessitate extensive modifications to both the LCS and main conveyor interfaces. A final problem with this concept is the added hydraulic and electrical connections needed for the linear actuators and sensors on the trolley assemblies.

Table 5.2 Ranking System Table for Conceptual Transfer Mechanism Designs

DESIGN REQUIREMENTS	Weight Factor	Concept 1 Simple Track	Concept 2 Actuator
<b>A. GENERAL REQUIREMENTS</b>			
1. Compatible with two horizontal cone stacks		Yes	Yes
2. Does not intrude in bucket seat areas		Yes	Yes
3. Easily retrofitted		Yes	Yes
4. Field serviceable		Yes	Yes
5. Handles climate extremes		Yes	Yes
6. Limited access to moving parts for safety		Yes	Yes
7. No patent infringement		Yes	Yes
8. Option for manual cone handling		Yes	Yes
9. System will operate on both sides of stack		Yes	Yes
10. Transfers cones from LCS to main conveyor		Yes	Yes
11. Works with standard Caltrans cone		Yes	Yes
<b>B. DETAILED REQUIREMENTS</b>			
1. Aesthetics	2	2	1
2. Anticipated time between failures	2	1	2
3. Cycle time of 1.5 seconds maximum	2	2	2
4. Maximum depth envelope of 0.015m (6 in)	3	2	1
5. Minimal alterations to LCS interface	3	2	1
6. Minimal alterations to main conveyor interface	2	2	0
7. Minimal electrical connections	2	2	1
8. Minimal force for cone transit	2	1	2
9. Minimal hydraulic connections	3	2	0
10. Minimum rotary actuators	2	1	1
11. Minimum working envelope	3	2	0
12. Minimum linear actuators	2	2	1
13. Minimum sensors	2	2	1
14. Minimal setup	2	2	2
15. Simplified mechanism	3	2	1
16. Storage capacity for 80 cones	1	0	2
17. System flexibility	2	0	2
18. System modularity	1	2	1
19. Tolerance of debris	2	2	2
20. Worker safety	3	1	1
<b>TOTAL</b>		<b>73</b>	<b>49</b>

### 5.2.4.3 Evaluation of Drive Concepts

Since the two concepts are identical except for the drive element, the only required evaluation is the determination of whether a belt or a chain is a better drive component for this application. Primary, secondary, and tertiary criteria and concept rankings are shown in Table 4.3 for the belt and chain drive concepts.

*Table 5.3 Ranking System Table for Conceptual Drive Designs*

DESIGN REQUIREMENTS	Weight Factor	Concept 1 Belt	Concept 2 Chain
DETAILED REQUIREMENTS			
1. Caltrans compatability	3	0	2
2. Cost	3	0	2
3. Friction	2	2	0
4. Load Capacity	3	2	0
5. Noise	1	2	0
6. Repairability	3	0	2
7. Slippage Resistance	3	1	2
8. Shock Resistance	2	2	0
TOTAL		19	24

While the belt concept provides the highest load capacity and shock resistance with the lowest friction and noise, it is not compatible with current Caltrans equipment and is expensive. Furthermore, belts are prone to slippage under severe loads and must be replaced when damaged. Complicated tensioning systems are often necessary to accommodate belt flex and stretch under load.

The chain concept is compatible with current Caltrans equipment practices, inexpensive, and repairable. Chain slippage is impossible when properly tensioned and chain links can be replaced or overall chain length shortened in the field with minimal trouble. However, chains have a lower load capacity and will break when subjected to high shock loading.

### 5.2.5 Selection of Stowage System Concepts

From the proceeding tables, the modified gripper concept combined with the simple track transfer concept and the chain drive concept proved to be the optimal solution for the second generation stowage system for the integrated prototype ACM. The final integrated concept combines all of these subsystems into a unified system that efficiently transfers cones between the LCS and main conveyor and provides a large improvement over the first generation system.

### 5.3 Detailed Design of the Second Generation Cone Stowage System

This section covers the four part detailed design phase of the second generation stowage system. Specifically, the steps of detail creation, material selection, strength calculations, and manufacturing documentation are explained for the stowage system subsystems.

#### 5.3.1 Design Requirements

After completion of the conceptual design phase with the selection of an integrated stowage system concept, the detailed design phase commenced. This design phase entailed the transformation of the integrated system concept into a detailed and documented design ready for manufacture. This transformation process was separated into the following four distinct actions: creation of design details, selection of components and materials, analysis of components, and documentation of design.

The first action, creation of design details, involved the generation of CAD models for all manufactured assemblies and subassemblies contained within the stowage system. These 3D models were used to perform motion studies and to assure that the completed stowage system design performed as intended. Furthermore, these models were also used to verify the correct tolerancing and dimensioning for all fabricated system components. Careful use of the Mechanical Desktop CAD package from Autocad, assured a proper fit and eliminated rework for design mistakes.

After creation of the CAD models, the material selection was made for the fabricated parts. For the second generation stowage system, structural steel was chosen as the material of choice for most parts and assemblies. Where required, other materials were used. Also, commercially off the shelf components like hydraulic actuators, bearings, mounting hardware were selected and purchased.

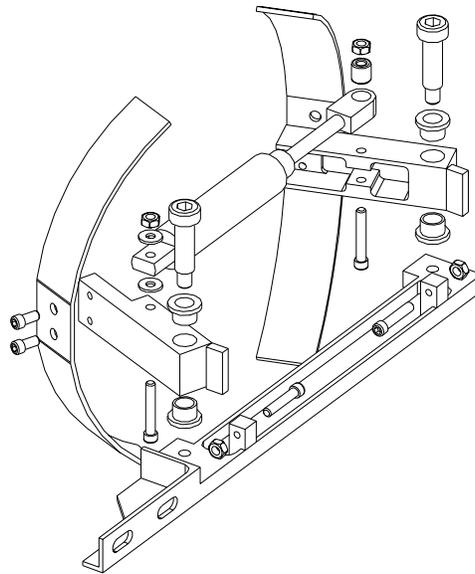
The third action, analysis of components, was performed only on critical assemblies due to the brevity of the detailed design phase. The trolley assembly was analyzed to assist in selection of commercial components and to assure robustness. All other components were over designed to avoid any complications with part failure or breakage due to underestimation of operational loads and forces.

The final action, design documentation, again utilized the Mechanical Desktop CAD package to produce 2D drawings from the 3D models. These drawings were sent to machining shops for quotation and manufacture of the fabricated components.

#### 5.3.2 Gripper Assembly Design

As the system with the most problems on the testbed, the detailed design of the gripper assembly was placed first. The upper arm subassembly from the trolley concept and the gripper assembly concept were unified into a single design by using the upper arm as the mounting frame for the grippers. To decrease the working envelope, the gripper arms were downsized and the pivots cantilevered from the side of the upper arm. Shoulder bolts inside of bronze sintered bushings formed the gripper arm pivots and bearing surfaces. Threaded holes were tapped into

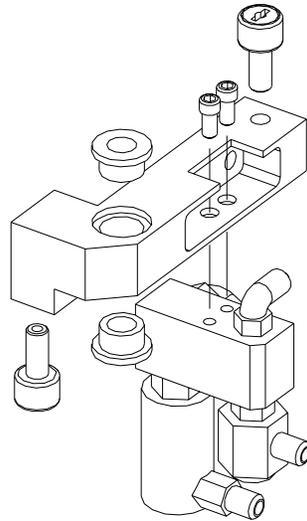
the gripper arms to allow for easy gripper shoe change out to facilitate testing while slots were machined into the backs of the gripper arms to provide mounts for the hydraulic actuators. The chosen hydraulic actuators were a shorter model of the AllenAir pistons used for the testbed grippers due to rapid availability and proven reliability. Welded bosses were added to the upper arm and threaded to provide hard stops for the gripper arms. The final gripper assembly produced a very compact envelope that only required a 0.06 m (0.21 ft) gap in the LCS belts for retrieval and deployment operations (see Figure 5.14). The created a very generous envelope for the central and outer gate mechanisms.



*Figure 5.14 Gripper Assembly*

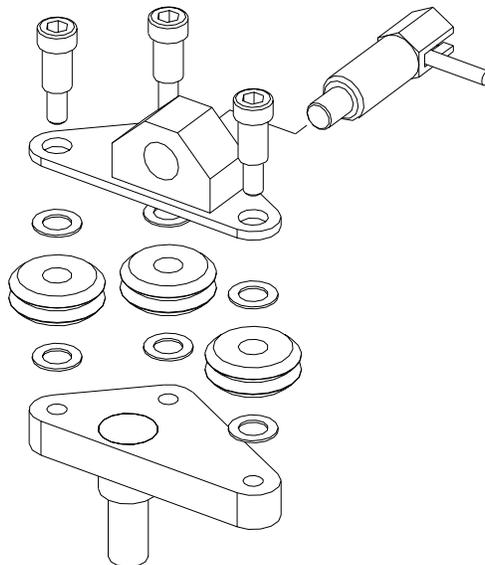
### 5.3.3 Trolley Assembly Design

With the integration of the upper arm subassembly into the gripper assembly, the trolley assembly was simplified into the lower arm and cart subassemblies (see Figure 5.15). The lower arm subassembly consisted of only the lower arm frame, bearings, hydraulic connection manifold, and cam followers for the upper and lower tracks. Again, bronze sintered bushings were chosen in place of roller bearing elements for their simplicity and self lubricating properties. As the bushings and cam followers were critical links in this subassembly, the resultant stresses in these components were analyzed for the worst case condition of a stalled upper arm during retrieval. Given these results, the bushings and cam followers were chosen accordingly. The small hydraulic manifold contained two rotary hydraulic fittings and two fixed fittings to provide hydraulic pressure from the vehicle system to the gripper assembly. Flexible hosing was to be routed to the gripper pistons after stowage system construction and installation on the prototype ACM to avoid interference with LCS systems.



*Figure 5.15 Trolley Lower Arm Subassembly*

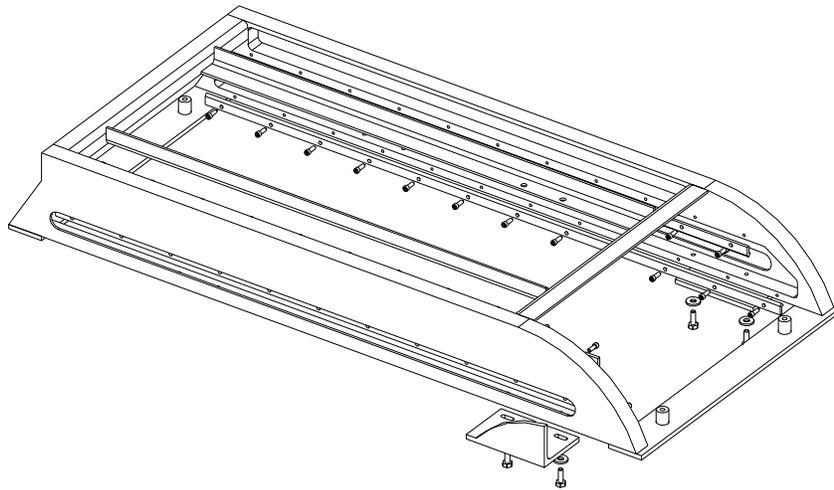
The cart subassembly provided the pivot location for the gripper assembly and the lower arm subassembly and the interfaces to the track and drive assemblies. This subassembly consisted of three v-wheels sandwiched between two mounting plates with shoulder bolts functioning as axles (see Figure 5.16). Proper selection of the v-wheel components required analysis of the cart subassembly for the same worst case condition. After solution of the forces, the v-wheels were sized accordingly. The thicker mounting plate contained the welded pivot shaft and mounting holes for the shoulder bolts while the thinner mounting plate contained the through holes for the bolts and a threaded spring plunger. This component was a commercial tooling fixture that supplied a removable connection to the drive assembly chain. During a system failure, the trolley assemblies could be removed from the drive assembly and positioned forward for manual operation.



*Figure 5.16 Trolley Cart Subassembly*

### 5.3.4 Track Assembly Design

As the track assembly evolved from the simple track concept, it was decided to reinforce the assembly to provide mounting locations for all other assemblies. This would allow assembly of the entire stowage system away from the vehicle except for the mounting interfaces, hydraulic plumbing, and electrical connections to the vehicle systems. The track frame was divided into three separate sections that were welded together to form the completed unit (see Figure 5.17). The right and left sections integrated the upper track geometry with the mounting holes for the v-wheel rails and lower track. These sections were machined from steel plate to assure strength and dimensional accuracy. The lower tracks were created from large structural angle with the required path curvature machined into them. The center section of the track frame was a weldment constructed from structural angle spaced to accommodate the drive assembly mounts. The final track assembly produced a compact and robust mounting frame.

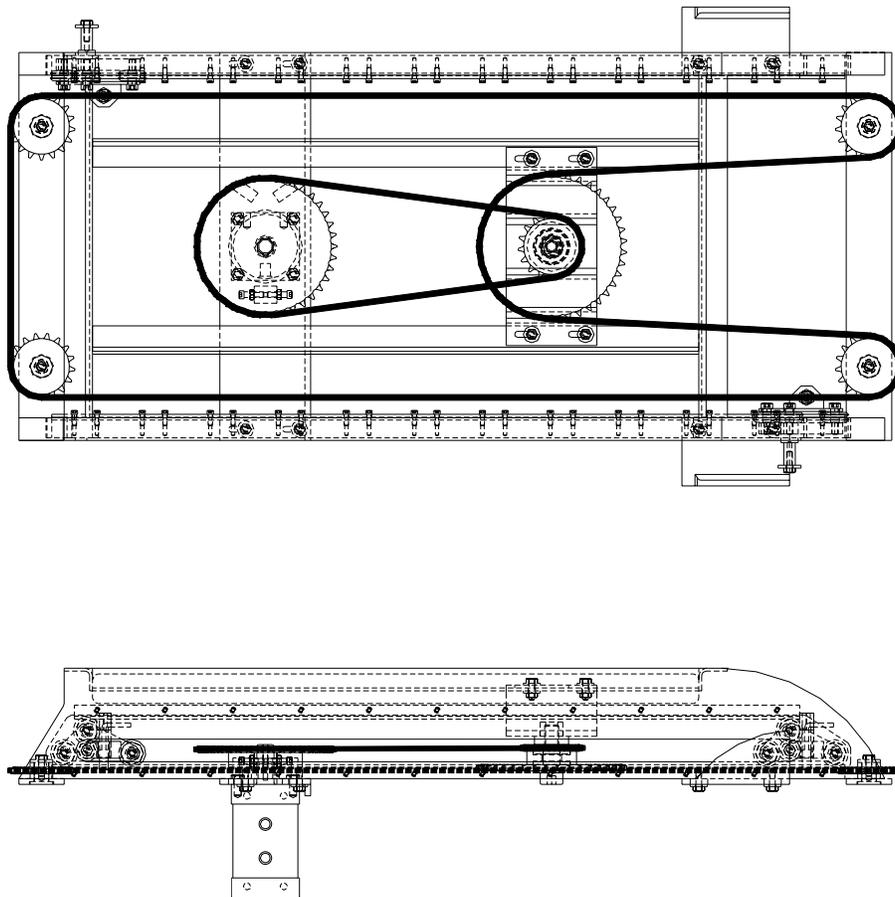


*Figure 5.17 Track Assembly*

### 5.3.5 Drive Assembly Design

The drive assembly required some modification from the original concept to fit into the decreased working envelope provided by the final design of the track assembly. To determine the amount of force required for the rotary actuator, the calculated values from the preceding design were used as a starting point. The gear motors used in the first generation system had large problems with stiction due to the high pressure and low flow rates observed. To alleviate this problem, a vane type of rotary actuator with taper bearings for higher load capacity was selected. This same actuator had been used very successfully on the previous system in different configurations. However, these types of actuators only provided  $270^\circ$  of rotation and gearing would be required to achieve the necessary 0.79 m (2.58 ft) of linear travel. Also, the size of the gears was limited by the frame spacing to accommodate the width of standard traffic cone bases. A multiple chain and gearing system needed to be used to accomplish the amount of gear augmentation required within the necessary envelope. For compatibility with other systems on the prototype ACM, a type forty chain was selected. The vane actuator was mounted to a flat plate that connected to the underside of the track assembly and a single gear sprocket was

attached to the keyed shaft of the actuator. For the gearing augmentation, two different gear sprockets were welded together and machined on both sides to permit insertion of sealed roller bearings. This completed subassembly was then connected by a large shoulder bolt to a mounting plate. The mounting plate then attached to the central section of the track frame. The first chain connected the gear sprocket on the actuator to the smaller of the two welded gear sprockets. The second chain routed around the larger of the welded gear sprockets and four idler sprockets located at the corners of the track assembly to connect to the trolley assemblies. The resulting drive assembly offered the capability of operation at various speeds without stalling (see Figure 5.18).



*Figure 5.18 Drive Assembly*

#### 5.4 Assembly and Testing of the Second Generation Cone Stowage System

In this section the assembly, testing, and modification of the second generation stowage system is discussed. Also, the issue of the stowage system mounting interfaces is addressed. System testing is divided into laboratory and road tests, while modifications are segregated by subsystem.

### 5.4.1 System Fabrication and Assembly

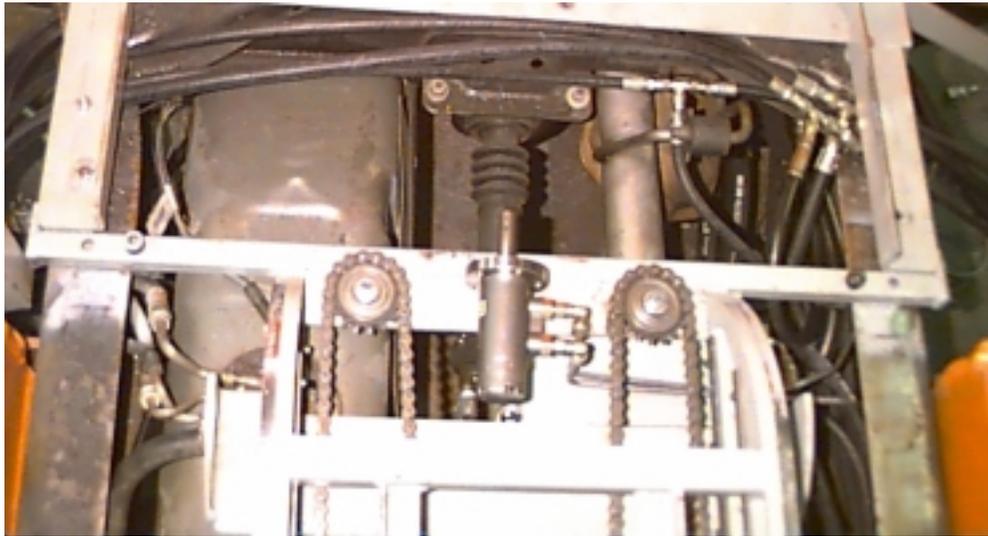
Upon completion of the detailed design phase of the ACM project, the stowage system parts were machined by outside vendors to facilitate rapid completion and consistent quality. As designed, the stowage system gripper, trolley, track, and drive assemblies were individually constructed and then assembled together prior to installation onto the prototype ACM vehicle.

### 5.4.2 System Mounting Interfaces

After fabrication and assembly, the stowage system interfaces to the ACM vehicle's frame rails, hydraulic system, and control system needed completion before operation was possible. These interfaces were divided into the following three areas: mounting interfaces, hydraulic interfaces, and electrical interfaces.

#### 5.4.2.1 Mounting Interfaces

The physical connection of the stowage system onto the vehicle frame rails involved the design and fabrication of front and rear mounting interfaces. A square steel tube was chosen for the front mount support and welded onto the track frame. This tube spanned between the cone body buckets and was drilled on each end to accept two threaded plates that clamped onto the frame rails (see Figure 5.19).



*Figure 5.19 Front Mounting Interface*

For the rear mount, a folded metal tray was welded onto the rear of the stowage track frame and bolted to a vehicle frame cross member (see Figure 5.20). To add additional rigidity and provide a mounting location for the main conveyor front roller, a rectangular tube was welded onto the metal tray with separate right and left mounts that rested on top of the vehicle frame rails. The left mount differed in design from the right mount due to the placement of the gasoline filler hose. The main conveyor front roller then mounted directly onto these right and left mounts and assured correct



*Figure 5.20 Rear Mounting Interface*

The last mounting interface involved the support of the stowage system hydraulic manifold. A set of angles was welded to the underside of the front mount and provided bolt holes for the manifold.

#### 5.4.2.2 Hydraulic Interfaces

The stowage system only required three sets of hydraulic connections that were routed after system installation. One set of hydraulic lines connected the vane actuator in the drive assembly to the hydraulic manifold located under the front right of the stowage system. The other two sets connected the same manifold to the right and left gripper assemblies (see Figure 5.21). Two steel weldments were constructed to provide guides for these hydraulic lines. The left side weldment also protected the vehicle gasoline filler tube from chaffing damage caused by the gripper assembly hydraulic lines. The gripper assembly hydraulic lines were later enclosed under a protective cover for system and worker safety.



*Figure 5.21 Gripper Assembly Hydraulic Interface*

#### 5.4.2.3 Electrical Interfaces

Control of the stowage system would require the addition of two sensor systems. Trolley assembly positioning required that two magnetic reed switches be used to determine which

assembly was located forward on the track system. The magnetic part of the sensor was placed on a welded extension connected to the inside mount plate of the cart subassembly. The wired part of the sensor was located on an outrigger welded to the front of the track assembly.

The second required sensor system was the photoeye arrays that are necessary to position the cones on the main conveyor for deployment and retrieval by the gripper assembly. Sheet metal boxes were constructed and then welded to form enclosures for the right and left side arrays. Each enclosure had a slot machined on the front for mounting of the two photoeyes per array. Cone guides were attached to the tops of both enclosures. A center cone guide was also constructed that contained the reflective tape strip required for photoeye operation (see Figure 5.22).



*Figure 5.22 Left Photoeye Array Enclosure*

### 5.4.3 System Testing

After installation of the stowage system onto the integrated prototype ACM was completed, extensive system testing was conducted using a two phase methodology. Phase 1 testing consisted completely of manual lab operation of the stowage system to discover problems and prove out the design. As testing progressed, computer control of the stowage system was slowly implemented after successful manual testing. Phase 2 testing consisted only of strenuous automated road testing at different locations and under various conditions. Modifications were made to subsystems as required to best satisfy the original design parameters and produce a functional prototype.

#### 5.4.3.1 Manual Testing

Lab testing of the stowage system in manual mode demonstrated that the first gripper shoe design did not provide sufficient retention capability to maintain cone engagement during transition for either deployment or retrieval operations (see Figure 5.23). After four different iterations of the gripper assembly, a working design was developed.



*Figure 5.23 Loss of Cone Retention by Gripper Assembly*

Although not seriously detrimental to retrieval system performance, it was observed that engagement of the gripper shoes was causing excessive cone deformation (see Figure 5.24). This deformation would make cone spacing on the main conveyor inconsistent during retrieval and would cause two cones to be engaged by the gripper system and removed from main stack during deployment. While the stowage system had enough torque to successfully transfer the double load, these cones would jam the LCS. A quick and simple solution to this problem was the implementation of a passive cone separator mounted above the main conveyor front roller. Following several successful weeks of manual testing, control of the stowage system was transferred to the microprocessor and testing continued



*Figure 5.24 Cone Deformation by the Gripper Shoes*

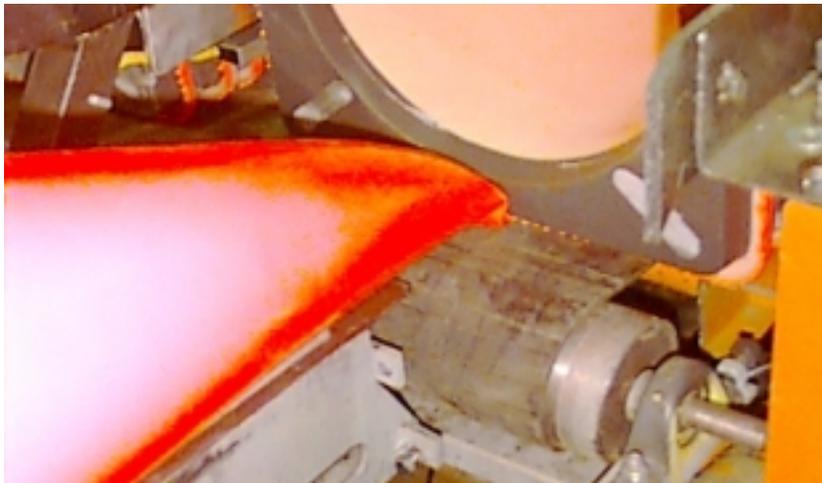
### 5.4.3.2 Automated Testing

Extensive debugging of the control software was required prior to the onset of automated testing. After exhaustive laboratory and road testing, three additional problems with the stowage system required immediate correction. The first problem was very serious and involved the repeated failure of the upper v-wheel component in the left side cart subassembly during cone retrieval (see Figure 5.25). Correction of this problem required careful investigation and a complete redesign of this subassembly.



*Figure 5.25 Upper V-Wheel Failure on Cart Subassembly*

The second problem manifested itself as jammed cones during retrieval. The tip on the retrieved cone would catch on the bottom base of the previous cone in the main conveyor stack and cause the stowage system to stall (see Figure 5.26). Although this problem appeared to be a repetition of the gripper assembly retention problem, the real cause of the problem was discovered to be the large amount of slop or gap between the upper cam follower on the lower arm subassembly and the upper track on the track assembly. Two separate modifications to the gripper and track assemblies were required to correct this situation.



*Figure 5.26 Jammed Cone Tip*

The third problem with automated testing of the stowage system was a slower than expected cycle time. The stowage system operation speed was limited by the transfer mechanism because

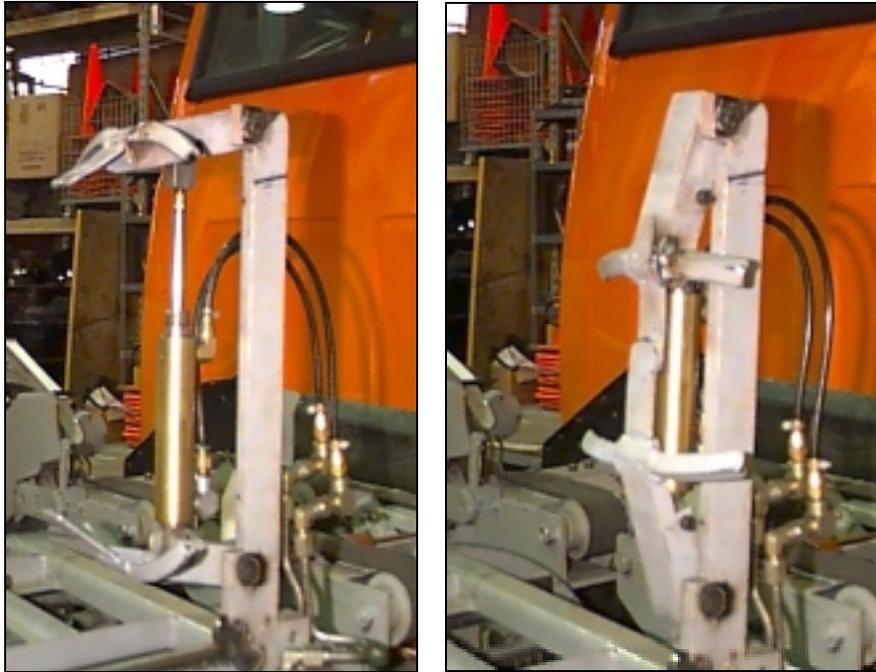
faster speeds made the transfer motion too abrupt to maintain cone retention. To increase cycle times, a high speed circuit was added to the drive assembly.

#### 5.4.4 Gripper Assembly Modifications

Given the poor performance of the initial gripper assembly design, a carefully structured iteration program was implemented to form an optimum solution within the shortest time frame. After trying several different coatings and surfaces for the gripper shoes, the original design of smooth curved plates was discarded. This system relied solely upon the friction developed between the gripper shoe surface and the inside of the cone base. The friction coefficient between the two surfaces varied too much based upon temperature and road conditions for the original design to ever be completely successful. A further complication was that new cones would still be coated with mold release that reduced the coefficient to almost zero.

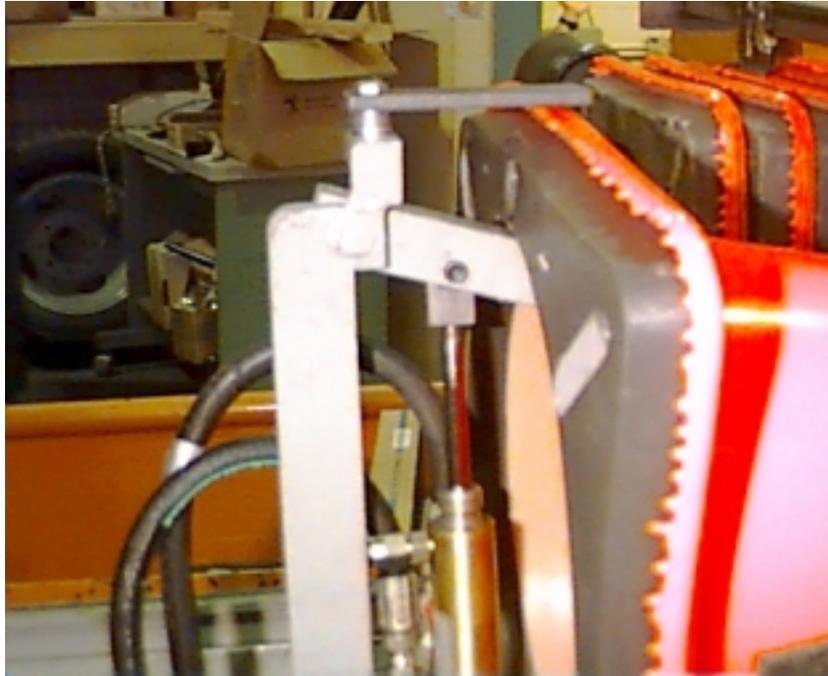
A new concept of selectively expanding the cone base was tried in the first iteration. Small gripper shoes constructed from steel rod were attached to the gripper arms and tested. These shoes were positioned to contact the cone at the interface between the base and conical sections. This interface area was less rigid than the cone base and provided more expansion to firmly encapsulate the gripper shoes and prevent a loss of cone retention. Although the concept was sound, the rods did not provide the correct shape for optimal retention. Testing of various shoe shapes demonstrated that small steel angles curved along the spine of the angle provided the best shape. These new shoes were welded to the gripper arms to become the second design iteration (see Figure 5.27).

Realization that the gripper system would occasionally lose cones, especially damaged ones, prompted the third gripper assembly iteration. Small fingers were welded onto the ends of the gripper arms to provide guides for cones that lost contact with the gripper shoes. As the stowage system continued retrieval motion, these fingers would keep the fallen cone centered and push the cone into the main conveyor stack.



*Figure 5.27 Second and Third Gripper Assembly Iterations*

While previous iterations did dramatically improve performance, the gripper assembly still did not perform at the necessary level that continuous automated operation required. A fourth iteration was designed that included a small plate at the top of the gripper assembly to provide a clamping action between the cone base and the upper gripper shoe (see Figure 5.28). This iteration finally provided the expected level of performance from the gripper assembly.



*Figure 5.28 Fourth Gripper Assembly Iteration*

The fifth and final iteration of the gripper assembly was required to partially alleviate the slop between the upper cam follower and the upper track. A small steel block was welded onto the bottom of the upper arm to provide a mount for a second cam follower (see Figure 5.29). The addition of this second cam follower provided another contact point on the upper track to prevent forward rotation of the cone during retrieval.



*Figure 5.29 Fifth Gripper Assembly Iteration*

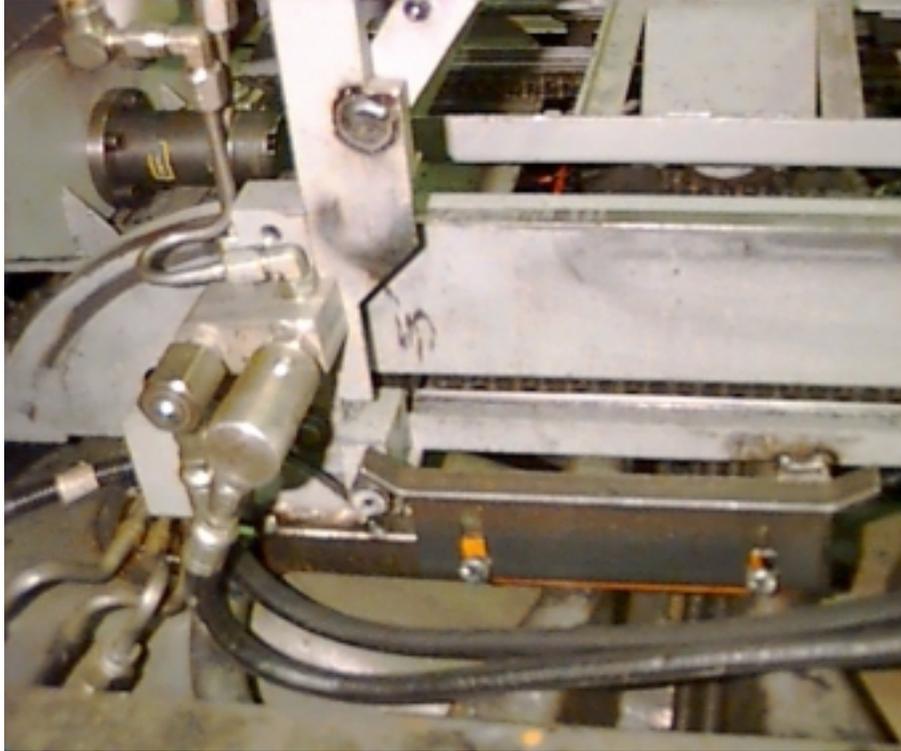
#### 5.4.5 Trolley and Track Assemblies Modifications

As the critical components of the trolley assembly, failure of one of the v-wheels would render the stowage system completely inoperable and would require manual operation to complete deployment or retrieval tasks. Very careful and painstaking investigation of the problem revealed that poor dimensional tolerances in the left and right sections of the track frame would allow uneven loading of the v-wheels to occur. Since most of the loading during the retrieval operation was centered on the upper v-wheel, this was the most likely candidate for failure. Due to time constraints, remanufacture of the track frame was impractical so the cart subassembly was redesigned to accommodate four v-wheels instead of three (see Figure 5.30). The right and left sections of the track frame required some modification to accept the redesigned cart subassembly. While this redesign improved the failure rate of the v-wheels, severe jamming could still cause a failure.



*Figure 5.30 Trolley Assembly Modification*

The second modification to solve the slop problem between the upper cam follower and upper track, required the addition of a guide rail for the lower cam follower on the lower arm subassembly (see Figure 5.31). The left and right guide rails were attached to mounts welded onto the sides of the track frame subassembly. The function of these rails was to force the cone to maintain an exact horizontal position for 0.3 m (0.75 ft) of motion after transition from vertical to horizontal during retrieval. The guide rails were only necessary until the cone tip entered into the base of the preceding cone in the cone stack. This modification, combined with the addition of a second cam follower onto the gripper assembly, provided a satisfactory solution to the cone tip jamming problem.



*Figure 5.31 Track Assembly Modification*

#### 5.4.6 Drive Assembly Modifications

Increased cycle times required the addition of a high speed hydraulic circuit into the manifold feeding the drive assembly actuator. When energized, a solenoid in the stowage system manifold increased the flow rate into the vane actuator to increase rotational speed. However, the high speed circuit could only be engaged after or before the circular segment of the upper track on the frame assembly depending on the operation type. For a retrieval operation the solenoid would not be energized until after the rotary transfer had occurred and would be de-energized prior to placing the cone into the main stack. A roller contact switch and guide track was added to the main drive gear sprocket to facilitate the necessary timing (see Figure 5.32). This modification reduced the average cycle times from 2.4 to just under 2 seconds.



*Figure 5.32 Drive Assembly Modification*

#### 5.4.7 Additional Modifications

The final modification to the stowage system involved the placement of the passive cone separator onto the cone body frame over the front of the main conveyor roller (see Figure 5.33). This system was designed with a set of specially placed hinges that prevented the gripper assembly from deploying two cones at the same time. The hinges were placed high enough for a free cone to pass under. Once the gripper system engaged the inside of the cone base for deployment, the deformation or expansion of the cone would cause the second cone to be stripped off the engaged cone as the system transferred. Although this solution was inelegant and required frequent adjustment, it did solve the problem.



*Figure 5.33 Passive Cone Separator Modification*

## 5.5 Conclusions and Recommendations

### 5.5.1 Conclusions

The development of the second generation stowage system for the ACM prototype produced a functional unit that successfully met most of the general and detailed design requirements. The new system increased cone retention during rotational motion, decreased the required transit forces, simplified the level of mechanism complexity, increased operational cycle times, and dramatically decreased the working envelope. However, the system was still restricted to one cone size and suffered from durability issues in the trolley assembly design. Even given these caveats, the second generation system was a vast improvement in performance from the testbed system.

A further issue was the success of the evolutionary design cycle that started with testing and modifications to the original testbed system and ended with a completed second generation prototype. The major strength of this methodology was the direct implementation of the experiences, data, and information gained from the previous design into producing the next design iteration in an accelerated manner while still producing a superior product. Lessons learned in the original system did not have to be painfully relearned in the next system.

### 5.5.2 Recommendations

Even though the second generation stowage system was successful, improvements need to be made in several areas to assure consistent performance. The key areas that require development are safety, reliability, and performance.

The ever critical issue of worker safety was not completely addressed during this development. Due to the rapid motion and relatively high forces inherent to the mechanism, the stowage system has the potential for injuring a worker by various impact or pinch scenarios. Due to the nature of mechanism required, the only way to completely guarantee worker safety is to place mesh guards around the stowage system while providing a large enough working envelope for cone transitions. This would make it more difficult to access jammed cones or quickly convert to a manual cone laying mode. Further refinement of the design could reduce the number of pinch points and forces such that with reasonable precautions the system can be operated safely without large guards.

System reliability is suspect in the gripper and trolley assemblies. The hydraulic connections to the gripper piston are constantly in motion and proved to be prone to leakage. Using a single acting piston would decrease the number of connections but would require extensive packaging to accommodate the additional piston length caused by the return spring. A design of linear sliding gripper arms and shoes would allow the gripper assembly to easily handle wide variation in cone sizes but would add system complexity.

As the weakest link in the system, the trolley assembly would require careful attention to improve reliability. Using higher capacity v-wheels and tracks and modifying the system to alleviate side loading would provide a large improvement in durability and decrease the failure rate.

A final recommendation is the design and addition of an active cone separator unit to positively control the interface between the stowage system and the main conveyor systems. This unit would prevent the engagement of multiple cones by the gripper assembly and could replace the photoeye array to provide a locating device for cone retrieval and deployment operations.

## CHAPTER 6 CONCLUSIONS AND RECOMMENDATIONS

### 6.1 Need for Automation

The need for mechanizing the cone handling process is well established throughout the world where automobiles are integral to society. Rising standards of living lead to higher standards of worker safety and tasks such as traffic cone handling require serious attempts to remove the human from unnecessary exposure to danger. Caltrans is seeking solutions to this and other road maintenance and construction tasks by supporting the research and development at the AHMCT center.

Handling the traffic cones is physically demanding and very unsafe. It is a difficult task that requires a significant amount of exertion and personnel are subject to repetitive stress injuries. Their exposed location when laying cones is subject to the hazards of the roadway, especially when forcing traffic over in a lane closure transition. Caltrans workers on the road are usually located close to a stream of high speed heavy objects guided by humans that often are more than likely concentrating on anything but the road. Since accidents do occur and the work must be accomplished, developing methods by which tasks can be achieved from within the relative safety of a vehicle is an obvious solution. This requires the application of mechanization and automation technology in order to give personnel a modicum of protection comparable to what the traveling public receives. By effective use of efficient and safe machines, the hazards to crews and the public will be reduced.

### 6.2 The Challenge to Automation of Cone Laying

There have been several significant attempts at developing methods to assist in the cone laying process. By locating the operator on the bed of a truck, the Addco Cone Wheel reduces exposure slightly and adds some mechanized assistance. Although some road work crews have used it, it is cumbersome and the set up needed is not acceptable. The Baliseur and Toyota machines are examples of fully automated machines but neither of them have succeeded in the market and cannot meet needs of Caltrans.

Initially the cone handling process appears to be ideally suited to mechanization. Never the less, it is very difficult to compete with the capabilities of the human cone handler in the current Caltrans cone truck. Cones are often damaged, knocked over, and coated with grit and tar. In the heat they are very gummy and flexible while at colder temperatures they are hard and almost brittle. An automated machine has to be very well designed to deal reliably with these variables. Due to the different cone laying situations and the inevitability of equipment failures, the ability to enable a manual operating mode is considered critical. The closure has to be put out before the work can begin. Dealing with a machine that is mal functioning will not be acceptable to the crew.

A successful machine has to meet a difficult set of requirements. Automated machinery cannot compromise the current cone laying methods. The machine has to conform to all the typical design criteria such as reliability, durability, maintainability, and low cost. The workers

must remain within the confines of the cab to limit their exposure and work site set up of the machine should not be required. The machine should be capable of operation by the driver alone and through a simple operator interface. The ability to retrofit existing Caltrans cone trucks is potentially a cost advantage. The machine has to be very reliable and operate in the dusty, wet, cold and hot conditions of California and the world. It must be capable of operating on an unimproved shoulder and function on typical road surfaces. When traveling to and from the work site the vehicle must be a fully functional truck with a minimum of features that interfere with driving across medians or in construction sites. It has to carry at least eighty regular traffic cones and be compatible with a means to increase this capacity to 200 cones or more. If the system breaks down, it should be possible to deploy the cones manually. The machine should be able to dispense cones from either side of the truck while driving forward. It must retrieve cones while traveling forward or backward and from either side of the truck without any manual set up. It must also be able to retrieve any cone that has been knocked over without manual intervention. It should operate at speeds comparable to manual operations.

These requirements have to be met to interest persons responsible for closing lanes. Anything less and the reliable manual cone laying method will be selected over a machine. A machine that is going to be used for handling cones has to be as simple and easy to use as possible. Cones are used for temporary closures that are often and they have to be put up and taken down quickly. Crews have signs and trailers to place and they have to maneuver the vehicle quickly on and off shoulders. A large bulky piece of equipment will not succeed.

### 6.3 The AHMCT Cone Machine

By working directly with Caltrans crews, the real world requirements were made clear and affected the selection of an ACM concept that clearly would meet the needs of the crews. By creating the working prototype that is described, the intent has been to readily demonstrate its effectiveness to persons who would want to use the machine and those that might commercialize it.

The journey of cones from the main conveyor belt system to the road and back is now achievable by using the totally automated ACM. Driver interactions with the machine are minimal. Four switches are set and the cones are dropped automatically as the vehicle is driven forward. Spacing is automatic and the driver does not attend to the machine except to change spacing and, at the end, turn the machine off. When picking cones up, the driver sets four switches and maneuvers the truck to align the cones with the primary funnel which positions the cones for retrieval. Only once in a while does he command the machine to interact with a tipped over cone in a particular orientation. The machine is simple to operate and is compatible with the Caltrans cone body. If equipment failure occurs, a worker can enter the bucket as before and place the closure manually.

With the single layer of cones, the ACM prototype machine is able to handle the vast majority of maintenance cone closures. Given the success of this concept, the addition of a means to extend the capacity of the machine was a natural development.

## 6.4 Multistack

Motivation for the project originated from the need to increase the cone storage capacity to deal with the less common but very physically demanding long closures that use over 100 cones. Typical in closures where lanes are being repaved, they are often performed at night, which increases the hazards. Numerous requirements and specifications were established to define the project scope and guide multistack development efforts. System concepts were generated and the best design was selected using comparison and trade off analyses. A test unit was fabricated, assembled and tested on the testbed cone body platform.

The multistack system layout is characterized by horizontally oriented cone stocks, which are stored in multiple, vertical layers. The system configuration is consistent with current methods for storing cones on the ACM and manually operated cone trucks. A forklift unit design was chosen to raise and lower the cone stacks within the storage framework. Successful integration and operation of the entire system can be mostly attributed to the simplicity of the forklift design. It effectively handles cone stacks and supports the reconfigured main conveyor. The simple design and operation of the retention hinges and retraction mechanism has also proven to be highly successful, though some redesign may be necessary to increase the hinge stiffness in order to robustly support the cone stacks.

Overall, the multistack system design fulfills the needs and desires of a cone storage system for cone laying vehicles. Preliminary testing and operation of the prototype system successfully demonstrates storage of a tripled cone load. The design is modular and compatible with the existing ACM. It can be easily modified in manufacturing to be lengthened or heightened to add capacity. By conceiving of it as an optional configuration, it is much more likely that the automated cone machine will become marketed. Since many crews will not need the larger vehicle and mechanism for most of their operations, the single layer ACM can be made available for a reduced cost. By using the ACM design for the complete range of capacities, multiple units would be fabricated thereby reducing the single unit costs.

## 6.5 Recommendations

The Automated Cone Machine design is a very viable design and should be used as the basis for a cone machine that can be marketed. Testing and demonstrations of the ACM can continue to be done by AHMCT, Caltrans and others to refine the design. The prototype is being used to identify weaknesses in mechanisms that will greatly assist in development of a robust commercial unit. This development effort is of high value to Caltrans who could manufacture the machine in-house if necessary. It is expected that commercialization of the machine will occur due to the ubiquitous use of the traffic cone and the success of this machine.

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APPENDIX  
SELECTED ASSEMBLY DRAWINGS

Left Front Funnel Assembly, ACM 4100

Left Rear Funnel Assembly, ACM 4200

Left Inner Funnel Assembly, ACM 4300

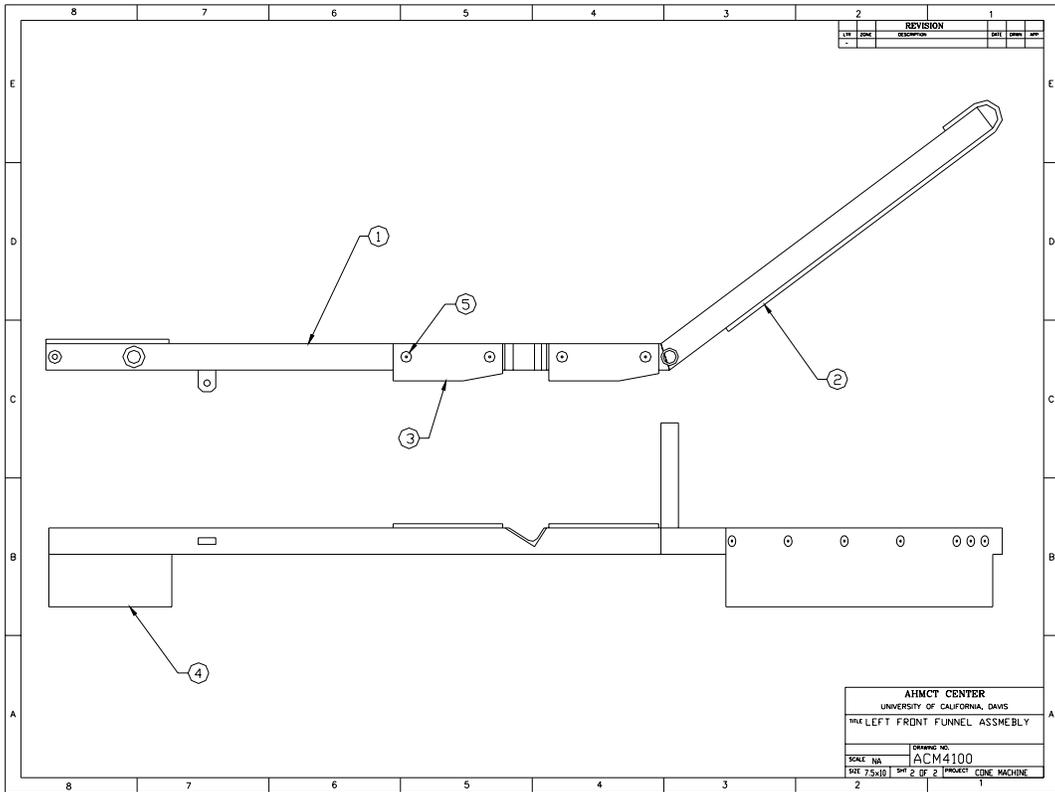
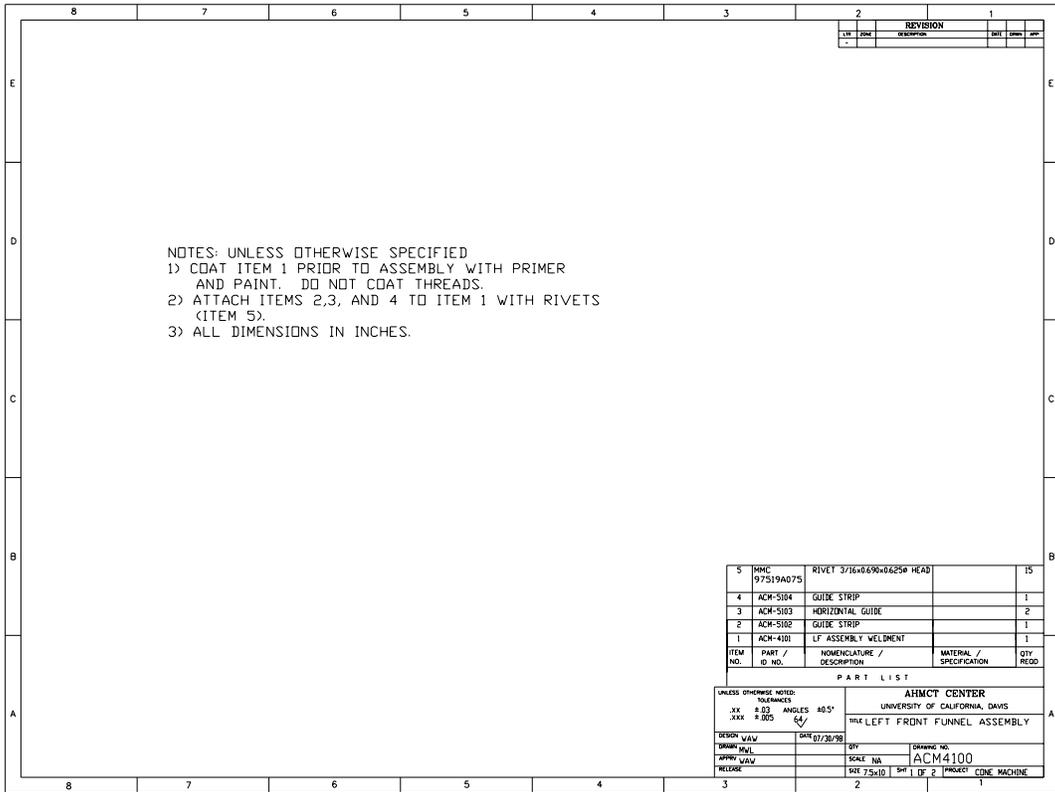
Left Drop Box Assembly, ACM 4400

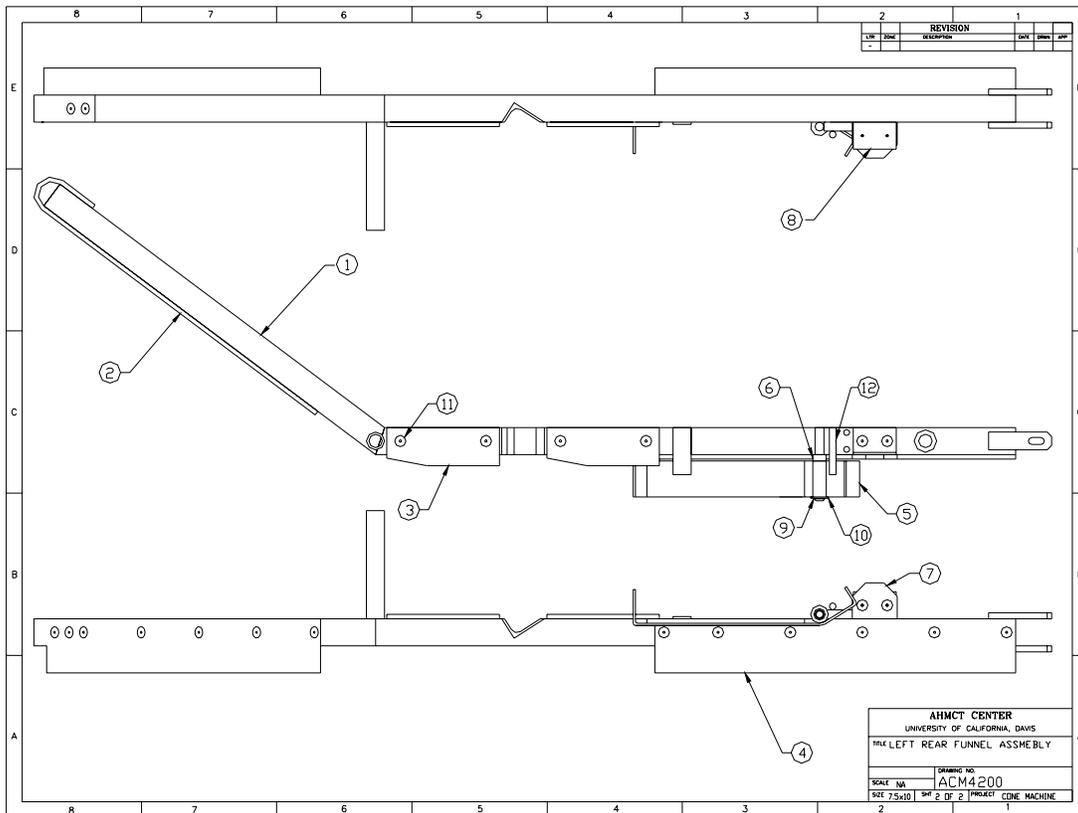
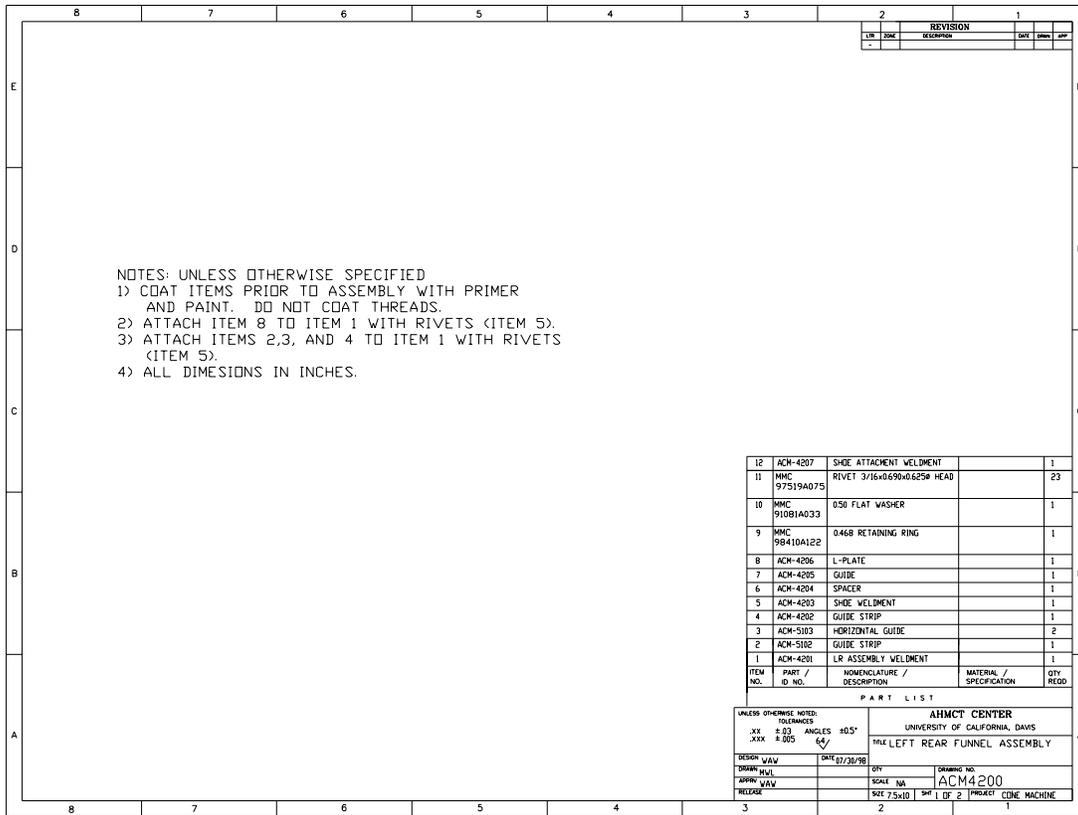
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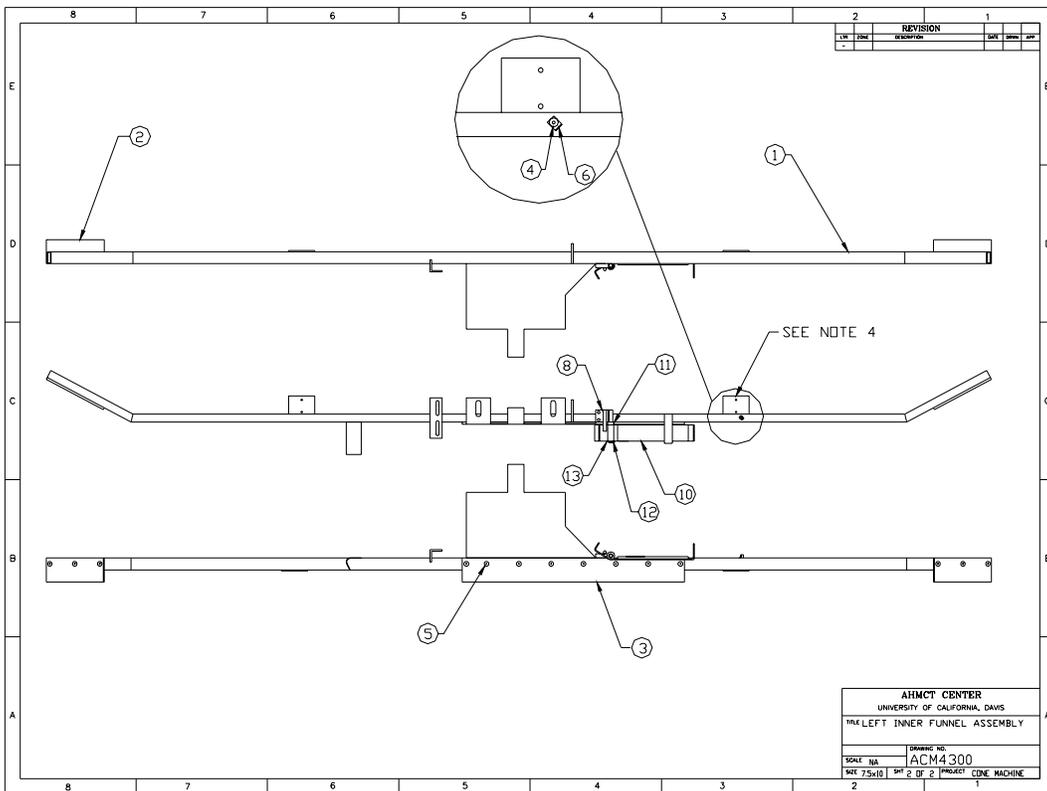
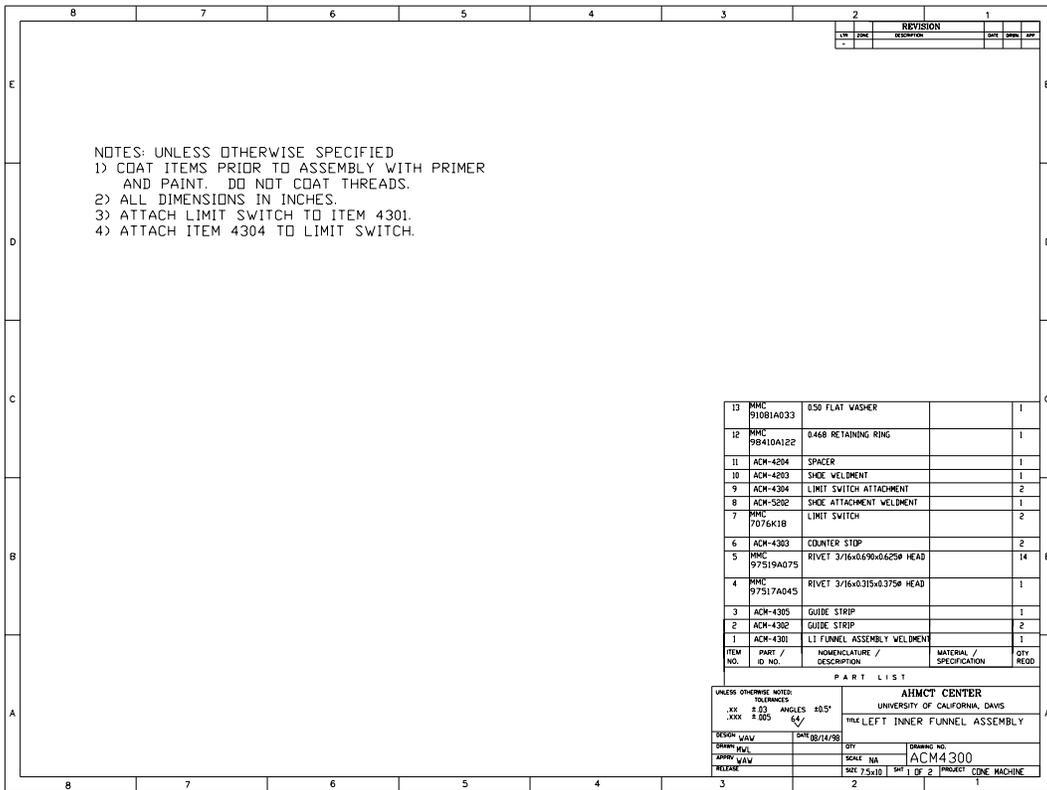
Primary Funnel Assembly, ACM 9000

Stowage System Assembly, BC 10054

Multistack System Assembly, CM-700-00







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28	BVC-B4	200x3185 BEARING	1
27	BVC-W4X	STATIONARY BUSHING	3
26	BVC-W4X	STEEL SEALED WHEELS	3
25	MMC 91251A101	0.3125-18x0.625 SOCKET HEAD CAP SCREW	4
24	MMC 90298A709	80.50x0.625, 0.375-16x0.625 SHOULDER SCREW	1
23	MMC 91251A581	0.3125-18x0.75 SOCKET HEAD CAP SCREW	3
22	MMC 90108A030	0.3125 FLAT WASHER	7
21	MMC 90108A027	#10 FLAT WASHER	11
20	MMC 90108A029	0.25 FLAT WASHER	8
19	MMC 91251A537	0.25-20x0.50 HEX HEAD CAP SCREW	8
18	MMC 92186A083	10-24x0.25 SOCKET HEAD CAP SCREW	11
17	MMC 92198A582	0.3125-18x0.875 HEX HEAD CAP SCREW	7
16	MMC 6072K23	0.375 MALE THD, PLAIN BALL END, W/ WRENCH FLATS	1
15	MMC 6072K46	0.375 FEMALE THD, PLAIN BALL END, W/ WRENCH FLATS	1
14	MMC 6072K42	0.3125 FEMALE THD, PLAIN BALL END, W/ WRENCH FLATS	1
13	MMC 6435K14	SHAFT COLLAR	1
12		0.50x1.70, 1.00x0.50, 0.3125-24	1
11	ACM-440	HORIZ. LINKAGE MOUNT WELDMNT	1
10	ACM-4409	VERTICAL LINKAGE BAR	1
9	ACM-4408	HOUSING	1
8	25T-30-tv	ACTUATOR, REUTER INC. #2-0013	1
7	ACM-4407	DEFLECTOR	1
6	ACM-4405	BEARING MOUNT	1
5	ACM-4405	ACCESS PANEL	1
4	ACM-4404	ACCESS PANEL	1
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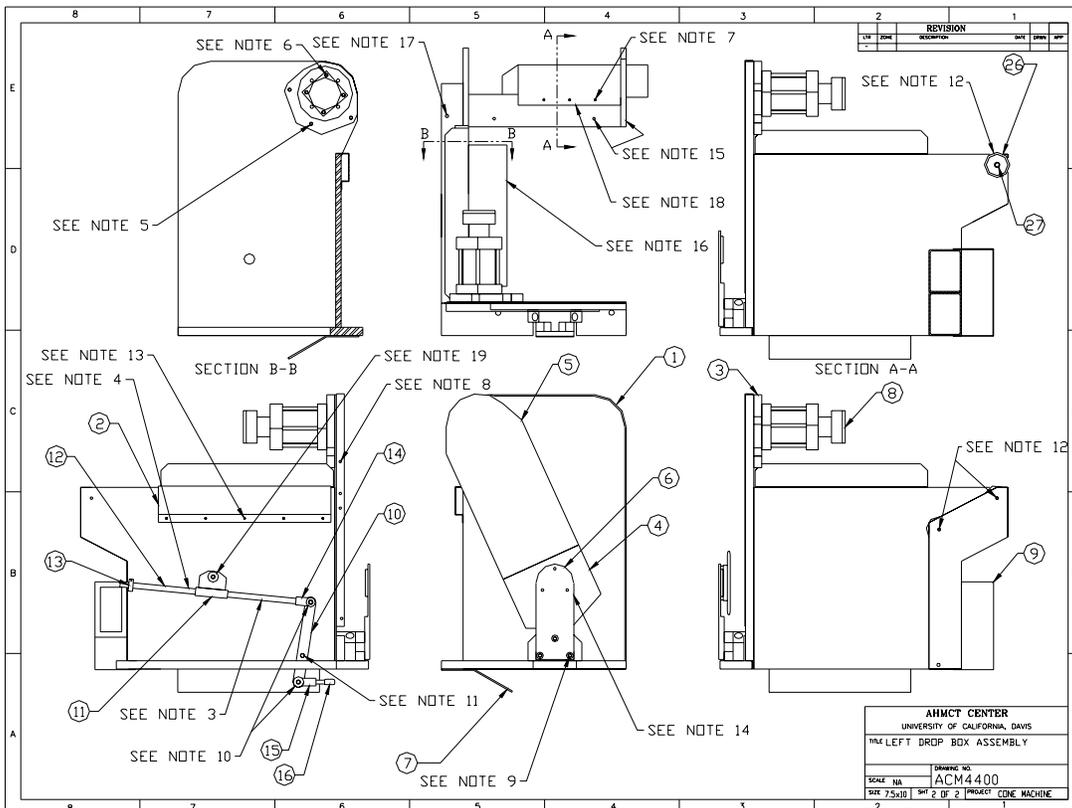
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31	MMC 91259A619	80.375x0.50, 0.3125-18x0.50 SHOULDER SCREW		1	
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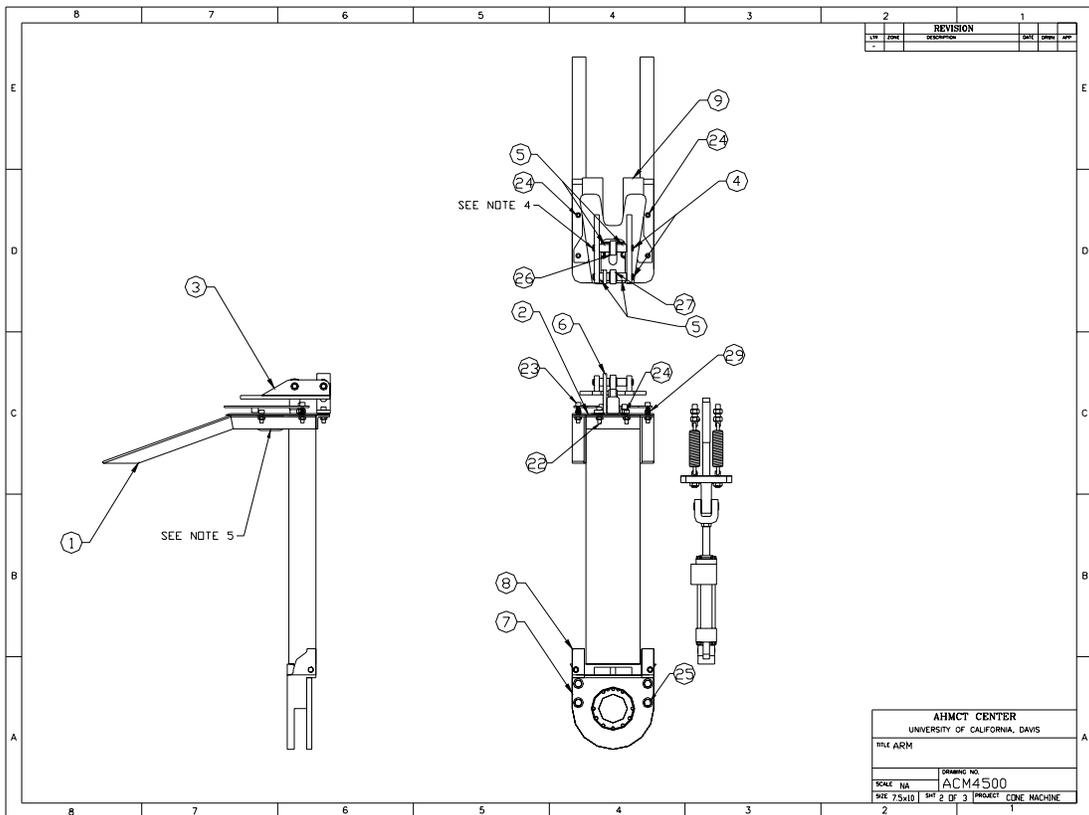
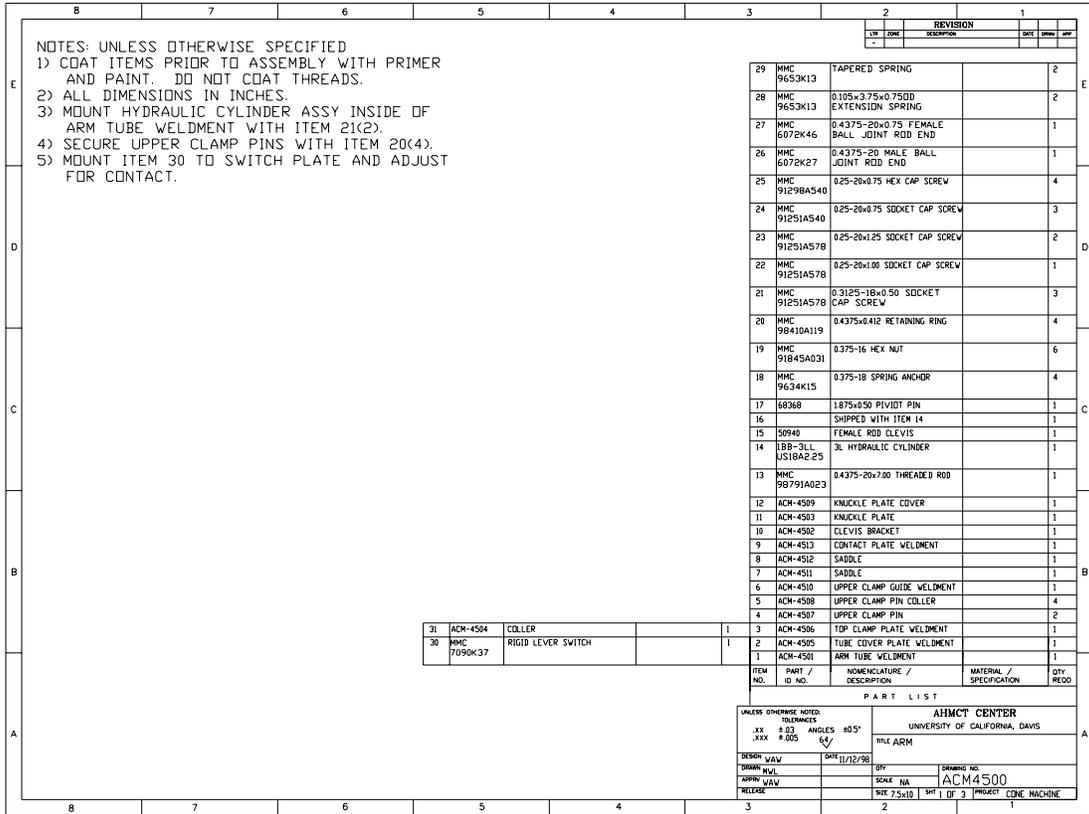
  

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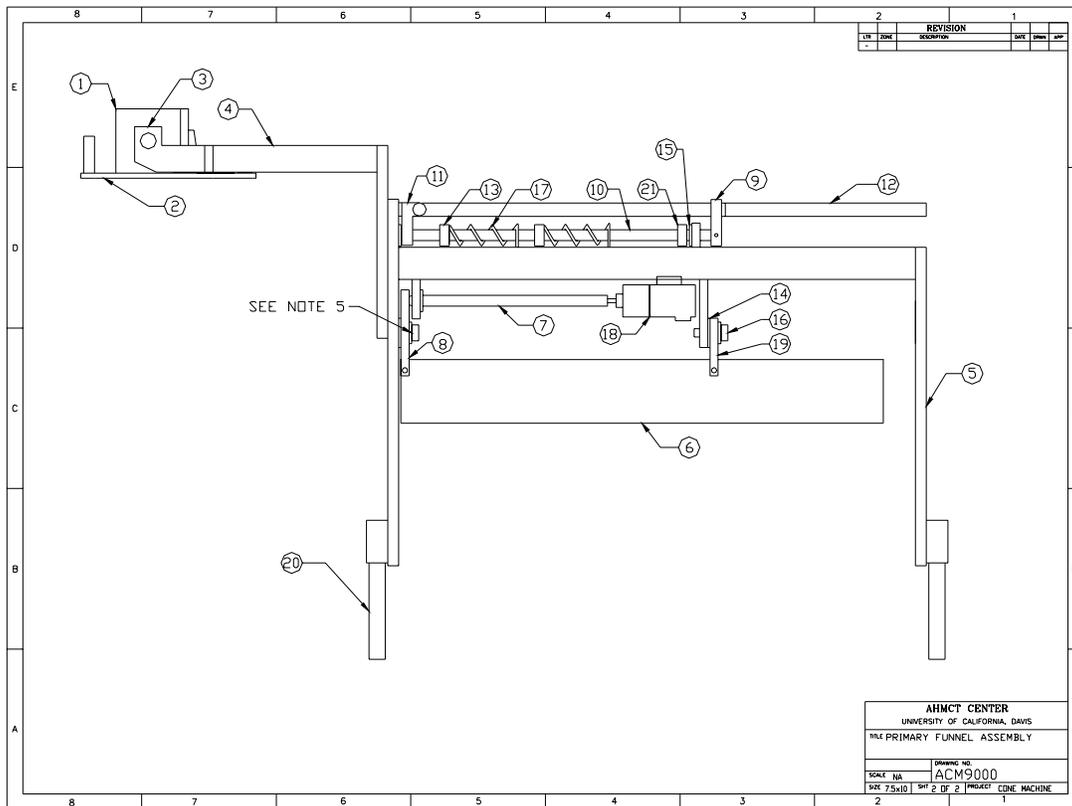
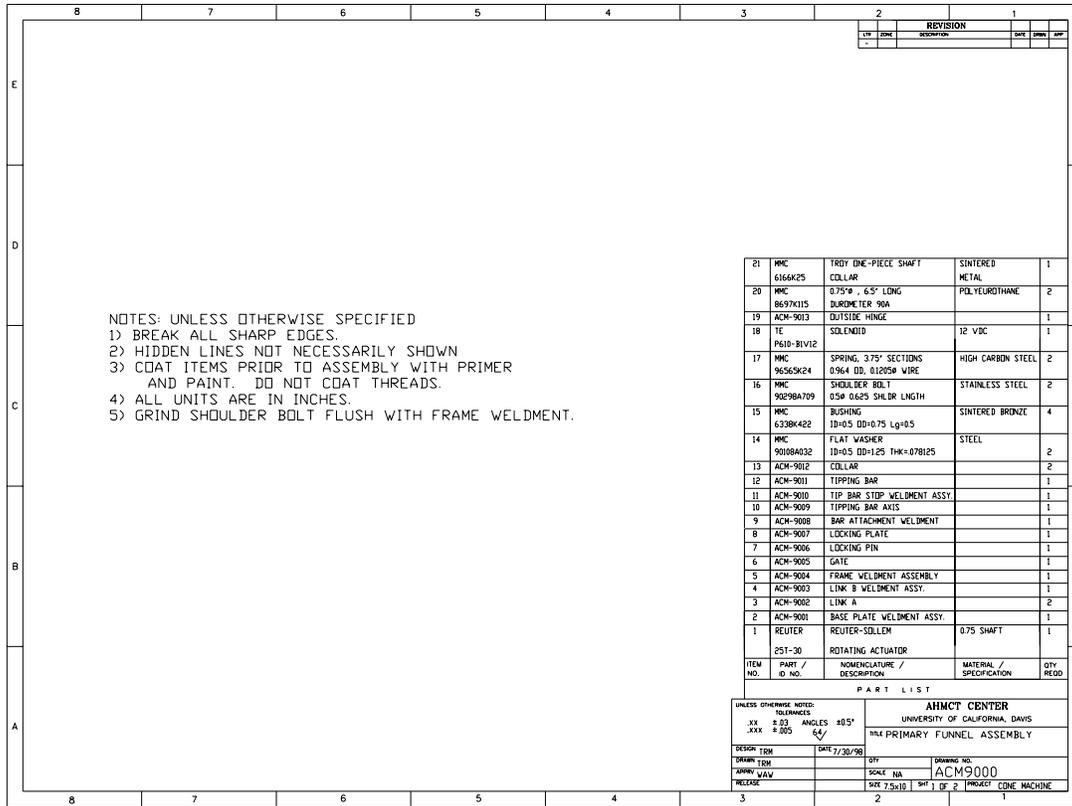
  

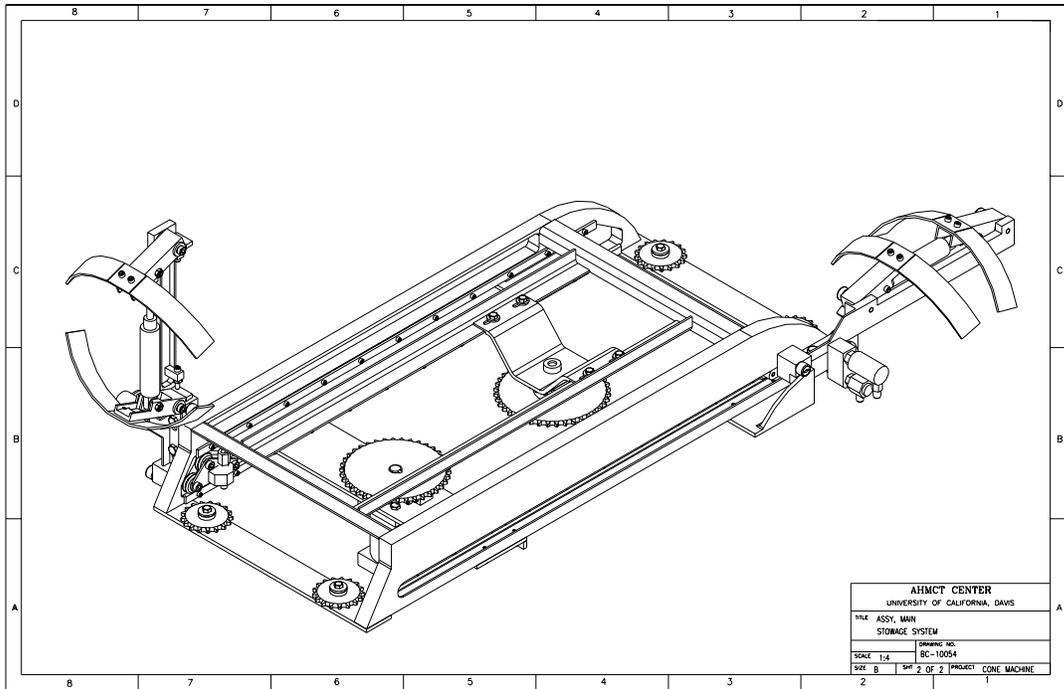
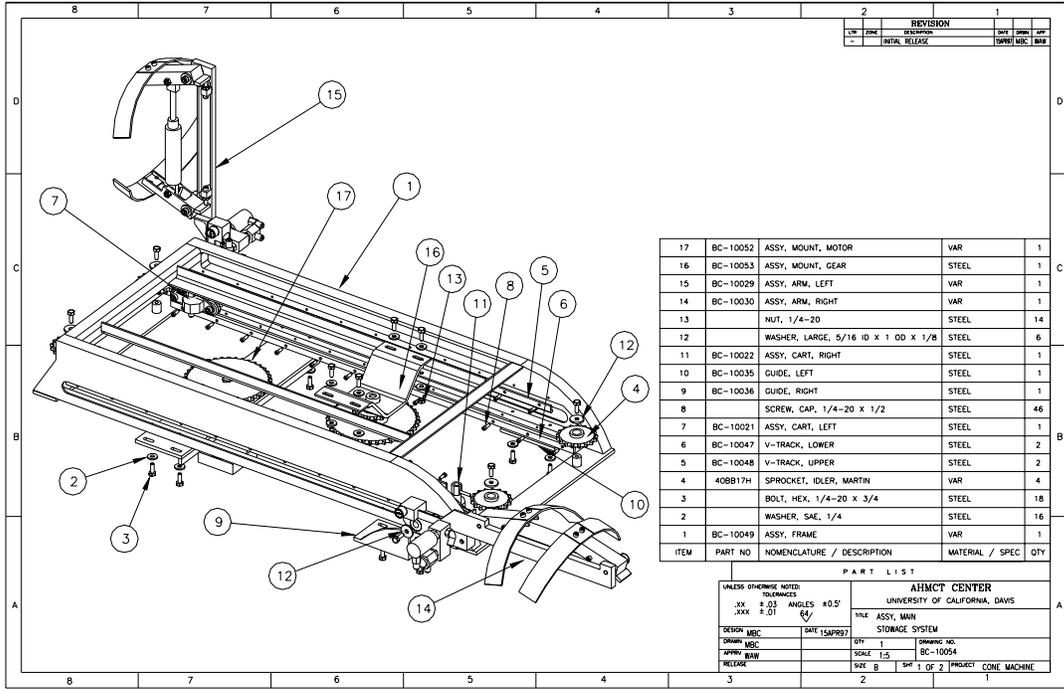
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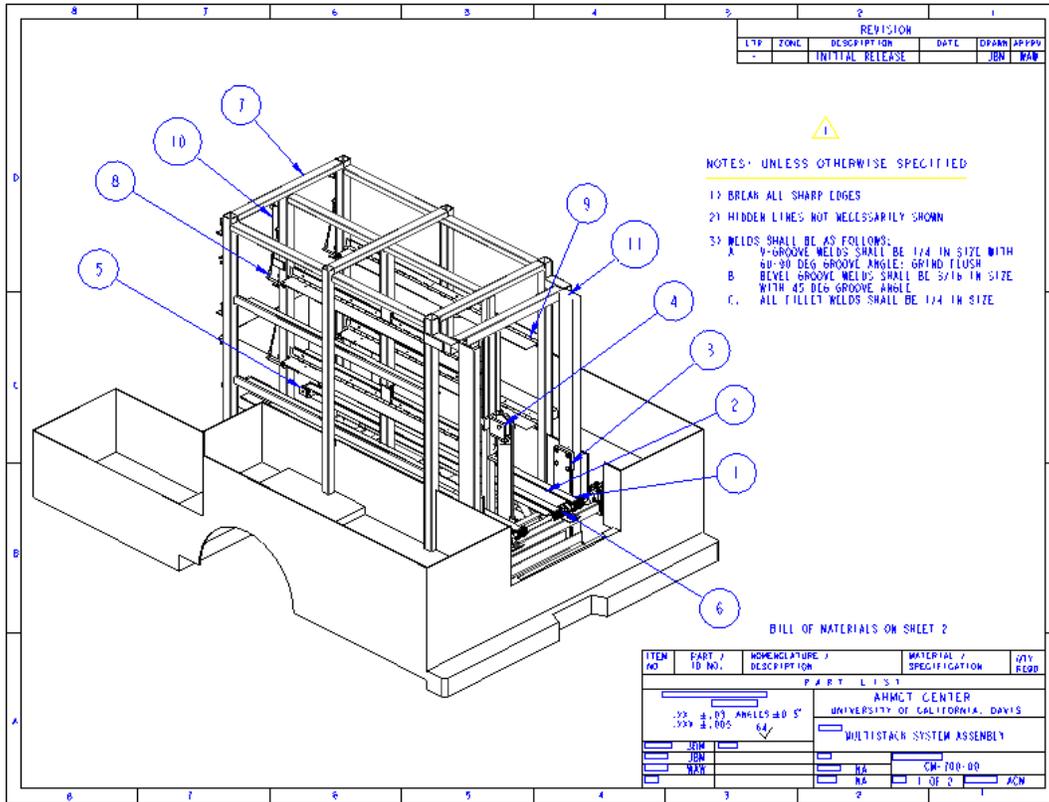












### MULTISTACK SYSTEM BILL OF MATERIALS

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10	CM-122-80	CYLINDER, 1" OD, 6" STRK		1
9	CM-122-50	HINGE, CONTINUOUS	304 SSFL	8
8	CM-122-00	ASSEMBLY, RET. MECHANISM		1
7	CM-121-00	ASSEMBLY, STORAGE STRUCT		1
6	CM-115-00	ASSEMBLY, DRIVE SHAFT		1
5	CM-114-00	ASSEMBLY, FRONT IDLER		2
4	CM-113-00	ASSEMBLY, CYLINDER		1
3	CM-112-00	ASSEMBLY, CARRIAGE		1
2	CM-111-00	ASSEMBLY, FORK		2
1	1700H200	TIMING BELT, GATES		4
ITEM NO.	PART NO.	DESCRIPTION	MATERIAL SPECIFICATION	QTY REQD

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MULTISTACK SYSTEM ASSEMBLY	
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	ACW