Division of Research
& Innovation

Development of Vegetation Cutting Tool Attachments for the Automated Roadway Debris Vacuums

Final Report

Report CA08-0253
December 2008

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California AHMCT Program
University of California at Davis
California Department of Transportation

DEVELOPMENT OF VEGETATION CUTTING TOOL ATTACHMENTS FOR THE AUTOMATED ROADWAY DEBRIS VACUUM

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AHMCT Research Report
UCD-ARR-05-12-31-02

Final Report of Contract
RTA 65A0132

December 31, 2005

This report has been prepared in cooperation with the State of California, Business Transportation and Housing Agency, Department of Transportation and is based on work supported by Contract Number RTA 65A0132 through the Advanced Highway Maintenance and Construction Technology Research Center at the University of California at Davis.

Technical Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
CA08-0253		

4. Title and Subtitle DEVELOPMENT OF VEGETATION CUTTING TOOL ATTACHMENTS FOR THE AUTOMATED ROADWAY DEBRIS VACUUM	5. Report Date December 31, 2005 6. Performing Organization Code AHMCT
7. Author(s): Rob Harker, Jason McPhee, Ryan Bieniek, W. A. White, S. A. Velinsky	8. Performing Organization Report No. UCD-ARR-05-12-31-02
9. Performing Organization Name and Address AHMCT Research Center Dept. of Mechanical & Aeronautical Engineering, UC-Davis Davis, CA 95616-5294	10. Work Unit No. (TRAIS) 11. Contract or Grant RTA-65A0132
12. Sponsoring Agency Name and Address California Department of Transportation Sacramento, CA 95819	13. Type of Report and Period Covered Final Report January 2002 – December 2005 14. Sponsoring Agency Code CALTRANS

15. Supplementary Notes

16. 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 design and analysis of vegetation cutting attachments for the Automated Roadway Debris Vacuum (ARDVAC), a tele-robotic litter removal system. This combination of ARDVAC system and cutting attachments is designed to operate in median divider areas, roadway shoulders, around guardrails, and on some embankments adjacent to roadways. It is capable of removing, by vacuum flow of air, trash, litter, loose vegetation and similar materials. The cutting attachments enable the removal of most vegetation found on the roadside. The machine is controlled from within the safety of the vehicle's cab, requires no on-site set-up, operates with controls of minimum complexity, and is a significant solution to the problem of roadway litter collection and vegetation removal. The development of two general cutting tool attachments and one tumbleweed removal attachment is described.

17. Key Words	18. Distribution Statement
Litter, trash, telerobotic control, roadside maintenance, vacuum collection, articulated nozzle, ditch cleaner, vegetation control	No restrictions. This document is available to the public through the National Technical Information Service, Springfield, Virginia 22161.

20. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	171	

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized
(PF V2.1, 6/30/92)

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 design and analysis of vegetation cutting attachments for the Automated Roadway Debris Vacuum (ARDVAC), a tele-robotic litter removal system. This combination of ARDVAC system and cutting attachments is designed to operate in median divider areas, roadway shoulders, around guardrails, and on some embankments adjacent to roadways. It is capable of removing, by vacuum flow of air, trash, litter, loose vegetation and similar materials. The cutting attachments enable the removal of most vegetation found on the roadside. The machine is controlled from within the safety of the vehicle's cab, requires no on-site set-up, operates with controls of minimum complexity, and is a significant solution to the problem of roadway litter collection and vegetation removal. The development of two general cutting tool attachments and one tumbleweed removal attachment is described.

EXECUTIVE SUMMARY

In 1993 alone, California spent $28 million (5.6% of its annual maintenance budget) to remove 218,000 m^3 (285,000 yd^3) of trash from its highways and freeways (Andres, 1993). Nationally, more than one-half billion American tax dollars were spent on litter removal from roads and public areas in 1989 (Andres, 1993). The California Department of Transportation (Caltrans) is the primary organization responsible for cleaning operations on the state's freeways. Litter clean-up operations reduce the money and man-power available for other maintenance activities. In general, clean up crews must work near high-speed traffic while they manually remove small articles of trash from roadways, one item at a time into garbage bags for subsequent collection.

To improve the safety and effectiveness of this and other highway maintenance operations, the Advanced Highway Maintenance and Construction Technology (AHMCT) Research Center at the University of California at Davis (UC-Davis), ties robotics and automation technology together with current proven maintenance methods to assist crews in their work. The AHMCT Center, a partnership between Caltrans and the University, researches and develops solutions that apply automation to highway maintenance and construction tasks, especially in cases where workers are prone to injury from awkward ergonomics or are exposed to the hazards of traffic.

This report describes the AHMCT Research Center design and analysis of vegetation cutting attachments for the Automated Roadway Debris Vacuum (ARDVAC), a tele-robotic litter removal system. This combination of ARDVAC system and cutting attachments is designed to operate in median divider areas, roadway shoulders, around guardrails, and on some embankments adjacent to roadways. It is capable of removing, by vacuum flow of air, trash, litter, loose vegetation and similar materials. The cutting attachments enable the removal of most vegetation found on the roadside. The machine is controlled from within the safety of the vehicle's cab, requires no on-site set-up, operates with controls of minimum complexity, and is a significant solution to the problem of roadway litter collection and vegetation removal.

The development of two general cutting tool attachments and one tumbleweed removal attachment is described herein. Caltrans has invested in the development by AHMCT of the ARDVAC system and is beginning the process of integrating this now commercially available machine into the fleet. The machines represent a major commitment by the state of California to the goals of improved worker safety and effectiveness and the advancement of the application of technology to human efforts. These machines are projected to be used in water quality control efforts and trash collection in hazardous zones. The vegetation cutting attachment tools developed here are expected to increase the usefulness of the ARDVAC, or a similar machine, to Caltrans roadside maintenance operations.

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CHAPTER 1 AUTOMATED ROADWAY DEBRIS VACUUM (ARDVAC)

1.1 ARDVAC Project Summary

The ARDVAC (Automated Roadway Debris Vacuum) is an ongoing project for the Advanced Highway Maintenance and Construction Technology Research Center (AHMCT). Now licensed by Clean Earth, LLC of Birmingham AL, the ARDVAC is a large, dexterous end-effector that attaches to an overhead boom vacuum truck for automated litter and debris removal on state highways. Figure 1.1 shows a picture of the ARDVAC attached to an overhead boom vacuum truck.

Figure 1.1 ARDVAC Attached to Boom Vacuum Truck

When the ARDVAC was first introduced and demonstrated, the California Department of Transportation (Caltrans) expressed some concern over the vehicle's inability to remove unwanted vegetation. Although not originally specified or designed for such an application, the desire to develop some means of enabling the ARDVAC with such a feature was expressed. Subsequently, the design process was started for development of a vegetation processing or cutting fixture to attach to the ARDVAC.

This chapter provides a brief overview of the preliminary research and work that has been done in designing and implementing an attachable cutting fixture for the ARDVAC.

1.2 Concept Development

Richardson [23] begins the overall process for designing a cutting fixture attachable to the existing ARDVAC end-effecter. He follows a good engineering design approach and provides a solid foundation for which future research and development can build on. At the end of the design process Richardson establishes a number of cutting fixture attachment concepts that could possibly be implemented onto the ARDVAC. Of these concepts the string-trimmer proves to be the most promising and is thus recommended for further development with respect to vegetation processing capabilities. In addition, Richardson mentions that possibly integrating multiple string-trimmers or cutting heads, concurrently on the cutting fixture, would be of benefit.

Richardson finalizes his study by offering recommendations on a number of items that should be addressed during future research. These include:

1. Establishing true cutting capability of the generated concepts.
2. Determining the cutting range of the system while establishing the overall power requirements.
3. Material properties of target vegetation.
4. Most effective means by which vegetation is processed.

Richardson then concludes with a few final design recommendations. He suggests the cutting fixture should operate with the same power and control scheme as the ARDVAC (namely hydraulic) and that mechanically the nozzle could require further reinforcement if a cutting fixture is attached. This is due to the added weight of the cutting fixture, which he recommends be kept under 333 N (75 lbf).

1.3 Concept Selection and Preliminary Integration

Latham [15] picks up the design process where Richardson ended and attempts to address the recommendations for future research and further design considerations posed. He conducts some field testing of various cutting heads which results in a bladed trimmer head utilizing custom high-strength steel blades. With the bladed trimmer selected as the cutting head for the cutting fixture, Latham then develops a mechanical assembly to transfer power to the cutting head. Following the design considerations established by Richardson, a hydraulic motor is implemented. In addition, Latham introduces the concept of articulating the cutting heads on the cutting fixture to offer multiple cutting orientations for possibly improving vegetation control on uneven terrain. This concept is presented but no ideas are given for actuating the articulating cutting heads.

Latham implements the idea of multiple cutting heads, a concept posed by Richardson, in the overall cutting fixture assembly. Using CAD he presents a three-cutter array concept with three cutting heads positioned around the base of the ARDVAC nozzle. This concept is shown in Figure 1.2 below.

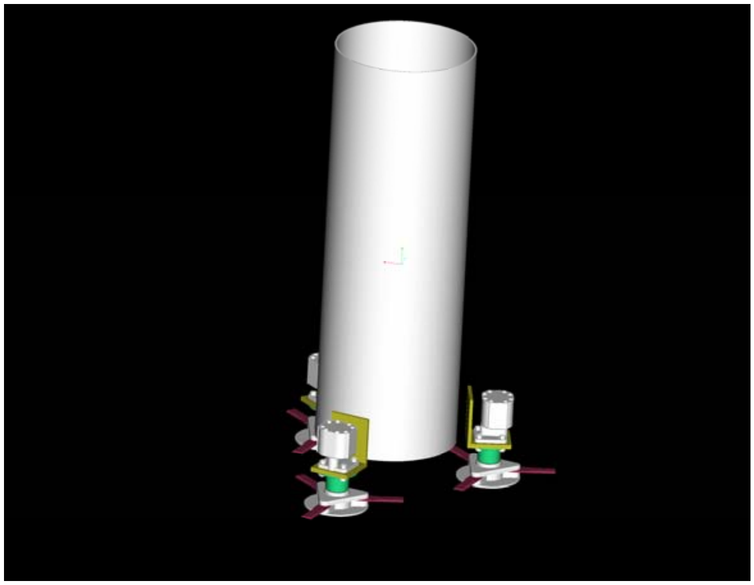


Figure 1.2 Three-Cutter Array Concept Posed by Latham

Richardson desired further work to be conducted in establishing the power requirements for the system and possibly establishing the mechanical properties of the target vegetation. Latham begins to answer these questions by using a laboratory setup for determining the minimum cutting head rotational velocity required for clean shearing of hardwood-doweling (used to simulate woody vegetation). Latham also conducted some impact and shear tests on the same wood-doweling to determine toughness and shear strength properties.

Latham, in taking the first step for integrating the cutting fixture attachment, also, like Richardson, poses some recommendations and future design considerations for the project. These include:

1. Finalizing the overall cutting fixture assembly design and implementing the multiple cutting head concept into the fixture.
2. Evaluation of the safety risks inherent in modifying the stock Grass Gator cutting head by replacing the plastic blades with steel blades.
3. Protective shrouding for moving motor mounts, pistons, and hydraulic lines.
4. Preventing the cutting fixture from striking the ground, which could break a cutting blade resulting in injury.

1.4 Cutting Fixture Design Process

The design process is often an iterative task. Iterating is necessary because new insight is gained as sections of the design process are repeated. Often this insight results in improvements on previous steps and leads to a better design.

The first step in building on Latham's work in finalizing the design of a vegetation processing attachment or cutting fixture, adaptable to the ARDVAC, is to take a step back and look once again at the design objectives and specifications. This ensures the project is on task and meeting the needs of Caltrans.

1.4.1 Meeting with Office of Roadside Maintenance

On September 2, 2003 a meeting took place between AHMCT and Dave Beach and Rick Houston [14], both of the Office of Roadside Maintenance at Caltrans' Headquarters in Sacramento. This meeting was to provide further guidance, for design purposes, in meeting specific maintenance needs with respect to vegetation control and help in iterating in the design process. The following list summarizes the desired capabilities of the cutting fixture:

1. Cut and remove Ice Plant and Ivy (types of landscape groundcover) present on interstate on and off-ramps and overpasses.
2. Removing vegetation under and around guardrails.
3. Removing vegetation and subsequent clippings from V-type ditches and canals, as herbicide use is undesirable in this application because of possible water contamination.
4. Cutting vegetation in a 1.22 m to 1.83 m (4ft to 6 ft) swath next to the right-of-way for fire prevention.
5. Adapting different types of cutting heads to the ARDVAC without having to manually change them.
6. The maximum after-cut height of vegetation should be no more than 15 cm (6 in)
7. Vegetation growing out of a crack (in the road surface) needs to be removed all the way down to the road surface.
8. Eliminate loose and/or rooted problematic tumbleweed.

1.4.1.1 Pictorial Identification of Problematic Areas

To get a better understanding of some of the problematic areas mentioned above, as a result of the meeting, pictures are given below which correspond to the list above.

Figure 1.3 Ice Plant Growing Onto Right-of-Way

Figure 1.3 shows Ivy ground cover that tends to overgrow onto the street and thus requires an edging application to remove it.

Figure 1.4 Undesirable Vegetation Around and Under Guardrail

Figure 1.4 gives a general idea of how vegetation can grow around and under the guardrails on interstates. An articulating nozzle, such as the ARDVAC, with a cutting fixture is a useful tool for getting in places where a conventional mower cannot.

Figure 1.5 V-type Ditch With Irrigation Water

Figure 1.5 shows a specific area where neither a mower nor herbicide is an ideal option for vegetation control and could possibly be addressed with a cutting fixture attached to the ARDVAC.

Figure 1.6 Swath Adjacent to Roadside1.22 to 1.83 m (4 to 6 ft.)

Figure 1.6 shows a typical swath area next to the roadside, found on state roads and is intended to prevent fire from a discarded cigarette or other ignition source. With the vacuum removing the discarded, cut vegetation, the possibility of fire is even less-likely.

Figure 1.7 Undesirable Rooted Tumbleweed

Figure 1.7 shows problematic tumbleweed, that, once dead, will detach from its roots and become mobile under the influence of wind.

1.4.2 Multiple Cutting Fixtures

Meeting with Caltrans provided excellent specifications for establishing a higher level of design for the cutting fixture. Probably the most important idea that came as a result of this meeting is that accommodating all of the desired capabilities in the established list, with a single cutting fixture design, such as a cutting fixture with the bladed trimmer head introduced by Latham, is unlikely. Consequently, multiple cutting fixtures, each of which having a unique vegetation processing capability, are needed.

Having multiple cutting fixtures has its strengths and weaknesses. Strengths could include the possibility of attacking a variety of vegetation problems. Weaknesses could include having to change the cutting fixture in the field, which, in the above list in 1.4.1 (see item 5), is undesirable.

1.4.2.1 Cutting Fixture Concepts

Three cutting fixture concepts surfaced as a result of iterating in the design process. These include:

1. The Rotary Impact Cutting Fixture
2. The Hedge Trimmer Cutting Fixture
3. The Tumbleweed Shredder

The Rotary Impact Cutting Fixture is based on the concept originally devised by Richardson and then further developed by Latham. The concept is unchanged, for the Rotary Impact Cutting Fixture, and the final mechanical design is complete. The recommendations and design considerations posed by Latham are addressed, to some extent, but considerable work is still needed in getting the ARDVAC, with vegetation processing capabilities, on the road and operating. For example, the hydraulic system for powering the fixture still needs to be finalized. In addition, a system for controlling the articulating cutting is still needed.

Figure 1.8 below is a picture of the Hedge Trimmer Cutting Fixture. It shares some of its components with the Rotary Impact Cutting Fixture described in detail in Chapter 2.

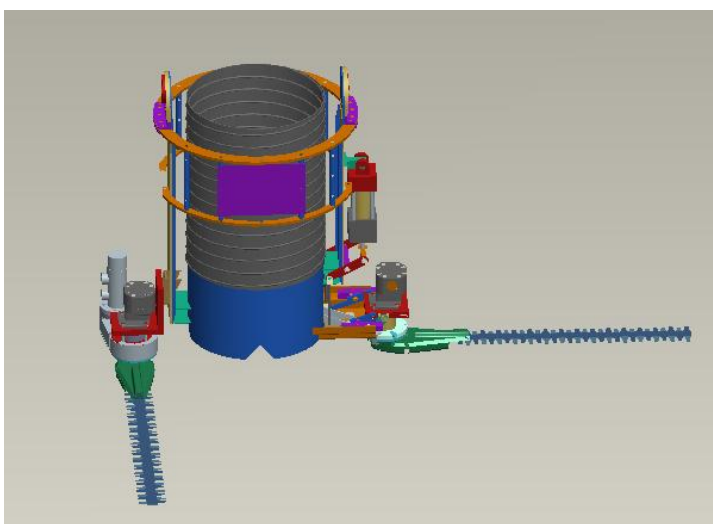


Figure 1.8 Hedge Trimmer Cutting Fixture

Figure 1.9 below shows a picture of the Tumbleweed Shredder cutting fixture. Where the Rotary Impact Cutting Fixture and the Dual Rotary Hedge Trimmer are more adaptable to a variety of vegetation problems, the Tumbleweed Shredder is more specific to item 8 in 1.4.1, which is reducing or eliminating tumbleweed.

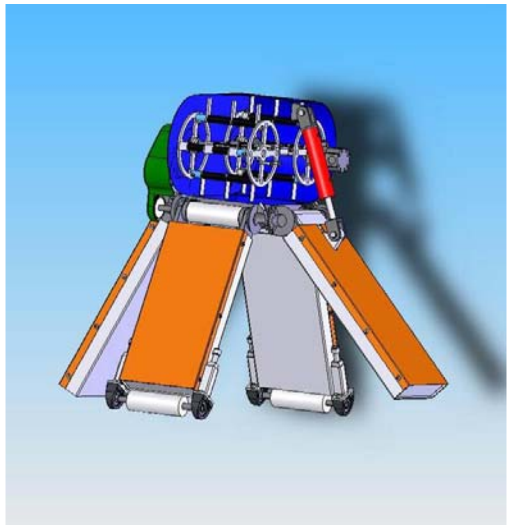


Figure 1.9 Tumbleweed Shredder

1.5 Chapter Summary

This Chapter provided a summary of the current state of the ARDVAC project and the contributions made by prior students in developing cutting fixture attachments for the ARDVAC. The design process has been given attention with respect to reevaluating the customer needs and identifying specific current vegetation control problems which might contribute to the overall final designs.

Latham developed a final concept for a cutting fixture, which developed into the Rotary Impact Cutting Fixture, but as a result of iterations in the design process, two more feasible concepts emerged: the Hedge Trimmer Cutting Fixture, and the Tumbleweed Shredder. Jason McPhee [16] has since developed the Hedge Trimmer Cutting Fixture as the Oscillating Cutter Attachment and Ryan Bieniek [2] has developed the Tumbleweed Processing Attachment.

The focus of this paper is to provide a detailed mechanical design and analysis of three different cutting tools for the ARDVAC. The three different cutting tool approaches, as mentioned above, are the Rotary Impact Cutting Attachment, the Oscillating Cutter Attachment, and the Tumbleweed Processing Attachment. This research builds upon previous ARDVAC research, including research by Richardson and Latham and attempts to address the design recommendations provided by both. The following provides an outline of this report.

Chapter 2 presents a look at vegetation in landscaping, including the different types of vegetation that are regularly encountered during roadside maintenance activities. In addition, the various types of vegetation control technologies are discussed. Also, methods and practices are addressed for designing the roadside to reduce or remove the need for vegetation maintenance.

Chapter 3 presents the original End-Effector System for the ARDVAC. The motion of the tube and nozzle are explained in detail. Extreme nozzle positions are briefly described. In addition, a discussion of the control system is included.

Chapter 4 presents the development and analysis of the Oscillating Cutter Attachment. Caltrans operating environment and demands are explained. Also included is a determination of the optimum cutting design. Major sub-assemblies and linkage assemblies are detailed in CAD. Descriptions of good and bad cutter profiles are included. The final design for the Articulated Oscillating Cutter using an adaptation of an aftermarket hedge trimmer is included.

Chapter 5 presents the detailed mechanical design of the Rotary Impact Cutting Fixture. Major sub-assemblies with CAD images are explained in detail, and relevant justification for the design, including driving design parameters, is given. A kinematical analysis of a linkage is presented. This linkage provides articulation for one of the cutting heads.

Chapter 6 presents a detailed failure mode analysis for the Rotary Impact Cutting Fixture. This analysis identifies four major components as possible modes of failure. Relevant stress analysis is performed and subsequent design changes are made to prevent the identified failure mode.

Chapter 7 performs a detailed dynamic analysis of the spinning cutting head designed by Latham. Kane's Method is used to derive the governing equations of motion and force equations, which are used to predict forces on the blade retaining pin of the cutting body sub-assembly. The input to the blade is found from a detailed analysis of the cutting mechanics involved with impact shearing of single stem vegetation. The results from the analysis lead to a re-designed cutting head with improved performance

Chapter 8 presents a design for the Tumbleweed Processing Attachment. The concept selection process is included. In addition, a detailed description of the design criteria is explained. Detailed CAD drawings of the major sub-assemblies are shown. The final assembly of the unit is addressed.

Chapter 9 will discuss the overall results of this research and how well the design objectives were met including the recommendations postulated by Latham. If not addressed, suggestions are given. Finally, ideas are presented that could result in future research and implementation of the three ARDVAC vegetation maintenance tool attachments.

CHAPTER 2 ALTERNATIVE METHODS OF WEED CONTROL

Alternative methods of weed control, for the purpose of this project, are defined as methods which do not utilize chemical herbicides. As mentioned previously, Caltrans has set a goal of 80% reduction in chemical herbicide usage for 2012, from 1992 levels.

During the course of the ARDVAC project, there has been some mention of different types of weed control methods. However, these methods were not considered in much depth, or as part of an overall strategy of weed control. Also, through the course of this research, it was noted that there are some alternative methods of weed control which were not examined. Therefore, this chapter is an attempt to give a more thorough overview of weed control options potentially available along California's highways. Upon testing of the ARDVAC's capabilities with respect to vegetation control, it is intended that this overview help provide a better understanding of where the ARDVAC project fits in to Caltrans' overall strategy of vegetation control. It is further hoped that this chapter will be helpful to those involved in future projects with AHMCT regarding vegetation control.

2.1 Types of Weed Control Methods

Methods uncovered through the course of this research include weed barriers, weed crushing, steam, vacuum, controlled burns, mulches, mechanical mowing, hand removal, cultivation, infrared radiation, biocontrols, and non-chemical herbicides. While this list may not include every method of weed control ever tried, it is believed to be quite thorough for the purpose of this project.

2.1.1 Weed Barriers

Using weed barriers as a method of weed control essentially involves placing a mat over the area upon which weed suppression is sought. Within this family of weed control products, there are several different material types and essentially two different mechanisms for suppressing weeds. The most common mechanism is simply depriving the weeds growing beneath the mat of sunlight and, to some degree, water. The other mechanism is called Soil Solarization [11]. Essentially, this method seeks to heat the soil beneath the mat to a high enough temperature to kill the weeds. Of the materials, there

Figure 2.1 Caltrans workers applying PolyPavement weed barrier

are spray on mats (Figure 2.1), such as a product called PolyPavement, where the material is literally sprayed over the bare ground, mats of varying thickness and porosity (Figure 2.2), and mats of various transparencies. Weed mats that are porous enough to allow water and air to transfer through are referred to as fiber mats or geotextiles. Geotextiles might be used where there is a desire to suppress weeds while allowing other plants to grow, such as trees. In the case of the freeway environment, there are many section of California where it is desirable to suppress weeds while allowing for existing Oleanders in the median area to survive.

Figure 2.2 Different weed mat products tested on California highways

All three of the products shown in Figure 2.1 and Figure 2.2 were tested in the Santa Cruz Pilot Project. This was a project in Caltrans District 5 designed to test alternative method of weed control while seeking to eliminate chemical herbicides. The project began in June of 2002 with the initial applications of the weed control products and is currently ongoing with respect to monitoring the results of the products' ability to suppress weeds.

One advantage to weed mats over some other types of weed control is that by sealing the mats at the location where guard rails or signs protrude from them, it is possible to eliminate the weed, which would normally sprout up next to these structures. When weeds sprout up in these locations, they are particularly hard to deal with without the use of chemical herbicides.

The basic problem plaguing all of these methods is that debris and dirt can accumulate on top of the mats allowing weeds to sprout up on their surface. In discussing results of the Santa Cruz Pilot Project with District 5 Landscape Specialist Roy Freer [10], it was determined that this problem could be overcome by sweeping the loose dirt and debris off of the mats. So far Caltrans believes this solution to eliminating the weeds growing on top of the mats has been successful, although only time will tell.

Another problem for these mats is their durability. In a dynamic environment such as a freeway, the mats would be subject to being torn or damaged by vehicles or debris, thus allowing weeds to crop up through these gaps in the weed barrier. A further problem inherent in weed mats is exposure damage. Because these mats are exposed to the sun, throughout the year in many cases, the materials may tend to degrade. Also, high winds can result in damage. Research revealed no references to performance in freezing climates. Many of these products have not been on the market long, so product life is hard to gauge with exact certainty. A general search on the web shows many manufacturers for various types of weed mats or geotextiles rate their products lives as anywhere from 1 to 5 years. It should be noted that most of these products were marketed toward garden type landscapes. Products currently being tested by Caltrans have been given life projections of 4 to 6 years in the case of fiber weed mats and 10 to 12 years in the case of Rubber weed mats. When these mats do start to deteriorate, some of them could present a clean-up issue of their own.

A problem with the soil solarization method is that it requires enough water to deeply saturate the ground before application. Due to the lack of irrigation in current median structures, this would most likely be cost prohibitive.

Initial cost is yet another hurdle with weed mats. For fiber weed mats, Caltrans has estimated the price to install this option would be \$25 - \$45/m^2. For rubber weed mats, the price is estimated at \$20 - \$40/m^2. However, it should be noted that if weed mats become more widely used, this price should drop substantially with higher production and increased competition.

Overall, these weed mats are being tested in over 40 states currently. Initial results are promising for some of these products. In determining success or failure of these mats, Freer [10] suggests that it may take as long as 10 years to truly know.

2.1.2 Weed Crushing

Weed crushing is a method of weed control which involves using a dozer or some other mechanism to kill weeds by physically crushing the cell structure of the plant. Although there are many references to weed crushing in weed control literature, it is almost always used in conjunction with other methods of weed control such as burning or mechanical cutting. This is because the mere act of crushing leaves the carcass of the weed behind, many times still attached to its roots to some degree. For this reason, weed crushing alone is not likely to be useful for weed maintenance along California's highways.

2.1.3 Steam

Treating weeds with steam is a method of weed control where the weeds are targeted with a steam blast to raise the water temperature inside of the cells of the weed high enough to kill it. According to UC Davis weed ecologist Joseph DiTomaso, this can be an effective way of killing a weed but the problem lies in making sure that the plant is exposed to the steam long enough to bring it up to a high enough temperature to finish it off. In an open uncontrolled environment like a busy freeway, this is easier said than done, as a plant's appearance is not likely to change significantly immediately after a brief application of steam to visually determine if it has been killed. Further, this method is complicated by the large supply of water and fuel needed to generate the steam.

2.1.4 Vacuum

Through the research done in gathering together alternative forms of weed control, no reference to the use of vacuum by itself, as a technique of weed control, was found. However, in discussing the idea of the ARDVAC as a potential means of vegetation control with UC Davis weed ecologist Joseph DiTomaso, he noted that a strong enough vacuum could have the potential of preventing weeds as a pre-emergent by sucking up weed seed before it could germinate. He suggested that this would have to be done at different times of the year for different types of weeds. Further, the vacuumed weed seed would have to be kept from venting out upon collection and attention to state and local laws would need to be investigated with regard to disposal of large amounts of weed seed.

One potential problem with the use of a high powered vacuum around guardrails, sign posts and medians is the potential conflict with the evermore present weed mats. It is possible that the suction of the vacuum could disturb the seals which bind weed mats to the ground and each other. Or it could destroy them altogether. Further studies would need to be conducted to determine this risk. Until then, great caution should be used when employing vacuum in the presence of weed mats.

2.1.5 Controlled Burns

Although burning can be an extremely effective means of weed control, there are many problems associated with this method when trying to apply it to the dynamic environment of a highway system (Figure 2.3) [26]. For starters, there is the danger associated with starting a fire in a dry environment and keeping it from spreading. Then there is the cost of the manpower and other firefighting resources which must be expended to control the burn. In many cases, burns require the supervision of the California Department of Forestry (CDF). According to DiTomaso, the CDF is responsible for all burns in the Northern part of California. Further, it is impossible to burn only the undesired vegetation while not affecting the desired vegetation when they are growing in close proximity to each other as can often be the case in roadway environments. Another limitation is that burns may only be applied during a restrictive range of temperature, humidity and wind conditions so as to guard against losing control of the fire. Thus, this technique should only be used in special circumstances.

Figure 2.3 Caltrans burning yellow star thistle weeds at Bear Creek

2.1.6 Mulches

Mulches can be an inexpensive method of weed control that is friendly to established desired vegetation. Depending on its purpose, it can be composed of small rocks or bio degradables such as wood chips. However, mulch material can be easily disturbed from its intended original application spot due to drainage, and possibly high winds. Also, errant vehicles may disturb the mulch. Displacement of the mulch can cause gaps in which weeds can sprout. Caltrans Roadside Management Toolbox [24] suggests that mulch be used in conjunction with a geotextile barrier for this reason. If left undisturbed, this combination is projected by Caltrans to have a life cycle of 5 to 8 years.

2.1.7 Mechanical Mowing

Mechanical mowing is a fairly economical way to control vegetation along California's highways. In an interview with Rick Houston, a maintenance manager for Caltrans, he cited that it costs approximately $150 per acre to mow. However, this does nothing to stifle future weed growth and leaves behind the cuttings or "duff" as it is sometimes referred to by Caltrans. This can be a potential fire hazard as it is left to dry. Further, mowers can only operate along wide shoulders and medians, leaving many areas unreachable.

Caltrans uses industrial grade mowing equipment when performing mowing operations (Figure 2.4). There are many different mechanisms available for mowing. These include rotary blades, discs, flails, cords/cables, and cutter bar (similar to a hedge trimmer). These methods of vegetation control were compared, and in some cases tested against each other, for application to the ARDVAC project by Richardson [23]. These results will be discussed in more detail in Chapter 4.

Figure 2.4 Industrial grade mowing machinery

2.1.8 Removal by Hand

One of the oldest methods of weed removal is hand pulling. People are able to get at all sorts of weeds in hard to reach areas that other methods miss. However the work is excessively laborious and also hazardous to the workers who may not have adequate protection from passing traffic. To be effective most of this work needs to be done in the daylight hours when traffic is heavy. Because lane closures are required, traffic tends to back up. Furthermore, this work is costly. The cost of hand removal is at least $500 per acre.

Figure 2.5 Weed crew in District 1

2.1.9 Cultivation

Cultivation, or tilling, is the process of disturbing the soil to aid in weed removal. Generally this can be performed by hand with a hoe. Its mechanism of weed control lies in the breaking of the seed cycle by killing the weeds before they get a chance to seed. In the interview conducted with Roy Freer [10], he recalled that Caltrans frequently employed this method of weed control in the past with the aid of heavy equipment employing large discs to cultivate the soil (Figure 2.6). Currently, Caltrans Maintenance policy forbids this form of cultivation. An interview with Jack Broadbent [4] conveyed that the reason for this ban is that cultivation can contribute to erosion which has ill effects on storm water quality. This restriction was adopted in response to the Clean Water Act.

Figure 2.6 Industrial soil cultivation equipment

2.1.10 Infrared Radiation

The Oregon Department of Transportation (ODOT) [20] has been experimenting recently with a device designed to control vegetation though intense heat via radiation (Figure 2.7).

Figure 2.7 Infrared vegetation control unit

The device is described in a research report completed by ODOT in December 2000 as follows:

"It applies an intense heat of about 1500º F (800º C), generated from a liquid propane fuel. The radiating unit is a steel deck measuring 4 ft wide x 6 ft long (1.22 m x 1.83 m). The width of the treated area is the same as the deck width. The bottom of the deck travels 2 - 4 in. (50-100 mm) above the ground. The distance allows infrared heat to radiate down to the target vegetation with no equipment-to-vegetation contact."

Essentially this kills the intended vegetation in a similar method as previously mentioned with steam. The internal water inside the plant is brought to a boil destroying the cell structure.

What makes this more effective than steam is the ability to more closely control the heat application due to the design of the application and the transfer mechanism being radiation as apposed to convection and conduction with steam.

One major draw back is a slow application speed of approximately 3 km/h (2 mph) over a 1.2 m (4 ft) wide application strip. A further draw back is that this device may cause fires due to the intense heat generation and should only be operated under certain temperature, humidity and wind conditions similar to controlled burns.

2.1.11 Biocontrols

Biocontrols refer to living organisms that attack the vegetation cycle of the plants in some way. Generally this involves consumption of the vegetation in whole or in part such that the reproductive seed cycle is disturbed. Current methods include grazing and biocontrol insects.

2.1.11.1 Grazing

Weed control through grazing of goats has been a traditional method employed throughout history. For obvious safety reasons goats cannot be turned loose on the medians or shoulders of California’s highways. However, they can have there uses for various areas of right of way that are far from traffic. ODOT has done testing with goats to remove noxious weeds and found them to be quite up to the task [30]. (Figure 2.8)

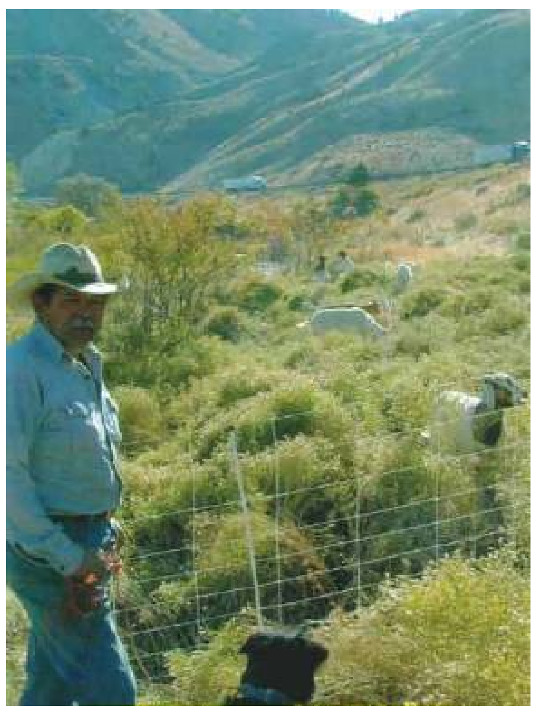


Figure 2.8 Goat graze on I-84 right of way

Apparently it is a very inexpensive proposition, at $300 for the use of 100 goats. Further, their digestive tracks destroy almost all of the weed seed that they consume. One drawback may be that the goats will eat desired vegetation too.

2.1.11.2 Biocontrol Insects

In various areas of the country, experiments are being carried out to test the effectiveness of insects as a method of weed control. In Seattle, Washington, several insects have been used to help control noxious weed populations (Figure 2.9).

Figure 2.9 Cinnabar moth caterpillar (left) and larvae (right) chow down on weeds

Figure 2.10 Aphthona flava flea beetle feeding on leafy spurge

It is important to note that biocontrol insects will not completely eradicate a plant population, but can be effective in keeping the targeted plant population under control. Further, it may take up to 4 to 5 years for results to become noticeable. Thus, this method should only be applied as part of a broader long range strategy of vegetation control. According to Jack Broadbent [4], there are several experiments currently being undertaken by Caltrans with respect to biocontrol insects.

2.1.12 Non-Chemical Herbicides

For the purpose of this paper, non-chemical herbicides are any substances that can be applied to undesired vegetation which are toxic to the target plant, but will have no other unintended ill effects upon the environment. A few such items include corn gluten, vinegar, and coconut oil.

2.1.12.1 Corn Gluten

Corn Gluten is a somewhat experimental new product [22]. Patented by Iowa State University Horticulturist Dr. Nick Christians, it has been demonstrated in some tests as an effective natural way to inhibit weed growth. However, according to Roy Freer [10], Caltrans applied this product to certain areas of roadway during the Santa Cruz Pilot Project, and after the first rain, weeds came up immediately. For now, Caltrans has scrapped this method. Further, this product is the same as the corn meal used in animal feed, and thus may cause unintended problems such as animals and birds coming into the road way to eat the corn gluten. Aside from eating away the weed herbicide, the animals could become a distraction and a danger to motorists. Further, there is the issue of cost. Bioscape.com [3] sells corn gluten at a current bulk price of $31 per 50 lb. bag (Note this is higher than a year ago when the price was $25 per 50 lb. bag in bulk). Further, they cite proper application quantity at 800 lbs. per acre. So not only is the product itself expensive, but costs to haul and apply that much of the herbicide properly would be quite high.

2.1.12.2 Vinegar

Vinegar as a means of weed control came up in my meeting with DiTomaso. According to him, vinegar can be effective but is caustic and so it would be hard on equipment. The kill mechanism is the high acid content.

2.1.12.3 Coconut Oil

Coconut oil also came up in my meeting with DiTomaso. He believed this could be effective but was extremely expensive. Acids are also the kill mechanism in coconut oil.

2.2 Design for Maintenance

One final note on this subject that needs to be covered is that of median and shoulder design with maintenance in mind. In viewing the maintenance problem from the highway design aspect, much of the problem of maintenance can be simplified to a smaller field of variables to which solutions can be much more easily focused. Note that design of highways is well beyond the scope of this paper; however, it is mentioned merely as an opportunity to view the problem of weed control along the highways from a system-wide level. Considerations in highway design which can affect weed control are hardscaping and applying uniform geometries to medians as much as possible. Further, by designing such that a uniform type of vegetation would inhabit the median areas of the highways that eliminated all other weed through competition, many variables to mechanical designs for maintenance could be eliminated.

2.2.1 Hardscapes

Hardscapes are places in medians or shoulders where the designer has chosen to pave with concrete or asphalt. In the past, according to Freer [10], this practice was not accepted in places where guard rails needed to be included because it could change the shear properties of the rails, as the asphalt would tend to be fairly non rigid around the base of the rails. By hardscaping, there is no place for the weeds to grow except when cracks develop or dirt deposits on top (Figure 2.11).

Figure 2.11 Asphalt concrete hardscape

2.2.2 Uniform Geometry in Medians

Instead of having many types of shoulders and medians, one could design the shoulders and medians to be much more uniform throughout the state, so that maintenance efforts could be designed with only a small range of working geometries in mind. Below are a few examples of some of the many types of highway medians throughout the state of California.

Figure 2.12 I-80 east bound through Davis, Ca.

Figure 2.12 shows medians with double guard rails with and without oleanders. Rails are spaced apart by approximately 6 feet.

Figure 2.13 Highway 113 north bound (L) and south bound (R) through Davis, Ca.

Figure 2.13 shows wide medians with and without guard rails. Wide medians without guard rails are often mowed.

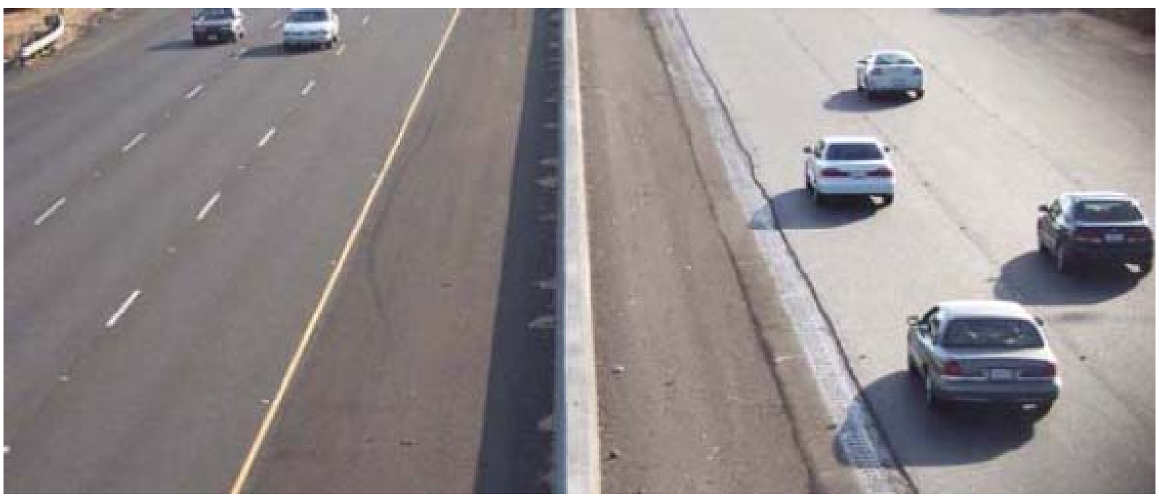


Figure 2.14 I-80 west bound through Davis, Ca.

Figure 2.14 shows a standard cement barrier on asphalt concrete median. This is also an example of hardscaping.

Figure 2.15 I-80 @ UC Davis.

Figure 2.15 shows a picture of poles strung together with cable.

Figure 2.16 I-5 north bound through San Diego, Ca.

Figure 2.16 shows a picture of medians with concrete dividers. One median has a protective fence while another has oleanders in between two concrete dividers similar to the oleanders sandwiched between two guard rails as in Figure 2.12.

Figure 2.17 I-5 north bound through Orange County, Ca.

Figure 2.17 shows a picture of a high concrete barrier on what appears to be a concrete median. This could also be considered a hardscape.

2.2.3 Uniform Vegetation in Medians

Many weeds could be eliminated through competition by employing one species, or several similar species, of plant growth in highway medians which would compete with, and eliminate all other types of weed growth within these areas. By doing this, the problem of weed maintenance will be more tenable, in that methods and machines could be designed with only one, or a very limited number of weeds and weed geometries in mind. Such a plant might be a ground cover which prevents sunlight from reaching competing weeds sprouting beneath it. For a ground cover to be a viable option it would have to be drought tolerant and slow growing so as to limit maintenance. Further, it would have to be fire resistant to some degree and it would have to have a relatively low height at maturity.

Roy Freer [10] indicated that District 1 is currently experimenting with different ground covers. Further, Jack Broadbent mentioned that research on ground covers for highway medians is currently being conducted under UC Davis Assistant Research Soil Scientist Victor Claassen Ph.D.

2.3 Summary

There are many possible methods of vegetation control which show some promise for specific situations. However, it is clear from the information presented in this chapter that there currently exists no one solution for all situations. Further, this research indicates that future consideration of mechanical methods of vegetation control may have to take into account the placement of weed mats around guard rails and sign posts as they seem to have demonstrated some promise so far. Further, it was shown that aside from any mechanical cutting and removal applications, the ARDVAC may have a previously unconsidered benefit in combating weeds. That benefit is the potential to remove weed seed through vacuum action. To measure any specific benefit which may be derived here would require further testing of the ARDVAC.

CHAPTER 3 ORIGINAL END-EFFECTOR SYSTEM

The following chapter describes the original end-effector hose positioning system designed for the ARDVAC project by Porterfield [21]and others at AHMCT. A system background followed by a detailed description of motion capabilities and controls will provide a basis for understanding further designs involving vegetation removal equipment, which will be augmented into the end-effector system.

3.1 System Background

The initial purpose of the ARDVAC project was to find a way to apply vacuum technology to remove trash and debris from California's highway shoulders and medians. In doing so, the need to have highway workers physically exposed to the hazardous environment of the highway could be greatly limited, helping to cut down on highway worker injuries and fatalities. Current commercial vacuum vehicles employed by municipalities in litter and debris removal require workers to manually guide the tip of the vacuum hose from outside the safety of the vehicle (Figure 3.1).

Figure 3.1 Workers guiding vacuum tip to remove leaf piles.

Early on in the ARDVAC project, it was determined that an automated hose positioning system would need to be designed in order to place the vacuum suction tube where it was needed for the purpose of retrieving litter and debris targeted by the driver. Although initially forced to use a vacuum vehicle with a fixed overhead boom as a test platform (Figure 3.2), ultimately, when put into commercial use, the ARDVAC vehicle had an overhead

Figure 3.2 Fixed boom vacuum vehicle.

boom with 3 degrees-of-freedom (Figure 3.3). It would have the ability to raise up and down, the ability to telescope for added reach and the ability to rotate about a vertical axis ± 45°.

Figure 3.3 Industrial vacuum vehicle with articulated boom.

However, even with the broad positioning capabilities of an articulated boom, it would be necessary to have a local hose positioning system located at the end of the vacuum boom which could perform fine adjustment of the hose tip and allow for automated sweeping actions. This resulted in the development of the ARDVAC's end-effector unit (Figure 3.4). The

Figure 3.4 End-effector from 3 different views.

end-effector was developed at AHMCT in 2001, and it has been the foundation for all further work on the ARDVAC system.

3.2 End-Effector System

In describing the end-effector system, both the end-effector motion and the original system controls will be described.

3.2.1 End-Effector System Motion

In order to better understand the hydraulics of the end-effector system, it is first necessary to understand the motion capabilities of the system, as these are the major design considerations for determining the hydraulic equipment and system flows. The original motion of the end-effector system is shown in Figure 3.5. Note, that motions 1, 2 and 3 are all related to the articulation of the boom, while motions 4 through 7 are all specific to end-effector motion.

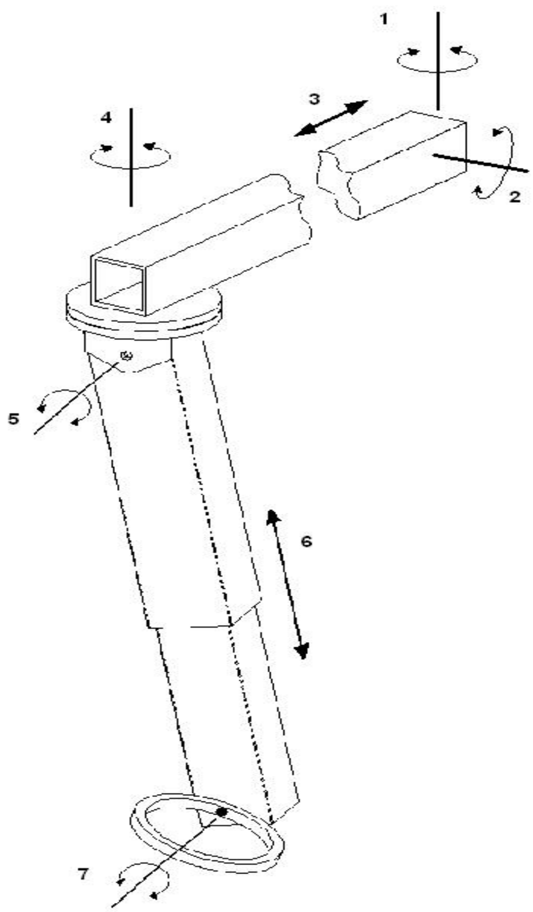


Figure 3.5 Initial end-effector motion

3.2.1.1 Motions 1 and 2

From Figure 3.5, rotations 1 and 2 represent the rotation capabilities of the boom about its mounting point to the vehicle, where axis 1 is the vertical axis and axis 2 is a horizontal axis with respect to ground. Functional specifications for the boom require rotation of the end-effector up to 45° on either side of the vehicle as measured from directly in front of the vehicle.

3.2.1.2 Motion 3

Motion 3 represents the telescoping capability of the boom. Many industrial vacuum vehicles allow for up to 1.8 m (6 ft) of telescoping action. Functional specifications require that the end-effector should be capable of being placed by the boom at 2.4 m (8 ft) to either side of the vehicle.

3.2.1.3 Motion 4

Motion 4 represents rotation of the end-effector about the central axis of the cylindrical tip of the boom to which the end-effector is attached. Originally, this motion was deemed necessary for keeping a constant forward orientation of the end-effector which is desirable for sweeping action. A mechanism was designed to simulate the motion of a four bar parallelogram mechanism (Figure 3.6) attached to the boom which would always keep the sweeping action in the same forward orientation with respect to the vehicle.

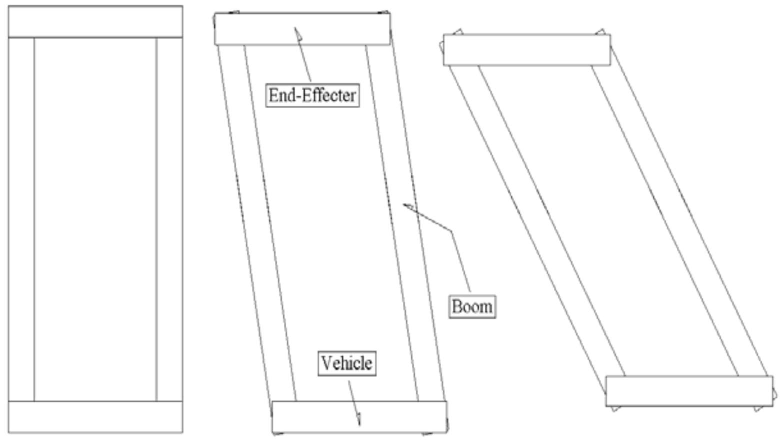


Figure 3.6 Parallelogram maintains constant forward sweeping orientation.

A ball bearing joint (Figure 3.7) was designed to allow for this type of motion, the 4 bar parallelogram mechanism was replaced by a direct acting motor that is independently controlled.

A potential restriction upon motion 4 is the limits imposed by the hydraulic lines and other conduits that pass across the joint. Roughly 360° of total rotation is necessary for positioning cutter heads or aligning the end-effector for the desired sweep orientations. Any rotation beyond this is redundant and would require additional complications to the routing of the lines. Thus the motor rotation for this circuit must have a designed stop to limit this end-effector motion to less than ± 180° from the neutral position.

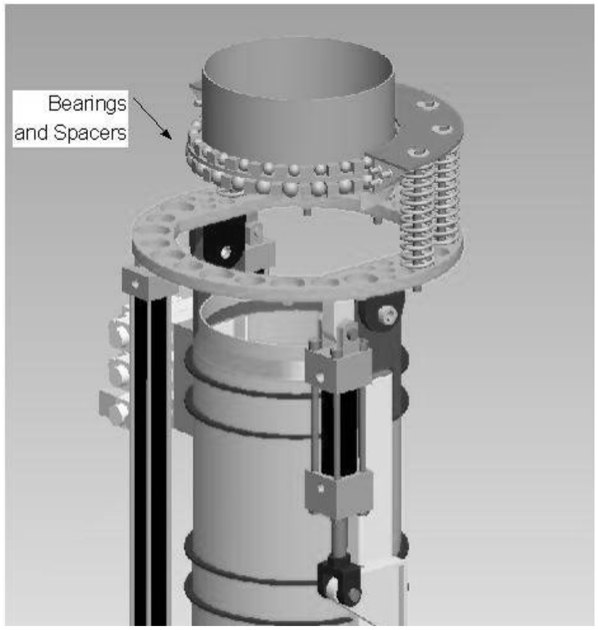


Figure 3.7 Cutaway of ball bearing joint for motion 4

3.2.1.4 Motion 5

Motion 5 represents rotation of the end-effector about axis 5 (Figure 3.5), located just below the Lower Tray Spring Plate (Figure 3.8). This motion is actuated by the Short Cylinder (Figure 3.8). The Short Cylinder stroke of 10 cm (4 in) allows for rotation of approximately 29.4° in one direction when retracting and 30.5° in the opposite direction when extending. This is very close to the 30° of motion in each direction as originally specified. The purpose of this motion is to position the tip of the end-effector when reaching under objects and to drive the sweeping motion of the end-effector.

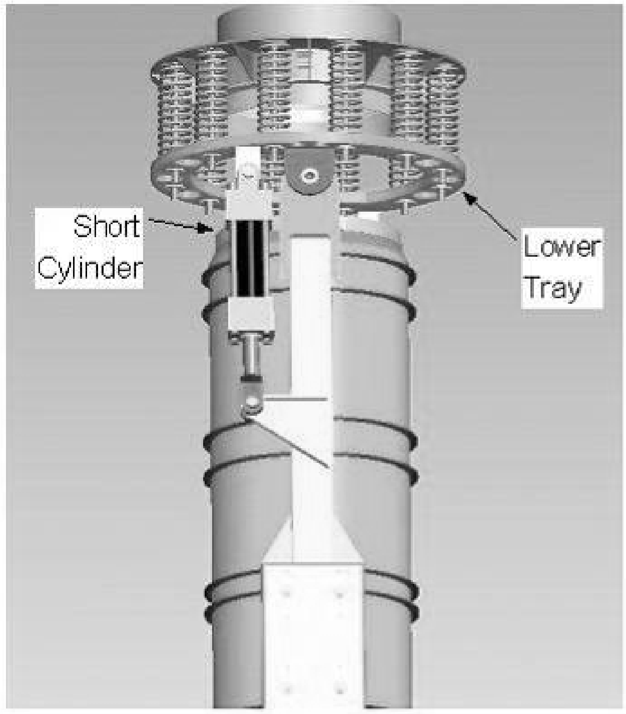


Figure 3.8 CAD model of the upper portion of the end-effector.

3.2.1.4.1 Sweeping Motion

The sweep motion is caused by reciprocating activation of the Short Cylinder. By driving the Short Cylinder back and forth and holding a fixed position on the Long Cylinders (Figure 3.9), the nozzle tip articulates in the opposite direction of the tube. With the Long Cylinders set at fixed lengths and since they are pinned at the Cylinder Pivot Flange, the system is reduced to a four bar mechanism and will exhibit the general motion shown in figure 3.10.

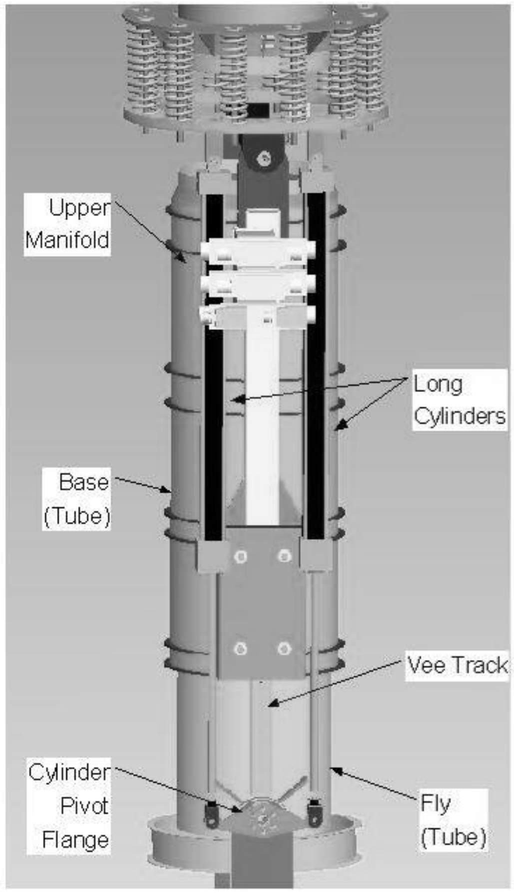


Figure 3.9 CAD end-effector profile.

If the Long Cylinders are set at equal length and since the Cylinder Pivot Flange and Lower Tray mounting points for the cylinders both have the same pin to pin length of .203 m (8 in), the shape formed by these components while in the sweeping mode will always be that of a parallelogram. Thus, opposite sides, by definition must remain parallel to each other. Since the Lower Tray is always roughly parallel to the ground, the Cylinder Pivot Flange and thus the opening of the Nozzle (Figure 3.11) will always be parallel to the ground throughout this motion.

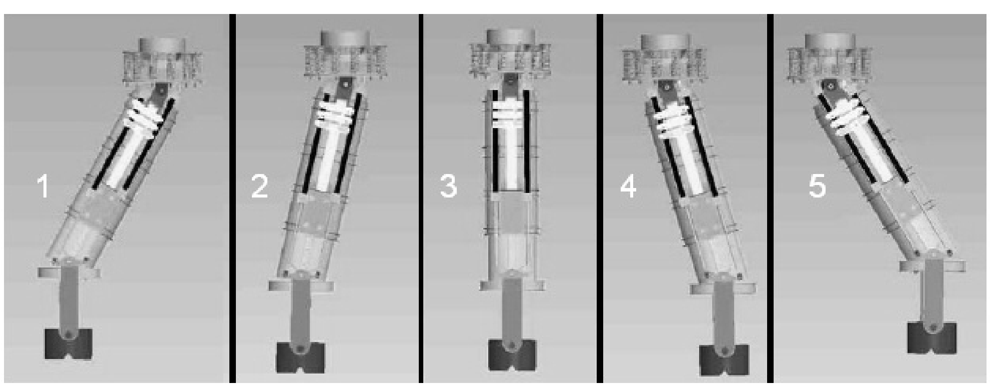


Figure 3.10 Sweep motion of the end-effector from CAD model.

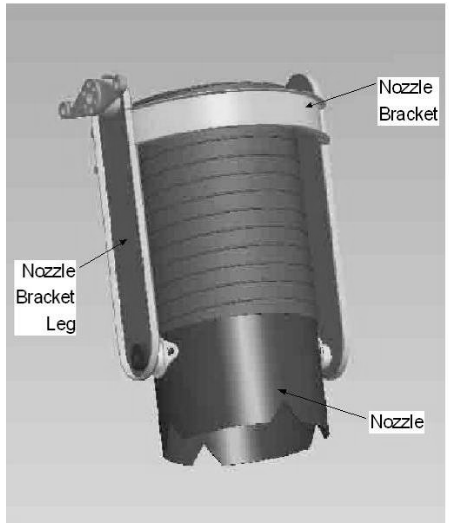


Figure 3.11 Original Nozzle assembly.

3.2.1.5 Motion 6

Motion 6 represents the telescoping action of the end-effector. This motion is affected by the Long Cylinders pushing/pulling in unison on the Pivot Cylinder Flange causing the Fly Tube (Figure 3.9) to extend and retract within the Base Tube (Figure 3.9) on rollers via the Vee Track (Figure 3.9). The Long Cylinders allow for .6 m (24 in) of telescoping range. Due to interferences set in the track by the Channel Seal Adapter, actual telescoping range was calculated to be .57 m (22.35 in) (Figure 3.12). This was further verified by measurements taken from the prototype. In conjunction with boom articulation, this range of motion is specified by Porterfield [21] to allow the Nozzle end to travel through a vertical distance of approximately 1.2 m (48 in), while being able to reach depths below the roadway surface .3 m (12 in). Reaching depths below the surface of the roadway is desirable for maintenance operations on culverts and ditches.

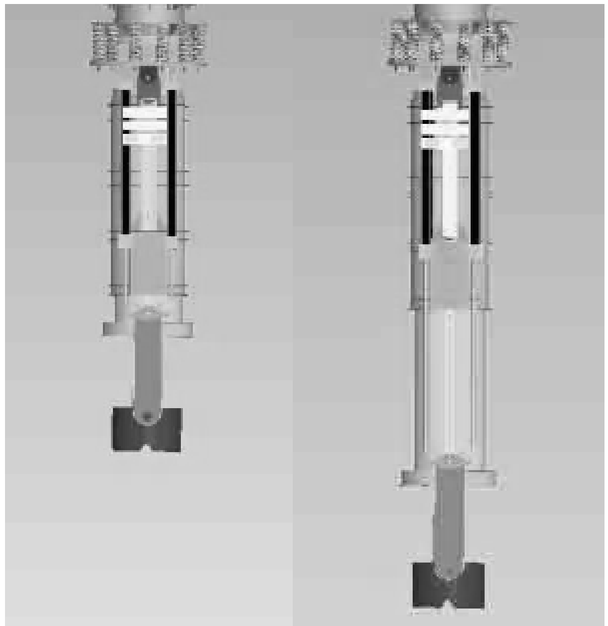


Figure 3.12 Telescoping Fly Tube (motion 6) shown fully retracted (Left) and fully extended (Right).

3.2.1.6 Motion 7

Motion 7 allows for positioning of the Nozzle independent from other motions of the end-effector. This allows the user to articulate the vacuum opening to the desired position once gross positioning has been achieved through boom articulation and other end-effector articulation. Nozzle Tip articulation is achieved by the use of the Long Cylinders acting through the Pivot Cylinder Flange, much in the same way as in the previously mentioned telescoping motion (motion 6). However, in this case one Long Cylinder is pushing on the Cylinder Pivot Flange while the other is pulling. This action causes a moment about the axis of the Cylinder Pivot Flange causing the Nozzle tip to articulate in one direction or the other depending upon which cylinder is doing the pushing or the pulling (Figure 3.13). In the old orientation, the Pivot Cylinder Flange was the limiting factor, causing interference at approximately 40° of Pivot Cylinder Flange Rotation. Further, the orientation of the Nozzle Bracket (ARD-500-01, Figure 3.11), depending upon the extent of telescoping of the Fly Tube, could cause interference at 41.3° of rotation. Redesign of both the Nozzle configuration and the Cylinder Pivot Flange has changed these numbers dramatically. The Cylinder Pivot Flange has been redesigned to allow for 57° of rotation before encountering interference at the rod clevis, and the Nozzle assembly has been redesigned such that its Nozzle rings do not encounter interference at the lip of the Base tube until 52° degrees of Nozzle rotation has occurred.

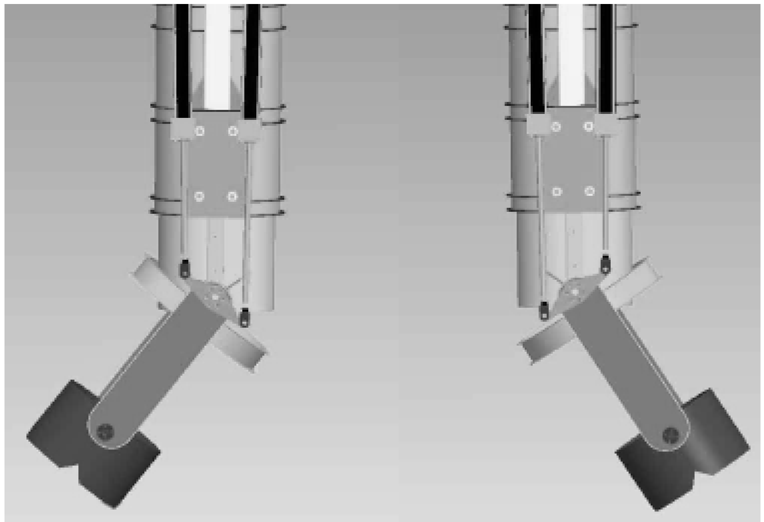


Figure 3.13 Motion 7 due to Long Cylinders applying torque through the Cylinder Pivot Flange.

3.2.2 Extreme Nozzle Positioning

When extreme Nozzle tip angles are desired, it is possible to first position the end-effector to an angle of 30° through motion 5 of the Short Cylinder, and then articulate the nozzle through motion 7, employing the Long Cylinders. However, due to the 4-bar motion, which first occurs due to the Long Cylinders maintaining their initial length through motion 5, clearances between the Upper Manifold (ARD-800-12, Figure 3.9) and the Long Cylinders become tighter. Thus, it is not possible to simply add the limits of the individual motion ranges to achieve the overall range of Nozzle motion if a new instance of interference is to occur. In the original design, this was not an issue, as the limitations previously discussed due to the original Pivot Cylinder Flange and the Nozzle Bracket did not allow for enough motion for this interference to occur. Thus, it was possible to just add the rotations. Thus, we see that the extreme limit was approximately 79° (Figure 3.14). Further, the original Long Cylinders employed on the end-effector had a smaller outer diameter which allowed for more clearance. New Long Cylinders were selected from Vickers to allow for a system operating pressure of 10.34 MPa (1500 psi) to avoid seal leaks which had occurred in the original long cylinders. To solve this interference problem, a mechanical stop will be added to the system to avoid this unintended interference and to further protect against the over flexing of the vacuum hose that must flex with the nozzle.

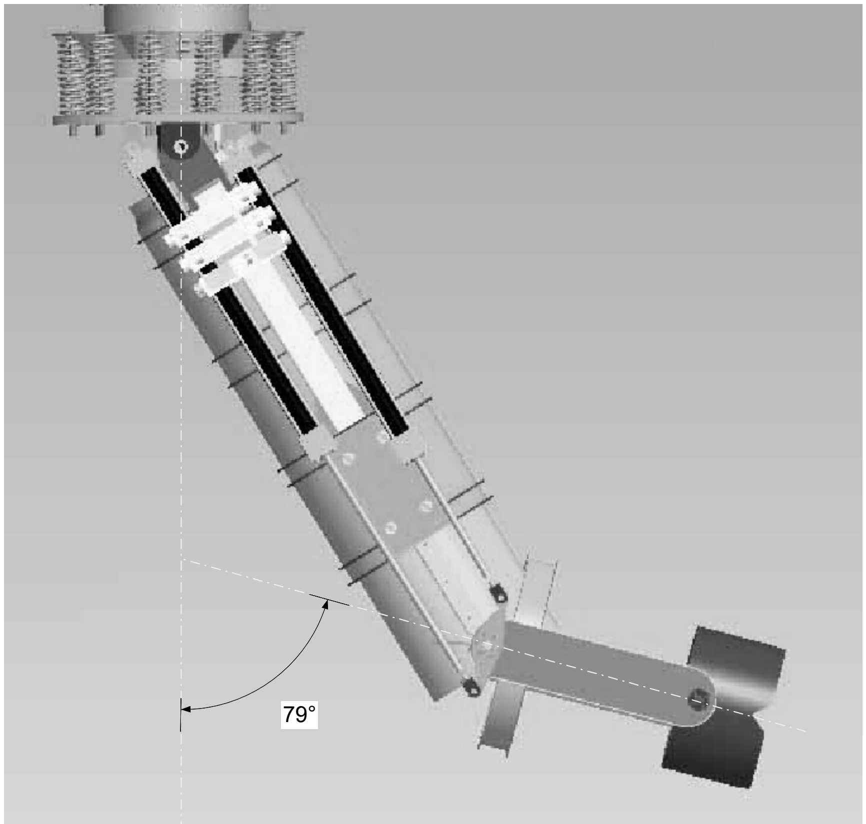


Figure 3.14 Extreme Nozzle positioning.

3.2.3 Original Controls

Originally, system controls were only designed for end-effector motions 5 – 7. These controls were incorporated into a three-axis joystick equipped with five momentary switches (Figure 3.15). Boom motions (motions 1 – 3) were left to be incorporated into this control system at a later date, upon implementation to a specific vehicle, and motion 4 was only to be applied through the four-bar boom mechanism (Figure 3.6), which was never built. From left to right on the joystick controller, the first button allows for motion 6 (Figure 3.12) raising the Nozzle, the second button allows for motion 6 lowering the Nozzle, the third button allows for motion 7 (Figure 3.13) in a clockwise orientation (when viewed from the cab), and the fourth button allows for motion 7 in the counterclockwise orientation (when viewed from the cab). Motion 5 is achieved through agitating the joystick right or left along the X-axis of the joystick. The sweep motion simply occurs when motion 5 is repeated over a frequent uniform interval. Through the use of motion controllers and proportional hydraulic valving, the speed of motion 5 can be controlled by the operator, up to fluid flow limits, through the degree to which the controller is agitated left or right (maximum speeds occurring at maximum perturbations).

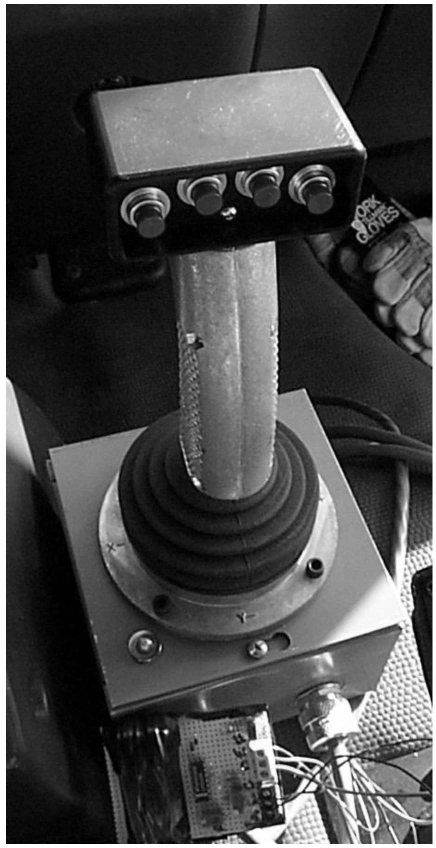


Figure 3.15 Three-axis joystick controlling motions 5 – 7

3.3 Summary

In this chapter, a thorough demonstration of original end-effector motions and capabilities has been illustrated. Further, motion limits and restrictions due to design modifications implemented since the original fabrication of the end-effector have been discussed. It is hoped that this chapter may serve as a guide to a better understanding of design considerations for vegetation removal systems, which have been undertaken to augment the end-effector system.

CHAPTER 4 DESIGN DEVELOPMENT OF OSCILLATING CUTTER ATTACHMENTS FOR USE WITH END-EFFECTOR SYSTEM

This chapter will discuss the design process leading to the selection of an oscillating cutter mechanism to be adapted to the ARDVAC vehicle for vegetation maintenance. A detailed discussion will be presented with respect to the design of the cutter assemblies, their capabilities, and how customer needs are satisfied through application of the design.

4.1 Defining Customer Needs

The driving customer for the ARDVAC project is Caltrans. Caltrans funds AHMCT through its Division of Research and Innovation to find automated, safe and efficient ways to deal with road maintenance and construction tasks. Other customers whose needs should be considered, though not directly tied to the project, include other Departments of Transportation (DOTs), municipalities, and private companies who contract through DOTs and municipalities to do roadside maintenance.

4.1.1 Caltrans Needs

Defining Caltrans' needs with regard to vegetation maintenance equipment is a difficult task in that Caltrans is a very large entity performing vegetation maintenance operations over a vast geographical region with many variations in terrain and climate. What presents a problem in one district may not even be encountered during operations in another district. Therefore, to understand Caltrans' needs as a customer with respect to this project, it is necessary to look at the range of environmental conditions Caltrans is operating under and the operating guidelines that Caltrans has set for itself with respect to vegetation control. Further, it is necessary to interview Caltrans personnel with respect to the problems they are currently facing in terms of roadside maintenance and vegetation control. Through consideration of these factors, a final design was adopted, which addressed the most commonly encountered problems.

4.1.1.1 Caltrans Operating Environment

Caltrans is responsible for maintenance of over 24,000 km (15,000 mi) of roadways and more than 931 km^2 (230,000 acres) of right-of-way. Regional environments for which Caltrans is responsible vary throughout California, from the desert regions of the southern part of the state, to the icy conditions of the Sierras, to the cool breezy coastal regions, to the flat and foothill humid regions of the central valley, to the many forest regions throughout the state. Roadside litter and debris removal functions of the ARDVAC may not differ significantly throughout these differing regions, however vegetation maintenance needs vary drastically as the different regions host a divergent range of plant-life (Figure 4.1). For example, drier, more arid regions may support infestations of yellow star thistle, or tumbleweeds. Coastal and cooler regions may see outbreaks of ground covers such as iceplant or Cape ivy. Warm and humid regions, such as those present in the central valley may see infestations of wild sunflower or combinations of any of the above mentioned pest plants. Colder high elevation climates may see an invasion of Klamathweed, while northern California regions may see infestations of Scotch broom. The above mentioned plants represent only a small sample of the different invasive pest plants in California. For a much more extensive list, see <u>The CalEPPC List: Exotic Pest Plants of Greatest Ecological Concern in California</u> [7].

From Figure 4.1, it is clear that there are a variety of different issues that must be addressed when designing a mechanical cutting system to apply to vegetation control throughout California. One of the issues most clearly present is the geometry of the plant being cut. A tumble weed is much differently shaped than a yellow star thistle, spreading out wide and round from its base, similar to a shrub, whereas the star thistle tends to grow more up than out. Thus, the base stem of the star thistle is much easier to access with cutting tools than the base stem of the tumble weed. Ivy grows very close to the ground, anchoring itself approximately every 15 cm (6 in) making it hard to locate a root to cut. Also, due to the ivies broad leaves and the necessity to cut close to the ground, blade damage is likely to occur if cutting equipment is employed to maintain it. Additionally, there is the issue of varying cutting characteristics of the different types of plants encountered due to diverse biological characteristics. Some plants, such as iceplant, can be highly dense due to water retention. Aside from causing heavy resistance to cutting motions, this may also present a problem in collection of the cuttings due to the weight capacity of the collection vehicle.

Figure 4.1 A selection of invasive weeds in California: (a) yellow star thistle, (b) tumble weed, (c) iceplant, (d) Cape ivy, (e) wild sunflower, (f) Klamathweed, (g) Scotch broom

Finally, there are the operational issues related to maintenance in a roadside environment. Maintenance must be performed in medians and on road shoulders while traffic is present. Median maintenance presents such operational issues as worker safety, cutting of vegetation near and around guard rails and posts (Figure 4.2), and limited

Figure 4.2 Vegetation and debris in between and around guard rails.

operational space for vehicles. Shoulder maintenance present such operational concerns as worker safety, cutting of vegetation in and around culverts to allow for proper drainage (Figure 4.3), cutting of vegetation near and around guard rails and posts, and cutting of vegetation along inclines.

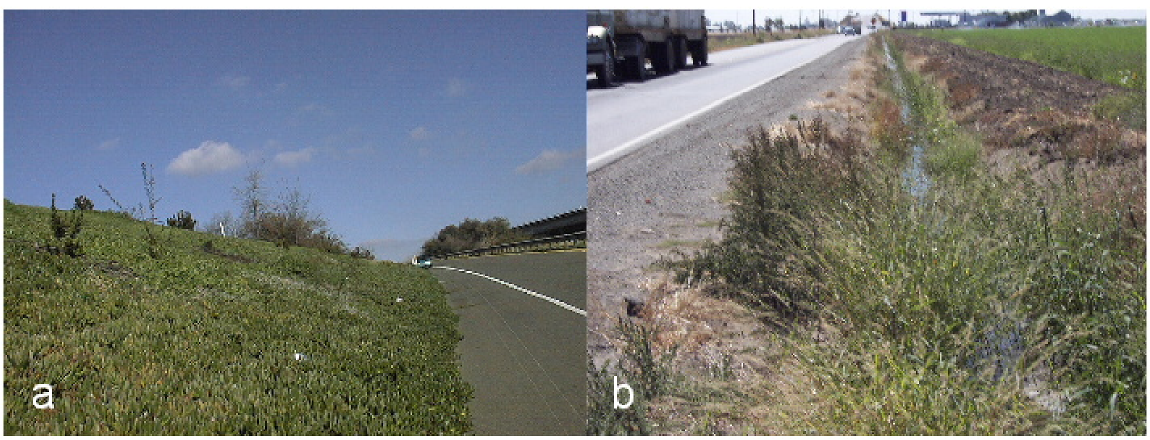


Figure 4.3 Shoulder vegetation along an incline (a) and in a culvert (b).

4.1.1.2 Caltrans Operating Guidelines for Vegetation Control

In meeting Caltrans' needs, it is useful to know what standards Caltrans has set for itself with respect to roadside vegetation maintenance. Caltrans' primary need for vegetation management is due to safety with respect to visibility and fire prevention. The most constraining guideline is Caltrans' mandate to reduce herbicide use from 1992 levels by 80% in 2012 due to environmental concerns. This is a problem in that herbicides tend to be the most efficient and cost effective way to control vegetation.

There are many other concerns that also impact operating guidelines. These include: worker safety, impact of operations on traffic flow, operational cost and efficiency, erosion control, prevention of the spread of noxious weeds, storm water quality, promotion of native species of vegetation, and aesthetics.

Caltrans also has many specific guidelines for particular maintenance operations laid out in The State of California Department of Transportation Maintenance Manual [6]. Chapter 2 of this manual addresses operational issues with respect to vegetation control. In this chapter, reference is given to state laws related to vegetation maintenance along streets, roads and highways. Further, guidelines are laid out with respect to specific operational procedures. For example, the manual states that when mowing, "Do not mow to a height of less than 4 in (102 mm). Mowing to a lower height risks scalping of the ground which …." To have a good overall understanding of Caltrans' operational guidelines with respect to vegetation control, it is necessary to read the above mentioned chapter.

4.1.1.3 Interview at Caltrans Maintenance Headquarters

On September 7, 2003, a meeting was held at Caltrans Headquarters in Sacramento between several representatives of AHMCT and Caltrans Maintenance Managers Rick Houston and Dave Beach [14]. The purpose of the meeting was to try to define more clearly Caltrans Maintenance Departments' needs with respect to equipment for vegetation maintenance along California's roadways. Several problem areas were discussed. These include:

1) Cutting and removal of vegetation in culverts along sides of roads.
2) Edging of iceplant and other groundcovers and removal of cuttings.
3) Cutting and removal of vegetation under and around guard rails.
4) Controlling sporadic weed growths through cracks in hardscapes and road surfaces.
5) Removing duff (cuttings from mowing) in a 1.2 m (4 ft) control strip along shoulders and wide medians to help prevent fires due to cigarettes tossed from passing vehicles.
6) Cutting of vegetation on gutters with berms without damaging equipment.

No conclusions were drawn at this meeting with respect to the specific solutions to these problem areas in vegetation maintenance. However, the information gained was important in formulating cutting mechanism designs for addition to the ARDVAC vehicle.

4.1.2 Other Customers' Needs

Other entities whose needs should be considered, though not directly tied to the project, include other DOTs, municipalities, and private companies who contract through DOTs and municipalities to do roadside maintenance, as these entities may be potential customers if the ARDVAC project proves to be a success with respect to vegetation maintenance. Other DOTs may have specific regional vegetation needs that dominate their maintenance programs. However, because California is such a large state of varying terrain and climate type, many of these other DOT's needs may be addressed in meeting a broad range of needs for Caltrans.

Some municipalities apply vacuum vehicles to the collection of vegetation debris (Figure 4.4) and take responsibility for maintaining roadside vegetation within their jurisdiction. A big difference between municipalities and DOTs is that maintenance equipment employed by municipalities must be capable of being used in a relatively close proximity to passing pedestrians, whereas most DOT vegetation control is done far away from pedestrians. Subcontractors who perform maintenance for DOTs and municipalities will have many of the same needs as DOTs and municipalities, as their work is generally performed under guidelines specified by those DOTs and municipalities.

Figure 4.4 Municipal workers manually vacuum up vegetation debris in Jackson, MS.

4.2 Determining the Optimum Cutting Mechanism

In determining the optimum cutting mechanism for this project, previous analysis performed by former AHMCT researcher Ben Richardson [23] was relied upon heavily. In his thesis on conceptual development of end-effector vegetation cutting devices, he did an extensive comparison of many commercially available mechanisms for cutting vegetation. He went so far as to build prototype bench models of many alternatives for a more thorough comparison. However, although his comparison analysis highly favored the manual hedge trimmer (oscillating cutter) in many comparison criteria, no bench prototype was built for this option and no further consideration was given to the oscillating cutter until being reassessed for use in this current design iteration. For a thorough description of the types of cutting mechanisms considered by Richardson, see Section 3.6 of Richardson's thesis [23].

4.2.1 Cutter Comparisons

In comparing cutters for his analysis, Richardson decided the appropriate comparison criteria were: cost of the cutter unit, cost per kW of cutting power, gauge of material that can be cut, power to weight ratio, and power to weight per dollar (cost). Upon reviewing these, it was determined that there are three more criteria that needed to be considered. These are: safety to passing vehicles in a roadway environment, safety to nearby workers or people not protected by being in a vehicle, and the ability to operate upon a wide range of vegetation. It is important to note that these criteria are not readily quantifiable as with the criteria specified by Richardson, however, by a simple check of these criteria, some options are easily eliminated, as safety and operability on a wide variety of vegetation are critical factors to the success of this project.

4.2.1.1 Cost

Table 4.1 graphically represents Richardson's comparison of off-the-shelf costs of the different cutter units considered viable for this project. In this case lower is better, and the hedge trimmer (oscillating cutter) comes in lowest at \$60. The actual hedge trimmer unit purchased for this project, the STIHL HL Adjustable Hedge Trimmer Attachment [25], ended up costing \$209. However, much of the price increase was due to choosing a hedge trimmer of industrial quality and modular design. Because of its modular design configuration, integrating this unit into a design became easier and cheaper than building much of the cutter mechanism from scratch. However, even at the increased price, it still comes in at 2nd best behind the string trimmer for this comparison.

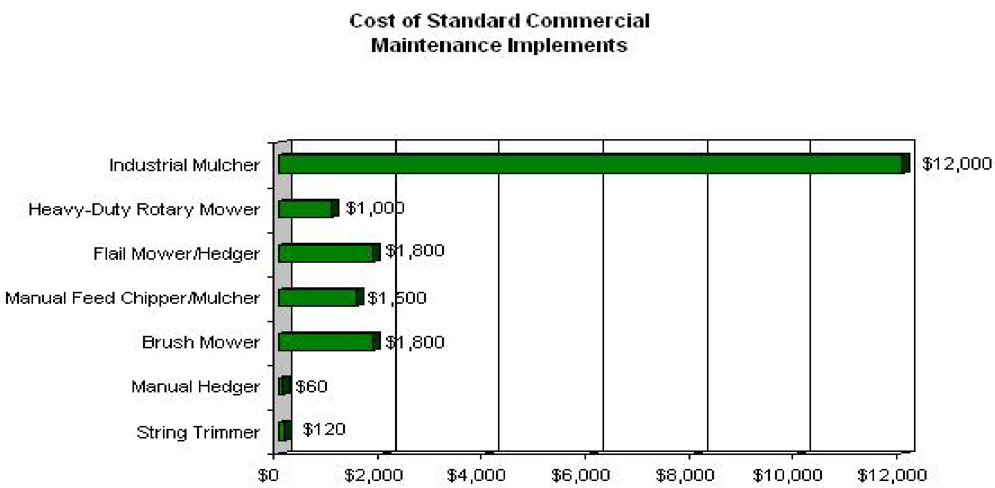


Table 4.1 Richardson's cost comparison.

4.2.1.2 Cost per kW of Cutting Power

Table 4.2 graphically represents Richardson's comparison of off-the-shelf costs of the different cutter units considered viable for this project. Once again, in this case lower is better, and the hedge trimmer (oscillating cutter) comes in at \$80/kW. However, because the actual hedge trimmer cost \$209 and has a power output of 1 kW, the actual price is \$209/kW. This leads to a ranking of 6th out of 7 for this criteria, placing it behind the brush mower, manual feed chipper/mulcher, string trimmer, flail mower, and the heavy duty rotary mower, in that order. However, it should be noted that the increase in price, and thus, cost per kW of the manual hedge trimmer was due to choosing a modular model that was readily adaptable for design. Since this has not been factored in for the other models, it is more appropriate to use Richardson's ranking of \$80/kW placing it 2nd behind only the rotary mower.

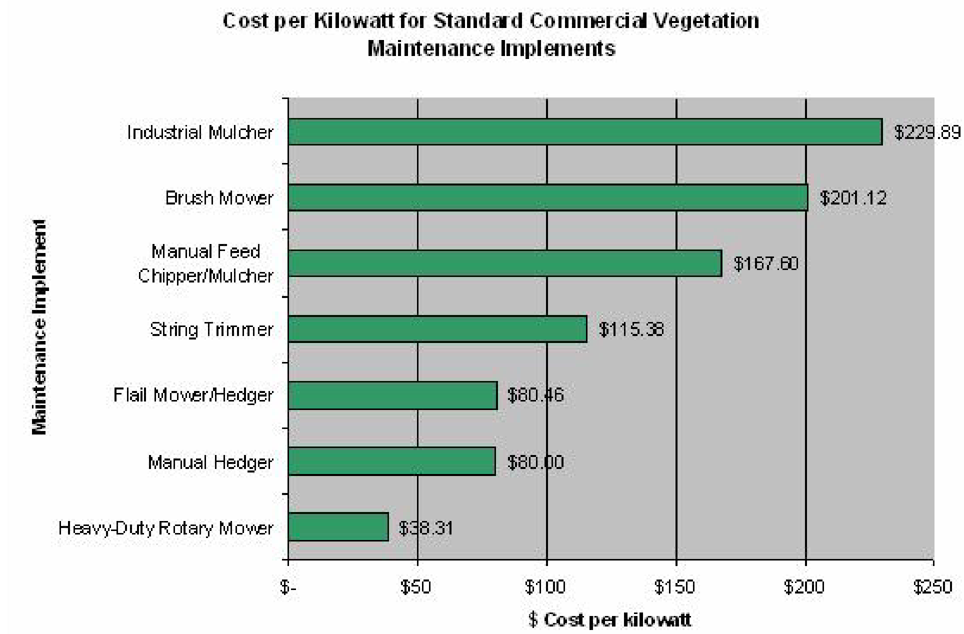
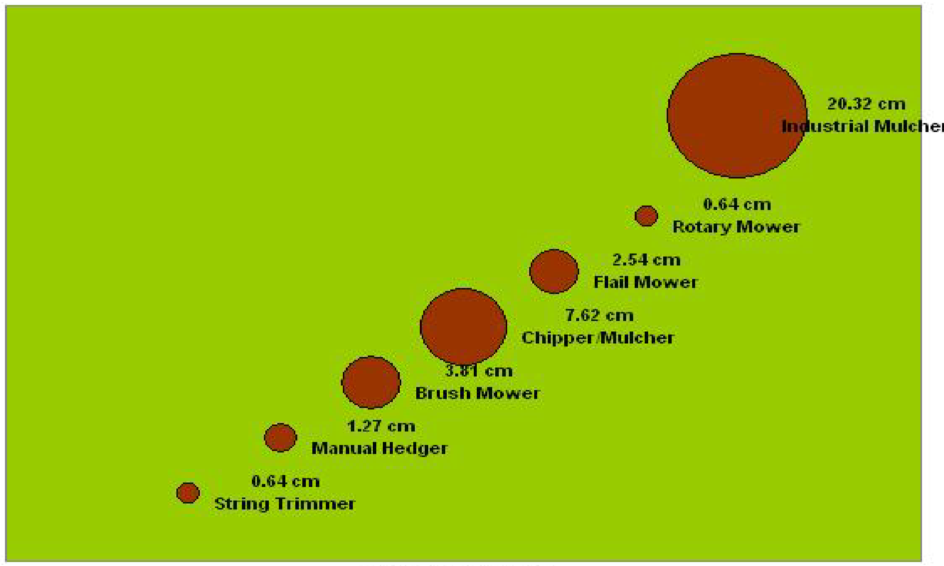


Table 4.2 Richardson's cost per kW comparison.

4.2.1.3 Gauge of Material Which Can be Cut

Table 4.3 graphically represents Richardson's comparison of the vegetation stem diameters (gauges), which may be cut by the various cutting mechanisms. In this case, bigger is better as it allows for a wider range of vegetation to be cut. In this comparison, the hedge trimmer is able to handle gauges of up to 1.27 cm (0.5 in) ranking it 5th out of the 7 possible cutters, behind the flail mower, brush mower, chipper/mulcher, and industrial mulcher in that order. However, it should be noted that the median blade edge to blade edge span for the STIHL hedge trimmer selected for this design is 2.54 cm (1.0 in). While STIHL may not be able to handle these gauges in hard wood, it is very likely that it can handle more fragile weedy vegetation approaching gauges of 2.54 cm (1.0 in). However, it should be noted that most vegetation likely to be encountered in the roadway environment do not have gauges above 2.54 cm (1.0 in).

Standard Cutting Gauges For Commercial Maintenance Implements



Cutting Gauges (cm)

Table 4.3 Richardson's cutting gauge comparison.

4.2.1.4 Power to Weight Ratio

Table 4.4 graphically represents Richardson's comparison of power to weight ratios of the different cutter units considered viable for this project. Once again, in this case bigger is better, and the hedge trimmer (oscillating cutter) comes in 2^{nd} at 236 W/kg. However, because the actual hedge trimmer used has a mass of 1.45 kg (3.2 lbm) and has a power output of 1 kW, the power to weight ratio is 690 W/kg. This far exceeds the performance of all other cutting mechanisms sampled by Richardson, where the next closest power to weight ratio is that of the heavy duty rotary mower at 287 W/kg. However, it should be noted that the numbers given by Richardson do not specify whether or not this include motor weight. If this is so, then it is more appropriate to use the numbers generated by Richardson.

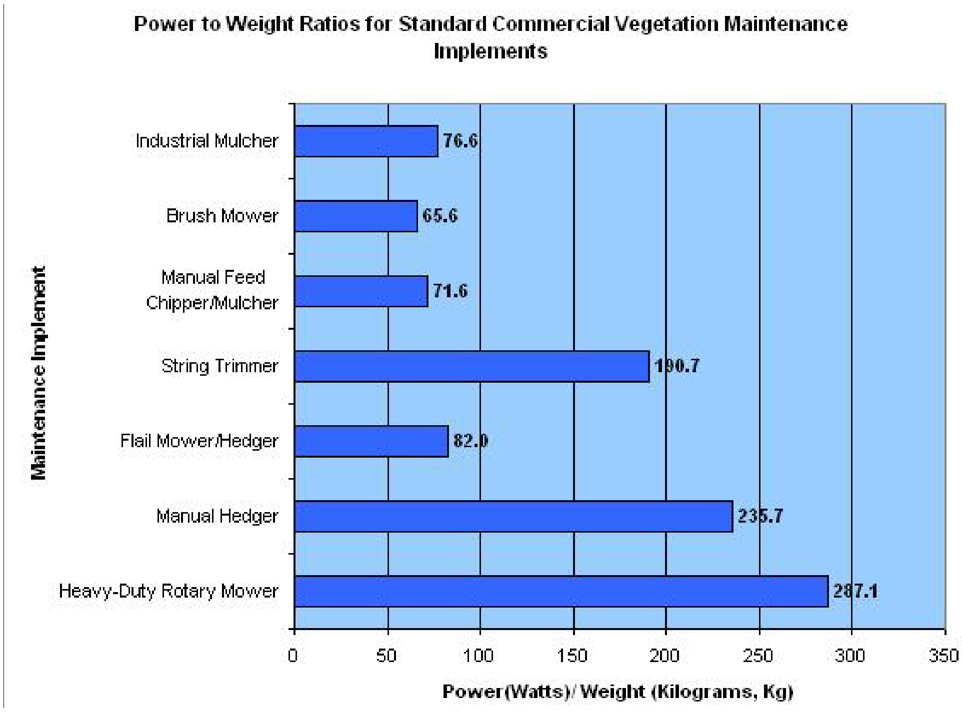


Table 4.4 Richardson's power to weight comparison.

4.2.1.5 Power to Weight per Dollar (Cost)

Table 4.5 graphically represents Richardson's comparison of power to weight per dollar ratios of the different cutter units considered viable for this project. Once again, in this case bigger is better, and the hedge trimmer (oscillating cutter) comes in 1st at 3.93 W/(kg · $). However, because the actual hedge trimmer used has a mass of 1.45 kg (3.2 lbm), has a power output of 1 kW, and a cost of $209, the power to weight per dollar ratio is 3.30 W/(kg · $). This is still far in excess of the performance of all other cutting mechanisms sampled by Richardson, where the next closest power to weight per dollar ratio is that of the heavy duty rotary mower at 1.59 W/(kg · $). The importance of this comparison is that it creates a scale from which the three important factors of cost, weight, and power can be observed in a combined metric.

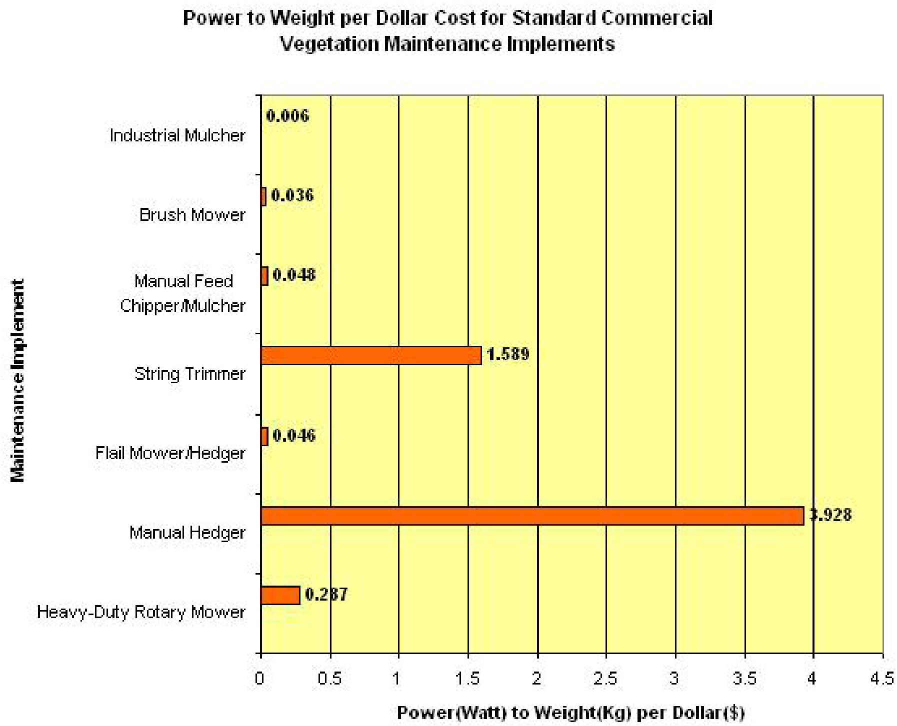


Table 4.5 Richardson's power to weight per dollar comparison.

4.2.1.6 Safety to Passing Vehicles in a Roadway Environment

Safety to passing vehicles is a highly important design criterion for this project as the operational environment is such that cars and trucks will be passing within a few meters of the ARDVAC vehicle at speeds of more than 97 km/hr (60 mph). All other proposed cutter mechanisms involve a spinning cutter blade or string. This cutting action, involving rotary speeds of up to 9000 rpm leading to maximum blade tip speeds of 144 m/s (471 ft/s), will tend to kick small debris, such as small rocks and broken glass, long distances at high speeds. Launching projectiles in such a manner into fast moving traffic can lead to driver distraction causing the potential for serious accidents. If shrouds are to be employed to reduce this likelihood, the cutter access to restrictive areas may be limited such as when cutting in and around guard rails and next to posts and barriers. The hedge trimmer (oscillating cutter) does not encounter this problem. The hedge trimmer's teeth oscillate back and forth with respect to each other with strokes that are approximately 3.3 cm (1.3 in) long and teeth that are 1.9 cm (0.75 in) in length and oscillate at 1364 cycles/sec leading to peak blade speeds of 3.0 m/s (9.9 ft/s). Thus the oscillating cutter can be utilized without the need for shrouding in the freeway environment allowing it to operate close to guard rails, posts and barriers. Further, with rotary cutters, there is also the danger of throwing a blade in the event of cutter damage. This is potentially more dangerous that projecting a rock into traffic. Due to the construction of the hedge trimmer, it is most likely to stall when damaged, thus, not presenting any danger to passing traffic.

4.2.1.7 Safety to Nearby Workers or Pedestrians

Safety to exposed workers and pedestrians is an important consideration in the design of cutting mechanisms for the ARDVAC in that it is possible there could be municipal applications at some point. However, for the freeway environment, this is not as great of a concern as there should be no pedestrians present. Further, one of the driving reasons for creating the ARDVAC is to remove maintenance workers from the exposed freeway environment, allowing them to remain in the safety of their vehicles.

With the oscillating cutters, there is the danger that small members, such as fingers, may be severed if placed in the operational area. Shrouding of rotary and oscillating cutters may cause them to be ineffective at rapid vegetation removal due to shroud interference with the vegetation to be cut and clippings becoming lodged in the shrouding. Rotary cutters, even when shrouded, may still present a danger of throwing projectiles at exposed workers or pedestrians.

Thus, the ARDVAC, when employed with oscillating cutters, should be safe for use in municipal areas provided people are kept a safe distance from the vehicle of at least 4.6 m (15 ft) from the work area of the end-effector. Although this work area is far greater than the approximately 1.5 m (4.5 ft) that the oscillating cutters protrude from the end-effector's central vertical axis, the extra area allows for any sudden change in boom orientation. This vehicle should never be operated when exposed workers or pedestrians are inside of the operational area.

4.2.1.8 Ability to Process a Wide Range of Vegetation

This is an important criterion for the simple reason that the more versatile the ARDVAC is in maintenance on various types of vegetation, the more attractive it will be as an overall tool for roadside maintenance. Cutting gauge is an important consideration when designing for the range of vegetation the cutter may process. All vegetation above the cutting gauge range of the cutting mechanism is not able to be processed by the cutter. The oscillating cutter is somewhat limited by comparison to the most of the other cutters in maximum cutting gauge, however, this limited maximum cutting gauge range of 1.3 – 2.5 cm (0.5 – 1.0 in) is adequate for processing most vegetation encountered in the freeway environment. Although cutting gauge is an important factor in this criterion, ability to articulate is even more important. Because the geometries of different varieties of vegetation may be diverse (Figure 4.1), along with other cutting variables such as terrain differences (Figure 4.3) and obstacles such as guardrails (Figure 4.2), posts and barriers, the ability to process vegetation may be limited by the ability of the cutting mechanism to articulate to it. The oscillating cutter is not limited in its ability to articulate due to shrouding concerns. Further, its light weight is somewhat ideal when designing joints to articulate it. All rotary cutters must be shrouded due to safety concerns, therefore to allow for articulation, shrouding must be able to adjust itself to the full variety of articulated cutting position to remain effective. This presents a major design obstacle for rotary cutters.

4.2.2 Establishing the Oscillating Cutter as the Ideal Cutting Mechanism for Use with the ARDVAC

When compiling all of the design criteria considered during this project, it is clear that the hedge trimmer (oscillating cutter) is the most ideal choice of the cutting mechanisms considered for use with the ARDVAC. The oscillating cutter is able to process a broad range of vegetation likely to be encountered along the roadway environment. It is the safest of the cutting mechanisms considered for use in a freeway environment. It is low in cost and has a high power to weight ratio when compared to the other cutting mechanisms.

4.3 New End-Effector Motions for Vegetation Removal

With the addition of cutting mechanisms to the end-effector to affect vegetation removal, several other motions must now be considered in order to provide for cutting action and articulation of the cutters. Currently, there are two cutting configurations that have been designed for the end-effector. These are, the oscillating cutter system (Figure 4.5), which is the focus of this paper, and the rotary cutter system (Figure 4.6), which is being designed by Kenneth R. Harker [12]. These two systems are being designed in a coordinated fashion to allow for modular designs that can be interchanged on the end-effector. The purpose of mentioning the rotary system here is so that it may be considered in the overall motion capabilities and the hydraulic design of the system.

The new cutting motions are named motions 8-11 (Figure 4.5 and Figure 4.6) and will be described in the following sections.

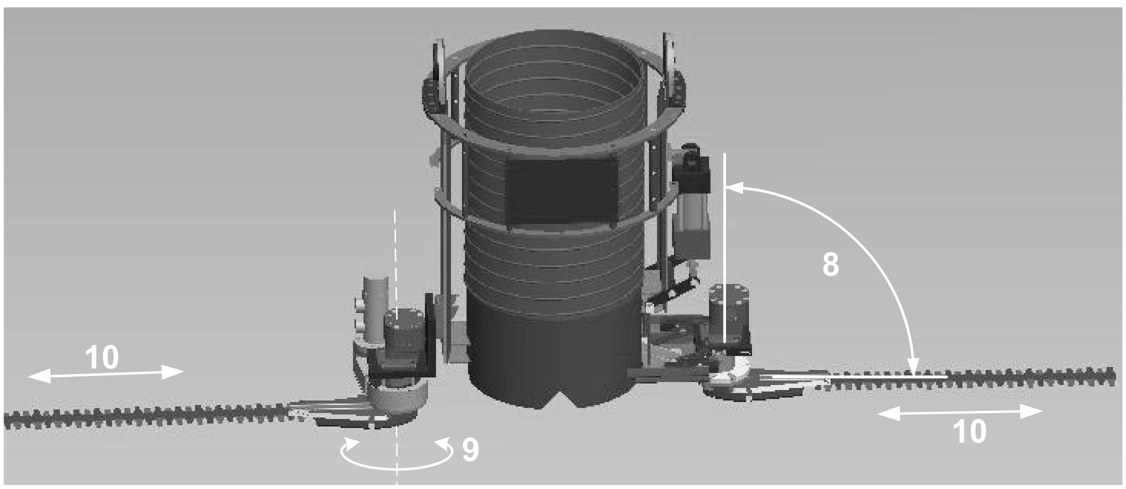


Figure 4.5 Motions 8-10 of oscillating cutter system.

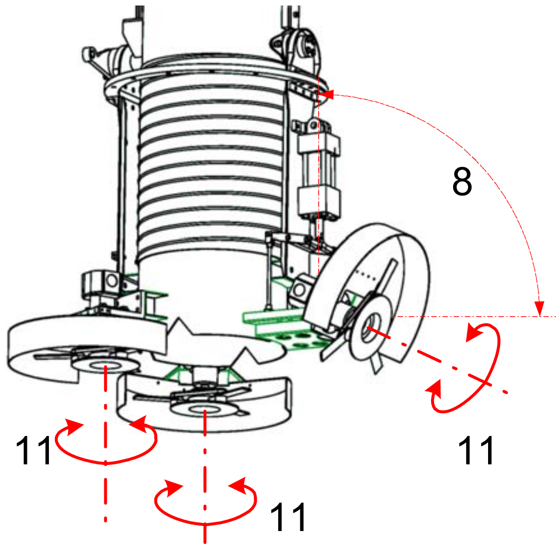


Figure 4.6 Motions 8 and 11 of the rotary cutter system.

4.3.1 Motion 8

In an attempt to be modular in design, both the rotary and the oscillating cutter systems were designed with mounting and hydraulic similarities. The most visible evidence in this is the Linkage Assembly (Figure 4.7) designed by Kenneth R. Harker, which was intended to allow cutter articulation in both designs.

4.3.1.1 Linkage Assembly and Angled Cutting Planes

The Linkage Assembly allows for cutting in the horizontal plane up through 90° until a cutting plane perpendicular with the ground is reached (Figure 4.8). This motion, motion 8, allows for cutting on inclines, along fences and walls, and normal cutting/mowing applications that are horizontal to the ground. For the rotary configuration there is the added ability to perform edging operations when cutting perpendicular to the ground. Note, that motion 8 is the only articulation in the rotary cutting system.

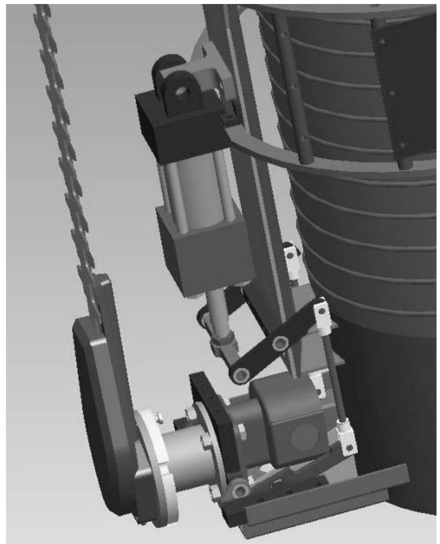


Figure 4.7 Linkage Assembly CAD close-up.

The Linkage is driven by a Vickers hydraulic cylinder. The cylinder's stroke length is .0397 m (1 9/16 in) and its piston and rod diameters are 0.0381 m (1.5 in) and .0159 m (5/8 in) respectively.

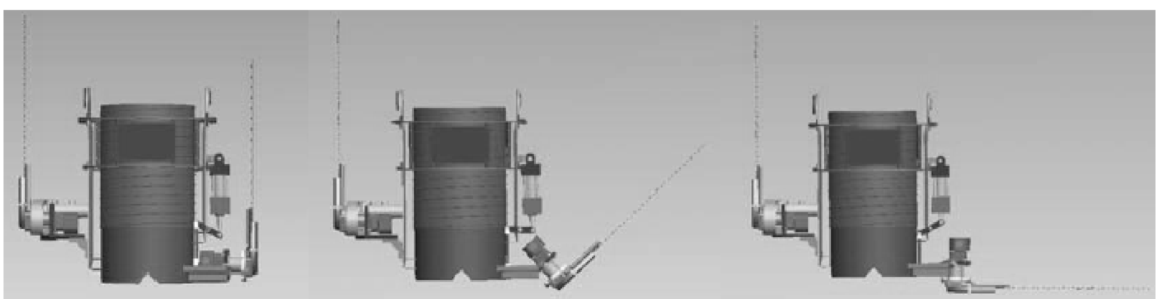


Figure 4.8 Motion 8 articulation through 90°.

An additional 90° of articulation can be gained from the oscillating cutter system when manually reversing the blade orientation of the cutter by 180° so that the cutting blade is in the vacuum stream (Figure 4.9). Note that motion 8 addresses Caltrans' need for cutting and removal of vegetation in culverts along sides of roads (4.1.1.3 and Figure 4.3).

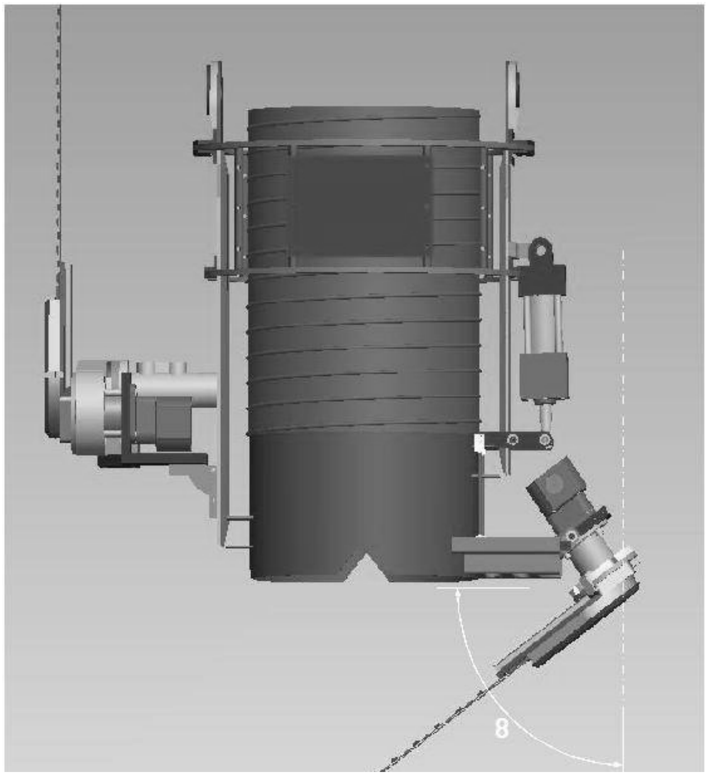


Figure 4.9 Additional 90° of articulation from motion 8 with oscillating cutting system.

4.3.2 Motion 9

Located on the nozzle, 180° from the Linkage assembly is the other oscillating cutter (hedge trimmer) and Rotary Articulation Assembly. The Rotary Articulation Assembly allows for continuous rotary motion of the hedge trimmer in the horizontal plane (Figure 4.10). Note that the oscillating cutters are staggered in height by 3.5 cm (1.4 in) when in their horizontal cutting positions so as to avoid interference.

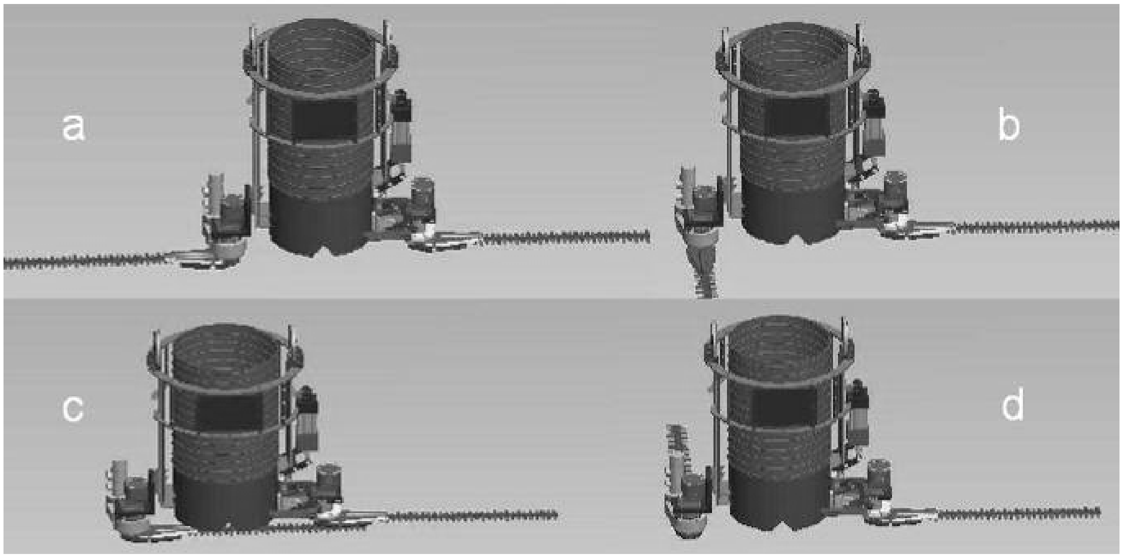


Figure 4.10 Motion 9 of oscillating cutter system.

The purpose of motion 9 is two-fold. First, this allows for a swath of 2.13 m (7 ft) to be cut while driving down the road at a continuous pace of approximately 3 – 8 km/hr (2 – 5 mph) when applying both oscillating cutters. Thus, in some narrow strips of median or roadside shoulder, mowing could be replaced where it may be difficult to insert a mower. Note that if the ARDVAC were to be employed for this purpose, cuttings would be picked up by the vacuum as the vehicle progressed down the road, thus addressing one of Caltrans' needs (Section 4.1.1.3), the removal of duff from mowing operations to reduce fire potential. Second, when vegetation growth is in light patches spread out in relatively short intervals, a mower would probably not be warranted. Further, it would be rather monotonous for a user to have to position the end-effector and boom for each little patch of growth. Because this motion requires no specific input from the driver once it is going, the vehicle would not have to stop for the driver to target each specific weed or patch of weeds with boom and end-effector articulation, thus generating higher efficiency.

4.3.2.1 Mowing Profiles

For this report, a computer model has been generated to demonstrate the cutter profiles when viewed from above given varying inputs for vehicle speed, rotary motor speed for the oscillating cutter, and time span of interest. Many plots were generated for different cutter profiles based on vehicle speed and cutter rotary speed; however, the plots that will be demonstrated here are for optimum cutter mowing profile, optimum cutter profile for intermittent vegetation, and a poor cutter mowing profile for comparison. Note that these scenarios are merely a few samples from a broad range of vehicle and cutter rotational speed configurations and may be altered as seen fit for cutting environment conditions.

4.3.2.1.1 Optimum Cutter Mowing Profile

The optimum cutter mowing profile is shown in Figure 4.11. This cutter profile is generated from a vehicle speed of 2 mph with an oscillating cutter rotational speed of 240 rpm for a time span of 0.75 seconds. Note by the x axis that the total cutting swath approaches 2 m (7 ft). The horizontal bar on the left represents the cutting path taken by the non-rotational oscillating cutter, while the cutter path on the right illustrates the cutting path of the rotational oscillating cutter. The red hue in the cutting path indicates area most recently cut in the time span. The white background demonstrates area that is not affected by the oscillating cutters. Figure 4.11 is considered optimum because the cutting circles overlap such that very little area is left uncut. Lower oscillating cutter rotational speeds are capable of achieving similarly satisfactory mowing results; however, the vehicle speed must be decreased for these lower rotational speeds.

Note the small gap between the rotational cutter swath and the fixed cutter swath. This gap is approximately 1 inch wide and is due to design constraints involved with using the same Nozzle configuration as the rotary cutting system (Figure 4.6). In future iterations of this design, this gap should be eliminated.

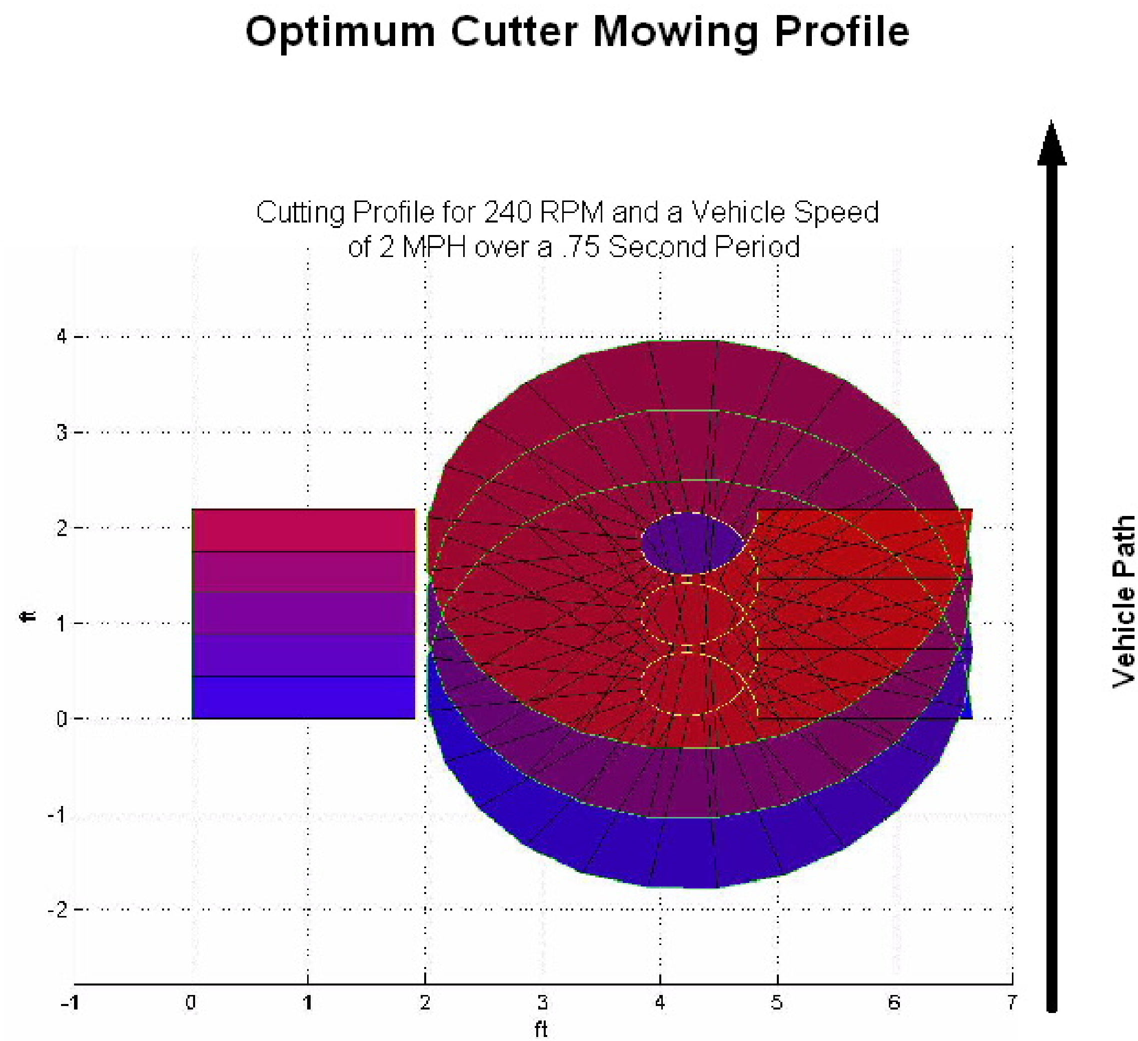


Figure 4.11 Optimum cutter mowing profile.

4.3.2.1.2 Optimum Cutter Profile for Intermittent Vegetation

The optimum cutter profile for intermittent vegetation is shown in Figure 4.12. This cutter profile is generated from a vehicle speed of 13 km/hr (8 mph) with an oscillating cutter rotational speed of 240 rpm for a time span of 0.75 seconds. As previously noted, the x axis indicates that the total cutting swath approaches 2m (7 ft). Again, the horizontal bar on the left represents the cutting path taken by the non-rotational oscillating cutter, while the cutter path on the right illustrates the cutting path of the rotational oscillating cutter. As noted before, the red hue in the cutting path indicates area most recently cut in the time span, while the white background demonstrates area that is not affected by the oscillating cutters. Figure 4.12 demonstrates that for increasing vehicle speed, while maintaining cutter rotational speed, much wider gaps will be left uncut as the rotational cutting paths do not overlap as much as previously (Figure 4.11). Thus, this configuration would not be ideal for mowing. However, there is sufficient cutter path area for picking up most intermittent vegetation growth that would lie in the 2 m (7 ft) swath.

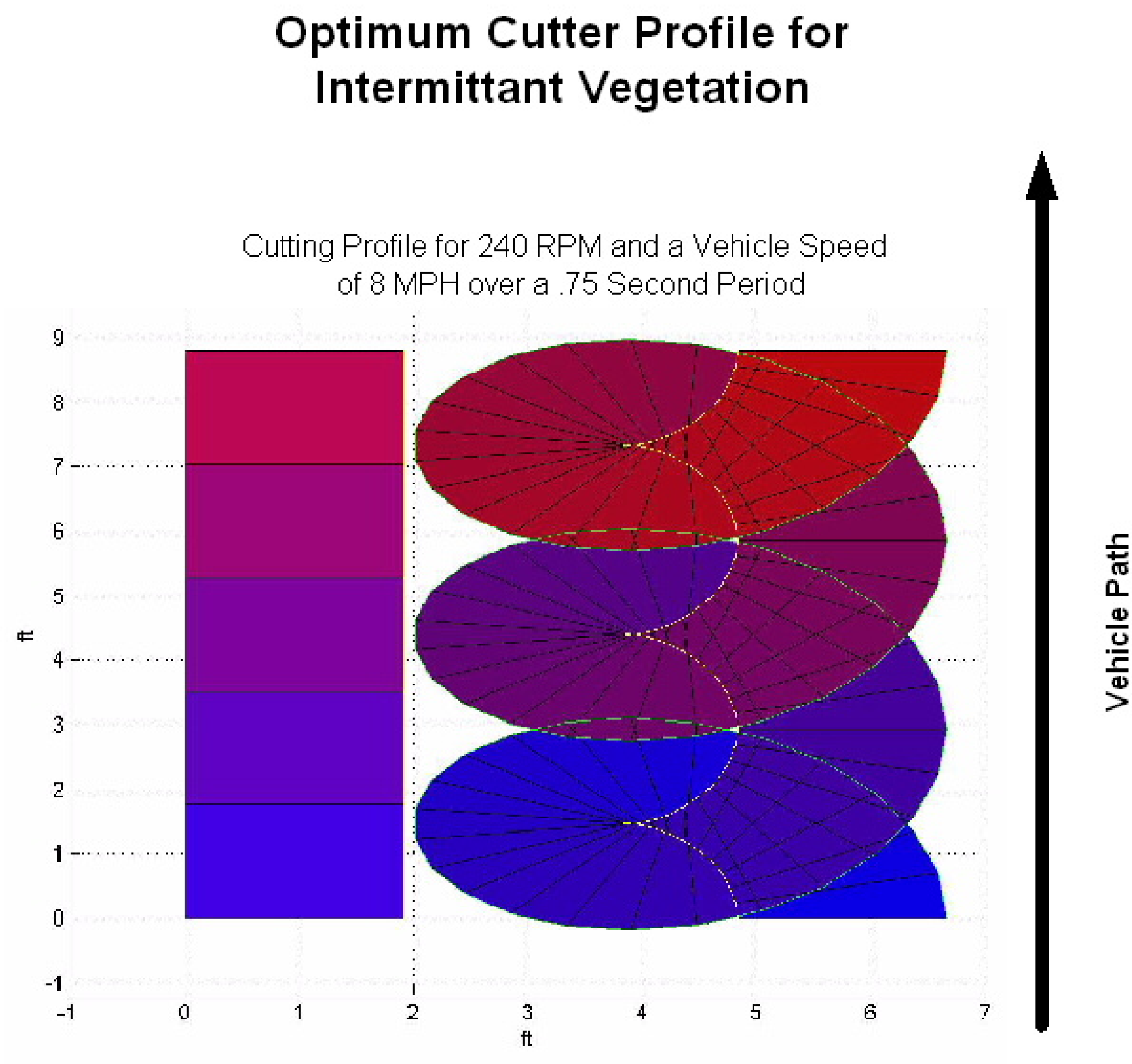


Figure 4.12 cutter profile for intermittent vegetation.

4.3.2.1.3 A Poor Cutter Mowing Profile

A poor cutter mowing profile is shown in Figure 4.13. This cutter profile is generated from a vehicle speed of 8 km/hr (5 mph) with an oscillating cutter rotational speed of 60 rpm for a time span of 0.75 seconds. As previously noted, the x axis indicates that the total cutting swath approaches 2m (7 ft). Again, the horizontal bar on the left represents the cutting path taken by the non-rotational oscillating cutter, while the cutter path on the right illustrates the cutting path of the rotational oscillating cutter. As noted before, the red hue in the cutting path indicates area most recently cut in the time span, while the white background demonstrates area that is not affected by the oscillating cutters. Note in this plot that the y axis is not one to one in relation to the x axis as in the previous plots. Figure 4.13 demonstrates that increasing vehicle speed, while decreasing cutter rotational speed, allows for extreme gaps in the 2 m (7 ft) cutting swath that will be left uncut. Thus, this configuration would not be ideal for mowing or intermittent vegetation cutting as too much vegetation would be missed.

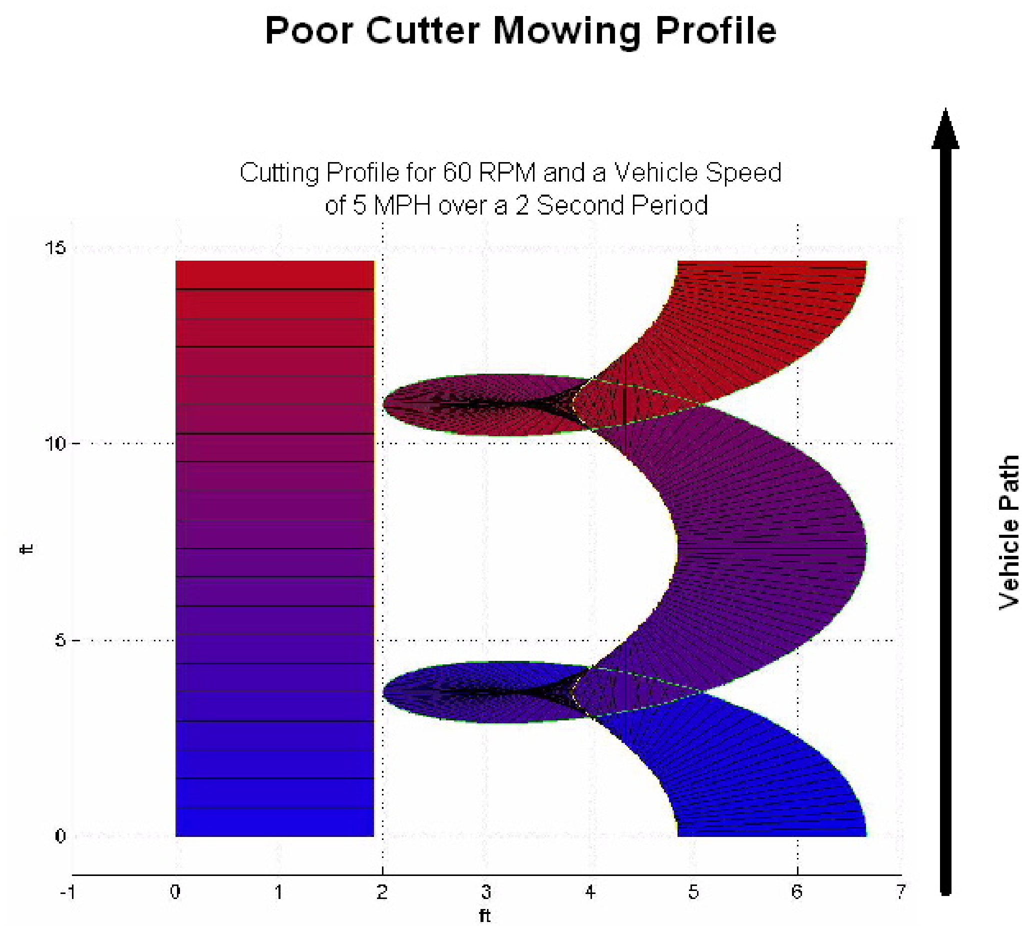


Figure 4.13 A poor cutter mowing profile.

4.3.3 Motions 10 and 11

Although there are two different cutter systems (Figure 4.5 and Figure 4.6), both employ a rotary high speed hydraulic motor to power the cutting action. The motors that were adapted for this project are Haldex GC series gear motors with displacements of 8.47 cm3 (.065 in3).

They are rated for a maximum of 5000 rpm and have a stall speed of approximately 750 rpm. Rotary cutters are driven through a direct drive while oscillating cutters have an integral gear reduction designed into them giving a gear reduction of 3.67:1, providing for 1364 oscillations per minute of the cutting blades.

4.4 Design of the Oscillating Cutter System

The oscillating cutter system design encompasses the adapting of two after-market hedge trimmer devices to the Nozzle assembly of the end-effector and the design of two mechanisms that will allow for articulations of motions 8 and 9 (Sections 4.3.1and 4.3.2) while allowing for power transfer to the cutting motors to allow for motion 10 (Section 4.3.3). These will be called Articulated Oscillating Cutter Assembly (motion 8) and the Rotational Oscillating Cutter Assembly (motion 9). In order to mount the mechanisms to the end-effector, design of a new Nozzle Tip was required.

4.4.1 Design of the New Nozzle Tip

Early on in the design process, it was determined through collaboration with Harker that cutting mechanisms should be mounted near the vacuum inlet, as this would allow for easy collection of cuttings by the ARDVAC end-effector. Thus the Nozzle Tip was an ideal mounting location. However, the original Nozzle Tip (Figure 3.11) was not originally designed for purposes other than collecting debris. Thus it was necessary to make several changes to the Nozzle Tip design, many of which were undertaken by Harker. Therefore, the four most significant changes will be pointed out here (Figure 4.14). For more detail on this design, it is recommended to see Harker's thesis. [12]

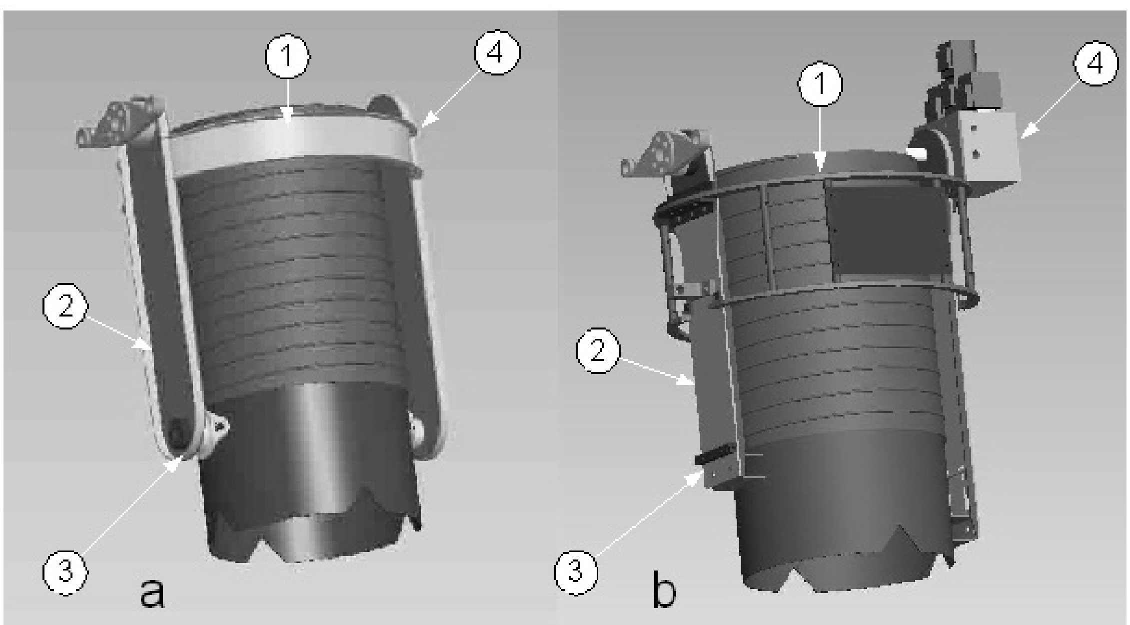


Figure 4.14 Four significant Nozzle Tip changes from the old design (a) to the new design (b).

4.4.1.1 Nozzle Change 1

The purpose of changing the Nozzle Bracket, as pointed to by arrow 1 in Figure 4.14, is for lighter weight and the addition of potential mounting locations for hydraulic components. The new design calls for lower and upper aluminum rings joined together by aluminum tubes threaded lengthwise (Figure 4.15). The tubes also have holes threaded transversely through them for the purpose of mounting the Nozzle Plates and hydraulic components. The plates were designed out of 16 gauge sheet steel for the purpose of mounting the Lower Hydraulic Manifold as shown by arrow 4 in Figure 4.14. However, the positioning of the manifold has been moved to the pin joint opposite the pivot cylinder flange for reasons to be explained later in 4.4.1.4. Thus, the Nozzle Plates may be applied to the Nozzle Assembly as deemed necessary for other mounting considerations.

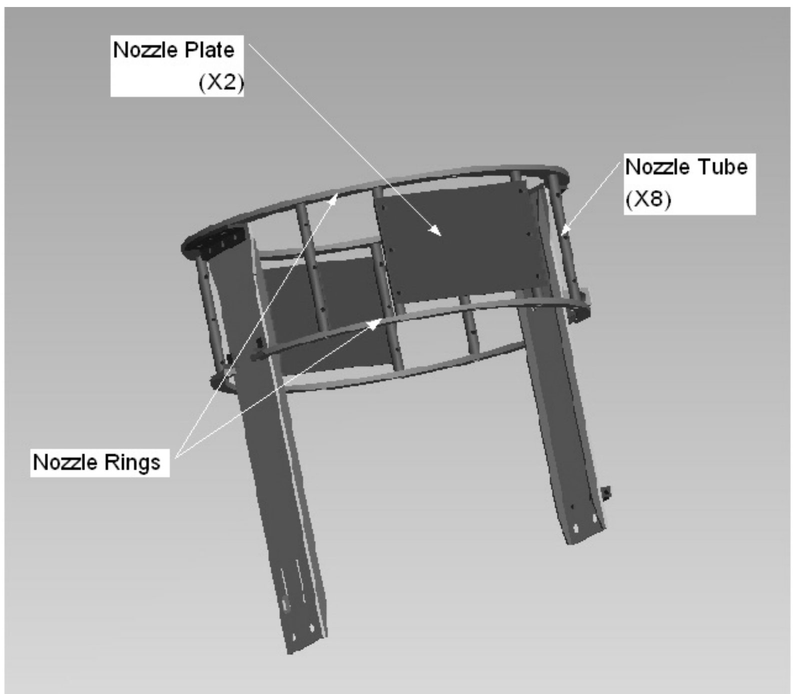


Figure 4.15 New Nozzle Bracket components.

4.4.1.2 Nozzle Change 2

The main purpose of changing the Nozzle Bracket Arms, as pointed to by arrow 2 in Figure 4.14, is to provide mounting locations for cutter assemblies (Figure 4.16). The other reason for the change was to allow for a greater degree of modularity by making it easier to remove the Nozzle Assembly from the end-effector, allowing for easy switching between the oscillating cutter system (Figure 4.5) and the rotary cutter system (Figure 4.6).

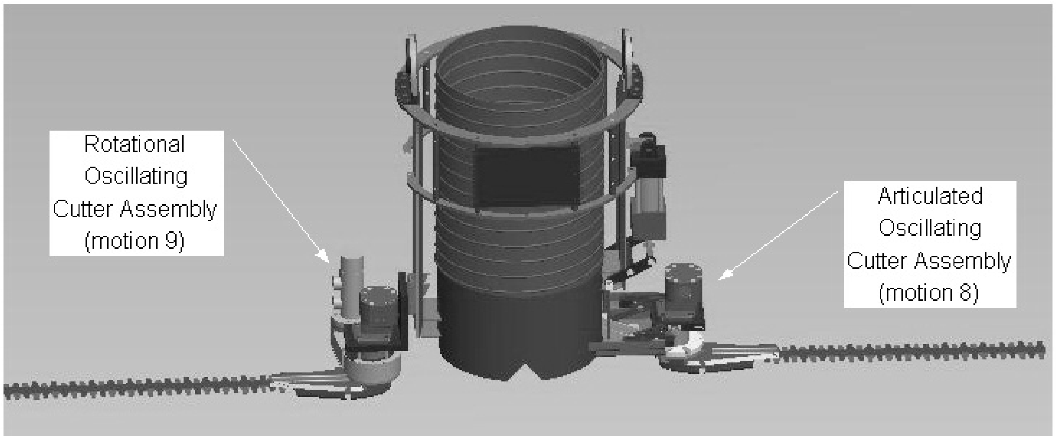


Figure 4.16 Oscillating Cutter Assembly locations.

Greater modularity is illustrated in Figure 4.17. The new Nozzle Bracket Arm is comprised of two major pieces, the Pivot Arm and the Mounting Plate. These are joined to each other and the rest of the Nozzle Assembly through fasteners. The old design is a one piece design that is welded to the Nozzle Bracket. The greatest advantage to the new design is the ability to remove the entire Nozzle Assembly below the Pivot Arms. This allows for relatively quick changes between the Nozzle assemblies for the rotary cutting system and the oscillating cutting system.

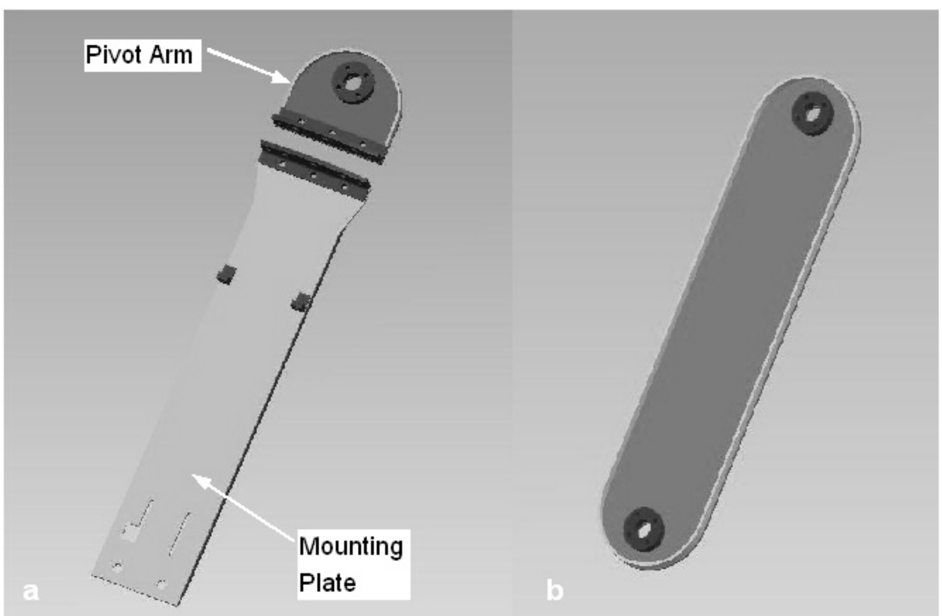


Figure 4.17 Nozzle Bracket Arms. a) New design and b) Old design.

4.4.1.3 Nozzle Change 3

This change, as pointed to by arrow 3 in Figure 4.14, alters the way the Nozzle tip is fastened to the Nozzle arms. The old configuration employs a pin joint that allows for rotation of the nozzle upon impact with objects encountered through normal operation. Because this allows for arbitrary changes in the horizontal cutting plane, this feature was deemed undesirable for this project. Thus, the pin joints were removed and replaced by brackets, which are welded to the Nozzle Tip. The Nozzle Bracket Arms are then joined to these brackets through fasteners.

4.4.1.4 Nozzle Change 4

The change indicated by arrow 4 in Figure 4.14 simply indicates that the Hydraulic manifold for the cutter applications will be mounted to the Pivot Arm opposite the Pivot Cylinder Flange. The previous design had no need for hydraulics on the Nozzle assembly.

4.4.2 Adaptation of the After-Market Hedge Trimmer

Once the oscillating cutter (hedge trimmer) was established as the optimum cutter for this project (4.2.2), it became necessary to design a cutter configuration that could be adapted to the end-effector. Because fabricating an end-effector from scratch would be an expensive process, it was decided to acquire after-market hedge trimmers and adapt them to the design. The hedge trimmer chosen was the STIHL HL adjustable hedge trimmer attachment (Figure 4.18). The main reason for the selection of this model was its modular design. This trimmer was originally designed to be part of a series of STIHL accessories that could be attached or removed from a motor through a quick disconnect coupling. This allowed it to be readily adaptable for use in the ARDVAC cutter assemblies. The motor, for which this trimmer as originally designed to be used with, is a 1 kW (1.3 hp) gasoline powered motor. The span of the teeth is approximately 3.3 cm (1.3 in).

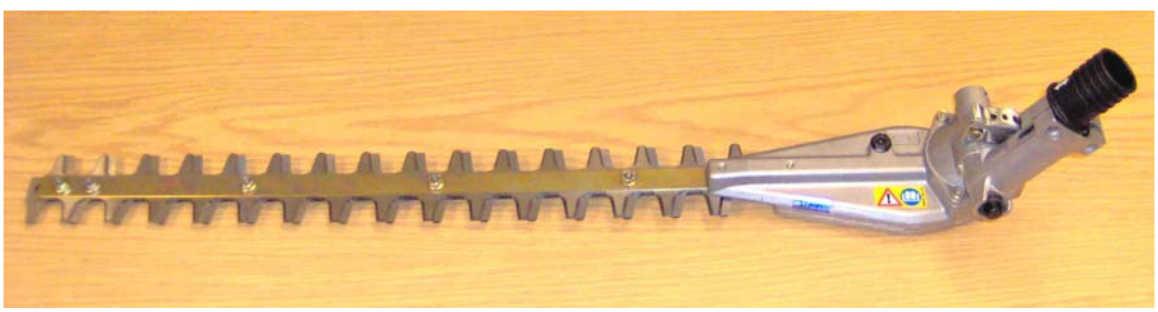


Figure 4.18 STIHL HL adjustable hedge trimmer.

After acquiring the hedge trimmer from the vendor, it was found that the easiest way to adapt the hedge trimmer to the designs for the cutter system was to remove the upper gear housing, which was designed to allow for 90° of articulation in its original intended application. This left the base configuration of the hedge trimmer with a smooth mounting surface (Figure 4.19).

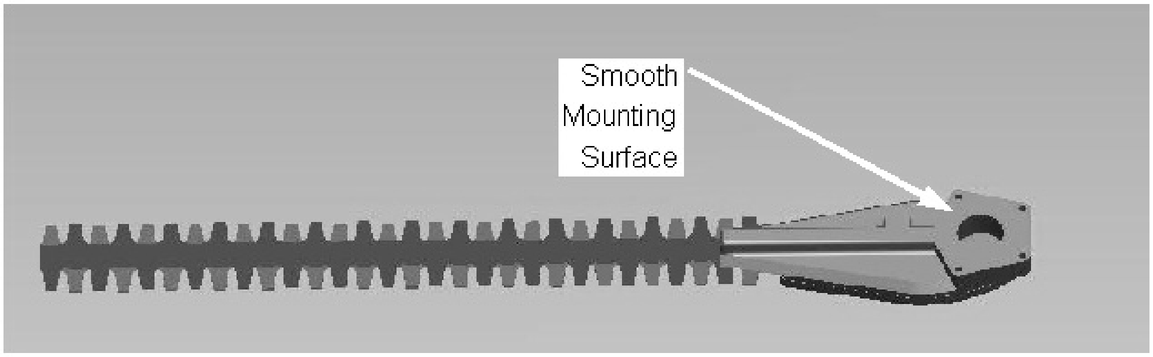


Figure 4.19 Base configuration of the hedge trimmer.

Upon the removal of the upper gear housing, the remaining gear reduction is 3.7:1. This reduction is directly applied from the cutter motor output to the cutting blades (4.3.3).

4.4.3 Design of the Articulated Oscillating Cutter Assembly

The Articulated Oscillating Cutter Assembly (Figure 4.16) includes the Linkage Assembly (Figure 4.7), the cutter motor (4.3.3), the hedge trimmer base configuration (Figure 4.19), and the Adaptor Cup Assembly (Figure 4.21). This assembly allows for motions 8 and 10 (Figure 4.5).

The Linkage Assembly was designed by Kenneth R. Harker, as described briefly in 4.3.1, and is responsible for generating motion 8. The Linkage Assembly begins with the Short Cylinder and its connection to the Nozzle Bracket Arm, and terminates at the Motor Mounting Plate (Figure 4.20). The Linkage Assembly allows for a rotation of the Motor Mounting Plate of 90°. Because the cutter motor, Adaptor Cup Assembly, and the hedge trimmer base configuration (indirectly) are fixed to this plate, they are therefore also articulated through the 90° of motion. For greater detail in describing the Linkage Assembly, see Harker's thesis [12].

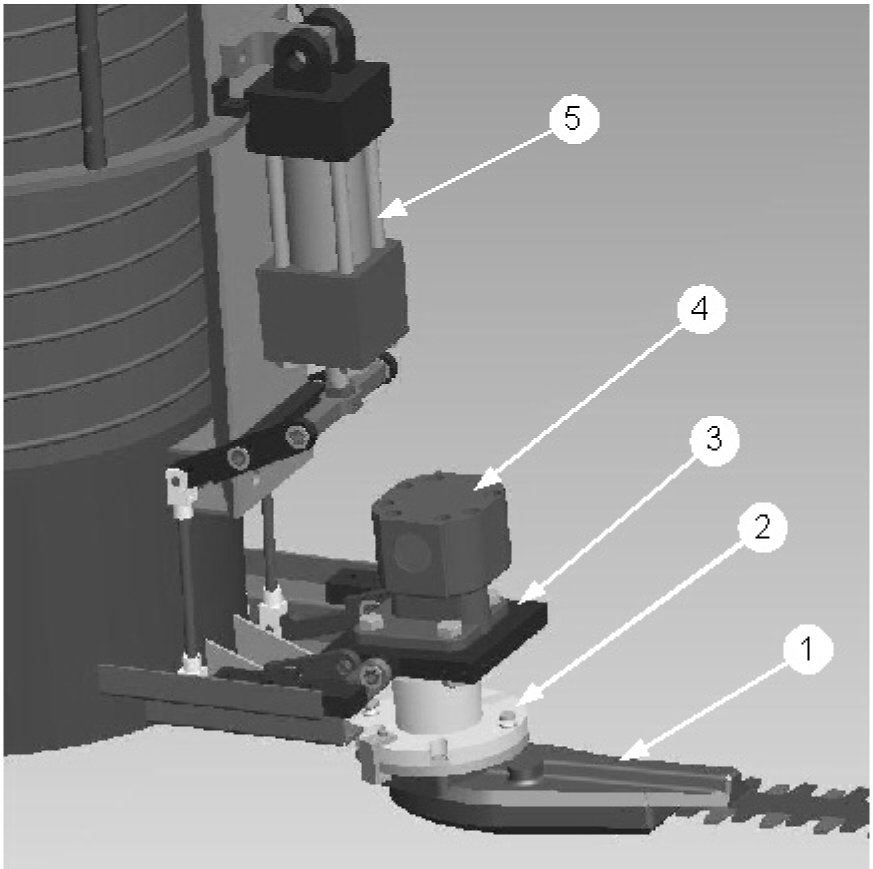


Figure 4.20 Articulated Oscillating Cutter Assembly components: 1) hedge trimmer base configuration, 2) Adaptor Cup Assembly, 3) Motor Mounting Plate, 4) cutter motor, 5) Linkage Cylinder.

The cutter motor is a Haldex GC series hydraulic motor as described in Section 4.3.3. It has a displacement of 8.47 cm3 (0.065 in3), a maximum speed of 5000 rpm, and a stall speed of 750 rpm. After hedge trimmer internal gearing of 3.7:1, the cutter motor provides for 1364 oscillations per minute of the cutting blades. The cutter motor is mounted to the top of the Motor Mount Plate.

4.4.3.1 Adaptor Cup Assembly

The Adaptor Cup Assembly (Figure 4.21) allows for power transfer from the cutter motor to the hedge trimmer base configuration allowing for motion 10 (Figure 4.5). Further, it allows for the added 90° of articulation to motion 8 as shown in Figure 4.9.

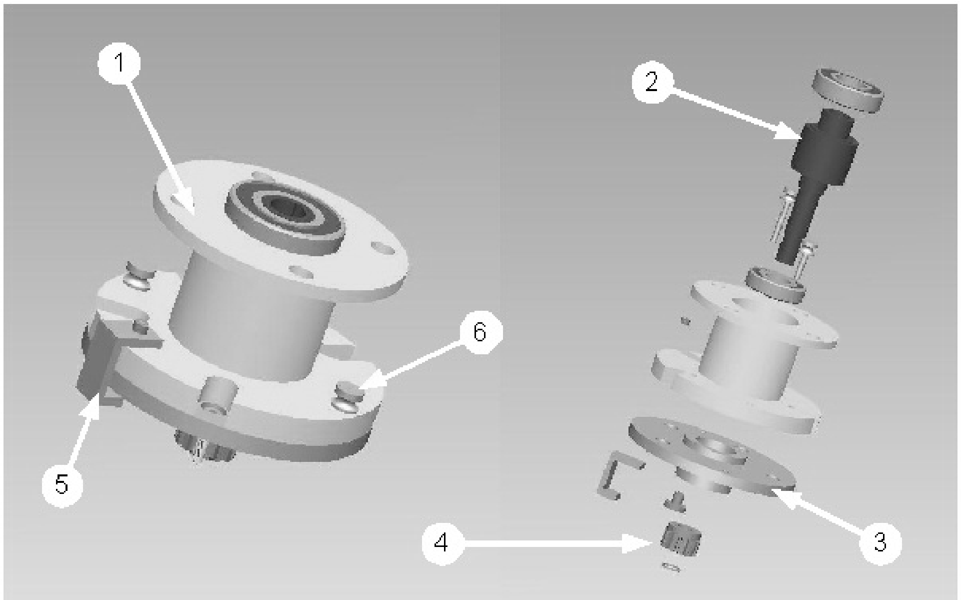


Figure 4.21 Adaptor Cup Assembly components of interest: 1) Bearing Cup, 2) Power Shaft, 3) Adaptor Plate, 4) cutter pinion, 5) HT-Bracket, 6) HT-thumb screw (X2)

4.4.3.1.1 HT-Bearing Cup

The face of the HT-Bearing Cup is mounted to the bottom of the Motor Mounting Plate, while the bottom of the Bearing Cup is mounted to the top face of the Adaptor Plate by two 6-32 thumb screws. The Bearing Cup is fabricated from Al-6061 and carries two steel ABEC-1 double sealed ball bearings, which should handle the low radial and thrust loads anticipated from vegetation cutting operations. The HT-Power Shaft is seated between these bearings. The bottom face of the Bearing Cup has four slots milled into it. These slots allow for access to the screws joining the HT-Adaptor Plate to the hedge trimmer base configuration.

4.4.3.1.2 HT-Power Shaft

The HT-Power Shaft is seated between the two ball-bearings of the HT-Bearing Cup and transfers rotary power between the cutter motor and the hedge trimmer base configuration. The upper end of the Power Shaft allows for the cutter motor shaft to be inserted inside of it and is driven by a keyway. The lower end of the shaft has the original hedge trimmer pinion gear splined to it and is retained by a c-clip. The HT-Power Shaft is fabricated from 1018-steel and its smallest diameter is 10 mm (0.39 in). This is well over designed for a maximum torque of 1.1 N-m (0.80 ft-lbf).

4.4.3.1.3 HT-Adaptor Plate

The top surface of the HT-Adaptor Plate is fastened to the bottom surface of the HT-Bearing Cup by two 6-32 thumb screws, while the bottom surface of the Adaptor Plate is fastened to the smooth mounting surface of the hedge trimmer base configuration by four low-profile M5 machine screws. The hub of the Adaptor Plate allows for a low clearance fit for proper alignment when connecting to the Bearing Cup and hedge trimmer. The Adaptor Plate is fabricated from 1018 Steel.

4.4.3.1.4 180° change in orientation for additional 90° augmentation to motion 8

The Adaptor Cup Assembly has the capability to alter the orientation of the cutter blade by 180° to allow for an additional 90° of articulation to motion 8 (Figure 4.9). This is accomplished through unfastening the thumb screws joining the HT-Bearing Cup to the HT-Adaptor Plate, then manually rotating the hedge trimmer 180°, and then reinserting the thumb screws. The purpose of the HT-Bracket (Figure 4.21), is to keep the HT-Adaptor Plate hub inserted into the Bearing cup while this manual action is performed. Without the HT-Bracket, the hedge trimmer and Adaptor Plate could become entirely separated from the rest of the Articulated Oscillating Cutter Assembly, allowing for introduction of foreign debris into the assembly while also causing unnecessary trouble in realigning the Adaptor Plate hub to the Bearing Cup during field operations. The HT-Bracket is fabricated from Al-6063 1″ X 1″ channel stock and is not intended to support a load during normal operations. It is mounted to the bottom plate of the Bearing Cup with a machine screw.

4.4.4 Design of the Rotational Oscillating Cutter Assembly

The Rotational Oscillating Cutter Assembly (Figure 4.16) includes the RHT-Hinge (Section 4.4.4.1), the cutter motor (Section 4.3.3), the RHT-Motor Mounting Plate (Figure 4.22), the hedge trimmer base configuration (Figure 4.19), and the RHT-Belt Drive Assembly, which is comprised of the sheet metal shrouding, the RHT-Rotary Adaptor Assembly (Figure 4.25), and the Rotary Motor Assembly (Figure 4.25). This assembly allows for motions 9 and 10 (Figure 4.5).

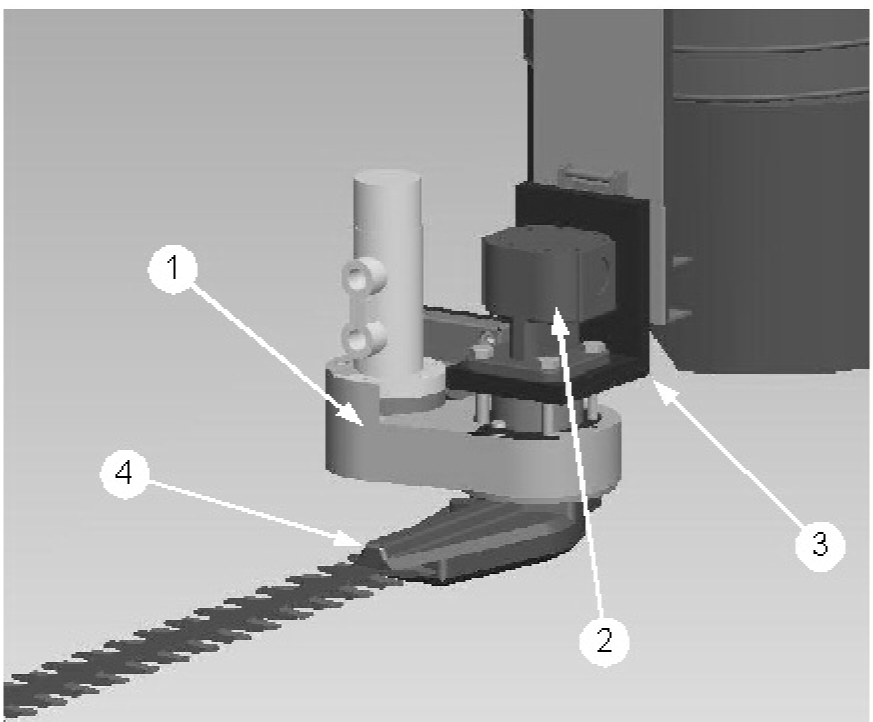


Figure 4.22 Selected Rotational Oscillating Cutter Assembly components: 1) RHT-Belt Drive Assembly (shrouding), 2) cutter motor, 3) RHT-Motor Mounting Plate, and 4) hedge trimmer base configuration

The cutter motor is a Haldex GC series hydraulic motor as described in Section 4.3.3. It has a displacement of 8.47 cm3 (0.065 in3), a maximum speed of 5000 rpm, and a stall speed of 750 rpm. After hedge trimmer internal gearing of 3.7:1, the cutter motor provides for 1364 oscillations per minute of the cutting blades. The cutter motor is mounted to the top of the RHT-Motor Mount Plate.

The RHT-Motor Mount Plate (Figure 4.22) is joined to the Nozzle Bracket Arm by the RHT-Hinge and is bolted to the cutter motor on its upper face, the RHT-Rotary Adaptor Assembly on its bottom face, and the Rotary Motor Assembly on its front face. It is fabricated from Al-6061.

4.4.4.1 RHT-Hinge

The RHT-hinge (Figure 4.23) allows for the Rotational Oscillating Cutter Assembly to be folded away when not in use (Figure 4.24). The hinge allows for 90° of rotation and locks in both the closed and opened positions. The hinge is constructed of two plates: one, which is fixed to a pin, and the other, which is free to rotate about that pin. The locking mechanism (not shown) is a spring loaded press bar with grooves placed in the press bar and the webbing affixed to one of the plates. As the spring loaded press bar is depressed, the grooves align allowing for one plate to rotate about the hinge. The grooves may only become aligned at the extreme positions, which are 90° apart.

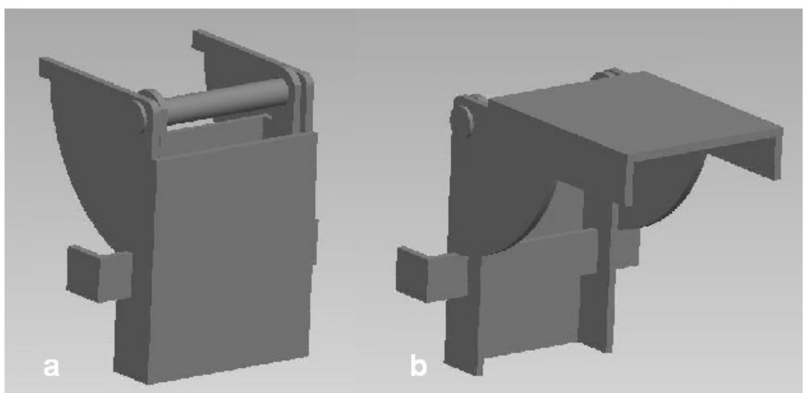


Figure 4.23 RHT-hinge close up (fastener holes not shown). a) closed, b) opened.

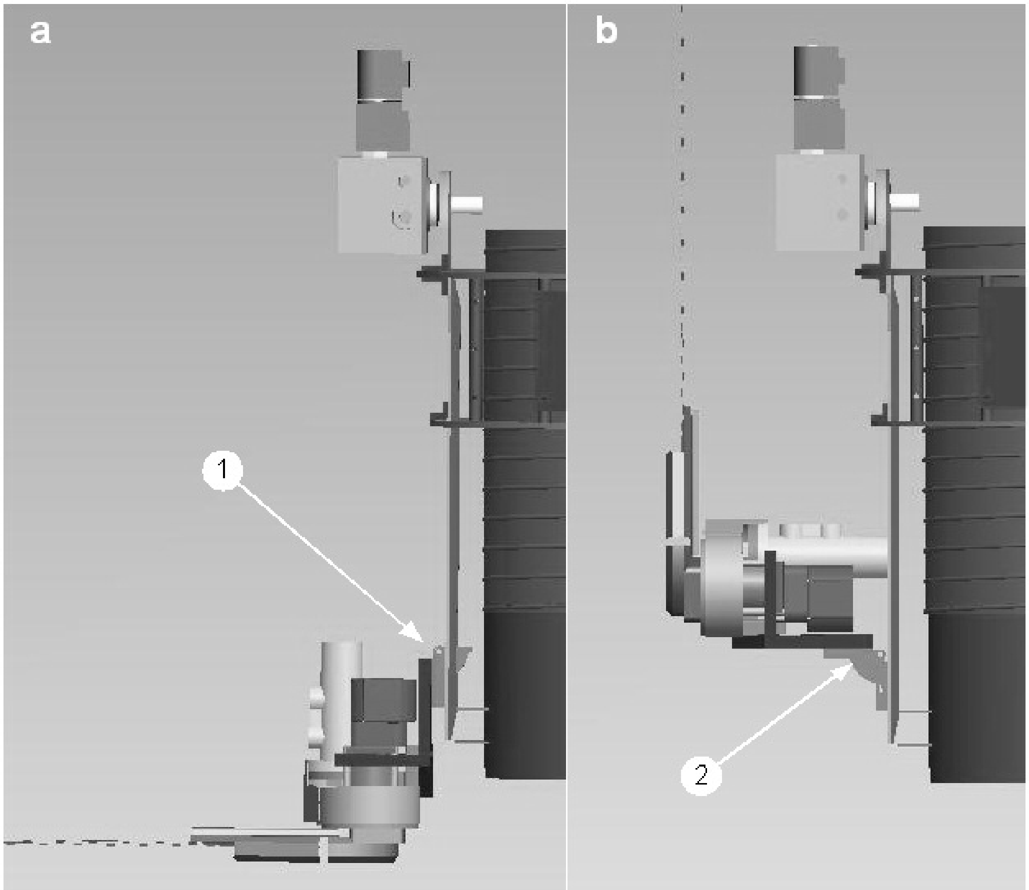


Figure 4.24 RHT-hinge operation on the Nozzle. 1) hinge closed, and 2) hinge opened.

The RHT-hinge is constructed of bronze plated steel. It was chosen from the McMaster-Carr online catalog and was attractive for use in this design because it satisfies the needs of providing a hinge joint and a locking mechanism. Also, it was very inexpensive, at approximately $3.71. However, it is difficult to know if this hinge will stand up to forces encountered while being employed in the roadway environment.

The hinge is rated to carry a static load of 333 N (75 lbf). The weight of the Rotational Oscillating Cutter Assembly, which it must support, is approximately 89 N (20 lbf). Thus, in static orientations and in normal cutting operations, the hinge will perform properly. However, if employed in the sweep motion (Figure 3.10), this hinge may not hold. Because of the irregular geometry of the hinge, it was not possible to get a reliable finite element analysis of this component.

Therefore, it is suggested that future design iterations address the problem presented by this hinge when dynamic loads are applied. One possible solution is to modify the hinge by using two hinges with one press bar. However, this may lead to modifications in the Nozzle Bracket Arm which it mounts to. Another possible solution is to design a more robust hinge with similar plate configurations, which could be locked by sliding a bolt through holes that would align only at 90° intervals. However, for proof of concept of the oscillating cutter assemblies, this hinge should be adequate.

4.4.4.2 RHT-Belt Drive Assembly

The RHT-Belt Drive Assembly includes the sheet metal shrouding, the RHT-Rotary Adaptor Assembly (Figure 4.25), and the Rotary Motor Assembly (Figure 4.25). The Rotary Motor Assembly drives a trapezoidal tooth steel timing belt pulley with 30 teeth and an outer diameter of 5.43 cm (2.14 in). This pulley is used to drive another trapezoidal tooth steel timing belt pulley with 60 teeth and an outer diameter of 9.65 cm (3.80 in). The 60 tooth pulley is part of the RHT-Rotary Adaptor Assembly, which ultimately drives motion 9 (Section 4.3.2). The pulleys are connected through a trapezoidal tooth neoprene rubber timing belt. The overall reduction in this drive is 2:1. A friction clutch (Figure 4.26) is attached to the smaller of the two pulleys to allow for slippage when the rotating cutter encounters unanticipated interference due to impact with roadside obstacles such as guardrail posts, rocks, etc. It can be adjusted to allow for a maximum load of 81 N-m (60 ft-lbf).

4.4.4.2.1 RHT-Belt Drive Assembly Shrouding

The RHT-Belt Drive Assembly shrouding is fabricated from 16 gauge sheet steel. The shrouding is comprised of three parts, which can be attached after the system is assembled. The shrouding protects the belt drive to the extent that large twigs, fingers etc. can not cause interference with the belt. Since the belt is softer that the steel, dust that enters the drive will most likely cause wear to the belt before the steel pulley. Because belts are relatively inexpensive, they may be replaced readily. This was the primary reason in choosing a belt drive for this design.

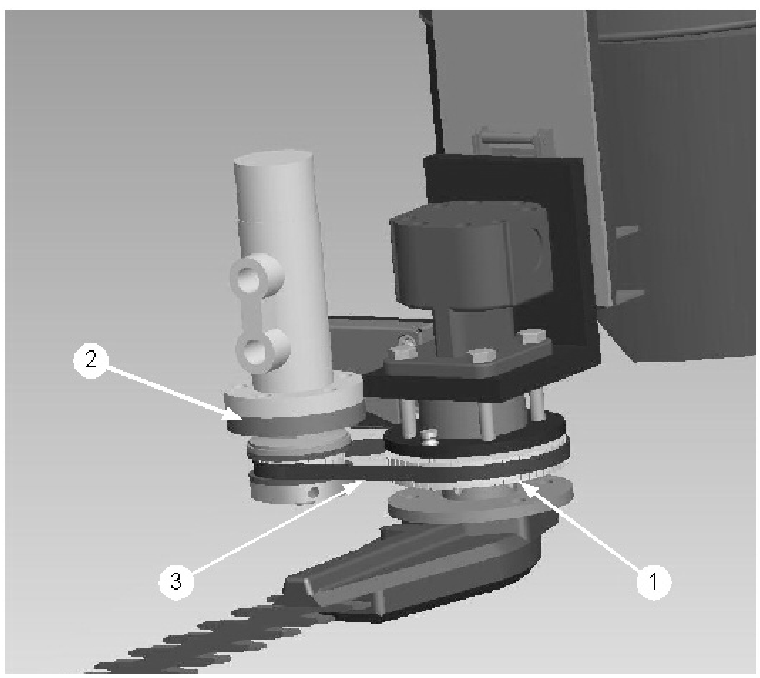


Figure 4.25 RHT-Belt Drive components with the shrouding removed: 1) RHT-Rotary Adaptor Assembly, 2) rotary motor assembly, and 3) neoprene timing belt.

4.4.4.2.2 Rotary Motor Assembly

The Rotary Motor Assembly consists of the Lamina hydraulic motor [1], the Lamina motor mount, and the friction clutch assembly (Figure 4.26). The hydraulic motor used in this design is Lamina motor A-100-FM. The Lamina motor is mounted to the Lamina motor mount with bolt fasteners, while the Lamina motor mount is attached to the front face of the RHT-Motor Mount Plate. The friction clutch assembly drives the smaller of the two timing belt gears and is in turn driven by a keyway in the Lamina motor shaft. If adjusted to maximum tightness, the friction clutch will slip at 81 N-m (60 ft-lbf). This is much higher than necessary for loads encountered in cutting.

Early errors in the design process caused a less than ideal motor to be chosen for the Rotary Motor Assembly. Lamina motor A-100-FM cannot generate high enough speed to accommodate the ideal rotary speeds desired for the optimum mowing configuration (Section 4.3.2.1.1) while limited to hydraulic design flows of 11 lpm (3 gpm). However, the fix for this flaw is relatively simple in that Lamina hydraulic motor A-37-FM will accommodate rotary speeds of up to 481 rpm at 11 lpm (3 gpm). After the RHT-Belt Drive's 2:1 reduction, this allows for 241 rpm of the rotary oscillating cutter, which meets the optimum mowing configuration.

Future design iterations of this assembly should consider elimination of the friction clutch assembly and designing the slip into the hydraulic system through the use of pressure bypass valves. The benefit of using the friction clutch assembly was to allow immediate slip in the cases of unintended interference noted above. However, design compromises were made to the system to incorporate the friction clutch, which may ultimately prove detrimental to the proper functioning of this assembly. Because the Lamina motor shaft was not long enough to support the friction clutch while seated in the Lamina motor mount, the Lamina motor mount was bored out to accommodate the friction clutch. This caused the removal of a bearing that was seated in the motor mount, which could have provided added radial support to the Lamina motor shaft. Currently, this shaft is only supported by internal motor bearings. Extra machining of the friction clutch to accommodate it to the size of the motor shaft, along with the fabrication of a specially sized wrench (necessary due to space constraints) to adjust the friction clutch, introduced extra costs to the assembly.

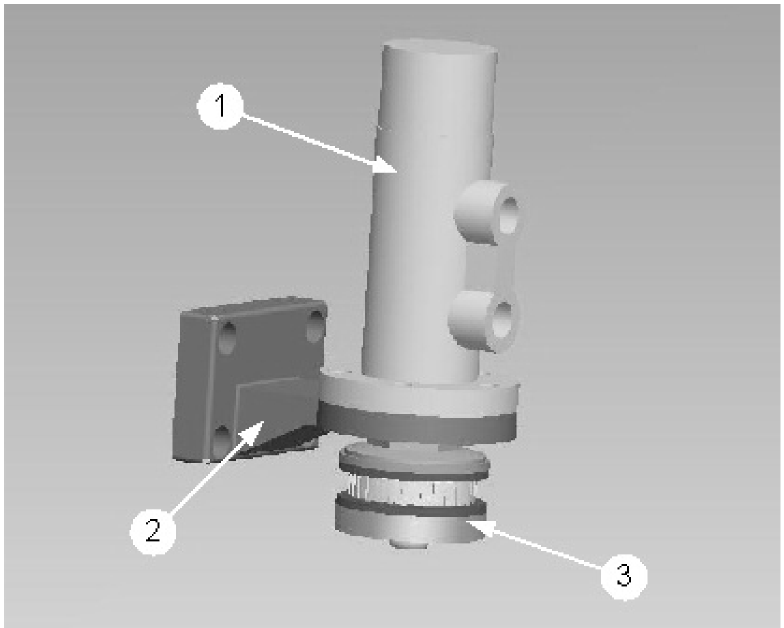


Figure 4.26 Rotary motor assembly components: 1) Lamina hydraulic motor, 2) Lamina motor mount, 3) friction clutch assembly.

4.4.4.2.3 RHT-Rotary Adaptor Assembly

The RHT-Rotary Adaptor Assembly (Figure 4.27) allows for power transfer from the cutter motor to the hedge trimmer base configuration allowing for motion 10 (Figure 4.5). Furthermore, it allows for the rotary articulation of motion 9 as shown in Figure 4.10.

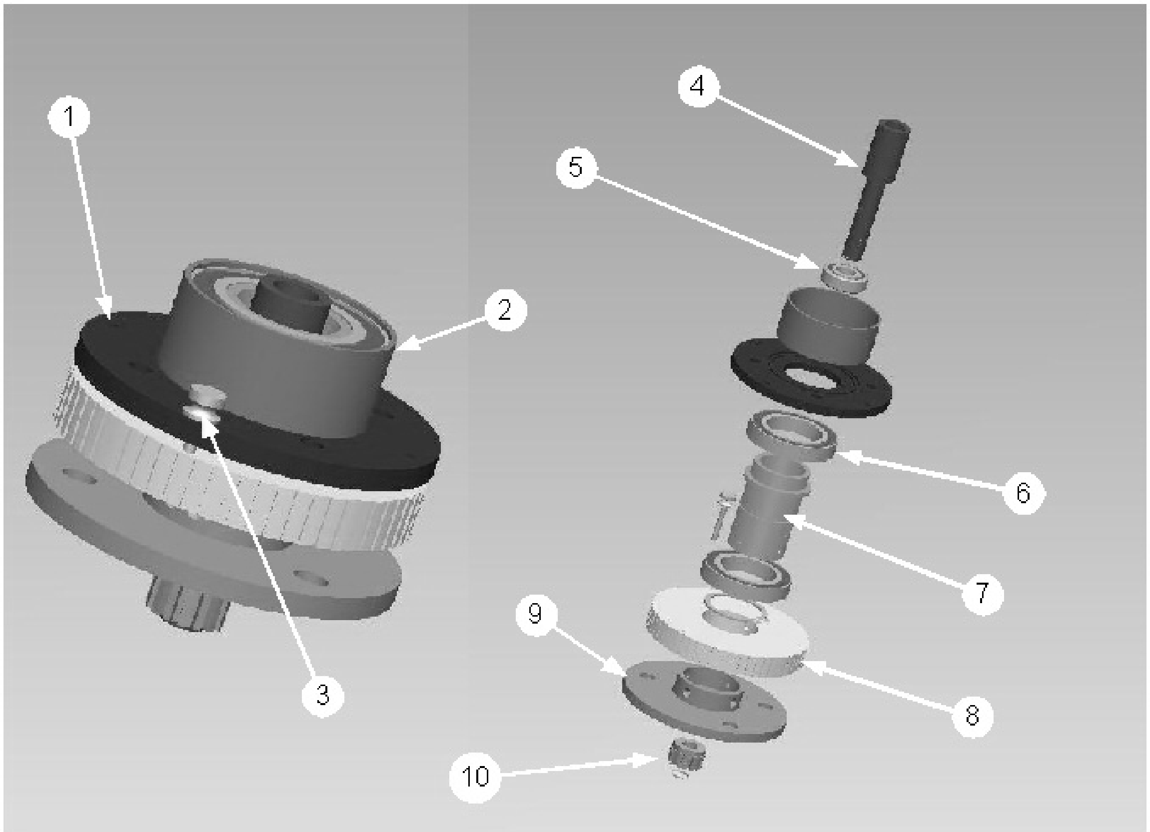


Figure 4.27 RHT-Rotary Adaptor Assembly components: 1) RHT-Bearing Plate, 2) outer bearing sleeve, 3) RHT-thumb screw, 4) RHT-Power shaft, 5) inner bearing, 6) outer bearing (X2), 7) RHT-Bearing Carrier, 8) large timing belt pulley, 9) RHT-Adaptor Plate, 10) cutter pinion.

4.4.4.2.3.1 RHT-Bearing Plate

The RHT-Bearing Plate (Figure 4.27) is joined to the RHT-Motor Mounting Plate (Figure 4.22) by four 5/16 – 18 bolts. It supports the outer bearings and the outer bearing sleeve directly, sandwiching them between itself and the RHT-Motor Mounting Plate. The RHT-Bearing Plate has a bore of 3.56 cm (1.40 in), allowing the RHT-Bearing Carrier to be inserted through it and rotate freely. The RHT-thumb screw may be inserted through a through-hole in the front of this plate to a threaded hole in the top face of the large timing belt pulley. This will allow the hedge trimmer to be fixed in one position while not in operation. This plate is fabricated from 1018 steel. Future iterations of this design should fabricate this feature from Aluminum for weight reduction.

4.4.4.2.3.2 RHT-Power Shaft

The RHT-Power Shaft is seated between the inner bearing of the RHT-Bearing Carrier and the cutter motor shaft. It transfers rotary power between the cutter motor and the hedge trimmer base configuration. The RHT-Power Shaft relies on the inner bearing and the internal cutter motor bearings for radial and thrust load support. The upper end of the Power Shaft allows for the cutter motor shaft to be inserted inside of it and is driven by a keyway. The lower end of the shaft has the original hedge trimmer pinion gear splined to it, and is retained by a c-clip. The HT-Power Shaft is fabricated from 1018-steel and its smallest diameter is 10 mm (0.39 in). This is well over designed for a maximum torque of 1.1 N-m (0.80 ft-lbf).

4.4.4.2.3.3 RHT-Bearing Carrier

The RHT-Bearing Carrier (Figure 4.28) is seated on its lip in between the outer bearings. This allows it to rotate with respect to the RHT-Bearing Plate, and thus the end-effector. It is fixed through four machine screws to the RHT-Adaptor Plate through which its lower end is inserted. The RHT-Adaptor Plate is in turn fixed to the hedge trimmer base configuration, allowing for rotary motion of the hedge trimmer. The large timing belt pulley is inserted onto the RHT-Bearing Carrier and is held in position on the shaft by a retaining ring on the top and upper ledge of the RHT-Adaptor Plate. The pulley is kept from spinning by set screws that are inserted through the pulley, contacting the flats of the RHT-Bearing Carrier surface. Internally, the inner bearing is seated within the RHT- Bearing Carrier, and the power shaft then spins within, while seated on the lower bearing. The RHT-Bearing Carrier is fabricated from 18-8 stainless steel, to avoid rust and corrosion while maintaining a smooth surface for the bearings to ride upon.

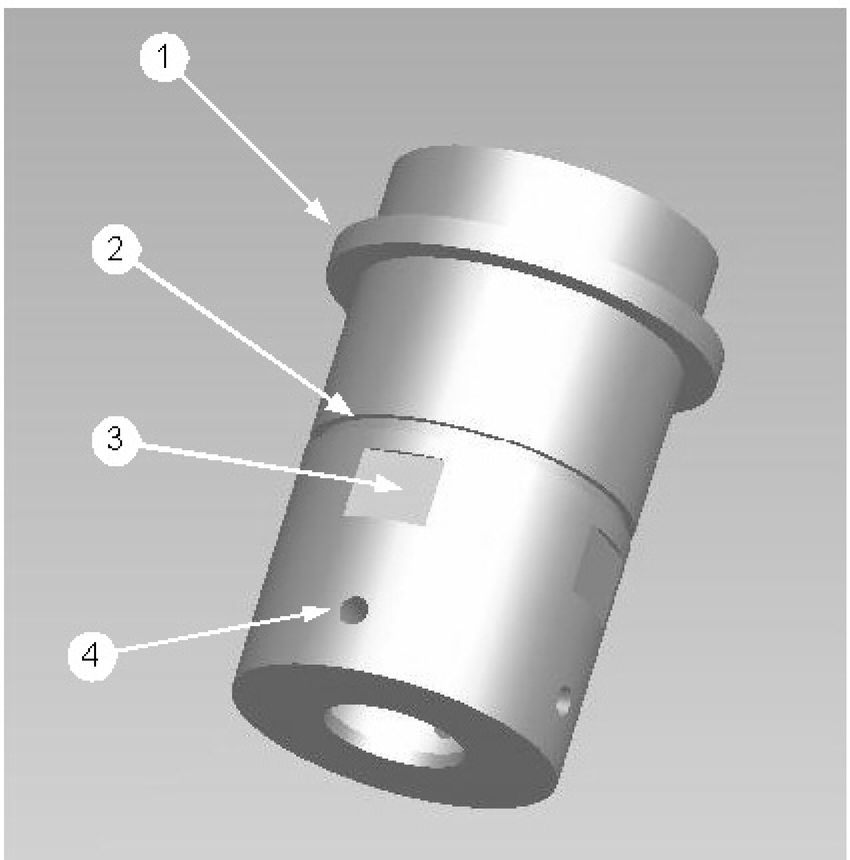


Figure 4.28 RHT-Bearing Carrier selected details: 1) BC-lip, 2) BC-snap ring groove, 3) BC-flat (X2), and 4) BC-machine screw hole (X4).

4.4.4.2.3.4 RHT-Adaptor Plate

The RHT-Adaptor Plate is fastened to the shaft of the RHT-Bearing Carrier by four machine screws, while the bottom surface of the Adaptor Plate is fastened to the smooth mounting surface of the hedge trimmer base configuration by four low-profile M5 machine screws. The RHT-Adaptor Plate is driven to rotate about its vertical central axis by its rigid connection to the RHT-Bearing Carrier. The RHT-Adaptor Plate is fabricated from 1018 Steel.

Future design iterations should consider adding a hub, as in the HT-Adaptor Plate (Section 4.4.3.1.3), to allow for more precise centering when connecting to the hedge trimmer base configuration and to allow for a location to include an additional bearing for added radial support of the RHT-Power Shaft. Furthermore, this plate should be fabricated from aluminum.

4.5 Prototype Testing

Due to delivery delays of products from vendors and machinists, it was not possible to perform road testing of the assembled prototype before the submission of this paper. However, final assembly and prototype testing should be completed by the end of January 2005.

4.6 Summary

This chapter has illustrated the design process of the oscillating cutter system, its capabilities, and how customer needs were satisfied. Furthermore, future modifications are suggested to improve upon the prototype. The oscillating cutter (hedge trimmer) ranked highest against all other cutter mechanism options when considering power, weight and cost together. Furthermore, the hedge trimmer also proves to be the safest of the cutting mechanisms considered when used in the highway environment, as peak blade speeds for the hedge trimmer are approximately 3.0 m/s (9.9 ft/s) versus the peak blade tip speed of the rotary cutter of 144 m/s (471 ft/s). Thus, the hedge trimmer is less likely to launch projectiles into traffic. The oscillating cutter system is capable of mowing a swath of 2.1 m (7 ft) at vehicle speeds of approximately 2 mph. Also, it is able to articulate to cut smaller strips on both inclines and declines. Cutters may alternatively be folded away to allow for normal debris vacuuming operations. In conjunction with the ARDVAC's vacuum, these operations satisfy Caltrans' need of removing duff (cuttings) generated during the mowing process, decreasing fire risks. Further Caltrans needs are satisfied such that cutting may be performed along inclines and culverts along roadways, and exposed workers will be removed from the hazardous areas of the freeway environment.

Future modifications should include: a) fabricating the RHT-Adaptor Plate, HT-Adaptor Plate, RHT-Bearing Plate and the belt pulleys from Aluminum for weight considerations, b) addition of an alignment hub to the RHT-Adapter Plate for more precise alignments of gears, c) elimination of the friction clutch in favor of a hydraulic pressure bypass valve on the Rotary Motor Assembly for better bearing support and cost reduction, d) switching from the Lamina A-100-FM motor to the A-37-FM motor on the Rotary Motor Assembly to allow for optimum mowing profiles, and e) redesign of the RHT-Hinge to allow for greater loading on the joint.

CHAPTER 5 ROTARY IMPACT CUTTING FIXTURE

5.1 Chapter Overview

In Chapter 1 three concepts are presented as possible cutting fixtures for a vegetation processing device adaptable to the ARDVAC system. These include the Rotary Impact Cutting Fixture, the Hedge Trimmer Cutting Fixture, and the Tumbleweed Shredder. In this chapter the detailed design of the Rotary Impact Cutting Fixture is presented.

5.2 Three Cutter Array Concept

Latham, as summarized in Chapter 1, conducted some preliminary lab experiments of the rotary impact cutting idea and developed a cutter array concept using CAD but did not continue the design beyond the concept stage. His concept uses a cutting array consisting of three independent cutting heads. Each cutting head consists of three free-swinging steel blades attached to a rotating body. Hydraulic motors power the cutting heads that are limited to cutting in the horizontal plane. For laboratory testing, Latham takes this concept and simplifies it to a single fixed cutter, which moves in a straight line parallel to the laboratory floor. The results from these laboratory tests established a minimum rotational speed of approximately 2000 rpm for clean shearing of dry vegetation.

5.3 Detail Design of Final Assembly

The Rotary Impact Cutting Fixture final assembly is displayed in Figure 5.1. The three-cutter array concept developed by Latham is inherent in the design. The major sub-assemblies are labeled and include the upper attachment plates, the hydraulic mounting cage, the nozzle weldment, both the stationary and articulating cutting heads, and the debris shield.

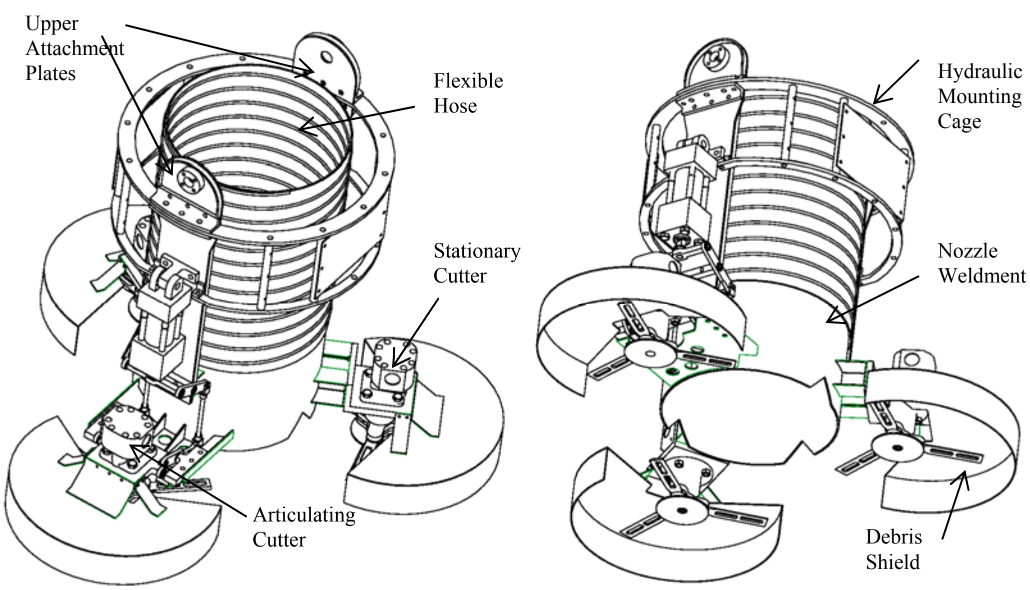


Figure 5.1 Rotary Impact Cutting Fixture Final Assembly

The assembly consists of components made from both mild and stainless steels, as well as aluminum. Aluminum is used wherever possible to keep the overall weight to a minimum. The sub-assemblies are subsequently explained in further detail.

5.3.1 Mechanical Linkage

The articulating cutter labeled in Figure 5.1, discussed in more detail at the end of this chapter, obtains its motion through a mechanical linkage. Figure 5.2 displays a detailed side view of the linkage with kinematical vector loops drawn.

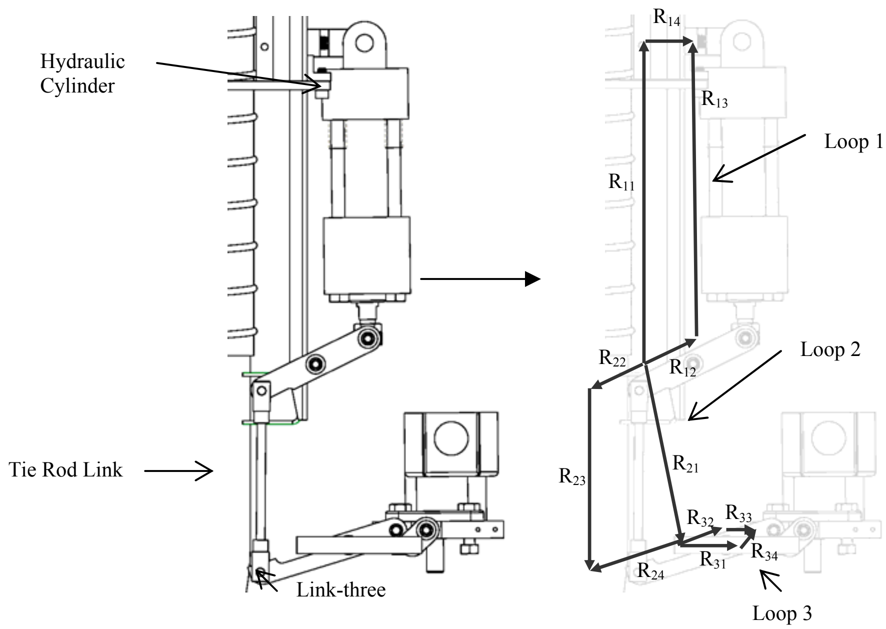


Figure 5.2 Mechanical Linkage and Corresponding Kinematical Vector Loops

Loop 1 constitutes a slider-crank mechanism with the input being applied at the slider or the hydraulic cylinder in this case. Loop 2 is a traditional four-bar linkage, which couples the input from the slider-crank to the rotation of the articulating cutter. This rotation is described by Loop 3, another four-bar.

5.3.1.1 Driving Design Parameters

The design of the mechanical linkage initially starts with loop 3. The specification governing the design is that the cutter head should be able to cut both in the horizontal and vertical orientation. In addition, for simplicity and adaptability to the current ARDVAC, as stated by Richardson [23], the input should be actuated with a hydraulic cylinder.

A simple device, that most people are familiar with, that fills these specifications is a metal folding chair that can be found in many school classrooms. The seat of the chair moves from being horizontal to vertical, when being folded, and when observed in action, the input is often at the crossbar, at the bottom of the chair's legs. An individual often applies pressure at this location with his/her foot and subsequently the linkage folds the chair. It generally requires very little input to move the seat through its full motion and is accomplished with nearly straight line input.

Another very important design issue is the accommodation of hydraulic hoses. This design issue is responsible for the orientation of loop 2. As the articulating cutter is rotated into the vertical cutting position, see Figure 5.3 below, the hydraulic motor swings into the nozzle. The circle on the hydraulic motor represents where the hydraulic hose extends out from the motor. In designing the linkage, the avoidance of these hoses, as the cutting head is rotated, is critical for the linkage to operate correctly.

5.3.1.2 Linkage Design

With the concept generated, the next stage is to use Working Model, a 2-D kinematical design software package. Working Model offers a quick way to model 2-D linkages and iterate with designs. Finally, after generating a working linkage, 3-D CAD is used to generate the final design. Pro Engineer is used to finalize the model and has a useful application called "Mechanism" that allows the user to simulate the motion of a linkage by simply clicking and dragging on components.

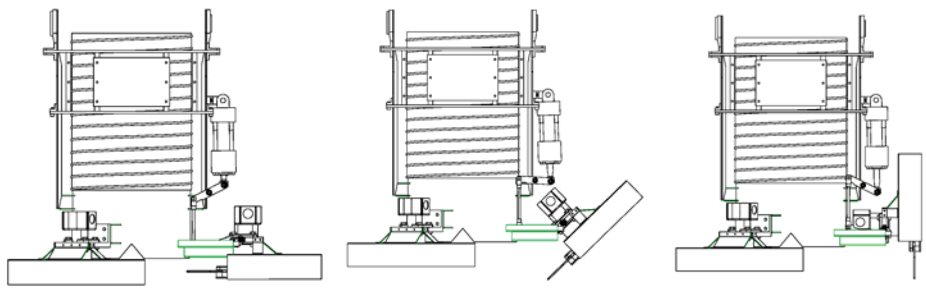


Figure 5.3 Articulating Cutter Motion

Figure 5.3 was generated using "Mechanism". This feature of Pro Engineer, makes it very easy to determine the approximate input required, kinematically speaking, to move the linkage. It is also very useful in obtaining a real-time sense of how well the linkage is providing the desired motion.

5.3.1.3 Linkage Analysis and Motion Verification

The amount of displacement required to move the articulating cutter from horizontal to vertical is found by simulating the movement of the linkage using "Mechanism", see Figure 5.3, and then verifying this value with a kinematical analysis. Using "Mechanism", full range of motion is achieved with an actuator displacement of 39.7 mm (1.56 in). Closing the vector loops shown in Figure 5.2 results in

$$\vec{R}_{31} + \vec{R}_{34} = \vec{R}_{32} + \vec{R}_{33} \text{ (Loop 3)} \tag{5.1}$$

$$\vec{R}_{21} + \vec{R}_{24} = \vec{R}_{22} + \vec{R}_{23} \text{ (Loop 2)} \tag{5.2}$$

$$\vec{R}_{11} + \vec{R}_{14} = \vec{R}_{12} + \vec{R}_{13} \text{ (Loop 1)} \tag{5.3}$$

Rewriting Equations (5.1), (5.2), and (5.3) in complex number format gives

$$r_{31}e^{i\theta_{31}} + r_{34}e^{i\theta_{34}} = r_{32}e^{i\theta_{32}} + r_{33}e^{i\theta_{33}} \tag{5.4}$$

$$r_{21}e^{i\theta_{21}} + r_{24}e^{i\theta_{24}} = r_{22}e^{i\theta_{22}} + r_{23}e^{i\theta_{23}} \tag{5.5}$$

$$r_{11}e^{i\theta_{11}} + r_{14}e^{i\theta_{14}} = r_{12}e^{i\theta_{12}} + r_{13}e^{i\theta_{13}} \tag{5.6}$$

where the θ_{ji}'s are all measured from the horizontal plane in a counter-clockwise (positive right-hand) direction.

Equations (5.4), (5.5), and (5.6) can each be written as two equations where one equation is the real component and the other is the imaginary component. This gives

$$r_{31} \cos\theta_{31} + r_{34} \cos\theta_{34} = r_{32} \cos\theta_{32} + r_{33} \cos\theta_{33} \quad (5.7)$$
$$r_{31} \sin\theta_{31} + r_{34} \sin\theta_{34} = r_{32} \sin\theta_{32} + r_{33} \sin\theta_{33} \quad (5.8)$$
$$r_{21} \cos\theta_{21} + r_{24} \cos\theta_{24} = r_{22} \cos\theta_{22} + r_{23} \cos\theta_{23} \quad (5.9)$$
$$r_{21} \sin\theta_{21} + r_{24} \sin\theta_{24} = r_{22} \sin\theta_{22} + r_{23} \sin\theta_{23} \quad (5.10)$$
$$r_{11} \cos\theta_{11} + r_{14} \cos\theta_{14} = r_{12} \cos\theta_{12} + r_{13} \cos\theta_{13} \quad (5.11)$$
$$r_{11} \sin\theta_{11} + r_{14} \sin\theta_{14} = r_{12} \sin\theta_{12} + r_{13} \sin\theta_{13} \quad (5.12)$$

where ***i*** has been divided out of the imaginary components.

Equation (5.7) through Equation (5.12) are six non-linear equations with eight unknowns. The unknowns are θ_{12}, θ_{13}, θ_{22}, θ_{23}, θ_{24}, θ_{32}, θ_{34}, and r_{13}. Two more equations are needed to solve for all of the unknowns and are given by

$$\theta_{12} = \theta_{22} - 180 \quad (5.14)$$
$$\theta_{24} = \theta_{32} + 180. \quad (5.15)$$

Using the desired angle of rotation for the articulating motor plate (θ_{33}) as the input, these eight unknowns can be solved for, where r_{13} is the total length of the cylinder. Rotating θ_{33} from 0 deg to 90 deg will define how much displacement is required from the cylinder.

Excel is used to solve this system of non-linear equations. From the solution the following plot (Figure 5.4), of cylinder displacement versus rotation of the articulating motor plate, can be made.

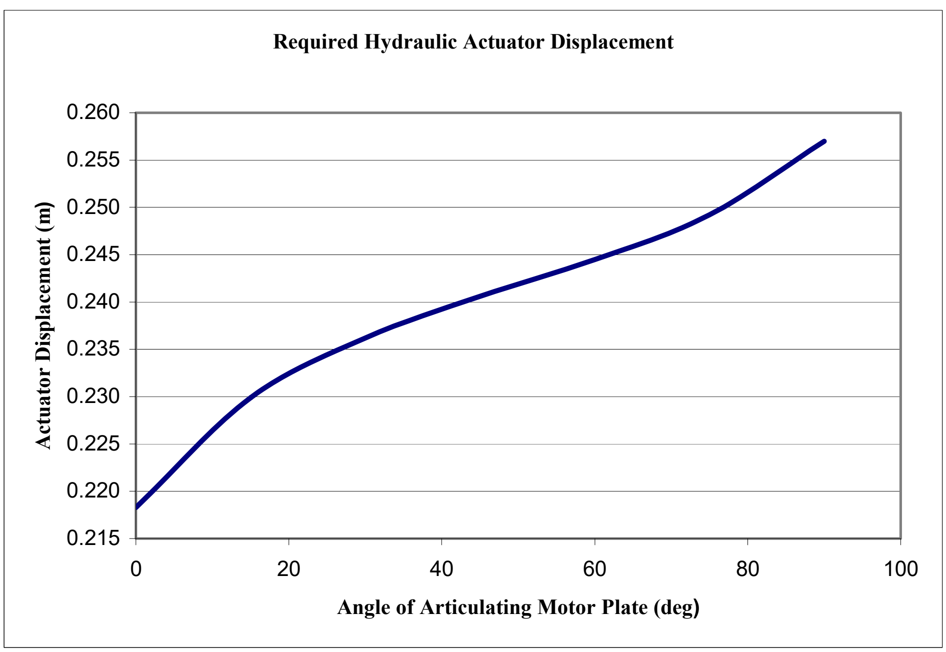


Figure 5.4 Required Actuator Displacement for Full Motor Plate Articulation

The required actuator displacement, from the kinematic analysis, is 38.7 mm (1.52 in), which is approximately 1 mm (0.04 in) different than what is measured using CAD. There precision associated with taking a measurement in the CAD software is relatively low because clicking and dragging on components and "visually" placing them leads to a different value from one measurement to the next.

There is mechanical advantage inherent in the linkage and can be calculated by using the fact that the rate of energy is conserved in the system or

$$\dot{E}_{in} = \dot{E}_{out} . \tag{5.15}$$

The rate of energy, or power, can be calculated from the product of torque and angular velocity. Equation (5.15) then becomes

$$T_{out}\dot{\theta}_{out} = T_{in}\dot{\theta}_{in} . \tag{5.16}$$

The mechanical advantage, as seen in Equation (5.17), can then be found by taking the ratio of the output torque to the input torque or the ratio of the input angular velocity to the output angular velocity.

$$MA = \frac{T_{out}}{T_{in}} = \frac{\dot{\theta}_{in}}{\dot{\theta}_{out}} . \tag{5.17}$$

In Equation (5.17), for this analysis, $\dot{\theta}_{in}$ is equal to $\dot{\theta}_{12}$ and $\dot{\theta}_{out}$ is equal to $\dot{\theta}_{33}$. $\dot{\theta}_{33}$ is assumed to be a constant 15 deg/s.

The angular velocities required by Equation (5.17) require differentiating the kinematic position displacement equations, see Equations (5.7) through (5.12). Differentiating gives

$$r_{34}\dot{\theta}_{34}\sin\theta_{34} - r_{32}\dot{\theta}_{32}\sin\theta_{32} = r_{33}\dot{\theta}_{33}\sin\theta_{33} \tag{5.18}$$

$$r_{34}\dot{\theta}_{34}\cos\theta_{34} - r_{32}\dot{\theta}_{32}\cos\theta_{32} = r_{33}\dot{\theta}_{33}\cos\theta_{33} \tag{5.19}$$

$$r_{24}\dot{\theta}_{24}\sin\theta_{24} = r_{22}\dot{\theta}_{22}\sin\theta_{22} + r_{23}\dot{\theta}_{23}\sin\theta_{23} \tag{5.20}$$

$$r_{24}\dot{\theta}_{24}\cos\theta_{24} = r_{22}\dot{\theta}_{22}\cos\theta_{22} + r_{23}\dot{\theta}_{23}\cos\theta_{23} \tag{5.21}$$

$$0 = r_{12}\dot{\theta}_{12}\sin\theta_{12} + \dot{r}_{13}\cos\theta_{13} - r_{13}\dot{\theta}_{13}\sin\theta_{13} \tag{5.22}$$

$$0 = r_{12}\dot{\theta}_{12}\cos\theta_{12} + \dot{r}_{13}\sin\theta_{13} + r_{13}\dot{\theta}_{13}\cos\theta_{13} \tag{5.23}$$

where, from Equations (5.14) and (5.15),

$$\dot{\theta}_{12} = \dot{\theta}_{22} \tag{5.24}$$

$$\dot{\theta}_{24} = \dot{\theta}_{32} . \tag{5.25}$$

Excel is used to solve this system of linear equations. They are linear because the angular displacements were calculated previously in the displacement analysis.

Figure 5.5 plots the mechanical advantage of the linkage versus cylinder displacement. As can be seen, the maximum mechanical advantage is approximately 1.5.

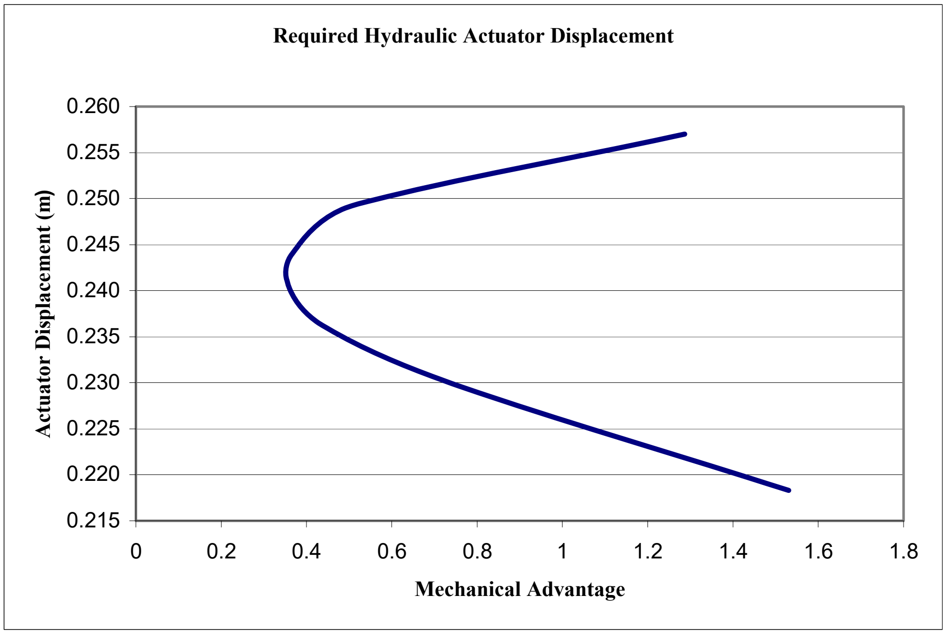


Figure 5.5 Mechanical Advantage for Full Actuator Input Displacement

The important information gathered from this plot is that the mechanical advantage never goes to zero. This ensures, throughout the full actuator or cylinder displacement, the linkage does not enter an undesirable change point. In addition, at the fully retracted and fully displaced positions, the mechanical advantage is greater than one.

5.3.2 Upper Attachment Plates

Figure 5.6 shows the upper attachment plates in more detail. These plates can stay on the ARDVAC nozzle permanently and serve as the attachment point for multiple cutting fixtures including the Rotary Impact Cutting Fixture.

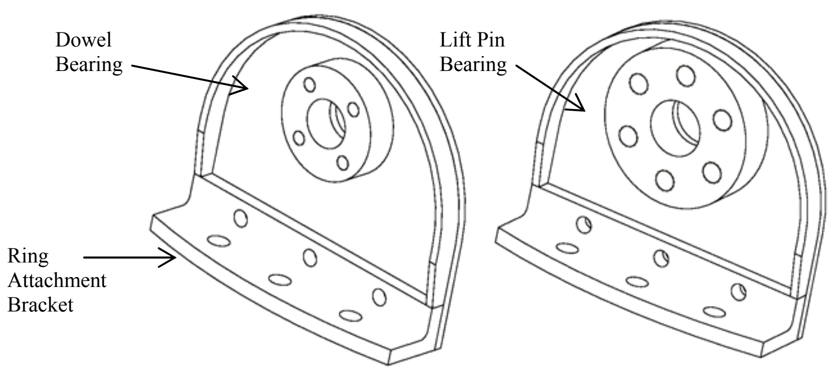


Figure 5.6 Upper Attachment Plate Sub-assemblies

The plate sub-assemblies use the same dowel bearing and lift pin bearing from the original design, and thus connection of the current ARDVAC is assured. The bearings are welded to the main plate of the sub-assembly. The components of the upper attachment plate sub-assemblies are made from mild steel.

The ring attachment bracket is the specific attachment location for the cutting fixture. Any cutting fixture assembly can attach to the ARDVAC if it is designed to mount to this location. In addition to this requirement, the cutting fixture needs clearance for the outer diameter of the fly weldment and the flexible hose. The lift pin bearing serves as the location for where the cylinder flange attaches. The cylinder flange provides the attachment location for the ARDVAC's long cylinders, which transfer motion to the ARDVAC's articulating nozzle and consequently the attached cutting fixture.

5.3.3 Hydraulic Mounting Cage

The hydraulic mounting cage serves as the connection between the upper attachment plate sub-assemblies, discussed above, and the nozzle weldment, where the cutting heads are attached. It also provides a location to mount the hydraulic cylinder for actuating the articulating cutter. Finally, it serves as a possible mounting location for all of the lower hydraulic plumbing. Figure 5.7 shows the hydraulic mounting cage sub-assembly with important components labeled.

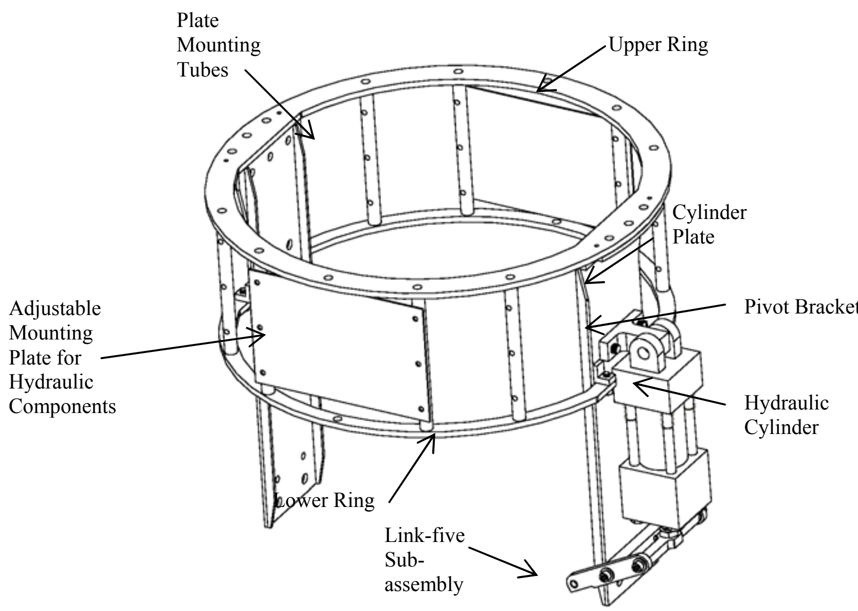


Figure 5.7 Hydraulic Mounting Cage Sub-assembly

The upper ring attaches to the ring attachment brackets found on the upper plates, as seen in Figure 5.6. Both the lower and upper rings are constructed from aluminum.

Until the cutting fixture is attached to the ARDVAC, the exact mounting scheme of the hydraulic manifolds, control valves, and hose routing is unknown. Because of the extra weight of these items, it is desirable to mount them on the main ARDVAC unit and keep them off of the cutting fixture completely. If this proves to be possible, the adjustable mounting plate, the plate mounting tubes, and the lower ring can all be removed. Nonetheless, these components have been included to provide additional mounting options for the hydraulic components.

If the above mentioned components are needed to mount hydraulic components, they are designed to be flexible with respect to the available mounting options. This is done by designing the hydraulic component mounting plate to be adjustable with respect to the mounting cage. Each side has three different mounting positions allowing greater flexibility than a simple stationary mounting scheme. In addition, the mounting plate is blank with respect to mounting holes and thus is easily adaptable to the hydraulic components chosen. The mounting plate is made from a mild steel, and the plate mounting tubes are made from aluminum.

5.3.3.1 Link-five Sub-assembly

The link-five sub-assembly in Figure 5.8 is an important component in the hydraulic mounting cage sub-assembly. Link-five and the hydraulic cylinder are the chief components in the slider-crank portion of the mechanical linkage or loop 1, see Figure 5.2. The slider crank mechanism is the input for the middle four-bar linkage or loop 2, which provides articulation in one of the cutting heads, see Figure 5.3.

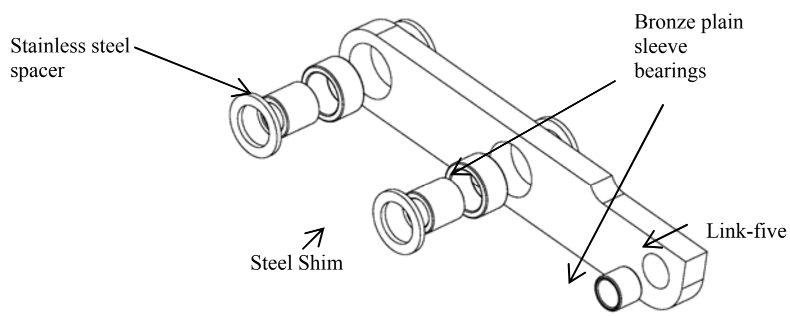


Figure 5.8 Link-five Sub-assembly

The bronze plain sleeve bearings are a standard bearing and the stainless steel spacer is a custom manufactured part. The bronze bearings are press fit into link-five, which is made from aluminum, while the spacer spins freely and allows the desired rotation of the linkage. The link is secured via the spacer with a standard ¼ - 20 UNC socket head cap bolt. The steel shim is included to inhibit lateral movement of the link, thus reducing the stress in the part, which will be discussed in greater detail later.

5.3.3.2 Hydraulic Cylinder

The hydraulic cylinder chosen to move the articulating cutter is a Vickers TZ10 clevis mount cylinder. This cylinder can operate at pressures up to 207 bars (3000 psi) where it generates about 22,000 N (5000 lbf). To bridle the relatively large force generated from this cylinder, the displacement is chosen to match the required articulation displacement, calculated earlier from the linkage analysis, see 5.3.1.3. This also ensures the cylinder's full range is utilized before the rotational limits of the linkage are met, thus preventing the possibility of breaking expensive linkage components.

Displacements are available in 1.59 mm (0.06in) increments. The cylinder displacement calculated earlier is 38.7 mm (1.52 in), thus a cylinder with a displacement of 39.7 mm (1.56 in or 1 - 9/16 in) is chosen.

5.3.4 Nozzle Weldment

Figure 5.9 shows the nozzle weldment sub-assembly. It serves as the tip of the ARDVAC nozzle as well as the foundation for all three cutting heads.

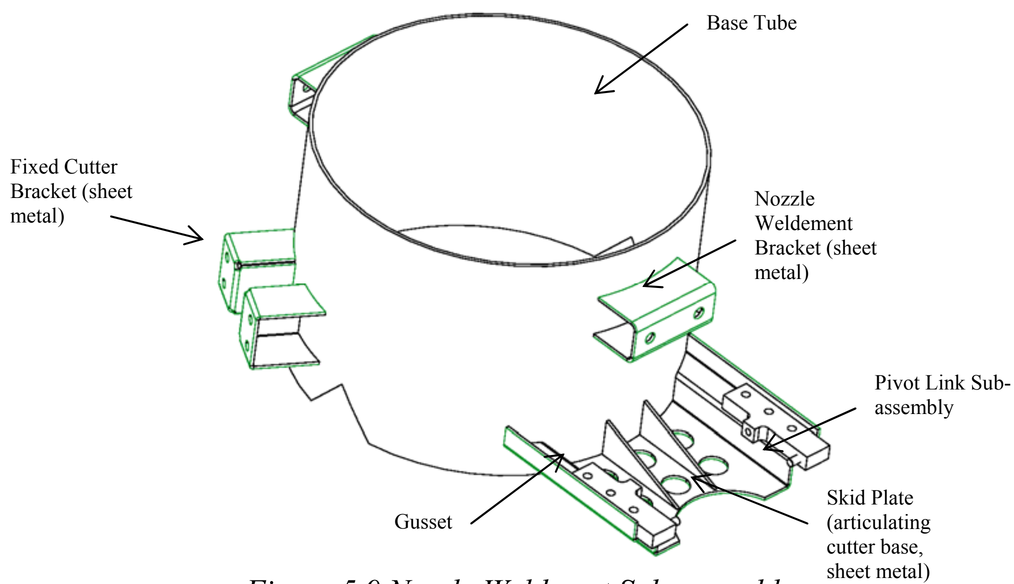


Figure 5.9 Nozzle Weldment Sub-assembly

The nozzle weldment has the same base tube as the original ARDVAC but instead of using bearing flanges for attachment to the fly weldment, a bracket is welded to the tube to make a more solid connection. The original ARDVAC design warranted a certain amount of compliance (the compliance coming from the flexible hose) at the base tube, but with the cutting fixtures attached, a more rigid connection is desired. These brackets are the nozzle weldment brackets, labeled in Figure 5.9, and provide a connection point for the hydraulic mounting cage.

Each of the sheet metal components in Figure 5.9 are welded to the base tube and are made from a mild steel. The fixed cutter brackets are used to attach the fixed cutters to the nozzle weldment. The skid plate is the base for the mechanical linkage discussed earlier. To make the skid plate more robust, two gussets are welded at the skid plate/base tube interface, see 6.2.1.

5.3.4.1 Pivot Link Sub-assembly

Bolted to the skid plate is the pivot-link sub-assembly which serves as the ground link for loop 3 of the mechanical linkage for the articulating cutter. Figure 5.10 shows this sub-assembly in more detail.

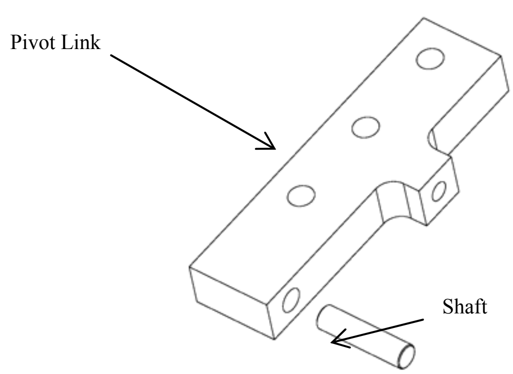


Figure 5.10 Pivot Link Sub-assembly

The pivot link component is made out of aluminum, and the shaft is from Inch Drive Components and is made out of stainless steel. The shaft is press fit into the pivot link. The protruding flange provides an adjacent mounting face for link-three. It has a tapped hole accommodating a ¼ - 20 UNC socket head cap bolt.

5.3.5 Debris Shield

The debris shield, shown in Figure 5.11, is to protect traffic and pedestrians from flying debris the cutting head might aggravate. The shield is made of mild sheet steel and is welded together.

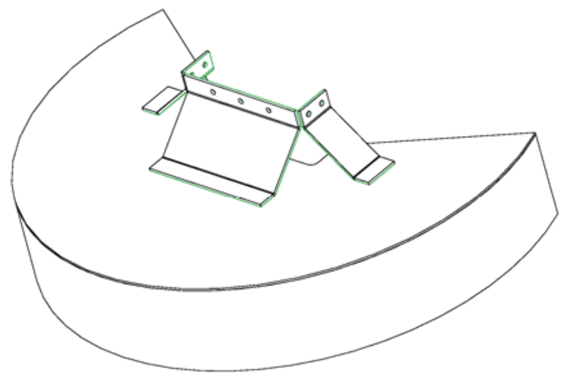


Figure 5.11 Debris Shield

Preliminary field testing of high speed rotary string trimmers proved that debris can be launched alarmingly far when struck by a high speed rotational cutting device. Even more so, using metal cutting blades, could magnify the problem. A protective shield, regardless of its impact on performance, is necessary.

5.3.6 Stationary Cutter

Figure 5.12 shows the stationary cutter sub-assembly with major components labeled. The debris shield has been removed for better viewing.

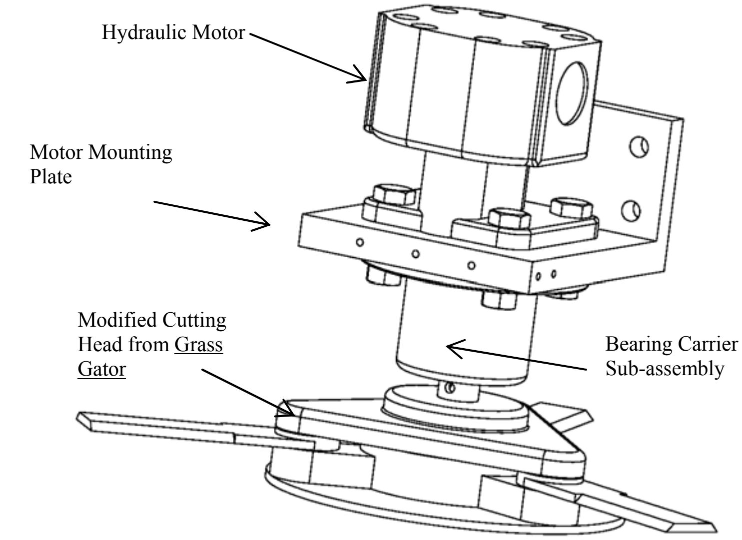


Figure 5.12 Stationary Cutter Sub-assembly

The motor used to drive the stationary cutter is a Haldex GC series hydraulic gear motor. It has a displacement of 1 cc (0.065 cu in) and is capable of 4000 rpm. It can operate at pressures up to 138 bars (2000 psi) where it produces about 15.8 N-m (140 lbf-in) of torque. It is chosen because it is very small yet still meets and exceeds the requirements for speed and torque.

Latham chose the Grass Gator cutting head shown in Figure 5.12 because he found it to be a viable option versus a standard string trimmer. The bladed cutting head design generally comes with plastic cutting blades, but these were found to be destroyed quickly when cutting heavy stalks. Consequently, steel blades were chosen to replace the plastic ones, see 1.3. The motor mounting plate is an aluminum plate and serves as the foundation to which all components are attached including the debris shield.

5.3.6.1 Bearing Carrier Sub-assembly

The bearing carrier sub-assembly shown in Figure 5.13 houses the shaft coupler and the radial ball bearings used to transfer the motor's input to the cutting head.

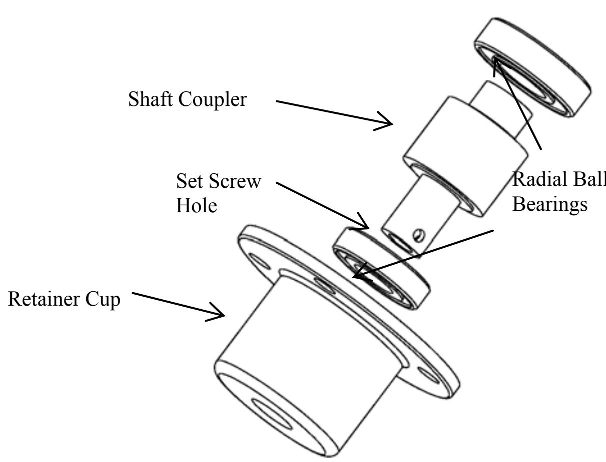


Figure 5.13 Bearing Carrier Sub-assembly

The shaft coupler has an internal keyway in one end to mate to the Haldex hydraulic motor and the other end has internal 3/8 – 16 UNC threads. The threads allow the attachment of the cutting head by means of a simple securing bolt. Because a simple bolt is used for attachment, a variety of cutting heads can be used such as a standard string trimmer or a brush cutter blade.

The set screw holes (one hole is hidden) are designed to restrain the securing bolt used to attach the cutting head to the sub-assembly. Generally trimmers do not have this feature, but it provides a safety feature in the event the power source seizes. Two small set screws clamp onto the securing bolt and restrain it from spinning loose. If the cutting head is spinning at a high rate and the motor seizes, the rotational inertia of the cutting head, exaggerated with metal blades, could unthread the bolt and the cutting head could come off. The set screws prevent this from happening.

5.3.7 Articulating Cutter

Figure 5.14 shows the articulating cutter sub-assembly.

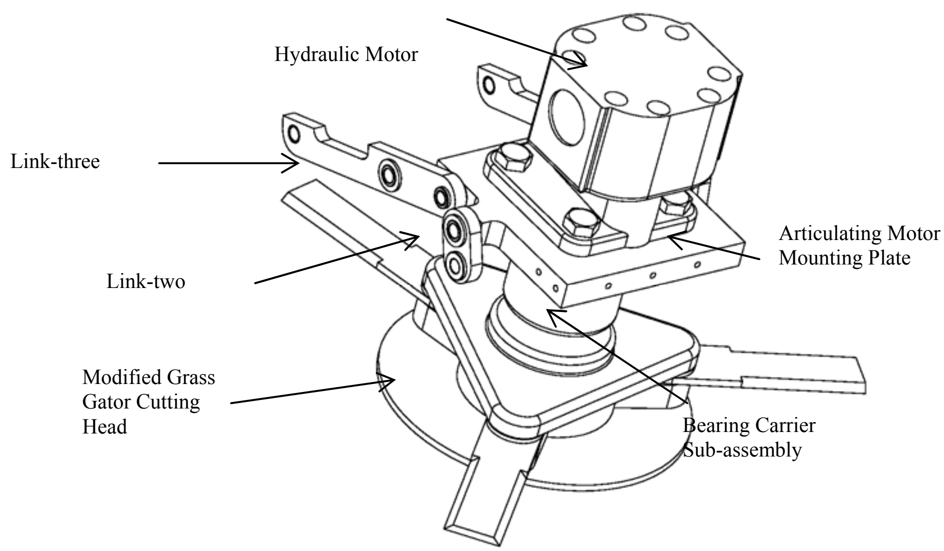


Figure 5.14 Articulating Cutter Sub-assembly

It utilizes the bearing carrier sub-assembly from the stationary cutter shown in Figure 5.12 and Figure 5.13. The articulating cutting head is the same as the stationary cutter but again could be substituted with a number of different rotary cutting devices. It is driven by the same hydraulic motor as the stationary cutter and uses a similar motor mounting plate. It also accommodates the same debris shield as shown in Figure 5.11.

5.3.7.1 Articulating Motor Plate Sub-assembly

Figure 5.15 shows the articulating motor plate. As can be seen in Figure 5.2, it is a component of kinematical loop 3. Similar to the stationary cutter, it serves as the connection base for the debris shield and the bearing carrier assembly.

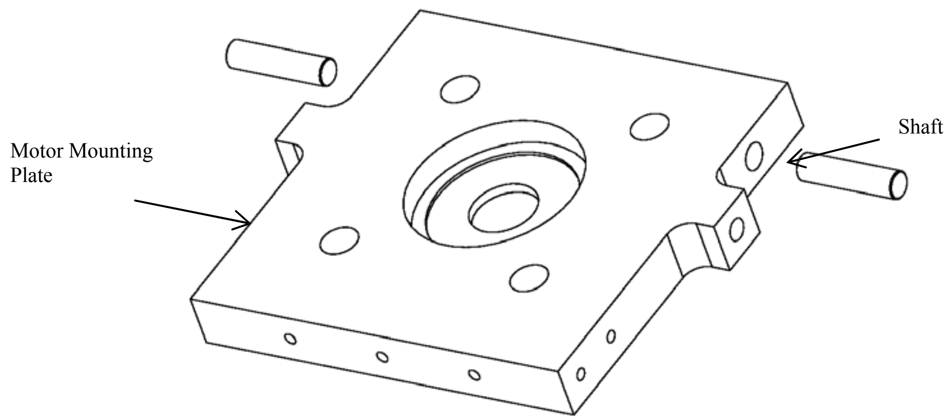


Figure 5.15 Articulating Motor Plate Sub-assembly

The shaft is the same shaft used in the pivot link sub-assembly as seen in Figure 5.10. It provides a point of rotation for link-three. The mounting plate is made of aluminum. The protruding flange is a point of attachment for link-two, again similar to the pivot link sub-assembly, and is a tapped hole accommodating a ¼ - 20 UNC socket head cap bolt.

5.3.7.2 Link-two Sub-assembly

Link-two, like the motor mounting plate, is a part of the kinematical loop 3. It is shown below in Figure 5.16.

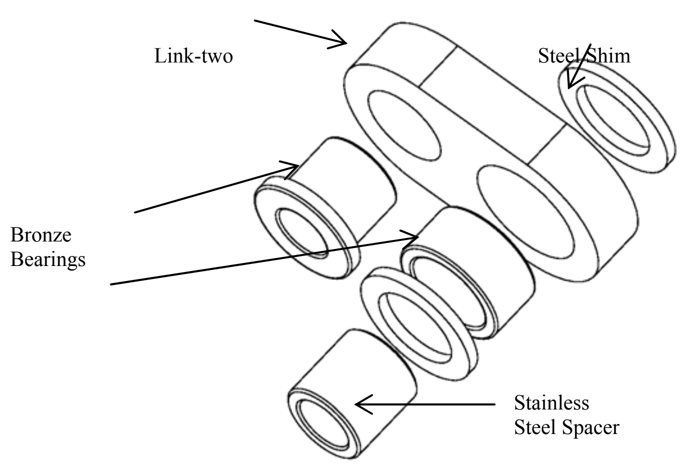


Figure 5.16 Link-two Sub-assembly

Link-two is constructed out of aluminum and, like link-five, has bronze bearings press fit into the link body. It is secured with a ¼ - 20 UNC socket head cap bolt to the articulating motor plate sub-assembly shown in Figure 5.15. The bolt secures the steel spacer that serves as a shaft for the link to rotate about.

5.3.7.3 Link-three Sub-assembly

Link-three is a component of both kinematical loops 2 and 3 and transfers the input from the hydraulic cylinder into the motion required for loop 3 to rotate the motor plate. Figure 5.17 shows the link-three sub-assembly.

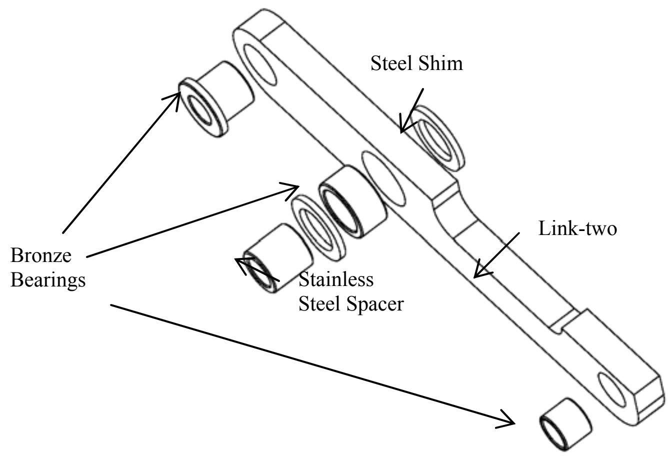


Figure 5.17 Link-three Sub-assembly

Link-three is very similar in design to both link-two and link-five. One distinguishing feature is the apparent large cutout along the top of the link. This cutout provides the needed clearance for the hydraulic lines that will be extending out of the motor. It was mentioned earlier that much of the linkage was designed to accommodate the hydraulic lines so as to avoid interference with other components.

5.4 Chapter Summary

This chapter covered the detailed design of the Rotary Impact Cutting Fixture. The cutting fixture provides extended capability to the current ARDVAC system by a means of a vegetation processing device to be used in conjunction with the vacuum. The cutting fixture is designed to attach directly to the current ARDVAC and provide a point of attachment for a variety of cutting fixtures such as the Hedge Trimmer Cutting Fixture or the Tumbleweed Shredder.

Fixed cutting heads are available for horizontal cutting as well as an articulating cutting head, which can operate both in the horizontal and vertical planes. Vertical cutting provides an option for edging applications such as ice plant or other ground cover type vegetation.

Finalized mounting schemes for the hydraulic system are not complete and are postponed until the hardware is built and attached to the ARDVAC. This provides flexibility in configuring the plumbing for the overall system of the current ARDVAC and the newly developed cutting fixture. The cutting fixture itself is also flexible in that the hydraulic attachment hardware is designed to offer multiple mounting options, see 5.3.3.

CHAPTER 6 FAILURE MODE ANALYSIS

A failure mode in engineering design is a failure of a component that would result in a product losing its ability to meet a customer's requirement. Modes of failure are identified for each component of a mechanical assembly. Having a mode of failure for every component in a mechanism could get exhausting, and thus obvious exemptions should be identified and excluded from the list. The Rotary Impact Cutting Fixture, as seen in Figure 5.1, is relatively simple and for each critical component a failure mode is identified.

6.1 Severity, Detection, and Occurrence

In identifying a failure mode for each component, a severity value (on a scale of 1 to 10) is assigned to that mode of failure. Each mode of failure is also given, again on a scale of 1 to 10, a detection value and an occurrence value. Detection refers to the ability to detect when failure has occurred, which will definitely be obvious for certain components, and occurrence refers to the likelihood of that failure. When rating detection, a 10 means the failure will not be detected if it occurs.

Severity is based on three criteria that includes the cost of replacing the component that failed, the relative difficulty in replacing the component (in the assembly), and the safety hazard that results from component failure. For example, a broken blade could be a potential hazard but is relatively inexpensive and easy to replace. In contrast, a damaged mounting bracket poses no real safety threat but is costly and difficult to replace because of the custom manufacturing required. These are the kinds of issues embedded in the severity rating.

6.2 Analysis of Failure Modes

Table 6.1 below summarizes the results of the failure mode analysis.

FMEA of Cutter Test Fixture					
Mode of Failure	**Severity**	**Detection**	**Occurance**	**RPN**	**Rank**
Fixed cutter sheet metal brackets fail in bending	7	4	2	56	**11**
Fixed motor mounting plate fails in bending	3	4	7	84	**8**
Shaft coupler fails in shear due to torsion	8	9	2	144	**2**
Steel cutter blade breaks	6	2	4	48	**13**
Stationary cutter sub-assembly mounting bolt shears	2	3	3	18	**16**
Lock collar fails	4	4	6	96	**5**
Key-way in shaft coupler tears	8	9	1	72	**9**
Hydraulic cylinder pivot bracket fails in bending	2	4	2	16	**17**
Hydraulic cylinder plates fail in bending	6	3	5	90	**6**
Skid plate fails in bending	7	4	6	168	**1**
Link-five fails in bending (Lat. or Horz.)	2	2	5	20	**15**
Tie rod link fails in tension	1	2	2	4	**18**
Link-three fails in bending (Lat. or Horz.)	5	4	6	120	**4**
Debris shield gets torn from assembly	7	1	7	49	**12**
Articulating motor mounting plate gets torn from assembly	6	2	7	84	**7**
Mechanical linkage socket head cap assembly bolts fail in shear	5	4	3	60	**10**
Upper ring attachment brackets fail in bending	9	3	5	135	**3**
Nozzle weldment attachment fasteners fail in shear	5	3	3	45	**14**

Table 6.1 Failure Mode Analysis of Rotary Impact Cutting Fixture

Analyzing every failure mode presented in Table 6.1 could be difficult and not entirely necessary because a majority of the components scored relatively low. Those components with an RPN (risk priority number) of 100 or higher will be given attention to reduce the possibility of failure. Totally eliminating failure is somewhat unreasonable because of the environment where the cutting fixture is used. There is always a possibility of an unforeseen impact type loading scenario that exceeds the strength of a component.

6.2.1 Skid Plate

The sheet metal skid plate, see Figure 5.9, is a critical component, and it would be relatively severe if it failed in bending. The cost and time to construct a new part would be high because it has a number of sheet metal bends and six critical mounting holes for the pivot link sub-assembly. In addition, the old part would have to be removed from the base tube, where it is welded, and the new part welded on. The sheet metal chosen for manufacturing the part is a 12 gauge sheet steel, which has an approximate thickness of 2.66 mm (0.105 in).

6.2.1.1 FEA of Skid Plate

To make the skid plate more robust and less likely to fail, two large gussets are added to the nozzle weldment assembly, see Figure 5.9. To show these gussets increase the strength of the skid plate, a finite element analysis (FEA) is performed.

The loading scenario used ensures the skid plate could withstand the weight of the ARDVAC, approximately 4450 N (1000 lbf), resting on top of it (in a cantilever fashion) and have a dynamic loading factor of approximately two. This dynamic loading factor can be found from Equation (6.1) below.

$$\frac{F_i}{W} = 1 + \sqrt{1 + \frac{2\eta h}{\delta_{st}}} \tag{6.1}$$

Equation (6.1) is a ratio of the impact force (F_i) on a part, or struck mass, to the weight (W) of a striking mass. η is a correction factor that relates the mass of the striking object (m) to the mass of the struck object (m_b). The equation for this correction factor is

$$\eta = \frac{1}{1 + \frac{m_b}{3m}} \tag{6.2}$$

The striking mass (m) is the mass of the ARDVAC. The struck mass (m_b) is the mass of the skid plate which is considerably less than the ARDVAC and thus as seen in Equation (6.2), η would be approximately equal to one. h in equation (3.1) is the height at which the striking mass is above the struck mass, and δ_{st} is the static deflection of the struck mass under the influence of a static load from the striking mass. It is interesting to note that in Equation (6.1), as h goes to zero the dynamic ratio approaches two. This is the dynamic loading factor mentioned previously.

Figure 6.1 and Figure 6.2 show the results from the skid plate FEA with and without the gussets, respectively. "Mechanica" (an analysis application of Pro/Engineer) is used for the FEA.

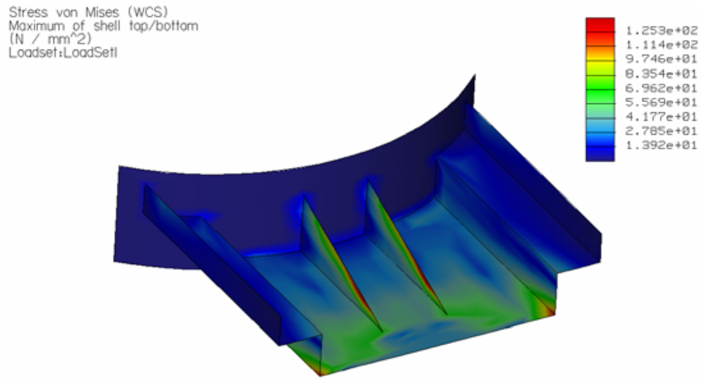


Figure 6.1 Result from Finite Element Analysis on Skid Plate with Gussets

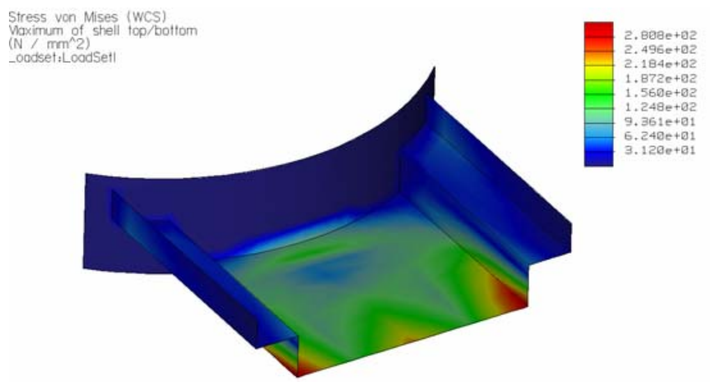


Figure 6.2 Result From Finite Element Analysis on Skid Plate with No Gussets

It is obvious that the finite element model is different from the actual CAD model. The holes and cutouts are not included for simplification and to reduce analysis time. To further simplify the analysis, shell elements are used, which is reasonable because of the thin sheet metal used to manufacture the part. The maximum stress in the skid plate with the gussets is approximately 140 MPa (20.2 kpsi) and without the gussets the maximum stress is approximately 310 MPa (45.3 kpsi), thus the stress (from this particular loading scenario) has been reduced below the yield (207 MPa or 30 kpsi) of the welded 1018 steel.

Figure 6.3 and Figure 6.4 show convergence of the analyses with the variable of interest being the maximum Von Mises stress. Convergence is determined by allowing the previous state of stress compared with the final state to be within approximately 5%. "Mechanica" uses an auto-mesh feature with P-version elements for meshing both solid parts and shell approximations. A P-version element is an element that has a variable order interpolating polynomial associated with it, which can be varied, in "Mechanica", up to the 9th degree. When running an analysis, the auto-mesh in "Mechanica" continues to refine the mesh, for each iteration of the stress calculation, called a P-loop pass, with higher order interpolating polynomials. As the interpolating polynomial is increased, with each iteration, the stress changes less significantly. If convergence has not occurred by the last iteration, there could be an error in the model and the stress value should not be completely accepted.

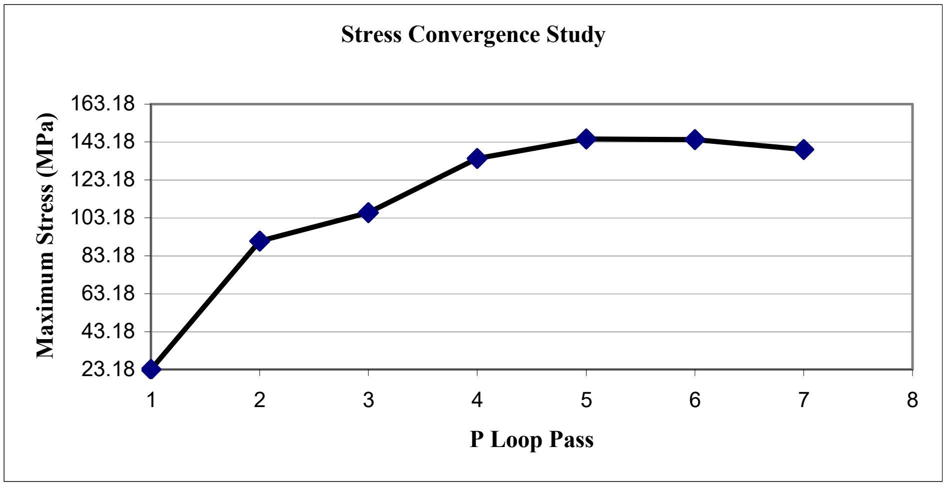


Figure 6.3 Stress Convergence for Skid Plate with Gussets

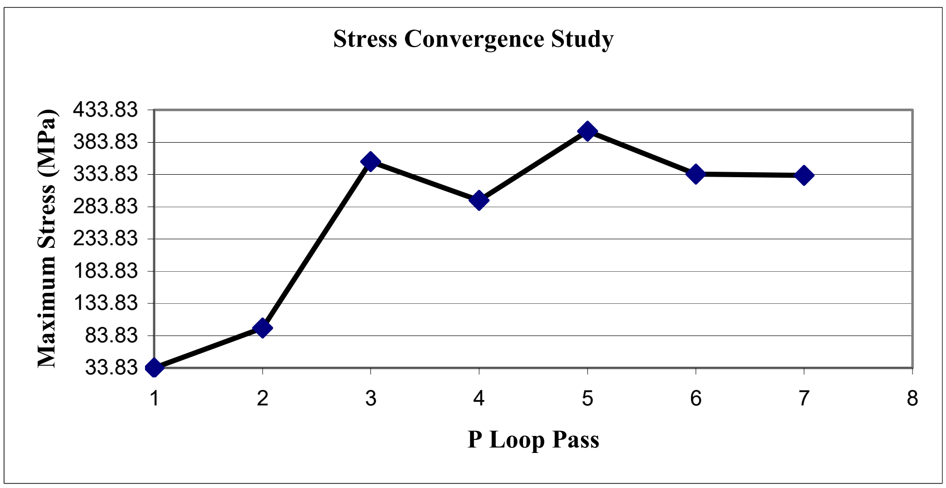


Figure 6.4 Stress Convergence for Skid Plate with No Gussets

6.2.2 Shaft Coupler

From Figures 6.3 and 6.4, the shaft coupler failing in shear ranked 2nd, thus requiring some attention be given to determine whether further design is necessary. It ranked high because detecting if failure occurs is difficult. The shaft coupler is the shaft driven by the motor, which then spins the cutter. It is enclosed in the bearing housing and spins on two ball bearings. Figure 5.13 shows how the shaft coupler fits into the overall assembly.

Upon further inspection of the overall assembly, it is apparent that the likelihood of torsional shear failure is rather low. The engineering reasoning behind this conclusion is that the cutting head attached to the shaft coupler contains blades (see Figure 5.12 or Figure 5.14) that swing free. Thus getting stuck in the cutting process and generating a large torque is unlikely. In addition, the Grass Gator cutting head has a nylon or plastic body and is attached to the shaft coupler with a bolt. The body has a cutout in its underside that fits the profile of the bolt head, and this serves to transfer power from the shaft coupler to the cutting head. This cutout will most likely undergo a type of tear-out failure long before the steel shaft couple fails in shear.

6.2.3 Upper Ring Attachment Brackets

The upper ring attachment brackets, see Figure 5.6, connect the whole rotary impact cutting fixture to the ARDVAC unit and are important pieces of hardware. Although relatively easy to replace, failure of this component could damage the whole assembly.

It was said earlier that the overall weight of assembly is to be kept at a minimum and consequently, aluminum is used wherever possible and was the case for this component prior to the failure mode analysis. With aluminum there is a concern with fatigue, because it's SN diagram does not have an endurance limit, especially with the ARDVAC's dynamic motion when in operation. Accordingly, to really understand the stress and resulting fatigue on the component, it would need to be tested specifically.

To ensure the bracket is as strong as possible and to avoid potential fatigue failure, three things are done. First the bracket is made from a mild steel instead of the 6061 aluminum of the original design, second a fillet is added to the internal corner, and third the horizontal flange is thickened from 4.76 mm (0.188 in) to 6.35 mm (0.25 in).

6.2.3.1 FEA of Upper Ring Attachments Brackets

The loading scenario for the brackets is found from modeling the cutting fixture as a point mass swinging from a rigid pendulum at ½ Hz and then suddenly stopped at the oscillation peak. This scenario simulates a type of motion the ARDVAC system can undergo. The mass of the cutting fixture is assumed to be 45 kg (100 lbm). With two brackets to attach the cutting fixture, each bracket will feel half of the weight or approximately 225 N (50 lbf). The pendulum is assumed to be approximately 1.52 m (5 ft) in length. With these parameters, Newton's Law, Equation (6.3), is used to find the dynamic force felt by the bracket.

$$\vec{F} = m\frac{\Delta v}{\Delta t} \tag{6.3}$$

In Equation (6.3) the time difference is assumed to be finite and is given a value of 0.1 sec. In addition, the change in velocity is found from taking the initial velocity to be the steady state oscillatory velocity of the pendulum and the final velocity to be zero. The steady state oscillatory velocity is defined as

$$v_{ss} = \omega r \tag{6.4}$$

where ω is the free swinging frequency (in rad/s) of the pendulum and r is the length of the pendulum, both of which are given values above. Substituting these parameters into Equations (6.3) and (6.4) results in a force of approximately 1085 N (244 lbf) felt by each bracket. This force is used for the FEA.

Figure 6.5 and Figure 6.6 show the results of the FEA of the brackets both before and after the strengthening modifications are made.

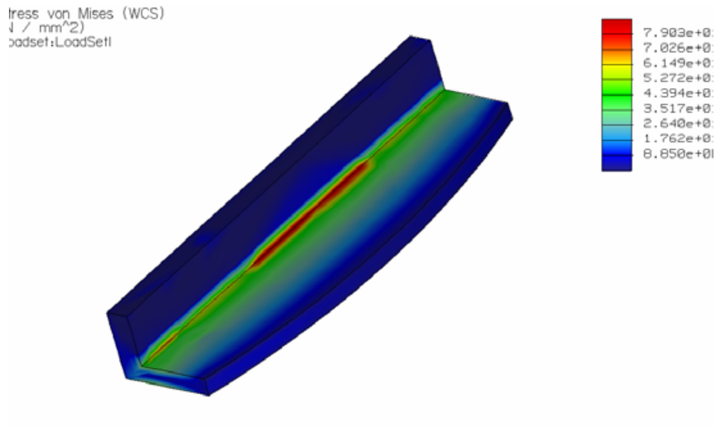


Figure 6.5 Result from Finite Element Analysis on Aluminum Bracket with No Flange nor Fillet

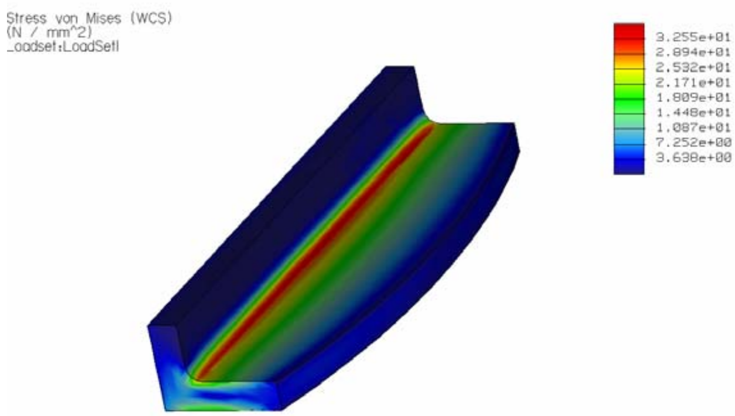


Figure 6.6 Result from Finite Element Analysis on Steel Bracket with Fillet and Thicker Flange

The steel bracket with the fillet and thicker flange has a maximum Von Mises stress of about 36 MPa (5.2 kpsi) and the aluminum bracket without the fillet has a max stress of about 89 MPa (12.7 kpsi). The aluminum bracket without the fillet is well below yield (310 MPa or 45 kpsi), but any increase in a margin of safety for the unknown fatigue properties is desired, and thus the redesigned steel bracket is an important and inexpensive upgrade.

Figure 6.7 and Figure 6.8 are convergence studies of the aluminum and the steel brackets, respectively.

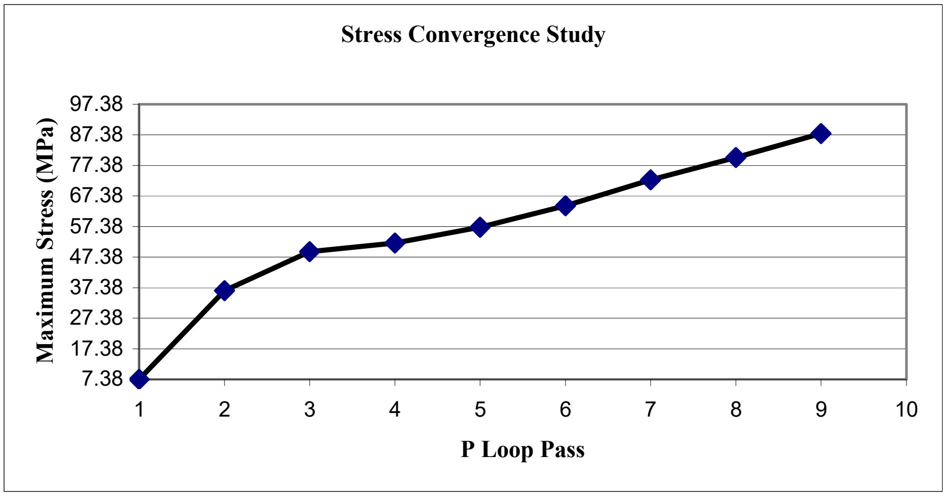


Figure 6.7 Max Stress Convergence for Aluminum Bracket with no Fillet

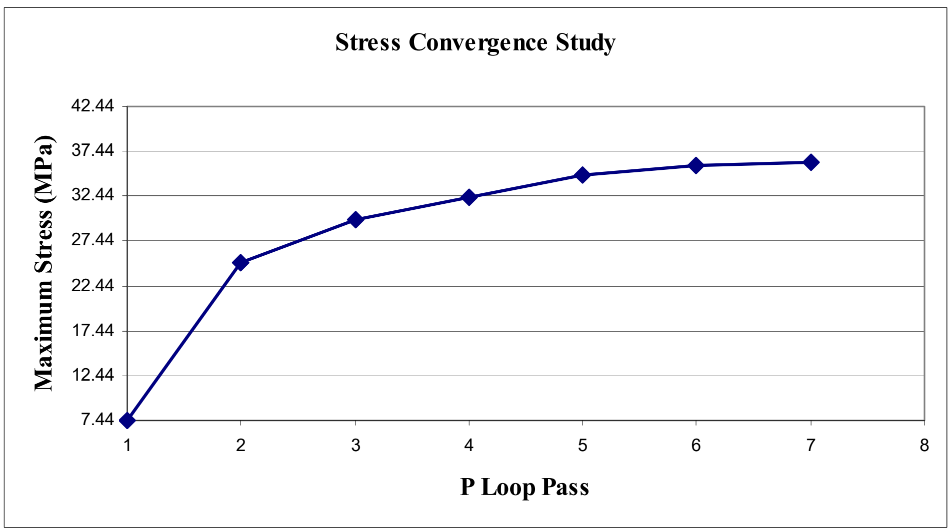


Figure 6.8 Max Stress Convergence for Steel Bracket with Fillet and Thicker Flange

Figure 6.8 shows convergence has been achieved to approximately 5% and thus the resulting max stress value can be trusted. However, Figure 6.7 shows that for the aluminum bracket with no fillet, the stress has not quite converged. Consequently, the value for the stress cannot be totally trusted and is most likely more than what is indicated. Thus the change in material, the added fillet, and the thicker flange are important, necessary features.

6.2.4 Link-three

The last failure model to be discussed is the horizontal and vertical bending of link-three. Recall link-three is a key component in the mechanical linkage which provides rotational motion for the articulating cutter, see Figure 5.14. Its high score or total in the failure mode analysis is largely dependent on the likelihood of occurrence.

6.2.4.1 Horizontal Bending

If the articulating cutter, while in operation, is subject to side loading from debris, most likely the loading will be transferred to link-three and link-two. Link-two is a very small component (small bending moment-arms) and probably not as likely to see failure size stresses as readily as link-three.

Because of the compact size of the mechanical linkage, the unavailability of space in and around the motor mounting plate and the hydraulic lines from the motor, not much can be done that has not already been done to increase the robustness of the design. To increase the rigidity of the linkage, which is all that is done in response to potential horizontal bending failure, steel shims, see Figure 5.17, are added to all linkage mounting points to limit horizontal deflection.

6.2.4.2 Vertical Bending

Vertical bending failure would occur if the articulating cutter became stuck while in the process of being rotated from a horizontal cutting position to a vertical cutting position. The hydraulic cylinder, capable of relatively large forces, would keep pushing and most likely cause the failure of expensive components. In this type of loading scenario, the first component to fail would most likely be link-three at the cutout, which provides clearance for the hydraulic hose.

To keep this failure from happening, the hydraulic system is designed with a pressure relief valve tuned to dump to the tank before reaching the bending stress allowance of link-three. Figure 6.9 shows the result from a FEA on link-three. This analysis is done to determine at what force failure size stresses are found in the component and consequently at what force should the relief valve be tuned.

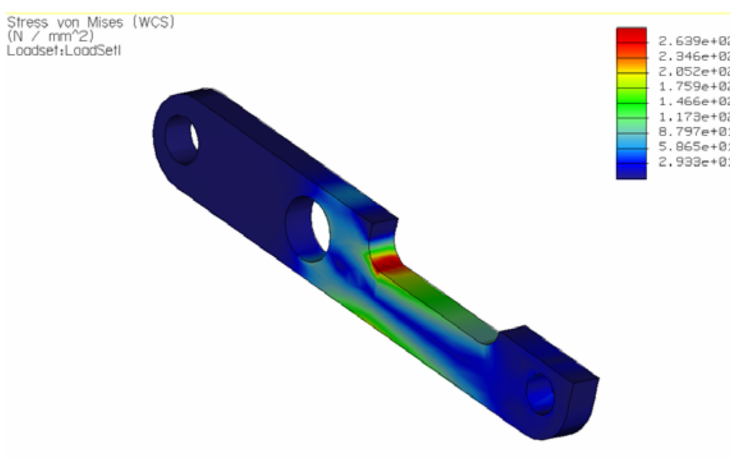


Figure 6.9 Result from Finite Element Analysis on Link-three

It is found with a load applied on the surface of the hole where the tie-rod link (see Figure 5.2) joins link-three, approximately 560 N (130 lbf) can be tolerated before yielding. This load is per link thus with two components, twice this load can be tolerated. Although not shown, convergence is achieved and a quick hand calculation can be done (for verification) using the basic bending stress equation, which states

$$\sigma_{max} = \frac{Mc}{I} \tag{6.5}$$

For a rectangular cross section, this reduces to

$$\sigma_{\max} = \frac{6M}{bh^2} \tag{6.6}$$

Using the max stress as the yield strength of 6061-T6 aluminum, which is 276 MPa (40 kpsi), Equation (6.6) results in a tolerated force of approximately 756 N (170 lbf). The discrepancy between the hand calculation and the FEA is attributed to the lack of accounting for the stress concentration factor in Equation (6.5). Using the FEA result should be conservative and provide a higher safety factor compared to the hand calculation result.

6.3 Chapter Summary

This chapter provided engineering analysis for key components of the Rotary Impact Cutting Fixture. A failure mode analysis was done to show four critical failure modes that required attention. Three of these components required additional features to help prevent failure.

The first of these components was the skid plate. Large Gussets were added to strengthen the joint at the skid plate base tube interface. Secondly, attention was given to the shaft coupler where it was shown further design was not necessary. Although a critical component, the likelihood of failure is extremely low. Thirdly, the upper ring attachment brackets were analyzed and redesigned. The bracket was given a new material, steel instead of aluminum, and a strengthening fillet was added. Lastly, link-three was given some attention. Shims were added to increase the horizontal bending strength of the link-three sub-assembly. To strengthen the part against vertical bending failure, a pressure relief valve is proposed to offer relief in the event the cutting head gets stuck when the cylinders attempts to rotate it from horizontal to vertical or vice-versa.

CHAPTER 7 ROTARY CUTTING HEAD DYNAMIC MODEL

7.1 Chapter Overview

In Chapter 2 the detail design of the rotary impact cutting fixture is developed. The design is based largely around implementing an array of three rotary impact cutters. A key component of each cutter is the actual cutting head used to cut the target vegetation, see Figure 5.12. This Chapter provides a detailed dynamic analysis of the rotational cutting head used in the Rotary Impact Cutting Fixture. The model is used to predict reaction forces generated at the retaining pin, which attaches the cutting blade to the cutting body sub-assembly, while cutting single stem vegetation. With an understanding of the underlying mechanics of the system, design changes can be made to improve cutting head performance.

Dynamic simulations show forces generated for both steady state spinning of the cutting head and when subject to a force input from cutting a stem of vegetation. This force is applied at the tip of the cutting blade. The development of the model, used to simulate this force, will be explained in detail.

7.2 Rotational Impact Cutting

Most vegetation cutting devices use momentum and subsequent impact as their means of cutting. Lawn mowers have a fixed, rigid blade that rotates at a high rate and cuts by shearing the grass. String trimmers have a flexible nylon string extending from a fixed, circular body. Spinning the nylon string at a high enough rate enables it to shear relatively thin vegetation. A recent development, in the quest for a more robust trimmer style cutting device, is the bladed trimmer head. These bladed trimmers are similar to a conventional string trimmer but use free-swinging plastic blades, instead of a string, for shearing of the target vegetation.

7.2.1 Modifying Stock Cutting Heads

Latham [15] chooses a bladed trimmer or cutting head manufactured by Grass Gator as an alternative to a conventional string trimmer. In his design recommendations he suggests there is potential risk involved in replacing the stock plastic blades with high-strength steel blades. Modifying a stock high speed rotational cutting head can be risky because of the lack of knowledge of the underlying mechanics of the high rotational momentum involved and related inertial forces.

7.2.2 Weed Wizard Cutting Head

On May 3, 2000 a cutting head named the Weed Wizard was recalled by the U.S. Consumer Product Safety Commission (CPSC) [29]. The Weed Wizard is a rotational cutting head similar to a string trimmer but instead of a nylon "string" a short metal chain, similar to a bike chain, is used as the mechanism for cutting. Figure 7.1 below shows a picture of this cutting device.

Figure 7.1 Weed Wizard Rotational Cutting Head

As can be seen in the picture the last link is a type of thin metal blade. The type of metal used for the chain is unknown. This cutting device was recalled because according to the CPSC "the end link of the trimmer's metal chain can rapidly and unexpectedly detach during use, propelling the link into the air at a high velocity." Forty-seven cases were reported of this occurrence that resulted in forty-one injuries and one death.

The Weed Wizard is an example of implementing a concept to make a product more effective but doing it without a thorough understanding of how it affects the safety of the device. The purpose of the dynamic analysis of the Grass Gator is to determine the safety risks of replacing a light weight plastic cutting blade with a steel blade and to make some final design changes with respect to the rotary impact cutting fixture. In addition, the analysis gives insight into the mechanics of impact cutting with bladed trimmer heads. Noticeable design changes and performance improvements develop as a result of the dynamic analysis.

7.3 Cutting Head Dynamic Model

Figure 7.2 shows the modified Grass Gator cutting head used by Latham. It is composed of a cutting body sub-assembly, which includes plastic base plates and steel retaining pins, and custom steel cutting blades.

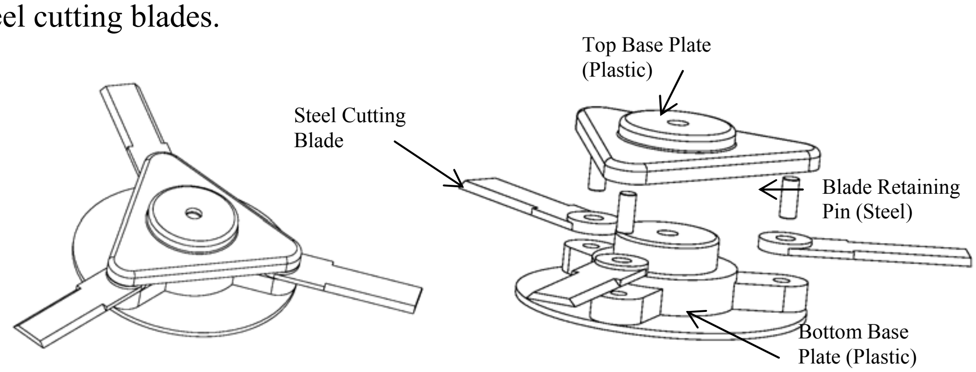


Figure 7.2 Grass Gator Cutting Head with Metal Cutting Blades

For the dynamic model of the cutting head, a single blade attached to the cutter body sub-assembly is used, see
Figure 7.3 and

Figure 7.4 below. This simplifies the model and seems reasonable because the other two blades are attached in exactly the same fashion. As stated earlier, the interest lies in the reaction forces at the blade retaining pin, thus all three blades need not be included in the model.

Mass properties for the cutting blade are obtained from Pro Engineer. This keeps the simulations as accurate as possible with respect to the cutting blade. The cutting body sub-assembly is modeled as a circular disk, but the mass is taken directly from the CAD model. The relevant output data from Pro Engineer includes locations of centers of gravity (CG) and mass properties including inertia tensors with respect to important points, such as the CG. It seems reasonable to make these simplifications to the model because the cutting body sub-assembly simply provides an attachment point for the cutting blade.

7.3.1 Dynamic Model Analysis Variables

Figure 7.3 shows the reference frames, generalized coordinates, and generalized speeds for the dynamic model of the cutting head.

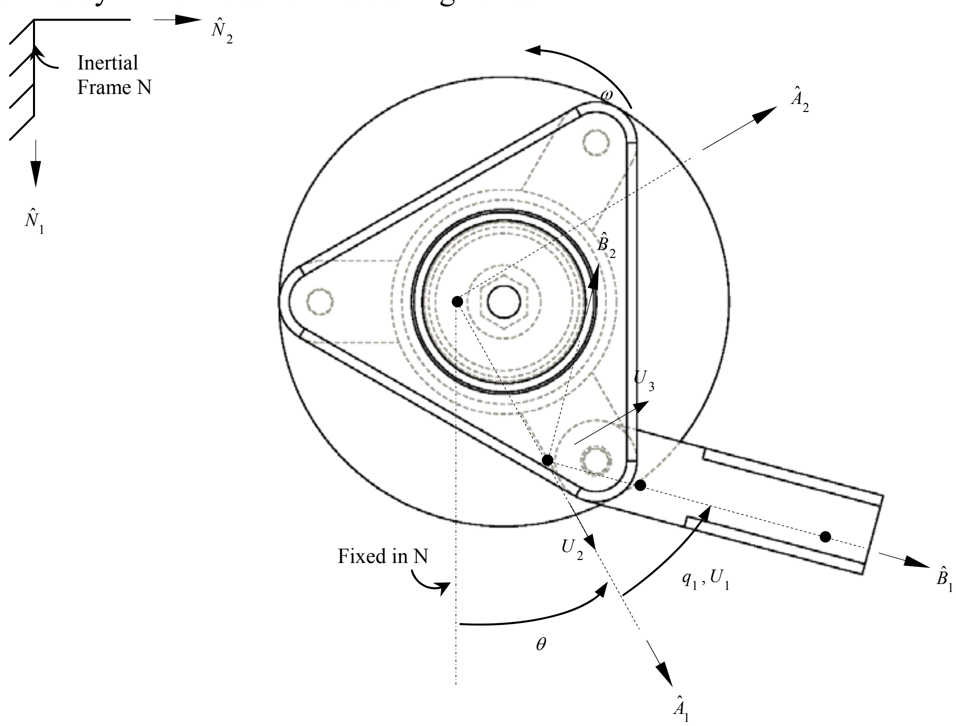


Figure 7.3 Reference Frames and Generalized Coordinates and Speeds

The reference frames and variables are summarized below.

$\hat{N}_1$, $\hat{N}_2$, and $\hat{N}_3$ - unit vectors which define a positive orthonormal inertial reference frame.

$\hat{A}_1$, $\hat{A}_2$, and $\hat{A}_3$ - orthonormal reference frame that is attached to body A, the cutting body sub-assembly.

$\hat{B}_1$, $\hat{B}_2$, and $\hat{B}_3$ - orthonormal reference frame that is attached to body B, the cutting blade.

θ - variable that defines the angular position for the simple rotation of body A (cutting body sub-assembly) about the positive $\hat{N}_3$ direction.

ω - rotational speed of the cutting body sub-assembly, which is assumed constant in the analysis that follows.
q_1 - generalized coordinate that defines the angular position of body B (cutting blade) about the positive $\hat{N}_3$ direction with respect to body A.
U_1 - generalized speed that defines the rate of angular rotation of body B, or the rate of change of q_1 where $\dot{q}_1 = U_1$, with respect to body A.
U_2 - auxiliary generalized speed used to determine the $\hat{A}_1$ or radial component of the retaining pin reaction force.
U_3 - auxiliary generalized speed used to determine the $\hat{A}_2$ or tangential component of the retaining pin reaction force.

7.3.2 Dynamic Model Geometry

Figure 7.4 shows the geometry of the cutting head dynamic model.

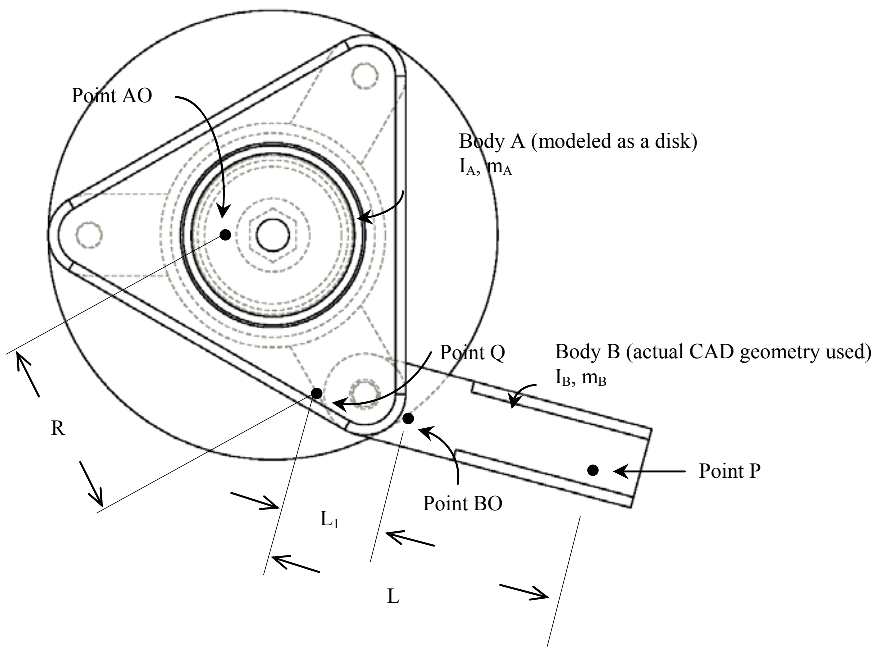


Figure 7.4 Geometry for Dynamic Model

The following list explains the associated geometry:
R – the distance, in the $\hat{A}_1$ direction, between the center of mass (AO) of the cutting body sub-assembly (modeled as a disk) and the center of the retaining pin (Q).
L – the distance from the center of the retaining pin (Q), in the $\hat{B}_1$ direction, to the tip of the cutting blade (P).
L_1 – the distance, in the $\hat{B}_1$ direction, between the retaining pin center (Q) and the center of mass (BO) of the cutting blade.

7.4 Equations of Motion

With the geometry, variables, and reference frames established, the equations of motion governing the dynamics of the cutter head can be formulated.

7.4.1 Kane's Method

Kane's Method is used to derive the governing equations of motion. Harker's thesis contains the detailed derivation of the equations and thus is not included here. It also contains a printout of the Matlab code used for numerical simulation and for plotting the dynamic response.

7.4.2 Dynamical Equations

From
Figure 7.3 and
Figure 7.4, the cutting head is very similar to a double pendulum where the upper link is essentially a circular disk rotating about its mass center. The circular disk is the cutting body sub-assembly and it is spinning at a constant angular velocity. The lower link is the cutting blade. Gravity, depending on the desired cutting orientation, vertical or horizontal, acts in either the positive $\hat{N}_1$, or the negative $\hat{N}_3$ direction, respectively.

7.4.2.1 Horizontal Cutting with Grass Gator

Equation (7.1) below is the governing ordinary differential equation for the dynamic response of the cutting blade. Equations (7.2) and (7.3) describe the reaction forces at the cutting blade retaining pin in the $\hat{A}_2$ (tangential) and $\hat{A}_1$ (radial) directions.

$$\left(I_{33\_B} + m_B L_1{}^2\right)\ddot{q}_1 + D\dot{q}_1 + m_B L_1 R\omega^2 \sin q_1 = -LF_{bld} \qquad (7.1)$$

$$F_{\hat{A}2} = F_{bld}\cos q_1 - L_1 m_B[\sin q_1(\omega + \dot{q}_1)^2 - \cos(q_1)\ddot{q}_1] \qquad (7.2)$$

$$F_{\hat{A}1} = -F_{bld}\sin q_1 - m_B[R\omega^2 + L_1\cos q_1(\omega + \dot{q}_1)^2 + L_1\sin(q_1)\ddot{q}_1] \qquad (7.3)$$

The mass properties and relevant parameter values in Equations (7.1), (7.2), and (7.3) are listed in Table 7.1 below. In addition, ω, for all subsequent simulations, is given the value of 3000 rpm.

Mass Properties and Relevant Parameters		
Cutting Body Sub-assembly		
Mass, kg (lbm)	0.2096	(0.4621)
$I_{11,}$ kg-m^2 (lbm-in^2)	1.71E-04	(0.5859)
$I_{22,}$ kg-m^2 (lbm-in^2)	1.71E-04	(0.5859)
$I_{33,}$ kg-m^2 (lbm-in^2)	3.43E-04	(1.1717)
R, m (in)	0.0572	(2.2520)
Cutting Blade		
Mass, kg (lbm)	0.0622	(0.1371)
$I_{11,}$ kg-m^2 (lbm-in^2)	5.75E-05	(0.1965)
$I_{22,}$ kg-m^2 (lbm-in^2)	6.05E-05	(0.2066)
$I_{33,}$ kg-m^2 (lbm-in^2)	3.19E-06	(0.0109)
L, m (in)	0.0889	(3.5000)
$L_{1,}$ m (in)	0.0348	(1.3701)

Table 7.1 Mass Properties and Parameter Values for Dynamic Equations

The mass of the cutting body sub-assembly is the total mass of the top and bottom base plates plus three steel retaining pins, all added together. Inertia values (for a thin disk) are calculated using Equations (7.4) and (7.5).

$$I_{11\_A} = I_{22\_A} = 1/4 m_A R^2 \tag{7.4}$$

$$I_{33\_A} = 1/2 m_A R^2 \tag{7.5}$$

It should be noted, that the products of inertia of the cutting blade and the cutting body sub-assembly, with respect to the axes chosen, are very small and they are thus neglected in the analysis. Furthermore, there are no angular velocity components out of plane with respect to the reference frames established, and thus the products of inertia do not affect the dynamics of the system.

Two of the parameters that have not been discussed yet are *D*, and F_{bld}. *D* is the viscous damping constant and for simulation purposes is given the value of 0.05 N-s-m (0.089 lbf-s-in). It is simply based on response results from simulations and subsequent iterating until the response seems reasonable in the sense that the cutting blade shows some small oscillations that quickly damp out. F_{bld} is the force applied on the blade, at point P, see Figure 7.4, which models the cutting blade shearing a single stem of vegetation. The development of this force is now explained.

7.4.2.1.1 Input Force Model

McRandal and McNulty [17, 18] suggest that cutting force varies linearly with penetration of the cutting blade into the stem, and cutting is completed when the penetration equals the stem diameter. They define the cutting force as

$$f = k(vt - y) \tag{7.6}$$

where *k* is the resistance to penetration, *v* is the cutter blade velocity, *t* is the time from when the blade first makes contact with the stalk to when cutting is complete, and *y* is the initial deflection of the stalk prior to penetration, at $t = 0$. They also suggest that the stem can be modeled as a particle with a mass equal to the product of the stem length *b*, above the cut, and the linear density *s* of the vegetation being cut. This results in

$$f = bs\ddot{y} \tag{7.7}$$

where $\ddot{y}$ is the particle acceleration. Combining Equations (4.6) and (4.7) results in a 2nd order ordinary differential equation as

$$\ddot{y} + (y - vt)\frac{k}{bs} = 0 . \tag{7.8}$$

McRandal and McNulty solve Equation (7.8) using the Laplace Transform which gives

$$y = v\left(t - \sqrt{\frac{bs}{k}}\sin\sqrt{\frac{k}{bs}}t\right). \tag{7.9}$$

Differentiating twice yields

$$\ddot{y} = v\sqrt{\frac{k}{bs}}\sin\sqrt{\frac{k}{bs}}t . \tag{7.10}$$

Substituting (7.10) into (7.7) results in a cutting force of

$$f = bsv\sqrt{\frac{k}{bs}}\sin\sqrt{\frac{k}{bs}t} . \tag{7.11}$$

Taking the limit of Equation (7.11) as $t \to 0$ gives

$$f = vtk . \tag{7.12}$$

Equation (7.12) resembles the equation for a linear spring where *k* is essentially a spring constant and *vt* is the spring displacement. Equation (7.9) is used to solve for the time in Equation (7.12) to complete the cut. McRandal and McNulty explain that cutting is complete when

$$vt - y = 2r \tag{7.13}$$

where *r* is the radius of the stem being cut. Solving Equation (7.13) for *y* and substituting this into (7.9) results in

$$t = \sqrt{\frac{bs}{k}}\sin^{-1}\left(\frac{2r}{v}\sqrt{\frac{k}{bs}}\right). \tag{7.14}$$

The linear density *s* in Equation (7.14) is found from the density ρ used for typical pine wood (approximately 550 kg/m^3 or 0.019 lbm/in^3, which is unexpectedly the same value used by McRandal and McNulty for perennial ryegrass). In addition, McRandal and McNulty define the resistance to penetration *k* as

$$k = \frac{W_c}{2r^2} \tag{7.15}$$

where W_c is the static shear energy of the stem being cut.

7.4.2.1.1.1 Static Shear Energy

The static shear energy W_c is one of two parameters that needs to be addressed before the input force on the cutting blade can be found, where the other parameter is the stem length *b*. In a related study, McRandal and McNulty perform quasi-static shear testing on perennial ryegrass to determine its shear properties. They utilize a mechanical test rig to develop force versus deflection curves. The area under these curves is the static shear energy. They also investigated the effect of grass maturity (age), size (stem diameter cross-section), and dry-matter content with respect to the grass properties. They correlated this data by linear regression, which resulted in an empirical equation for the static shear energy given by

$$W_c = -39.6 + 8.18a - 253z + 3.51t$$

where a in Equation (7.16) is the total cross-sectional area (mm^2), z is the fractional dry matter content, and t is the age (weeks). Typical average stem values include 23.5 weeks for age and 0.25 for fractional dry matter content. In addition, Latham, in finding a minimum blade speed for complete shearing, performed cutting tests on wood doweling with diameters of 3.175 mm (0.125 in), 4.763 mm (0.1875 in), and 6.350 mm (0.25 in). Using the 6.350 mm (0.25 in) diameter doweling, for simulating the most severe case of the three, for which the cutting blade is to shear, and the stem values listed above, the static shear energy W_c is approximately 240 mJ (2.12 lbf-in).

7.4.2.1.2 Cutting Time

Substituting the calculated static shear energy into Equation (7.15) yields a resistance to penetration k of 11.84 kN/m (67.61 lbf/in). With k calculated, the last parameter needed is the stem length b. The stem length for target vegetation could vary tremendously. Consequently, the cutting time is calculated for various stem lengths using a cutting head velocity v equal to 18.1 m/s (59.4 ft/s) (based on an ω of 3000 rpm). The results are then plotted and shown below in Figure 7.5.

It is interesting to note that as the stem length increases, the cutting time seems to approach an asymptotic cutting time between 0.35 and 0.36 ms. This is explained by looking at Equation (7.14). b in the inverse sin term is in the denominator. As b gets larger and larger, the inverse sine portion of the equation will asymptotically approach zero. This seems reasonable considering the particle model. Most of the mass relevant to the analysis is centered around the point of impact of the blade. Mass further from the point of impact is less relevant.

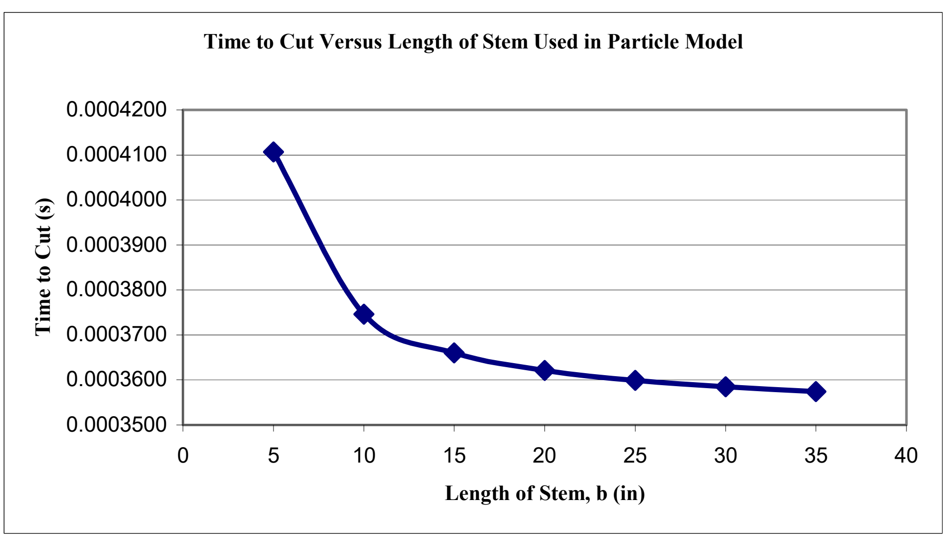


Figure 7.5 Cutting Time versus Length of Stem Being Cut

7.4.2.1.3 Input Force

Implementing the cutting force into the numerical analysis is done by assuming a constant input force on the cutting blade over a time period of 0.4 ms, which is slightly longer than the time period calculated above. The cutting input force is applied at 0.1000 s and is taken away at 0.1004 s. In order to capture this brief impulse, the time step in the numerical simulation needs to be at greatest 0.0001 s. To use the exact time of 0.36 ms, the time step would need to be dropped to 0.00001 s, which could dramatically increase computation time and round-off error, and thus the exact time is not used. Substituting a cutting time *t* of 0.4 ms into Equation (7.12) results in a force on the cutting blade edge of approximately 87 N (19.6 lbf) at 3000 rpm.

7.4.2.1.4 Horizontal Cutting Simulation Results

Figure 7.6, Figure 7.7, and Figure 7.8 show the results from a cutting simulation with the cutting head spinning at 3000 rpm and with an input force and cutting time applied as specified above.

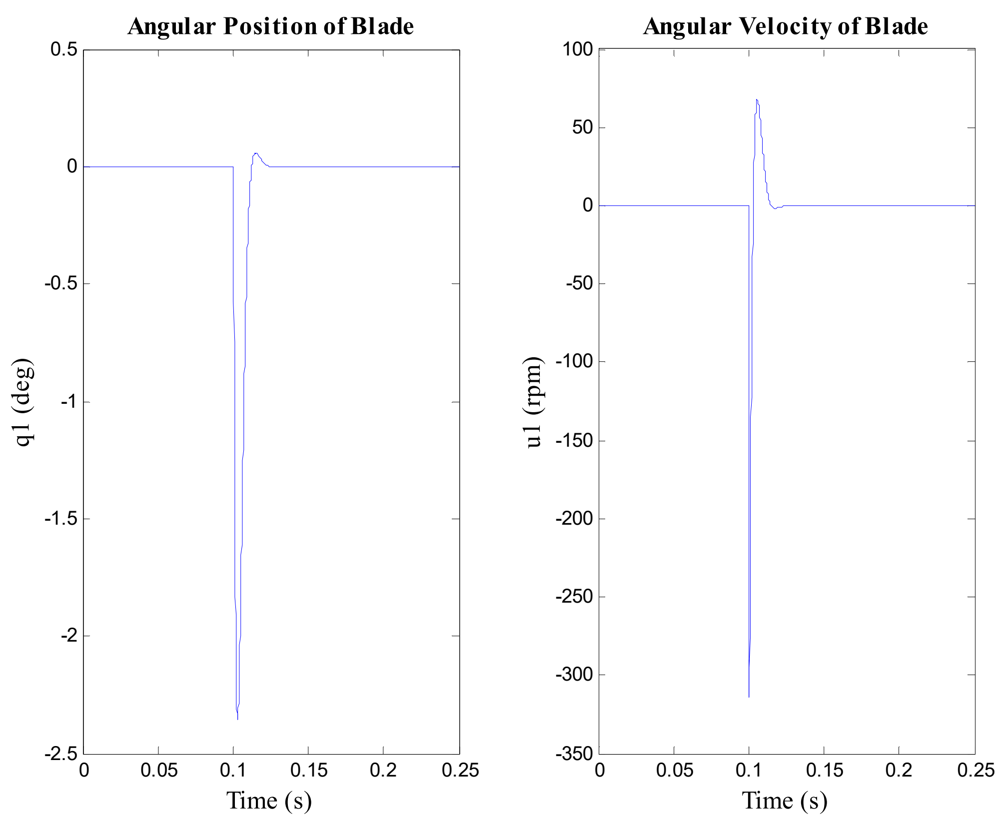


Figure 7.6 Dynamic Response of the Cutting Blade for Horizontal Cutting of a 6.35 mm (0.25 in) Stem Diameter

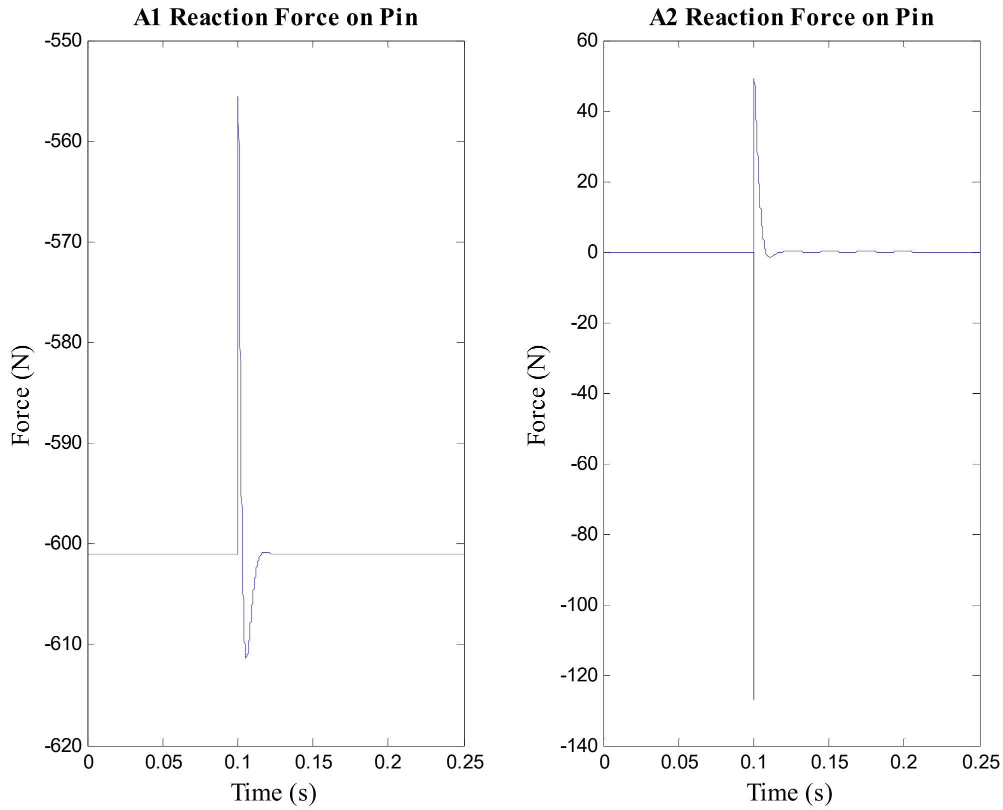


Figure 7.7 Retaining Pin Reaction Force Components for Horizontal Cutting of a 6.35 mm (0.25 in) Stem Diameter

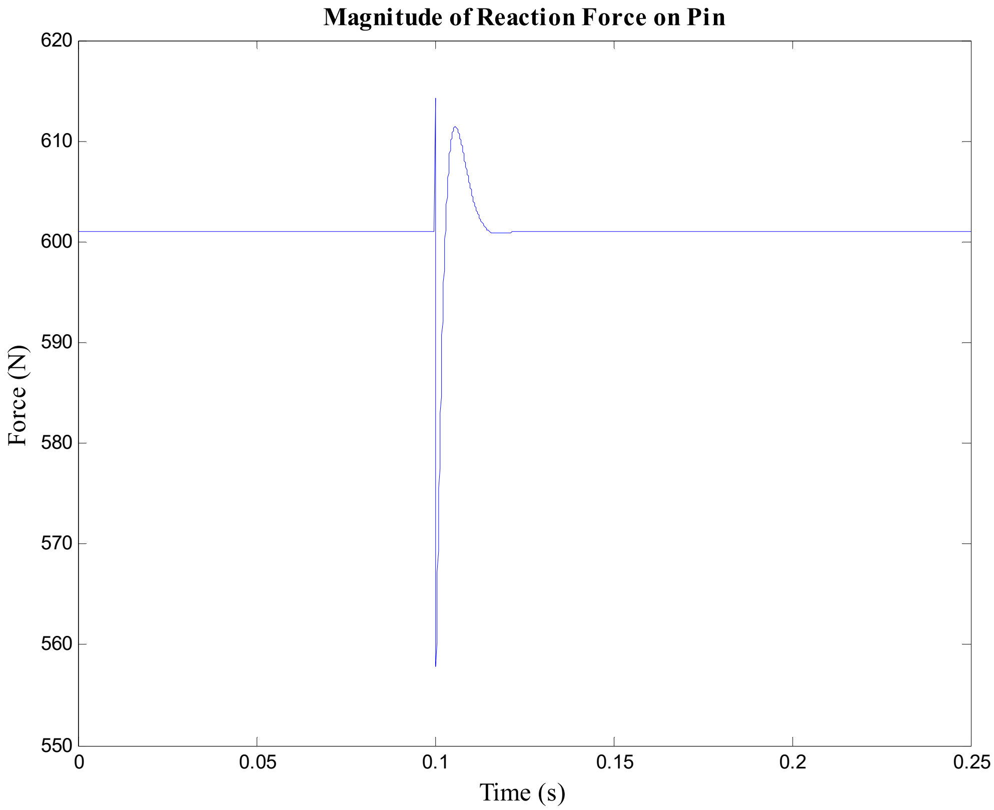


Figure 7.8 Magnitude of Force on Retaining Pin for Horizontal Cutting of a 6.35 mm (0.25 in) Stem Diameter

In Figure 7.6 the left plot shows the cutting blade's angular displacement and the right plot shows the blade's angular velocity over time. The large spike represents the stem striking the blade.

Figure 7.7 shows the retaining pin reaction force components for the radial and tangential directions, respectively. It seems reasonable that the tangential component is initially zero. There are no tangential acceleration components. Then, after impact, it damps back out to zero. In addition, a quick calculation can be made to verify the radial force component on the steel blade. At steady-state, Equation (4.3) reduces to

$$F_{\hat{A}1} = -m_B \omega^2 \left(R + L_1\right) \tag{7.17}$$

which is the force due to the radial component of acceleration. Substituting in the parameters results in a force magnitude of approximately 600 N (135 lbf), which agrees with Figure 7.7 for the cutting blade.

Figure 7.8 shows the magnitude of the resultant reaction force vector, on the retaining pin, formed from both the radial and tangential components. The maximum force is approximately 620 N (140 lbf). This makes it apparent that the largest contributor (for relatively small impact forces), to the retaining pin reaction force, is the radial or centrifugal force component summarized above. From this it can be concluded that reducing the mass of the cutting blade will be the most significant modification if trying to reduce the reaction force on the pin, for normal operating conditions.

7.4.2.2 Vertical Cutting with Grass Gator

A cutting simulation is done with the cutting head rotated into the vertical position, see Figure 5.3. Equation (7.18) below is the governing ordinary differential equation for the response of the cutting blade and Equations (7.19) and (7.20) describe the reaction forces at the cutting blade retaining pin in the $\hat{A}_2$ (tangential) and $\hat{A}_1$ (radial) directions.

$$gL_1 m_B \sin(q_1+\theta) + L_1 m_B R\omega^2 \sin q_1 + D\dot{q}_1 + \left(I_{33\_B} + m_B L_1{}^2\right)\ddot{q}_1 = -LF_{bld} \tag{7.18}$$

$$F_{\hat{A}2} = gm_B \sin\theta + F_{bld}\cos q_1 - L_1 m_B[\sin q_1(\omega+\dot{q}_1)^2 - \cos q_1 \ddot{q}_1] \tag{7.19}$$

$$F_{\hat{A}1} = -gm_B\cos\theta - F_{bld}\sin q_1 - m_B[R\omega^2 + L_1\cos q_1(\omega+\dot{q}_1)^2 + L_1 \sin q_1 \ddot{q}_1] \tag{7.20}$$

Comparing these equations to those for horizontal cutting, there is the obvious difference from the presence of the gravity term.

7.4.2.2.1 Vertical Cutting Simulation Results

Figure 7.9, Figure 7.10, and Figure 7.11 show a similar simulation as that performed initially, e.g. cutting a 6.35 mm (0.25 in) diameter stem, except in this simulation the cutting head is in the vertical orientation.

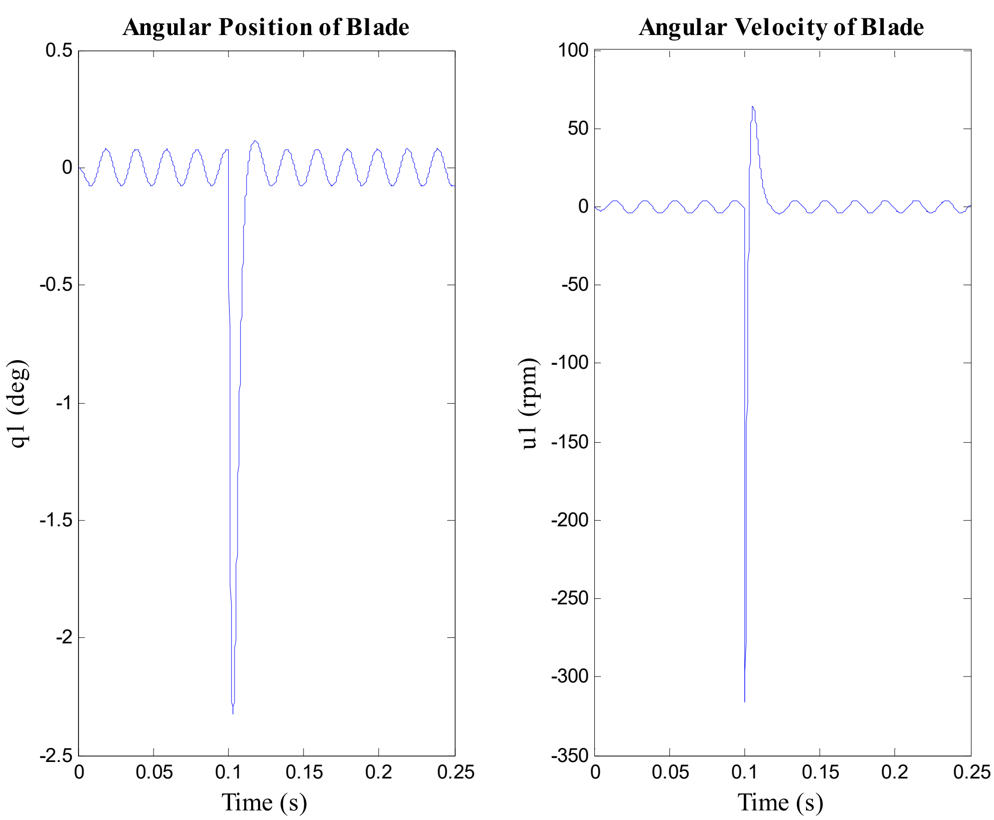


Figure 7.9 Dynamic Response of the Cutting Blade for Vertical Cutting of a 6.35 mm (0.25 in) Stem Diameter

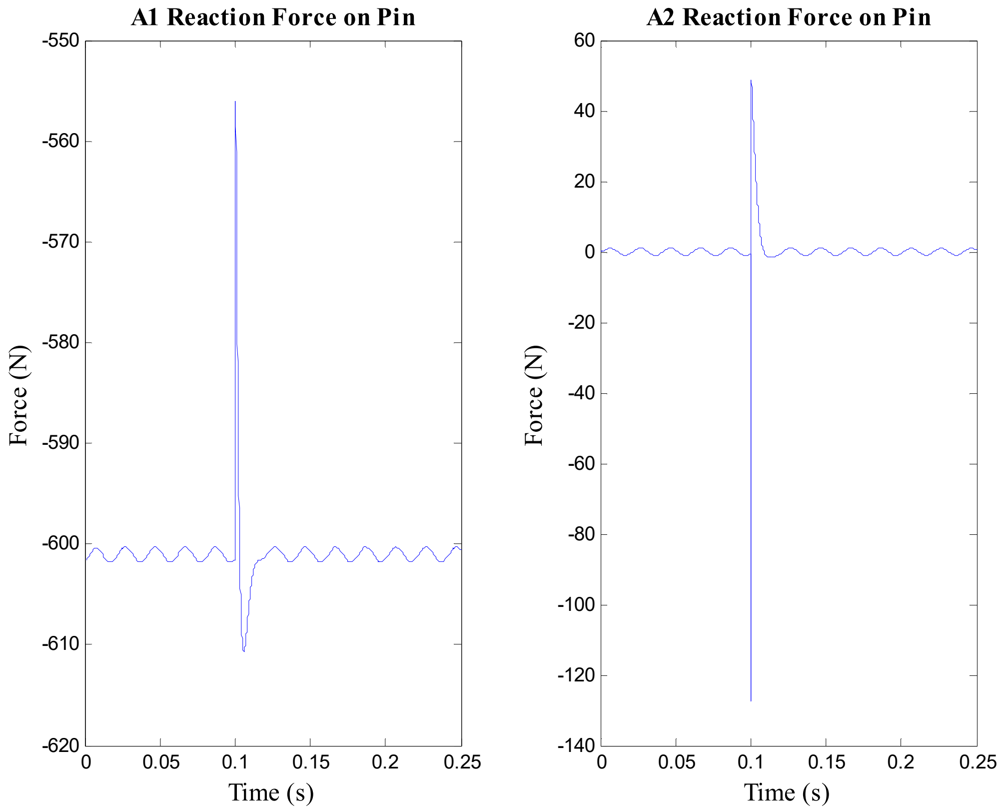


Figure 7.10 Retaining Pin Reaction Force Components for Vertical Cutting of a 6.35 mm (0.25 in) Stem Diameter

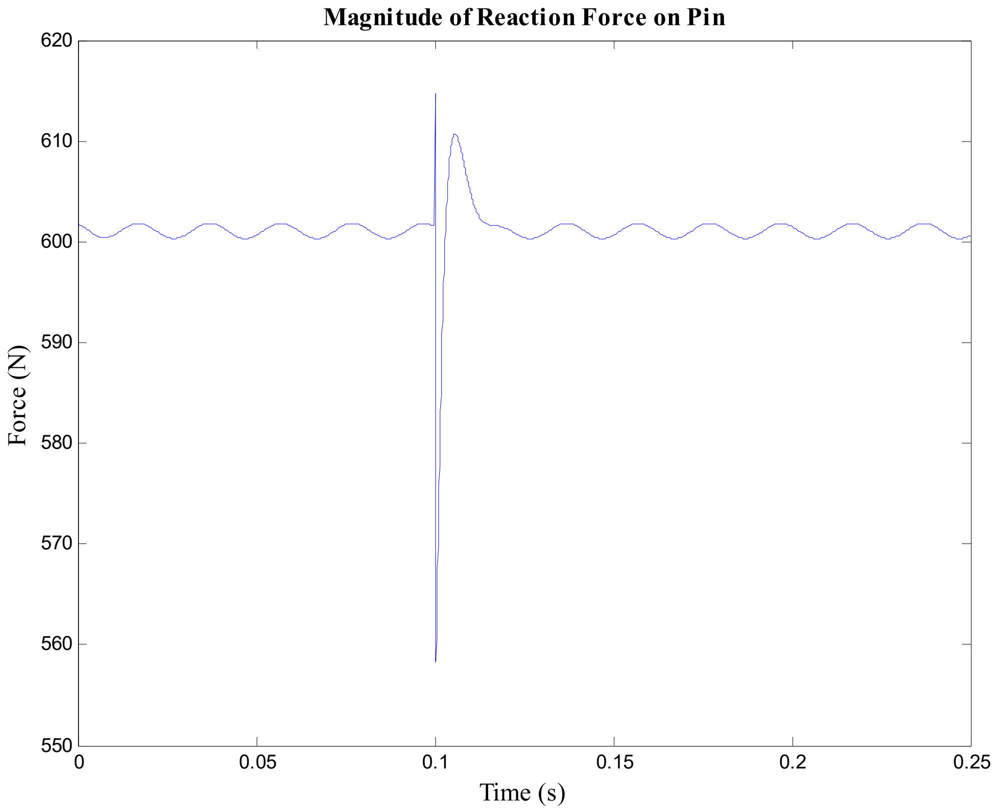


Figure 7.11 Magnitude of Force on Retaining Pin for Vertical Cutting of a 6.35 mm (0.25 in) Stem Diameter

The obvious difference between horizontal and vertical cutting is the contribution of the gravitational component in the dynamic response and force results. It appears as a sinusoidal component, which is expected as the blade rotates around and gravity pulls on the blade.

The vertical cutting orientation is included to possibly provide a means of utilizing the ARDVAC in edging applications such as cutting Ice Plant ground cover, see Figure 1.3. In this cutting scenario, the possibility of the blade striking the ground is likely, which would be similar to the blade striking a large stationary rock in the horizontal cutting configuration.

Spinning the cutting head at 3000 rpm and subsequently cutting a 6.35 mm (0.25 in) stem diameter can produce reasonably large dynamic forces. The rotary impact cutting fixture is designed to be attached to the ARDVAC and used as a tool in removing vegetation debris on state highways. It is to be operated in close proximity to the right-of-way, often in the median on interstates. Vegetation that is to be removed or eliminated is often long and dense, and thus the cutting fixture is likely to strike hidden objects. These objects could include rocks and other hard objects. These objects could also be loose or fixed in the ground. If the cutting blade impacts these types of objects, the forces developed could be very large, much larger than just cutting a single stem of woody vegetation.

The conclusions presented above justify redesigning the overall cutting head to increase the safety factor when operating the cutting fixture. Ensuring the safety of nearby vehicles, possible pedestrians, as well as the system operator is of prime importance.

7.5 High Performance Cutting Head

It was discussed earlier that the majority of the force felt by the retaining pin is caused by inertial forces from accelerations of the cutting blade. Consequently, to reduce this force some of the mass of the cutting blade must be eliminated. Reducing mass does not necessarily mean a drop in performance but can lead to a better design.

Figure 7.12 shows a picture of a high performance cutting head design based on the modified Grass Gator design. Designed for a more industrial type application, such as what the rotary impact cutting fixture is designed for, the Industrial High Performance (IHP) cutting head is lighter, and stronger, and has more cutting surface than its predecessor.

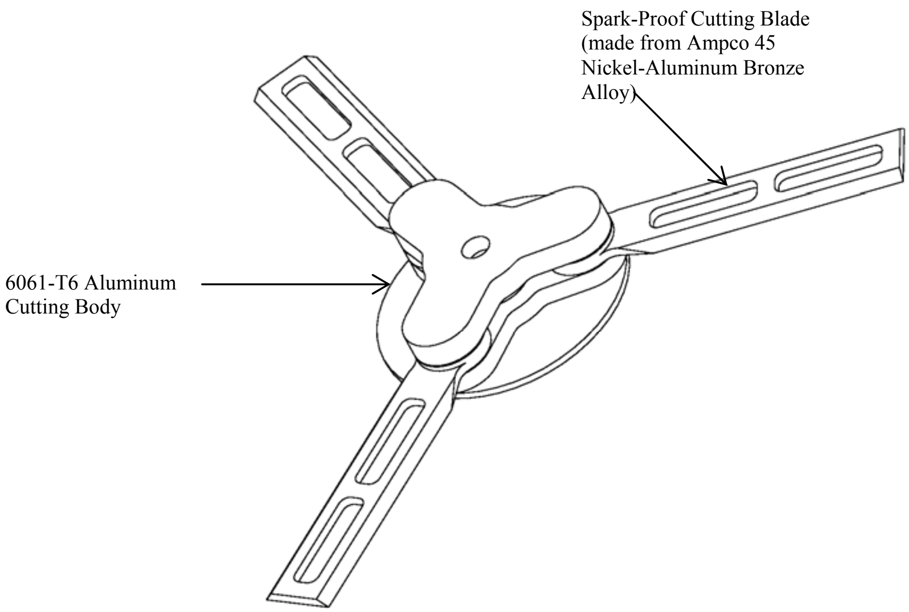


Figure 7.12 Industrial High Performance (IHP) Cutting Head

Table 7.2 provides a comparison between the Grass Gator and the IHP cutting head. The table shows properties relevant to the dynamic analysis.

Cutting Head Comparison				
	Grass Gator		Industrial High Performance	
Cutting Body Sub-assembly				
Mass, kg (lbm)	0.2096	(0.4621)	0.1925	(0.4244)
I_{11}, kg-m^2 (lbm-in^2)	1.71E-04	(0.5859)	4.85E-05	(0.1658)
I_{22}, kg-m^2 (lbm-in^2)	1.71E-04	(0.5859)	4.85E-05	(0.1658)
I_{33}, kg-m^2 (lbm-in^2)	3.43E-04	(1.1717)	9.70E-05	(0.3316)
R, m (in)	0.0572	(2.2520)	0.0318	(1.2500)
Cutting Blade				
Mass, kg (lbm)	0.0622	(0.1371)	0.0576	(0.1270)
I_{11}, kg-m^2 (lbm-in^2)	5.75E-05	(0.1965)	8.70E-05	(0.2973)
I_{22}, kg-m^2 (lbm-in^2)	6.05E-05	(0.2066)	9.00E-05	(0.3075)
I_{33}, kg-m^2 (lbm-in^2)	3.19E-06	(0.0109)	3.15E-06	(0.0108)
L, m (in)	0.0889	(3.5000)	0.1143	(4.5000)
L_1, m (in)	0.0348	(1.3701)	0.0423	(1.6654)

Table 7.2 Mass Properties and Geometry Comparison of Cutting Heads

This table presents five overall design enhancements of the IHP versus the Grass Gator. The first and most important is improvement is that the mass of the IHP's cutting blade is less than the Grass Gator's. This is accomplished by machining small pockets in the blades, see Figure 7.12, and using a lighter material. The blades are made from a material called Ampco 45 (or often referred to as AL 630) and are a nickel-aluminum bronze alloy. The second design enhancement is that this blade material is spark-proof, which eliminates the danger of starting a fire in the event that the blade strikes a rock or other hard object. Third, in addition to being light weight and spark proof, the alloy is extremely strong and boasts a yield strength of 517 MPa (75 kpsi) and a tensile strength of 814 MPa (118 kpsi). Fourth, the cutting blade of the IHP is longer than its predecessor, which means the cutting area is increased (by approximately 22%). A fifth design improvement is that the cutting body sub-assembly is smaller and lighter.

It is desirable to keep the overall diameter of the cutting head unchanged, thus the cutting body sub-assembly is considerably smaller in diameter to compensate for the increased blade length. The higher strength is a result of the base plates being made from aluminum rather than plastic.

7.5.1 Dynamic Comparison

Figure 7.13 and Figure 7.14 compare the performance of both cutting heads. The cutting heads are cutting in the horizontal plane while cutting a 6.35 mm (0.25 in) diameter stem.

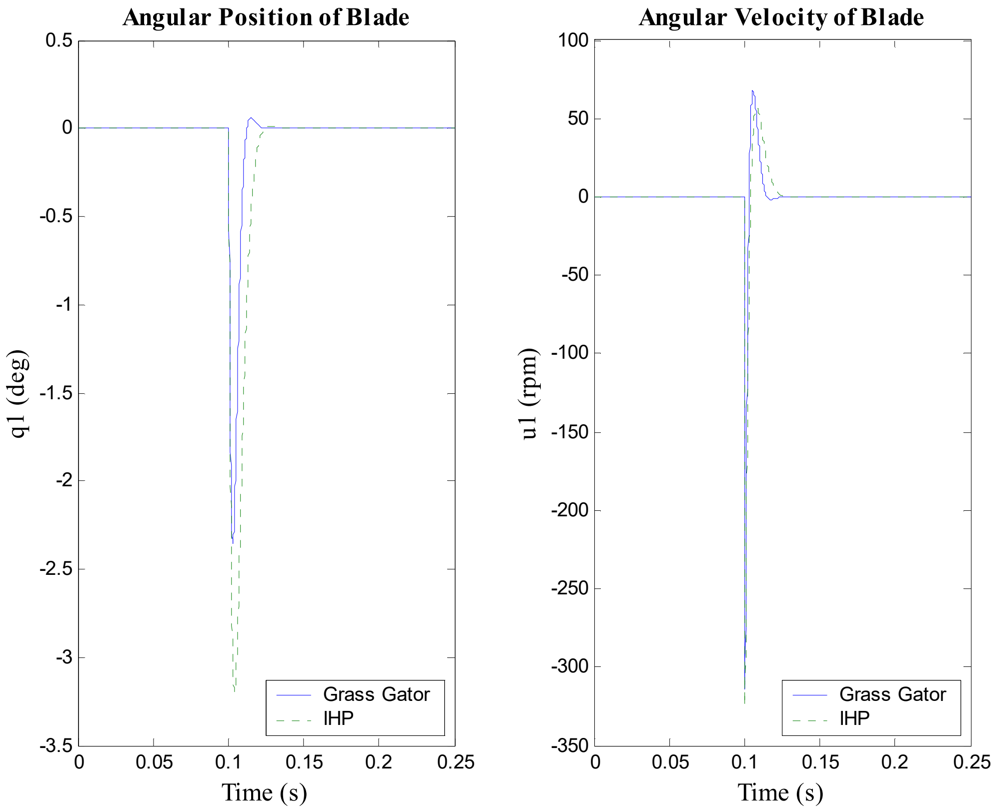


Figure 7.13 Dynamic Response Comparison of Modified Grass Gator and Industrial High Performance Cutting Head

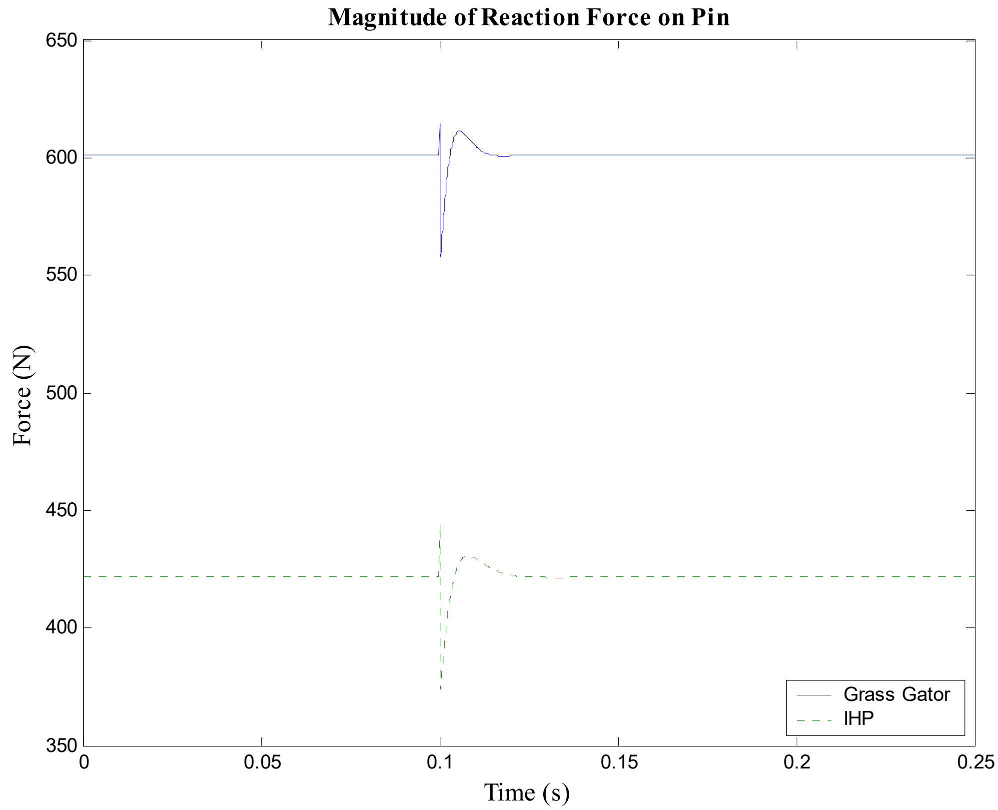


Figure 7.14 Comparison of Retaining Pin Reaction Force of Modified Grass Gator and Industrial High Performance Cutting Head

Figure 7.13 compares the dynamic response of the cutting blade. The lighter IHP cutting blade is subject to more displacement and a slightly faster rotation. In Figure 7.14 the IHP cutting head reduces the force on the retaining pin by approximately 150 N (33.7 lbf), which is not much, but considering the overall assembly is stronger, more efficient, and less likely to start a fire, it is a considerable improvement. With the reduced load on the retaining pin, the safety of operating the rotary impact cutting fixture is increased.

A final simulation is to provide insight into what might happen in the event a blade is struck with a considerably more substantial input force (maybe from a rock) rather than from what a stem of vegetation can provide. Figure 7.15 shows the dynamic response and Figure 7.16 shows the retaining pin reaction force for both cutting heads. The blade input is 1110 N (250 lbf) and is applied over 0.001 s.

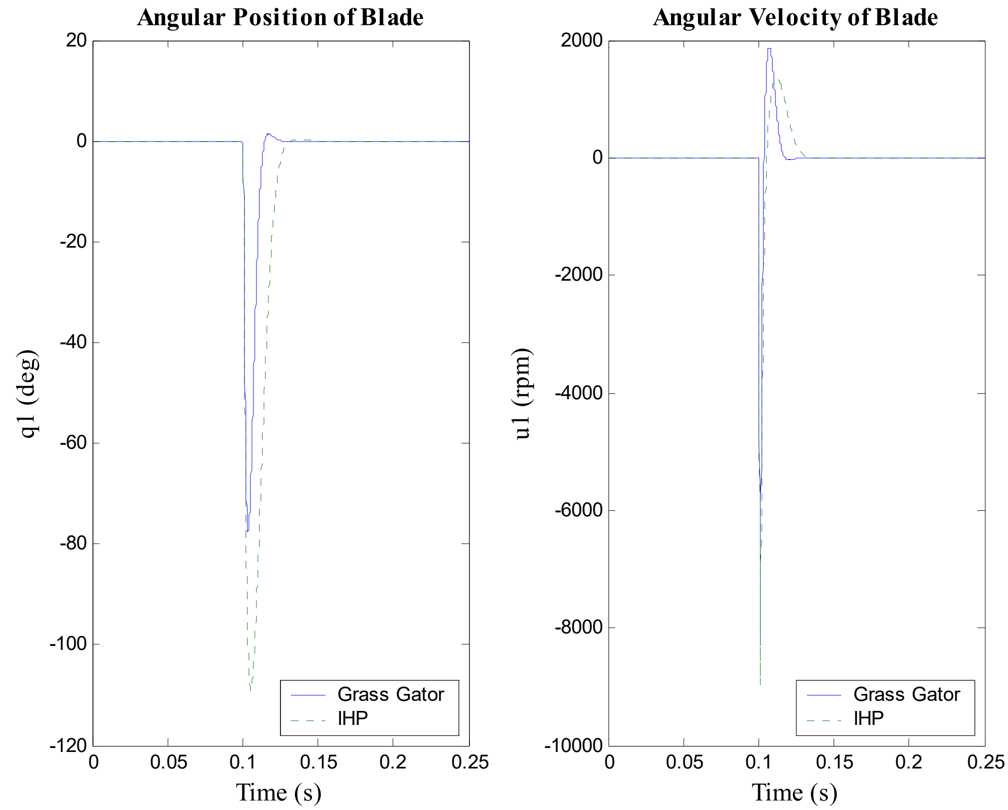


Figure 7.15 Dynamic Response Comparison When Cutting Blades are Subject to a Large Input Force

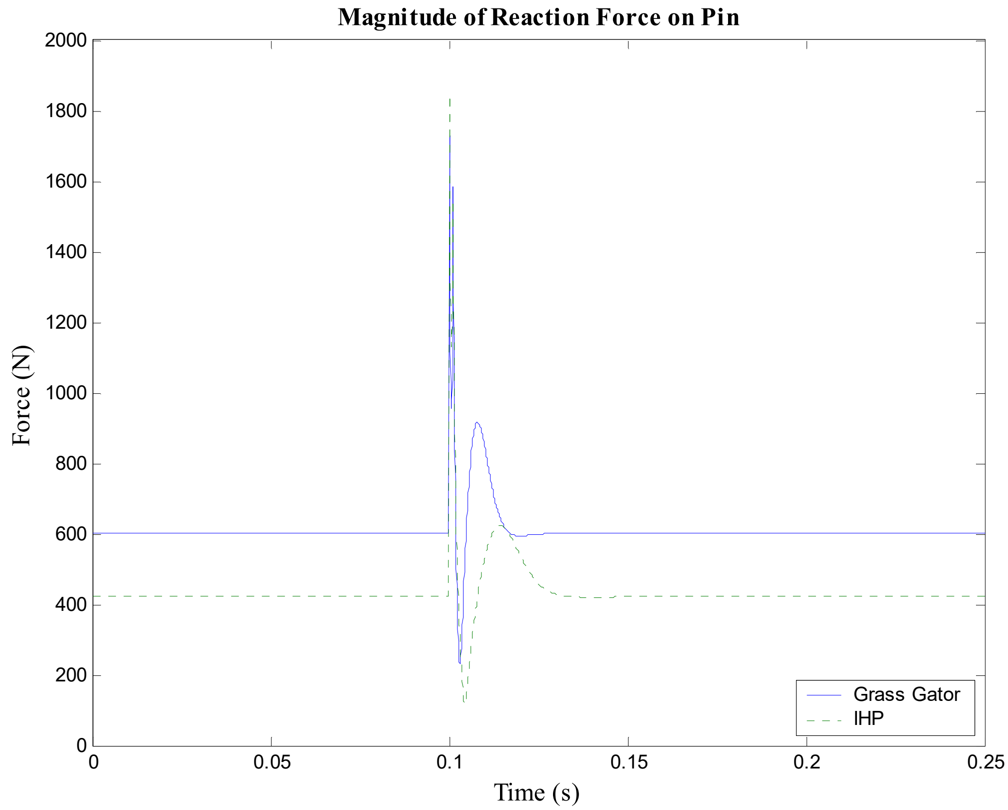


Figure 7.16 Comparison of Retaining Pin Reaction Forces When Cutting Blades are Subject to a Large Input Force

Unlike Figure 7.14, in Figure 7.16 the IHP blade produces a larger reaction force on the pin. The difference is barely noticeable but regardless of which blade is observed, the reaction force is alarmingly large. The maximum force on the IHP cutting blade retaining pin is approximately 1800 N (400 lbf).

7.5.1.1 Retaining Pin Shear Stress

The following defines the shear stress for direct shear application, such as that observed with the retaining pin in the cutting body sub-assembly

$$\tau = \frac{P}{A} \tag{7.21}$$

where P is the force on the retaining pin and A is the cross sectional area. Using a diameter of 7.94 mm (0.313 in) for the retaining pin and reaction force given above, Equation (7.21) gives a shear stress of 36.4 MPa (5.3 kpsi). This is well below the shear yield stress (276 MPa or 40 kpsi) of the pin's 18-8 stainless steel. Repeatedly subjecting the cutting head to input forces of such magnitude could result in fatigue failure of the IHP cutting body sub-assembly where the pins are encapsulated. It has been mentioned that aluminum, such as what the cutting body sub-assembly is made of, does not have an endurance limit.

7.6 Chapter Summary

This chapter presented a numerical dynamic model for cutting single stem vegetation with a bladed rotary cutting head. Considerable detail and research is provided for the development of the blade input force as a result of shear cutting of the stem. In contrast, no basis for the viscous damping constant, between the cutting blade and the retaining pin, is given. This value is based purely on observing the blade response in the simulation output and adjusting it accordingly for reasonableness.

The dynamic model is then used for designing a more robust, efficient cutting head. This design resulted in what is called the Industrial High Performance (IHP) cutting head. Compared to the Grass Gator, the IHP has lighter, stronger, safer cutting blades. It has a lighter and stronger cutting body and when cutting single stem vegetation produces less stress on the blade retaining pin. When the cutting blade is subject to large input forces, it is found the IHP puts slightly more stress on the retaining pin, versus the Grass Gator. The increased force is rather insignificant, in comparison, but the IHP is a justifiably better design for it exhibits such favorable characteristics in other aspects of the design.

CHAPTER 8 DESIGN OF A TUMBLEWEED PROCESSING ATTACHMENT

The objective of this chapter is to go through a detailed description of the design of an automated end effector (EF) attachment that can pick up and process tumbleweeds. Previous work was done on this by Au and others for a senior design project at UC Davis. Their work focused on selecting a concept and coming up with a preliminary design. Interested readers can read their report on the design and concept selection process, as it will not be covered in detail here. While their design was a good starting point, it had many critical flaws.

An objective of this report is to find these problem areas of the original Tumbleweed Machine and redesign them as necessary. Many of these problematic areas were not due to structural integrity or possible failure, but due to a complex design, or lack of necessary parts to make the original design functionally sound. The Tumbleweed Machine concept will be described in detail in this chapter as well as the modifications to the previous design.

8.1 Concept Selection and Design Needs

The requirements for this design were the following:

- Able to pick up tumbleweed
- Able to cut up tumbleweed
- Able to attach to EF
- Keep weight of the machine to a minimum

Au et. al. met three of these four requirements for this design. The one requirement that has not been entirely met is the ability to attach to the EF: It has been partially met in that it was designed with an area for an interface to the EF to be designed and implemented.

The overall concept of the Tumbleweed Machine, which was loosely based on commercial chipping/shredding machines (Figure 8.1), relies on 3 main components: Cutters, Arms, and the Housing. These three components can be seen in Figure 8.2 below.

Figure 8.1 Cutting wheel from a commercial Chipper/Shredder

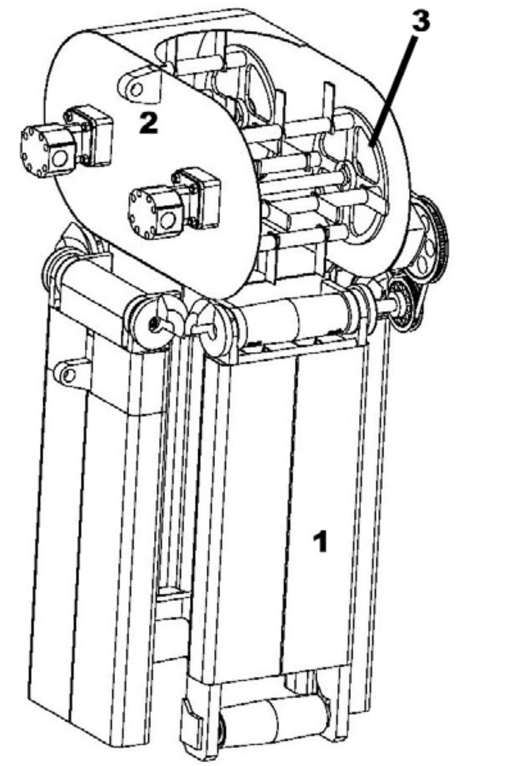


Figure 8.2 Tumbleweed Machine - 1) Arms, 2) Housing, & 3) Cutter Assembly

8.1.1 Overall Concept

The following is a description of what the machine would look like as if one were watching it in action. First, the arms (labeled 1 in Figure 8.2) would be in an open position as in Figure 8.3. The EF boom would lower the machine on top of tumbleweed, and the arms would begin to close. While they are closing, the motors turn on the cutters, and the conveyor belts (not shown, but on the arms with rollers at the end) start moving as well. When the arms close down tight enough on the tumbleweed, the conveyor belts will feed the tumbleweed into the housing, where the cutters begin to chop up the tumbleweed. When it is finished, the motors will turn off, and the arms will remain in a closed position.

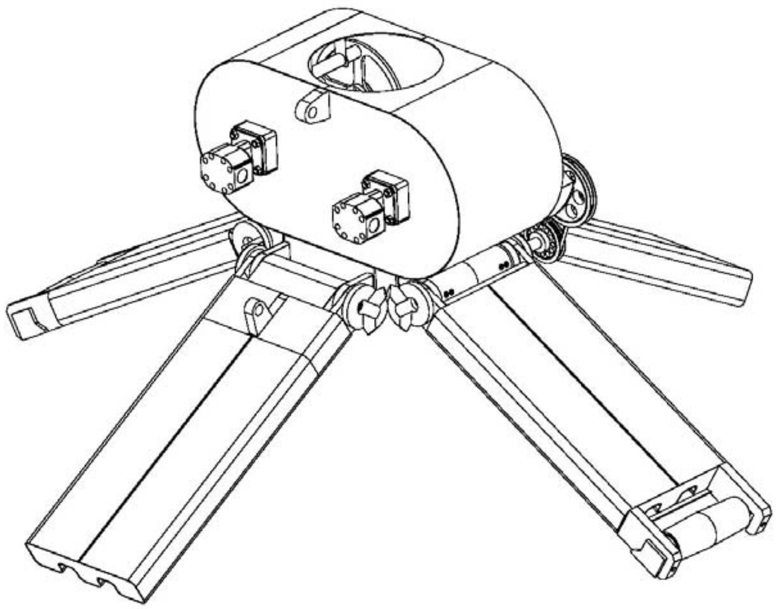


Figure 8.3 Open arm position

8.1.2 Cutters and Commercial Chipper/Shredders

The chipping/shredding mechanism seen in Figure 8.1 is standard for chipper/shredders, and consists of three main parts: A flywheel, which provides a high rotational inertia, shafts, and blades that are attached to the shafts. Chipper/shredders come with all types of variations on the size of the flywheel, blade type, number of blades, etc., but the concept remains the same. This concept of free swinging rotating blades is the basis for the tumbleweed machine.

The cutters are simply a modification of the chipper/shredder device. The chipper/shredder device uses the massive flywheel primarily for the chipping function of the machine, which the tumbleweed machine does not need. Thus, the cutters used for the tumbleweed machine have a much lighter flywheel and many more blades than any of the chipper/shredders that are commercially available. The tumbleweed machine uses two cutter assemblies whose axes of rotation are parallel to each other. Further, they both spin in a direction that will help pull the tumbleweed into the cutters.

8.1.3 Housing

The housing is what holds the Tumbleweed machine together. The hydraulic motors are mounted to the outside of the housing. The cutters serve as a shaft to connect the motors to the pulley system on the rear side of the housing. This pulley system enables the hydraulic motors that power the cutters to power the conveyor belts as well. The hydraulic cylinder that actuates the arm position is also mounted onto the housing.

8.1.4 Arms

The Tumbleweed machine consists of four arms. Two arms have conveyor belts. One of the arms is a plain arm, and the hydraulic cylinder is attached to the last arm. The four arms are connected to each other via a series of mitre gears. This allows all four arms to be controlled by one hydraulic cylinder. The idea with the arms is for them to be in an open position as the EF boom lowers the tumbleweed machine on top of resting tumbleweed. When it gets close, the arms begin to close down, compressing the tumbleweed, and the conveyor belts will begin to feed the tumbleweed into the cutters.

8.2 Detailed Design Description

The previous section served as a basis for understanding how the machine as a whole works. This section will go through a more rigorous description of the Tumbleweed Machine. It also describes individual assemblies and gives a more detailed description of their design. Further, this section will discuss the changes from the old design. First, the 3 individual sub-assemblies will be described. After that, the final assembly, and more specifically how the three subassemblies are connected, will be described. This way of describing it is best since it is how it would be assembled in reality.

8.2.1 Cutter

The Tumbleweed Machine consists of two identical Cutter Assemblies. As mentioned in 8.1.2, the cutters are based on the design of commercial chipper/shredder cutters, and consist of free swinging blades, shafts to hold the blades, spacers, two flywheels to hold the shafts, and a center drive shaft. These components can be seen in the Figure 8.4 below. The design of the cutter was not changed from the previous design iteration.

The two flywheels are slightly different to accommodate mounting to the hydraulic motor and the pulley system. This mounting is discussed in 8.2.4.1. All parts of the cutter assembly will be made of AISI 314 Stainless Steel except for the cutters and spacers. The cutter blades are the same as those seen in Figure 8.1, are made of non-stainless steel, and are readily available at any garden tool reseller. The spacers are made of Delrin® and are nothing more than tubes to fit around the shafts to keep the blades from sliding axially along the shaft.

The cutter shaft and all four knife shafts are welded to the motor flywheel to provide a rigid foundation for the cutter. The knives and spacers are then slid onto the knife shafts such that the knives on any two adjacent shafts are staggered. There are three different spacer lengths to accommodate this. This staggering is needed so that blades on adjacent shafts cannot hit each other when freely swinging around. Finally, the rear flywheel is slid onto the four knife shafts. This will be fastened more securely with screws after it is installed in the housing.

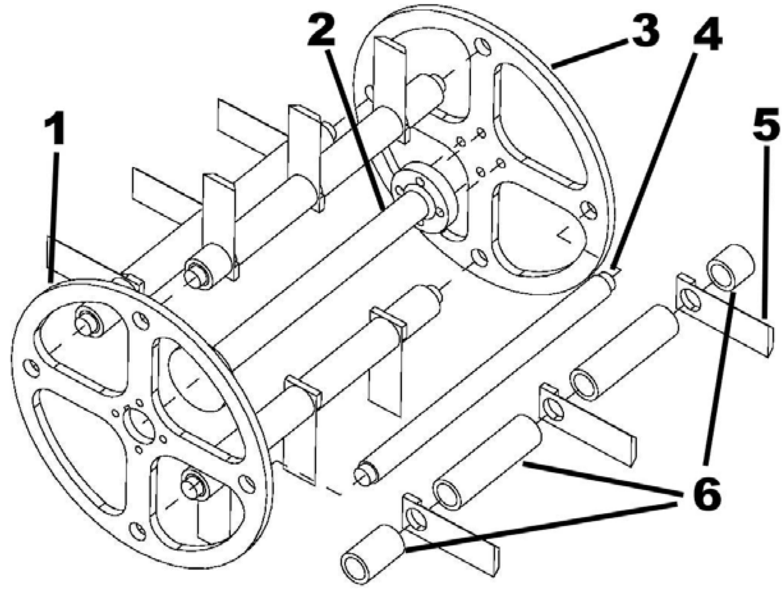


Figure 8.4 Cutter: 1) Motor Flywheel, 2) Drive Shaft, 3) Rear Flywheel, 4) Knife Shaft, 5) Blade, & 6) 3 Spacer sizes

8.2.2 Housing

The Housing is what holds everything together and is what will be attached to the EF. The housing consists of two subassemblies: the Cutter Housing and the Arm Frame, seen in Figure 8.5. The entire housing is to be made out of AISI 1020 cold rolled steel.

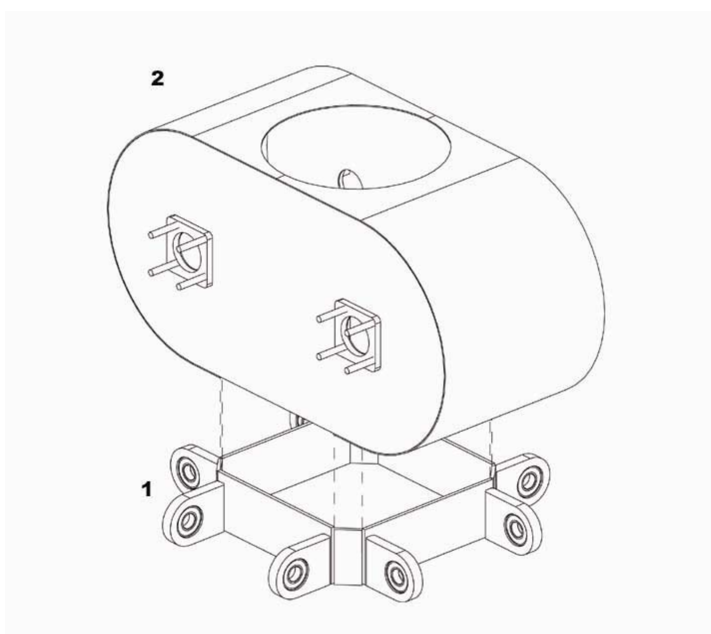


Figure 8.5 Housing assembly: 1) arm mounting ring & 2) cutter housing

8.2.2.1 Cutter Housing

The Cutter Housing (shown in Figure 8.6) is made of 16 gauge sheet steel all around. It consists of a front panel (front-left in Figure 8.6) to which the motors are attached, two identical side panels (sheet steel cut to shape then bent), and a rear panel to which the pulley reduction system for the conveyor belts is mounted. The mounting boss plates (labeled 2 and 4 in Figure 8.6) are welded to the housing sides to provide a stiffer interface for the mounts (5/16"-18 UNC threaded steel rods labeled 1 and 5 in Figure 8.6) to weld to. This part of the assembly is mostly the same as the original design. The only change made to this was removal of the original mounting holes, which were replaced by boss plates.

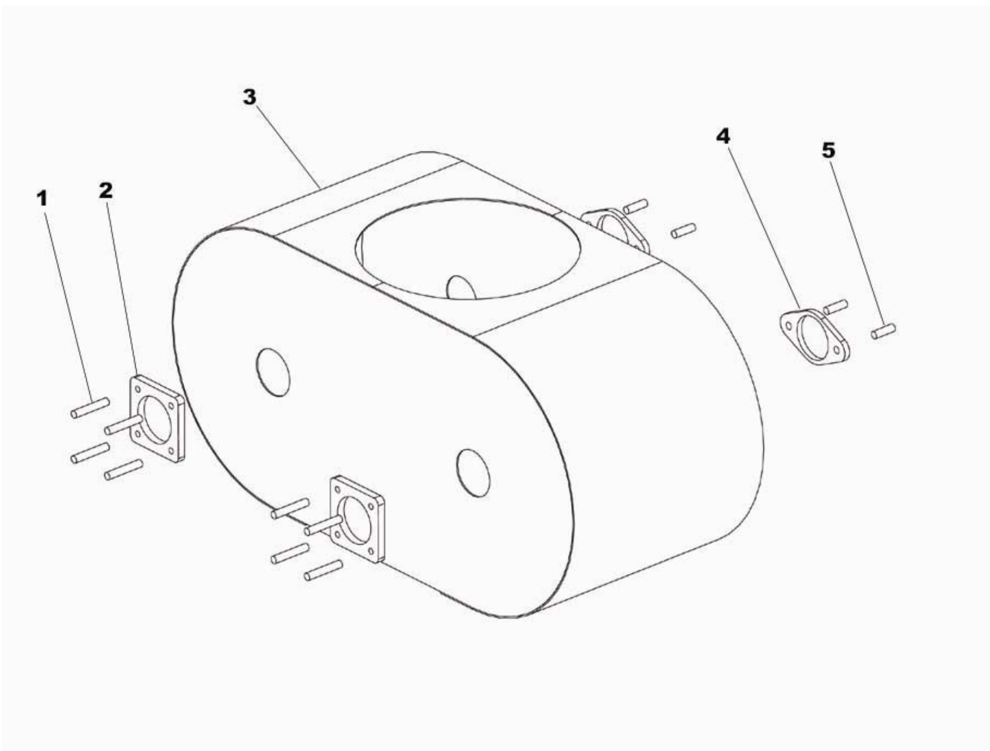


Figure 8.6 Cutter Housing: 1) Motor Mount, 2) Motor Mount Boss Plate, 3) Cutter Housing, 4) Pulley Mount Boss Plate, & 5) Pulley Mounts

8.2.2.2 Arm Mounting Ring

The lower portion (labeled 2 in Figure 8.5) of the Housing Assembly connects the Housing to the Arms. Since the arms are what will generate the majority of the load on the machine, this portion of the frame needs to be stronger than the cutter housing. The ring itself is constructed of 0.64 cm (0.24 in) thick steel all the way around. The diagonal corners are there to make clearance for the mitre gears used for the arm actuation. There are also eight custom machined pillow blocks welded to the main ring. These serve as mounting points for each of the four arms. A double sealed ball bearing is pressed into each pillow block and secured with a ¼”-20 set screw. It is critical that the centerlines of the bearings are co-planar and that the plane they lie on is parallel to the bottom of the mounting ring. This is important because the axes of the mitre gears used in the arms need to be aligned to function properly. A local metalworker was consulted about the feasibility of manufacturing this, and assured us that a fixture could be constructed to make the mounting ring while maintaining accuracy and minimizing residual stresses from welding operations.

8.2.3 Arms

The Tumbleweed Machine consists of three different arm designs: one plain arm, an arm with a conveyor belt (two of these are used), and one actuation arm. The actuation arm is the arm to which the hydraulic cylinder is attached, which actuates this arm, and in turn actuates the other three via a series of mitre gears. The parts common to all three arms will be described first, followed by descriptions of the three unique arms.

8.2.3.1 Arm Frame

The Arm Frame is the main structural element for all of the arms. It provides the length and strength needed to compress the tumbleweeds. The original version of the Arm Frame consisted of a section of bent 10 gauge 6061 aluminum with thinner aluminum plates on the inside (see Figure 8.10). This needed to be changed for two key reasons: Aluminum is not as easily welded as steel, and steel components needed to be attached via the use of fasteners instead of welding.

The new design of the frame consists of two elements: the Arm Frame and the End Caps. The difference in the Arm Frames between arms is simply the overall length. They all consist of the same bent sheet metal cross-section (only differing in length) and the End Caps. The End Caps are welded to the frame, which provide torsional stiffness as well as a good mounting surface for other components. The frame is made of 16 gauge AISI 1020 sheet steel, and the end caps are machined from AISI 1020 steel as well. These two components can be seen in Figure 8.7 below.

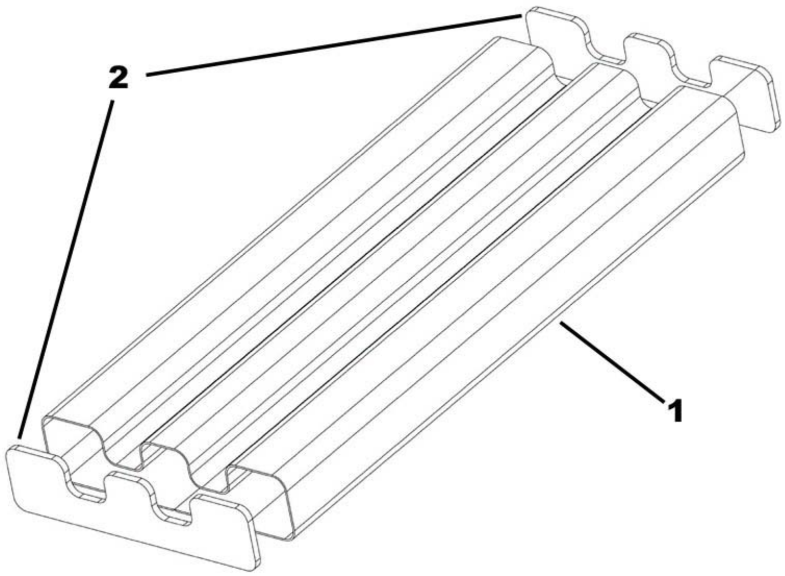


Figure 8.7 Arm Frame: 1) Frame & 2) End Caps

8.2.3.2 Gap Angles

The Gap Angles are small pieces of 16 gauge 1020 sheet steel that fill the two grooves in the Arm Frame to help prevent sliding objects from catching on sharp edges.

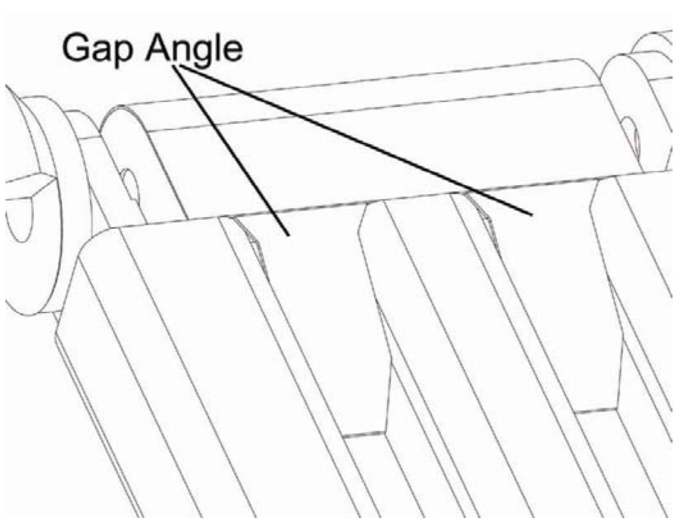


Figure 8.8 Gap Angles

The Gap Angle reduces the sharp 90 degree angle of the Gap Blocker to a 20 degree angle. This will greatly reduce the drag forces that the tumbleweeds encounter on their way to the housing.

8.2.3.3 Conveyor Belt Arms

There are two Conveyor Belt Arms on the tumbleweed machine. They provide the means to feed the tumbleweed into the housing. This arm (Figure 8.9) underwent heavy modification from the original design (Figure 8.10).

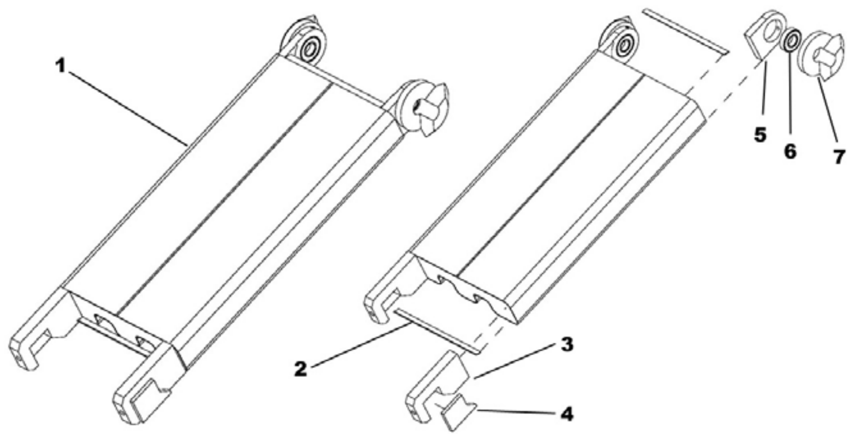


Figure 8.9 Conveyor Belt Arm: an unassembled arm is on the left, and to the right is an exploded view of the unassembled arm. 1) Arm Frame, 2) Belt Scoop, 3) Tension Bracket, 4) Bracket Stiffener Plate, 5) Pillow Block, 6) Arm Bearing, & 7) Mitre Gear

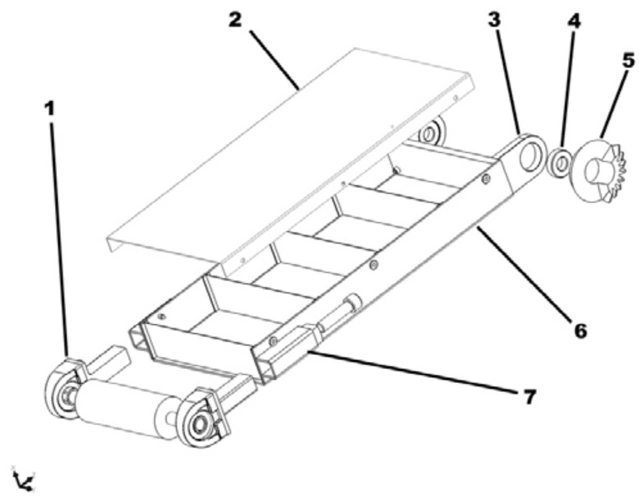


Figure 8.10 Old design of Conveyor Arm. 1) Take-up Frame, 2) Arm Cover, 3) Pillow Block, 4) Bearing, 5) Mitre Gear, 6) Arm Frame, 7) Take-up Housing

The Arm Frame underwent the most drastic change from the two design iterations. These changes are described in section 8.2.3.1. Flaws in the conveyor belt arms were that the conveyor belt was to be run underneath the cover, and on top of the interior frame plates causing a lot of drag and wear. Another was the conveyor belt take-up frame being very bulky and heavy.

8.2.3.3.1 Belt Scoop

The first flaw was fixed by changing the design of the arm frame as discussed before. Since the conveyor pulleys are tapered to keep the belt centered, the diameter at the end of the pulley is smaller than the thickness of the arm [by 3.2 mm (0.125 in)], thus causing the belt to be dragged on a corner of metal. To minimize drag and wear on the belt, small curved sections of metal (labeled 2 in Figure 8.9) called Belt Scoops were welded flush to the top and bottom of the Arm Frames at the ends, giving the belt a smoother curved surface to be picked up by.

8.2.3.4 Arm Mounting

Figure 8.9 shows the mounting system used for all four arms. The mounting system was designed so that all of the arms move synchronously. This was accomplished by using a series of mitre (or bevel) gears (two per arm for a total of eight). The axes of rotation of the four arms are all coplanar and form a square (these are the same as the four axes formed from the holes in the pillow blocks on the arm mounting ring seen in Figure 8.5). The three parts are a pillow block made of 1020 Steel, a double-sealed ball bearing, and a mitre gear made of 0.40 carbon steel with hardened teeth. The gear has a pitch diameter of 152.4 mm (6 in) with 24 teeth. Analysis of this gear will be done in a later section. The gear shown in the figure has a 240^{o} sector machined out of it as well as 20.6 mm (0.81 in) machined off the bottom of the hub for purposes of weight reduction [13.3 N (3.0 lbs) per gear]. This leaves 120^{o} of teeth, more than enough for the 75^{o} range of motion for the arms. The gear is first welded to the pillow block, and then this assembly is welded to the Arm Frame as shown in Figure 8.9. Finally the bearing is pressed into the pillow block and secured in place with a set screw.

8.2.3.5 Bare Arms

The other two arms do not have conveyor belts on them, primarily because it would be difficult to drive the belts, and secondly due to weight considerations. These arms serve to help compress the tumbleweed and funnel it into the housing. One of the bare arms is also the arm onto which a hydraulic cylinder is mounted in order to actuate the other four arms. These two arms have one thing in common that the conveyor arms do not: the Gap Blocker. This is simply to fill in the gap between the top of the arm and the side of the Arm Mounting Ring (seen in Figure 8.3 and Figure 8.11). The Gap Blocker is made of AISI 1020 sheet steel and is welded to the top of the arm.

8.2.3.5.1 Non-Actuating Bare Arm

The Non-Actuating Bare Arm shown in Figure 8.11 is the only arm on the machine that does not have a unique feature. Its only function is to compress the tumbleweeds. Its arm frame is 0.61 m (24 in) long, longer than the other frames since it does not have a pulley tensioning assembly at the end or a reinforced cylinder mount section.

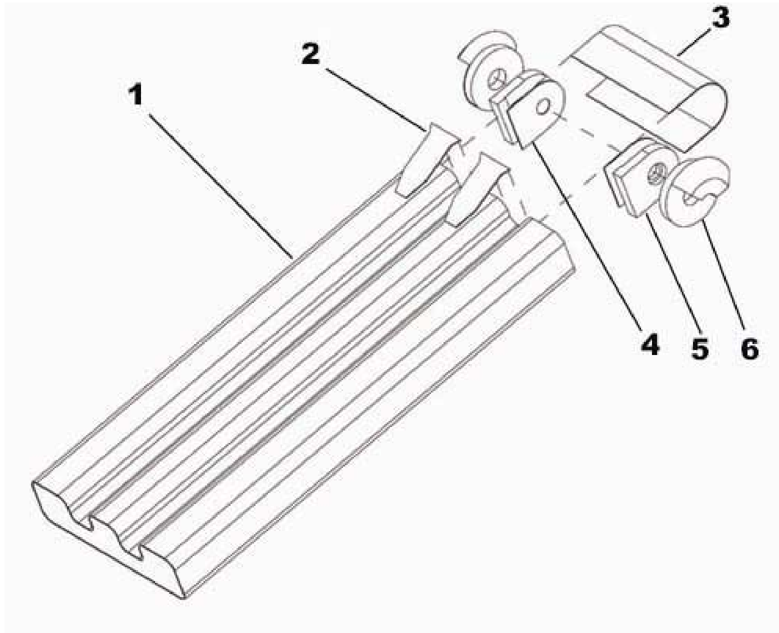


Figure 8.11 Non-actuating Bare Arm: 1) Arm Frame, 2) Gap Angle, 3) Gap Blocker, 4) Gap Blocker Side, 5) Shaft Block, & 6) Bevel Gear.

8.2.3.5.2 Actuating Bare Arm

The Actuating Bare Arm is very similar to the Non-Actuating Arm; however it has a reinforced upper section to which the hydraulic cylinder is attached. This short section holds the mitre gears, pillow blocks, and bearings that connect to the housing.

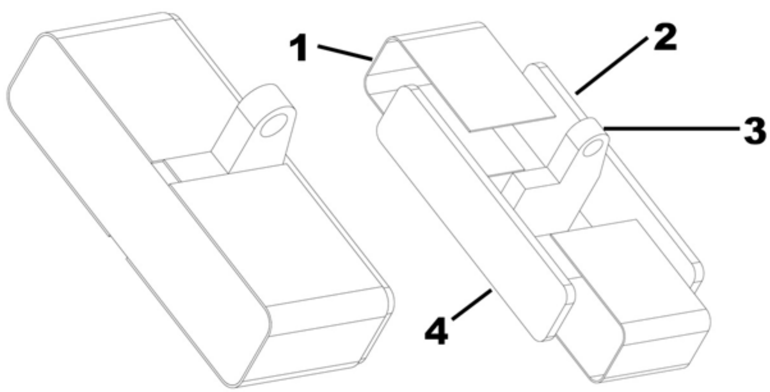


Figure 8.12 Cylinder Mount Section: 1) Cylinder Block Cover, 2) Lower Cylinder End Cap, 3) Cylinder Mount, & 4) Upper Cylinder End Cap

Figure 8.12 shows this shorter section with its four components. The hydraulic cylinder attaches to the Cylinder Mount through the top hole, and the mounting hardware is attached to the Upper Cylinder End Cap. The Lower Cylinder End Cap has a cut-out on the other side matching the cross section of the arm frames. This was done to give the frames a surface to positively identify its welding location (this can be seen in Figure 8.13). Figure 8.13 below shows the complete Actuating Arm assembly.

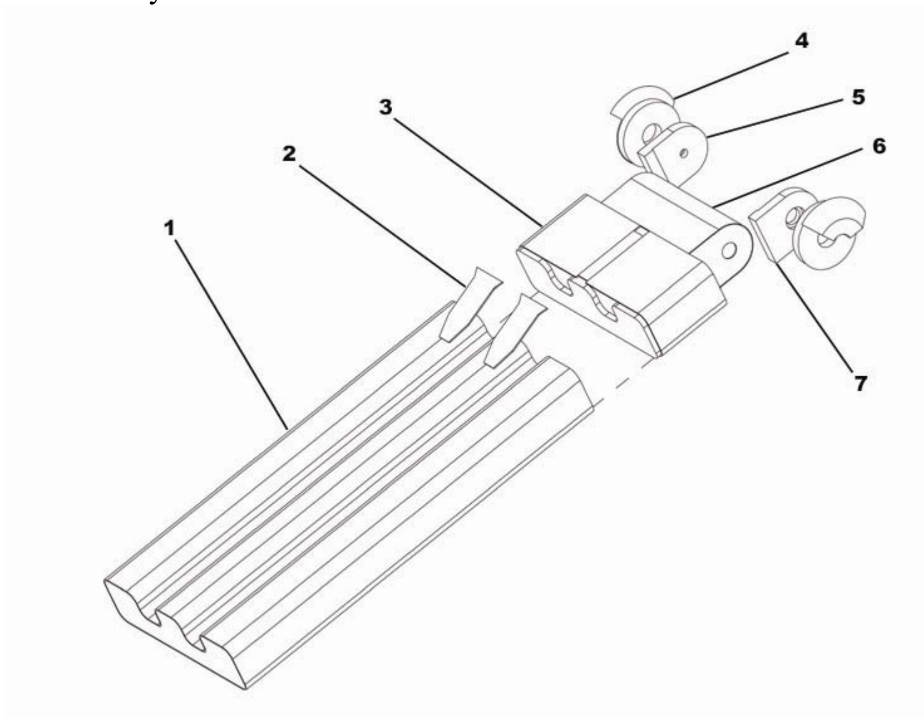


Figure 8.13 1) Arm Frame, 2) Gap Angles, 3) Cylinder Mount Block, 4) Bevel Gear, 5) Right Shaft Block, 6) Gap Blocker, & 7) Left Shaft Block

8.2.4 Assembly of Housing, Cutter, and Arms

This section of the chapter will explain how the three major components fit together. It will also cover the power train from the cutter shafts to the conveyor belts.

8.2.4.1 Connecting the Cutters to the Housing

Figure 8.14 shows a cutaway exploded view of how the Cutter is attached to the Housing Assembly. It should be noted that neither cutter can be attached to its mounting until the other cutter has been lifted inside the housing as well due to geometric constraints (despite Figure 8.14 showing otherwise; this was done to show a complete cutter mounting and an exploded cutter mounting). The design of this mounting system allowed for the cutters to occupy almost the entire width of the housing (22.9 cm out of 25.4 cm [9 in out of 10 in]) instead of having the input and output shafts occupy part of it. Because the Cutter Shaft is prevented from moving outward on both sides of the housing, it cannot move along its axis of rotation at all, thus fixing it in place only allowing for the rotation of the cutter.

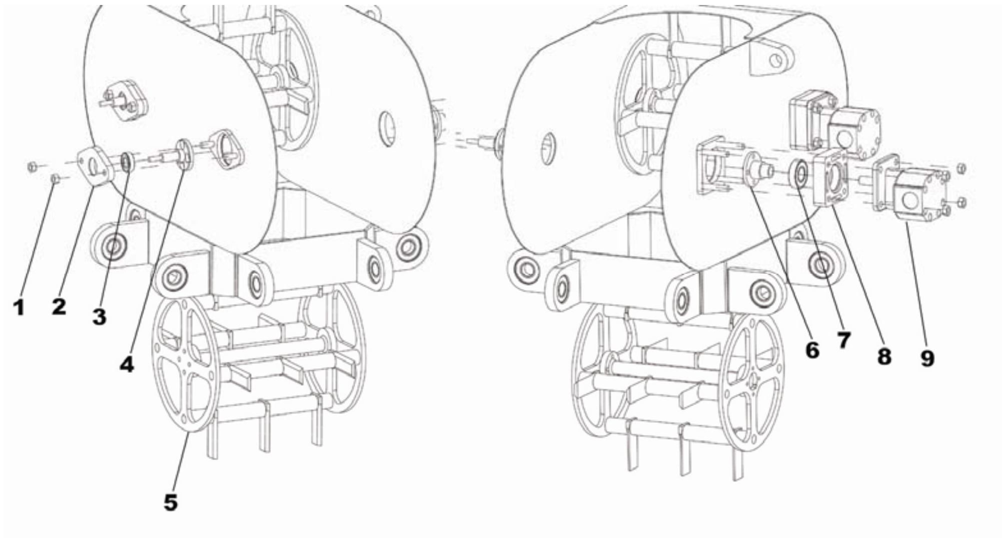


Figure 8.14 Cutter Mounting: 1) Mounting Nut (5/16"-18), 2) Pulley Bearing Mount, 3) Pulley Cutter Bearing, 4) Pulley Shaft, 5) Cutter, 6) Motor Coupler, 7) Motor Bearing, 8) Motor Mount, & 9) Hydraulic Motor

The left side of Figure 8.14 shows the pulley side mounting of the cutter. It consists of an output shaft that is made to drive the pulley system (explained in 8.2.4.4), a double sealed ball bearing, a bearing mount, two nuts, and four ¼"-20 screws not shown in the figure. Once the cutter is lifted into place within the housing, the pulley shaft (made of AISI 314 Stainless Steel) is screwed onto the cutter through the housing with the four screws (not shown). Next the bearing is slid over the shaft (machined for a slip fit with the bearing) and the pulley bearing mount (made of AISI 314 Stainless Steel) is slid onto the mounting rods and over the bearing, fastened with the two nuts.

The right side of Figure 8.14 shows the motor side mounting of the cutter as well as the motor mounting. Conceptually, it is the same mounting as the pulley side mounting; however, the input to the cutter is not a shaft but a 1.27 cm (0.5 in) diameter hole with a 0.32 cm (0.125 in) keyway for the motor shaft to be inserted into. The mounting consists of the motor coupler just described, four hex head ¼"-20 bolts not shown, a double sealed ball bearing, the motor mount, the hydraulic motor, and four nuts (same size as the ones on the pulley side). The four bolts (not shown) are used to fasten the coupler to the cutter. Then the bearing is slid over the coupler with a slip fit, and the motor mount is then slid over the 4 mounting rods and the bearing. The motor shaft is then fitted with a 0.32 cm (0.125 in) square key (3.18 cm [1.25 in] long) and inserted into the shaft coupler. The motor is then secured to the motor mount with the four nuts.

8.2.4.2 Connecting the Bare Arms to the Housing

The Bare Arms (section 8.2.3.5) both connect to the Housing in the same fashion (shown in Figure 8.15): The holes in the shaft blocks (Figure 8.11 and Figure 8.13) are first aligned with the bearings coming out of the flat face of the housing (the actuating arm is attached to the motor side and the non-actuating arm on the pulley side), and a 1.90cm (0.75 in) diameter shaft with a 2.54 cm (1.00 in) diameter shoulder on one end is inserted through the blocks and the bearings. The non-shouldered end of the shaft has a 3/8"-16 UNC hole drilled along the shaft axis. Once the shaft shoulder is resting on the shaft block, a washer and screw are fastened to the other end to prevent the shaft from sliding around on the bearings in its axial direction.

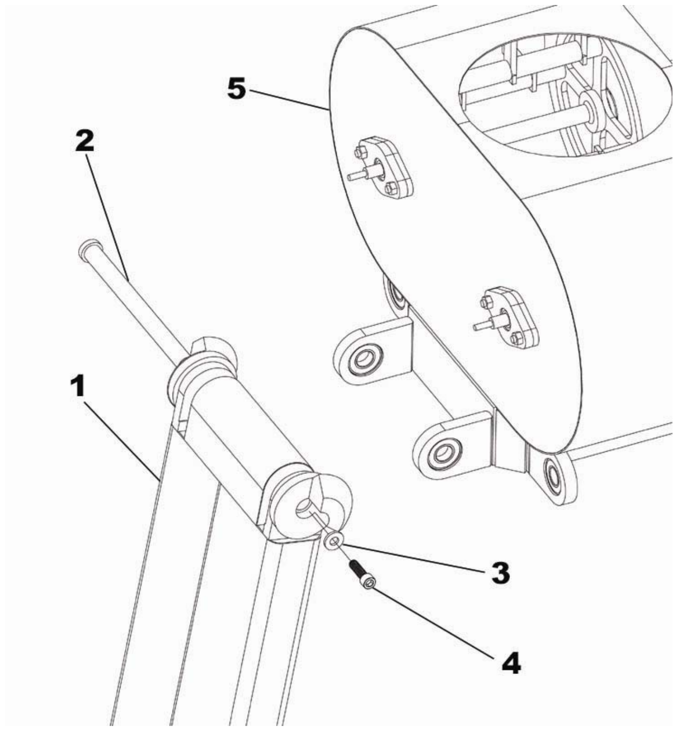


Figure 8.15 Bare Arm Connection: 1) Arm, 2) Arm Shaft, 3) Restraining Washer, 4) Retaining Screw, & 5) Housing.

8.2.4.3 Connecting the Conveyor Belt Arms to the Housing

Attaching the Conveyor Belt Arms to the housing is a bit more complicated than for the Bare Arms. This is due to the fact that the shaft that connects the arm to the housing is also the drive shaft for the conveyor belt, which means that the conveyor belt (not shown) needs to be on the arm (albeit loosely) when it gets mounted. Aside from that, these arms mount in the same fashion as the others as seen below in Figure 8.16. The drive shaft is machined with a small shoulder and has a washer acting as a shoulder on the other end to prevent lateral movement of the shaft when it's installed. The belt drive pulley, labeled 4 in Figure 8.16, is tapered to keep the belt centered, and has four counter-bored 0.64 cm (0.25 in) holes drilled intersecting the axis perpendicularly. These four holes line up with four ¼"-20 UNC holes drilled into the drive shaft. When installing, the belt needs to be folded over itself to expose the holes, and the ¼"-20 socket cap screws are screwed in to lock the pulley into place on the shaft.

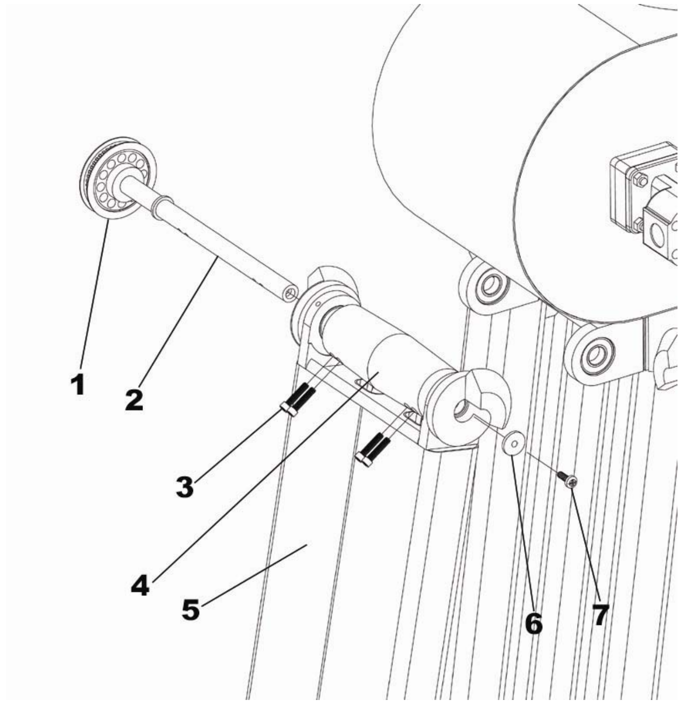


Figure 8.16 Conveyor Belt Arm Installation: 1) Conveyor Drive Pulley, 2) Pulley Drive Shaft, 3) Pulley Screws, 4) Drive Pulley, 5) Conveyor Arm, 6) Washer, & 7) Screw. Not shown: Conveyor Belt.

8.2.4.3.1 Conveyor Tensioning System

The new steel Arm Frame allowed for a custom take-up frame to be mounted to the end of the arm as opposed to the sides of the arm in the previous design. The take-up frame in the previous design (labeled 1 & 7 in Figure 8.10) is a scaled down version of what is standard in commercial conveyor belt systems. Even this scaled down version was bulky and heavy since it was mounted on the outside of the arms and used a commercial pillow block/bearing. A 3/8"-18 cap screw was threaded into the housing and pushed the take-up frame out until the belt was properly tensioned.

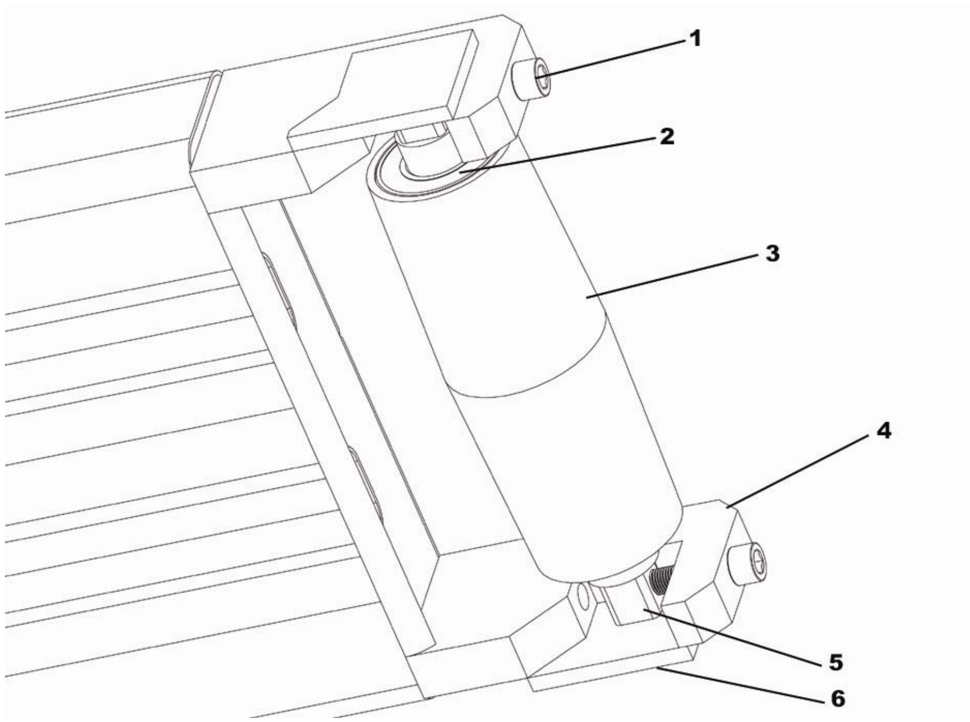


Figure 8.17 New Tensioning System: 1) Take-up Screw, 2) Pulley Bearing, 3) Pulley, 4) Tension Bracket, 5) Tension Pulley Shaft, & 6) Stiffener Plate

The new belt tensioning mechanism is shown in detail in Figure 8.17. This design relies on pulling the belt shaft instead of pushing it, and is made of 1020 steel. The pulley shaft has two 3/8"-24 threaded holes on each end (perpendicular to the shaft axis) that line up with the 9.5 mm (0.375 in) holes at the end of the tensioning brackets. Two 3/8"-24 cap screws are then used to pull the shaft until the belt is properly tensioned. The stiffener plate is added to strengthen the bracket and is a 6.6 mm (0.25 in) thick plate of 1020 steel welded to the bracket.

8.2.4.3.1.1 Pulley and Shaft

The Tensioning Pulley shown in Figure 8.17 is made of 63.5 mm (2.5 in) 6061 Aluminum (to reduce weight) round stock. The diameter is tapered by 6.4 mm (0.25 in) at both ends to keep the belt centered. The inside of the pulley is bored out to allow the shaft to pass through, and two holes are drilled at each end into which a sealed ball bearing will be pressed. These 2 bearings allow for the pulley to rotate about the shaft.

The Tensioning Shaft is made of 31.8 mm (1.25 in) round Stainless Steel stock. The two ends are machined square, and 3/8"-24 holes are drilled and tapped into the square ends to accommodate the tensioning screws. The shaft diameter is 25.4 mm (1 in); however, there is a small shoulder on the shaft that is 31.8 mm (1.25in) in diameter to prevent the pulley bearing from sliding around. Movement of the pulley in the other direction is prevented by a 3.2 mm (0.125 in) hole and a cotter pin (not seen in figures above).

8.2.4.4 Installing the Pulley System (Conveyor Belt Drive)

One of the unique features of the Tumbleweed Machine is that it uses one hydraulic motor to power a single cutter and conveyor belt. The shaft of the cutter transfers the power from the motor to a system of timing belt pulleys on the other side of the housing. The timing belts are the XL Series from McMaster, with a pitch of 0.51 cm (0.2 in). One belt has 88 teeth and the other has 100.

The first pulley in the system is an 11 tooth pulley that is driven by the cutter shaft. This drives the composite pulley in the system, which consists of a 72 tooth pulley with an 11 tooth pulley welded to it concentrically. The 11 tooth pulley of the composite pulley drives a 44 tooth pulley on the conveyor drive shaft. This series of pulleys yields a gearing ratio of 0.038 from the following equation (where n_i is defined as the number of teeth on gear i):

$$\frac{n_1}{n_2} \bullet \frac{n_3}{n_4} \qquad (2\text{-}1)$$

Figure 8.18 shows an exploded view of one of the pulley systems and a fully assembled view on the other side. The Composite Pulley Housing (labeled 2 in Figure 8.18) is first welded to the housing at a 32^{o} angle from the horizontal. This housing consists of an AISI 1020 steel block with a hole bored in it to fit two 1.27 cm (0.5 in) inner diameter ball bearings. The underside is machined out to reduce its weight, and on the top are four ¼"-20 UNC tapped holes drilled for the cover plate to be attached. Two tensioner brackets (protruding element on the side of the housing) are welded to two sides of the housing. These brackets are on the line of action of the pulleys, and serve as the basis for the tensioning system explained in 8.2.4.4.1. The lines of action can be seen on the right side of Figure 8.18 and are perpendicular to each other.

After the housing is in place, the composite pulley is fitted with a bearing on the top and bottom, and slid into the housing. The bending load caused by the belt tensions on the shaft is the reason for using two ball bearings. The top bearing on the shaft is flushed with the top of the pulley housing and held in place by the housing cover. The cover is fastened with four ¼"-20 flat head screws. The composite pulley is then secured to the shaft by means of a set screw. Finally the small 11-tooth pulley (labeled 1 in Figure 8.18) is slid onto the shaft and secured with a set screw.

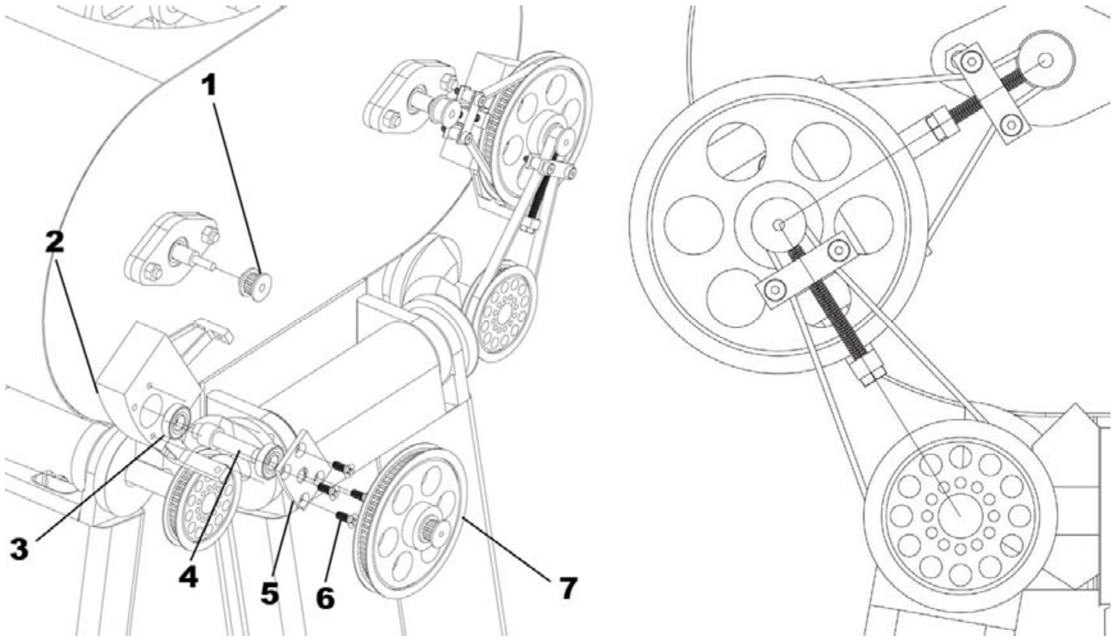


Figure 8.18 Left: Installation of pulley system: 1) 11-Tooth Pulley, 2) Composite Pulley Housing, 3) Composit Pulley Bearing, 4) Composite Pulley Shaft, 5) Pulley Housing Cover, 6) Pulley Housing Cover Screw, & 7) Composite Pulley (11 & 72 Tooth Pulleys). Right: Frontal view of pulley system.

8.2.4.4.1 Pulley Tensioners

The original design of the Tumbleweed Machine actually lacked a proper tensioning system for the pulleys. This tensioning system works by sliding two small rolling bronze bushings that are close to each other along the line of action of the pulley, in effect pinching the belt together. As the tensioner gets closer to the large pulley, the belt will become tighter. This effect can be seen on the right side of Figure 8.18.

Figure 8.19 shows an exploded view of a belt tensioner. It consists of a machined tensioner, two shoulder bolts, two bronze bushings (tensioner bearings), and an adjusting bolt (1/4"-20 hex cap screw). The tensioner has a threaded ¼"-20 UNC hole in the middle through which the adjusting bolt goes. The timing belt goes around the center and between the wings of the tensioner. The wings have holes drilled through them to accommodate the shoulder bolts, which hold the bushings in place and provide a smooth surface for them to roll on. The shoulder bolts are held in place by two nuts (not shown) on the underside of the wings.

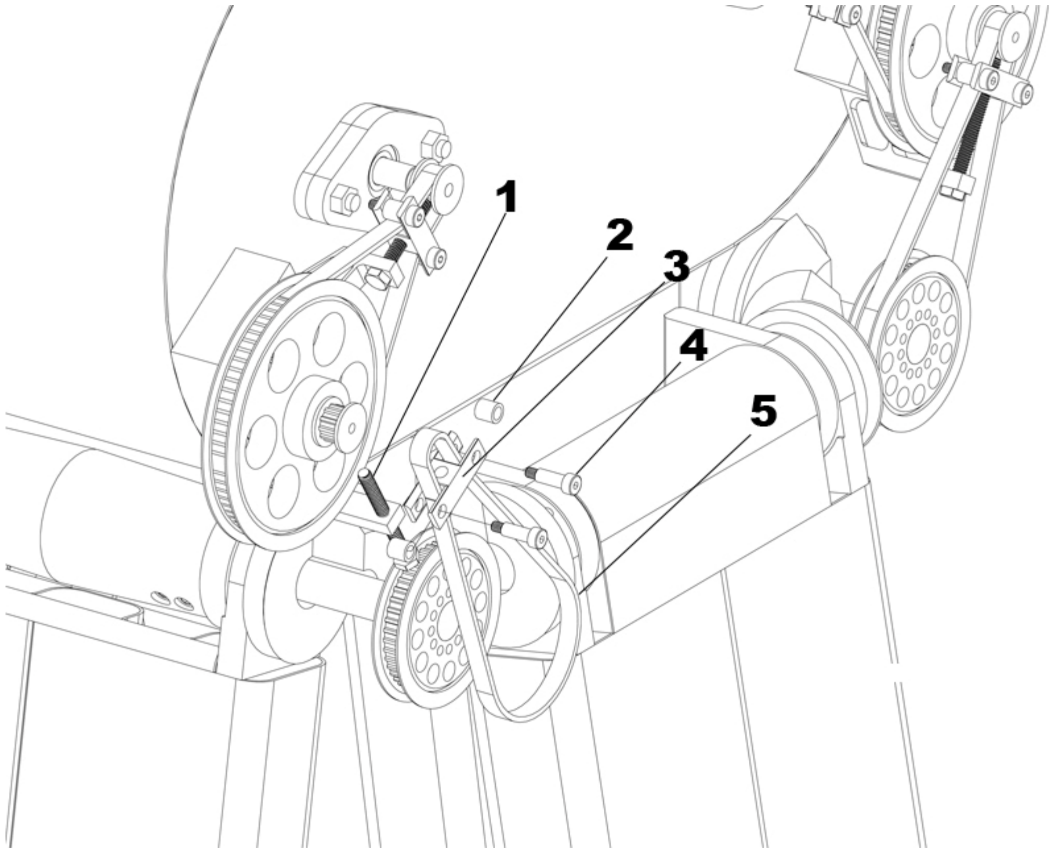


Figure 8.19 Pulley Tensioning System: 1) Tension Adjusting Bolt, 2) Tensioner Bearing, 3) Belt Tensioner, 4) Tensioner Shoulder Bolt, & 5) Timing Belt.

8.2.4.5 Actuating the Arms

As explained in 8.1.4, the arms are actuated by a single hydraulic cylinder. This is seen below in Figure 8.20. The cylinder is from the Wizard line made by Prince Manufacturing Corporation. This cylinder was selected because it is very low profiled and light weight compared to competing cylinders. Another feature of it is the customizability of the base end and the rod end attachments of the cylinder, in that the cylinder comes without them, and we can fit it with what works best with the design. In this case, a clevis with a pin is used at each end. The cylinder itself has a 3.81 cm (1.5 in) bore with a 10.16 cm (4.0 in) stroke length. This accommodates approximately a 75^o range of motion.

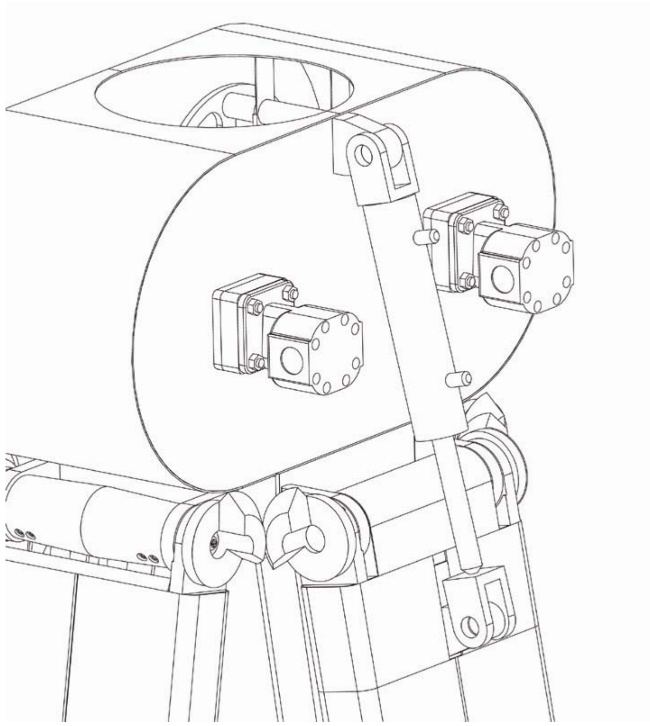


Figure 8.20 Actuating Cylinder

8.2.5 Interface to the End-Effector

The interface/connection to the End-Effector (EF) was intentionally left open. As of now, the top surface of the Tumbleweed Machine consists of a flat surface with a 22.86 cm (9.0 in) hole in it. The reason for not designing an interface is that an automated tool changer is among the projects proposed here at AHMCT and that interface has yet to be designed. Harker and McPhee [16] designed an interface for their EF tools (Rotary Impact Cutter and Oscillating Cutting System respectively) seen below in Figure 8.21. The two upper brackets hang from the EF and are bolted to the upper mounting ring. The problem with this mounting design is that it is very long and when coupled with the overall length of the tumbleweed machine (1.0 m [39 in]), this becomes an extremely long attachment.

One possibility of using Harker's/McPhee's mounting design is only using the two upper brackets and attaching them to the top surface of the Tumbleweed Machine. Another more drastic solution to the mounting problem is removing the inner tube from the EF since this machine does not need the motion provided by it, and to simply attach it to the main tube of the EF (interested readers can refer to Porterfield's thesis [21] for more detailed information on the EF). This however would make the tumbleweed machine impossible to include in any automated tool changer.

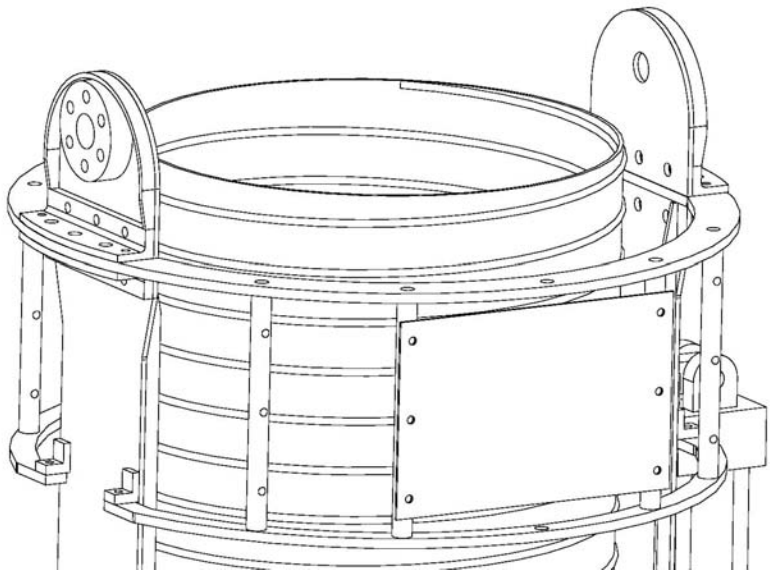


Figure 8.21 EF Mount Ring from Harker/McPhee.

8.3 Summary

This chapter provided an in depth description of the concept and design of a Tumbleweed Mulching Device (also called Tumbleweed Machine) to be connected to the ARDVAC. The design work done for this weldment was improving upon an existing design done by Au and others. This new design, however, is still a work in progress. Areas that still need to be addressed are the mounting surface, an automated control system, and the hydraulic system.

CHAPTER 9 CONCLUSIONS AND RECOMMENDATIONS

9.1 Conclusions

The development of vegetation cutting attachments is a logical extension to the development of the ARDVAC, a machine system that will improve the safety and effectiveness of CALTRANS highway maintenance operations. The problem of collecting vegetation was defined as a valuable feature during initial testing with the ARDVAC prototype, and the cutting attachments developed for this project and presented in this report are optimal solutions to the problem given the knowledge available at this time. The concepts were selected and developed using engineering design principles and processes that have been refined and used successfully at the Advanced Highway Maintenance and Construction Technology (AHMCT) Research Center at the University of California at Davis (UC-Davis). The application of robotics and automation technology together with current proven maintenance methods is an engineering challenge that will ultimately lead to a safer worker environment, while improving work conditions and worker effectiveness.

9.2 Recommendations for Testing, Evaluation and Development

Caltrans has initiated the purchase of the first ARDVAC machines, which are presently available through CleanEarth LLC and are expected to be delivered in 2006. Investing in the machines represents a major commitment by the state of California to the goals of improved worker safety and effectiveness and the advancement of the application of technology. These machines are projected to be used in water quality control efforts and trash collection in hazardous zones. It is reasonable to expect that these machines will be used for a variety of new tasks that take advantage of the dexterity of the ARDVAC nozzle. The ARDVAC machines will ultimately be required for testing of the tools developed in this project and any other similar designs.

Limited in-house laboratory testing and limited field testing will be pursued to maximize the benefit of the effort and investment of this project to date. Since full scale testing without the ARDVAC machine is not ideal, further development of tools is limited to concept development until further integration of the ARDVAC into the Caltrans fleet. Once the ARDVAC has been obtained and tested within Caltrans, it is recommended that the ARDVAC vendor be involved in further development. These tools should be recognized in all investigations related to roadside maintenance and considered for the ‘tool box’ of the future.

Since integration of the ARDVAC into the Caltrans fleet is yet to occur, continuing efforts to projecting the benefits versus costs should be pursued in order to support deployment of the system. By making the effort to formally quantify the value of a system such as this, Caltrans will be able to more quickly bring technologies such as this into use. In the immediate future, engineers, machine operators and others should be closely monitoring the use of the machines in order to more completely develop the potential for tele-robotic systems that will improve safety and effectiveness.

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9.3 Final Comments and Future Goals

Future goals of the AHMCT Center will be to support Caltrans maintenance operations in the evaluation and integration of the ARDVAC machines. The use of automation as represented by the ARDVAC and the tools developed in this project is a significant leap forward in roadside maintenance operations. Many unforeseen benefits and challenges to implementation can be expected, but the efforts reported here represent a valuable foundation to the development effort. The ARDVAC based system will have continued value as both a system for vacuuming and a system to manipulate tools such as those described here. This type of system is by design, expected to have unique applications in the road maintenance and construction industry, and it is recommended that Caltrans and others responsible for this work continue to investigate the potential of this and similar machines.

REFERENCES

[1] Anchor Lamina Inc., Lamina Hydraulic Motors, 5, Dec. 2003, 4, http://www.anchorlamina.com/pdf.d/hydrmotr.pdf

[2] Bieniek, R.S. Personal Communication. 2004

[3] Bioscape.com, 14 Oct. 2004, http://www.bioscape.com/asccustompages/products.asp?categoryid=4

[4] Broadbent, Jack, Telephone interview, 15 Oct. 2004

[5] California Department of Transportation (Caltrans) 12, Aug. 2003, http://www.dot.ca.gov

[6] California Department of Transportation (Caltrans), Maintenance Manual, 5, Aug. 2003, http://www.dot.ca.gov/hq/maint/manual/maintman.htm

[7] "California Exotic Pest Plants of Greatest Ecological Concern, 1999", 4, Sep. 2002, California Exotic Pest Plant Council, http://www.caleppc.org/

[8] "Controlling Weeds in your Lawn and Garden: You really don't need all of those chemicals, do you?", 21, Jan. 2000, The Garden Helper, 12, Oct. 2004, <http://www.thegardenhelper.com/weeds.html>.

[9] Dieter, G.E. Engineering Design A Materials and Processing Approach. 3rd ed. Boston: McGraw-Hill, 2000

[10] Freer, Roy. Telephone interview, 27 Aug. 2003

[11] G. S. Pullman, et al., Soil Solarization: A Non-chemical Method for Controlling Diseases and Pests, 27, Jan. 2003, (University of California: Small Farm Center. Feb 1984) 1, http://www.sfc.ucdavis.edu/library/details.asp?pub_ID=1826

[12] Harker, Kenneth R. "Mechanical Design and Analysis of a Rotary Impact Cutting Fixture for an Automated Roadway Debris Vacuum", (Thesis, UC Davis, 2004)

[13] Hibbeler, R.C. Engineering Mechanics Dynamics. 8th ed. New Jersey: Prentice Hall, 1998

[14] Houston, Rick and Dave Beach, Personal interview, 2 Sept. 2003

[15] Latham, M.W. Detail Design and Testing of a Vegetation Processing End Effector for an Automated Roadway Debris Vacuum Vehicle. Thesis. University of California, Davis. 2003

[16] McPhee, J.A. Personal Communication. 2004

[17] McRandal, D.M. and P.B. McNulty. "Impact Cutting Behavior of Forage Crops." Journal of Agricultural Engineering Research, 23 (1978): 313-328

[18] McRandal, D.M. and P.B. McNulty. "Mechanical and Physical Properties of Grasses." Transactions of the ASAE. (1980): 816-821

[19] Norton, R.L. Machine Design An Integrated Approach. 8th ed. New Jersey: Prentice Hall, 2000

[20] Oregon Department of Transportation (ODOT), EVALUATION OF INFRARED TREATMENTS FOR MANAGING ROADSIDE VEGETATION, Online Image, Dec. 2000, Oregon Department of Transportation, 10 Sept. 2004, 11, http://ntl.bts.gov/sec508/infrared_veg.pdf

[21] Porterfield, A.A. Development of a Telerobotic Hose Positioning System for an Automated Roadway Debris Vacuum Vehicle. Thesis. University of California, Davis. 2001

[22] Ransom, Viveka. "Corn Gluten Meal -- Byproduct to Wonder Product", Iowa State University, 15 Oct. 2004, http://www.**gluten**.iastate.edu/pdf/iowahort.pdf

[23] Richardson, B.A. Conceptual Design of a Vegetation Processing End Effector for an Automated Debris Vacuum Vehicle. Thesis. University of California, Davis. 2002

[24] Caltrans Roadside Management Toolbox http://www.dot.ca.gov/hq/LandArch/roadside/

[25] STIHL Inc., 2003, STIHL: Power Tools and Accessories, 10, Sept. 2003, http://www.stihlusa.com

[26] Thomsen, Craig and George Hartwell, Prescribed burning for invasive weed management: The Bear Creek Burn, California Department of Transportation, 14, Aug. 2003, http://www.dot.ca.gov/hq/LandArch/CaliforniaWILD/BearCreekBurn.pdf

[27] Toogood, R. Pro/MECHANICA Tutorial Structure. Release 2001 – Integrated Mode. Edmonton, Alberta: ProCAD Engineering Ltd., 2001

[28] Ulrich, K. T and Eppinger S. D. Product Design and Development. McGraw-Hill, Inc., 1995

[29] U.S. Consumer Product Safety Commission. Office of Information and Public Affairs. "CPSC, Weed Wizard Acquisition Corp. Announce Recall of Weed Trimmer Heads With Metal Chains." News from CPSC. June 21, 2001 http://www.cpsc.gov/cpscpub/prerel/prhtml00/00104.html

[30] United States Department of Transportation - Federal Highway Administration (US DOT-FHA), Oregon Goats Gnaw on Knapweed, United States Department of Transportation - Federal Highway Administration, 14, Oct. 2004, http://www.dot.ca.gov/hq/LandArch/roadside-detail-gm.htm

[31] Velinsky S.A., 1998, "A Proposal for: Development of a Telerobotic System for Debris Vacuuming Positioning", 1998