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December 11, 2008

Franklin French  
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Dear Mr. French:

Enclosed is a copy of “Spar mill Work holder Solution.” This report is a summary of our findings from the work that we have completed during the summer and fall of 2008 on a possible solution to the spar mill work holder failure problems.

Several concept designs were incorporated into our final design. We have included a passive system along with a cylinder quantity reduction system featuring a common shaft. We are planning to use one cylinder for every two clamps, which will only be active during the unclamping of the spar. This will dramatically decrease the amount of time that the cylinders are under pressure, and in turn significantly decrease the failure rate. This system is discussed in more detail in the design report.

If you have any questions and/or comments regarding the interpretation of this report, please feel free to contact us at the email address above or contact our instructor Jay McCormack. Thank you for supporting this project. Hopefully this project will be a benefit to you.

Sincerely,

Team Sparmill  
-Andy Florence  
-Ben Puyleart  
-Ryan Mathews

Enclosure: Final Report
Spar mill Work holder Solution:
A Project for the Boeing Company

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December 11, 2008
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Executive Summary

The spar mill work holder for the Boeing Company has been in slow degradation for the past 15 plus years. They are experiencing several problems with the failure of the hydraulic cylinders that are used to clamp the spar to the base. On average, they have to replace or rebuild 50 cylinders per week, so this is obviously a cost and maintenance issue. We were approached with the opportunity to design a new system of clamping the spars to the base. To keep changeover costs to a minimum, Boeing recommended that we try to maintain the current configuration of the base and top castings.

After a thorough concept generation, the system that we elected to explore in more depth is the passive system in combination with a common shaft system. The reason for this is that it would be very inexpensive to implement and would significantly reduce the amount of time that the cylinders were under pressure. Our design only uses one cylinder to deactivate every two clamps and a series of springs to handle the required 1500 lb clamping force.

This new system introduced a common shaft system which was able to cut the number of cylinders in half. It also featured a passive spring system which reduced the amount of time the hydraulic system is pressurized. These features yielded in a potential of 1/32 of the current maintenance time.

Some recommendations have been stated to improve the viability of the new design. Such things as cutter interference, cylinder pressure, and clamping force need to be addressed and further researched.
Background

The current spar mill work holder in place at the Boeing Company's Auburn skin and spar division has been causing a maintenance problem for the last 15 plus years. The original plan was to phase out this plant and disperse the operations throughout other facilities, but high demand has prevented this from happening. The issue is that the hydraulic cylinders used in the clamping of the spars to the work holder are failing at an accelerated and costly rate. On their current system, each machine can contain over 300 cylinders which are pressurized for over 16 hours per day. From data given to us, we calculated that the average maintenance effort in their facility is 1500 minutes per week on hydraulic repairs.

When the work holders were originally implemented, they were outfitted with shields that covered the openings at the cylinder locations. This was designed to prevent contaminants from entering the area around the cylinders, and hopefully eliminate many of the cylinder failures. The problem with this system is that instead of preventing contaminants from entering, it actually held them in this area. To make matters worse, the trapped material would end up hardening together with the flood of cutting fluid that is used in the process. Obviously this system was counterproductive and had no benefits in preventing cylinder failure.

Boeing revealed potential solution to the problem. The most prominent idea that we have collaborated on is the possibility of using springs to create a more passive clamping system. The spar would be clamped by the springs, and the hydraulic cylinders would be used to release the clamps during changeover of the spars.

Problem Definition

The efforts of the Sparmill team are to significantly reduce the amount of maintenance required to keep the mill up and running. This project will save the company money and repair time in the long run. The final goal which we achieved in this design period was a working prototype of our final design of a 1 foot clamping section. Our primary client for this project is The Boeing Company, while the secondary client is The University of Idaho.

A detailed table of the requirements and spec's are illustrated in Figure 1 of appendix A. A summary of the requirements are:

- Hold work piece in place during clamping
  - Provide a clamping force of 1500 lbs
- Maintain current clamping positions
  - Allow for 12 inch spacing between clamps
- Reduce maintenance efforts
  - Less than 1500 minutes per week
• Survive hostile environment  
  o Has a service life of at least 10 years  
• Fit in standard machine base  
  o No greater than 7 inches by 4.5 inches in size  
  o Provides 1.5 inches of linear travel  
• Allow for manual clamping  
• No obstruction of the cutters  
  o Fits in current geometry  
• Provide signal that maintenance is needed  
  o Applied force deviation of no more than 200 lbs  
• Safe operating environment  
  o Does not have any stored force in the clamps when not in use

The Boeing Company specified that they would prefer to not make any radical changes to their current system. Basically this means that any item which has been casted should remain in its current configuration. This will help prevent a major overhaul and costly redesign of the system. They, like anyone, want to keep the costs to a minimum, while still implementing an effective solution.

By satisfying these requirements we will assure that the solution will provide all of the necessary features and will satisfy the customer’s needs.

Project Plan

Our project was guided by a six step process illustrated to the right in figure 2. Tasks for project learning were specified by the team and given to each team member use MS project. Many preliminary tasks needed to be completed before others could be started which resulted in team commitment to deadlines. The MS Project schedule used to date can be seen in figure 3 in the appendix.

During the project learning phase, a hydraulic mockup was constructed for testing using an actual spar mill cylinder, some fittings, a pressure gauge and the use of a hand pump. Next our team calculated the pressure needed in the cylinder to apply the client specified 1500lbs of force. The pressure was then verified with the use of a force testing platform and the purchased pressure gauge. With a method of measuring force found and hydraulic pressures necessary for achieving this force verified, preliminary research for conceptual design could be performed. A problem
tear down was performed to determine what was going wrong within the system. From this we could see what some potential solutions could involve. Cam profiles, hydraulic motors and a digital mockup of the top holder, top casting and riser were studied. Space constraints were identified with the digital mockups and compatibility of hydraulic motors to meet these size constraints was found.

In the conceptual design phase of the project, further concepts were studied and established with the use of the specifications. Initial torque and force calculations were made to ensure that each conceptual design would meet the requirements however, detailed calculations were not performed as we weren’t sure which design our clients would lean towards as a result of our design review. With initial research performed on our conceptual designs, ground work was made for future work. Contacts were made for Belleville springs and an easy to build base mockup design was completed.

The final design began after our design review with Boeing. Along with Boeing engineers and operators, we devised key features and a plan for the final design. We designed and fabricated our prototype during the fall of 2008. We then went on to test the prototype to ensure that it would meet all of the specifications.

**Concepts Considered**

In search for solutions to meet our client’s needs, a problem teardown was performed to identify exactly what functions needed to be performed in order to achieve the end result, a clamped spar. Of course, these functions needed to be achieved while maintaining the specifications agreed upon with our client. The client specifications and problem teardown can be seen in figures 1 and 4 respectively in the Appendix A. The problem teardown helped us in realizing the role that each current clamping part played in the clamping of the spar. With each role identified, the linear motion currently provided by the hydraulic cylinder became the main focus of our redesign. By maintaining the same base and much of the top assembly, client specifications and changeover could be achieved more easily. With linear motion identified as the main focus of the redesign, a morphological chart was assembled which showed the necessary functions and solution principals for achieving these functions. These solution principals could then be used in multiple ways to achieve the desired functions and therefore provided our design team with multiple solution paths to consider. The morphological chart can be seen in figure 5 in appendix A. As seen in the morphological chart, actuation options were electric motors, hydraulic cylinders, hydraulic motors and ball nuts. Pneumatics weren’t considered due to the probability of the same failures with no true advantage. Open and close clamping forces could be provided with hydraulic cylinders, Belleville washers, or helical compression springs.

With solution principles identified and client specifications considered, the conceptual design moved forward. Creative design concepts were presented during our project review on July 18, 2008 at Boeing’s Auburn plant. Excellent input was
received by clamping operators, maintenance technicians, project managers and the equipment engineer, which shaped our concept selection and definitely narrowed down our solution path, which will be described in the concept selection portion of this report. The concepts delivered at the design review are described in the following paragraphs. More information on these concepts can be viewed in appendix B (figures 6-12).

The cheapest and easiest solution was considered first and included retrofitting the current design (figure 13). By adding a conical shield, the hydraulic cylinders’ seals (the major failure point of the current setup) would be subjected to less aluminum shavings and coolant (believed to be the main contributors for cylinder failure). Additionally, a rubber mat applied to the top casting was proposed for the same effect.

Fittings were also considered to reduce the bend radius that the current hydraulic hoses are subjected to. Finally, side shielding could be provided with thin sheet of metal applied. View ports could be installed in these side shields to allow for a means of visually realizing failure in the form of fluid leakage on the top of the cylinders. These retrofitting ideas could of course be incorporated into all of our design concepts in one form or another to help protect from the harsh environment of the clamp setup. An operator at the design review expressed a concern with the side shielding. The operator stated that side shielding had been applied before and did help in reducing the quantity of chips and coolant near the cylinders, however, the shielding also resulted in a cement-like formation of chips that was extremely difficult to clean up. Total cost of this retrofitting option was estimated at $100 per 6 foot module.

The next design concepts (figure 14) involved a hydraulic or electric motor, which would provide rotational motion to cams or a crankshaft. The cams or crankshaft would support 6 ft. modules and convert the rotational motion to linear motion. The modular design was considered to reduce the number of moving parts and in turn reduce the occurrence of failure. This design would also be cost effective and provide for ease in the changeover process. As required in
the client specifications, 3 inches of travel could be achieved with common cams or a 1.5 in crankshaft throws. As in the majority of our design concepts, Belleville springs were considered to allow for variance in necessary linear motion. Of course with a variance in linear motion, the springs would result in a variance of force provided and the question of allowable variance in applied force was asked during the design review with our client. We have currently not been given the answer to aid in further consideration of such designs. The cam setup could provide a passive clamping force with the addition of a dwell in the cam design. This flat spot or "dwell" would hold the clamp in place without the need for continuous hydraulic or electric energy. The crank could not utilize this aspect; however, with the use of an electric brake, the electric motor could provide for a passive system as well. Estimated cost per 6 ft. module was $3,500 for either design. A concern realized by our team members when we visited the Auburn plant was the staggering of the current hydraulic cylinders. With this modular design concept, staggering the linear motion effect might prove to be difficult and impractical.

Following the passive system idea, the next design concept utilizes hydraulic cylinders only when opening the clamping mechanism to allow for spar insertion and removal (figure 15). Belleville springs would be used to provide the 1500lbs of force required in our client specifications and would be compressed with the use of a hydraulic cylinder. By utilizing the cylinders only during opening of the clamps, the need for 8 hours of continuous hydraulic energy is diminished and a lesser occurrence of hydraulic failure would follow. Also, failure of this design results in a clamped spar, ideal for finishing a milling operation as opposed to the current setup which results in an unclamped spar during failure. Concerns for this design include the ability of Belleville springs to be compressed the required distance to allow for a part to be inserted or removed. Another concern is for safety. The stored energy of the spring packs could be dangerous for the maintenance personnel. The estimated cost of the Belleville passive spring design is $400 per 6 ft. module.

Next, a ball nut and spring design was proposed (figure 16). The screw stock for such a design could be turned with a number of different mechanisms however; a worm gear was considered to aid in self locking to again provide for a passive system. The ball nut converts rotational energy into linear
motion with a major advantage of relatively low torque required to provide linear motion. Lower torque motors would of course take less space than the motors necessary to turn a crankshaft or cam, especially in a modular design and could again be electric or hydraulic. Belleville springs were again considered to allow for the variance in linear travel and the entire ball nut/screw/spring assembly would be housed in a module with dimensions similar to the hydraulic cylinders currently used. This would allow the new module to simply slip into the current base for ease of changeover. The estimated cost for this concept design is $16,000 per six foot module.

Further research of the ball nut and spring assembly resulted in our finding of fully enclosed electric actuators (figure 17). These electric actuators use a ball nut and power screw design again allowing for low torque motors. The Exlar Corporation is one manufacturer of such a device which could be designed to fit in the current base setup. Exlar’s product could also provide further capabilities of position and force sensing. These linear actuators might be the ideal solution however cost was a concern and as expected, it was expressed at the design review by our client. The estimated cost per 6 ft. module is $72,000.

The last conceptual design presented at our design review involved cylinder quantity reduction (figure 18). An illustration of the reduction concept can be seen in figure 9a of the appendix. By using half of the number of cylinders, half of the number of hydraulic lines would be used. As a result, the number of failures would also be reduced. Two options for this reduction in cylinders were considered. One, use the same cylinders and increase the hydraulic pressure in the current system. Our design team had concerns with this option from an expected increase in hydraulic failure due to the higher pressures. The second option would be to use a new cylinder with double the cross sectional area of the old cylinder. This would provide for twice the force as necessary for the design to work and would allow the system pressure to remain the same. The downfall to this method would be the initial cost of
buying the new larger cylinders and fitting them with a new flange for mounting purposes.

**Concept Selection**

Before our preliminary review, the team created a morphological chart, which is shown in appendix A (figure 5). This figure shows how the different designs contribute to each required function. We then evaluated the pros/cons and cost of each design. This can also be seen in the appendix. From this we saw which concepts had the most advantages for the application with respect for cost, fabrication, and performance. Table 1 clearly and simplistically represents this data. We could see how some concepts had extremes which reduced they’re validity. Other designs such as the passive system, retrofitting, and cylinder reduction stood on good ground because they were cheap, easy to fabricate, and had a reasonable opportunity for performance. The retrofitting concept yielded less performance but it can be easily applied to other concepts to improve their performance.

<table>
<thead>
<tr>
<th>Concept</th>
<th>cost</th>
<th>fabrication</th>
<th>performance</th>
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<tbody>
<tr>
<td>retrofitting</td>
<td>$100</td>
<td></td>
<td></td>
</tr>
<tr>
<td>cam/spring</td>
<td>$3,500</td>
<td></td>
<td></td>
</tr>
<tr>
<td>crank/spring</td>
<td>$3,500</td>
<td></td>
<td></td>
</tr>
<tr>
<td>passive</td>
<td>$400</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ballnut</td>
<td>$16,000</td>
<td></td>
<td></td>
</tr>
<tr>
<td>linear actuator</td>
<td>$72,000</td>
<td></td>
<td></td>
</tr>
<tr>
<td>cylinder reduction</td>
<td>$3,400</td>
<td></td>
<td></td>
</tr>
</tbody>
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Table 1 - Comparison of cost, fabrication and performance of each design concept

A lot of our concept selection occurred during the preliminary design review. We were able to look at the current set up at the Auburn plant and hear from the operators to get an idea of what would work and what would not work. The operators gave us details such as how the side panels on the retrofitting idea not only keep material out, but hold material in. They also gave us insight on how some designs, such as the crank and cam, would not work for their staggered cylinder pattern which is prevalent on some of their machines but not on the one we are set to design around. As we ended our design review, we then took the ideas and insights which they gave us and worked to produce a new concept. This concept included some ideas from retrofitting, passive system, and cylinder reduction concept.
System Architecture

System Design

<table>
<thead>
<tr>
<th>Feature</th>
<th>Description</th>
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<tbody>
<tr>
<td>Common Shaft:</td>
<td>Rigidly connects two clamps to reduce the quantity of hydraulic cylinders</td>
</tr>
<tr>
<td>Spring System:</td>
<td>Springs supply 1500 [lb] clamping force rather than the hydraulic cylinders</td>
</tr>
<tr>
<td>Release Arm:</td>
<td>Allows for release of the spar. Prevents side loading of the hydraulic cylinder.</td>
</tr>
</tbody>
</table>

Table 2

Figure 19
The final concept, shown in more detail in appendix C (figures 20-24), was not one of our initial concepts. As stated before, it is a combination of a few. The 3d model in figure 19 is for the reader to get a basic idea of the structure and features. A summary of the key features is expressed in table 2 above. One key feature on this design is the common shaft, which is shown in detail in figures 20-24 of appendix C. This shaft reduces the number of required cylinders which translates into fewer components for failure and less maintenance. Our design reduces the number of hydraulic cylinders by 50 percent. To prevent a constant pressure being exerted on the pistons while it is in the machining process, we switched it to a passive system so that springs (located inside spring housing), guided by a pushrod will push up against the rocker arm to create the force required. The pistons are therefore only pressurized in order to pull down on the rocker arm, compressing the springs to disengage the clamps. The systems parts are shown in figure 19. One requirement for the springs is to supply the force required to clamp but also allow for the vertical actuation required when the pistons disengage the clamps. To make it possible for only two pistons to actuate the clamps on one side, we incorporated a common shaft. This shaft is rigidly connected to the rocker arms and pivots about the same axis as the clamps. The clamps are attached to the shaft with keys (figure 25) at the desired location by a center hole which the shaft can run through. The key for the keyway is designed to be the failsafe in the system. For the prototype, it was designed to have the lowest safety factor of 2.3. This ensures that we know where the failure will occur if something goes wrong which results in no safety issues to the operator.

Since this system is still hydraulic based, we have formulated ideas to minimize the hydraulic failure. The current hydraulics can withstand higher pressures, but this could prove to increase the failure rate. So we created a longer release arm (shown on figure 19) on the common shaft which will create more leverage and reduce the required pressure in the cylinders. To keep contaminants from ruining the seals on the hydraulic cylinder, a boot or deflection cone can be inserted on top of the cylinder (as seen in figure 6 of appendix B). New hydraulic fittings on the cylinder end of the hoses can also be beneficial. With ninety degree free swivel ends, the hoses are put in much less of a bind which contributes to a longer service life. This feature is also shown in figure 6 of appendix B.

This setup has been designed so that there are numerous ways to compress the springs for installation and removal. This is because many different failure modes could leave the system vulnerable to only a few methods of disassembly. The easiest method of removal occurs when the system is in working order. For removal, the springs are fully compressed and the two safety device pins are inserted into the
spring housing. Then the cylinders are de-pressurized slowly. The spring’s movement is stopped as it pushes against the pins. Then the spring housing subassembly is simply removed. The same concept can be applied for installation. Another method of installation and removal is available for the instance of failure. This technique for removal is more time consuming and should only be used when needed. It requires a spacer tube and a nut/washer to be attached on the bottom of the pushrod. The bottom of the pushrod has certain length of threads incorporated onto it (as seen in figure 26). The spacer is placed between the base and the nut and the nut is screwed onto the pushrod. The nut is then screwed down, compressing the spring. The safety pins are once again installed when the spring is compressed enough and removal can then be performed.

This concept is also fairly cheap and easy to fabricate. Only the top mounting module will require some machining. A new mounting system for the common shaft will need to be incorporated on to the top clamping module. We do not have enough data to do a detailed cost analysis. However, we deduced that this design could potentially cut the maintenance time down to 47 minutes per week. This is 1/32 of the current maintenance time so it could potentially cost only 1/32 of the current costs. The design itself, cost us around $1000 for two clamps. We assume that this price can be drastically dropped when buying in bulk. More in-depth cost data is provided in figure 27 in appendix C.

**Stress Analysis**

To ensure that the system would not fail, numerous methods and calculations were performed to validate the design. Major components where the stress would most likely be highest were the main focus. These areas included the common shaft, keys, clamp and release arm welds, and the bolts used for the guides holding up the common shaft.

The common shaft was analyzed using a TK solver math model. This model analyzed the section of the shaft which underwent the highest torsion which is in between the clamp and the release arm. The maximum force which would occur, which is 2000 [lbf], was used in the model. With this math model, it was determined that we had a safety factor of 2.6. The bolts used in the common shaft guides were analyzed the same way but with different equations. The end result yielded a very high safety factor of around 140. Other loading situations underwent slightly different stresses. However, since it was found that the bolts had such a great safety factor for one
loading set, it was assumed that it would be safe against failure for all other loading sets.

For the keys it was found the bearing force needed to be analyzed due to the highest stresses. A math model again using TK solver was used to acquire a safety factor of 2.3. Once again, this is the lowest safety factor which prevails in the design. This is to ensure that the keys fail instead of any other major components. This analysis used the highest spring force just as in the common shaft analysis.

The welds on the clamps were the most difficult to analyze. Weld stress equations use parameters derived for each weld shape. There are parameters for full circle welds which are incorporated into the release arm, but there are no parameters for half circle welds. So an ALGOR model (figure 28 in appendix C) was set up to analyze the stresses. The ALGOR model uses an old half circle weld model of the release arm but since the clamps are very similar except for half the loading, it is still a valid analysis. The stresses that occur on the ALGOR Half Circle Release Arm model will be double for that of the clamp. This model shows that the stresses around the welds are close to 40 [Kpsi]. This is under the yield strength of steel and gives a safety factor of 1.5 (3 for a full circle weld). The clamps will only undergo half of the stress (FOS = 3) so they are within safe limits as well.

**Force Analysis**

To start the force validation, the helical compression springs were first tested to ensure a valid spring rate. A SATEC T5000 outfitted (figure 29) with a 5000 lbf Interface load cell (model number 1210-a) was used to acquire data on the spring. This data was then used to determine the spring rate. Three runs were performed where the spring was compressed from 0 inches up to 4 inches. Data points were collected at every 1/4 inch of deflection. This data is illustrated in figure 30 of appendix C.

Once the spring rate was known, the clamping force was then calculated. This was calculated using a set of equations relating the hydraulic pressure, spring force, and frictional effects. Illustrated in figure 31 is a diagram which shows the relevant
forces which we used to derive the static equations. The frictional effects were determined to have a significant impact and therefore were included in the calculations. The hydraulic pressure was acquired directly off a pressure gauge which was plumbed into the hydraulic system. In the equations which were derived, the independent variable was the spring compression distance \( x \) and dependent variables include the hydraulic pressure \( P_h \), clamping force \( F_c \), and friction force \( F_f \). These equations are illustrated below.

"Determining Friction forces"
"Sum of Moments-no clamping force"
\[
0=F_hL_h-2F_sL_s-(F_f)D_s/2 \\
F_h=P_hA_h \\
A_h=\pi/4(2.5^2-1^2) \\
F_s=k\text{deltax} \\
\text{Deltax} = x_{\text{free-x}}
\]

"Determining Clamping Forces"
"Sum of Moments-clamping force"
\[
0=2F_cL_c-2F_sL_s+(F_f)D_s/2
\]

To Determine the Error in the calculations, a RSS method was used. Using the two moment equations (1 and 6) illustrated above, the error propagation equations were formed. These two equations are illustrated below.

"Error in the force of friction"
\[
\text{Error}_{F_F} = \left( (L_h^2/D_s)*\text{Error}_{F_h} \right)^2 + \left( (L_s^2/D_s)*\text{Error}_{F_s} \right)^2
\]

"Error in the Clamping Force"
\[
\text{Error}_{F_c} = \left( (L_s/L_c)*\text{Error}_{F_s} \right)^2 + \left( D_s/(4L_c)*\text{Error}_{F_f} \right)^2
\]

"Deviation due to uncertainty in clamping force"
\[
\text{Deviation}_{F_c} = F_c*\text{Error}_{F_c}
\]

Through the calculations, it was discovered that our actual clamping force is 1244 [lbf]. This is due to the friction effects which are present in the shaft mounts. The expected error in our measurements is ±30 [lbf]. This error is coming from the uncertainty in the pressure, spring compression, and spring rate.

In the end we have a product that meets or exceeds all of our requirements. The most prevalent requirement that our design meets is the resulting lower maintenance due to fewer cylinders and a better environment for the hydraulic cylinders. The design also fills many requirements by maintaining the same base and clamping module, allows room for the cutter, fits the current base, and provides
for manual clamping. With the safety pins, a safe working environment is achieved. Even though 6 inch spacing clamps are not incorporated into our current design, it can possibly be applied into a later design. A substantially lower change over cost will also result due to the limited redesign and fabrication.

**Future Work:**

For future integration, the final design will need to have a few additional features incorporated. One main feature which needs to be addressed is protection from contaminants. Refer to the DFMEA (design failure mode effects analysis) for this and other design recommendations. This is located in appendix C (figure 32) and also on the website at [http://seniordesign.engr.uidaho.edu/2008_2009/sparmill/](http://seniordesign.engr.uidaho.edu/2008_2009/sparmill/). To avoid contaminent build up around the spring, a boot or some other type of shield needs to be fitted on top of the spring housing. If this shielding is not present, metal shavings will eventually fill the housing and the spring will be rendered unmovable. A shield is also suggested to be placed on top of the hydraulic cylinder. Even though the cylinder is inactive during machining, contaminants can still build up and cause failure of the seals while unclamping.

Safety is another important focus area for future consideration. A very large force is present within the compressed spring. If something were to happen during unclamping, such as hydraulic failure or some other type of system failure, there is a large risk for injury. The new design features safety pins which are inserted into the spring housing. This is an effective safety method but it could prove to be a time consuming process if it is incorporated into a larger scale. A automatic lock out system would provide a safe, yet time efficient method of retaining the store spring force. This lock out mechanism could simply lock the pushrod like safety pins, or it could attach to the clamp itself. Other safety issues such as this can be seen in the DFMEA located in the appendix (figure 32 of appendix C).

Another possible safety hazard is failure within the metal. The prototype uses weaker metals than intended due to budget and time constraints. For future models, it is recommended that stronger metals should be used for the common shaft and keys. The welds around the clamp and release arm might also need to be further investigated to increase their strength.

An issue which was not accounted for is interference with the cutter. It was learned at the final presentation that on the current design, it is possible for the cutters to run into the clamps. This is a problem for the new-design because it is made of steel and the release arm is more exposed than the clamps. To elevate this problem, it may be necessary to change the material and change the design of the release arm to leave more room for the cutters.

Force determination is another issue which needs to be explored more. From the data which was received from boeing, the current system would need a hydraulic pressure of over 900 [psi] to supply a clamping force of 1500 [lbf]. Boeing has stated at the final presentation that they do not go higher than 500 [psi]. There might be
some error in the pressure data which was given to the team for the machine #2 workholder. Force determination is also recommended to be further researched and experimented with on the prototype.

As presented before, it was determined that the clamping force was under the expected value. This was due to frictional forces. For later research and design, it is recommended that design features be incorporated to relief these effects. Such things as bushings, bearings, or grease fittings could be beneficial. The frictional effects on the current design also need to be investigated to insure that they are maintaining their desired clamping force.
### Appendix A

<table>
<thead>
<tr>
<th>Requirements (a.k.a Voice of the customer or Customer requirements)</th>
<th>Clamp force</th>
<th>Clamping Spec</th>
<th>total downtime</th>
<th>Repair time</th>
<th>X Accessibility for repair</th>
<th>Service life</th>
<th>Fits base dimensions</th>
<th>Fits manual clamp</th>
<th>Average clamping force</th>
<th>Applied force deviation</th>
<th>provide linear movement</th>
<th>stored clamping force when not in use</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hold work piece in place during cutting</td>
<td>X</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maintain current clamping positions</td>
<td>X</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Reduce maintenance effort</td>
<td></td>
<td></td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fit in standard machine base</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Survive hostile environment</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Support different work holders (dovetail and flat) with a common actuator base</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Allow for the manual clamping</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Provide signal that maintenance is needed</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Safe operating environment</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>X</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Units of measure</th>
<th>lb</th>
<th>inch</th>
<th>hr/ failure</th>
<th>min/wee</th>
<th>min/failure</th>
<th>years</th>
<th>in</th>
<th>hours</th>
<th>lb</th>
<th>in</th>
<th>lb</th>
</tr>
</thead>
<tbody>
<tr>
<td>Target values</td>
<td>1500</td>
<td>12</td>
<td>1 to 2</td>
<td>1500</td>
<td>10</td>
<td>10</td>
<td></td>
<td>8</td>
<td>200</td>
<td>1.5</td>
<td>0</td>
</tr>
<tr>
<td>Measured values</td>
<td>1244</td>
<td>12</td>
<td>1 to 2</td>
<td>47</td>
<td>10</td>
<td>10</td>
<td>74.5</td>
<td>8</td>
<td>256</td>
<td>1.3</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 1: Client Specifications
<table>
<thead>
<tr>
<th>Week 1</th>
<th>Week 2</th>
<th>Week 3</th>
<th>Week 4</th>
<th>Week 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design</td>
<td>Drafting</td>
<td>Analysis</td>
<td>Review</td>
<td>Final</td>
</tr>
</tbody>
</table>

**Figure 3**
**Morphological Chart**

<table>
<thead>
<tr>
<th>Function</th>
<th>Solution Principles</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allow for adjustment</td>
<td>Use current base in combination with compact design</td>
</tr>
<tr>
<td>Actuation</td>
<td>Electric motor</td>
</tr>
<tr>
<td>Open/close Clamp Force</td>
<td>Hydraulic cylinder</td>
</tr>
<tr>
<td>Indicate Failure</td>
<td>residual fluid leakage</td>
</tr>
<tr>
<td>Modularity</td>
<td>Cam</td>
</tr>
<tr>
<td>Shielding</td>
<td>Top Cover</td>
</tr>
</tbody>
</table>

Figure 5: Morphological chart showing solution principals to desired functions.
Appendix B

Retrofitting

Pros

- Low changeover costs
- Easy installation
- Lessens failure rate due to contaminants

Cons

- System is still prone to hydraulic failure

Price/6 ft. module

$100

Figure 6

Cam/Spring

Pros

- No complex components
- Easily manufactured

Cons

- Contaminants can damage/pit cam
- Cam size may be too large for current base design
- Changeover could prove to be costly

Price/6 ft. module

Motor/drive-$1000
Machining-$2000
Materials-$500
Total-$3500

Figure 7
Crank/Spring

Pros

- Modularity- Many pushrod actuations for each motor
- Simple linkage design
- Less components at risk for failure
- Contaminants place less of a roll in failure

Cons

- Alignment issues-time
- Failure could require an entire section to be replaced

Price/6 ft. module

Motor/drive-$1000
Machining-$2000
Materials-$500
Total-$3500

---

Passive System

Pros

- System is only active during clamp release
- Quick and inexpensive changeover

Cons

- System still prone to hydraulic failure
- Contaminants can still cause damage
- Some redesign of upper clamping fixture may be required
- Larger cylinder or greater pressure would be needed to compress springs

Price/6 ft. module

Springs-$200
Machining-$200
Total-$400
**Ball nut/Spring design**

**Pros**
- No torque on motor while clamped
- Possibility of no hydraulics
- Quick changeover
- Precise force control

**Cons**
- Highly complex components
- Contaminants can damage ball nut

**Price/6 ft. module**

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft</td>
<td>$1200</td>
</tr>
<tr>
<td>Motor</td>
<td>$5000</td>
</tr>
<tr>
<td>Ball nut</td>
<td>$4800</td>
</tr>
<tr>
<td>Springs</td>
<td>$100</td>
</tr>
<tr>
<td>Machining</td>
<td>$5000</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>$16000</strong></td>
</tr>
</tbody>
</table>

**Integrated linear actuator design**

**Pros**
- Fully enclosed to survive hostile environment
- AC or DC power
- Mounting flange built to customer specs
- Force and position sensing capable

**Cons**
- Cost
- Require electrical upgrade
- Require software

**Price/6 ft. module**

$72000
Cylinder Quantity Reduction

Pros

- Reduction in Cylinders/hoses-less opportunity for failure
- Ability to invert cylinder-preventing contamination in seals
- Maintain current base as well as ability to adjust for different clamps

Cons

- Still possibility of hydraulic failure
- Possibility of problem with side loading if one pushrod is inactive or in different orientation

Price/6 ft. module

Material-$500
Cylinder-$2400
Machining-500
Total-$3400

Figure 12
Appendix C

Figure 20

Figure 21

Figure 22
**Cost per 2 clamps**

- Rocker Arm and Shaft Assembly $510.00
- Pushrods $24.00
- Springs (2 each) $115.00
- Hardware (bolts/pins/chain) $40.00
- Spring Housing $80.00
- Labor $250.00
- **Total** $1019.00
- (Costs in bulk would likely be considerably less)

---

**Figure 27**

**Figure 28**
Force 1

\[ y = 429.46x - 17.039 \]

\[ R^2 = 0.99927 \]

Figure 30
### DESIGN FAILURE MODE AND EFFECT ANALYSIS (DFMEA)

<table>
<thead>
<tr>
<th>Item/Function</th>
<th>Potential Failure Mode(s)</th>
<th>Potential Effect(s) of Failure</th>
<th>Potential Cause(s) of Failure</th>
<th>Current Design Controls</th>
<th>Proposed Improvement</th>
<th>Recommended Actions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Connecting mechanism to clamps for the duration of the milling process</td>
<td>See components</td>
<td>See components</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bolts</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Springs</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Used to connect chain to hydraulic cylinder</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chain/Cylinder Adapter</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>lockout mechanism</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Used to allow compressed springs to be pinned</td>
<td>See components</td>
<td>See components</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Provide linear motion to a specified point on the rocker</td>
<td>See components</td>
<td>See components</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Linear motion to tangent point on central cam</td>
<td>See components</td>
<td>See components</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Components)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Internally hydraulic cylinder to central cam for the purpose of opening the clamp</td>
<td>Breakage of chain</td>
<td>Clamps could not be opened with cylinder</td>
<td>3 Yielding of pins (shear)</td>
<td>2 Material Analysis FOS=5</td>
<td>16 Tensile test of chain</td>
<td>None</td>
</tr>
<tr>
<td>Fracture</td>
<td>Clamps would remain in a closed clamp position</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fracture of keyseats necessary to open clamp</td>
<td>Load</td>
<td>Clamps could not be opened with cylinder</td>
<td>1 Yielding from buckling</td>
<td>2 Material Analysis FOS=2.1</td>
<td>4 No change</td>
<td>None</td>
</tr>
<tr>
<td>Fracture of chain link necessary to open clamp</td>
<td>Load</td>
<td>Clamps would remain in a closed clamp position</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Load bearing</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rocker Arms/Hubs</td>
<td>Yielding from buckling</td>
<td>Thread failure</td>
<td>1 High stress</td>
<td>2 Material Analysis</td>
<td>2.9 Yielding of material near pin hole</td>
<td>None</td>
</tr>
<tr>
<td>Loads</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Load on keyseats</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Common Shaft</td>
<td>Plastic deformation</td>
<td>Fracture</td>
<td>2 Material Analysis FOS=2</td>
<td>4 Load testing of threads</td>
<td>6 Stress to reduce tension</td>
<td>None</td>
</tr>
<tr>
<td>Common Shaft/Cam &amp; Rocker (Sub-System)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Internally</td>
<td>Fracture</td>
<td>Load on keyseats</td>
<td>1 Material Analysis FOS=2.5</td>
<td>4 Load testing of threads</td>
<td>6 Stress to reduce tension</td>
<td>None</td>
</tr>
<tr>
<td>Internally</td>
<td>Fracture</td>
<td>Load on keyseats</td>
<td>1 Material Analysis FOS=2.5</td>
<td>4 Load testing of threads</td>
<td>6 Stress to reduce tension</td>
<td>None</td>
</tr>
<tr>
<td>Internally</td>
<td>Fracture</td>
<td>Load on keyseats</td>
<td>1 Material Analysis FOS=2.5</td>
<td>4 Load testing of threads</td>
<td>6 Stress to reduce tension</td>
<td>None</td>
</tr>
</tbody>
</table>

### Appendix

**Figure 32**

**Table 1:** Design Failure Mode and Effect Analysis (DFMEA)

- **Yield due to bending and/or shear:** Spars would not be clamped, pushing and springs would default, pushrods could potentially be forced off of rocker.
- **Shear and bending stress:** Material analysis FOS=1.4
- **6150 lbf:** Grade 8 bolts used
- **63:** Change design chip shielding mechanism
- **Operational testing:** Grade 8 bolts used
- **ALGOR FEA testing:** Grade 8 bolts used
- **Full load operational testing:** Grade 8 bolts used
- **Tensile test of chain:** Grade 8 bolts used
- **Operational testing:** Grade 8 bolts used