GEORGIA DOT RESEARCH PROJECT 10-25
FINAL REPORT

DEVELOPMENT OF A METHOD TO REMOVE RAISED-PAVEMENT MARKERS (RPMs) FROM ROAD SURFACES

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16. Abstract:
The Georgia Department of Transportation (GDOT) uses raised pavement markers (RPMs) widely on roads throughout the State to increase road safety. Each of the approximate 3 million RPMs in Georgia was placed manually. Unfortunately, RPMs do not last as long as the road surface meaning they need to be replaced several times throughout the life of a road. There is a strong desire to remove the RPMs prior to placing new ones. GDOT contracted with the Georgia Tech Research Institute (GTRI) for a feasibility study of new methods of removing RPMs.

Currently, RPM removal is a manually intensive process. To remove RPMs, workers use either a reciprocating chisel or an eradicator/scarifier. The approach that GTRI proposed was to use a high speed machining technique to mill the marker off of the road surface. This technique had not been attempted before and the machining rates were well beyond standard machining operations published in literature. Due to the unknowns, a prototype cutting cell was developed that demonstrated the ability to machine markers as proposed. This operation of the concept was confirmed through testing but some challenges that were highlighted from testing are discussed. Lastly, a conceptual design was completed that contains features needed to adapt the laboratory prototype into a mobile platform such as a truck.

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Final Report

DEVELOPMENT OF A METHOD TO REMOVE RAISED-PAVEMENT MARKERS (RPMs) FROM ROAD SURFACES

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Executive Summary

The Georgia Department of Transportation (GDOT) uses raised pavement markers (RPMs) widely on roads throughout the State to increase road safety. Each of the approximate 3 million RPMs in Georgia was placed manually. Unfortunately, RPMs do not last as long as the road surface meaning they need to be replaced several times throughout the life of a road. There is a strong desire to remove the RPMs prior to placing new ones. GDOT contracted with the Georgia Tech Research Institute (GTRI) for a feasibility study of new methods of removing RPMs .

Currently, RPM removal is a manually intensive process. To remove RPMs, workers use either a reciprocating chisel or an eradicator/scarifier. The approach that GTRI proposed was to use a high speed machining technique to mill the marker off of the road surface. This technique had not been attempted before and the machining rates were well beyond standard machining operations published in literature. Due to the unknowns, a prototype cutting cell was developed that demonstrated the ability to machine markers as proposed. This operation of the concept was confirmed through testing, but the research did highlight a flaw in the initial approach taken with the cutting prototype, which was the existence of sand captured in a composite matrix in particular markers. The existence of sand led to dulling of the cutting edge nullifying the ability to use basic tool steel as a cutting tool for this application. Although this issue became a major limitation to completing data capture as initially planned, the general concept is still valid. Additional research would be required to build and test a new blade design leveraging different materials with the special consideration of making the cutting edge harder than sand. This can be achieved by using carbide materials such as titanium carbide, which is a common material used for machining.

Lastly, a conceptual design was completed that contains features needed to adapt the laboratory prototype into a mobile platform such as a truck. Although only a single cut was successfully performed with the system prior to dulling the blade on a sand filled marker, the general findings support the notion that machining RPMs from the roadway is a feasible approach to road maintenance.

Acknowledgements

GTRI would like to extend their thanks to all those at GDOT that have helped to make this research possible from the initial decision to fund this project to the technical manager. The technical manager, Mr. David Jared, has been extremely helpful by staying engaged in the project through detailed review of all reports delivered throughout the project. The regular involvement of the sponsor in this RPM removal development helped to improve the quality of the research performed.

Additional thanks go to the members of the Internal Research and Development (IRAD) team at GTRI of Mr. Steven Robertson and Mr. Wiley Holcombe. Their hard work helped lay the groundwork for the initial feasibility tests that led to this sponsored work.

Introduction

The Georgia Department of Transportation (GDOT) uses raised pavement markers (RPMs) widely on roads throughout the State to increase road safety. Each one of the RPMs is placed manually. Though GDOT has not yet had an occurrence of an injury during RPM placement, GDOT employees consider placing RPMs to be one of the highest risk jobs. The person placing RPMs is riding on a seat about 6 inches off the ground, cantilevered off the side of a trailer adjacent to traffic, even while placing RPMs along interstate highways.

There are about 3 million RPMs in service in the State of Georgia. GDOT would like to replace RPMs on an 18-month cycle. RPMs must often be replaced after snowplow use, after pavement crack sealing, or after lane restriping. Three central crews and 6 district crews install approx. 900,000 markers per year. Furthermore, GDOT desires that RPMs are removed prior to or during the process when new markers are placed on the roadway.

Currently, RPM removal is a manually intensive process. To remove RPMs, workers use either a reciprocating chisel or an eradicator/scarifier. With a reciprocating chisel, the operator manipulates the chisel edge underneath an RPM to pry it off the road. A scarifier falls under the category of surface preparation equipment for asphalt or concrete. GDOT generally uses it to remove paint striping from roadways. It has a rotating drum with carbide cutters and is powered electrically, hydraulically or by a gas engine. The scarifier grinds the RPM as it is pushed over it. This generally requires multiple passes at gradually lower heights to completely remove the RPM from the roadway.

After understanding the needs of GDOT, The Georgia Tech Research Institute (GTRI) began an effort to address the challenges of RPM removal through an Internal Research and Development (IRAD) effort. This IRAD was initiated to evaluate the performance of using milling operations as opposed to the use of a scarifier for RPM removal. To estimate the power required to machine the RPM from the roadway, GTRI generated an order-of-magnitude estimate of the power required. The result of building and testing a small scale cutter resulted in an estimated power to machine it in 40 milliseconds of around 200 HP. The 40 millisecond time is based on the time to travel 3.5 inches, the length of the RPMs manufactured by 3M Corporation™, at 5 mph.

The results of these efforts led to confidence in being able to machine RPMs from the road at the feed rate expected for a vehicle moving at 5 mph. At this point, a proposal was submitted to GDOT for the proof of concept prototype related to the machining of RPMs.

This final report details the design, fabrication and testing of the previously mentioned RPM removal prototype. Most importantly, the results of testing exercises proved the concept is feasible by being able to machine RPMs at accelerated speeds similar to what would be expected in operation on a plastic marker. However, challenges remain in addressing sharpness and wear of the cutting surfaces in typical operation such as when cutting sand filled markers. The last section of the report covers the development of a conceptual design or path forward to transition the prototype laboratory cutter to a mobile platform such as a truck or trailer.

Objectives

The objectives of the proposed effort included:

- Work with GDOT to develop a specification for RPM removal.
- Scale up the GTRI-proprietary cutter design from the existing 2-inch-diameter by 0.220-inch-wide cutter to a full-scale (approximately 12-inch-diameter by 5-inch-wide) cutter.
- Instrument and test the full-scale cutter on RPMs using an electrically-driven spindle and base fabricated for this purpose.
- Evaluate test data and refine requirements for a RPM removal system.
- Develop conceptual design for vehicle and supporting systems necessary for RPM removal using the GTRI-proprietary cutter.
- Identify commercial sources for major system components including high-speed spindle, vacuum system, and vehicle platform for RPM removal system.

System Requirements

Operational System Requirements

The project proposal contains the following list of requirements for an operational system:

- Remove the majority of the RPM to within a fraction of a 1/16 inch above the surrounding road surface.
- Collect the RPM and transfer into a container for disposal.
- Average speed will be 5 mph.
- Operate only in dry conditions.
- On multi-lane highways, remove the single RPM along the dividing skip lines.
- On two lane highways, remove one or both RPMs along the centerline in one pass. This will allow for concurrent removal and replacement operations.
- Replacement RPMs will not be placed at the same location as a previous RPM. The placement operation will provide for offsetting new RPMs from the previous placement location.
- Accommodate obstacles such as manhole covers, rumble strips, etc.
- Minimize damage to paint stripes.
- Minimize overhang into adjoining lane.

On May 9, 2011, a meeting was held with Georgia DOT representatives to review the systems requirements. The results of that review are included below. Two topics from the May 9 discussion are most relevant to the cutter development effort. One is the suggestion to make measurements of the height above the surface of installed markers. The second is to assess the cutter blade wear and to consider methods that might be used to evaluate cutter wear on cutters that are in service.

Requirements for RPM removal system agreed upon at May 9, 2011 meeting

Several requirements were discussed that resulted in the following list of considerations and/or requirements. Most of these only affect the system level conceptual design; however, some of the requirements will be implemented into the laboratory testing device being built under the contract.

1. The RPM removal system needs to be able to accommodate concrete and asphalt based pavements. In particular, the heaves that are present in concrete

surfaces and the height changes where asphalt lanes meet need to be considered in the system prototype design. One operation agreed upon was that the cutter should be lifted between cutting activities to limit the chance of accidental crashes or cutting of items such as the steel in the expansion joints of bridges.

2. A target of 1/16” maximum material remaining on the road seemed reasonable. There is some concern that on concrete pavement surfaces the 1/16” may not be enough to fully remove the marker and leave some plastic visible on the adhesive. A suggestion was made to support a small field exercise that would provide traffic control to take measurements on marker heights and adhesive thicknesses on different pavement types. A specialized height gauge could be fabricated to make this measurement easy and repeatable to capture several thicknesses on different marker and pavement types.

3. The thickness of the thermoplastic material can be as much as .090”, which is thicker than the 1/16” (.062”) that will be used in the design. Some provision may need to be made to prevent milling the paint stripe on the conceptual design.

4. Most major roadways will have a maximum of 2 to 3 markers at a single location on the road. The goal of the system will be to remove all these markers at one time, and at the desired speed of 5mph. There are some cases where 6+ markers can be in the same location on less travelled roads. In these areas, it was agreed upon that the speed could be varied and slowed to 3 mph at times to allow the system to handle these situations. The case where there are 6+ markers was considered an exceptional case.

5. Removal of the marker will likely generate fine plastic particles after the removal operation. Opinions were expressed as to whether or not these plastic particles would be a problem. Possible solutions discussed included blowing the particles from the roadway and incorporating a vacuum system into the marker removal system to collect the plastic particles as they are generated.

6. Several situations were discussed as far as marker placements on the road and the need for flexibility to handle double lines, skip lines, etc. The two cases that need to be supported are skip lines where a single marker is placed on every other skip line and a solid single line where two markers are placed approximately 12” apart. Although there was discussion about addressing the single line case with a cutter, this may be undesirable due to the potential for uneven pavement on a paving seam.

7. Overhang needs to be considered carefully. The current overhang of the RPM placement system camera was not preferable. Overhang will need to be minimized in order to minimize impact to traffic in the adjacent lane.
8. One goal of the laboratory prototype tests is to characterize the wear of the cutter blade. The diameter of the blade can be measured precisely then tool wear equations will be used for building a model to estimate the wear properties of the cutter. It is essential that some estimate of wear is made as the viability of the unit will depend on the ability for the cutter to survive some reasonable length of time. There was also discussion that the existing cutter could potentially be remachined and reused several times. This was an interesting suggestion and one thought was this could possibly be accomplished with electrical discharge machining operations. This also created an implied requirement of making the cutting blade easily replaceable. One final consideration for wear is how to inspect the wear on an existing blade. Perhaps there could be wear indicators present on the blade to provide maintenance staff with a visual indicator of extreme wear.

The previously described requirements list will serve as design guidance for the test device as well as the conceptual design.

Experimental System Design

Results of GTRI-funded IRAD project

During the course of the GTRI-funded IRAD project, the investigation focused on either removing whole RPMs with a blade-type device or machining RPMs into chips with a scarifier type device. Any device that removed the whole RPM was eliminated based on GDOT's experience with a similar device and its potential for damaging the road.

The resulting recommendation is a solution that would remove RPMs by machining them from the roadway. In practice, the removal device might have one or more cutting heads. Each cutting head would have a cutter or stack of cutters about 6 inches wide. The cutter(s) width should exceed the width of the widest RPM by at least 1 inch. A single head cutter would have to machine a RPM in about 40 milliseconds. However, the energy to machine a RPM is not substantial; the power to machine it in 40 milliseconds is still over 200 HP. One potential solution is to use the cutter as a flywheel to store the energy. A 12-inch-diameter cutter, 6-inches-wide, made from steel and rotating at 4000 rev/min will store 8.5 times the energy required to machine one RPM. The cutter rotating at 4000 rev/min would lose 240 rev/min after cutting one RPM. Further development of the experimental cutter was also recommended, including testing a 1-inch-wide, 12-inch-diameter version of the cutter at 5 mph. The cutter used in the IRAD is pictured in Figure 1 with the expanded cutter for the prototype test depicted in Figure 2.

The summary of the test results from the GTRI-funded IRAD project are as follows:

- the adhesive force holding the RPM to the roadway varies from virtually zero to 1200 lbf,
- the specific cutting energy for the plastics used to manufacture RPMs is around 0.04 hp-min/in3,
- the experimental saw has the potential to effectively and safely machine a RPM from the roadway whether it is bonded to the roadway or not,
- extrapolating the cutting force data to much higher feed rates also means much higher surface speeds which could melt the RPM plastic.

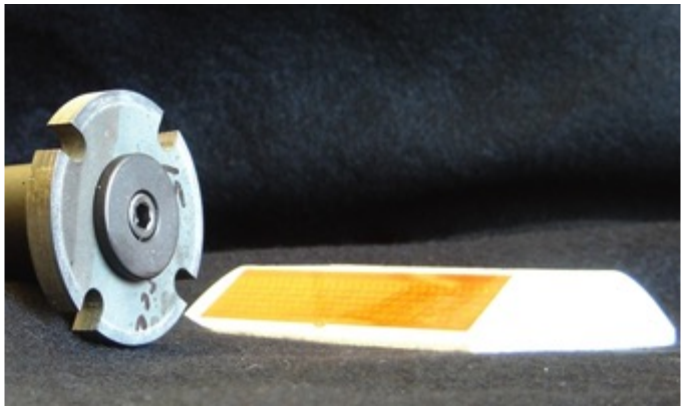


Figure 1: 2-inch-diameter, 0.220-inch-wide Proprietary Cutter Alongside RPM

Figure 2: Illustration of 12-inch-diameter, 5-inch-wide, Full-Scale Cutter

Conceptual Design of Prototype Cutter

The conceptual design for the experimental system was based on the results and recommendations from the GTRI-funded IRAD project. The conceptual design for the cutter, as described in the IRAD project final report, was for a stack of six, 1-inch-wide, 12-inch-diameter cutters, made from steel with carbide teeth inserts, rotating at 4000. The tooth geometry would be based on the geometry of the 2-inch-diameter cutter tested during the IRAD project. The marker would be mounted on a carriage that would accelerate it to 5 mph as it moved toward the rotating cutter. The marker would be supported by a load cell to measure the cutting forces. A high-speed video system would be used to capture images of the cut along with force data and any other velocity or position data that was to be collected. Brushless DC servo motors, motor controllers, and mechanical transmissions would be used to drive the cutter and the marker carriage. A physical containment would be used to protect the surrounding people and facilities from the hazards posed by the energy stored in the rotating cutter, the moving cutter teeth, and debris and projectiles generated during the cutting operation.

During the conceptual design phase, many of the major design decisions were made. Most of the components were modeled in a 3-D, solid-modeling package. An assembly model was completed that established a feasible geometry for the system. At this point, little analysis had been done; many design details were not complete on the components to be fabricated; many components to be purchased had yet to be selected.

Figure 3 and Figure 4 show assembly models from early in the conceptual design phase. Figure 5 shows the assembly model near the beginning of the detailed design phase with major components labeled.

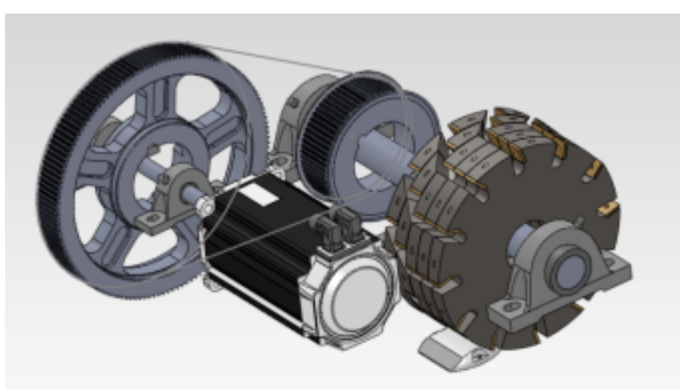


Figure 3: Early Conceptual Design with Cutter with Carbide Inserts

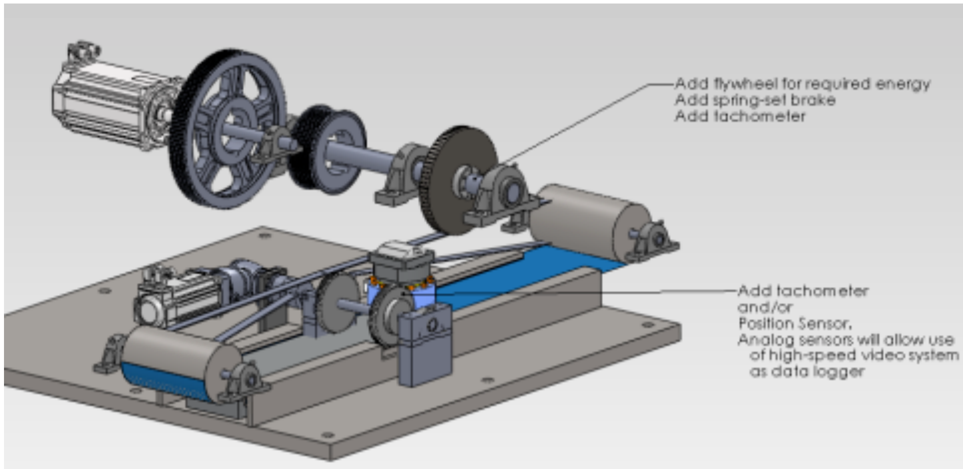


Figure 4: Early Conceptual Design with Moving Way Covers

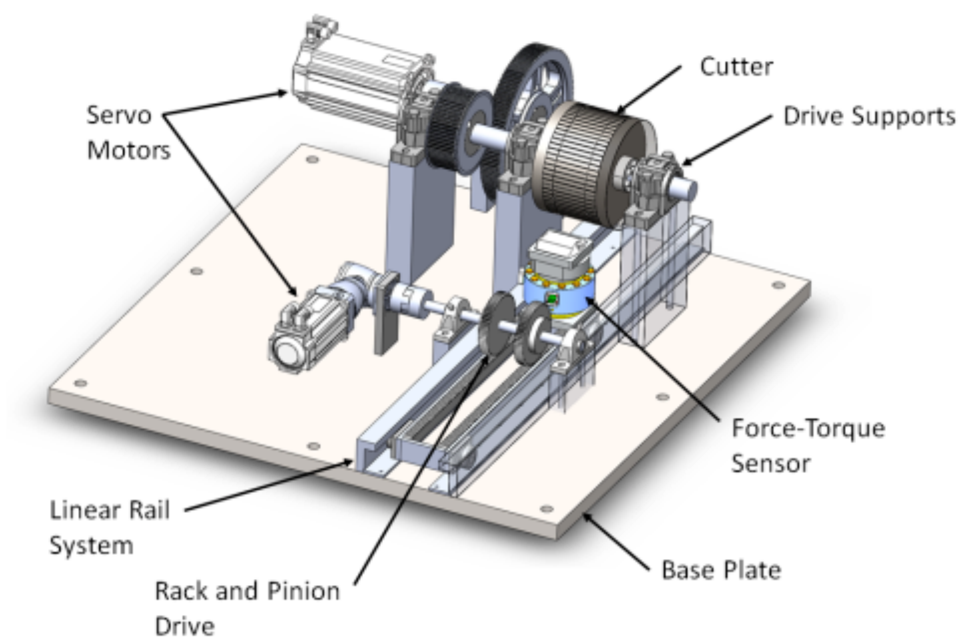


Figure 5: Late Conceptual Design with Major Components Labeled

Detailed Design, Analysis, and Review

A detailed design, based on the conceptual design shown in Figure 5, was generated. Design details were finalized on each fabricated components and dimensioned shop drawings were generated. Issues were resolved as they were identified. The remaining purchased components were selected, modeled, and added to the assembly model. Analyses were performed on several major components and subsystems.

Once these analyses were completed, an internal design review was scheduled to evaluate the system design and the analysis results. Design review participants included not only the project team, but also several other engineers from GTRI/ATAS, both senior and junior engineers. A total of seven engineers participated in the two design review sessions. A design review handout was generated and distributed to the participants several days before the initial meeting on July 01, 2011 between the hours of 9:00 AM and 11:30 PM. The design review handout is included as Appendix A. The participants reviewed the handout prior to the initial meeting. A major portion of the meeting was spent addressing the participant's questions that resulted from their review of the handout. Then, the handout was reviewed by section in document order. Only a portion of the handout was covered during the first meeting. Prior to the second meeting, attempts were made to resolve the issues identified during the first meeting. The handout was revised to include the issues raised during the first design review meeting, to track the resolution of the issues, and to document any additional analyses that were performed. Participants were provided with the pages necessary to update their copy of the design review handout; pages that had been covered during the first meeting were not exchanged. The appendix includes the latest revision of the design review handout. A second design review meeting was held on July 26, 2011 between the hours of 3:00 PM and 5:00 PM. There, the efforts made to resolve the issues raised in the first meeting were reviewed. The remaining sections of the design review handout were reviewed.

A major portion of the design and analysis effort was expended on the cutter. Ten pages of the handout are devoted to the cutter design and analysis. The handout also covers the following items:

- The selection and analysis of the shaft, bearings, and keyless bushing that support the cutter,
- The selection and analysis of the motor, motor controller, and transmission for both the cutter drive and the marker carriage drive, and
- The selection of the load cell for measuring cutting force and the track rollers that support the marker carriage.

The following subsections provide summaries of some of the design and analysis results.

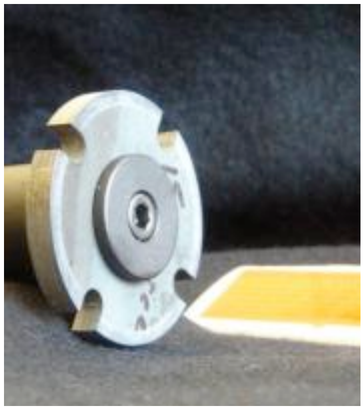
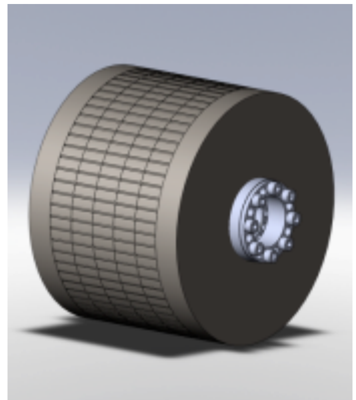
Full-Scale Cutter Design

The conceptual design for the cutter, as described in the IRAD project final report, is for a stack of six, 1-inch-wide, 12-inch-diameter cutters, made from steel with carbide teeth inserts, rotating at 4000 rev/min. Such a cutter would store 8.5 times the energy required to machine one RPM and would lose 240 rev/min after cutting one RPM.

Initial design effort was based on these values and the use of carbide inserts as well as an attempt to arrange the teeth in a helical pattern such that the chips could be cleared axially. The use of replaceable carbide inserts added significant complexity to the design and limited the number of cutter teeth. The 12-inch-diameter dimension would necessitate fabrication by an outside vendor. The cutter design presented during the July 2011 design review was based on the following design decisions:

- Use the same feed per tooth (i.e. chip size) as that of the experimental cutter, 0.02 inch per tooth. 66 teeth are required for 5280 inch/min at 4000 rev/min.
- Do not use carbide tooth inserts. Fabricate the cutter disks from hardened or case hardened tool steel. Machine the cutting teeth into the cutter disk.
- Reduce cutter diameter to 9.6 inches such that cutter disks can be fabricated on the GTRI Machine Services wire EDM machine.
- Clear the chips radially rather than axially.
- Use a stack of six 1-inch-wide cutter disks with two 1-inch-wide end disks for additional flywheel inertia.

Table 1: Cutter Parameter Comparison

	2006 Experimental Cutter	Full-Scale Cutter Design	
			
Diameter	2	9.6	inch
Number of teeth	4	66	
Radial rake angle	15°, 30°, 45°	15°	
Radial clearance	0 at feed of 0.02 inch/tooth	0 at feed of 0.02 inch/tooth	
Axial rake angle	0°	0°	
Axial depth of cut	0.22	4.54	inch
Radial depth of cut	0.125	0.8	inch
Feed per tooth	0.02	0.02	inch/tooth
Spindle speed	417-884 rev/min	4000	rev/min
Feed rate	50	5280	inch/min

A comparison of the cutter parameters of the 2006 experimental cutter and the full-scale development cutter are given in Table 1. The parameters for the full-scale development cutter are the result of the design decisions listed above. These parameters and design decisions dictate most of the geometry of the cutter. The remaining geometry is determined by design decisions about the flute width and depth, the chip breaker style and size, and the selection of the cutter shaft and keyless bushing.

A review was done of the extrapolation of cutting forces from those measured on the 2006 experimental cutter to those estimated for the full-scale development cutter. Based on the experimental results from the IRAD project, a decision was made to assume that cutting force is proportional to axial depth of cut; radial depth of cut; and feed per tooth. This calculation was previously executed and presented in the IRAD project final report.

Table 2: Cutter Force Extrapolation

	2006 Experimental Cutter	Full-Scale Cutter Design	
axial depth of cut multiplier		20.64	
radial depth of cut multiplier		6.4	
feed per tooth multiplier		1	
combined multiplier		132.1	
x-direction peak force measurement	22	2906	lbf
y-direction peak force measurement	79	10434	lbf

Note: Many of the analyses and component selections were based on the extrapolated cutting forces. The forces have been extrapolated by a factor of 132 beyond the experimental conditions. Initial testing using a single, 1-inch-thick cutter disk will be required to assess the validity of this large extrapolation.

A review was done of the estimated cutting energy required. This calculation was previously executed and presented in the IRAD project final report. The specific cutting energy would be on the order of 0.05 HP min/in^3 at an undeformed chip thickness of 0.02 inch. For the marker manufactured by 3M™, the energy required to mill the marker would be 12780 J.

Given the cutter geometry and the operational rotational speed, the flywheel energy storage of the cutter could be calculated. The mass moment of inertia of the cutter assembly was estimated to be 0.4756 kg m^2. This cutter assembly would store 60083 J at 4800 rev/min. The required cutting energy would be 21.3% of the total kinetic energy. The cutter speed would be reduced to 4259 rev/min after extracting 12780 J from the cutter kinetic energy.

The mass moment of inertia of all of the rotating mass in the development system was estimated to be 0.2444 kg m^2. This included the 1-inch-wide cutter, the auxiliary flywheel, the shaft and hubs, the brake rotor, and the drive pulleys and motor rotor.

Given the cutter geometry, the operational rotational speed, and the estimated cutting forces, stress analyses could be performed. The cutter disks will be subjected to loads from several sources. The cutter disks will spin at the design rotational speed currently set at 4800 rpm. The cutter will be connected to the main shaft by one or more Fenner B-LOC™ bushings that use wedges that result in an internal pressure load on the cutter disk bore. The cutting operation will subject the cutter teeth and rim to large tangential and radial loads. Analyses were completed for both the centrifugal load alone and for the combined centrifugal and pressure loads. For these cases, closed-form stress analysis results are compared to results from SolidWorks Simulation™. Only a finite element analysis was done for the combined centrifugal, pressure, and cutting loads.

The results of the analyses are presented graphically in Appendix A. The closed-form solutions are in good agreement with the finite element analysis from SolidWorks Simulation™. The hoop and radial stresses from the closed-form solution are the first and second principal stresses calculated by the finite element analysis. Both analyses show that the internal pressure loads generated by the Fenner B-LOC™ bushing are very significant as compared to the centrifugal loads caused by rotation. The close agreement between the closed-form solution and the finite element analysis results for the first two cases provide confidence that finite element analysis results for the combined centrifugal, pressure, and cutting loads will be representative of the actual cutter under load.

A simplified model of the cutter was used for the finite element analysis. A constant diameter was used for the rim of the cutter. The flute and chip breaker geometry were replaced with simple notches with semi-circular bottoms. The loads applied in the finite element analysis were:

- Centrifugal Load: -419 radians/sec (4000 rev/min) and 0.0006860 radians/sec^2 about axis of rotation,
- Pressure load: 150 N/mm^2 on bore, Inertia-relief constraint,
- Normal force: 814 N on each of 7 teeth,
- Normal force: total of 8977 N applied to a segment of the rim, and
- Normal force: total of 8977 N applied to a segment of the bore.

Several graphical plots of the results of this analysis are presented in Appendix A including the principal stress and displacement plots. The von Mises stress plot is shown in Figure 6. The von Mises stress is commonly used to predict yielding of materials under any loading condition based on the yield stress measured by simple uniaxial tensile test.

The extent to which the detailed design was completed accounted for a successful prototype development as no issues other than blade integrity were discovered in operation.

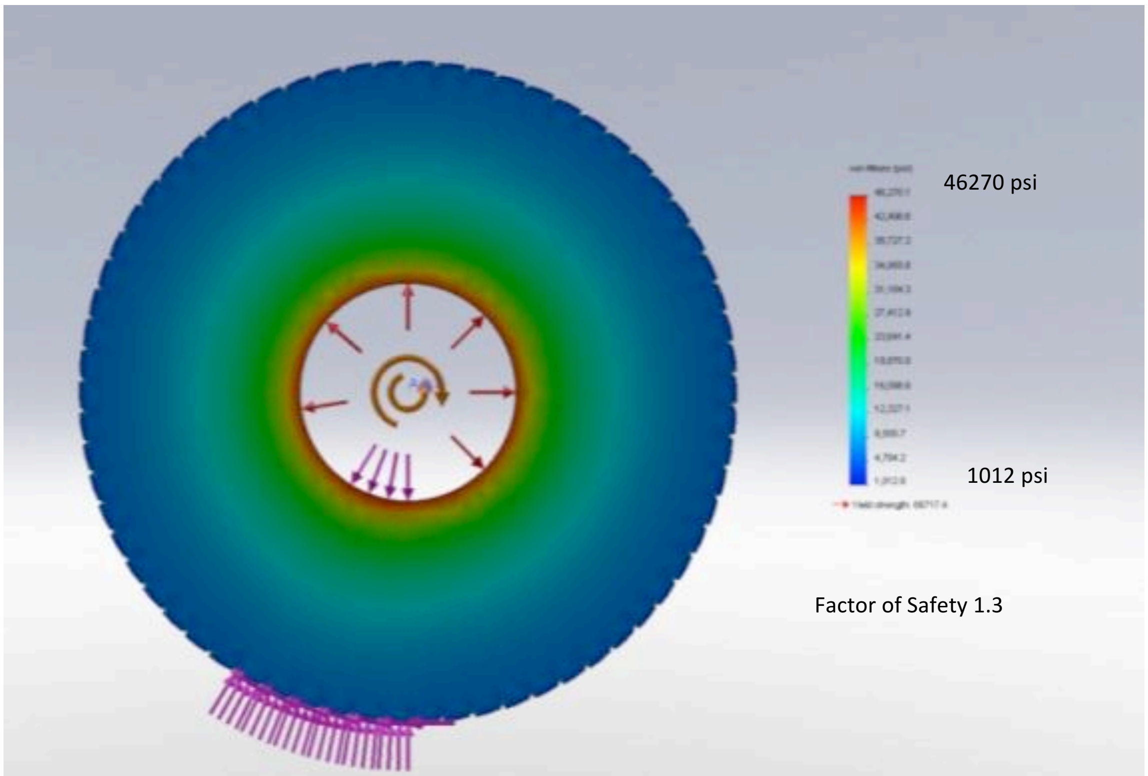


Figure 6: Von Mises Stress Plot for Combined Centrifugal, Pressure, and Cut Loads

Fabrication and Assembly

Based on the design that was generated in the conceptual design phase for the prototype cutter (See Figure 5), the detailed design was generated focusing primarily on parameters with little or no changes to the previously described conceptual design. There were a number of custom parts that were required for fabrication as well as several purchased items that were needed for the assembly. The following figures contain a number of those individual parts.

Figure 7: Machined Components for Prototype Cutter

Figure 8: Drive Components for Prototype Cutter

Figure 9: Purchased Parts for Roller Elements and Force Measurements

Beyond the basic components, a considerable effort was required to assemble and program all of the drive hardware. The following image was taken inside the electronics cabinet. This cabinet contains the controls for the motors used to drive the cutter shaft and the marker carriage. This cabinet was reused from existing work done at GTRI, but several modifications were required to tailor the equipment for the prototype cutting system.

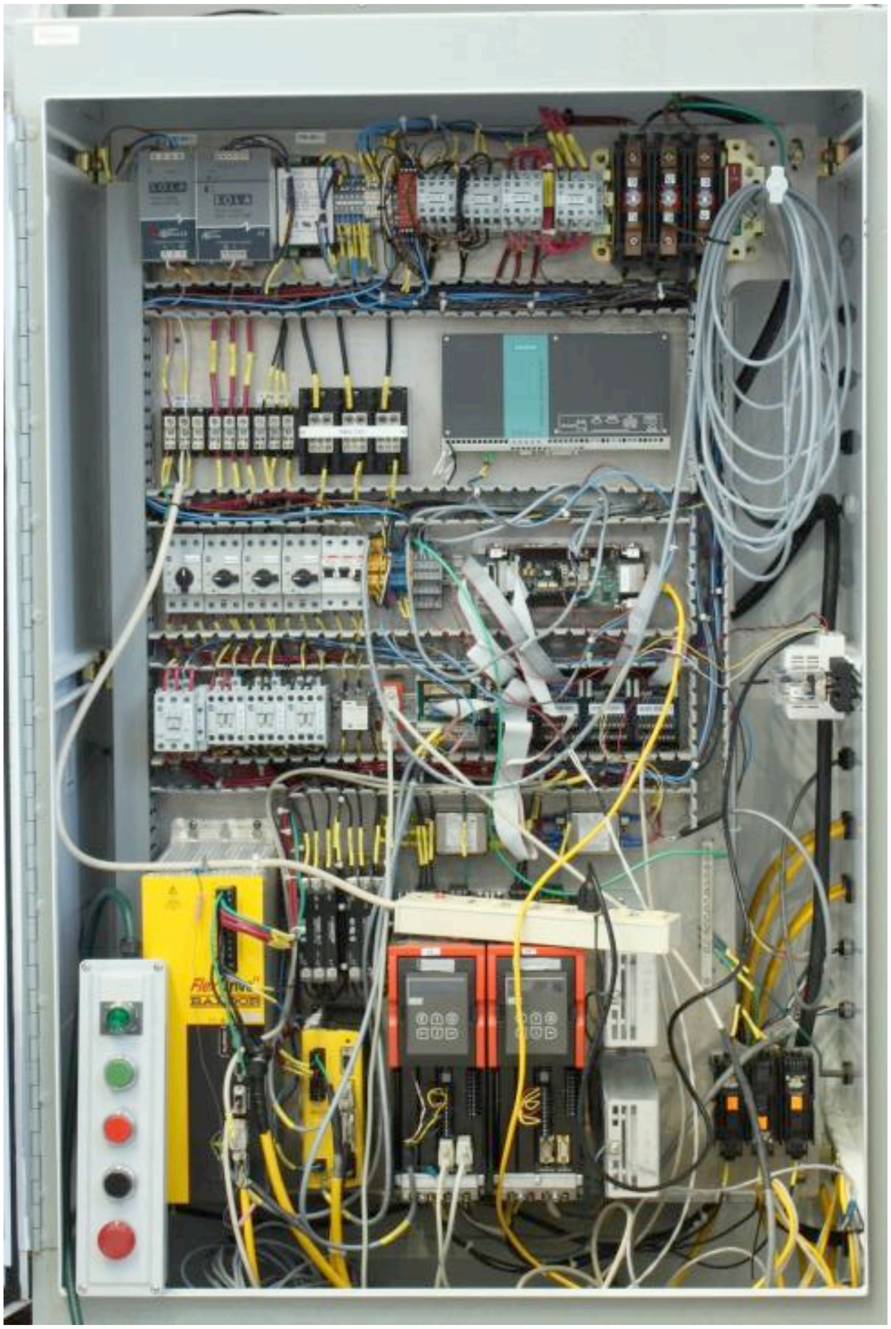


Figure 10: Electronic Hardware for Prototype Cutter System

With the major components in place, the full assembly of the system was able to be completed. The full assembly was registered to a large steel plate that was attached to the floor of the laboratory with concrete anchors (See Figure 11). Numerous photographs are included to capture the complex assembly of the motors, drive systems, brake, and cutter.

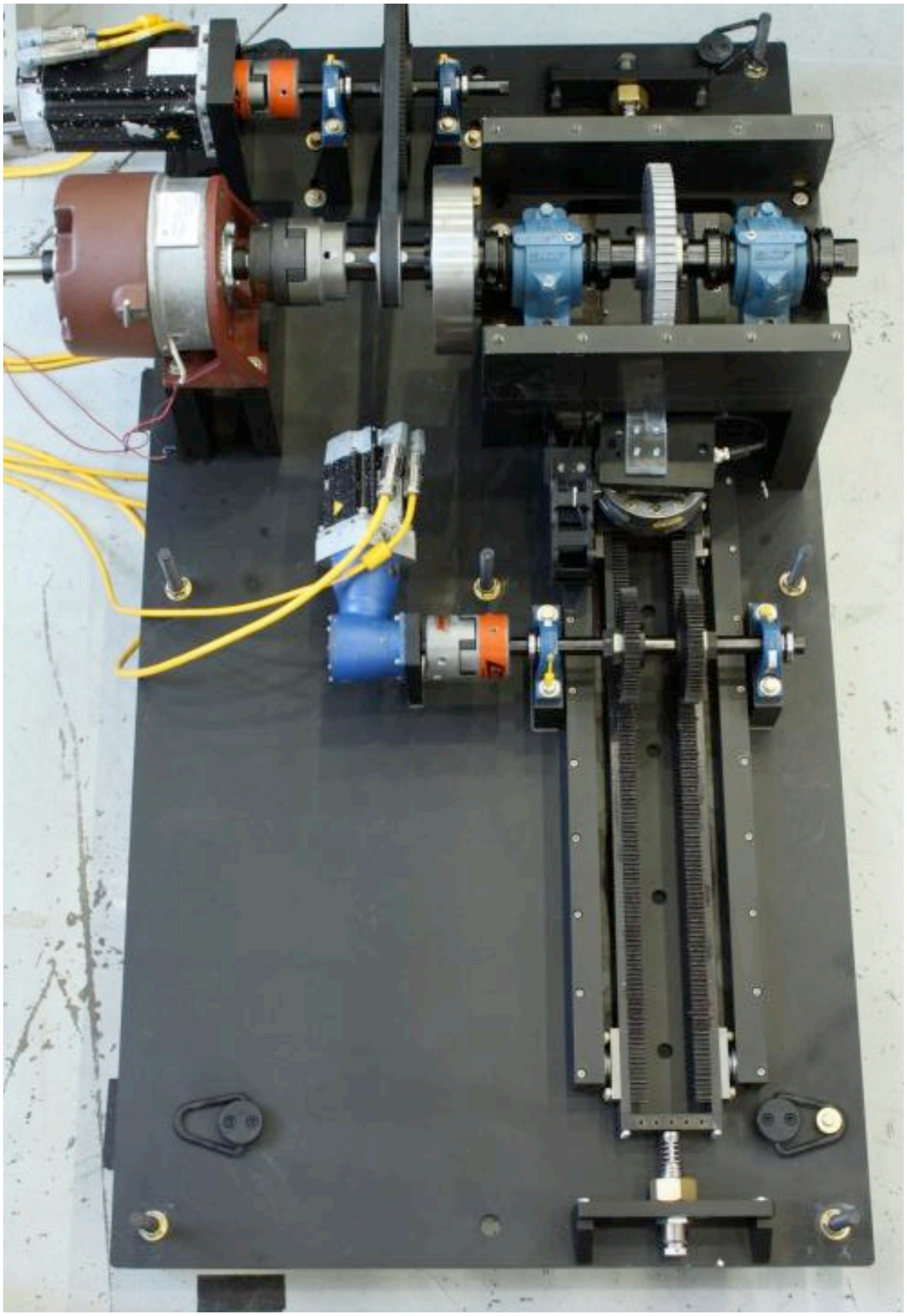


Figure 11: Top View of Cutting Prototype

Figure 12: Linear Drive Including Shock Absorbers

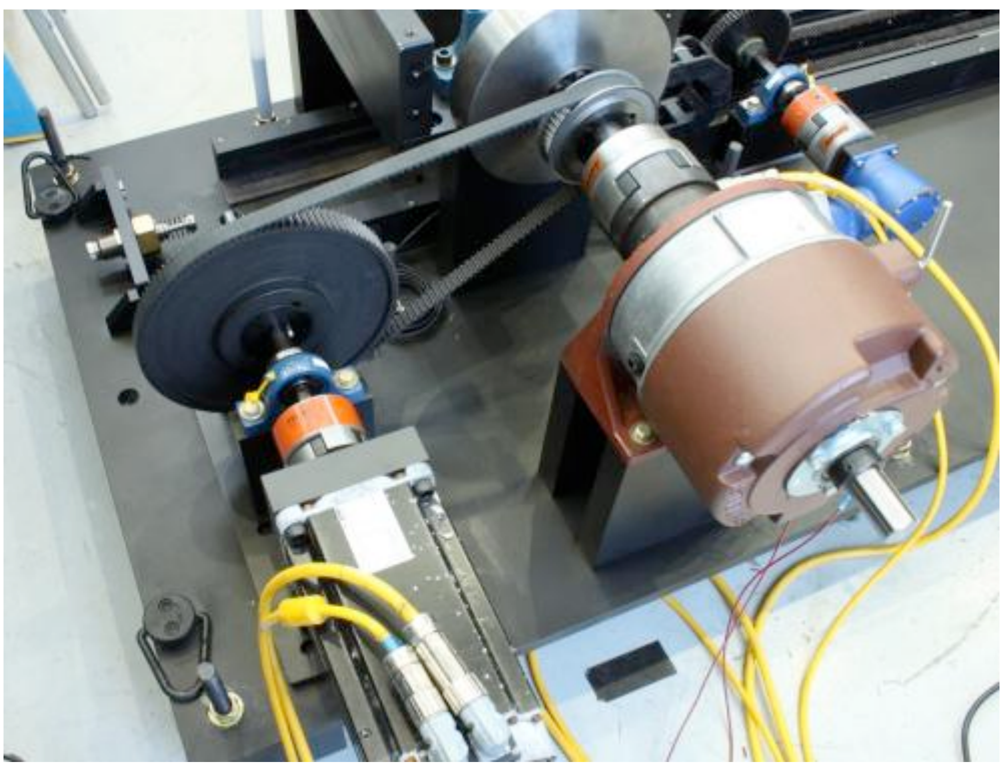


Figure 13: Rotary Drive System with Brake

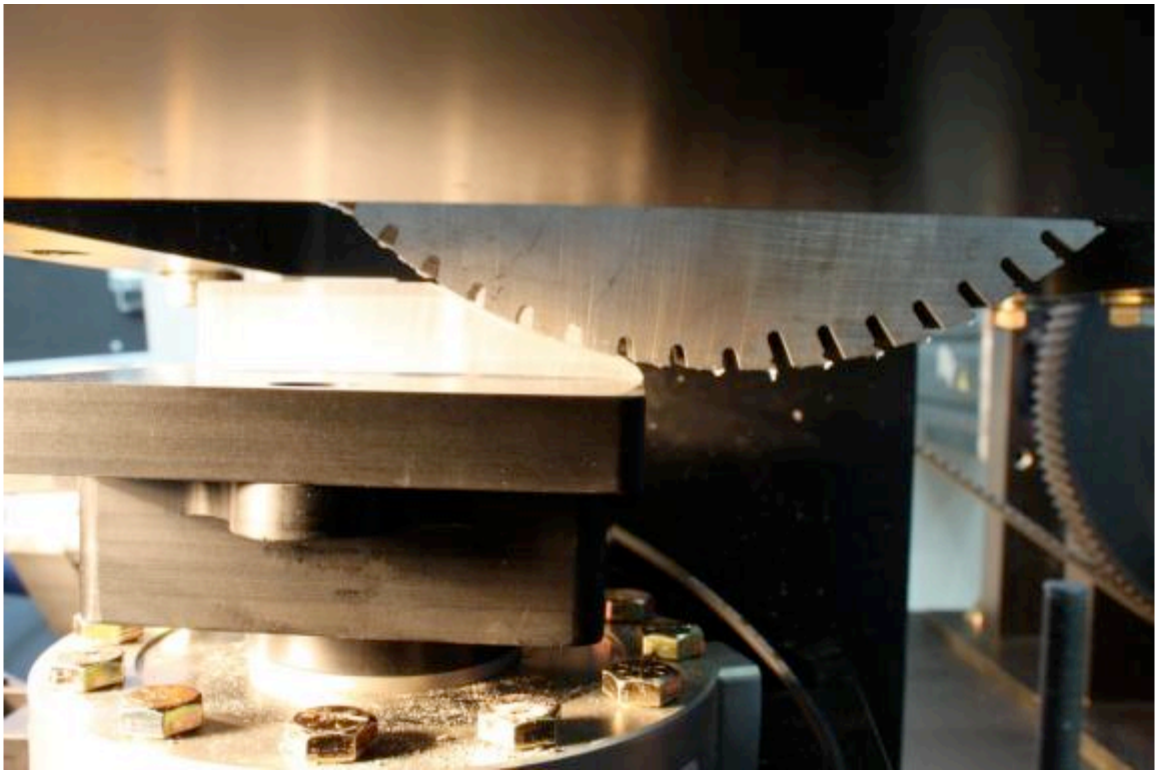


Figure 14: Side View of Cutter in Contact with Test RPM

Figure 15: Side View of Cutter Showing Multiple Tooth Types

Testing and Results

Initial testing exercises with the prototype cutter yielded successful results. A number of high-speed videos were captured to evaluate the performance of the tooth design shown in Figure 15. The smaller tooth, with a chip breaker, appeared to be efficient for cutting. There was no significant buildup of plastic on the cutter after several cutting attempts. Future blade types could be made simply with only the one type of tooth.

Cutting operation was tested at varying speeds and feed rates using a solid piece of polycarbonate as well as Avery Dennison™ RPMs. At lower speeds, the blade only partially cuts the RPM due to the lower cutting energy stored in the blade. However, once full speed of the cutter was assessed, the marker was completely milled away in a fraction of a second as initially planned. Figure 16 shows one of the markers from the initial cutting tests. This particular marker was subjected to two separate cuts. The initial cut was made with the cutter set to a lower speed (60% rotation speed) and only partially cut the marker while the second cut with the cutter set to full speed fully cut the marker. Figure 17 includes labels that clarify the different cut types. One thing of importance is the speed at which the cut was made. The marker needs to be removed in approximately 100 milliseconds to achieve a cut rate of approximately 5 mph, whereas the cut depicted in Figure 17 was determined to be approximately 200 milliseconds based on a review of the high-speed video of the cut demonstrating a feed rate of approximately 3 mph.

Figure 16: Milled Marker Shown with Resulting Chips

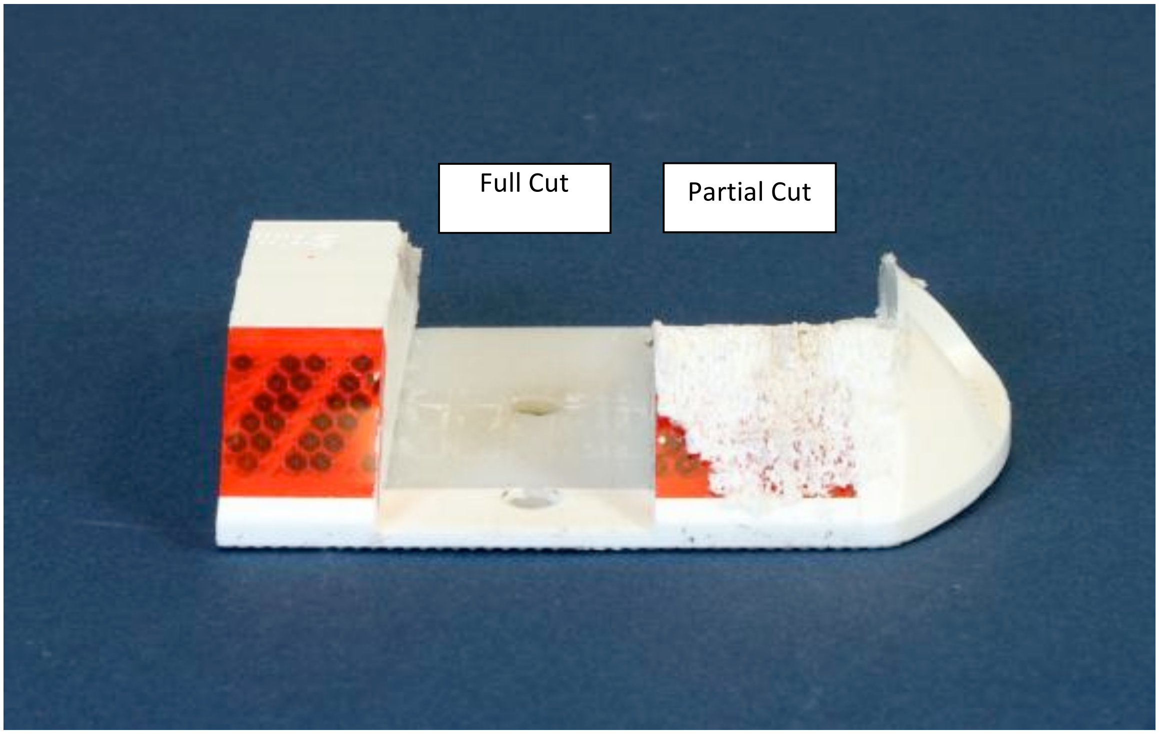


Figure 17: Cut Description on Test RPM

On April 25, 2012, a single Avery-Dennison (Ennis) C80 marker was cut using a nominal cutter speed of 4870 rpm. High-speed video was captured at 4000 fps. The motor controller for the cutter drive recorded commanded and actual motor speed at a rate of once every 0.071 s. No force data was recorded. Before and after weights of the marker were not measured. The resulting cut is pictured in Figure 16 and Figure 17. One point to make is that during the cuts on April 25, the loadcell had not been integrated in the system yet meaning no force data was collected.

The high-speed video was used to estimate the marker velocity during cutting; it could not be used to estimate the time required for the cut. The average marker velocity was estimated to be 23.1 in/s or 1.3 mph. The travel distance required for the cut was estimated to be 4.637 inches based on the geometry of the cutter and the marker. The cutting time was estimated to be 201 milliseconds based on the average marker velocity and the travel distance.

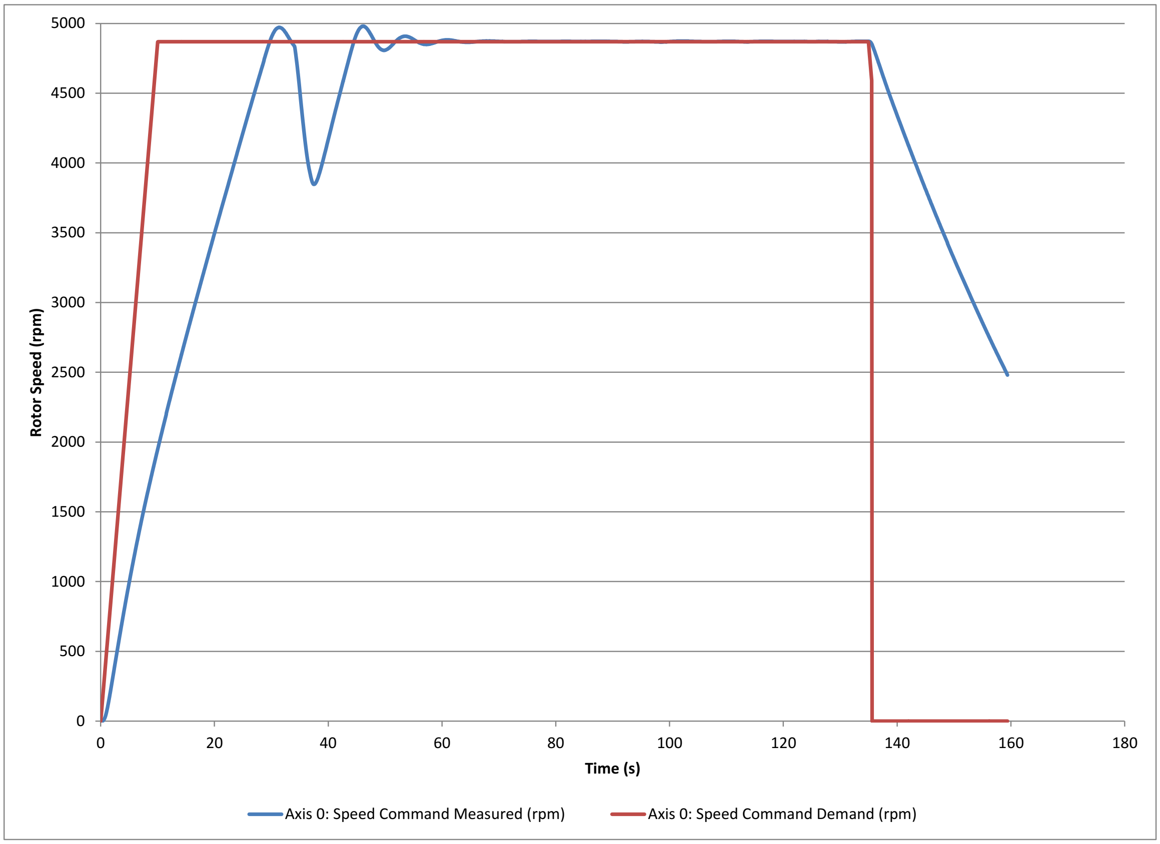


Figure 18: Graph of Cutter Speed

It was expected that the cutting energy could be estimated from the drop in cutter speed caused by the cut along with the rotational inertia of the rotating masses. A knee or corner can be seen in the cutter speed plot at around 34 s. However, the cutter speed falls for over 3 seconds while the cutting time was estimated to be only 0.2 s. One possible explanation is that the speed reduction due to cutting was confounded with the motor controller behavior. Calculating the kinetic energy change over only periods of 201 milliseconds gives cutting energy estimates of between 660 and 980 J depending on the selected start time of the cut.

In order to estimate the amount of mass removed during this cut, a groove that duplicated the cut made with the marker removal machine on April 25 was cut in a whole marker using a milling machine. The before and after weight for that marker are 87.897 g and 68.227 g respectively. The specific cutting energy is typically calculated on a volumetric basis rather than a mass basis. The C80 marker is assembled from several molded plastic pieces; it contains a number of voids. The density of Lexan™ polycarbonate was used to estimate the volume of material removed based on the mass of material removed. The specific cutting energy can be calculated based on the cutting energy and the volume of material removed. Calculating the specific cutting energy based on the kinetic energy change over only periods of 201 milliseconds gives specific cutting energy estimates of between 0.015 and 0.02 HP min/in^3 depending on the selected start time of the cut. The system design was based on a specific cutting energy of 0.049 HP min/in^3 which was measured for polycarbonate during the GTRI-funded IRAD project.

Table 3: Change in Kinetic Energy Over 0.201 Second versus Cut Start Time

Start Time (mins)	Initial Cutter Speed (rpm)	Final Cutter Speed (rpm)	Cutting Energy (J)	Specific Cutting Energy (J/mm^3)	Specific Cutting Energy (HP min/in^3)
34009	4771.6	4737.7	432.0	0.0264	0.0097
34080	4768.1	4716.2	659.9	0.0403	0.0147
34151	4754.9	4694.0	771.2	0.0470	0.0172
34222	4734.1	4671.0	795.2	0.0485	0.0178
34293	4712.4	4647.6	812.8	0.0496	0.0182
34364	4690.1	4623.2	835.5	0.0510	0.0187
34435	4667.0	4597.6	861.3	0.0525	0.0192
34506	4643.6	4571.4	891.7	0.0544	0.0199
34577	4618.9	4544.5	914.2	0.0558	0.0204
34648	4593.2	4516.8	932.1	0.0569	0.0208
34719	4566.8	4488.5	950.1	0.0580	0.0212
34790	4539.8	4459.6	967.8	0.0590	0.0216
34861	4512.0	4430.4	978.6	0.0597	0.0219
34932	4483.6	4401.1	982.7	0.0600	0.0220
35003	4454.6	4372.0	976.9	0.0596	0.0218

On April 25, a single cut was made on an Apex™ Model 921 marker. The resulting cut is pictured in Figure 19. It was later observed that the epoxy fill of these markers contains sand throughout. Earlier, it was believed that the sand was confined to a layer at the bottom surface of the marker. A polished section of this same marker is pictured in Figure 20. The epoxy fill has been lightly stained. The sand particles can be seen as whiter shapes within the stained epoxy. Sand particles can also be seen in the cut surface in Figure 19.

Figure 19: Apex Model 921 Cut

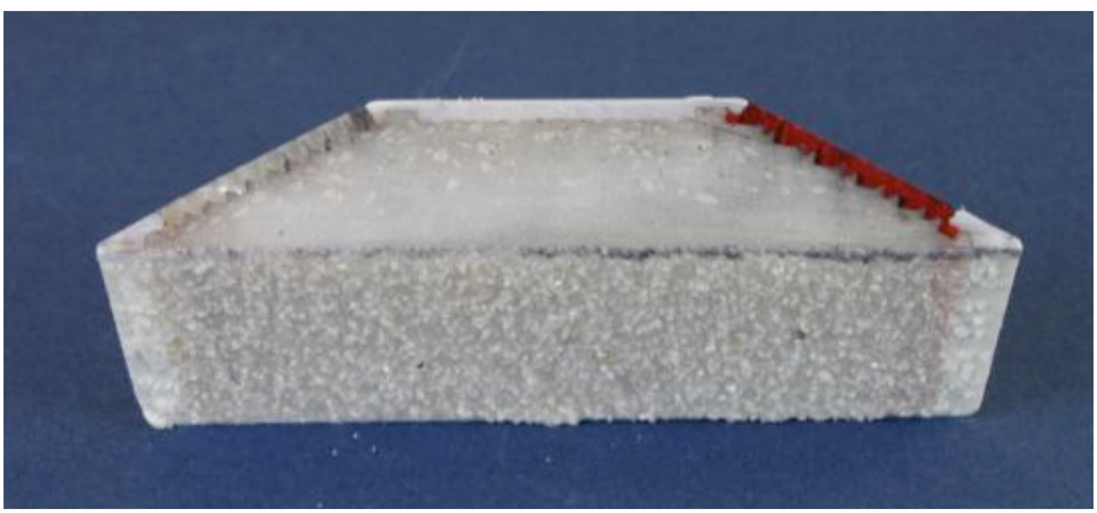


Figure 20: Apex Model 921 Cross Section

A polished section of an Apex Model 921 marker that was sawed into two pieces in 2006 is pictured in Figure 21. No sand is apparent in the epoxy fill of this marker.

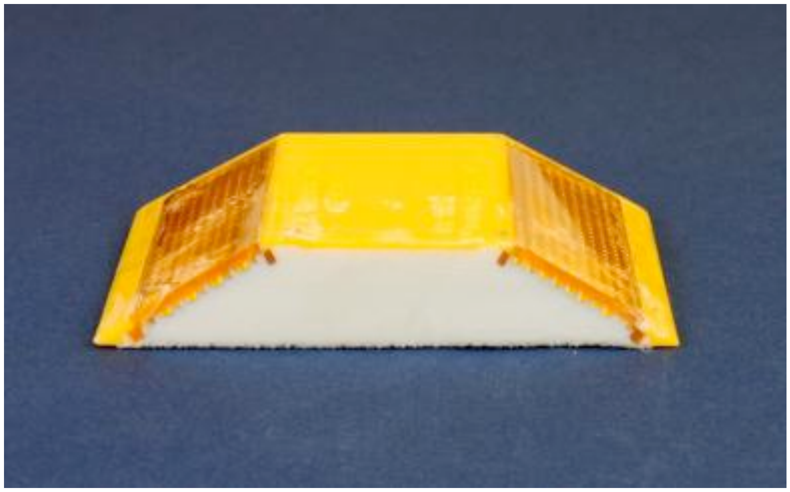


Figure 21: Apex Model 921 from 2006

On June 14, 2012, two additional Avery-Dennison (Ennis) C80 markers were cut using a nominal cutter speed of 4870 rpm. High-speed video was captured at 4000 fps. Force data was recorded by the high-speed video system twice per image frame. The motor controller for the cutter drive recorded commanded and actual motor speed at a rate of once every 0.071 s. Before and after weights of the marker were measured. The resulting cuts are pictured in Figure 22. The performace of the system was very different on these cuts than it was on the full-speed cut made on April 25. Some of the differences can be seen in a comparison of Figure 16 and Figure 22. The chips produced on April 25 are 1-inch-wide ribbons that are rolled and folded. Considerable melting is apparent on the markers and chips from the cut on June 14. Also, on June 14, considerable smoke and odor was generated during the cutting process. The marker velocity was considerably slower on June 14 than it was on April 25.

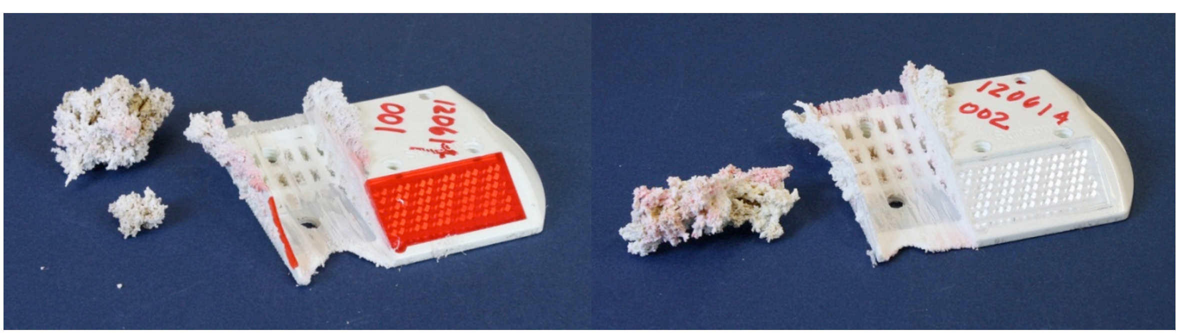


Figure 22: Markers Cut at Full Speed on June 14, 2012

A feature of the high-speed video software was used to automatically track the position of the marker in the image frame over time. The position was scaled in inches based on the dimensions of the marker. In Figure 23, the position of the marker in the direction of carriage travel is plotted verus time for the three cuts. The registration of position and time are arbitrary from cut to cut in this graph. The direction of travel and the marker velocity can be compared from cut to cut. For the two cuts made on June 14, the marker travels rapidly towards the cutter, then reverses direction upon impact with the cutter. After several direction reversals, the marker moves in an approximately constant velocity. For the cut made on April 25, the initial impact of the marker with the cutter could not be tracked in the available high-speed video. The marker velocity during the portion of the cut that can be tracked is very much higher than the velocity of the other two cuts. Using a line-fitting routine, the slope of the position data from April 25 can be estimated to be 27.0 in/s; that is the average marker velocity during the cut is 27.0 in/s. For comparison, the slope of the position data from the second cut on June 14 can be estimated to be 0.31 in/s.

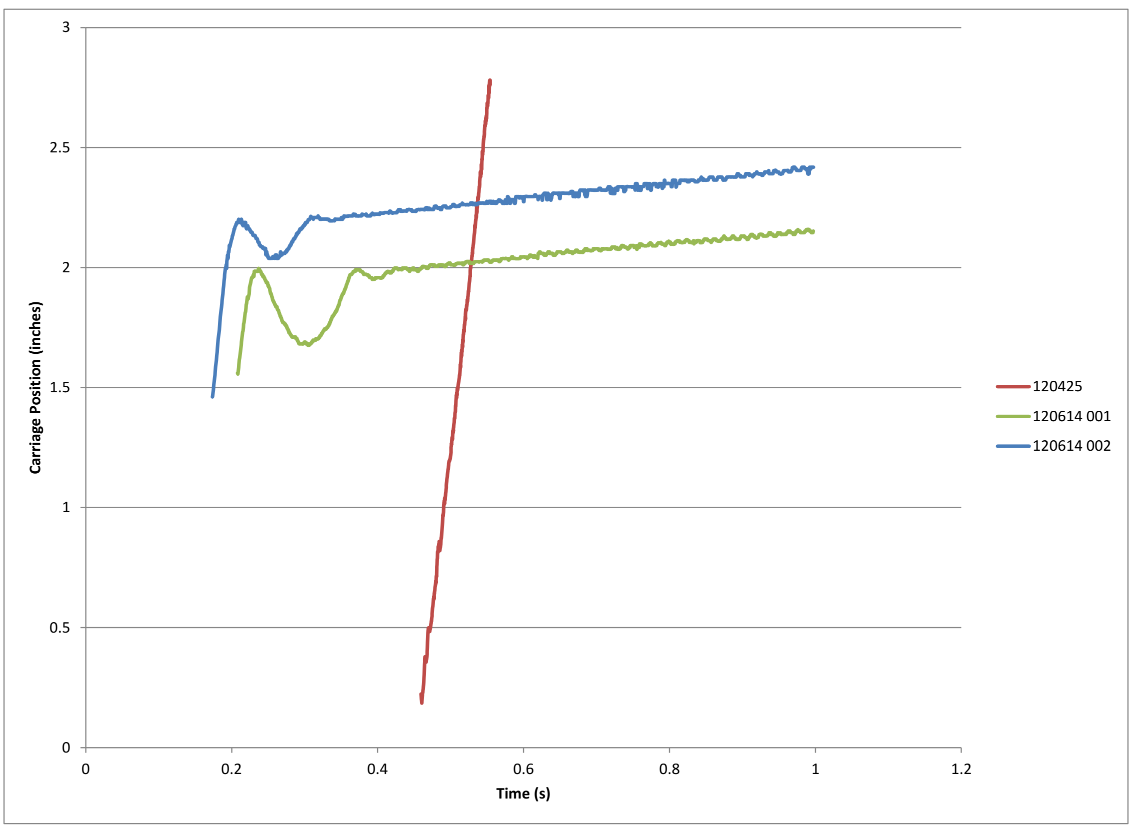


Figure 23: Comparison of Marker Motion

In summary, there was a significant change in the performance of the cutter when cutting the Avery Dennison™/Ennis™ markers between the cut made on April 25 and the cuts made on June 14. The observed changes included:

Table 4: General Comparison of Results Before and After Blade Dulling

	April 25, 2012	**June 14, 2012**
Cutting speed	> 25 in/s, 0.2 second cut	About 0.3 in/s, cut lasted several seconds
Chip appearance	Distinct, 1-inch-wide chips folded and rolled, limited melting	No distinct chips, considerable melting
Cut appearance	Very little melted material attached to marker, clean cut surface	Significant melted material attached to cut, smeared cut surface
Smoke and odor	Very little smoke and odor	Significant amount of smoke and odor

The current belief is that the one cut made on the Apex marker on April 25 caused significant damage to the cutting edges of the teeth on the cutter because of the sand in the epoxy.

The loadcell was used to measure cutting forces during the cuts made on June 14, 2012. The forces measured during the second cut on June 14 are plotted along with marker position in Figure 24. The force in the z direction is the vertical force; the sign is negative to indicate the force is down towards the carriage. The force in the x direction is the force of the cutter pulling the marker into the cut. The graph shows the rapid approach of the marker, the rise of both the x-direction and z-direction forces with the initial contact of the marker and cutter, the fall of both forces as the marker moves away from the cutter, and later, a second smaller peak in both forces as the marker once again contacts the cutter.

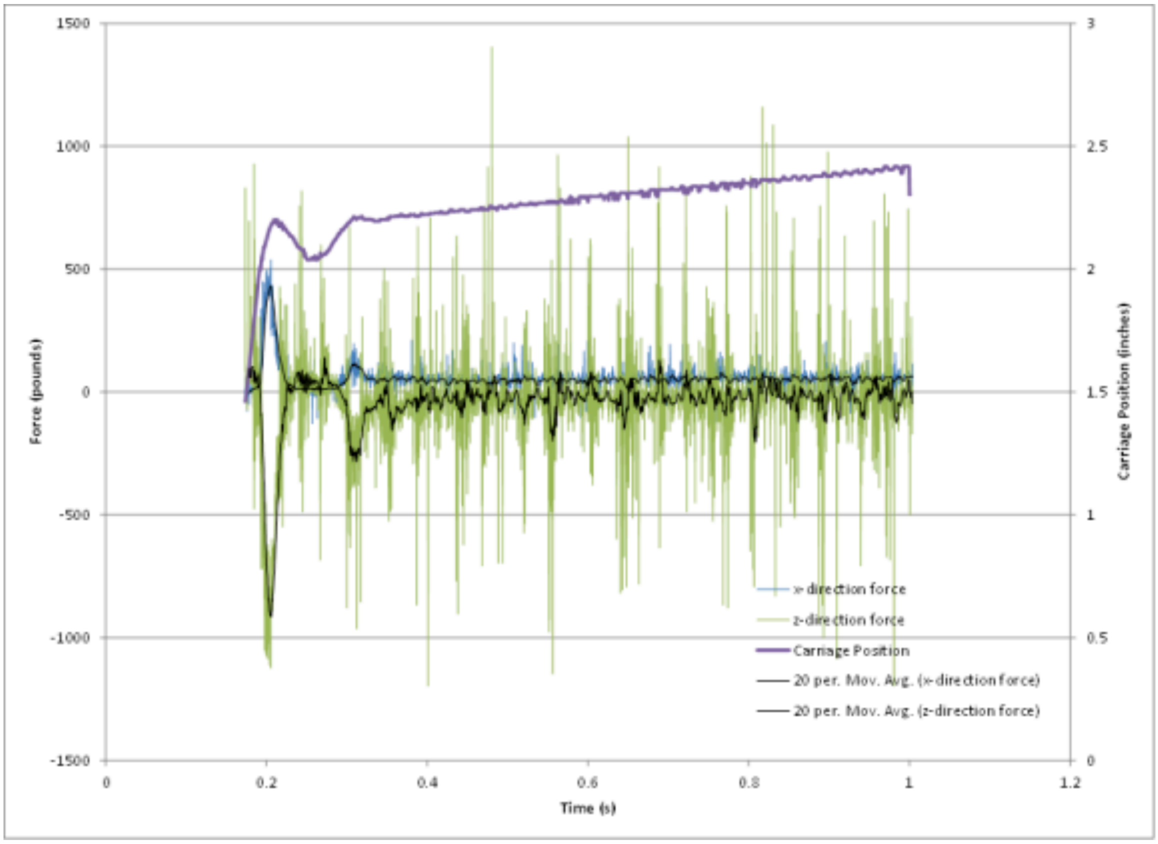


Figure 24: Force and Position from Second Cut on June 14

Conceptual Design of Full-Scale System

With the full-scale cutter/flywheel tested successfully, a number of issues related to incorporating the cutter into an RPM removal system still remain. For example:

- How will the position of the cutter, relative to the pavement, be set to maximize the RPM removal while minimizing the frequency of pavement strikes?
- Our current estimate of the vertical reaction force of the cutter is 10,000 pounds. How will this force be reacted through the spindle to the vehicle frame?
- Will the cutters be lifted above the pavement between RPMs? If so, how?
- Will the operator align the cutters with the RPMs or will an automatic alignment system be used?
- How will the center-to-center distance between the two cutters be adjusted to accommodate the various centerline stripe patterns?
- How will the cutter be protected from debris such as rocks and steel fasteners on the roadway? Will a combination of brushes and nozzles be sufficient? Will sensing used to trigger lifting of the cutter to avoid the debris be necessary?

A conceptual design was completed to address some of these questions with some questions requiring additional investigation. For instance, the ability to cut markers side by side may not be reasonable due to the variation evident in the crown of the road. Otherwise, the conceptual design includes the general design details that would exist for such a machine. Because the platform is yet to be determined the general concept was created to be placed upon either a large truck or possibly a trailer.

Figures 25 – 27 contain detailed images of the concept developed under this contract. The primary feature that is different from the laboratory prototype generated is the inclusion of a hydraulic cylinder that controls the height of the system during motion. The cutter and shroud (identified in Figure 25) become enclosed with the front portion of the shroud having the potential to absorb pavement strikes in case there are unexpected bumps in the road. Because the cutter size would be larger for the full scale system, there is no need for an additional inertial load as required in the prototype system.

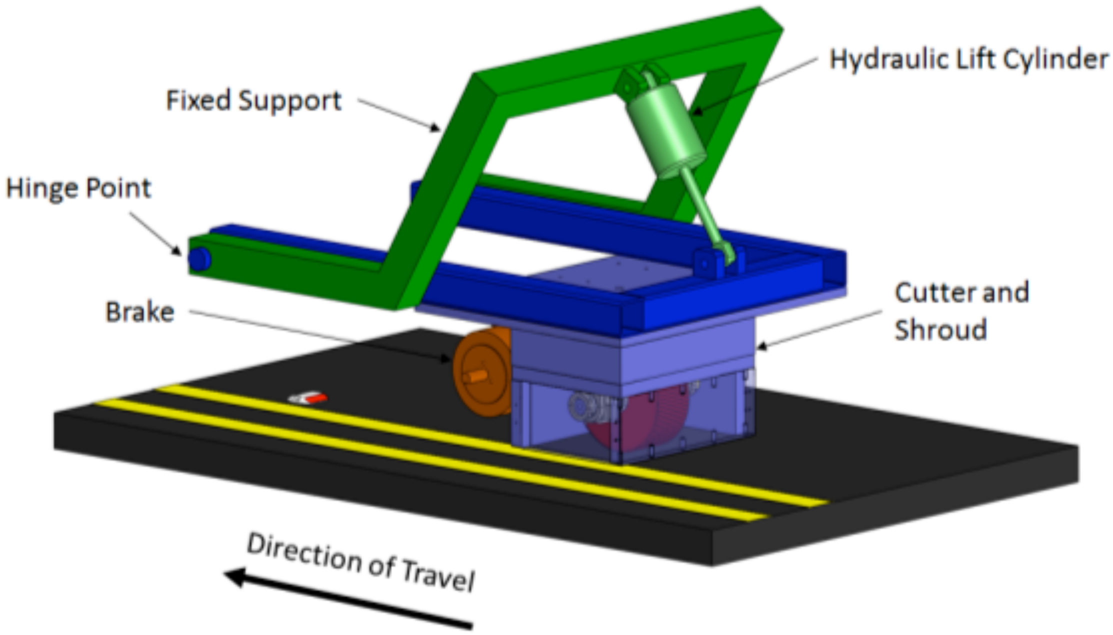


Figure 25: Rear View of Mobile Cutting System

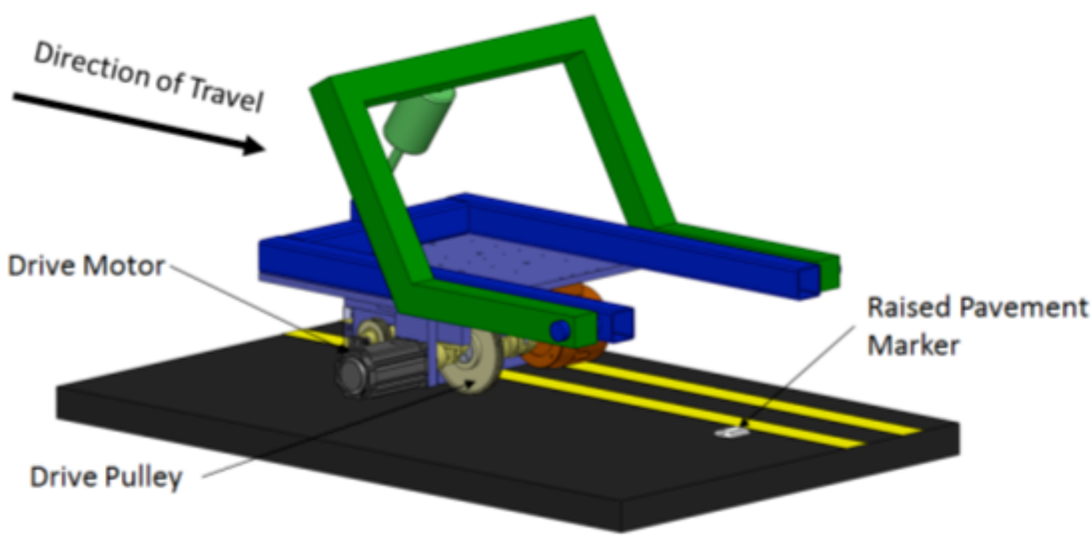


Figure 26: Front View of Mobile Cutting System

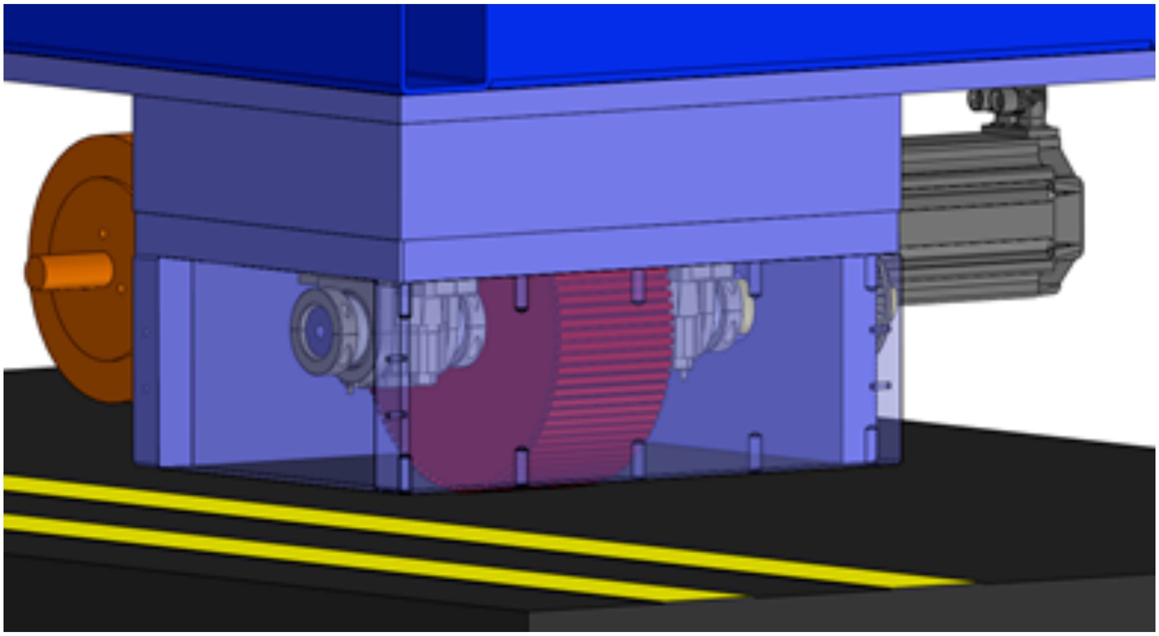


Figure 27: Detailed View of Cutter Showing Bearing Support

The estimated cost of the machine would be driven by the cost of the cutter, the mechanical structure to support the cutter cell, and the drive components such as the high powered rotary motor and hydraulic system. The cost of these components alone would be approximately $20-25k. Additional support equipment such as a hydraulic pumping station, safety circuit, or electric generator would be required to complete the system. In total, the cost of the system would be expected to be approximately $40-60k including the labor to install the system on a truck or trailer in limited quantities. Additional costs related to consumable items such as carbide inserts or new blades may be on the order of $10k annually.

Summary

The RPM removal research did highlight a flaw in the initial approach taken with the cutting prototype, which was the existence of sand captured in a composite matrix in particular markers. The existence of sand led to dulling of the cutting edge nullifying the ability to use basic tool steel as a cutting tool for this application. Although this issue became a major limitation to completing data capture as initially planned, the general concept is still valid. Additional research would be required to build and test a new blade design leveraging different materials with the special consideration of making the cutting edge harder than sand. This can be achieved by using carbide materials such as titanium carbide, which is a common material used for machining. Although only a single cut was successfully performed with the system prior to dulling the blade on a sand filled marker, the general findings support the notion that machining RPMs from the roadway is a feasible approach to road maintenance.

References

1. Schey, John A., Introduction to manufacturing processes, New York: McGraw-Hill, c1977, pp. 230-240.

Appendix A
Design Review Handout Document

Appendix A – Detailed Design Review Document

Design Review Objective: Identify any remaining issues and action items that should be resolved prior to release of drawings and purchase orders for fabrication and construction of the full-scale cutter and supporting components.

Contents

Revision Notes

Rev	Description and Comments	Date
–	Initial Release	6/30/2011
A	*Update handout with pages i, ii, 2, 15, and 19 and higher.* Specify loads on stress plots on page 15. Add section on load cell selection. Add section on salvaged motor for main drive. Add section on carriage. Update Action Item List. Add resolutions for Action Items 14, 15, 4, 22, and 25.	7/21/11

Project Background

There are about 3 million raised-pavement markers (RPMs) in service in the State of Georgia. Georgia DOT would like to replace RPMs on an 18-month cycle. Currently, RPM removal is a manually intensive process. To remove RPMs, workers use either a reciprocating chisel or an eradicator/scarifier. With a reciprocating chisel, the operator manipulates the chisel edge underneath an RPM to pry it off the road.

In the original proposal to Georgia DOT for development of an automated RPM placement system, the proposed concept for RPM removal used a scarifier. The requirement was to remove RPMs while moving continuously at 5 mph or faster. Based on the information at the time of the proposal writing, this concept seemed feasible. After further investigation, we questioned whether the scarifier was adequate for RPM removal for several reasons. The main concern was whether or not the scarifier had the power required to machine an RPM from the roadway. In addition, Georgia DOT found that scarifiers did not work well on open graded asphalt used on Interstate highways. The scarifier tended to pull out chunks of the asphalt. This is not a problem on the “smooth” asphalt used on most secondary roads. Towards the end of the RPM placement effort in Fiscal Year 2006, an RPM removal internal research and development effort (IRAD) was initiated to evaluate the performance of using milling operations as opposed to the use of a scarifier for RPM removal.

One set of experiments was conducted to measure the specific cutting energies of the plastics used to manufacture RPMs. This property was then used to estimate the cutting power required to machine RPMs from the roadway surface at a travel speed of 5 mph.

Another set of experiments was conducted to measure the shear force required to remove RPMs from the roadway surface, both newly place markers and markers that had been in service for a year or more. Based on these measurements, we concluded that the adhesive could not serve to fixture the marker during a machining operation.

During the course of the IRAD project, Steve Robertson, the lead engineer on the project, developed a proprietary cutter design for use in machining RPMs from the roadway surface. This cutter would not require that the marker be fixtured for machining.

In a final set of experiments, the cutting forces were measured for several small-scale variations of this proprietary cutter design. This cutter was scaled in both dimensions and performance. [1]

Table 5 Cutter Parameters

	Small-scale Cutter	Full-scale Cutter
Depth of cut	0.125 inch	0.8 inch
Width of cut	0.22 inch	5 inch
Feed rate	50 inch/min	5280 inch/min (5 mph)

IRAD Project Results and Recommendations

The summary of our test results are as follows:

The adhesive force holding the RPM to the roadway varies from virtually zero to 1200 lbf,

- the specific cutting energy for the plastics used to manufacture RPMs is around 0.04 hp-min/in^3,
- the experimental saw has the potential to effectively and safely machine a RPM from the roadway whether it is bonded to the roadway or not,
- extrapolating the cutting force data to much higher feed rates also means much higher surface speeds which could melt the RPM plastic.

Our approach to RPM removal focused on either removing whole RPMs with a blade type device or machining RPMs into chips with a scarifier type device. Any device that removed the whole RPM was eliminated based on GDOT's experience with a similar device and its potential for damaging the road.

Our recommendation is a solution that would remove RPMs by machining them from the roadway. In practice, the removal device might have one or more cutting heads. Each cutting head would have a cutter or stack of cutters about 6 inches wide. The cutter(s) width should exceed the width of the widest RPM by at least 1 inch. A single head cutter would have to machine a RPM in about 40 milliseconds. Although, the energy to machine a RPM is not substantial, the power to machine it in 40 milliseconds is still over 200 HP. One potential solution is to use the cutter as a flywheel to store the energy. A 12-inch-diameter cutter, 6-inches-wide, made from steel and rotating at 4000 rev/min will store 8.5 times the energy required to machine one RPM. This cutter rotating at 4000 rev/min would lose 240 rev/min after cutting one RPM. We also recommend further development of the experimental cutter. This would include a 12-inch-diameter version that is at least 1-inch wide for testing at 5 mph.[2]

Proposal History

A proposal based on the IRAD project was prepared and submitted to Georgia DOT in 2006. A supporting Needs Statement was submitted and was revised one or more times in the intervening years. Early in 2010, Rick Deaver of Georgia DOT notified Jonathan Holmes that the project had been selected for funding. Jonathan resubmitted a revised version of the 2006 proposal.

Project Objectives

- Work with Georgia DOT to develop a specification for RPM removal.
- Scale up the GTRI-proprietary cutter design from the existing 2-inch-diameter by 0.220-inch-thick cutter to a full-scale (approximately 12-inch-diameter by 5-inch-wide) cutter.
- Instrument and test the full-scale cutter on RPMs using an electrically-driven spindle and base fabricated for this purpose.
- Evaluate test data and refine requirements for a RPM removal system.
- Develop conceptual design for vehicle and supporting systems necessary for RPM removal using the GTRI-proprietary cutter.
- Identify commercial sources for major system components including high-speed spindle, vacuum system, and vehicle platform for RPM removal system.[3]

Operational System Requirements

The project proposal contains the following list of requirements for an operational system. On May 9, 2011, a meeting was held with Georgia DOT representatives to review the systems requirements. The results of that review are included on Page A-7. The requirements that are relevant to the full-scale cutter development are highlighted below. Two topics from the May 9 discussion are most relevant to the cutter development effort. One is the suggestion to make measurements of the height above the surface of installed markers. The second is to assess the cutter blade wear and to consider methods that might be used to evaluate cutter wear on cutters that are in service.

Proposed preliminary requirements for RPM removal system

- Remove the majority of the RPM to within a fraction of a 1/16 inch above the surrounding road surface.
- Collect the RPM and transfer into a container for disposal.
- Average speed will be 5 mph.
- Operate only in dry conditions.
- On multi-lane highways remove the single RPM along the dividing skip lines.
- On two lane highways, remove one or both RPMs along the centerline in one pass. This will allow for concurrent removal and replacement operations.
- Replacement RPMs will not be placed at the same location as a previous RPM. The placement operation will provide for offsetting new RPMs from the previous placement location.
- Accommodate obstacles such as manhole covers, rumble strips, etc.
- Minimize damage to paint stripes.
- Minimize overhang into adjoining lane.[4]

Requirements for RPM removal system agreed upon at May 9, 2011 meeting

Several requirements were discussed that resulted in the following list of considerations and/or requirements. Most of these only affect the system level conceptual design; however, some of the requirements will be implemented into the laboratory testing device being built under the contract.

1) The RPM removal system needs to be able to accommodate concrete and asphalt based pavements. In particular, the heaves that are present in concrete surfaces and the height changes where asphalt lanes meet need to be considered in the system prototype design. One operation agreed upon was that the cutter should be lifted between cutting activities to limit the chance of accidental crashes or cutting of items such as the steel in the expansion joints of bridges.

2) A target of 1/16" maximum material remaining on the road seemed reasonable. There is some concern that on concrete pavement surfaces the 1/16" may not be enough to fully remove the marker and leave some plastic visible on the adhesive. Grady Jones suggested supporting a small field exercise that would provide GTRI traffic control to take measurements on marker heights and adhesive thicknesses on different pavement types. Wiley and Jonathan see value in this exercise and would like to fabricate a specialized height gauge to make this measurement easy and repeatable to capture several thicknesses on different marker and pavement types.

3) The thickness of the thermoplastic material can be as much as .090", which is thicker than the 1/16" (.062") that will be used in the design. Some provision may need to be made to prevent milling the paint stripe on the conceptual design.

4) Most major roadways will have a maximum of 2 to 3 markers at a single location on the road. The goal of the system will be to remove all these markers at one time, and at the desired speed of 5mph. There are some cases where 6+ markers can be in the same location on less travelled roads. In these areas, it was agreed upon that the speed could be varied and slowed to 3 mph at times to allow the system to handle these situations. The case where there are 6+ markers was considered an exceptional case.

5) Removal of the marker will likely generate fine plastic particles after the removal operation. As a component of a marker replacement system, there was some concern that these plastic particles may be problematic. If a problem arises, it was agreed upon that blowing the excess particulates from the road would be acceptable prior to placing a new marker in a fully automated system.

6) Several situations were discussed as far as marker placements on the road and the need for flexibility to handle double lines, skip lines, etc. The two cases that need to be supported are skip lines where a single marker is placed on every other skip, and a solid single line where two markers are placed approximately 12" apart. Although there was discussion about addressing the single line case with a 12"+ wide cutter, this may be undesirable due to the potential for uneven pavement on a paving seam.

7) Overhang needs to be considered carefully. The current overhang of the RPM placement system camera was not preferable. Overhang will need to be minimized in order to minimize impact to traffic in the adjacent lane.

8) One goal of the laboratory prototype tests is to characterize the wear of the cutter blade. The diameter of the blade can be measured precisely then tool wear equations will be used for building a model to estimate the wear properties of the cutter. It is essential that some estimate of wear is made as the viability of the unit will depend on the ability for the cutter to survive some reasonable length of time. There was also discussion that the existing cutter could potentially be remachined and reused several times. This was an interesting suggestion and one thought was this could possibly be accomplished with electrical discharge machining operations. This also created an implied requirement of making the cutting blade easily replaceable. One final consideration for wear is how to inspect the wear on an existing blade. Perhaps there could be wear indicators present on the blade to provide maintenance staff with a visual indicator of extreme wear.

The previously described requirements list will serve as design guidance for the test device as well as the conceptual design.

Proposed Conceptual Design

The results of a GTRI-funded IRAD project on RPM removal have led to a preliminary selection of a RPM removal method. Key aspects of this method include:

- A proprietary cutter design will be used to prevent the cutting forces from causing the marker to be dislodged and launched,
- A high-speed, horizontal-axis spindle will be used to turn the cutter,
- The cutter drum will be designed as a flywheel to provide the necessary energy to mill away the entire marker within 40 msec.
- Carbide cutter inserts will be used to reduce maintenance requirements and to minimize the damage resulting from pavement strikes,
- Abrasion-resistant, steel shoes will support the mechanism off the pavement.
- A vacuum system will be used to remove the chips,
- Powered brushes, compressed air nozzles, and sensors will be used to prevent debris from impacting and damaging the cutter.[5]

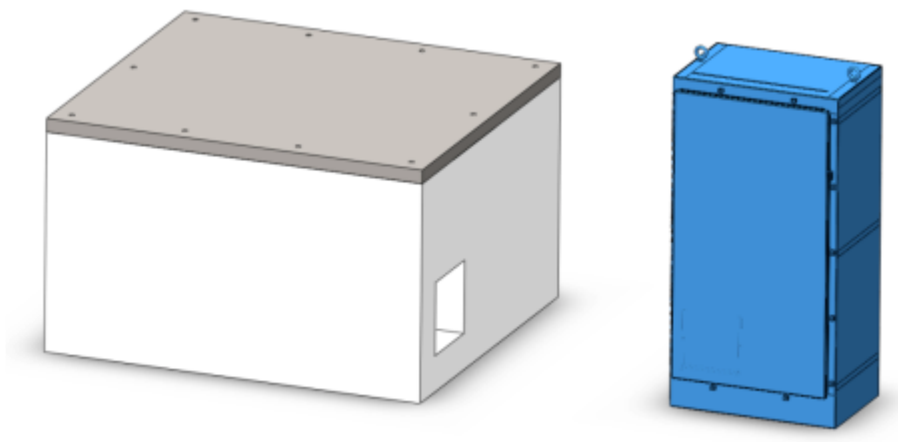


Figure 28. Concrete and steel containment and cabinet for controls

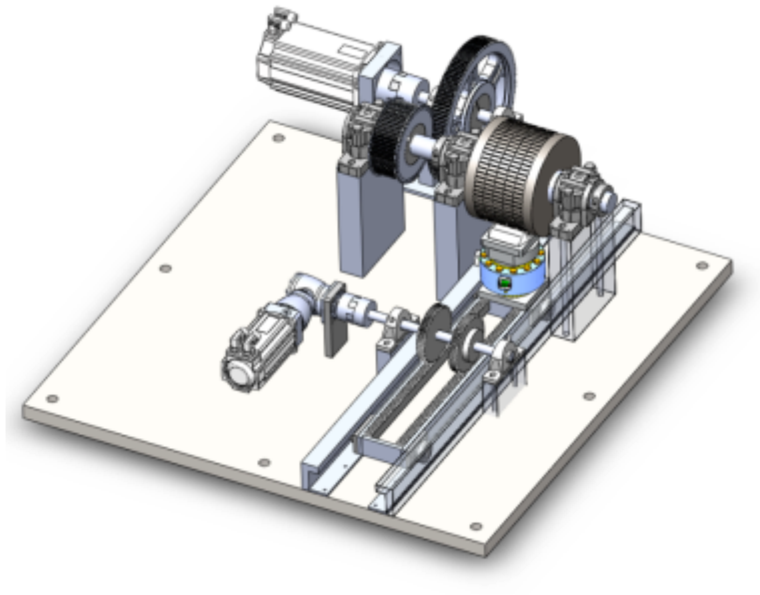


Figure 29. Cutter, cutter drive, marker carriage, and carriage drive

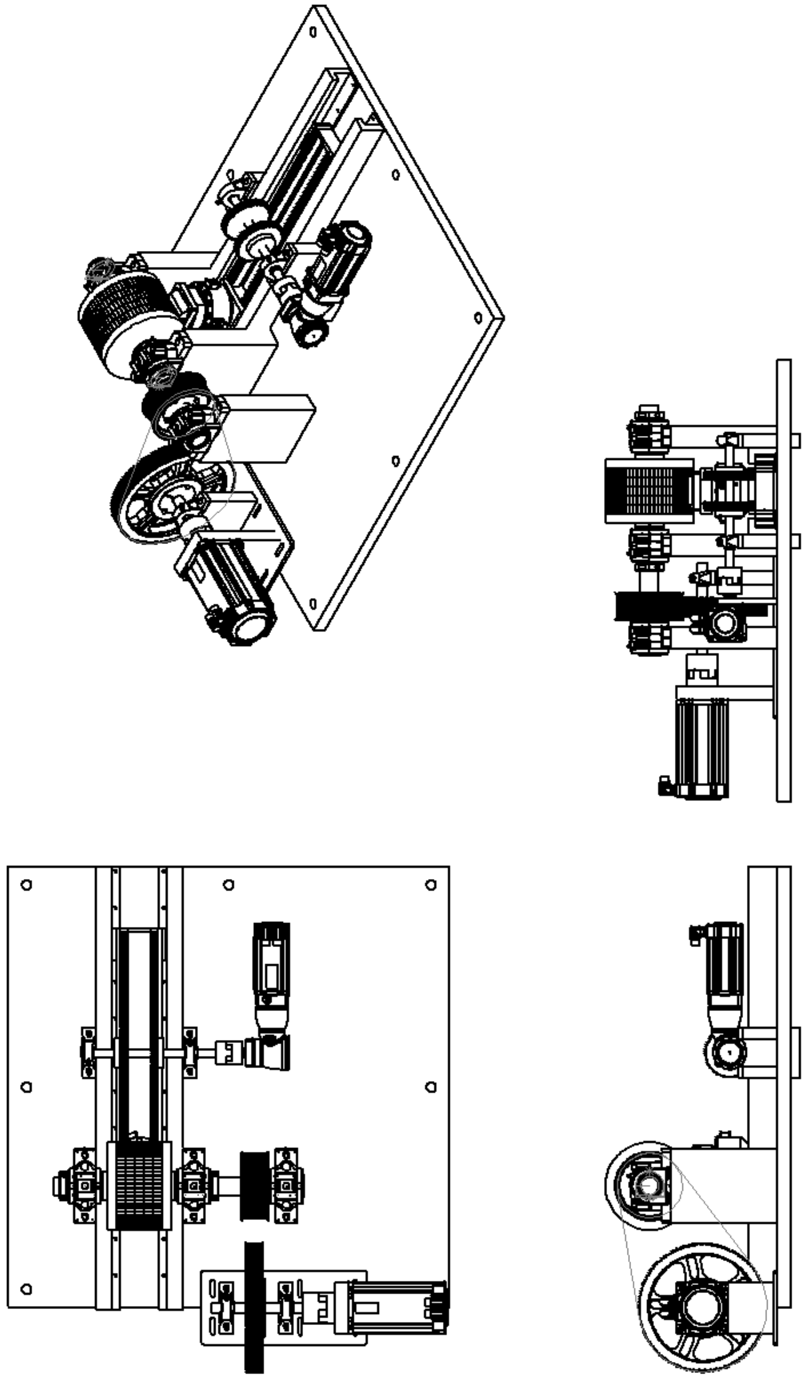


Figure 30. 3-view drawing of experimental system

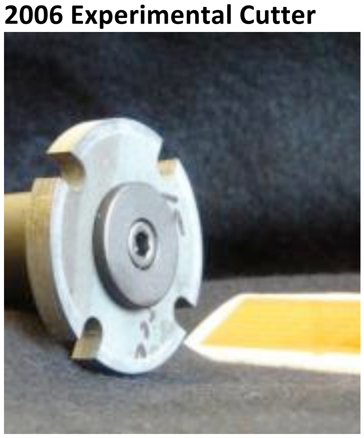
Full-Scale Cutter Design

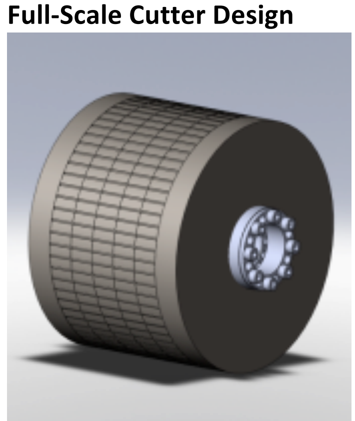
The conceptual design for the cutter, as described in the IRAD project final report, is for a stack of 6, 1- inch-wide, 12-inch diameter cutters, made from steel with carbide teeth inserts, rotating at 4000. (See highlighted text on Page A-4.) Such a cutter rotating at 4000 rpm would store 8.5 times the energy required to machine one RPM. This cutter rotating at 4000 rpm would lose 240 rpm after cutting one RPM.

Initial design effort was based on these dimensions and the use of carbide inserts as well as an attempt to arrange the teeth in a helical pattern such that the chips could be cleared axially. The use of replaceable carbide inserts added significant complexity to the design and limited the number of cutter teeth. The 12-inch-diameter dimension would necessitate fabrication by an outside vendor. The cutter design presented in this design review is based on the following design decisions:

- Use the same feed per tooth (i.e. chip size) as that of the experimental cutter, 0.02 inch per tooth. 66 teeth are required for 5280 inch/min at 4000 rpm.
- Do not use carbide tooth inserts. Fabricate the cutter disks from hardened or case hardened tool steel. Machine the cutting teeth into the cutter disk.
- Reduce cutter diameter to 9.6 inches such that cutter disks can be fabricated on the GTRI Machine Services wire EDM machine.
- Clear the chips radially rather than axially.
- Use a stack of six 1-inch-wide cutter disks with two 1-inch-wide end disks for additional flywheel inertia.

Table 6 Cutter Parameters[6]





	2006 Experimental Cutter	Full-Scale Cutter Design	
Diameter	2	**9.6**	**inch**
Number of teeth	4	**66**	
Radial rake angle	15°, 30°, 45°	**15°**	
Radial clearance	0 at feed of 0.02 inch/tooth	**0 at feed of 0.02 inch/tooth**	
Axial rake angle	0°	**0°**	
Axial depth of cut	0.22	**4.54**	**inch**
Radial depth of cut	0.125	**0.8**	**inch**
Feed per tooth	0.02	**0.02**	**inch/tooth**
Spindle speed	417-884 rpm	**4000**	**rpm**
Feed rate	50	**5280**	**inch/min**

Extrapolation of cutting forces

Our cutting force measurements show the actual cutting force is proportional to the axial depth of cut, radial depth of cut and feed per tooth (or uncut chip thickness).[7]

Design decisions:

- Estimate cutting force based on extrapolation of force measurements made on experimental cutter during IRAD project. Assume that cutting force is proportional to axial depth of cut. Assume that cutting force is proportional to radial depth of cut. Assume that cutting force is proportional to feed per tooth.

Table 7 Cutter Force Extrapolation[8]

	2006 Experimental Cutter	**Full-Scale Cutter Design**	
axial depth of cut multiplier		20.64	
radial depth of cut multiplier		6.4	
feed per tooth multiplier		1	
combined multiplier		132.1	
x-direction peak force measurement	22	**2906**	**lbf**
y-direction peak force measurement	79	**10434**	**lbf**

Note: Many of the following analyses and component selections are based on the extrapolated cutting forces. The forces have been extrapolated by a **factor of 132** beyond the experimental conditions. Initial testing using a single, 1-inch-thick cutter disk will be required to assess the validity of this large extrapolation.

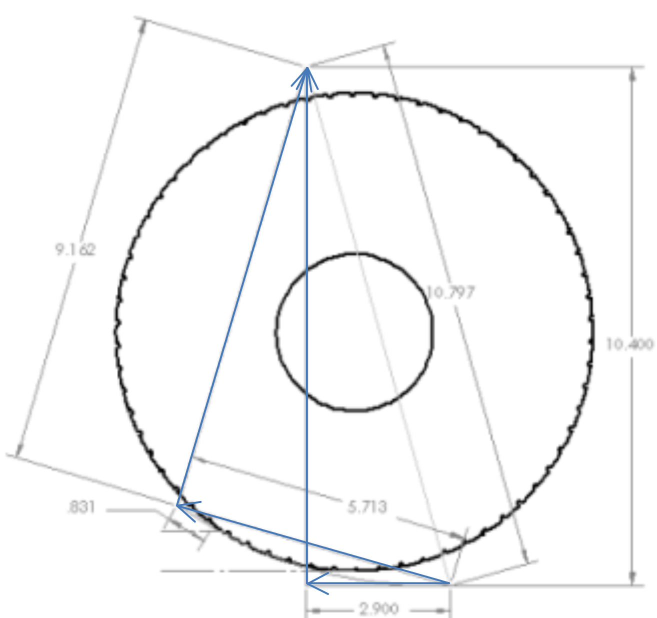


Figure 31. Graphical Force Vector Solution, Extrapolated x-, y-forces and equivalent tangential-, radial-forces 1 inch = 1000 pounds

A SolidWorks sketch has been generated to provide a graphical solution of the cutting forces. The forces are scaled by a factor of 1/1000. The cutting force dynamometer resolved forces in the vertical and horizontal directions. The extrapolated forces are 10,400 and 2900 pounds. The resultant is 10800 pounds. For a radial depth of cut of 0.8 inches with a 66-tooth cutter, a maximum of 7 teeth are engaged. Assume that the cutting force on each tooth is tangential and that the cutting forces are equal. The radial force component results from contact between the edge of the cutter disk and the cut face of the marker. The graphical solution gives a resultant tangential force of 5713 pounds at the center tooth. The resultant radial force is 9162 pounds. The tangential force results in a moment of 27,422.4 inch pounds (3,098.3 N m).

Cutting Energy[9,10]

During the IRAD project, the specific cutting energy was determined for three plastics commonly used to manufacture RPMs. All were on the order of 0.04 HP min/in^3 at an undeformed chip thickness of 0.04 inch. An empirical formula can be used to adjust the specific cutting energy for other chip thicknesses. The specific cutting energy would be on the order of 0.05 HP min/in^3 at an undeformed chip thickness of 0.02 inch. For the marker manufactured by 3M, the energy required to mill the marker would be 12780 joules.

specific_cutting_energy at .04 inch/tooth	0.04	hp*min/in^3				
specific_cutting_energy at .02 inch/tooth	0.049	hp*min/in^3				
3MRPM_volume	5.8	in^3				
milling_energy at .04 inch/tooth	0.232	HPmin	7656	lbf ft	10380.14	Joules
milling_energy at .02 inch/tooth	0.285626	HPmin	9425.642	lbf ft	12779.45	Joules

Flywheel Energy Storage

Note: The specific cutting energy of representative plastics was determined experimentally during the IRAD project. No cutting experiments were performed on the epoxy-filled markers manufactured by Apex. To provide an upper bound, the total milling energy was scaled by the mass ratio of the markers. Initial testing using a single, 1-inch-thick cutter disk will be required to assess the validity of this cutting energy requirement for the 3M markers and to measure the cutting energy requirement for the Apex markers.

	Avery-Dennison C80		**Apex 921, energy scaled by mass**			
marker mass	88		229		g	
mass moment of inertia of cutter assembly	0.4756				kg m^2	
starting angular velocity	**4000**	**4800**	**4000**	**4800**	rev/min	
starting angular velocity	419	503	419	503	rad/sec	
starting kinetic energy	41724	60083	41724	60083	N m (J)	
ending kinetic energy	28945	47304	8447	26806	N m (J)	
ending angular velocity	349	446	188	336	rad/sec	
ending angular velocity	3332	4259	1800	3206	rev/min	
cutting energy requirement as % of total	30.6%	21.3%	79.8%	55.4%		

Cutter Disk Stress Analysis

Design decisions:

- The cutter will consist of a stack of one-inch thick disks.
- The disks will be fabricated from tool steel plate using a wire EDM machine.
- After fabrication, the disks will be hardened.
- After fabrication, the disks will be balanced.
- The cutter assembly will be connected to the main shaft by one or more Fenner B-LOC bushings.
- Initial testing at full rotational speed and full feed speed will be conducted on a single, one-inch thick disk.

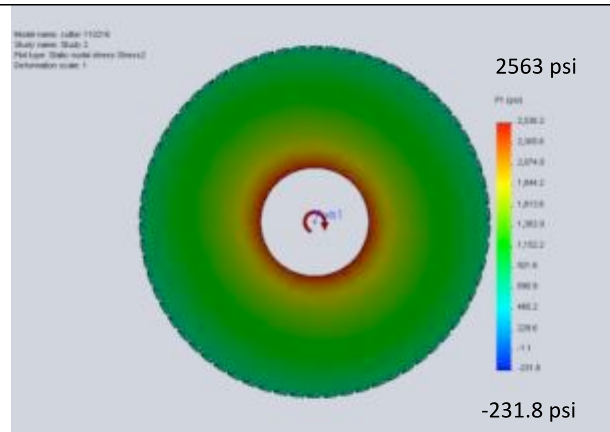
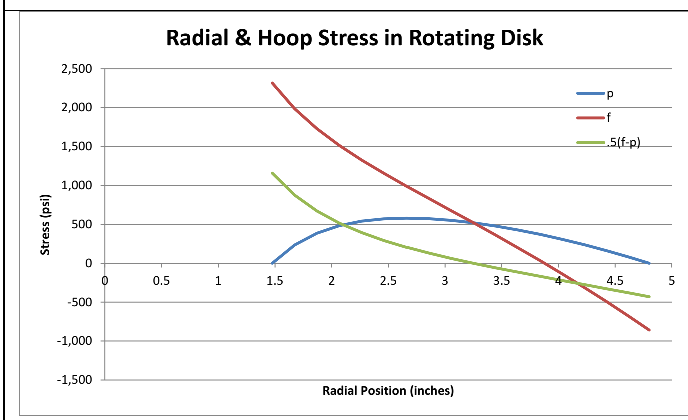
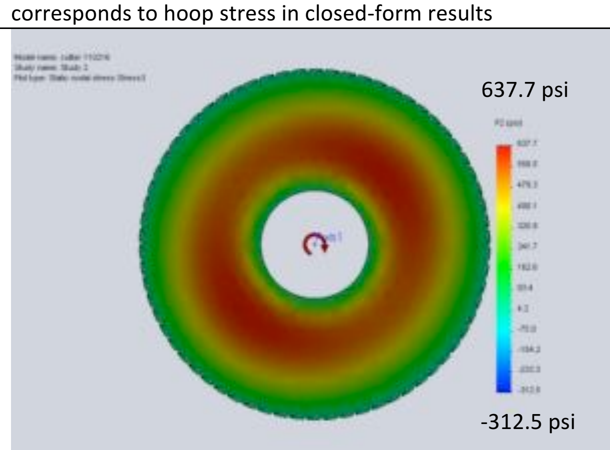
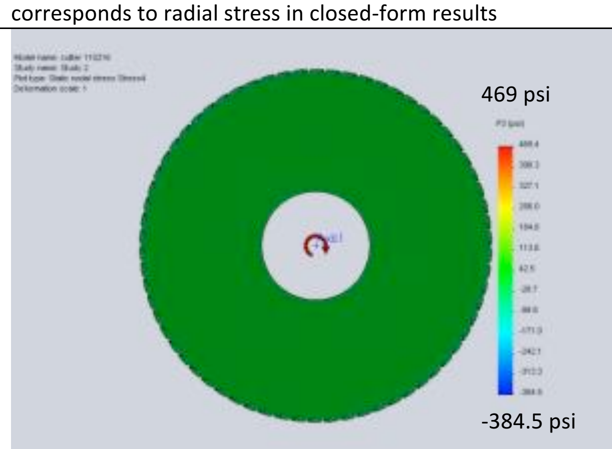
Remaining design decisions:

- How many Fenner B-LOC bushings will be used to connect the cutter stack to the shaft, one, two, or three?
- How will the individual cutter plates be connected to each other? Will there be a circular pattern of fasteners connecting the stack of cutter disks?
- How will the cutter assembly be balanced; as an assembly or as individual components?
- Can the cutter disks be balanced after hardening?

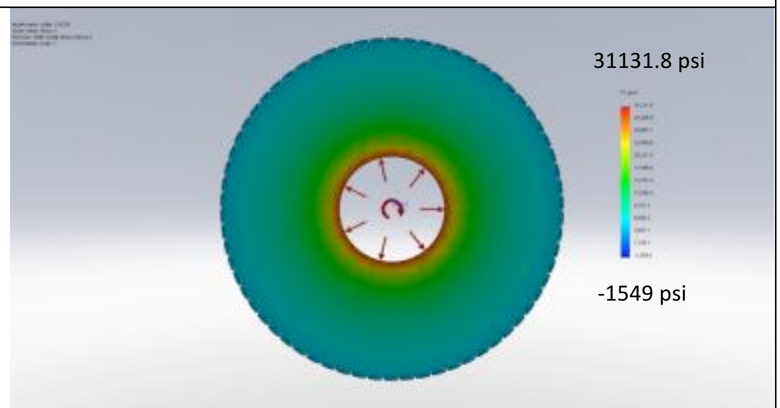
The cutter disks will be subjected to loads from several sources. The cutter disks will spin at the design rotational speed currently set at 4000 rpm. The cutter will be connected to the main shaft by one or more Fenner B-LOC bushings that use wedges that result in an internal pressure load on the cutter disk bore. The cutting operation will subject the cutter teeth and rim to large tangential and radial loads. Results are presented for the centrifugal load and for the combined centrifugal and pressure loads. Closed-form stress analysis results are compared to results from SolidWorks Simulation. Only a finite element analysis was done for the combined centrifugal, pressure, and cutting loads.

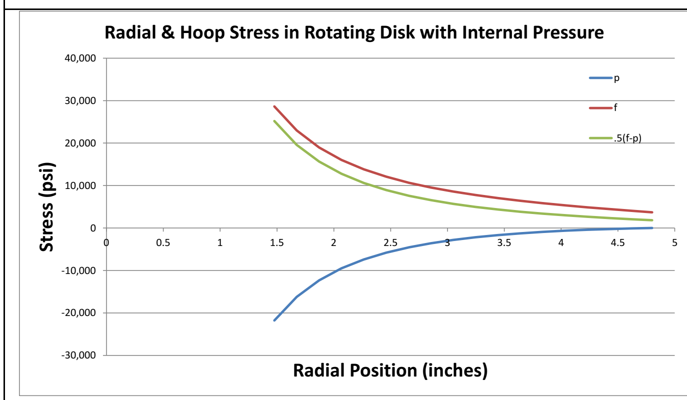
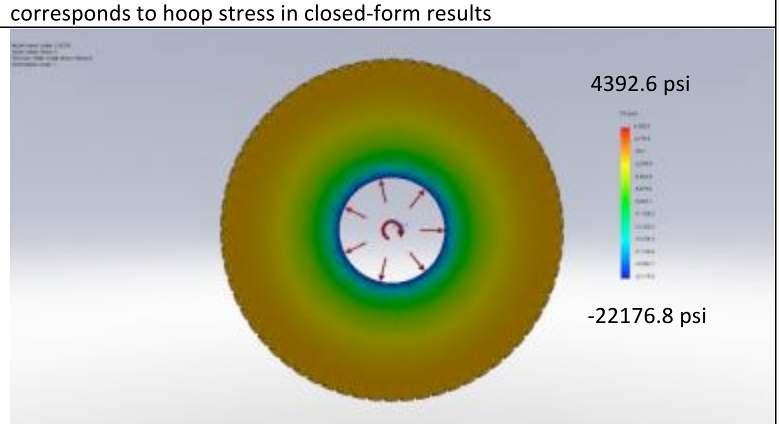
We expect to manufacture the cutter disks from 1-inch-thick alloy plate such as AISI 4140. The yield stress for annealed 4140 is on the order of 60,500 psi.

Inertia Stresses in Rotating Cutter Disk

Closed-form solution[11,12,13]	**SolidWorks Simulation Results, Study 2-1**[14]
R_1 0.0375m R_2 0.12192m nu 0.3 g 9.807m/sec^2 omega 418.9rad/sec (4000 rpm) rho 7750.37332kg/m^3	***Load:*** Centrifugal Load, Inertia-relief constraint -419 radians/sec (4000 rpm) about Axis 1 ***Part File:*** S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter 110216.SLDPRT ***Material:*** AISI 4340 Steel, annealed
	 1st Principal Stress, corresponds to hoop stress in closed-form results
	 2nd Principal Stress, corresponds to radial stress in closed-form results
	 3rd Principal Stress, corresponds to axial stress, equal to 0 in closed-form results

Combined Stresses in Rotating Cutter Disk from Inertia and Internal Pressure from Keyless Bushing

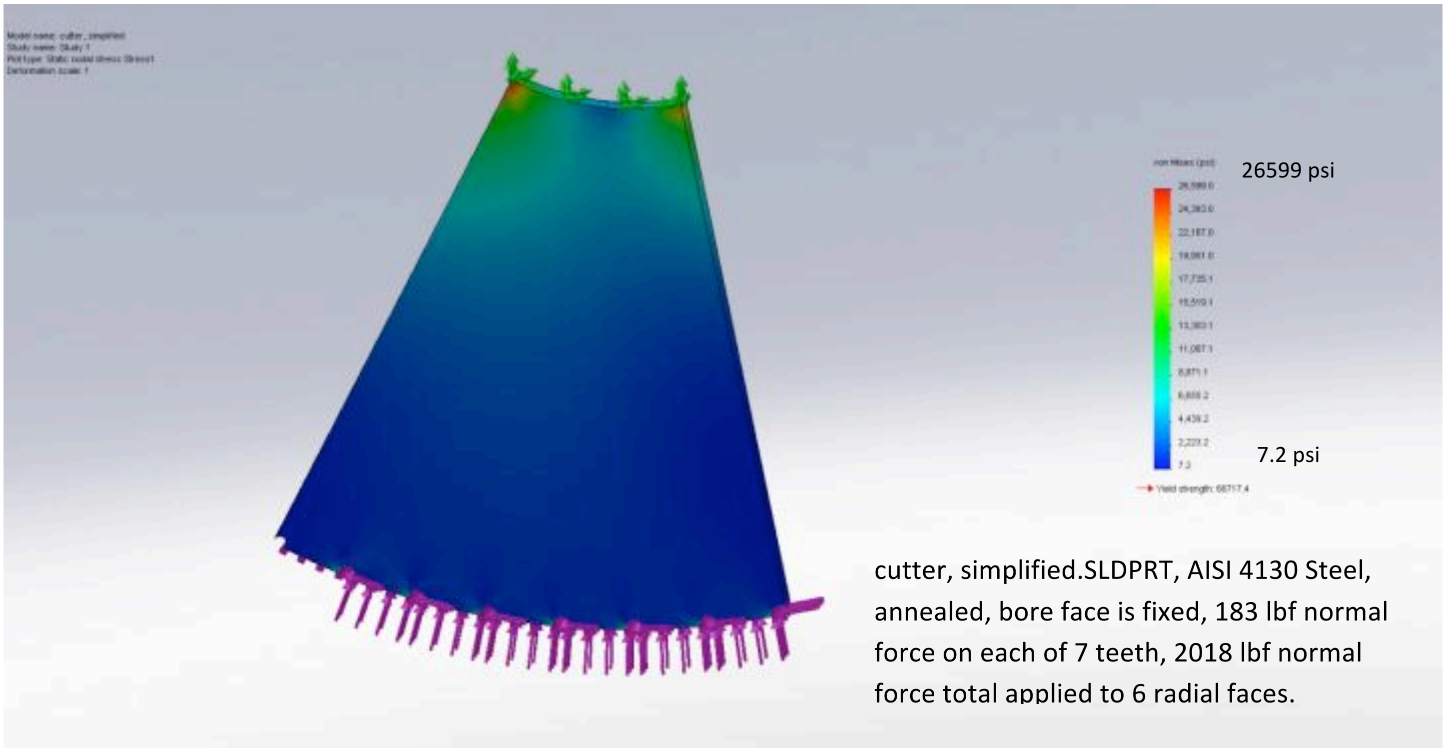
<table>
<tr><th>Closed-form solution</th><th>SolidWorks Simulation Results, Study 3-1[15]</th></tr>
<tr><td>R_1 0.0375m
R_2 0.12192m
nu 0.3
g 9.807m/sec^2
omega 418.9rad/sec (4000 rpm)
rho 7750.37332kg/m^3
q 150N/mm^2</td><td>Load: Centrifugal Load: -503 radians/sec (4800 rpm) about Axis 1, Pressure load: 150 N/mm^2 on bore, Inertia-relief constraint
Part File: S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter 110216.SLDPRT
Material: AISI 4340 Steel, annealed</td></tr>
<tr><td></td><td>

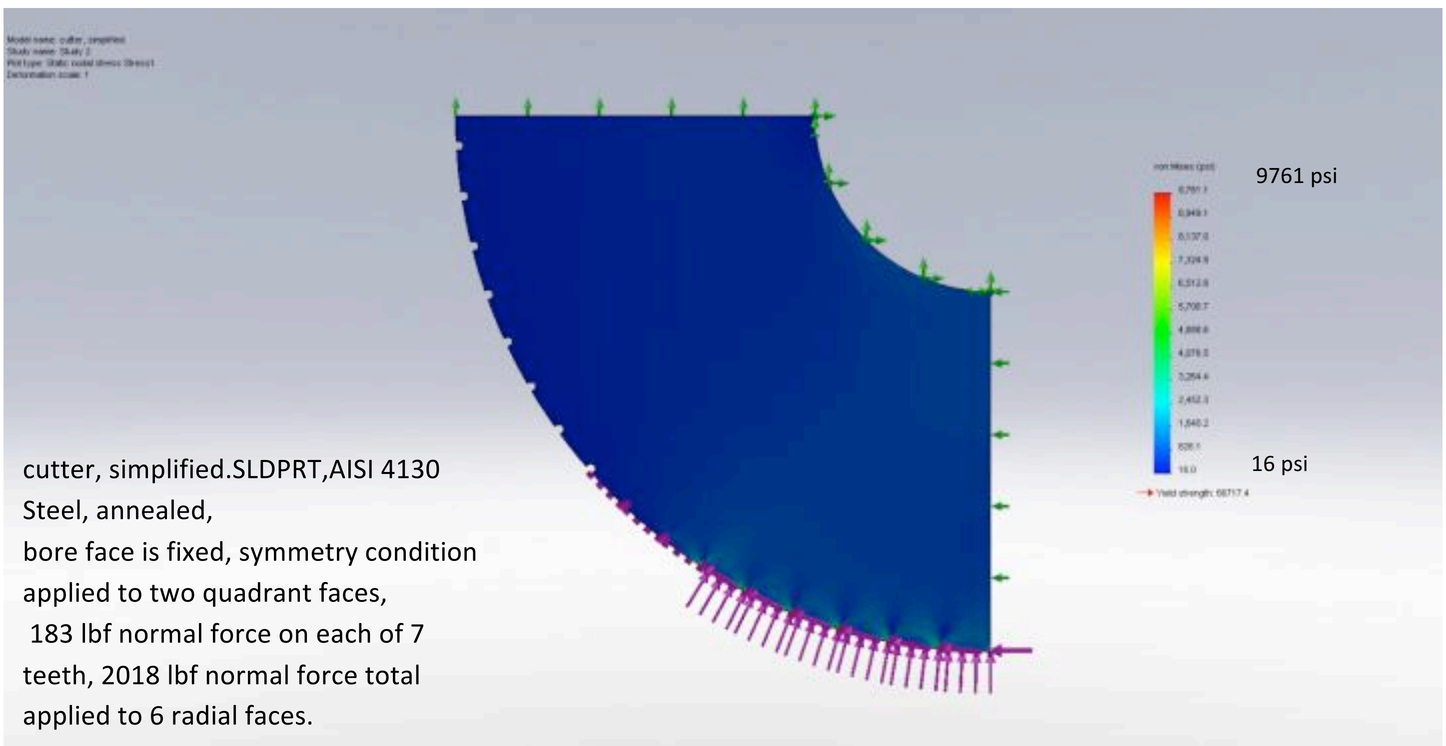
1st Principal Stress,
corresponds to hoop stress in closed-form results</td></tr>
<tr><td>
</td><td>

2nd Principal Stress,
corresponds to radial stress in closed-form results</td></tr>
<tr><td></td><td>

3rd Principal Stress,
corresponds to axial stress,
equal to 0 in closed-form results</td></tr>
</table>

Stresses in Cutter Disk from Cutting Forces[1,2]





[1] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter, simplified-Study 1-1.htm
[2] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter, simplified-Study 2-1.htm

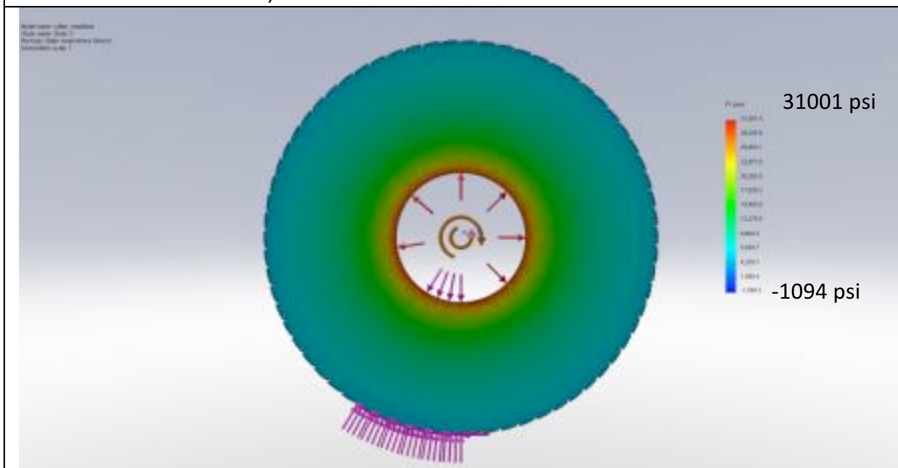
Combined Stresses in Rotating Cutter Disk from Inertia, Internal Pressure from, Bushing and Cutting Force[16]

SolidWorks Simulation Results, Study 3-1

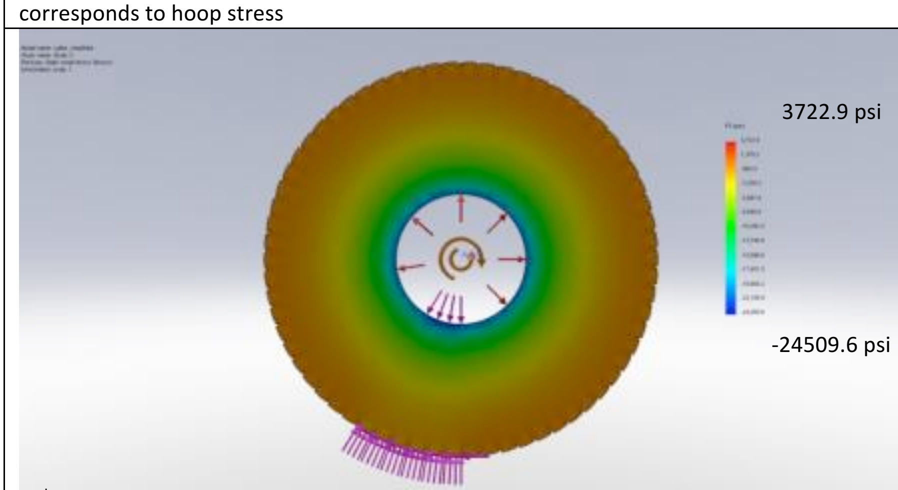
Load: Centrifugal Load: -419 radians/sec (4000 rpm) and 0.0006860 radians/sec^2 about Axis 1, Pressure load: 150 N/mm^2 on bore, Inertia-relief constraint, Normal force of 814 N on 7 teeth, total Normal force of 8977 N applied to a segment of rim, Normal force of 8977 N applied to a segment of bore

Part File: S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter, simplified.SLDPRT

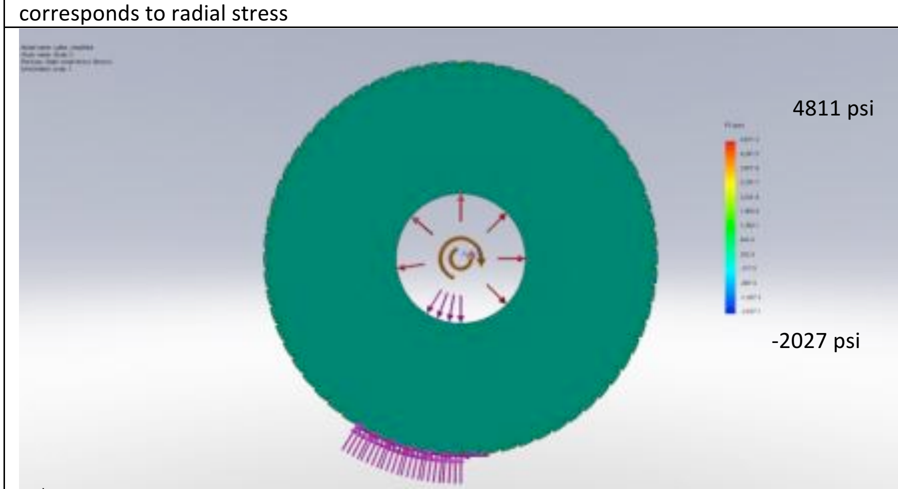
Material: AISI 4340 Steel, annealed



1st Principal Stress,
corresponds to hoop stress



2nd Principal Stress,
corresponds to radial stress



3rd Principal Stress,
corresponds to axial stress

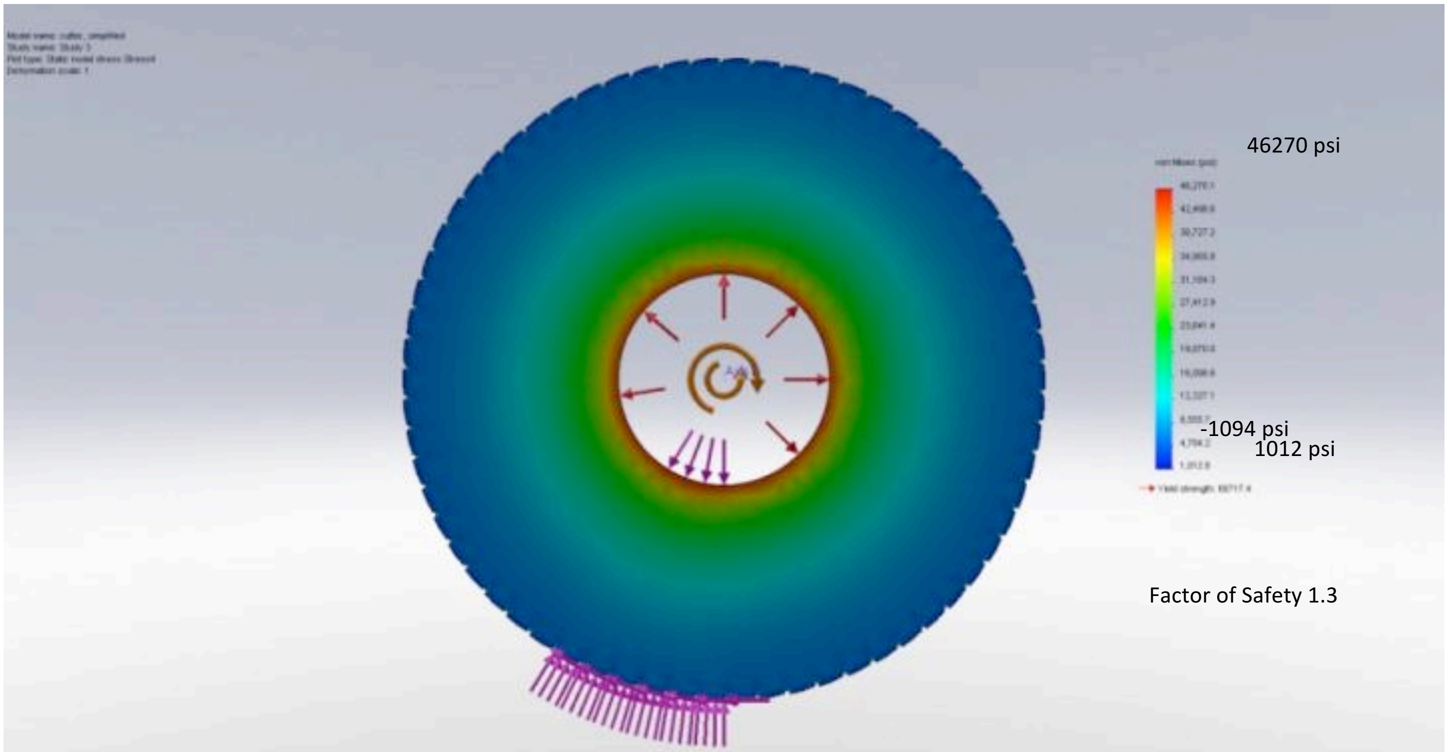


Figure 32. von Mises Stress

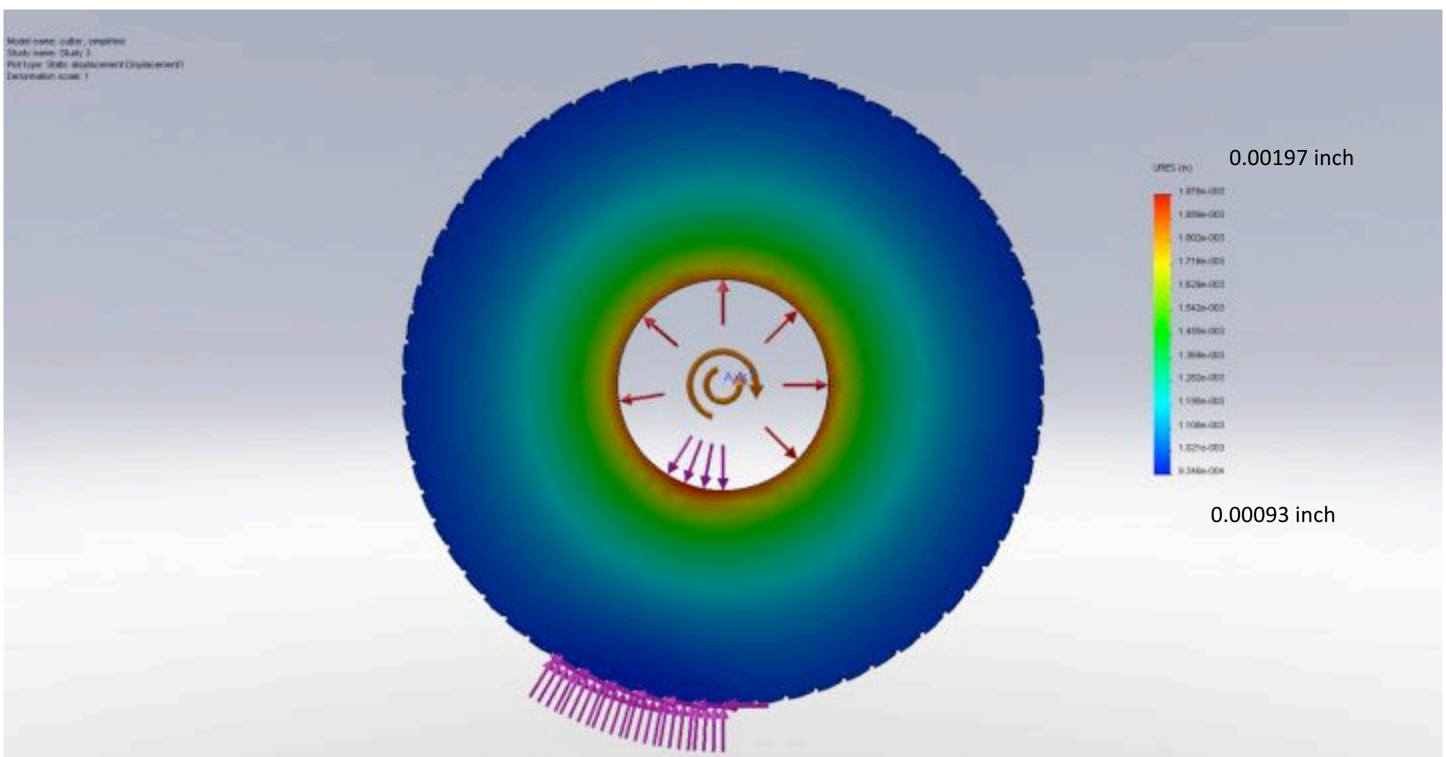


Figure 33. Resultant Displacement

Design of Mechanical Elements to Support the Full-Scale Cutter

The design and selection of the supporting mechanical elements are coupled with conflicting requirements. Rolling element bearings with larger bores can support higher radial loads, but have lower limiting speeds. Initial exploration began with SKF metric-dimension, self-aligning ball bearings with a 45 mm bore. The shaft length was based on the system conceptual design. The span between supporting bearings was constrained by the size of the load cell, the carriage bearings, and the rack-and-pinion carriage drive. Preliminary shaft stress analysis results along with a survey of readily available shafting materials suggested that the shaft diameter should be larger. The following selections are for 50 mm bore bearings, 50 mm diameter shaft, and 50 mm bore Fenner B-LOC keyless bushings.

Selection of Cutter Shaft Bearings

Selection criteria:

- Support main shaft on an identical pair of self-aligning, rolling element bearings.
- Add an identical third bearing to support the drive belt tension load. Use alignment to minimize rotating misalignment load.
- Base the dynamic radial load requirement on extrapolated cutting force, 10,830 lbs (48,174 N).
- Assume axial load is low.
- Rotational speed requirement – minimum 4000 rpm, desirable 5000 rpm.
- Bearing life can be short.

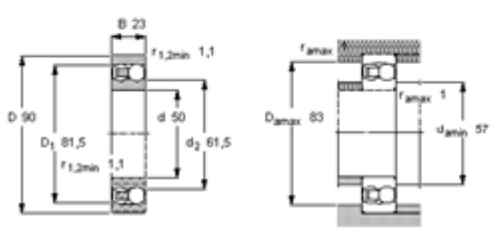
Selection tools available on the SKF website were used to first search for a 45 mm bore self-aligning bearing, then later for a 50 mm bore self-aligning bearing. In addition, an application engineer at SKF was consulted. An SKF *2210 ETN9* bearing was selected. An SKF *SNL 210* housing is required to support the bearing and provide lubrication. Locating rings can be included or not such that the bearing is located axially or floats axially. Seals can be added to the housing. The seals are designed to contact a 60 mm diameter shoulder. Fabricate and use 50 mm ID X 60 mm OD sleeves in pairs for each bearing. Provide internal O-ring groove to prevent lubricant from passing between sleeve and shaft. The pair of sleeves around the main bearing that is located axially with locating rings will locate the shaft axially. One sleeve will be located by the keyless bushing; the other by a shaft collar or by the drive pulley bushing.

Self-aligning ball bearings, cylindrical and tapered bore, cylindrical bore, unsealed
Product information

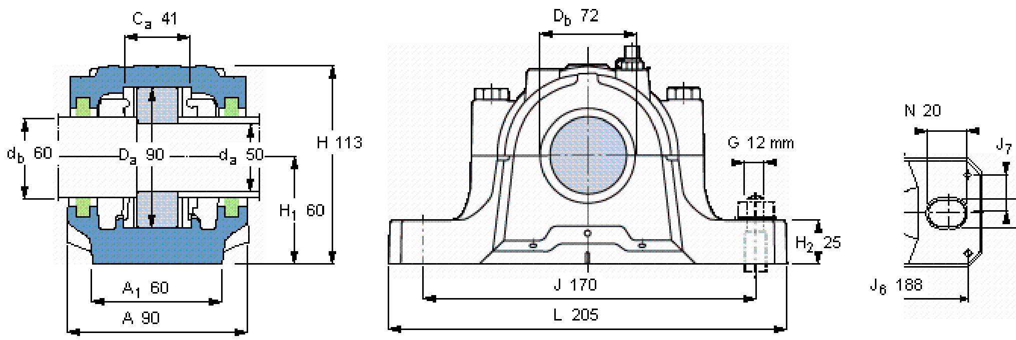
SKF

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	-
50	90	23	33,8	11,2	0,57	14000	9500	0,60	2210 ETN9

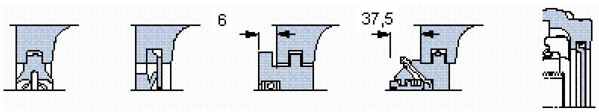


Calculation factors
e 0,23
Y_1 2,7
Y_2 4,2
Y_0 2,8



Order designations
Housing only SNL 210
Seals **End cover**

-	TSN 210 A	2 x TSN 210 S	2 x TSN 210 ND	ASNH 512-610



Appropriate SKF bearings and accessories

Bearing	Lock nut	Locking device	Locating rings
1210 ETN9	KM 10	MB 10	2 x FRB 10,5/90
2210 ETN9	KM 10	MB 10	2 x FRB 9/90
-	-	-	-
22210 E	KM 10	MB 10	2 x FRB 9/90
C 2210 TN9	KMFE 10	-	2 x FRB 9/90

Cap bolts

Size [mm]	M 10x55
Rec. tightening torque [Nm]	50

Max permissible load [kN]

$F_{120°}$	50
$F_{150°}$	30
$F_{180°}$	25

Appropriate attachment bolts

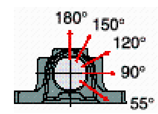
Size [mm]	12
Rec. tightening torque [Nm]	80

Dowel dimensions

Diameter (max)	6
Centre distance J_6	188
Centre distance J_7	22

Breaking loads, housing [kN]

$P_{0°}$	170
$P_{55°}$	265
$P_{90°}$	155
$P_{120°}$	120
$P_{150°}$	110
$P_{180°}$	130
P_a	85



The limit for $P_{0°}$ applies only when the housing is not supported over its entire base area

Grease quantities [kg]

Initial fill	0,075
Regreasing	0,01

Sizing of Cutter Shaft[17,18,19]

A simple beam theory analysis was conducted for the main shaft. Both the cutting force loads and the bearing supports were modeled as uniformly distributed loads. The drive torque load is ignored. The external pressure load from the Fenner B-LOC bushing is ignored. Any stiffness added by the cutter assembly itself is ignored. The singularity function method was used to write the load, shear, moment, slope, and displacement equations. A spreadsheet was used to solve for the constants of integration and to plot the values of the equations over the length of the beam. The results from the singularity function method were compared to the results given in Roark for a simply supported beam under a uniformly distributed load.

$$\frac{d^2M}{dx^2} = +p = -p_0\langle x-0\rangle^0 + p_0\langle x-.9\rangle^0 + p_1\langle x-2.65\rangle^0 - p_1\langle x-8.65\rangle^0 - p_0\langle x-10.4\rangle^0$$

$$\frac{dM}{dx} = -V = -p_0\langle x-0\rangle^1 + p_0\langle x-.9\rangle^1 + p_1\langle x-2.65\rangle^1 - p_1\langle x-8.65\rangle^1 - p_0\langle x-10.4\rangle^1 + C_1$$

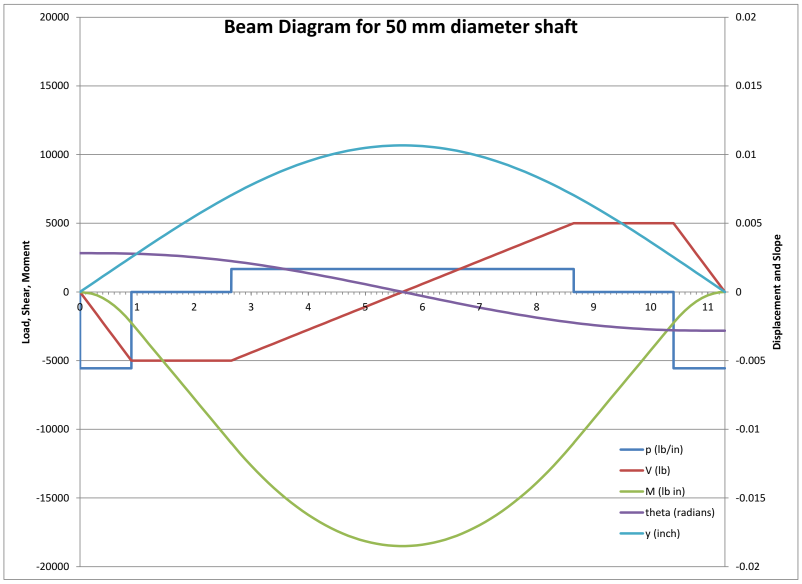
$$M = -\frac{p_0}{2}\langle x-0\rangle^2 + \frac{p_0}{2}\langle x-.9\rangle^2 + \frac{p_1}{2}\langle x-2.65\rangle^2 - \frac{p_1}{2}\langle x-8.65\rangle^2 - \frac{p_0}{2}\langle x-10.4\rangle^2 + C_1x + C_2$$

$$\theta = \frac{1}{EI}\left[-\frac{p_0}{6}\langle x-0\rangle^3 + \frac{p_0}{6}\langle x-.9\rangle^3 + \frac{p_1}{6}\langle x-2.65\rangle^3 - \frac{p_1}{6}\langle x-8.65\rangle^3 - \frac{p_0}{6}\langle x-10.4\rangle^3 + \frac{C_1x^2}{2} + C_2x + C_3\right]$$

$$y = \frac{1}{EI}\left[-\frac{p_0}{24}\langle x-0\rangle^4 + \frac{p_0}{24}\langle x-.9\rangle^4 + \frac{p_1}{24}\langle x-2.65\rangle^4 - \frac{p_1}{24}\langle x-8.65\rangle^4 - \frac{p_0}{24}\langle x-10.4\rangle^4 + \frac{C_1x^3}{6} + \frac{C_2x^2}{2} + C_3x + C_4\right]$$

							C1	C2	C3	C4
x	11.3	0	0.9	2.65	8.65	10.4	-5.63E-13	0	60268.75	0
		-5555.56	5555.555556	1666.666667	-1666.67	-5555.56				
p	-5555.555556	-5555.56	5555.555556	1666.666667	-1666.67	-5555.56				
-V	-2.38239E-12	-62777.8	57777.77778	14416.66667	-4416.67	-5000	-5.63E-13			
M	-2.42338E-27	-354694	300444.4444	62352.08333	-5852.08	-2250	-6.36646E-12	0		
theta	-0.002819546	-1336016	1041540.741	179781.8403	-5169.34	-675	-3.59705E-11	0	60268.75	
y	5.44624E-18	-3774244	2708005.926	388778.2296	-3424.69	-151.875	-1.35489E-10	0	681036.9	0

The shaft analysis assumes that the cutter assembly does not provide additional stiffness to the shaft. Given the high thrust capability of the connecting bushing (165,000 N), we expect the cutter assembly to provide significant stiffness to the shaft, primarily because bending of the shaft will put a portion of the cutter stack into axial compression. An Action Item has been added to consider analyzing this effect using SolidWorks Simulation and to consider how to



l	11.3	in			
wsuba	-884.956	lb/in			
wsubl	-884.956	lb/in			
shaft diameter	1.968504	in	50	mm	
2nd moment of inertia	0.737081	in^4	306796.2	mm^4	
section modulus	0.748874	in^3	12271.85	mm^3	
area	3.043424	in^2	1963.495	mm^2	
E	2.90E+07	psi			
alpha, section factor	1.333333				
		from singularity function equation			
Maximum M at l/2	-14125	-18498	in lbs		
Maximum theta at l	-0.00249	-0.00282	radians		
Maximum y at l/2	0.008789	0.01067	in		
Maximum V		5000	lbs		
Maximum normal stress	-18862	-24701	psi		
Maximum shear stress		2191	psi		

size the through bolts for the cutter stack.

Material Selection – Material selection will be influenced by availability. One potential source of a 50mm diameter shaft up to 31.5 inches in length is Misumi. The material is listed as 4137 Alloy Steel available with black oxide or electroless nickel plating finishes. I have not found any properties listed on the website; I inquired about properties through their website. (Misumi inquiry was acknowledged on 6/27/11 at 7 p.m.)

The calculated stress is less than ½ of the tensile strength at yield, 52200 psi, of AISI 4130 steel in the annealed state.[20]

Shaft Critical Speed[21,22,23,24]

The shaft critical speed comes from a calculation of the first natural frequency of the shaft as a beam. One method estimates the shaft critical speed using the deflection due to concentrated gravity loads and concentrated weights (or masses). A second method is based on a uniformly distributed gravity load. The shaft has been sized for the cutting force resultant of almost 11,000 pounds. The deflection due to the cutter static load is very small.

	14.67037	lbs	shaft weight		
	136.65	lbs	cutter weight		
W	136.65	lbs			
l	11.86	in			
a	5.93	in			
E	29000000	lb/in^2			
I	0.7370807	in^4			
y	-0.0002222	in			
g	386	in/sec^2			
f	209.77758	cycles/sec			
f	12,587	rpm	Concentrated Load		
f	18,119	rpm	Uniformly Distributed Load		

Critical Speed (rpm) of simply supported shaft with 137 lb cutter

	Cutter Mass	Cutter and Shaft Masses
Concentrated Load	12,587	12,271
Uniformly Distributed Load	18,119	17,218

Comments on Fatigue – The cutting force will apply a cyclic load to the shaft during the cutting operation. We expect that this cyclic load will be applied for a very small fraction of the running time of the system. The timing belt tension load will be a continuous cyclic load. That load will be very small compared to the cutting load. Once the timing belt design is complete, a fatigue analysis could be done.

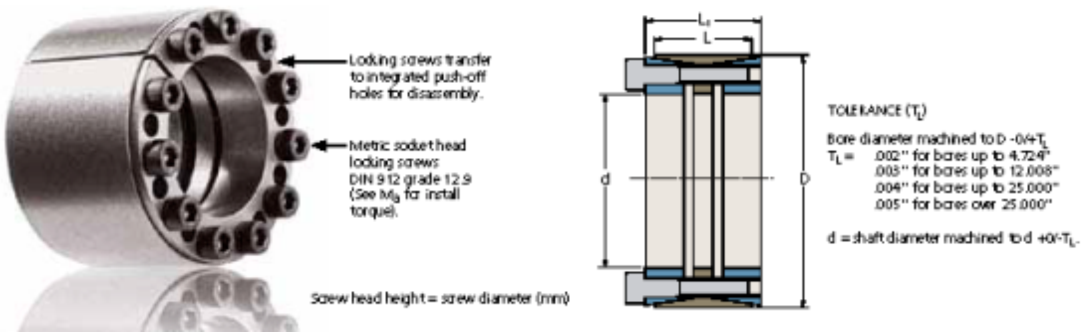
In a typical shaft application, the shaft would be designed for a constant torque load and a cyclic radial load. In this application, the motor used to drive the cutter shaft cannot supply the torque required for cutting. Kinetic energy stored in the cutter will power the cutting operation. For this application, a fatigue analysis would be done for the cyclic load only.

Selection of Keyless Bushing for Cutter

Use Fenner B-LOC Part Number B121050 (B112, Heavy Duty, Metric) keyless bushing to connect cutter assembly to main shaft. Use straight shaft with no shoulders, no keyway.

Series B112, B113& B115 ....

- Wide, double taper design for enhanced bending moment capacity
- Exceptional concentricity with thru-bored hubs
- No axial movement during installation
- Available in Standard, Heavy and Extra-Heavy Duty models



PartNumber	d	D	L	L1	Locking Screws		Ma	Mt	Th	Ph	DN*	Shipping
	(mm)	(mm)	(mm)	(mm)	Qty	Size	Install Torque	Maximum Transmitted		Hub	Minimum	Weight
								Torque	Thrust	Pressure	(mm)	(kg)
							(Nm)	(Nm)	(N)	(N/mm2)		
B121050	50	80	56	66	8	M8 x 55	41	4123	164918	124	122.3	1.2

The graphical solution on Page A-12 gives a resultant tangential force of 5713 pounds at the center tooth. The resultant radial force is 9162 pounds. The tangential force results in a moment of 27,422.4 inch pounds (3,098.3 N m). All cutting torque will come from the inertia of the cutter assembly. Even if the cutting torque was provided by a driven shaft, two Fenner B-LOC B121050 keyless bushings would be able to drive the cutter.

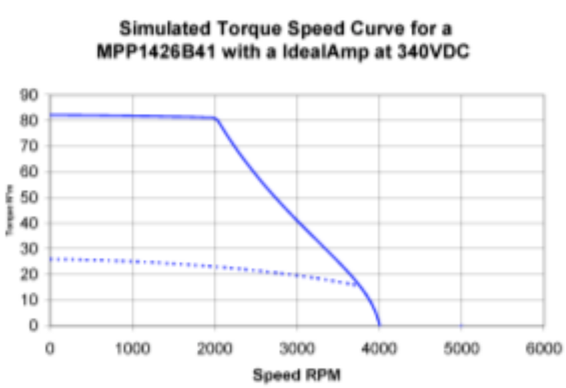
Design of Main Drive

We plan to salvage an existing drive system that was built to drive a custom, 2-axis, washdown robot. This includes not only the brushless DC servomotors, gearheads, and amplifiers, but also the complete, working electrical panel. This should save both a significant amount of money and a significant amount of labor that would have been required for design, fabrication, and integration.

Motor performance

The larger of the two brushless DC servomotors, a Parker MPP1426B1E-NPSN, will be used for spinning the cutter to the target rotational velocity, thereby charging it with the kinetic energy to be used for cutting. The motor performance curve is shown below. We plan to use a step-up transmission such that the cutter shaft velocity is at least twice that of the motor shaft. The table at the upper right shows the time required to charge the flywheel initially as well as the time required to recharge the flywheel after extraction of the necessary cutting energy. The charging times are shown for both peak and continuous motor output and for target cutter speeds of both 4000 and 4800 rpm.

The motor data is provided in the table at the right for a specific motor amplifier.



	Acceleration due to motor torque		
mass moment of inertia of cutter assembly	0.4756	$kg\ m^2$	
	Peak	**Continuous**	
motor torque	82.1	18.9	N m
shaft torque, 1:2 ratio	41.05	9.45	N m
acceleration	86.31	19.87	rad/sec^2
time, 0-4000 rpm	4.9	21.1	sec
time, 3332-4000 rpm	1.7	7.4	sec
shaft torque, 1:2.4 ratio	34.21	7.88	N m
acceleration	71.93	16.56	rad/sec^2
time, 0-4800 rpm	7.0	30.4	sec
time, 4259-4800 rpm	1.7	7.2	sec

Combined Motor and Amplifier Specification Sheet				
Motor Model Number				MPP1426B41
Amplifier Model Number				IdealAmp
Motor Thermal Test Condition				MPP1426 With 12x12x0.5 Heatsink
Description	**Notes**	**Symbol**	**Units**	**Value**
Stall Torque Continous	1,2,3	Tcs	Nm	25.96
Stall Torque Continous	1,2,3	Tcs	in-lb	229.7
Stall Torque Continous	1,2,3	Tcs	oz-in	3675.5
Stall Current Continous	1,2,3	Ics(rms)	Arms	26.2
Stall Current Continous	1,2,3	Ics(trap)	Amps DC	32.1
Peak Torque		Tpk	Nm	82.18
Peak Torque		Tpk	in-lb	727.3
Peak Torque		Tpk	oz-in	11636.0
Peak Current		Ipk(rms)	Arms	82.7
Peak Current		Ipk(trap)	Amps DC	101.3
Rated Speed	1,2,3	Sr	rpm	3352
Rated Torque	1,2,3	Tr	Nm	17.91
Rated Torque	1,2,3	Tr	in-lb	158.5
Rated Torque	1,2,3	Tr	oz-in	2535.6
Rated Shaft Output Power	1,2,3	Pout	kW	6.3
Current at Rated Speed	1,2,3	Ir	Arms	18.5
Voltage Constant	4,9	Kb	V/rad/s	0.8118
Voltage Constant	4,9	Ke	Vrms/krpm	60.12
Torque Constant	4,9	Kt(sine)	Nm/Arms	0.994
Torque Constant	4,9	Kt(trap)	Oz*in/Amp DC	114.96
Resistance	4,9	R	ohm	0.162
Inductance	5,9	L	mH	3.3
Max DC bus Voltage	6	Vmbus	VDCmax	340
Max AC Voltage	6	Vs	VAC	240
Winding-Amb Thermal Resist	6	Rthw-a	°C/W	0.43
Ambient Temp at Rating		Tamb	°C	25
Max Winding Temp		Tmax	°C	155
Winding Temp at Rating	7	Twr	°C	125
Motor Thermal Time Constant	6	tth	minutes	80.0
Rotor Shaft Viscous Damping	6	B	Nm/krpm	0.1130
Rotor Shaft Dynamic Friction	6	Tf	Nm	0.0706
Rotor Inertia	6	J	kg-m2	2.147E-03
Rotor Inertia	6	J	in-lb-sec2	1.900E-02
Number of rotor maget poles	6	Np	#poles	8
Motor Weight	6	weight	kg	20.2
Motor Weight	6	weight	lb	44.4
Motor UL Class		F	UL class	H
Winding Number				W00565
Environmental Protection Rating	8	IP		IP40 - IP65
Amplifier Bus Voltage		Volts DC		340
Amplifier Continous Current		Ica(rms)		141.42
Amplifier Peak Current		Ipa(rms)		424.26
Amplifier Switching Frequency		kHz		8

Timing belt selection

We plan to use a toothed-belt drive for the step-up transmission between the motor and the cutter shaft. A preliminary toothed belt selection was performed using an on-line tool from Gates-Mectrol. The tool selected a FAT5NT belt: AT5 tooth profile, 5 mm pitch, 50 mm width, 500 tooth length, Steel Cord, PU 92 Shore A Resin, Nylon Tooth Facing, Metric Standard, F Type Belt. The tool selected a 256-tooth drive pulley and a 130-tooth pulley for a transmission ratio of 1:1.97, step-up. An excerpt of the selection tool report is provided in the table below.

The model on Page A-8 shows pulleys for an 8 mm pitch, HTD tooth profile with a ratio 1:1.97. The final component selection will be completed once we decide on the main shaft speed (4000-4800 rpm) and motor output torque (18-80 Nm) settings.

	Driver Pulley	**Driven Pulley**
Pitch diameter	0.4074 m	0.2069 m
Number of teeth	256	130
Number of teeth in mesh	138	59
Center distance	0.762 m (30 inch)	-
Belt wrap angle	3.405 radians	2.878 radians
Angular velocity	209.44 rad/sec (2000 rpm)	412.435 rad/sec (3938 rpm)
Angular acceleration	36.651 rad/s^2	72.175 rad/s^2
Torque	31.4 Nm driven, 61.8 Nm driving	
Power	12939 W	
Initial belt tension	213.87 N	
Effective tension	303.26 N	
Shaft force	425.35 N (95.6 pounds)	
Mesh Frequency	8533.3 Hz	

Cutter shaft brake

We plan to connect a mechanical brake to the cutter shaft so that the cutter can be stopped rapidly. We have identified several candidates from Stearns Division of Rexnord Industries, datasheet shown on the following page. They are available in torque ratings of 1.1 to 3.5 times the torque provided by the drive motor through a 1:2 step up transmission. These brakes are self-contained, bearing-supported, through-shaft, spring-set, electrically-released units. For the time to set and release the brake, see the graph and table below for Series 87,000, static torque 35 lb-ft. The brake will be mounted on a pedestal at the pulley end of the main shaft, connected to the main shaft with a shaft coupling that provides for angular and radial misalignment, and operated by the monitored, emergency-stop relay. We plan to identify an acceleration-activated switch to add to the emergency-stop loop that would automatically trigger an emergency-stop should an out-of-balance condition arise. The through-shaft could be used as a mounting location for the main shaft tachometer.

Set and Release Times

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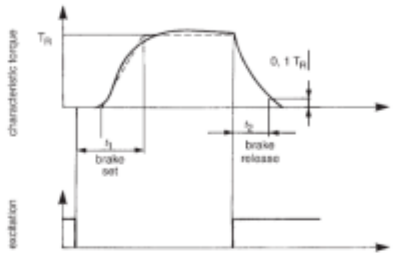
The models listed below were tested for typical set and release times. Times listed below are defined as follows:

T1 = Total set time to 80% of rated static torque

T2 = Release time, measured as the time from when the power is applied to the brake to the time that the solenoid plunger or armature is fully seated.

NOTE: Times will vary with the motor used, and brakes tested with factory-set air gap. The times shown should be used as a guide only.



SAB T1/T2 Time in Milliseconds

Series	Static Torque lb-ft	Coil Size	T1 AC	T2 AC
56,000	1½ – 25	K4, K4, K4+, M4+	25	14
87,000	10,15, 25,50	5 & 6	42	20
87,000	35,75,105	8	48	20
81,000 82,000	All	9	56	27

Brake and motor are switched separately.
All brakes tested in horizontal position.
Coil is energized for >24 hours before testing.
Ambient temperature 70°F at time of test.

Series 87,200 (1-087-2XX) Foot Mounted, Bearing-Supported Thru-Shaft

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Static Torque: 6 through 105 lb-ft.

Enclosure Material: Cast Iron Endplate and Housing

Release Type: Side Lever, maintained with automatic release.

Enclosure Protection: IP 23 & 54 (formerly referred to by Stearns as NEMA 2 & 4, respectively).

Installation and Service Instructions: P/N 8-078-927-00

Parts List: P/N 8-078-917-02

Specifications: Page 17

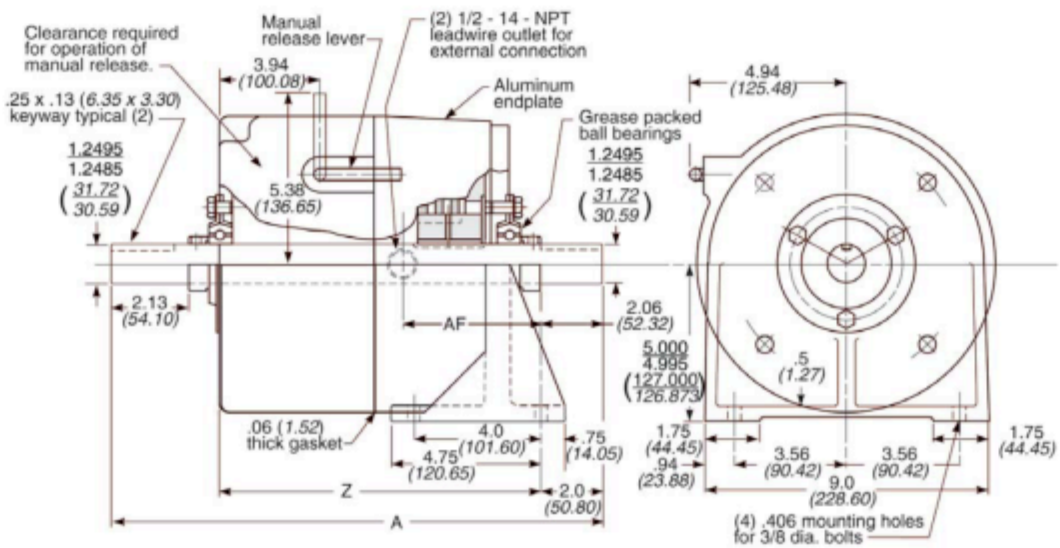
Modifications: Pages 51-60
Self adjust - see SAB Modifications for new manual adjust.

For vertical mounting modification see SAB Modification Section.

Maximum overhung, or side load measured at one inch from end of shaft: 100 lbs on brake housing side, 150 lbs on endplate/foot mount side

Brake set and release times in milliseconds, when brake and motor are switched separately (for T1/T2 definitions, see page 98):

Static Torque	Coil Size	T1	T2
10, 15, 25, 50	5 & 6	42	20
35, 75, 105	8	48	20



*Keyseats made to ANSI B17.1 standard.

Dimensions for estimating only. For installation purposes request certified prints.

Dimensional Data and Engineering Specifications/Unit Pricing (Discount Symbol A2)

Nominal Static Torque lb-ft (Nm)	Enclosure	Basic Model Number and List Price*				Dimensions in Inches (Dimensions in Millimeters)			Thermal Capacity (hp-sec/min)	Inertia Wk² (lb-ft²)	Wt. lbs (kg)**
		AC	AC List Price	DC	DC List Price	A	Z	AF			
10 (14)	IP 23	1-087-211-00	$2,475.00	1-087-215-00	$3,045.00	14.56 (369.82)	9.32 (238.13)	3.56 (90.42)	17.5	.049	72 (33.0)
	IP 54	1-087-212-00	2,675.00	1-087-216-00	3,245.00		9.38 (328.25)				
15 (20)	IP 23	1-087-221-00	2,525.00	1-087-225-00	3,095.00	14.56 (369.82)	9.32 (238.13)	3.56 (90.42)	17.5	.049	72 (33.0)
	IP 54	1-087-222-00	2,725.00	1-087-226-00	3,295.00		9.38 (328.25)				
25 (34)	IP 23	1-087-231-00	2,600.00	1-087-235-00	3,170.00	14.56 (369.82)	9.32 (238.13)	3.56 (90.42)	17.5	.049	73 (33.0)
	IP 54	1-087-232-00	2,800.00	1-087-236-00	3,370.00		9.38 (328.25)				
35 (47)	IP 23	1-087-241-00	2,750.00	1-087-245-00	3,320.00	14.56 (369.82)	9.32 (238.13)	3.56 (90.42)	17.5	.049	73 (33.0)
	IP 54	1-087-242-00	2,950.00	1-087-246-00	3,520.00		9.38 (328.25)				
50 (68)	IP 23	1-087-251-00	3,050.00	1-087-255-00	3,620.00	15.06 (382.50)	9.81 (249.94)	4.06 (103.12)	17.5	.083	78 (35.0)
	IP 54	1-087-252-00	3,250.00	1-087-256-00	3,820.00		9.88 (250.95)				
75 (102)	IP 23	1-087-261-00	3,550.00	1-087-265-00	4,120.00	15.06 (382.50)	9.81 (249.94)	4.06 (103.12)	17.5	.083	78 (35.0)
	IP 54	1-087-262-00	3,750.00	1-087-266-00	4,320.00		9.88 (250.95)				
105 (142)	IP 23	1-087-281-00	4,250.00	1-087-285-00	4,820.00	15.56 (395.20)	10.32 (262.13)	4.56 (115.82)	17.5	.117	81 (37.0)
	IP 54	1-087-282-00	4,450.00	1-087-286-00	5,020.00		10.38 (263.65)				

*See "Ordering Information", previous page.

Design of Carriage

In an operational system, a vehicle will propel the rotating cutter along the road surface at 5 mph. A means, e.g. wear blocks in contact with the road surface, will be provided to set the cutter height above the road surface. The vehicle weight, mass, and drive traction will be used to oppose the cutting force. The experimental system for cutter development, presented here, will feature a stationary cutter axis. To simulate the cutting operation, the marker will be moved toward the cutter axis at 5 mph.

Requirement	Design decisions
1. The carriage must support the resultant cutting force.	Carriage to be supported by 4 or 6 wheels sized for expected cutting force.
2. The carriage must support one or more markers at various target heights.	Provide adapter blocks, sacrificial blocks, and shims to support marker on carriage platform with or without load cell with plastic, aluminum, steel, asphalt, or concrete substrates.
3. The carriage must be instrumented to measure cutting forces with an appropriate frequency response.	Provide a properly-sized load cell to measure cutting forces.
4. The carriage must be instrumented to measure linear velocity and either a single position or continuous position.	Provide a tachometer on the carriage drive shaft or other means to measure carriage velocity. Provide a position switch for recording the time that the carriage passes a known spot OR provide a transducer for continuous position feedback.
5. Structure must be provided to carry the cutting load from the carriage to the cutter shaft.	Load path includes adapter plate, load cell, carriage structure, carriage bearings, bearing ways, base plate, main-bearing pedestals, main bearings, cutter shaft, and cutter assembly.
6. The carriage must be accelerated from stopped to a velocity of 5 mph prior to marker contact with the cutter and then decelerated to a stop after the cut is complete.	Use salvaged brushless DC servo motor, salvaged right angle gearbox, drive shaft, pinion gears, and gear rack to drive carriage. Approximate the velocity profile shown on Page A-32.

Selection of Load Cell for Cutting Force Measurement

Load cells from several manufacturers were considered including Kistler, Futek, and ATI Industrial Automation. The cutting force measurements made during the IRAD project were made at the Precision Machining Center of the GIT Manufacturing Research Center using a Kistler force dynamometer. The load cell capacity requirement was based on the extrapolated cutting forces, Page A-12. The data output requirement was based on a plan to use an existing high-speed video system with an 8-channel, analog input capability for data collection. We selected a Futek Model MTA505, Low Profile Thrust and Moment Load Cell with capacities: Channel Fz: 25000 lb; Channel Mx: 10000 in-lb; Channel My: 10000 in-lb. It provides analog outputs for each channel through the use of strain gage amplifiers. The solution from Kistler was considerably more expensive, in part, because of the cost of the required signal conditioning unit.

The sole-brand justification provided with the purchase request and the vendor quotation are provided below. The datasheet and photograph of the load cell follows.

Sole-Brand Justification from Purchase Request*: We have based the system design for our deliverable system on this component. This thrust and moment load cell has sufficient load capacity for the cutting force that we need to measure. The load cell in conjunction with the strain gage amplifiers included on this order can provide an analog output to an existing data logger that we plan to use to collect data from the experiments. The system is moderately priced. We compared this product to products available from Kistler. The Futek quotation is attached. Also, notes from a conversation with a Kistler representative and Kistler data sheets are attached.*

Line	Description	Quantity	Unit Price	Disc	Extension
1	ITEM #: FSH02727 Model: MTA505 , Ch Fz: 25000 lb; Ch Mx: 10000 in-lb; Ch My: 10000 in-lb , Low Profile Thrust and Moment Load Cell , Material - 17-4 PH S.S. , 1 1/4-12-Thread , 6 Pin Bendix Receptacle, PT02A-10-6P Delivery Time : In Stock	1	$4,800.00		$4,800.00
2	ITEM #: FSH01757 Model: ZCC950 , 15 ft Long , 6 Pin Bendix PT06A-10-6S-SR to Cable Assembly , CC1 Code, Molded Strain Relief , Material - Polyurethane , 28 Awg 6 Conductors , Braided Shielded Delivery Time : In Stock	3	$90.00		$270.00
3	ITEM #: FSH01449 Model: CSG110 , Strain Gage Amplifier , Standard , Enclosed With Din Rail Mount , With DB9 Connectors , ABS-94HB Black Enclosure , Analog Output , +/-5 VDC , +/-10 VDC , 0-20 mA , 4-20 mA , 0-16 mA , 5-25 mA , 1 KHZ Bandwidth Delivery Time : In Stock	3	$350.00		$1,050.00
4	ITEM #: SLB00023 Model: NIST Traceable System Calibration w/ Amplifier & Certificate, Tension & Compression, Voltage Output, 5 points Delivery Time : 1 to 2 Days A.R.O.	1	$300.00		$300.00
5	ITEM #: STB00025 Model: NIST Traceable Torque System Calibration w/ Amplifier & Certificate, Clockwise & Counter Clockwise, Voltage Output, 5 points Delivery Time : 1 to 2 Days A.R.O.	2	$600.00		$1,200.00

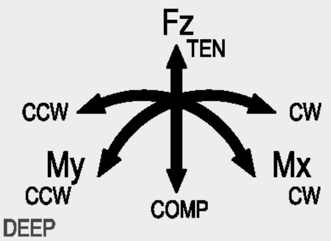
FUTEK MODEL MTA505

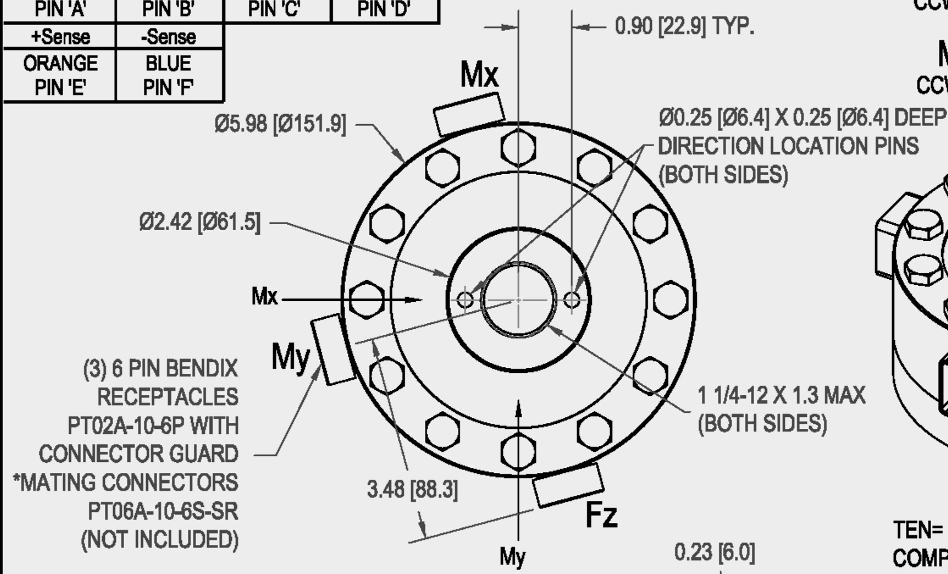
LOW PROFILE THRUST AND MOMENT LOAD CELL

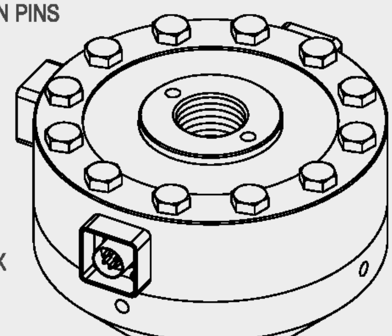
Drawing Number: FI1288-B

INCH [mm] | R.O.= Rated Output

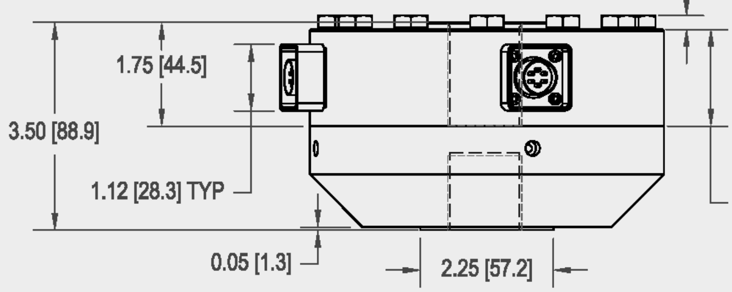
WIRING CODE (CC1)			
+Excitation	-Excitation	+Signal	-Signal
RED PIN 'A'	BLACK PIN 'B'	GREEN PIN 'C'	WHITE PIN 'D'
+Sense	-Sense		
ORANGE PIN 'E'	BLUE PIN 'F'		







TEN= TENSION (+OUTPUT)
COMP= COMPRESSION (-OUTPUT)
CW= CLOCKWISE (+OUTPUT)
CCW= COUNTER CLOCKWISE (-OUTPUT)



SPECIFICATIONS:

RATED OUTPUT	SEE CHART
CAPACITY	SEE CHART
SAFE OVERLOAD	150% of R.O.
ZERO BALANCE	±2% of R.O.
EXCITATION (VDC OR VAC)	18 MAX
BRIDGE RESISTANCE	350 Ω nom.
NONLINEARITY (Mx, My)	±0.5% of R.O.
NONLINEARITY (Fz)	±0.2% of R.O.
CROSS TALK	±2% of R.O. OR LESS
TEMP. SHIFT ZERO	±0.01% of R.O./°F (0.018% of R.O./°C)
TEMP. SHIFT SPAN	±0.01% of LOAD/°F (0.018% of LOAD/°C)
COMPENSATED TEMP.	60 to 160°F (15 to 72°C)
OPERATING TEMP.	-65 to 200°F (-50 to 93°C)
MATERIAL	17- 4 ph S.S.
WEIGHT	20 lb [9.3 kg]

CONNECTOR: (3) 6 Pin BENDIX Receptacle (PT02A-10-6P)

ACCESSORIES AND RELATED INSTRUMENTS AVAILABLE

CALIBRATION (STD)	5pt. COMPRESSION, CW ; SEE CHART FOR SHUNT CAL. VALUE
CALIBRATION (OPTIONAL)	5 pt. TENSION, CCW
CALIBRATION TEST EXCITATION	10 VDC

	CAPACITY		RATED OUTPUT		SHUNT	
	lb	in-lb	mV/V		KΩ	
ITEM #	(Fz)	(Mx, My)	(Fz)	(Mx, My)	(Fz)	(Mx, My)
FSH02453	10,000	2000	2.5	0.5	43.575	301
FSH02727	25,000	10,000	4	2	30	60.4

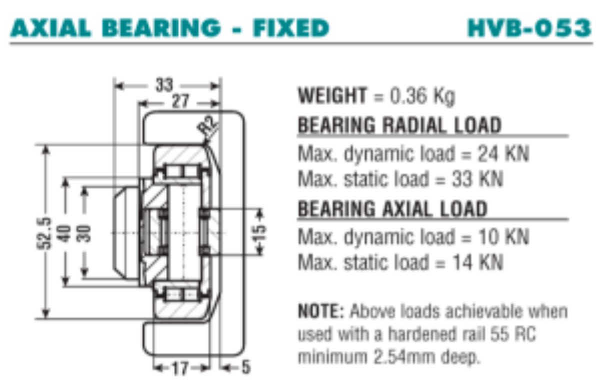
FUTEK
ADVANCED SENSOR TECHNOLOGY, INC.

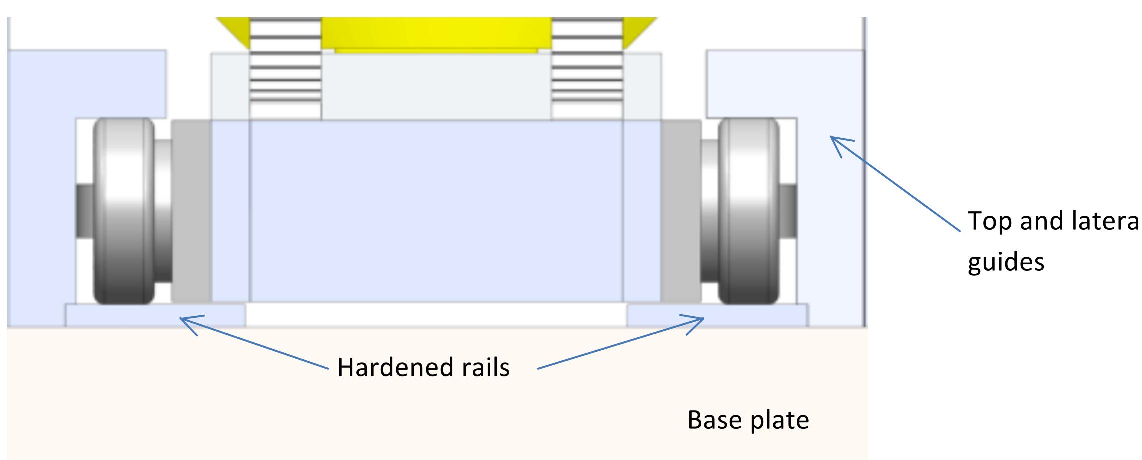
10 THOMAS
IRVINE, CA 92618 USA
1-800-23-FUTEK (38835)

INTERNET:
http://www.futek.com

Selection of Carriage Bearings

The carriage is supported by four hardened wheels commonly referred to as track rollers. The wheels are integral units such that the wheel incorporates the outer race of the supporting ball or roller bearing. The selected carriage bearings incorporate a second wheel with axis perpendicular to the main wheel to guide the carriage laterally. The carriage bearing capacity requirement was based on the extrapolated cutting forces, Page A-12. The cutting force must be carried by a pair of bearings. The selected carriage bearings are Pacific Bearings Heavi-Rail HVB-053 with a maximum radial load capacity of 24000 N when used on a rail hardened to 55 Rockwell C. The section view below shows the wheel and roller, their supporting bearings and a U-channel rail. The guides for the wheels will be built up out of two or more components and will incorporate hardened rails on top of the system base plate to achieve the maximum rated load capacity of 24000 N. An end view of the carriage with carriage bearings and guide rail is shown below.

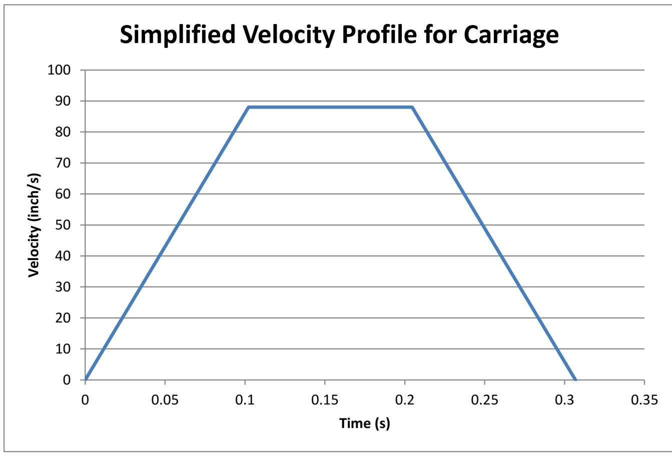


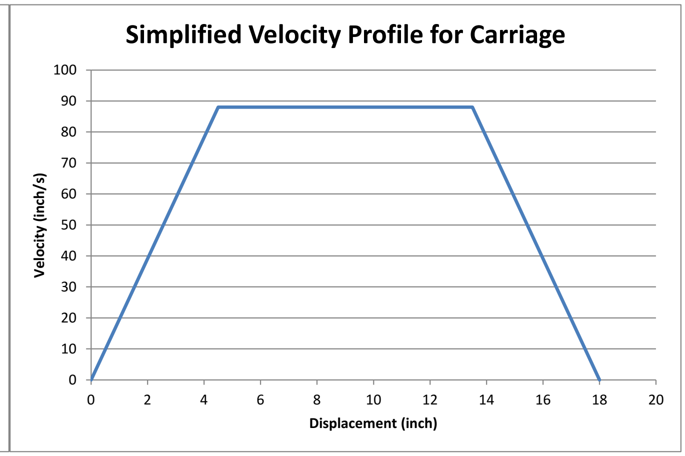


Design of Carriage Drive

The carriage drive must accelerate the carriage from 0 to 5 mph prior to contact between the marker and the cutter and then stop the carriage after completion of the cut. A simplified velocity profile is shown below, both as a function of time and of displacement. This velocity profile is not physically achievable; in an actual velocity profile, the sharp corners will be replaced by curves.

Note: This target velocity profile provides for 9 inches of travel at the target velocity of 5 mph (88 ft/sec) with a total travel of 18 inches. It could be used for an attempt to cut two markers in a row, but no more than 3 markers in a row.





The smaller servo motor and right-angle gearbox of the salvaged drive system, mentioned on Page A-24, will be used for the carriage drive. The Parker motor, MPP1002D1E-NPSN, performance characteristics are provided on Page A-34. The Alpha Gear Drives 16:1 gearbox, SK075S-MF2-16-OE1, performance characteristics are provided on Page A-33. The gearbox output shaft will connect to the pinion shaft. The pair of pinion gears will drive a pair of gear racks on the carriage.

The table below gives the pinion pitch diameter required to drive the carriage at 5280 inch/per minute for various motor speeds. For each combination of motor speed and pitch diameter, the driving force and mass that could be accelerated at 2.2 g is given for both the continuous and peak torque rating of the motor. Friction is not accounted for in these calculations. The carriage shown on Page A-8 weighs 60 lbm.

Design Decision: Select pinion gear pitch diameter between 5.375 and 7.625 inch for motor input speed between 3500 and 5000 rpm. Action Item 32 has been added to estimate the expected friction from the wheels and rack and pinion prior to selecting the pinion gear pitch diameter.

Note: From the specifications provided on Page A-33, gearbox nominal input speed is 3500 rpm and the maximum input speed is 6000 rpm; maximum continuous input speed is not provided for 2-stage gearboxes.

				Torque in foot pounds			
				Continuous	***Peak***		
				47.2	165.2	***Continuous***	***Peak***
Motor speed	Output speed	Pitch Diameter for 5280 in/min	Pitch radius	**Available Force**		**Mass at 2.2 g acceleration**	
rpm	rpm	inch	feet	lbf	lbf	lbm	lbm
1000	62.5	26.891	1.1205	42.1	147.5	19.1	67.0
3000	187.5	8.964	0.3735	126.4	442.4	57.4	201.1
3500	218.75	7.683	0.3201	147.5	516.1	67.0	234.6
4000	250	6.723	0.2801	168.5	589.8	76.6	268.1
5000	312.5	5.378	0.2241	210.6	737.3	95.7	335.1

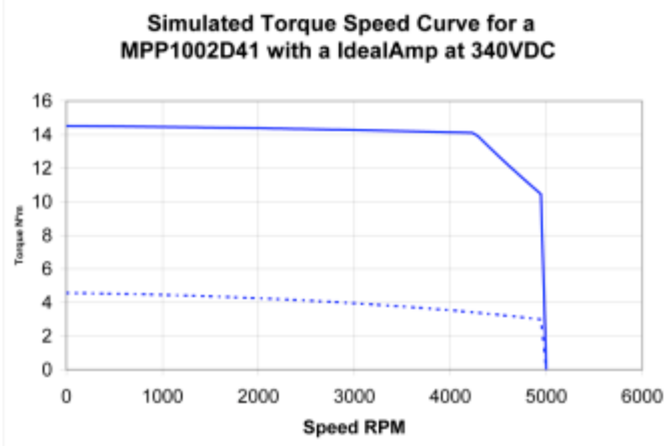
Performance characteristics for Alpha Gear, 16:1, 2-stage gearbox, SK075S-MF2-16-OE1.

Technical Data **SK+ 075**

			1-stage					2-stage								
Ratio	i		**3**	**4**	**5**	**7**	**10**	**16**	**20**	**25**	**28**	**35**	**40**	**50**	**70**	**100**
Max. acceleration torque (max. 1000 cycles per hour)	T_{2B}	in.lb (Nm)	620 (70)	620 (70)	620 (70)	531 (60)	443 (50)	620 (70)	620 (70)	620 (70)	620 (70)	620 (70)	620 (70)	620 (70)	531 (60)	443 (50)
Nominal output torque	T_{2N}	in.lb (Nm)	443 (50)	443 (50)	443 (50)	398 (45)	354 (40)	443 (50)	443 (50)	443 (50)	443 (50)	443 (50)	443 (50)	443 (50)	398 (45)	354 (40)
Emergency stop torque (Permissible 1000 times during the lifespan of the gear reducer)	T_{2Not}	in.lb (Nm)	841 (95)	841 (95)	841 (95)	664 (75)	575 (65)	841 (95)	841 (95)	841 (95)	841 (95)	841 (95)	841 (95)	841 (95)	664 (75)	575 (65)
Nominal input speed at T_{2N} * (At 20 °C ambient temperature) **	n_{1N}	rpm	2500	2500	2500	2500	2500	3500	3500	3500	3500	3500	3500	3500	4500	4500
Max. continuous speed (At 20 °C ambient temperature) **	$n_{1N\,cym}$	rpm	3500	3500	3500	4000	4000	For higher mean speed, contact alpha								
No-load running torque (n_1=3000 rpm) (At 20 °C gear reducer temperature)	T_{012}	in.lb (Nm)	17.7 (2.0)	15.0 (1.7)	13.3 (1.5)	17.7 (2.0)	15.9 (1.8)	2.7 (0.3)	-	-	-	-	-	-	-	-
Max. input speed	n_{1Max}	rpm	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000	6000
Torsional backlash	j_t	arcmin	≤ 4													
Torsional stiffness	C_{t21}	in.lb (Nm)/arcmin	40 (4.5)	-	55 (6.2)	-	49 (5.5)	Please contact alpha								
Max. axial force ***	F_{2AMax}	lb_f (N)	719 (3200)													
Max. radial force ***	F_{2RMax}	lb_f (N)	854 (3800)													
Max. tilting moment	M_{2KMax}	in.lb (Nm)	3868 (437)													
Efficiency at full load	η	%	96					94								
Service life (see alpha's "Technical Basics" catalogue for calculation)	L_h	h	> 20 000													
Weight (incl. ADP)	m	lb_m (kg)	10.6 (4.8)					12.1 (5.5)								
Noise level (n_1=3000 min^{-1}) ****	L_{PA}	dB(A)	≤ 66													
Max. permissible housing temperature		F (°C)	+194 (+90)													
Ambient temperature		F (°C)	32 (0) to 104 (+40)													
Lubrication			Synthetic gear oil													
Paint			Blue RAL 5002													
Direction of rotation			Input and output sides in opposite directions													
Type of protection			IP 65													
Mass moment of inertia (refering to the drive)	J_1	in.lb.s² (kgcm²)	0.0013 (1.46)	0.0010 (1.19)	0.0009 (1.06)	0.0008 (0.95)	0.0008 (0.90)	0.0002 (0.27)	0.0002 (0.23)	0.0002 (0.23)	0.0002 (0.20)	0.0002 (0.20)	0.0001 (0.18)	0.0001 (0.18)	0.0001 (0.18)	0.0001 (0.18)

* Higher mean speeds are possible at reduced nominal torque.
** Please reduce the n_{1N} speed at higher ambient temperatures.
*** In reference to the center of the output shaft.
**** Measured at ratio i = 5 (without load).

Performance characteristics for Parker MPP1002D1E-NPSN



Combined Motor and Amplifier Specification Sheet				
Motor Model Number				MPP1002D41
Amplifier Model Number				IdealAmp
Motor Thermal Test Condition				MPP1002 With 12x12x0.5 Heatsink
Description	**Notes**	**Symbol**	**Units**	**Value**
Stall Torque Continous	1,2,3	Tcs	Nm	4.57
Stall Torque Continous	1,2,3	Tcs	in-lb	40.5
Stall Torque Continous	1,2,3	Tcs	oz-in	647.3
Stall Current Continous	1,2,3	Ics(rms)	Arms	7.9
Stall Current Continous	1,2,3	Ics(trap)	Amps DC	9.6
Peak Torque		Tpk	Nm	14.51
Peak Torque		Tpk	in-lb	128.5
Peak Torque		Tpk	oz-in	2055.3
Peak Current		Ipk(rms)	Arms	24.9
Peak Current		Ipk(trap)	Amps DC	30.5
Rated Speed	1,2,3	Sr	rpm	4780
Rated Torque	1,2,3	Tr	Nm	3.10
Rated Torque	1,2,3	Tr	in-lb	27.4
Rated Torque	1,2,3	Tr	oz-in	438.8
Rated Shaft Output Power	1,2,3	Pout	kW	1.6
Current at Rated Speed	1,2,3	Ir	Arms	5.6
Voltage Constant	4,9	Kb	V/rad/s	0.4776
Voltage Constant	4,9	Ke	Vrms/krpm	35.36
Torque Constant	4,9	Kt(sine)	Nm/Arms	0.585
Torque Constant	4,9	Kt(trap)	Oz*in/Amp DC	67.62
Resistance	4,9	R	ohm	0.960
Inductance	5,9	L	mH	4.3
Max DC bus Voltage	6	Vmbus	VDCmax	340
Max AC Voltage	6	Vs	VAC	240
Winding-Amb Thermal Resist	6	Rthw-a	°C/W	0.80
Ambient Temp at Rating		Tamb	°C	25
Max Winding Temp		Tmax	°C	155
Winding Temp at Rating	7	Twr	°C	125
Motor Thermal Time Constant	6	t_{th}	minutes	40.0
Rotor Shaft Viscous Damping	6	B	Nm/krpm	0.0282
Rotor Shaft Dynamic Friction	6	Tf	Nm	0.0318
Rotor Inertia	6	J	kg-m2	2.599E-04
Rotor Inertia	6	J	in-lb-sec2	2.300E-03
Number of rotor maget poles	6	Np	#poles	8
Motor Weight	6	weight	kg	4.3
Motor Weight	6	weight	lb	9.5
Motor UL Class		F	UL class	H
Winding Number				W00553
Environmental Protection Rating	8	IP		IP40 - IP65
Amplifier Bus Voltage		Volts DC		340
Amplifier Continous Current		Ica(rms)		141.42
Amplifier Peak Current		Ipa(rms)		424.26
Amplifier Switching Frequency		kHz		8

Action Items

Action Item List

Action Item Description	Open/ Close Date	Priority	Comments
1. Stress analysis of cutter disk for combined loads – centrifugal, hub pressure, and cutting force.	6/30/11		Complete Before release of drawing for first cutter disk
2. Correct shape of cutter tooth. Replace circular arc with spiral.	Open		Complete Before release of drawing for first cutter disk
3. Modify design to add means of alignment and connection for stack of cutter disks.	Open		Complete Before release of drawing for first cutter disk
4. Calculate the maximum volume of a chip. Make the flute volume ≥ the maximum chip volume. Investigate the use of a chip breaker in the flute design.			The maximum volume of a chip = 0.01579 $inch^3$. The current flute volume = 0.004 $inch^3$. Complete Before release of drawing for first cutter disk
5. Evaluate the feasibility of eliminating the 3rd bearing on the main shaft. Add belt tension load to shaft analysis. Calculate moment in two planes per Spotts pp. 152-153.	Open		Complete Before release of purchase order for bearings
6. Test single cutter disk at full rotational speed and full feed speed.	Open		
7. Contact Fenner about price and lead time for special keyless bushing for cutter stack. For example, d = 50 mm bore, L = 7 inches, D ≥ 80 mm.	Open		Complete Before release of drawing for full-scale cutter
8. Add internal O-ring groove to 50X60 mm sleeves. Axial location between B-LOC and shaft collar on bearing with locating rings.	Open		
9. Add centers to main shaft.. No end treatments are available from Misumi for AISI 4137 steel shafts.	Open		Complete Before release of shaft purchase order
10. Complete field measurements of in-service heights of markers with adhesive.	Open		
11. Water jets could be used to cool blade and prevent melting of the marker.	7/5/11		Allow for the use of water jets cooling in experimental setup. Provide water supply, drain, spray shields. May not be possible with load cell installed.
12. Pavement temperatures can be much higher than ambient air temperatures.	7/5/11		Provide heated platform to support heater. Raise marker temperature prior to cutting. May not be possible with load cell installed.
13. Machine Design textbook (Spotts, pg 523) just mentions that, for a spinning disk, "stress concentration occurs at the bolt holes".	Open		Add bolt circle to model and rerun.
14. Can you put an error bound on the cutting force extrapolation?			
15. For the extrapolation of forces, what is the effect of going from 1-			

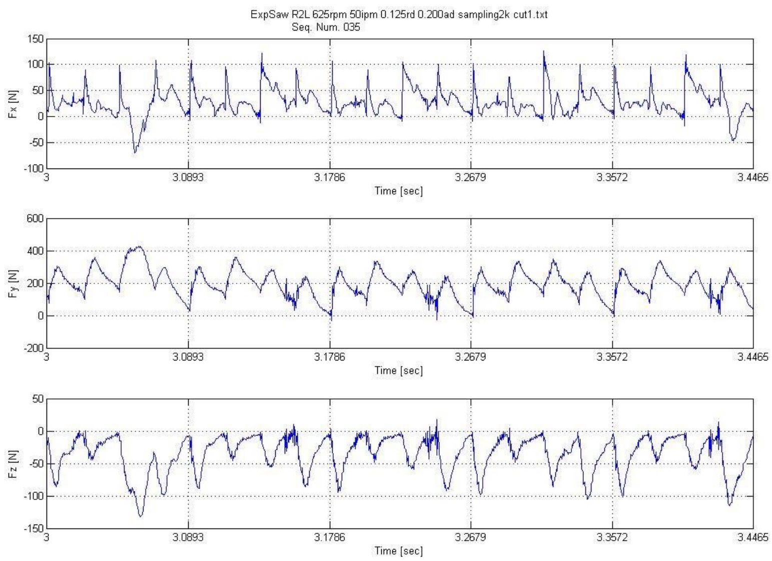
tooth engagement to 7-tooth engagement?			
16. LS Dyna could be used to model cutting forces.	7/5/11		No. Depend on single disk cutting trials to confirm force extrapolation.
17. Assign task to Burt to duplicate stress analysis of spinning disk with internal pressure and cutting forces for verification.	Open		
18. Do trade analysis on cutter diameter versus rotational speed. Can consider stresses, kinetic energy, number of teeth, etc.	Open		
19. Could use retroreflective marker on cutter to measure rotational speed with high-speed video.	7/5/11		Add inductive proximity switch for tachometer sensor for high-speed-video data logger. Sense feature on timing belt pulley.
20. What about bolt design for main bearings? Does the pillow block have a shear pin feature?	8/9/11		Assign bolt analysis to Chris Haile. Do we need a 6mm dia. dowel? Is one enough?
21. Calculate the shaft critical speed.	7/20/11		
22. What happens when feed is faster than 5 mph?			Does this result in a requirement and specification for speed control of the operational system vehicle during removal operations?
23. Test Futek load cell with horizontal static load.	Open		
24. Add window in main bearing pedestal for high-speed camera periscope.	Open		
25. What is the effect of an out-of-balance cutter?			
26. What is the impact of cutter assembly stiffness on shaft deflection and stress? How should the through bolts for the cutter assembly be sized?	Open		
27. Consider doing a thermal analysis on the cutter drive motor. How long would the motor need to cool after running a single test at peak torque?	Open		
28. Identify an acceleration-activated switch ("knock" sensor) that could be used to initiate an emergency stop for an out-of-balance condition.	Open		
29. Add shock absorbers to each end of the carriage to absorb impact energy.	Open		
30. Generate a detailed material estimate and send authorization request to David Jared. (Futek, brake unit, etc.)	Open		
31. Review design with Dennis Denney. Designate fabrication method for each fabricated part.	Open		

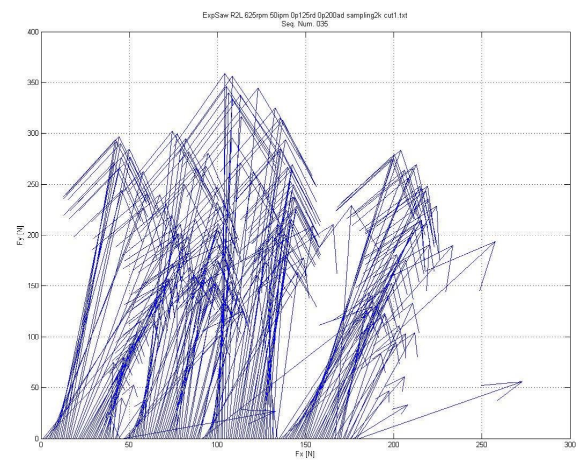
32. Estimate the expected friction from the wheels and rack and pinion prior to selecting the pinion gear pitch diameter for the carriage drive.	Open		
33. Consider service loop or “energy chain” for force-torque (load cell) sensor cables.			
34. Obtain force-torque (load cell) sensor stiffness parameters to estimate deformation under load.			
35. Make containment structure large enough for a 14-inch diameter cutter.			
36. Get contact information from Burt for balancing the cutter assembly. Does the main shaft need centers?			http://atlantaspeedshop.com/, Tony Jape’s cell is 770-778-8373
37. For *Action Item 15*, recalculate equivalent force to cut single chip of marker volume based on an average chip thickness of 0.005”. Include specific cutting energy adjustment for undeformed chip thickness.			0.005-inch thickness comes from volumetric thickness calculation on model – chip.sldprt.
38. Unless we can obtain a 7-inch-long B-LOC from Fenner, use both bolts and dowel pins to join the cutter assembly. Consider disassembly.			Not pertinent for the initial, single-disk cutter.
39. Verify braking resistors on primary servo amplifier are adequate to stop cutter in approximately 5 seconds.			
40. Find maximum angular velocity rating of candidate brake unit. Compare thermal rating to the maximum kinetic energy of the cutter. Is a brake-status switch available?			
41. Establish reasonable clearance between the carriage track rollers and the upper and lateral guides.			
42. Add third pair of track rollers to carriage. Verify that extrapolated cutting force resultant fall between the pair of track rollers at the marker end of the carriage.			
43. Add way covers for the carriage gear rack, e.g. telescoping way covers.			Review concept for driven way covers considered for protection of hydrodynamic bearing.
44. Discuss tolerances of base plate when talking about part fabrication with Dennis Denney.			
45. Include a single dowel to carry shear load for all pedestals.			
46. Allow for the use of shims to set the clearance between the pair of pinion gears and the gear rack. Couple pinion gears to shaft with Tran-Torque hubs to facilitate clocking the pair of gears.			
47. Add feature such that main-drive timing belt can be tensioned by driving a screw.			

Action Item Resolution

Action Item 14. *Can you put an error bound on the cutting force extrapolation?*

The cutting force varies with time. The force dynamometer resolved the forces vertically and horizontally as F_x, F_y. For a physical interpretation of the forces, it would be better to resolve the forces tangentially and radially for cutting force and rubbing normal force. (For tangential force, cutting force and friction force will be confounded.) The F_x, F_y, and F_z graphs below show the forces on the 4-tooth experimental cutter as a function of time for five revolutions. The "feather" graph at the right shows the resultant vectors for a single revolution.





For Run 035 on the experimental saw, depicted in the graphs above, data was collected for 11.5 seconds. The cutting operation occurred between seconds 2.75 and 8.5. The table below shows the average, maximum, and minimum for forces F_x, F_y, and F_z between seconds 3 and 4. The force extrapolation shown on Page A-12 is based on F_x= 22 lbf and F_y = 79 lbf. The maximum forces are significantly higher than the average forces.

	Time	Time		
	>3	<4		
	Fx	Fy	Fz	pounds
Average	3.155824	41.37628	-6.73385	
Maximum	30.50489	103.1153	6.690089	
Minumum	-12.2238	-23.2874	-34.4036	

Action Item 15. *For the extrapolation of forces, what is the effect of going from 1-tooth engagement to 7-tooth engagement?*

Another way to ask this question is "should the force be seven times as high since there are seven times as many teeth engaged?" Let's compare the amount of energy expended by the extrapolated loads to the cutting energy required.[25]

The table on the left calculates the amount of energy that would be expended if a tangential force of 5000 lbs, corresponding to the extrapolated resultant of 10,800 pounds, was sustained for 40 ms. The energy expended is 3.6 times as much as the energy required to mill away the marker based on the measured specific cutting energy.

The table on the right calculates the cutting force that would generate the calculated total cutting energy in cutting the marker volume into a single chip 0.02 X 4.54 X 63.88 inches. A force of 1770 pounds applied over a distance of 63.88 inches is equivalent to 12779 J. The ratio of 5000:1770 is 2.8:1.

Each of these calculations is based on a different set of experiments, the first from the cutting force experiments and the second from the cutting energy experiments. Additional study would be required to determine if there should be better agreement between the two resulting ratios.

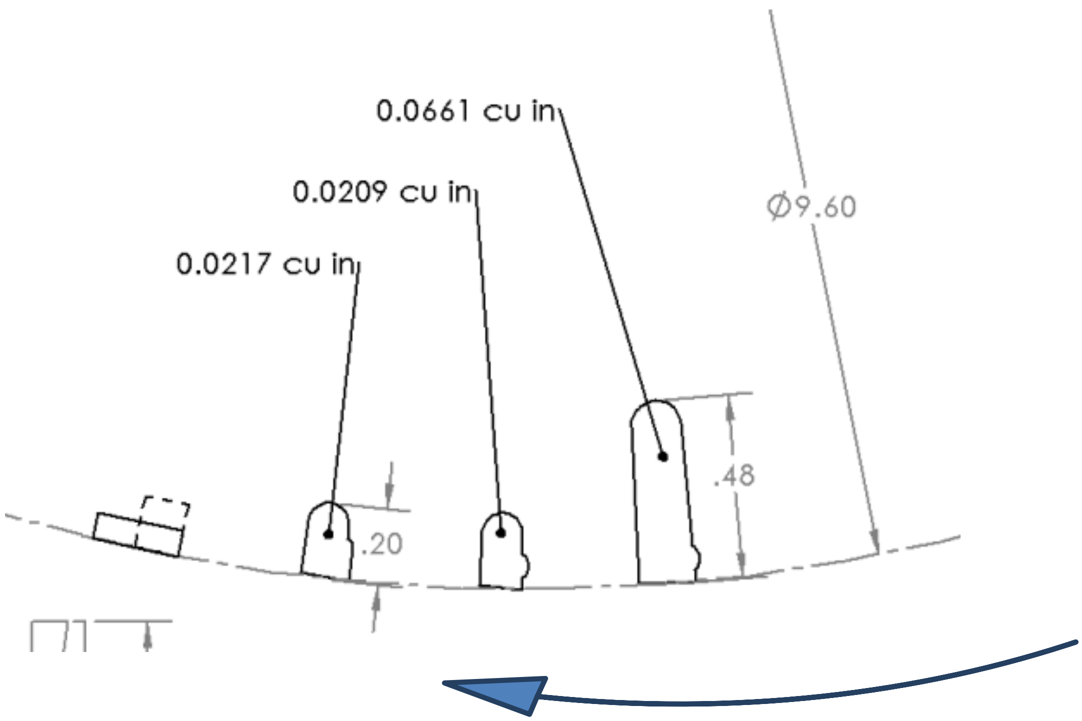
Both methods suggest that the extrapolated peak forces are well in excess of the average force needed to mill the marker based on the measured specific cutting energy. This does not prove that the peak force generated by a 7-tooth engagement is no higher than the peak force generated by a 1-tooth engagement.

5000 pounds force = 22 241.1081 newtons				
22,250	N	force		
0.12192	m	radius		
2,713	Nm	torque		
419	rad/sec			
1,136,630	watts			
0.04	sec			
45,465	J			
12779	J	5.8 in^3 at .049 HP min/in^3		
3.5578048	ratio of energy			

0.02	inch	chip thickness		
4.54	inch	axial depth of cut		
63.88	inch	equivalent chip length		
5.8	$inch^3$	marker volume		
1.62	m	equivalent chip length		
12779	J	5.8 in^3 at .049 HP min/in^3		
7876	N	equivalent force required		
7 876.277491 newtons = 1 770.65762 pounds force				
2.82	ratio of force			

Action Item 4. *Calculate the maximum volume of a chip. Make the flute volume ≥ the maximum chip volume. Investigate the use of a chip breaker in the flute design.*

The maximum volume of a 1-inch-wide chip = 0.01579 inch^3. For cutter 110216.sldprt, the flute volume = 0.004 inch^3. Machinery's Handbook, 25th ed., pp. 726-727, describes three types of chip breakers – angular shoulder, parallel shoulder, and groove type chip breakers. The drawing below shows three candidate flutes, one with a shoulder chip breaker and two with groove chip breakers. The flute volume for a one-axial-inch cutter is shown. Two flutes are greater than the volume of one chip. One flute is greater than the volume of three chips. All three flutes are open at a 3° angle. The blue arrow shows direction of rotation of the cutter.



Action Item 22. *What happens when the feed is faster than 5 mph?*

At higher than design feed rates, the forces will go up. One possible consequence might be that the force breaks the adhesive bond such that the cutter pushes the marker along the pavement as it cuts it. If this occurs, the programmed raising of the cutter from the pavement surface might occur before the marker has been machined completely.

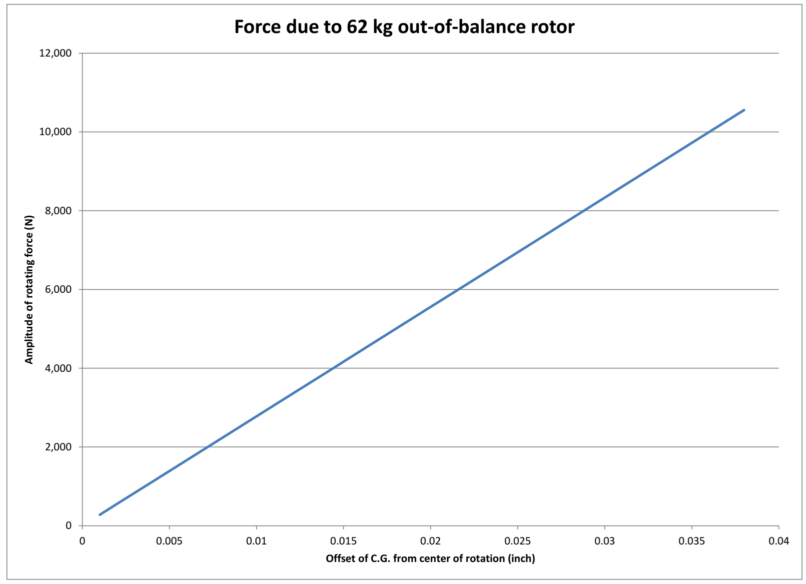
It may be desirable to include an automated speed control requirement for the operational system vehicle. Automated controls for lowering and raising the cutter might require marker-sensing sensors.

On July 6, 2006, experimental were made cuts with 0.220-inch-wide experimental saw at feed rated higher than the design rate of 0.02 inch per tooth. (The feed rates were all 50 inch per minute with the spindle speed adjusted for the nominal inch per tooth feed rate.)

Feed (inch per tooth)	Approximate maximum force in x direction (N)	Approximate maximum force in y direction (N)
0.020	100	400
0.030	100	1000
0.040	100	1300
0.050	100	1300

Action Item 25. *What is the effect of an out-of-balance cutter?*

Condition	C.G.-to-rotation-center offset
Cutter disk stack	0.0 inch
Cutter disk stack with keyless bushing	0.000017 inch
Cutter disk stack with keyless bushing with 0.1 $inch^3$ tooth missing from rim of 1 of 7 disks.	0.001 inch (=> force = 278 N)



Action Item 32. Estimate the expected friction from the wheels and rack and pinion prior to selecting the pinion gear pitch diameter for the carriage drive.

Conversation with Dave at Pacific Bearing on 2/22/2011 re: Hevi-Rail HVB-053 bearing – coefficient of friction 0.005 rolling, 0.01 breakaway.

In order to arrive at an estimate, which provides a good first approximation with minimum calculation, we follow the recommendation of Buckingham (Spur Gears, McGraw-Hill, 1928), which is still a good one even today. This, in effect, states that for average operating conditions, the power loss at each mesh can be approximated as 1 % of the potential power transmitted through the mesh. Figures quoted in the literature vary from less than 1/2% to 2% and the reader can always adjust the percentage if desired.[26]

carriage weight	60	lbf
acceleration	2.2	g
pinion force, tangential	132	lbf
pressure angle, 20°	0.349066	radians
pinion force, radial	48.04407	lbf
coefficient of friction, wheel	0.005	
friction force, wheel	0.54022	lbf
friction force, pinion, 1%	1.32	lbf
friction force, pinion, 2%	2.64	lbf
friction force, wheel, cutting	50.54022	lbf

Action Item 20. What about bolt design for main bearings? Does the pillow block have a shear pin feature?

S:\Projects\GDOT RPM Removal\Designs\DesignReview110630\Bolt Design, Main Bearing, 110728.docx

Bolt Design for Main Bearings for Raised Pavement Marker Removal Cutter

S:\Projects\GDOT RPM Removal\Designs\DesignReview110630\Bolt Design, Main Bearing, 110728.docx

The cutter shaft will be supported by 2 each, SKF, self-aligning ball bearings in SKF housings. The bearing housings will each be supported by a steel pedestal with tapped holes top and bottom. The pedestals will be bolted to a steel base plate at least 1.5 inches thick.
The bearing selection is discussed on pages 18-19 of Design Review Handout Revision -, 6/30/2011.
The data sheet for the bearings and housings are in folder: S:\Projects\GDOT RPM Removal\Vendor Literature\SKF

Objective:
For both bearing-housing-to-pedestal and pedestal-to-base connections, recommend a bolt configuration including: bolt size, number of bolts, bolt material, and torque specification. For each connection, also size a single dowel pin to carry the shear load.

Tasks:

1. Size and analyze the pair of bolts that will hold each bearing housing to its pedestal.
2. The bearing housing will accommodate a 6mm diameter dowel pin. Will a 6mm diameter dowel pin carry the entire shear load experienced by the bearing housing?
3. Size and analyze the 2 or more bolts that will hold each pedestal to the base plate.
4. Will the bolt design change if the plate thickness is increased?
5. Size a single dowel pin that can carry the entire shear load at the surface of the base plate.

Loads:

Use the loads provided by Figure 4 on page 10 of Design Review Handout Revision -, 6/30/2011.
Vertical force applied to the cutter of 10400 pounds, horizontal force of 2900 pounds.

SW Model Filenames:

Bearing assembly:
S:\Projects\GDOT RPM Removal\Designs\Vendor STP files\SKF\SNL_210.sldasm

Pedestal:
S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\mount, main bearing.SLDPRT Configuration: *Default*

Main Assembly:
S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\Concept4b.SLDASM

I am now working in a new folder preparing the model for drawing release. Do not modify the files in this folder.
Main Assembly:
S:\Projects\GDOT RPM Removal\Designs\DesignForRelease110728\CutterDevelopmentSystem.SLDASM

S:\Projects\GDOT RPM Removal\Designs\DesignReview110630\GDOT RPM Removal Bolt Design.docx
GDOT RPM Removal – Bolt Calculation
Christopher Haile
8/8/2011

The RPM Removal test stand has 2 SKF bearing blocks (SKF 210) supporting the cutting shaft. When in operation a vertical load of 10434 lbf and a horizontal load of 2906 lbf are applied to the shaft. These forces are distributed over 4 bolts (2 per bearing housing) and the interface regions between the two bearing blocks and two pedestals. The slots in the bearing blocks provide sufficient clearance for ½"-20 socket head cap screws (SHCS). The SHCS should meet ASTM A574 strength requirements. Fasteners meeting this requirement can be found at McMaster Carr, manufactured by HoloKrome. To provide sufficient holding power fasteners should be installed with hardened washers, McMaster # 90850A300, and a thread locker such as Loctite 242, and torqued to a minimum of 67 ft-lb of torque. Minimum thread engagement is 4 threads, recommended bolt length is 1.75", McMaster # 91251A018 (provides for ½" of thread engagement). This configuration yields a minimum safety factor of 2. No pins are required to resolve the shear forces in the test apparatus. The chart below shows additional torque values for higher safety factors. Maximum torque should not exceed 172 ft-lb. Recommended torque value of 100 ft-lb ± 10% guarantees minimum safety factor of 3. All numbers were calculated with values taken from HoloKrome Technical Handbook and the Concise Metals Data Handbook by J.R. Davis.

Bolt Parameters											
HoloKrome Cap Screw			Torque @ FS=2			Torque @ FS=3			Torque @ FS=4		
Size	1/2" - 20 UNRF		FS_{min}	2			3			4	
F_{yield}	25905.0	lbf	FS_{shear}	2.00		FS_{shear}	3.00		FS_{shear}	4.00	
	115.2	kN	$FS_{tension}$	2.80		$FS_{tension}$	3.70		$FS_{tension}$	4.59	
$F_{tensile}$	28780.0	lbf	F_{fric}	25.9	kN	F_f	38.8	kN	F_f	51.7	kN
	128.0	kN		5812.0	lbf		8718.0	lbf		11624.0	lbf
τ_{THD}	18105.0	lbf	F_{ytotal}	92.8	kN	F_{ytotal}	139.2	kN	F_{ytotal}	185.7	kN
	80.5	kN		20868.0	lbf		31302.0	lbf		41736.0	lbf
τ_{BODY}	21175.0	lbf	$F_{preload\text{-}fric}$	32.5	kN	$F_{preload\text{-}fric}$	42.9	kN	$F_{preload\text{-}fric}$	53.3	kN
	94.2	kN		7295.6	lbf		9639.1	lbf		11982.7	lbf
T	172.083	ft-lb	$F_{preload\text{-}tension}$	23.20637	kN	$F_{preload\text{-}tension}$	34.80956	kN	$F_{preload\text{-}tension}$	46.41274	kN
	232.5	Nm		5217.0	lbf		7825.5	lbf		10434.0	lbf
$F_{Preload}$	17800.0	lbf	T	66.8763	ft-lb	T	88.35883	ft-lb	T	109.8414	ft-lb
	79.2	kN		90.4	Nm		119.4	Nm		148.4	Nm

The ½" - 20 SHCS torqued the corrected value and installed as noted above will be sufficient to secure the pedestals to the mounting plate. Only 2 fasteners are required per pedestal. Plate thickness does not have an effect on fastener sizing.

No pins are required to resolve shear forces, friction provides sufficient resistive force. If desired, a minimum of a 3/16" pin can be installed to resolve all of the shear force in the apparatus (HoloKrome manual provides single shear values for the pins).

Additional calculations for rack and pinion selection (See Page A-32)[27]

Given 18 inches of travel for the carriage, at 2.23 g acceleration, the carriage will travel at 5 mph for 9 inches. Given 18 inches of travel for the carriage, at 4.46 g acceleration, the carriage will travel at 5 mph for 13.5 inches.

Per table below, for 20 inches of travel and a maximum pinion diameter of 8 inches, the rack must be at least 28 inches long. Given the hole locations on the QTC Gears KSRFD2-1000 rack, the rack will be about 30 inches long.

Remaining design decisions:

1. Grade, material, hardness.
2. Pinion size.
3. Straight or helical.

travel stroke	18	inch
over travel	2	inch
maximum pinion diameter	8.031496	inch
rack length	28.0315	inch
rack length	712	mm
carriage length	37.5	inch
rail length	57.5	inch

feed velocity	mph	in/sec	ft/sec	in/min	
	5	88	7.333333	5280	
trapezoidal velocity profile, 18 inch of travel in 3t seconds					
t=	0.102273	sec			
accel	860.4444	in/sec^2	2.2289	g	
	0	0	0		
s1	4.5	0.102273	88	4.5	
s2	13.5	0.204545	88	9	
s3	18	0.306818	0	4.5	
				18	
trapezoidal velocity profile, 18 inch of travel at 4.45g acceleration					
t=	0.051136	sec			
accel	1720.889	in/sec^2	4.457799	g	
	0	0	0		
s1	2.25	0.051136	88	2.25	
s2	15.75	0.153409	88	13.5	
s3	18	0.051136	0	2.25	
		0.255682		18	

Action Item 31. *Review design with Dennis Denney. Designate fabrication method for each fabricated part.*

Braddock Metallurgical on Thursday, August 11, 2011 at 9:00 a.m.
Brian Waters, Tim Waters, Dennis Denney, and Wiley Holcombe

We reviewed drawings for 1-inch-thick cutter with candidate tooth designs, 5-inch-thick cutter, and track roller rail. We discussed material options, heat treatment options, and achievable hardness.

Track Roller Rail
They recommended against case hardening the track roller rail; the rail will not be flat after processing. They suggested using a D2 tool steel and through hardening the rail. They predicted that the rail would remain flat. 55 HRC is achievable.

5-inch-thick cutter, Option 1
Fabricate from 8620 steel alloy; balance before hardening; harden through to 30-35 HRC; and carburize to 60-62 HRC, 0.060 inch deep. The ID of the bore will tighten up during carburizing. They can mask the bore to prevent carburizing the bore and allow for post machining. Dennis suggested that bore could be machined with wire EDM even in hard state.

5-inch-thick cutter, Option 2
Fabricate from D2 tool steel; balance before hardening; harden through to 60^{+} HRC.

1-inch-thick cutter
Fabricate from D2 tool steel (or possibly M2, M42, M4 tool steel). Harden through to 60-62 HRC.

Note:

Lead time will likely be on the order of one week. They harden D2 nearly every day.

Carburized cutter might be more brittle, less tough, than through hardened cutter. Tim noted that the candidate teeth with the groove-type chip breakers are quite thin.

Consider fabricating a second 1-inch-thick cutter and using carburizing process prior to fabricating a 5-inch-thick cutter that will be hardened by carburizing.

We did discuss pavement strikes.

Action Item 29. *Add shock absorbers to each end of the carriage to absorb impact energy.*[28]

Enidine Shock Absorber						**Coil Spring**		**Max**	**Weight**
Catalog No./Model	**Stroke**	**Range**	**Max.**	**Max.**	**Max.**	**Extended**	**Compress**	**Propelling**	**(mass)**
			Energy	**Energy**	**Reaction**			**Force**	
					Force				
	in.	***in./sec.***	***in.-lbs./cycle***	***in.-lbs./hour***	***lbs.***	***lbs.***	***lbs.***	***lbs.***	***oz***
ΔOEM 1.25 x 1	1	12-130	1,700	808,000	2,500	12.5	20	500	20
	(mm)	***(m/s)***	***(Nm/cycle)***	***(Nm/h)***	***(N)***	***(N)***	***(N)***	***(N)***	***(g)***
ΔOEM 1.25M x 1	25	(0,3-3,30)	195	91000	11120	56	89	2220	567
carriage mass	27.50	kg							
velocity	2.24	m/s	88	in/s					
KE	68.70	Nm							
maximum propelling force	592.90	N	motor capable of driving load at 2.2 g						
work energy	15.06	Nm							
Total energy	83.76	Nm	83.7562968 (N m) = 741.305691 pound force inch						

References

[1] *Development of a method to remove raised-pavement markers (RPMs) from road surfaces, GTRI Proposal Number ATAS-FPTD-11-1306*, Georgia Tech Research Institute, September 2010, p. 2.
[2] *RPM* Removal, IRAD Final Report, I7000.6.3.8.0, August 01, 2006, pp. 23-24.
[3] *Development of a method to remove raised-pavement markers (RPMs) from road surfaces, GTRI Proposal Number ATAS-FPTD-11-1306*, Georgia Tech Research Institute, September 2010, p. 3.
[4] *Development of a method to remove raised-pavement markers (RPMs) from road surfaces, GTRI Proposal Number ATAS-FPTD-11-1306*, Georgia Tech Research Institute, September 2010, p. 3.
[5] *Development of a method to remove raised-pavement markers (RPMs) from road surfaces, GTRI Proposal Number ATAS-FPTD-11-1306*, Georgia Tech Research Institute, September 2010, p. 4.
[6] S:\Projects\GDOT RPM Removal\Requirements\calculations 110126.xlsx, Tab *prototype*.
[7] *RPM* Removal, IRAD Final Report, I7000.6.3.8.0, August 01, 2006, p. 20.
[8] S:\Projects\GDOT RPM Removal\Requirements\calculations 110126.xlsx, Tab *prototype*.
[9] *RPM* Removal, IRAD Final Report, I7000.6.3.8.0, August 01, 2006, p. 12-13.
[10] S:\Projects\GDOT RPM Removal\Requirements\calculations 110126.xlsx, Tab *cutterenergy*.
[11] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\flywheel calculation check.xlsx, Tab *flywheel (2).*
[12] Roark and Young, *Formulas for Stress and Strain*, 5th edition. New York: McGraw-Hill, 1975, pp. 567.
[13] http://www.codecogs.com/reference/engineering/materials/rotating/discs/and/cylinders/rotating_discs_and_cylinders.php
[14] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter 110216-Study 2-1.htm
[15] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter 110216-Study 3-1.htm
[16] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutter, simplified-Study 3-1.htm
[17] S:\Projects\GDOT RPM Removal\Requirements\calculations 110126.xlsx, Tab *50mm-shaft*
[18] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\main shaft calculation check.xlsx, Tab *sheet1*
[19] Roark and Young, *Formulas for Stress and Strain*, 5th edition. New York: McGraw-Hill, 1975, pp. 100, Table 3, result 2e.
[20] http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=M4130S.
[21] S:\Projects\GDOT RPM Removal\Requirements\calculations 110126.xlsx, Tab *crtitcalspeed.*
[22] Spots, M.F., *Design of Machine Elements*, 5th ed., 1978, pp 157-158.
[23] Timoshenko, Stephen, D. H Young and William Weaver, *Vibration Problems In Engineering*, 4th ed. New York: Wiley, 1974, pp.42-44.
[24] http://www.engineersedge.com/bearing/critical-speed-distributed-load.htm
[25] S:\Projects\GDOT RPM Removal\Designs\Conceptual Design\cutting force extrapolation.xlsx, Tabs *Sheet1* & *Sheet2*.
[26] *Speed Reducers and Gear Trains*, http://www.sdp-si.com/D220/D220cat.htm#T58
[27] S:\Projects\GDOT RPM Removal\Designs\Design Spreadsheets\carriage calculations 110728.xlsx, Tabs *RPMvprof* & *stroke.*
[28] S:\Projects\GDOT RPM Removal\Designs\Design Spreadsheets\carriage calculations 110728.xlsx, Tab *shockabsorber*