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Design, analysis and testing of a radial-axial hybrid active force compliant tool head for deburring turbine engine parts

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Design, Analysis and Testing of a Radial-Axial Hybrid Active Force Compliant Tool Head for Deburring Turbine Engine Components

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A thesis
presented to Ryerson University
in partial fulfillment of the requirements for the
degree of Master of Applied Science
in the Program of
Aerospace Engineering

Toronto, Ontario, Canada 2011

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ABSTRACT

Design, Analysis and Testing of a Radial-Axial Hybrid Active Force Compliant Tool Head for Deburring Turbine Engine Parts

A thesis of the degree of Master of Applied Science in Aerospace Engineering, 2011

By
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In this thesis, a new concept and design is presented for a tool with the purpose of deburring gas turbine engine parts. This new concept utilizes both axial and radial active force compliance to accomplish the burr removal in a more robust manner. The axial and radial components are integrated in a manner that allows them to be decoupled, reducing the complexity of the system.

The tool is designed around a pneumatic spindle that is affixed to pneumatic axial actuators. The axial motion system is then affixed to the radial system which makes use of a 2 axis rotary gimbal, acting as a 2-D pivot. Sensors for the axial and radial components of the tool are independent of each other. Axial sensing is accomplished using a commercial string-potentiometer and radial sensing is accomplished using magnets and magnetic field sensors.

Burr formation and methods of removal are discussed. Different deburring tool designs available commercially and through literature are then explored. The design process of selecting axial and radial actuation and sensing and integrating them together while keeping the systems decoupled is outlined. Modeling of the tool is then developed and a simulation of the tool is presented to illustrate the deburring mechanics of the decoupled axial and radial components. Experimentation to determine the stiffness qualities of the tool as well as calibration of the sensors are presented and used within the simulation.

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NOMENTCLATURE

a	Major axis of Hertzian contact ellipse
AFC	Active Force Compliance
ADT	Adaptive Deburring Tool
b	Minor axis of Hertzian contact ellipse
b_l	Lower pivot length
b_u	Upper pivot length
c	Hertzian Contact Ellipsoid third axis
c_1, c_2	Integration constants
C_{Cut}	General Damping coefficient due to cutting
C_{PRA}	Damping co-efficient of the PRA
C_{z-Air}	Damping co-efficient of FESTO cylinders
C_{z-Cut}	Damping co-efficient of cutting in the axial direction
CADET	Chamfering And Deburring End of arm Tool
d_{gk}	Diameter of any given grain k
f_k	Cutting force of any given grain k
F	Total Hertzian contact force applied
F_c	Total abrasive cutting force
F_{C-PRA}	Damping force of PRA
F_{K-PRA}	Stiffness force of PRA
g_i	Grain number i
h_1	Largest grain protrusion
h_k	Protrusion of grain k
HB	Brinell Hardness

I	Moment of Inertia
K_{PRA}	Stiffness of the PRA
K_z	Stiffness of axial components
k	Hertzian contact ellipse ratio
m_z	Mass of moving axial components
M	Hertzian Ratio
M_{Cut}	Moment due to cutting force
M_{C-PRA}	Moment due to PRA damping
M_{K-PRA}	Moment due to PRA stiffness
M_{PIVOT}	Total moment about pivot axis
n	Hertzian Ratio
N	Vector direction in Global Coordinates
PRA	Pneumatic Ring Actuator
P_o	Hertzian contact pressure
P_m	Mean Hertzian contact pressure
P_{seg}	Hertzian contact pressure distribution
Δr_l	Change in lower radial position
r_l'	New lower radial position
r_l	Lower Radial Position
$\dot{r}_l$	Lower Radial Velocity
$\ddot{r}_l$	Lower Radial Acceleration
r_u	Upper Radial Position
$\dot{r}_u$	Upper Radial Velocity
$\ddot{r}_u$	Upper Radial Acceleration
r_{gi}	Radius of grain i

R_t	Tool bit radius
R_1	Radius of Hertzian Contact Disc 1
R_1'	Edge Radius of Hertzian Contact Disc 1
R_2	Radius of Hertzian Contact Disc 2
R_2'	Edge Radius of Hertzian Contact Disc 2
UTRC	United Technologies Research Centre
w_B	Burr width
X_G	Global x axis
Y_G	Global y axis
Z_G	Global z axis
Z	Depth from surface of Hertzian disc
z	Axial position
$\dot{z}$	Axial velocity
$\ddot{z}$	Axial acceleration
α	Euler angle about X_G
β	Euler angle about y'
θ	Pivot axis angular position
$\dot{\theta}$	Pivot axis angular velocity
$\ddot{\theta}$	Pivot axis angular acceleration
ϕ	Hertzian contact angle
ψ	Angle of action plane wrt X_G
ω	Spindle speed

CHAPTER 1 - INTRODUCTION

1.1 BURR FORMATION

In order to understand deburring, it is first necessary to understand what a burr is and how a burr is formed. Once this is understood, all measures possible within the manufacturing process can be taken in order to reduce the size and recurrence of these burrs. This will minimize the requirements for deburring, thereby cutting costs in production and increasing efficiency. Focus should always be given to eliminating the problem of the burr at the source before attempting to provide a fix for the after effects. Once these avenues have been explored, deburring methods should be examined with great care, as many different methods exist and work well for different scenarios.

Work on the understanding and reduction of burrs began in the 1970s. F. Shafer, K. Nakayama and M. Arai, L.K. Gillespie and P.T. Blotter (Gillespie, 1999) were the pioneers of burr research, modelling and theory. They set the foundation for a growing body of research aimed at understanding, reducing and removing burrs from work pieces. These researchers categorized burrs according to their geometries (Schäfer, 1978), cause of formation (Gillespie, 1996) and cutting edges involved and directions of formation (Nakayama, 1987).

Burrs are generally features of a work piece that lie outside the desired boundaries of the part, set by its geometry, i.e. the rough and jagged edges left after a piece has been cut. Burrs are an unavoidable consequence of the loss of support at the edges of a work piece in material removal operations. Unfortunately, after this introductory statement, the problem gets increasingly complex. An understanding of what burrs are and how they are formed must be understood before their remedy can be considered.

1.1.1 Burr Geometry

Looking at burr geometry is useful in understanding its formation. Figure 1-1 contains a typical burr. Its generation can be the product of a variety of different machining operations. There are four main characteristics of a burr. The burr height dictates the overall length of the burr. The burr root thickness is the measure of the depth within the part that the deformation penetrates, perpendicular to the face opposing the cutting plane. The burr thickness is a measure of the thickness of the burr and the burr radius is the radius of the curve that the burr forms with the face opposing the cutting plane.

Figure 1-1 – Burr Geometries (Min, 2004)

The values of these geometries are dependent upon many things in the cutting process including the material strengths, shapes of the contacting components, the cutting speeds, feeds and cutter parameters. The size of the burr is proportional to the cutting edge radius and the applied pressure.

1.1.2 Burr Formation Mechanisms

Burrs can be formed in many different ways from different cutting operations. There are four main burr types that will be explored next.

Poisson Burrs

Poisson Burrs, as shown in Figure 1-2 are formed from the deformation of material during cutting. The material is deformed in a lateral direction forming extensions along the cutting plane and making it impossible for the cutter to remove these pieces.

Figure 1-2 - Poisson Burr Formation during turning (Gillespie, 1999)

Entrance Burrs

Entrance burrs, as shown in Figure 1-3 are burrs formed by plastic deformation as a tool enters a work piece. This is due to the material that the tool initially displaces as it enters the work piece before shearing has fully initiated. Strain hardening plays an important role in the formation of both Poisson burrs and Entrance burrs.

Figure 1-3 - Entrance Burrs forming during a cutting operation (Gillespie, 1999)

Rollover/ Exit Burrs

Rollover burrs are generated when a cutting tool is exiting a piece and it is easier for the piece to bend and deform than it is to cut or fracture the edge. This ease of deformation occurs because at the edge of the work piece, no more material is available to provide the resistant shear force that facilitates the removal of material through the rest of the cut. Rollover burrs are very common and are the burr modelled in Figure 1-1

Tear Burrs

Tear burrs, as shown in Figure 1-4 occur when chips are torn from the work piece instead of sheared off in the proper manner. Tear burrs are smaller than other burrs and generally resemble small jagged edges where the material separated according to its grain structure instead of the cutting tool. Tear burrs are common in the stamping process as well as when side milling a part.

Figure 1-4 - Tear burrs caused by a chip torn from the part (Gillespie, 1999)

Other Unwanted Edge Projections

Unwanted edge projections are those that occur not from the cutting process. These can include recast material, cut-off projections, flash, cratered edging, nicks, dings, scratches and other accidental damage. Recast material is formed when molten metal from cutting, electrical discharge machining or some other process gathers on a work surface or edge. Cut-off projections are the protrusions formed when a piece of bar stock is cut, typically with a band saw or on the lathe. Flash is created when casting a material. Excess material gathers in the seam created by the two separate parts of the mould. If proper pressure and alignment exists, this can be minimized or eliminated. Cratered edges occur when the

piece being machined is brittle and breaks easily. A cratered edge is essentially the opposite of a burr. Excess material is inadvertently removed. Although the part does not exceed the geometric tolerances, the surface will still have to be smoothed out according to specification (in most cases, if the material removed is greater than the allowable minimum tolerance, the part will have to be scrapped or weld repaired). Dings, scratches or other accidental damage must be dealt with on a case by case basis. Some damage, like a scratch, can be buffed or polished out. An excellent illustration of burr formation can be seen in Figure 1-5, showing the progression of the burr formation as the cutter passes through the edge of the work piece.

Figure 1-5 - a.) Rollover Burr formed by orthogonal cutting of copper b.) Negative Burr created by the same cutting of more brittle Aluminum (Gillespie, 1999)

1.2 DEFINING THE PROBLEM

Once efforts to mitigate burr formation have been attempted, it is time to address the issue of burr removal. It is not possible to completely eliminate burr formation and even if this was possible, many parts require a finishing edge chamfer for safety and ease of assembly. Deburring and edge finishing are unavoidable processes.

When deburring is done by manual operators, there is no need to fully understand and define the difficulties encountered when removing burrs and smoothing edges because by nature, humans are highly adaptable to the variety of complex geometries that are encountered. A systematic way to approach the problem is difficult because of the variation in methods from operator to operator.

In the case of automation, a systematic approach is required. Automation relies heavily on measurement and consistency. Consequently it is necessary to quantify the difficulties encountered in deburring and edge finishing. Issues of why a feature is difficult to deburr and what characteristics cause this difficulty are very important to developing an automated process to produce results that are more consistent than manual processes. These processes must also overcome the shortcomings of automation such as a lack of adaptability.

Eight criteria have recently been established that effectively describe the difficulties encountered in burr removal (Petz, 2010). These criteria are as follows; the size of the feature to be deburred, the tolerance applied to that feature, its proximity to other features, its complexity and machinability of the material, accessibility of the feature, confinement of the feature and the severity of the burr on the feature edge.

In designing a tool to automate precision deburring, it is wise to consider these criteria. If a tool can be developed that addresses all of the criteria that cause a burr to be difficult to remove, that tool will be very useful in automating the process, decreasing process time, increasing accuracy and ultimately saving industry money. Even if a tool may only be able to address a few or even one of these problems, significant savings can still be made.

This thesis addresses issues that pertain to the deburring of turbine discs for turbine engines. Turbine disc, like the one illustrated in Figure 1-6 are employed in the hot section of the engine and are where the turbine blades that harness the forces of the hot expanding gasses to drive the engine are mounted. The incredible heat from the combustion of jet fuel coupled with the centripetal forces imparted on the disc from the high rotational speeds create an environment high in stress, temperature, corrosion and

fatigue. Turbine discs are subject to some of the most inhospitable environments known to engineering. Because of this, the materials used are incredibly tough and difficult to form, the geometries must be complex and the tolerances are among the highest applied in any macro scaled machining industry. Materials used in manufacturing turbine discs are nickel super alloys like Waspaloy and Inconel.

Figure 1-6 - CAD Rendering of a High Pressure Section Turbine Disc

Adding to the imperative of this issue is that deburring is, after all, a parts finishing process. This implies that the part has passed through all other stages of manufacturing. The part has come as a cast ingot, has passed through all of the lathe and mill work, the honing and broaching, any chemical treatments and all quality inspections after each process. All of these processes are automated to the utmost degree of accuracy, repeatability and quality. By the time the disc reaches the deburring department, the value of the part exceeds that of an entire automobile. At this stage a human operator deburrs the part manually with a file and a pneumatic rotary tool.

Currently, tremendous resources are expended in the manual deburring of turbine discs and other turbine engine components. The total cost of deburring and parts finishing can be 10-20% of the total cost of the part (Tomastik, 1997). Highly skilled technicians meticulously scrape and sand every edge in order to chamfer and smooth them to an acceptable edge profile using tools resembling dentistry equipment and jeweler tooling. This process is time consuming, holds environmental health and safety issues and is prone to human error. Training new technicians is costly and takes months. The limited number of skilled technicians causes capacity shortages and bottlenecks in the production line.

There are many solutions to the problem of burr removal on turbine discs. These include design alterations to mitigate the creation of burrs, specialized manual tooling available to operators and automated processes both of a mass finishing and a localized nature that can either replace or more likely compliment the manual deburring component.

The nature of burr formation is one that is inconsistent and varies greatly between parts. Variables like material microstructure, resonant frequencies / chatter and tool wear all play factors in the size and shape of burrs that are formed when cutting materials. Burrs can be similar but no two burrs are the same and this causes great problems when attempting to automate the deburring process. By nature, automation is repeatable and consistent and is therefore juxtaposed to the problem of burr removal. The challenge in automating a process of this nature is to add robustness to the automation i.e. to create an automated process that is not only repeatable and consistent but also adaptable to varying geometries in order to produce an acceptable edge. The challenge is illustrated in the diagram in Figure 1-7.

Figure 1-7 - Combining the positive attributes of both manual and automated systems can help solve the deburring problem

1.3 PROBLEM STATEMENT

The perceived solution to the problem of automating the deburring process lies in Active Force Compliance (AFC). A system applies AFC by taking a measurement of current conditions and adjusting the cutting force applied to the edge based on that measurement. A human operator uses their sense of touch and sight to position the cutting tool in the appropriate position and then applies a cutting force to the work piece. The operator can then sense the force that they are applying as well as the displacement that the tool incurs due to the burr. The operator varies the applied force based on what has been sensed in what is in essence, an AFC system. In creating a robust automation process, specialized tooling must be developed that will allow the system to mimic the senses of the operator that allow the operator to perform so robustly while still maintaining the level of control that makes an automated process effective. As mentioned above, there are effectively two manners in which the operator collects feedback information about the cutting conditions. The operator uses the resistance encountered in the muscles in their hand and arm to feel the cutting force they are applying to the work piece and those same muscles are used to sense the relative displacement of the tool. The information is obviously not quantified but is used intuitively by the operator to adjust the level of force they apply and produce an acceptable edge profile.

The operator employs both a force measurement and a displacement measurement method in order to affect the desired tool control. Currently, various 6-axis force measuring devices exist that can and are being implemented for advances in deburring technology. However, there is no device available that measures the minute displacements caused by the burr and adjusts the cutting forces according to those measurements. In light of this, the purpose of this thesis is to develop, build and test a deburring tool that will measure displacement and augment a deburring force output in both the radial and axial direction. The standards set within the design parameters will be taken from the perspective of deburring gas turbine engine parts like the one in Figure 1-6.

CHAPTER 2 - LITERATURE SURVEY

2.1 METHODS OF DIRECT BURR REMOVAL

Methods of direct burr removal refer to those used by manual operators and have been thoroughly documented in the aptly titled “Hand Deburring” by LaRoux Gillespie (Gillespie, 2003). Gillespie documents with great detail the methods and tools used in the hand deburring industry as well as a number of empirical metrics with which to measure a hand deburring department’s performance. Asada and Asari (1991) developed a method of acquiring/measuring the compliance that a manual operator applies within their arm and hand when deburring work pieces. Liu and Asada (1991) developed an adaptive control system for robotic deburring based on the measurement of motion and compliance of a deburring operator as they finished various pieces.

Various robotic control theories are fascinating and show that looking to manual deburring can provide valuable insight into improved methods of automated deburring. The tools used in manual deburring can be used for automated deburring work with the proper modifications and will be incorporated into the design of the deburring tool. These tools can be sorted into two types and two methods of implementation. Abrasives (silicon dioxide, aluminum oxide) and hard tools (metal blades) can be used. These tools can be used either as the tool head of a rotary tool or as a stationary tool.

Table 2-1 – Methods of Manual Deburring

	Stationary	Rotary/Mechanized Tool
Abrasive	<ul style="list-style-type: none">• Sand Paper• Sanding Blocks	<ul style="list-style-type: none">• Grinding Stones• Sanding Discs• Abrasive laden vinyl• Butterfly flap wheels• NAF Brushes• Belt Sanding
Hard Tool	<ul style="list-style-type: none">• Files• Scrapers	<ul style="list-style-type: none">• Rotary files• Cutters

Looking into Table 2-1, any mechanized tooling can be adapted for automation however for the specific type of detailed deburring work that is the subject of this thesis a more specific type becomes practical;

only those that have a more consistent geometry, suited for detail work. They are given Table 2-2 with images of an example of each type:

Table 2-2 – Deburring tool pieces

- Grinding stones

- Sanding Discs

- Rotary files

- Cutters

It is understood that grinding stones and sanding discs wear down and diameters change based in the state of wear, however this can be measured, predicted and compensated for.

For the basis of the design of this tool, it will be assumed that the tool bit will be one that will maintain its geometry and be relatively compact, that being a file or grinding stone that basis its material removal methods on abrasive material removal.

2.2 PRINCIPLES OF COMPLIANT TOOL HEADS

2.2.1 NON-COMPLIANT / HARD TOOLING

Non-compliant tooling is the simple case of a robot with a hard-tool deburring end effector (a cutting or sanding disc or drum, rosette or mounted point that does not deform when a force is applied to it, mounted on a spindle). The robot follows a distinct tool path in order to deburr the part. There is no feedback between the robot and controller to ensure that the burrs are being removed effectively, the robot simply moves through a predetermined tool path and upon completion, returns to its original position. This approximates a CNC type machining method. Robots are far cheaper to implement but also much less accurate.

Problems arise even if the robot is properly calibrated and the tool path is exact and repeatable. The burrs that the robot will encounter are not the same for each part. If a simple, uniform tool path is implemented on the non-uniform surface, the results will not be repeatable from piece to piece and may not meet specifications, requiring further manual deburring. Hard tool deburring by robotic means is widely used but has limitations. It requires a large amount of development in order to calibrate the process to produce an acceptable result and the process is not robust. A change in the parameters will alter the finished product, causing need for recalibration.

2.2.2 COMPLIANT TOOLING

Compliant tooling is a way to deal with the issues that arise from non-compliant / hard tooling.

Compliant tooling requires a tool path that informs the robot of the location of the edge. The robot moves the tool along the edge like it would with a hard tool end effector, however the tool has the ability to alter its performance to match the surface characteristics and the burrs that are encountered in order to produce a smooth and uniform edge. There are two types of compliance tooling, passive compliance and active compliance.

There are also two different facets of compliance; the field of compliance and the manner in which the parameters of the cut are controlled. Fields of compliance include Axial and Radial. In the axial field, the compliance and cutting force is directed along the axis of the tool. In radial compliance, the compliance is directed perpendicular to the axis of the tool. This can be seen in the Figure 2-1.

Figure 2-1 - Axial and Radial Compliance (ATI Industrial Automation)

Passive Compliance

Passive compliant tools alter their shape and the force they apply to a burr in an uncontrolled manner. Generally the tool is attached to a spring that applies a force in the axial or radial direction. The stiffness of this spring can be controlled, however no feedback is present. If there is a burr, the tool will be forced up, displacing the tool and deforming the spring. A deformation will cause an increased cutting force applied back onto the burr, aiding in its removal. The main principle of passive compliance is based on Hooke's Law. Another good example of a passive compliant tool is a nylon brush or belt which works on the same principle but covers more area than the single point hard tool and uses the stiffness of its bristles as the spring stiffness.

Active Compliance

Active compliance tools have the ability to control the various parameters that define the material removal on the work-piece. The alterations are determined through some form of burr measurement. Since burrs are randomized, an active compliance tool must somehow alter the parameters of its

function in order to produce a consistent edge or surface that conforms to the ideal edge and tool path (existing in virtual space).

There are several types of active compliance. Parameters that can be altered when deburring a surface include the force applied to the surface, the feed rate that the tool is moving at and the spindle speed of the tool. Furthermore, the force being applied to the surface can come directly from the robot (coupled) or from the tool (uncoupled). There are also different ways to sense a burr on a surface and ensure its removal. Methods include force sensors, cameras, and position sensors to name a few.

Active coupled force compliance requires a robot with a force sensing device to measure the forces and moments that are created when the tool interacts with the work piece. If the robot experiences a burr, the forces change. The robot then directly alters the amount of force that it is applying to the work piece until the force that it senses equals that of the specified force. This method couples the position of the tool with the force applied as both are handled by the robot simultaneously. This method does not directly measure burr height and hence the quality of the surface cannot be ensured. It is therefore more applicable for polishing, where a constant force is more important than a constant or consistent geometry, as this geometry will already have been brought into compliance through other finishing techniques.

Decoupled force compliance uses separate systems to provide the overall positioning and deburring tasks. A robot or CNC machine is used to provide the tool path for the deburring tool and the deburring tool is used to provide any compliance necessary to perform the material removal. This system allows for greater versatility. When relying on the robot to provide the cutting force, very specific characteristics about its performance must be known and continuously monitored. By removing this requirement and placing it within an end effector that can be switched out to another positioning system, this complexity is removed. In a decoupled system the robot's only function is to maintain the proper tool path.

2.3 EXISTING COMPLIANT TOOLHEADS

There are many existing compliant tool heads both in industry and in research. The focus of this section will be on tool heads that are directly relevant to the design of this deburring tool.

First is the Adaptive Deburring Tool (ADT) built by TriKinetics Inc of Waltham MA (UTRC, 1992). This tool was an active force compliant tool used in a study by United Technology Research Center in partnership

with Pratt & Whitney in an attempt to produce deburring and edge chamfering of an acceptable quality for turbine engine parts . Although the tool failed to produce the appropriate edge quality, its design is worth investigating.

Figure 2-2 - Adaptive Deburring Tool by TriKinetics (UTRC, 1992)

As can be seen in Figure 2-2 above, the tool has a number of features worth noting. Its actuation is produced by two mechanical motors operating on a drive screw, it has a force sensor for force feedback of the cutter and it has a 2-axis gimbal to allow for smooth pivoting that translates approximately into planar motion normal to the tool length. This tool did not meet the design criteria in simulation trials due to the bandwidth of the tool being too great and its reaction time too slow (Pratt & Whitney, 1996).

The United Technologies Research Center also produced a design called CADET (Chamfering And Deburring End-of-arm Tool) from the same study (Pratt & Whitney, 1996). In the same simulation as the ADT tool above, it passed the requirements for chamfer edge conditions and so development of this tool progressed (this tool was also Active Force Compliant). The CADET tool utilizes a two axis planar direct drive voice coil actuator, a spider/universal joint linkage that will transfer the actuated motion towards the cutter tip with a small enough rotational angle (5 deg) and vertical displacement (0.37mm) that motion of the cutter can be considered planar. The CADET is shown below in Figure 2-3 and Figure 2-4.

Figure 2-3 - CADET tool developed at UTRC (Pratt & Whitney, UTRC, 1996)

Figure 2-4 - CADET Schematic (Pratt & Whitney, 1996)

The CADET has similar attributes to the ADT however there are some marked improvements. The CADET also has a two axis gimbal however offset in this case. The CADET has four button load sensors for force feedback on the cutter through the Force Transducer (labeled in Figure 2-4). The main attribute of the CADET that allows a greater accuracy, swifter response and lower bandwidth are the voice coil actuators that created the motion for cutting compliance (Pratt & Whitney, 1996).

ATI Industrial Automation has two deburring tools commercially available. Both mount to a robot and are passive force complaint tools. The first is the radially compliant “Flexdebur” (ATI Industrial Automation, 2009). This tool, seen in Figure 2-5 uses compressed air to drive both the spindle and the compliance of the tool.

Figure 2-5 - ATI's Flexdebur, a passive radially compliant tool

Figure 2-6 - Flexdeburr Assembly, Ring Actuator highlight (ATI Industrial Automation, 2009)

Inside this tool, there is what is known as a Ring Cylinder Assembly. This assembly is a series of pneumatic pistons arranged radially about the center of the ring. The pistons fill with air to provide a level of compliance related to the air pressure from the air supply. The ring cylinder assembly can be seen in the assembly drawing of the Flexdeburr (Figure 2-6).

Referring to Figure 2-6 there are some common themes present. Firstly is the presence of a 2-axis rotational joint. In this case it is in the form of a grooved ball joint instead of a gimbal. As well, an actuator is used here to control the movement of the cutter radially, which will translate into approximate planar motion. Because this tool is passive compliant, any feedback is absent. The effective stiffness of the tool is adjusted by controlling the pneumatic pressure input.

The second ATI compliant deburring tool is the Speedeburr, as shown in Figure 1-7. This tool is a passive force compliant tool with motion only in the axial direction. The tool's design is relatively simple. It is composed of a pneumatic spindle set in a simple spring piston cylinder. The pneumatic input pressure determines the level of compliance of the system (ATI Industrial Automation, 2009).

Each of these tools is mountable to a robot that performs a specific cutting tool path, these tools are generally in use to remove flash around metal castings and do not provide the precision required for the task of deburring turbine engine components.

Figure 2-7 – ATI's Speedeburr, a passive axially compliant tool

The final tool that must be examined is the tool developed at Ryerson University by Liao and Xi (Liao, 2008). This tool is the precursor to this thesis and so it holds valuable insight. This tool is an axially compliant tool mounted on a tripod robot. It is pneumatically driven.

As can be seen in Figure 2-8 and Figure 2-9, this tool is composed of an aluminum cylinder in which grooves are machined for three FESTO pneumatic actuators. These actuators are essentially spring damper systems that contain a spring and an air piston. When air flows into the piston, it extends to a length dependent on the pressure of the air flow. The movable component of the cylinder is attached to a moving mount that holds the rotary air spindle in place with set screws, effectively forming a linear actuator that moves the spindle tool piece, in this case a grinding stone bit, up and down along the axis of the tool. The air spindle is stabilized by a linear bearing within the main aluminum cylinder. As air is supplied to both the spindle and the linear actuator, the spindle spins at a prescribed RPM and the actuator moves to an operating location and provides a level of stiffness/compliance based on the pressure and air flow provided from the air supply. The controller determines the level of compliance based upon the readings from an extensometer that are processed through a controller.

Figure 2-8 - Active Compliant Pneumatic Axial Tool

Figure 2-9 - CAD Model of compliant tool head (Liao, 2008)

The valve used to control the air supply for the actuation of the tool is a proportional directional control valve with a response frequency of 100Hz. This provides adequate response time for this tool. The tool head control is best represented in Figure 2-10.

Figure 2-10 - Schematics of the tool head control system (Liao, 2008)

There are many important features of this tool that will be repeated for the design of the hybrid tool. This tool was designed and constructed using materials that are readily available and some of the components will be transferred to the new tool for cost savings. With those components, design features will follow.

The pneumatic spindle will be transferred to the tool and similar FESTO cylinders will be employed. Similar if not the same control valves will also be utilized and an effort to use as many other components will be made in order to save costs.

It is important, however to evaluate the design based on its merits to ensure that positive aspects are carried on and negative ones are not and to avoid basing the design solely on what has been previously done. In this case, the FESTO cylinders work well. They are durable and allow for even actuation and an acceptable response time in the modeling (Liao, 2008). Similar can be said for the proportional directional control valve. These systems can be mimicked. The tripod robot that was used to provide position control is not desired in this case. Tripod robots are not common in industry and other, more commercially available options exist that will allow for an easier experimental implementation. Other design similarities will arise and other items will be changed due to necessity as will be seen later.

2.4 DESIGN STRATEGIES

In the previous section a number of deburring tools were reviewed. Design features of these tools will now be examined in order to organize them in a way that can be used when considering the new tool's features. First, the axial actuation of applicable tools will be examined, radial actuation will follow, structure and movement and finally sensing.

Tools that had axial movement were the ATI Speedeburr and Ryerson University's active force compliant polisher/deburrer. The ATI Speedeburr uses a large spring damper pneumatic piston to provide compliance while deburring. The Ryerson tool uses a series of spring damper pneumatic pistons. In each of these cases, this set up works well.

Radial actuation was seen in the ADT by TriKinetics made for UTRC, the CADET by UTRC and ATI's FlexDeburr. The ADT had actuation provided by two motors operating a drive screw. This tool proved to be too slow and cumbersome to provide appropriate actuation for the needs of the research team. The UTRC team decided instead to use voice coil actuators in two planar directions set on bearings in order to actuate the radial movement on the CADET tool. This method proved successful for a task similar to that presented in this thesis. The ATI tool used a series of pneumatic spring pistons, arranged radially in what they referred to as a Ring Actuator Assembly (Figure 2-8). This ring actuator allowed for passive compliance in any radial direction due to the distributed force provided by the series of pistons. This product is successful in its applications of providing passive radial compliance and has a more simplistic implementation than the voice coil actuators. However the response time is presumably less. Specific details for this are not available as the FlexDeburr does not involve any feedback.

There are many common themes with structure and movement between the tools. The ADT had a two-axis gimbal, allowing the tool to pivot and the tool piece to effectively traverse in a working plane normal to the axis of the tool. The CADET had two offset axes providing the equivalent to a two axis gimbal. This was somewhat more complex to allow for the type of sensing that was chosen. The ATI FlexDeburr used a grooved ball joint to allow for the same two axis rotation. The groove within the ball joint prevented the tool from rotating about its axis. If this groove was not present, the tool would be unconstrained and unable to provide the torque and cutting force required to remove material.

In sensing and feedback, the ATI tools are not applicable as they are passive compliant tools. The ADT tool used a force sensing mount, ie a 6-Axis force transducer, similar to one available, aptly, through ATI Automation (ATI Automation, 2010). This type of sensor is able to measure forces and moments in six

axes. No displacement sensors were utilized. The CADET tool used button load cells placed in a way that the radial forces were measurable (see Figure 2-4). The Ryerson tool used an extensometer to measure the displacement, not the force, resulting from deburring operations.

Table 2-3 is a summary of the attributes of each tool as previously discussed:

Table 2-3 - Summary of deburring tool design features

	ADT	CADET	FlexDebur	Speedeburr	Ryerson Tool
Axial Actuation	N/A	N/A	N/A	Pneumatic	Pneumatic
Radial Actuation	2 Drive Screws	Voice Coil Actuators	Air Ring Piston	N/A	N/A
Movement	2 Axis Gimbal	Offset 2 Axis Gimbal	Grooved Ball Joint	Air Piston	Air Piston
Sensing	6-Axis Force Transducer	Button Load Cells	N/A	N/A	Extensometer

Reviewing Table 2-3, some common themes are identified. Movement of these tools is provided by gimbal axes for radial tools and axial tools use an air driven spring piston method to provide movement. Pneumatic actuation is a popular choice for passive compliant systems. The tool developed at Ryerson showed it can also be used for active compliance (Liao, 2008). Other options for actuating the radial components are available and will be considered in the next section.

CHAPTER 3 – TOOL DESIGN

In order to design the tool, the highest level requirements must be examined. These requirements are fundamentally simple and grow more complex as the design evolves to more specific systems and sub-systems and their accompanying requirements. In the most basic concept, there is a burr on a work piece and it is desired that the burr be removed, creating a smooth edge.

In setting out to design a tool to meet the high level requirements, the design will address the requirements placed on the tool itself and the interactions between the tool and the part that will produce the desired smooth edge. In this tool, a requirement is that it be compliant in both the axial and radial directions to allow for more robustness in the deburring process. This requirement can cause the tool to become complex and difficult to manage. In order to eliminate the complexities of integrating axial and radial AFC systems, those systems will be decoupled to as great a degree as possible. By decoupling the two systems and limiting their interaction the architecture of the tool as well as the supporting control and sensing systems are greatly simplified. From this starting point, further requirements can be placed on each system.

3.1 SYSTEM REQUIREMENTS AND PARAMETERS

The design parameters for this tool were formed by considering the task of manual deburring and how this process can be improved upon as well as the existing designs and previous work in this area that was discussed in the previous chapter.

Firstly, the tool must be able to work with resources that are readily available in the lab in order to reduce costs. The budget of the tool was to be kept to within \$7000. Several FESTO pneumatic cylinders as well as tubing and a WESPRO pneumatic spindle tool were available and so were chosen as the stock parts to be used. Then, based around these parts, the size of the tool must be minimized. A smaller tool will be more versatile and be able to fit into smaller geometries. The tool must be kept as small as possible.

The tool must have the capabilities to sense displacements, control displacement and provide cutting forces in both the radial and axial directions. Displacement can be controlled through cutting forces and tool stiffness. In accordance with industry standards, a $\pm 0.012''$ edge profile tolerance is applied to the

finished edge (Tomastik, 1997), see Figure 3-1. Because of this, a sensory resolution of $<0.012''$ is required and preferably $<0.004''$ to have a reasonable idea of the quality of the edge.

Figure 3-1 - Edge Profile Tolerance

The size of a burr to be removed is another issue to be considered. Considering the largest burr will dictate the minimum required travel in the axial and radial directions. Because the maximum displacement from the tolerance chosen was $0.024''$, a minimum displacement value of $0.025''$ was chosen. This value allows the full range of motion within the tolerable zone. Any protrusion greater than $0.025''$ will be removed by the tool due to the nature of the deburring setup. In consideration of the calibration and testing, the tool must be able to be mounted in some manner to a CNC machine or robot in order to move the tool along a designated tool path.

To recap, the following design requirements have been imposed:

- Must have both axial and radial controlled displacement
- Must have axial and radial displacement sensing
- Must provide cutting force in axial and radial direction
- Must have a sensing resolution of less than $.012''$, preferably less than $0.004''$
- Must be capable of displacement greater than $0.025''$
- Tool must be able to mount on robot or CNC for position/tool path control
- Tool must be produced and developed on a budget of \$7000

Another consideration that is not considered a design parameter but is a matter of practicality is that components for this tool will obviously have to be produced. The components should be designed in a way that will allow for the easiest and most cost effective manner of manufacturing.

3.2 DESIGN IDEAS

The main design ideas must center on controlling and sensing radial and axial displacement and ensuring that these two systems remain decoupled. Since previous work had already determined that pneumatic cylinders were a good way to control the axial displacement (Liao, 2008), this leaves the radial displacement as well as the axial and radial sensing to be considered (the extensometer used in previous work was deemed too large to meet the versatility requirement).

For radial actuation, many different approaches were considered. These ideas can be broken into two types; linkage actuation and axi-symmetric actuation. The linkage actuation relies on some form of bar linkages to provide planar motion that can control the tool's radial position on the radial plane. The axi-symmetric actuation relies on the principle of a centering force that works against any forces displacing the tool from the center axis.

Figure 3-2 - Radial actuation concepts

As seen in Figure 3-2-a, the linkage actuators form what can be considered two five bar linkages, i.e. two prismatic joints connected by the outer frame and connected at the center joint. In theory, only one of these linkages is necessary however in practice, two would be required for symmetric performance. The manner of actuation of the prismatic joints is a matter of consideration. Drive screw, pneumatic, solenoid, electrical (voice coil) and mechanical/servo actuation are all considered viable possibilities by

which to enact the force required to provide cutting forces to drive the tool towards the center after it has been offset by a burr. An important consideration here is that the mirrored linkages would have to perform in precise tandem with each other and each axis (x and y) requires a separate input channel of control. However this aspect also allows for direct control of the position of the tool, the tool would not have to be centered.

Examining the other option, in Figure 3-2-b, the axi-symmetric actuation works on the principle of a force gradient with respect to displacement from the center. This option has the benefit of minimizing any mechanical linkage to the center of the tool that could cause coupling with the axial AFC system. The further from the center the tool bit is displaced by a protruding burr, the greater the force to center the tool becomes. This concept is applied using pneumatics in the ATI Flexdeburr tool's Ring Actuator Assembly component seen in Figure 2-6. Possible actuation methods for this configuration include magnetic, pneumatic and more exotic materials whose properties are altered by the flow of electric current (much like synthetic electromechanical muscle tissue (Hirai, 2007)). In this configuration, only a single channel would be required, altering the stiffness of the actuator on the basis of the level of displacement sensed. For the current design requirements, this is sufficient. Specific positioning was not made a design parameter. Magnetic means of actuation were immediately ruled out due to complexity, cost and issues that could arise with interference with sensory electronics. Exotic materials that are by nature expensive and hard to come by were obviously not a practical option and so pneumatic actuation was deemed the most logical route at this stage of development.

Examining the FESTO cylinders in their employment as axial actuators from a perspective of axial radial coupling, these cylinders are small and can be designed to function within the radial system in such a way that the two AFC systems will be independent of each other.

The mechanism to facilitate the movement and pivoting of the radial component while accommodating the axial components and avoiding any coupling is a matter that requires more consideration. The previously considered devices used were the 2 axis gimbal, an offset 2 axis gimbal and a grooved ball joint (a type of universal joint). Of the three, the grooved ball joint is mechanically the simplest. Unfortunately such a product is not commercially available. The offset 2-axis gimbal was a necessity of that particular design configuration and shares the same principles as the regular gimbal with an added level of complexity. The concept of a decoupled radial and axial system is illustrated in Figure 3-3. In this simple diagram the axial AFC system is affixed to the pivot rod of the radial system, allowing the axial AFC system to operate independently of the radial system.

Figure 3-3 - Concept of decoupled axial-radial AFC deburring configuration

3.3 TRADE STUDY

In Table 3-1, the methods that were found in the literature have been examined are presented. Considerations of the quality of the option, the availability, the cost and the complexity of its implementation have been considered and the option has been deemed either acceptable or not:

Table 3-1 - Existing Options for design implementation

	Option	Availability	Cost	Quality	Complexity	Acceptable?
Axial Actuation	Pneumatic	ALREADY AQUIRED	NONE	PROVEN	LOW	YES
	Drive Screw	DESIGN / BUILD	MED	SLOW	MED	NO
Radial	Air Ring Piston	PURCHASE	HIGH	HIGH	LOW	NO
	Voice Coil	DESIGN / BUILD	HIGH	HIGH	HIGH	NO
	Drive Screw	DESIGN / BUILD	MED	SLOW	MED	NO
Radial Movement	2 Axis Gimbal	DESIGN / BUILD	LOW	MED	LOW	YES
	Ball Joint	PURCHASE	HIGH	HIGH	LOW	NO
Sensing	Extensometer	ALREADY AQUIRED	NONE	PROVEN	LOW	NO, TOO LARGE

Examining the chart, many of the options for the design have been eliminated. Turn screws are too slow to react appropriately, as was found with the TriKinetics tool, voice coils were deemed too expensive and complex. The air ring piston was a possible option however to acquire one, the entire Speedeburr tool would have to be purchased and disassembled. Likewise with the grooved ball joint. According to correspondence with ATI, the ballpark purchase price of one of these tools was \$3800-\$4400. This price relative to the budget cost makes this option prohibitive. Furthermore, future patents and marketability were taken into account when the decision was made not to go with those options. What is thus evident in this chart is that aspects of the tool do not yet exist “off the shelf” and so will have to be designed from scratch, most notably, the radial sensing and actuation.

Since the axial AFS system already exists and has been proven, the simplest method of design was to design the radial actuation and sensing around the axial AFS while ensuring that the two systems remain decoupled. The overarching tool concept was sketched and then a 3D model was created in CATIA. The tool concept sketch was then formalized into Figure 4-5. The initial tool concept is seen in Figure 3-4:

Figure 3-4 - Initial Tool Concept

Seen in the cutaway model is an outer casing, a 2 axis gimbal fastened to a long shaft. On one end of the shaft is the axial actuators and the deburring tool piece. On the other end is a concept for the radial actuator, set at a distance from the axial system. The idea behind this configuration is to use the gimbals as a 2-D pivot point that will allow for a mechanical advantage when exerting radial force and when sensing displacement. Both force and displacement will be amplified if the upper pivot length is greater than the lower pivot length. Since the method of actuation and sensing are not known at this stage, built in mechanical amplification of each seems advantageous as well as providing the added benefit of ensuring that the two systems are decoupled. After the initial concept was created, the existing hardware was modeled and then the concept was modified to match the geometric limitations that this hardware introduced. This model is considered the Mark I model of the tool and is seen in Figure 3-5.

Figure 3-5 - Mark I Tool Design

3.4 DEVELOPMENT OF THE PRA

Concepts for the radial sensing and actuation were explored earlier. The most attractive concept was that which mimics the Air Piston ring due to its simplicity. In this vane a concept for a “Pneumatic Ring Actuator” or PRA was conceived. The PRA in essence is a volume of liquid or gas configured in a torus shape, contained within an elastic material constrained in such a way that when the inner pressure

increases, the volume and elastic containment material will expand inward, exerting a centering force on the pivot rod. The inner pressure is dictated by the level of offset of the pivot rod, sensed by the radial sensors. In Figure 3-4, a PRA concept that uses four different chambers made of silicone that fills with air, expanding to restrict the movement of the center rod. In the Mark I Tool Design, this concept was changed to a full torus, which would expand based on the dynamic pressure of air flow. The air would enter through four input ports and exit through four exit ports, inputs and outputs alternate around the ring. The flow would be controlled by an electric flow valve and back pressure valve.

Development of a Silicone PRA was done as a separate thesis project (Kroeker, 2010). Ultimately the idea of the Silicone PRA did not work at this design stage due to manufacturability problems and so other materials were considered including surgical tubing and latex however bicycle inner tube was chosen as the most promising candidate due to its low cost and easy availability. The tubing was cut into an appropriate length and then it was stretched around the conduit ring as seen in this diagram:

Figure 3-6 - PRA Bicycle Innertube stretched around Conduit Ring

Many methods of fastening the bottom of the PRA were attempted. Glue, wire and zip ties were all attempted. In the end, a small pipe clamp proved most effective at preventing any air leakage. The best bicycle tube to suit this purpose was found to be the Axiom 26x2.125-2.40" bike inner tube.

Once the inner tube was stretched around the conduit ring, small holes were burned into the sides at the site of the threaded holes. Then threaded pneumatic barbs were fitted in and tightened. The process was tedious and difficult to accomplish without splitting the rubber and having it tear, rendering it useless. After several attempts the process was refined and properly executed. This configuration provides an airtight seal with relatively uniform performance. Once the full assembly of the tool is illustrated (Section 3.6) the exact manner of function of the PRA will become clearer.

3.5 SENSOR DESIGNS

Based on the preliminary design, sensors were required for sensing both the radial displacement of the cap on the pivot rod and the axial displacement of the tool piece. Several ideas were considered on a conceptual level. These included force measurement strips, optical encoders, laser measurement systems, potentiometers and magnetic field sensors.

The field of search was first narrowed by practicality. After searching the market for optical encoders, none were found to exist that were small enough to be applicable to the design. Furthermore, these devices were very expensive. Laser measurement systems were also prohibitively expensive.

This left force measurement strips, potentiometers and magnetic field sensors to be considered. All of these were available and considered economical options. These options were explored more explicitly through conceptual CAD modeling.

3.5.1 RADIAL SENSING

The force strips use a piezoelectric film to generate a voltage from displacements. These strips are made by Measurement Specialties. The strips measure in the range of millivolts per unit micro strain. A conceptual CAD model illustrates how a force strip sensing model would be considered (Figure 3-7). In this case, force (or film strip deformation) and center rod displacement would have to be related through voltage output by proper calibration in order to determine the nature of the sensors. These strips would be arranged at 90 degrees to each other (2 along the x axis and 2 along the y) and be displaced by single points of contact extending from the center rod. When the rod is displaced, two of the strips are deformed, indicating a displacement that could then be determined by the outputs of the strips. The strips are non-saturated, i.e. they do not continuously output voltage and will settle to a zero state after some period of time, regardless of the deformation state.

Figure 3-7 - Force Strip Concept

The Honeywell HMC1501 magnetic field sensors are each composed of four anisotropic magneto-resistant (AMR) sensing elements configured in a Wheatstone bridge circuit. This configuration allows for a varying in impedance dependent on the incidence of magnetic field lines with respect to the flow of current. As the angle at which the field lines impend on the sensor varies, a change in voltage across the bridge can be measured. See Figure 3-8 for an illustration. By fixing small magnets to the top of the pivot rod small movements can be measured through the change in their magnetic field line direction relative to the stationary sensors. This measured voltage remains a function of the field lines and so, unlike the force strips, a continuous output from the sensors is available.

Figure 3-8 - HMC1501: A - Wheatstone Bridge Circuit. B – Application illustration. (Honeywell)

It is because of this continuous output that these sensors were chosen for further study as the radial sensing components. The sensors still had to conform to the design parameters outlined in Section 3.1. According to the literature provided by Honeywell, the sensor had a resolution of up to 0.002". This cited value is twice as good as the 0.004" requirement outlined in the design parameters.

The sensor has a sensing range of $\pm 45^\circ$ and as such, in general use, several are arranged in a line in order to track the position of a magnetic object. In this case, only one in each direction will be necessary as the displacements of the pivot rod are not estimated to exceed 0.100".

For further technical specifications on this device and the sensory electronics, see Appendix B.

Figure 3-9 shows the configuration of the HMS1501 sensors in relation to the magnets mounted on the top end of the pivot rod. These sensors are positioned on a round circuit board to allow the sensors to be in the same plane as the magnet. The four cylindrical spacers seen on the round circuit board allow for an additional board to be mounted on top, which holds the additional electronics. This sensor configuration allows sensing of the radial displacement only and is fully independent of the axial AFC system. A wealth of information about the HMS1501 magnetic sensor can be found through the Honeywell website (Honeywell, 2010).

Figure 3-9 - Configuration of magnetic sensors for radial displacement sensing

3.5.2 AXIAL SENSING

In investigating axial position sensors, the objective was to find the smallest sensor possible that would indicate position to within at least 0.004". No extensometer or optical encoder was available that was deemed of appropriate size. A single product was located and so was procured and incorporated into the design; The Celesco M150 cable extension position transducer (a.k.a. String Potentiometer or "StringPot"). This product was an out of the box "plug and play" that would return an output proportional in scale to the input based on the level of extension of its attached cable.

Figure 3-10 - Celesco M150 "String-Pot" Mounted and Un-mounted

As seen in Figure 3-10, the String-Pot mounts to the side of the spindle housing and the extension cable is connected to the spindle ring to measure the displacement between the casing and the spindle ring, effectively measuring the axial displacement between the cutting bit and the tool. This device was small enough to be incorporated onto the axial AFC system without affecting the radial in any way, again ensuring no coupling.

3.6 FINAL DESIGN

The final design of the tool can be seen in Figure 3-11 in this cutaway illustration:

Figure 3-11 - HFCDT Cutaway Illustration

In Figure 3-11, many of the components mentioned before can now be seen integrated together and examining the configuration, it is clear that the radial system was designed around the axial one, not with it, allowing the benefits that decoupled tool architecture provides. Some components have been left out and are shown in Figure 3-12 for clarity.

Figure 3-12 - Cutting End of the HFCDT for clarity

In creating the final design, the geometric constraints were set by the hardware that was initially provided. The FESTO cylinders for example were of a certain size that to configure only three, evenly spaced, would have meant that there would not have been sufficient room on the circumference of the spindle housing to also accommodate the gimbal axle mounts. This forced a configuration of 4 evenly spaced FESTO cylinders to allow a direct line for the axles. Grappling with this geometric constraint as well as the diameters of the linear bearing and the spindle diameter resulted in the final spindle housing geometry and subsequently the diameter of the tool in general.

The PRA can be seen at the top of the tool in Figure 3-12 and also in Figure 3-6. The PRA has 8 barbed pneumatic ports. Four of these are for air input and four are for exhaust. This is to provide an even distribution of air flow throughout the ring as the flow and pressure changes and the PRA expands or contracts, exerting a centering force about the pivot rod.

Materials used were Aluminum 6061 and SAE-1020 Steel. Aluminum was chosen for its light weight and ease of machinability and the steel was chosen for components where stiffness or wear were considered more important than being light weight.

The idea of using the gimbal as a pivot point in order to magnify the displacement caused by an offset burr so that the radial sensors could, in effect, have a higher resolution was employed in the final design and is a marked distinction between this design and other radial deburring tools investigated in Section 2.3. In this case, the combined distance from the gimbal plane to the radial sensors is 9.25". The distance from the point of contact with the tool piece and the gimbal axis will vary depending on burr size however the nominal distance is 2.25". Any variation in the lower axial distance caused by deburring operation would be trivial in determining the scaling factor however can be accounted for using direct online measurements from the axial sensor. This means that a radial displacement of 0.001" on the tool tip will result in a radial sensor reading of 0.0041". This is important considering that displacement in different directions will be measured differently because the radial displacement is determined through the x and y measurements from each of the magnetic sensors. The smallest reading for any given radial displacement would occur if that displacement was 45 deg from each HMS1502 sensor.

These values are theoretical and do not take into account any interplay between mating parts or the bending of components. These additional considerations are very difficult, complex and time consuming to model and would have required experimental verification. As such, it was decided to forego modeling and determine this relationship directly through experimentation. As will be seen later, this relationship is significant.

Considerations for manufacturing and assembly were accounted for as well. Manufacturing considerations will be explored in Section 5. Considerations for assembly included fitting Spindle Housing A and B together with the linear bearing inside and positioning this assembly within the outer casing. The spindle housing assembly then had to be fastened to the gimbal ring and the gimbal ring to the outer casing. All of these items had to fit while providing clearance for the accessories that are fastened to the main components as well as provide space for pneumatic tubing to supply the air spindle and the FESTO cylinders. Set screws are employed to fasten the air spindle to the spindle ring and the linear bearing within Spindle Housing A. As mentioned before, each component was modeled in CATIA and the entire tool was assembled virtually in order to ensure clearance during assembly and proper fitting.

Design of the End Cap (Figure 3-13) had to allow for enough space for the sensors, electronics, cable connectors and air circulation to cool the electronics. This was accomplished by increasing the height of the end cap, thereby increasing its internal volume.

Figure 3-13 - End Cap with view of internal volume

Another consideration was the location and configuration of the tool mount. The tool mount is a forked piece fastened to a cylinder sized at 1" in order to accommodate a tool chuck, see Figure 3-14. This mount was situated as close as geometries would permit to the gimbal axis in order to avoid as much bending of the tool as possible. The mount was made of steel for increased stiffness. The mount is fastened to the outer casing using two threaded rods and four accompanying nuts. The mounting holes in the casing are also threaded for added stiffness.

Figure 3-14 - Tool Mount

Also seen in Figure 3-14 is the opening for the air spindle pneumatic feed. Special consideration was taken in making sure that after the spindle was inserted the elbow joint for the spindle could be attached using its threaded connection. The opening is elongated so that the spindle can move freely in the axial direction while the spindle housing and the outer casing both remain fixed.

The outer casing of the tool tapers to a radius slightly larger than that of the axle bearings. This is to maximize the clearance of the tool. By having a minimum amount of material close to the work piece, the tool will be able to fit into more confined spaces than if this taper did not exist. This also allows for much better access to components.

Overall the design of this tool successfully combined axial and radial AFC systems in a decoupled manner that followed the functions and parameters that were outlined with as many off the shelf components as was possible. CATIA design software was instrumental in the successful design and assembly of the components of the tool as well as the generation of technical drawings that were essential for the machining of each component. The final machined and assembled tool can be seen in Figure 3-15.

Figure 3-15 - Final tool assembled

CHAPTER 4 – ANALYSIS

The following chapter will develop a model that will allow the tool to react to a burr by removing it through abrasive cutting. This model will require three separate theories to be developed / presented, see Figure 4-1. The first is the mechanical modeling of the tool. This modeling will predict the theoretical behavior of the tool by splitting the model into the decoupled radial and axial components and laying out the mechanics of each system. The tool-part interaction will be accounted for using Hertzian Contact theory and abrasive cutting theories will be presented to model the removal of the burr. Once this modeling is developed it is presented in a Matlab Simulink simulation to illustrate the ideal behavior of the tool. Values derived from Chapter 5, the testing of the tool, are used within this simulation to validate the model and infer further characteristics.

Figure 4-1 - Areas of theory development presented

4.1 TOOL MODELING

The overarching structure of the decoupled axial and radial AFC systems is very prevalent in modeling the tool. Due to their decoupled nature, each system can be modeled separately. In no part of the modeling do the two systems interact. While the two systems encounter an identical burr, the interaction of each system with that burr is independent of the other.

4.1.1 GLOBAL REFERENCE COORDINATES

To begin the modeling, it is necessary to establish a global positioning system for the tool. This system will use the two gimbal axes as the global X (X_G) and global Y (Y_G) co-ordinates with the positive global Z (Z_G) axis in the direction of the cutting end of the tool. This global position can be seen in Figure 4-2.

After establishing these global coordinates, gimbal rotations can be considered as Euler Angles in an XYZ convention. Identifying the direction of the tool (Z_G) as the theoretical vector direction N, a rotation of angle α about X_G can be made, followed by a rotation of angle β about y' . This is shown in Eq. (4.1).

$$N = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos \alpha & -\sin \alpha \\ 0 & \sin \alpha & \cos \alpha \end{bmatrix} \begin{bmatrix} \cos \beta & 0 & \sin \beta \\ 0 & 1 & 0 \\ -\sin \beta & 0 & \cos \beta \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} \quad 4.1$$

The resultant Euler angle transformation is seen in Eq. (4.2)

$$N = \begin{bmatrix} \cos \beta & 0 & \sin \beta \\ \sin \alpha \sin \beta & \cos \alpha & -\sin \alpha \cos \beta \\ -\cos \alpha \sin \beta & \sin \alpha & \cos \alpha \cos \beta \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} \quad 4.2$$

Figure 4-2 - Global Coordinate System of the Tool

Multiplying Eq. (4.2) through, the unit vector is achieved with respect to the Global Axes:

$$N = \begin{bmatrix} \sin \beta \\ -\sin \alpha \cos \beta \\ \cos \alpha \cos \beta \end{bmatrix} \quad 4.3$$

At this point it is useful to introduce the concept of the “Action Plane”. Due to the axi-symmetric nature of this tool, all forces and motions can be placed within this plane, regardless of the gimbal angles. The action plane is a plane formed by the Z_G axis and the vector produced from the X_G axis being rotated an angle ψ about Z_G . This vector will be considered as x' . The angle ψ is dictated by the unit vector Eq. (4.3) through the following equation:

$$\psi = \tan^{-1} \left(\frac{-\sin \alpha \cos \beta}{\sin \beta} \right) \quad 4.4$$

This relationship is illustrated graphically in Figure 4-3.

Figure 4-3 - Vectors and Values illustrating the action plane angle ψ

and the formation of the Action Plane can be seen in Figure 4-4

Figure 4-4 - Action Plane defined from global coordinates

This action plane rotates to be parallel to the force incident on the tool head and is dependent on that force.

4.1.2 ACTION PLANE MODELLING

Having established the Action Plane with respect to the Global coordinate system, the axi-symmetric properties now allow the tool to be modeled within that plane, effectively reducing the complexity of the tool modeling to a 2-dimensional system. This allows a simplification of the tool kinematics. Seen in Figure 4-5 is a 2-D schematic view of the tool. This is the tool as it exists, ideally, within the action plane and it is from this view that the remainder of the tool modeling will take place.

Figure 4-5 - HFCDT Schematic

In modeling the mechanics within the action plane, the axial and radial components will be modeled separately because they are decoupled. Successful modeling of the two systems in this manner will illustrate their decoupled nature. The radial AFC system is more complex and will be undertaken first. In modeling the radial AFC it will be useful to understand the relationship between the radial motion of the upper end of the pivot (upon which the radial sensors will measure displacement) and the lower end of the pivot (where the tool piece contacts the burr). In this sense, r_u can be taken as the displacement sensed by the radial sensors and r_l can be taken as the displacement of the tool piece, caused by the presence of a burr. The relationship between the two can be expressed in Eq. (4.5).

$$r_l = \frac{b_l}{b_u} r_u \quad 4.5$$

The velocities of the upper and lower radial positions are expressed in a similar fashion in Eq. (4.6).

$$\dot{r}_l = \frac{b_l}{b_u} \dot{r}_u \quad 4.6$$

The acceleration of the upper and lower radial positions are also expressed in similar fashion in Eq. (4.7)

$$\ddot{r}_l = \frac{b_l}{b_u} \ddot{r}_u \quad 4.7$$

The principle behind the radial modeling in the 2-D action plane is the summation of the moment forces about the pivot axis. The relationship of the pivot axis angular position and movement with the upper and lower radial positions and movement is necessary to understand before introducing these moments. The relationship between the angular position and the radial positions are expressed in Eq. (4.8). Figure 4-6 illustrates the location of these values for a better understanding.

$$\sin \theta = \frac{r_u}{b_u} \quad 4.8$$

From Eq. (4.8) r_l can be determined using Eq. (4.5).

Figure 4-6 - Idealized pivot values

Considering the scale of r_u compared to b_u the small angle approximation can be applied; $\sin \theta \cong \theta$, Eq. (4.9) results.

$$\theta = \frac{r_u}{b_u} \quad 4.9$$

Taking the derivative of Eq. (4.9) provides the relationship for velocity in Eq. (4.10).

$$\dot{\theta} = \frac{\dot{r}_u}{b_u} \quad 4.10$$

And taking the second derivative of Eq. (4.9) provides the relationship for acceleration in Eq. (4.10).

$$\ddot{\theta} = \frac{\ddot{r}_u}{b_u} \quad 4.11$$

The values for $\dot{r}_l$ and $\ddot{r}_l$ can be found using the same pivot length ratio as was employed in Eq. (4.5).

The summation of moments, shown in Eq. (4.12), will now be introduced and the derivations of each will be presented.

$$M_{PIVOT} = M_{K-PRA} + M_{C-PRA} + M_{Cut} \quad 4.12$$

Where M_{PIVOT} is the moment about the pivot axis, M_{K-PRA} is the moment related to the stiffness of the PRA, M_{C-PRA} is the moment related to the PRA damping and M_{Cut} is the moment related to the damping due to abrasive cutting / burr removal.

The moment related to the stiffness of the PRA, M_{K-PRA} comes from the force caused by Hooke's Law through the displacement r_u and the stiffness of the PRA, K_{PRA} , through the distance of the upper pivot length b_u and is shown in Eq. (4.13)

$$M_{K-PRA} = F_{K-PRA} b_u = K_{PRA} r_u b_u \quad 4.13$$

The moment from the damping of the PRA, C_{PRA} comes from the force of the damping through the rate of change $\dot{r}_u$ and the same pivot length b_u and is shown in Eq. (4.14).

$$M_{C-PRA} = F_{C-PRA} b_u = C_{PRA} \dot{r}_u b_u \quad 4.14$$

The moment created from cutting is expressed in Eq. (4.15)

$$M_{Cut} = C_{Cut} \dot{r}_l b_l \quad 4.15$$

Where C_{Cut} is the damping due to cutting that will be determined through the abrasive cutting theory in Section 4.3.

The moment about the pivot axis, M_{PIVOT} is useful in determine the angular acceleration, velocity and position that can then be applied to Eq. (4.13-4.15) to determine the above mentioned moments as well

as the new burr height r_l' . Angular acceleration is determined through Eq. (4.16) where I is the angular moment of inertia about the pivot axis of the moving components of the tool. The added 2π term is used to express the angular acceleration $\ddot{\theta}$ in radians.

$$\ddot{\theta} = \frac{M_{PIVOT}}{I} 2\pi \quad 4.16$$

Once the angular acceleration is known, the angular velocity can be determined by integration as seen in Eq. (4.17).

$$\int \ddot{\theta} dt = \dot{\theta} + c_1 \quad 4.17$$

The angular position can be determined by integrating Eq. (4.17) once more, as seen in Eq. (4.18).

$$\int \dot{\theta} = \theta + c_2 \quad 4.18$$

By assuming c_1 and c_2 are equal to zero, the angular velocities and position are now known. The angular velocity can be used to calculate the damping effects on M_{PIVOT} through Eq. (4.14) and Eq. (4.15) and the angular position can be used to determine the new burr height through Eq. (4.19) using Eq. (4.5) and Eq. (4.9).

$$\Delta r_l = \theta b_l \quad 4.19$$

The term Δr_l from Eq. (4.19) represents the change in distance of the tool head from its original position r_l . This change in position must be subtracted from the original position to determine the new position r_l' as in Eq. (4.20).

$$r_l' = r_l - \Delta r_l \quad 4.20$$

In order to produce a standard second order DE that represents the forces of the tool from the perspective of the tool tip, the relationships presented in Eq. (4.5) to Eq. (4.15) are used to produce Eq. (4.21).

$$\left(\frac{I}{b_l}\right)\ddot{r}_l + \left(C_{PRA}\frac{b_u^2}{b_l^2} + C_{r-Cut}\right)\dot{r}_l + \left(K_{PRA}\frac{b_u^2}{b_l^2}\right)r_l = 0 \quad 4.21$$

The axial modeling for this tool can be transferred from the previous tool, the Active Force Compliant Axial Deburring Tool (Liao, 2008) and adapted for the different FESTO cylinder model and a change in the number of cylinders. The model is essentially a spring damper system, modeled using the axial components in the schematic seen in Figure 4-5. The basic equation governing the behavior of the axial components is seen here in Eq. (4.21).

$$m_z\ddot{z} + (C_{z-Cut} + C_{z-Air})\dot{z} + K_z z = 0 \quad 4.22$$

In Eq. (4.22) z represents the position of the tool head in the axial direction. The mass of the moving system, m_z , includes the air spindle and tool bit, the spindle ring that fastens the air spindle to the tool and the moving parts of the FESTO cylinders. The cutting damping is represented by C_{z-Cut} and the damping due to air pressure within the FESTO cylinders is C_{z-Air} . The stiffness that the FESTO cylinders provide is given by K_z .

By combining the radial and axial components, Equation 4.29 expresses the ideal model of the tool.

$$\begin{bmatrix} \frac{I}{b_l} & 0 \\ 0 & m_z \end{bmatrix} \begin{pmatrix} \ddot{r}_l \\ \ddot{z} \end{pmatrix} + \begin{bmatrix} C_{PRA}\frac{b_u^2}{b_l^2} + C_{r-Cut} & 0 \\ 0 & C_{z-Cut} + C_{z-Air} \end{bmatrix} \begin{pmatrix} \dot{r}_l \\ \dot{z} \end{pmatrix} + \begin{bmatrix} K_{PRA}\frac{b_u^2}{b_l^2} & 0 \\ 0 & K_z \end{bmatrix} \begin{pmatrix} r_l \\ z \end{pmatrix} = 0 \quad 4.23$$

The decoupled nature of the tool can be noted here by the presence of zeros in the backwards diagonals of each term in Eq. (4.23). These terms represent the coupling between the radial and axial systems, indicating that there is no coupling.

4.2 TOOL – PART INTERACTION

To begin with, consider the Hertzian contact model; with two discs, of radii R_1, R'_1, R_2, R'_2 being the disc radii and disc edge radii respectively. According to the Hertzian contact model, when these two discs are forced together the discs deform, creating an elliptical stress area. See Figure 4-7 for an example of idealized Hertzian Contact.

Figure 4-7 - Hertzian Disc Contact

The contact area of the two discs forms an elliptical contact surface. This surface is seen in Figure 4-8. The size and shape of the contact ellipse is dependent on several factors including the material properties of the opposing discs as well as the contact force between them. First the compressive principle stress will be defined for this area, followed by the mechanics that define the dimensions and the stress distribution within the elliptical contact area and finally, the forces in terms of one disc impeding upon the other will be presented.

The compressive principle stress of this area can be defined by the following equation:

$$\sigma_{zz} = - \left[\frac{M}{2} \left(\frac{1}{n} - n \right) \right] \frac{b}{\Delta} \quad 4.24$$

Where M is defined in Eq. (4.25). Note that k is the ratio of major to minor ellipse axes, as noted in Eq. (4.30).

$$M = \frac{2k}{k'^2 E(k')} \quad 4.25$$

And n is defined in Eq. (4.26)

$$n = \sqrt{\frac{k^2 + k^2(Z/b)^2}{1 + k^2(Z/b)^2}} \quad 4.26$$

Take note that Z is the depth from the surface of the disc to be considered. Moving further, the value Δ can be found through Eq. (4.27).

$$\Delta = \frac{1}{A + B} \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right) \quad 4.27$$

And A is defined within Eq. (4.28).

$$A = \frac{1}{4} \left(\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R'_1} + \frac{1}{R'_2} \right) \quad 4.28$$

$$- \frac{1}{4} \sqrt{\left[\left(\frac{1}{R_1} - \frac{1}{R'_1} \right) + \left(\frac{1}{R_2} - \frac{1}{R'_2} \right) \right]^2 - 4 \left(\frac{1}{R_1} - \frac{1}{R'_1} \right) \left(\frac{1}{R_2} - \frac{1}{R'_2} \right) \sin^2(\varnothing)}$$

The term B is defined within Eq. (4.29)

$$B = \frac{1}{4} \left(\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R'_1} + \frac{1}{R'_2} \right) + \frac{1}{4} \sqrt{\left[\left(\frac{1}{R_1} - \frac{1}{R'_1} \right) + \left(\frac{1}{R_2} - \frac{1}{R'_2} \right) \right]^2 - 4 \left(\frac{1}{R_1} - \frac{1}{R'_1} \right) \left(\frac{1}{R_2} - \frac{1}{R'_2} \right) \sin^2(\phi)}$$

Note that ϕ is the angle formed between the discs. Now, considering the elliptical contact surface in Figure 4-8.

Figure 4-8 - Hertzian Elliptical Contact Area

It is mathematically advantageous to define the term k in (Eq 4.30) before continuing.

$$k = \frac{a}{b} \quad 4.30$$

Where a is seen as the major axis of the ellipse and b is defined as the minor axis. The term b can further be defined in (Eq 4.31).

$$b = \sqrt[3]{\frac{3k E(k')}{2\pi} (F\Delta)} \quad 4.31$$

Within Eq. (4.31), $E(k')$ can be defined as it is in Eq. (4.32)

$$E(k') = \int_0^{2\pi} \sqrt{1 - k'^2 \sin^2(\theta)} d\theta \quad 4.32$$

These relationships are used to determine the value of a , b and k , allowing the full shape of the ellipse to be known. Knowing this allows the full stress profile to be known as well as the maximum principle stress.

Since the surface is what is of importance here, $Z = 0$ and hence from Equation 32, $n = k$. We can also consider R_1 & $R'_1 = \infty$ when comparing the size of the cutting tool to the size of a burr or a knife-edged corner. From the simple definition of pressure, it can be further defined in Eq. (4.33).

$$P_0 = \left| -\frac{b}{E(k')\Delta} \right| \quad 4.33$$

Substituting $\emptyset = 0$ into Eq. (4.28) and Eq. (4.29) and dividing B by A ,

$$\frac{B}{A} = \frac{R'_2}{R_2} \quad 4.34$$

The elliptical stress distribution pattern can be defined in Eq. (4.35).

$$P_{seg}(x, y) = P_0[1 - (x/a)^2 - (y/b)^2] \quad 4.35$$

This equation stems from the ellipsoid equation Eq. (4.36).

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} + \frac{z^2}{c^2} = 1 \quad 4.36$$

From this pattern, the mean stress across the area will be two thirds, as shown in Eq. (4.37).

$$P_m = \frac{2}{3}P_o \quad 4.37$$

Substituting Eq. (4.31) and Eq. (4.33) into Eq. (4.37) produces Eq. (4.38).

$$P_m = \sqrt[3]{\frac{4kF}{9\pi E^2(k')\Delta^2}} \quad 4.38$$

From Eq. (4.38) the force F can be isolated, as in Eq. (4.39).

$$F = \frac{9\pi E^2(k')\Delta^2 P_m^2}{4k} \quad 4.39$$

F is the force encountered by the tool as it encounters the part. In this case, it is the force the tool must exert for deburring.

4.3 ABRASIVE CUTTING THEORY

There has been much recent work on abrasive cutting theory and the idea of the cumulative effect of varying sizes of grain particles affecting a work surface. Williams and Xie (Williams & Xie, 1996) created a simulation model that combines the contacts between a soft work surface and grains of sizes varying through a Gaussian distribution. The model predicts the overall coefficient of friction and the wear rate on the soft surface by employing three types of interaction, Elastic deformation (shakedown) through basic friction modeling, plastic deformation (through empirical approximation) and micro chipping through a cutting model (Williams & Xie, 1996). Zhou and Xi further the concept of a Gaussian distribution of grain sizes in order to create a theory base and software that accurately demonstrates grinding and polishing surfaces and operations when compared to empirical results (Zhou & Xi, 2002); (Xi & Zhou, 2005).

Figure 4-9 - Polishing stone topography (Xi & Zhou, 2005)

The abrasive cutting theory presented in the above mentioned literature can be drawn on and combined with the Hertzian contact modeling to explain the deburring process of the edge deburring tool assuming a grinding stone tool piece is employed. After a stone topography is generated through Gaussian distribution (Figure 4-9) a semispherical grain profile is assumed and the Brinell hardness scale is employed as the basis of the depth of penetration of the grains.

An algorithm called the “Search Method” (Xi & Zhou, 2005) first determines the maximum depth of cut from the largest grain size. In Eq. (4.40) this is shown, where F_c is the total force (cutting force) imposed on the tool piece, r_{g1} is the radius of the largest grain and HB is the Brinell Hardness of the work piece material.

$$h_1 = \frac{F_c}{2\pi r_{g1} HB} \quad 4.40$$

This value is usually unrealistically large. After h_1 is found, the difference $(r_{g1} - h_1)$ is compared to the next largest grain size r_{g2} . If $(r_{g1} - h_1) > r_{g2}$, then the second grain is not in contact with the work piece and the whole of F_c will be imposed on the single grain g_1 . Otherwise (and nearly always the case) the second stone will distribute some of F_c . The same is repeated for the difference of h_2 and r_{g2} and so on. This algorithm is repeated until $r_{gk} - h_k > r_{g(k+1)}$ where h_k is then considered the depth of cut, Eq. (4.41) and Eq. (4.42) illustrate this. This was employed in the software written by Xi and Zhou.

$$f_i = \frac{F r_{gi}}{\sqrt{\sum_{i=1}^k r_{gi}^2}} \quad 4.41$$

$$h_k = \frac{f_k}{2\pi d_{gk} HB} \quad 4.42$$

Where f_i is the force applied on grain g_i . Using this method, micro depth of cut calculations were tabled alongside experimental data and assumed to equal surface roughness. For deburring applications however, it is advantageous to use these values as depth of cut.

Table 4-1 - Comparison of modeled data with experimental (Xi & Zhou, 2005)

Grit number M	Predicted h μm	Experimental h μm
320	0.4-0.8	0.7-0.8
400	0.25-0.6	0.5-0.6
600	0.2-0.5	0.25-0.3

These depth of cut values represent the material removal rate per cutting area proportional to work piece area. That is to say that for 1mm^2 of work surface, 1mm^2 of abrasive performing one full pass shall remove material to the depth specified by the depth of cut. This proportionality allows for simplification in a 2D interpretation whereby 1mm lengthwise abrasive surface passing once over a 1 mm length of cross sectional work surface length will remove material equal to the cutting depth. Applying this

relationship to a cylindrical or spherical grinding stone, the rate at which the tool progresses to remove the burr can be determined. This is illustrated in Figure 4-10.

Figure 4-10 - Tool - Burr contact

By comparing the burr width, w_b , to the circumference of the cutting stone (assuming $R_t \gg w_b$) and multiplying by the cutting depth and then by the rate of rotation of the tool, the rate at which the burr is removed can be defined, as in Eq. (4.43).

$$\dot{z} = \frac{2\pi R_t}{w_b} h_k \omega \quad 4.43$$

In Eq. (4.43) it is assumed that cutting depth value is accurate and that the width of the burr is known. As discussed in earlier sections, the nature of the burr is that it is irregular and rarely known ahead of time, hence the need for an adaptive, active compliant tool. For the purpose of theory and simulation though, this is acceptable.

The final value to be determined is the cutting damping ratio, C_{cut} to be applied alongside the tool damping ratio. This value is found in Eq. (4.44) where F_c is the cutting force applied.

$$C_{Cut} = \frac{F_c}{\dot{z}} \quad 4.44$$

4.4 SIMULATION MODEL

The basis of the simulation model comes from an accumulation of the above theory, modeled in Matlab's Simulink. The simulation has two components to it, the axial portion and the radial portion and is limited to the tool itself; it does not model the air system that supports it. Integrating an air system model would add a level of complexity that is beyond the scope of the thesis. A complete air system model is available for the axial polishing and deburring tool from (Liao, 2008).

Because the Radial and Axial AFC systems are decoupled, they are modeled separately. The radial portion is based around the summation of moments about the pivot axis. Stiffness values were used that reflect the measurements provided from Section 5.5.3. The axial model is straightforward and also uses measurements from Section 5.5.3 in order to ensure that the input was reflective of the experimental results.

In reference to Figure 4-11, the simulation begins from the generation of the burr. The signal representing the displacement of the tool head from the work piece edge that is being deburred is summed with any chamfer offset provided and the calculated burr reduction is subtracted, Eq. (4.20) can be equated to this. The new burr height is then converted to the cap displacement where the PRA actuator is, employing Eq. (4.5). Using Hook's law, the force that the PRA applies to the end cap is determined. This is then multiplied to the upper pivot length to determine the moment applied to the gimbal axis, as with Eq. (4.13). This moment is summed with the calculated effects of the damping from the PRA and the cutting Eq. (4.12).

The summed moment is multiplied by 2π to place the value into radians and divided by the moment of inertia, I , about the pivot axis, which was determined through CATIA modeling Eq. (4.16). This value now represents the angular acceleration, in radians. This value is integrated to find the angular velocity Eq. (4.17). The angular velocity or rate of rotation about the gimbal axis is useful in finding the effects of damping on the moment through Eq. (4.14) and Eq. (4.15). Damping values from cutting are calculated using the equations developed from Section 4.3. Damping values from the PRA are not known and so the system is calibrated to determine these values by adjustment to a level that provides the desired edge profile from the burr profile readout.

Results of the radial deburring simulation can be seen in Figure 4-12. In this figure, the tool encounters the burr and effectively removes it. The values used for this simulation are seen in Table 4-2.

Table 4-2- Values employed in Radial Simulation

Term	Value	Term	Value
b_u	9.25"	K_{r-Cut}	600
b_l	2.19"	C_{r-PRA}	10
I	15.03 lb-in ²	$Burr\ Size$	0.025"
C_{r-Cut}	0.285

Figure 4-12- Radial AFC Simulation Output (Red: Burr Input, Blue: Tool Reaction/Output)

The axial AFC simulation, seen in Figure 4-13, closely mirrors the radial one except that it is simpler as it does not have to account for the exchange of moments about the pivot axis.

The values in this simulation are found in Table 4-3.

Table 4-3 - Values used for Axial AFC

Term	Value	Term	Value
m	0.422	K_{z-Cut}	10
C_{z-Cut}	0.285	C_{z-Air}	4
Burr Size	0.025"		

Viewing the results of the simulation, it can be seen that the simulation correctly demonstrates the burr removal ability of the tool and that the systems can function well in a decoupled manner.

CHAPTER 5 – FABRICATION AND TESTING

The original planned outline of this thesis was to fully design, manufacture and test this tool by attempting to deburr features of a turbine disc. This deburring was to occur on a CNC machine with the tool performing online adjustments in stiffness based on the data collected from its sensors. Time constraints and certain practical limits made this plan too ambitious. Instead, it was decided that the testing will conclude with static testing of the tool's behaviors in the radial and axial directions. The decoupled design of the tool allows for the testing of the axial and radial components to be entirely decoupled from each other and performed separately. The sensors and stiffness will be investigated only in the settled state. The transient behavior of the tool and any time variable characteristics will be left to others to investigate.

This section will also explain certain considerations taken when designing for manufacturing, the manner in which the tool was calibrated (including the design considerations and the final design of the test rig) and the final test results which explain the behavior of the tool with respect to different pressures and forces as well as investigate whether certain assumptions in the theoretical modeling in Section 4 are actually applicable.

5.1 TOOL FABRICATION

Tool components were machined at Apollo Machining in Mississauga, Ont. The total cost of machining the tool components came to \$3500. This price did not include the test rig which will be discussed later.

Special care was taken to minimize the total cost of the machining when possible by considering the number of tool changes and set up changes that would be required for the features on each component. Many radii and dimensions were driven by the practical capabilities of the milling and turning operations and the effects that those dimensions would have on the process.

For example, if one sized milling bit could be used to machine the features of an entire part, this would reduce the process time and number of tool changes. If a part could be machined on a lathe or mill with only one set up, additional cost savings could be found there. This is the manner in which the stipulation that "components should be designed in a way that will allow for the easiest and most cost effective manner of manufacturing" from Section 3.1 was met.

5.2 CALIBRATION

5.2.1 Test Rig Design

Prior to calibrating the tool, a test rig to calibrate and test the tool had to be conceived. This test rig had the following design parameters placed on it:

- Must be able to apply and measure force in multiple directions for radial testing
- Must be able to measure small displacements of the tool tip (0.001" resolution)
- Must be able to measure small displacements at the sensor end (0.001")
- Must be rigid enough that bending of the rig would be negligible
- Must be adjustable for versatility

These parameters would be accommodated by the fact that an optics table would be used to mount all of the equipment and that the tool would be held steady in place by a vice, fixed to the table. Some adjustments would be made available by the vice and positioning on the table.

The test rig was designed on CATIA in the same manner in which the actual tool was. Figure 5-1 shows an image of the test rig, with parts labeled and with the tool also present where it would be during testing. Seen on the right is a turn-screw. This turn-screw is used to apply a displacement to the tool tip. Directly in contact with the turn-screw, between the turn screw and the tool tip is a load cell to measure the force. The load cells are both Full Bridge Thin-Beam load cells made by Omega. Both 1 lb and 10 lb load cells were acquired and both are able to fit in the rig.

Opposite to the turn-screw is a dial indicator capable of measuring to within 0.0001" with a range of 0.025". The base of the rig is adjustable so that the dial indicator, which is fixed to the C-Mount, can be positioned directly adjacent to the tool piece and the turn-screw then adjusted to position the load cell directly opposite. Figure 5-2 shows this configuration, which is used for radial testing. Note that the shape of the C-Mount allows the mount to be adjusted to a 45° or 90° angle, to test for radial uniformity in performance.

Figure 5-1 - Testing and Calibration Rig

Figure 5-2 - Radial Stiffness Testing Set Up

5.2.2 Load Cell Calibration

The dial indicator came from the manufacturer as calibrated and so the only part of the test rig that needed calibration was the load cells. To calibrate these, weights were suspended from the cell in appropriate increments based on the total weight rating of the load cell.

Figure 5-3 - Calibrating the load cells (10 lb cell pictured)

The load cell power supply was adjusted with each load cell so that the total load of 1 lb or 10 lbs would provide a 10V output to the DAS. Weights between 0 and the maximum load were also loaded onto the cell to ensure that the relationship was linear. The 1 lb calibration chart is shown here. The 10 lb load cell was similar however scaled up x10.

Figure 5-4 – Calibration Chart

5.2.3 Tool Sensor Calibration

Radial and Axial tool sensors had to be calibrated in order to properly interpret the collected data. These sensors were calibrated using the test rig and another displacement sensor which had a longer probe with more travel and was hence more versatile. The end cap was set up with the test rig as shown in Figure 5-5.

Figure 5-5 - Radial Sensor Calibration

Because this displacement probe was calibrated to millimeters, the axes are in Volts and Millimeters.

Figure 5-6 - Calibration of Radial Sensor X Axis

The Y-Axis calibration yielded similar results:

Figure 5-7 - Calibration of Radial Sensor Y Axis

Note that the two graphs have some differences. The X-sensor graph has a more refined plot of points while the Y-sensor graph has a larger spread. In each case, the relationship is linear and the rate at which the voltage changes is 1.1 V/mm for the X-sensor and 1.2 for the Y-sensor. These values can be used in order to calibrate online response to tip displacement when the tool is running a deburring tool path.

While these tests were run, cross talk between the sensors was measured. By this, what is meant is that the magnetic field used to sense displacement in X is also impeding on the Y sensor. Movement of the magnet will effect changes in voltage of both sensors. The amount of unintended change was recorded determined and is shown here. The effect is minimal however can still be compensated for if deemed necessary.

Figure 5-8 - Crosstalk sensed by X-Sensor while testing Y Sensor

The axial sensor used was the M-150 CeleSCO String-Potentiometer as mentioned in section 3.5.2. This sensor was calibrated by displacing the spindle ring with spacers of known thicknesses and comparing this to the output of the M-150. The results were linear, as expected. The calibration is shown below:

Figure 5-9 - M150 Celesco Calibration

The final sensor values are listed here in V/in and will be used in the following section to interpret the data collected through the Data Acquisition System.

Radial X Sensor 28.04 V/in

Radial Y Sensor 30.44 V/in

Axial Sensor 5.64 V/in

5.3 TESTING

5.3.1 Data Acquisition System

The Data Acquisition System used was a USB1208-FS unit from Measurement Computing. This unit was used in its 4 channel 12bit differential input mode to collect data from the PCB boards within the HFCDT as well as from the load cell.

Figure 5-10 - DAS USB1208-FS unit from Measurement Computing

The USB1208-FS connects directly to the computer via USB. Signals were read using a Simulink program and then MATLAB was used to analyze the collected data.

5.3.2 Testing Method

Several different tests were done in order to determine the characteristics of the tool. All of the sensors were calibrated (Section 5.2.3). Then the overall radial tool stiffness was measured at different PRA pressures. Tests were then done to determine the level of internal bending that was occurring within the tool with no PRA pressure and at the maximum PRA pressure tested. These tests were followed by axial stiffness tests to determine the relationship between the FESTO cylinders and the axial stiffness of the tool. The following images illustrate each of the tests concisely. The test set up to measure axial stiffness is seen in Figure 5-11. The test set up to measure the internal bending of the tool is shown in Figure 5-12. The test set up to measure the axial stiffness of the tool is shown in Figure 5-13.

Figure 5-11 - Radial Stiffness testing. PRA Gauge pressure varied for various stiffness curves.

Figure 5-12 - Testing internal tool bending. End cap removed, replaced with retainment plate for PRA.

Figure 5-13 - Axial Stiffness Testing. Various pressures tested for Stiffness Curve

5.3.3 TEST RESULTS

The test results of the tool were tabulated and plotted in Excel. The graphs are displayed in the following pages along with some comments interpreting the behavior of the tool. All pressures are gauge pressure.

Figure 5-14 - Stiffness Plot of PRA at pressures 0 -22 PSI

Test results of the radial stiffness were intuitive for the most part. The stiffness is very nearly linear for each of the pressures measured and what is most interesting is that each of the curves, once within its linear portion, tends to have a very similar slope, showing that the escalation of the stiffness is equal, regardless of the pressure. Ideally, each line indicating a pressure level would have been evenly spaced with respect to each other however the limitations of the tool in its construction instead created the spread that is seen here.

Another way to view this data is by looking at the stiffness based upon different distances as data lines. The following graph shows each distance as a data line, the pressure along the x axis and the stiffness

along the y. This interpretation could be useful when selecting a pressure to stiffen the tool to a stiffness based upon the deflection of the tool and the cutting parameters.

Figure 5-15 - Stiffness vs Pressure

The plotted lines for displacements 0.015" to 0.024" were only partially plotted due to the limitation of the 10 lb force sensor. The data points available were extrapolated to show the stiffness up to 22 psi.

Comparison of tip displacement and end cap displacement shows that there is a significant amount of internal bending within the tool components. With the exception of no gauge pressure, the movement sensed on the end cap with that of the tip displacement was similar for all pressures and was significantly less than the ideal amount of displacement. Figure 5-16 shows the measurements in relation to one another and Figure 5-17 shows a curve indicative of the trend (created by averaging the characteristics of each quadratic trend line) with respect to the ideal level of displacement.

Figure 5-16 - Measured displacements

Figure 5-17 - Measured tool bending in comparison with the ideal, if no bending were to occur

Axial stiffness testing revealed that the tool will provide an equal amount of force, regardless of the axial displacement of the tip. This creates a relationship where the axial force applied is solely dependent on the gauge pressure of the pneumatic cylinders.

Figure 5-18 - Axial Force vs Displacement

Measurements indicated that the force applied in the backward direction, i.e. returning to the original position after displacing the tool tip to its maximum point was significantly less than the forward force applied. All of the forces are tabulated below in Figure 5-19 to showing the relationship between the pressure within the FESTO pneumatic cylinders and the amount of force that is applied. The relationship of the two plotted lines (forwards and backwards) with respect to each other indicates that an additional force is at play. This could be a frictional force attributed to non-ideal construction of the tool's components as the force is working in both directions.

Figure 5-19 - Axial Force vs. Gauge Pressure

CHAPTER 6 – CONCLUSIONS AND FUTURE WORK

6.1 CONCLUSIONS

After viewing the results, valuable insight has been gained into the characteristics of the tool, its performance and aspects of its design and construction. From this study and design exercise, the following insights were gained:

- The automation of deburring and edge finishing is not a trivial process. It requires the precision and repeatability of an automated process incorporated with the robustness and adaptability of a manual process. Because of the transient nature of the machining process (constantly varying levels of tool wear, varying cutting paths and microstructure, etc...) no two burrs are alike and as a result, no two deburring passes will encounter the same conditions.
- Many forms of compliant tooling exist for deburring and edge finishing, both manual and automated, most of which are passive compliant tools. Throughout the literature no other tool was found that attempts to incorporate both axial and radial active compliance into a single tool.
- A variety of different types of actuation and sensing were investigated. Various Pneumatic methods of actuation, turn screw, voice coil and magnet actuation were explored for both effectiveness as well as cost. Methods of motion were also looked at. Gimbals, universal joints, ball joints and other modes of transmission were explored based on both literature and creative design. Through defining specific design parameters and with use of trade studies, a design was configured that balanced effectiveness with complexity and cost.
- CATIA is a highly effective and efficient method of generating the specific geometries of design. Within CATIA, the configuration for the deburring tool was developed, all stock items were defined and then original part dimensions were configured. Then the technical drawings were generated to produce a complex tool with many different moving parts. All of these parts fit together on the first try with only minor, designer based issues that were easily resolved. It is

highly recommended that any future design be based through this design tool or a comparable CAD software package.

- The sensor system designed for this tool proved remarkably effective. The radial sensing, centered around the Honeywell HMC1501 magnetic sensors and the electronics that were built around these devices functioned beyond their expected accuracy and attained a resolution $>0.002''$. This resolution could be further enhanced through greater signal filtering and processing.
- The Pneumatic Ring Actuator, while in concept is of sound principle, needs further development. Fashioning a precision deburring actuator through tedious, laborious means from bicycle tubing is not a preferred manufacturing method. More shall be mentioned in the Section 6.3 about the development of the PRA.
- The mechanical design of the tool was sound in theory however in practice several things were revealed to need improvement. The center assembly of the tool consisting of the two spindle housings needs to be much stiffer. When dealing with such small displacements as those encountered in deburring, even the slightest bending can skew the data and although such bending can be accounted for, it reduces the accuracy and response of the tool while complicating any modeling.
- The tool overall was an excellent first iteration of a radial-axial active force compliant deburring tool. Experimentation demonstrated its effectiveness at sensing burr displacement and thereby its potential to respond. Future design iterations should take note of frictional loss/interference from the size and configuration of the joints and moving parts (configured and dimensioned to reduce cost and provide an ease in demonstration) and incorporate the lessons learned from this work into future design iterations.

6.2 MAIN RESEARCH CONTRIBUTIONS

As was demonstrated in the literature review, there has, up until now, not been a deburring tool that has incorporated both the axial and radial components of active compliant deburring in a single tool. This thesis presents the process of designing, manufacturing and testing such a tool in a way that is transparent and reproducible.

- The active force compliant deburring tool designed here is based on the principle that the axial and radial components can be incorporated with little to no coupling, allowing minimal interface between the two systems. This decoupling allows for a simpler design and control architecture. Through this work, it has been proven that the decoupled configuration presented is a practical and viable method of employing axial and radial active force compliant deburring.
- Models of the radial and axial AFC systems were created and successfully implemented. These models demonstrate the mechanics of the burr removal systems and their successful implementation illustrates the decoupled nature of the tool. The modeling of the tool was also used to infer values of the tool itself.
- Within this design, the axial deburring system is incorporated into the pivoting portion of the radial system. Although pivoting itself was utilized on the CADET device from UTRC, in this case it was used for mechanical advantage in both actuation and sensing. Although bending within the device negated this mechanical advantage, the premise remains and can be improved upon. This configuration is unique and has not been attempted before.
- The proposed concept of the PRA is also new. Within the FlexDeburr tool from ATI (see Figure 2-6) their ring actuator uses a series of air pistons arranged radially about a center rod. The tool itself is also a passive compliant tool with no feedback or control. The PRA presented here uses a volume of gas (or liquid) configured in a torus shape, contained within an elastic material constrained in such a way that when the inner pressure increases, the volume and elastic material will expand inward, exerting a centering force on the pivot rod. The inner pressure is dictated by the level of offset of the pivot rod sensed by the radial sensors. This type of actuator is not apparent in any literature and as such is a novel idea.

- The radial sensing system that was conceived for the purpose of this thesis is unique. By mounting magnets on the cap of the pivot rod, magnetic sensors could determine the displacement in the X and Y directions. The circuitry that was designed for this sensing system allowed a resolution of $>0.002''$ to be achieved. It was unclear what type of accuracy such sensors would provide. This testing has provided performance specifications on this sensor configuration that will be invaluable in future designs. No sensing system resembling this type was apparent in the literature and as such this too is a unique and novel contribution.
- Combining all of the new design contributions with those that were collected through review of existing tools, a configuration was produced that is unique in its architecture and illustrates the first iteration of a design tool that will incorporate both radial and axial active force compliant deburring. The design process and the lessons that can be inferred from the results and conclusions of this thesis, the methods that worked well and those that need improvement will provide valuable insight for future iterations of this design.

6.3 FUTURE WORK

The focus of this research was to develop a tool configuration that would allow for active force compliance both in the axial and radial direction. There are many other aspects of a system that would provide this active compliance to be developed as well as improvements upon the current design that can be pursued.

- The PRA should be further developed to provide a more consistent shape and a more conventional and convenient application. Medical silicone product manufacturers should be investigated for this purpose. Further exploration on this topic can be found in (Kroeker, 2010). By manufacturing a consistent rubber or silicone product, the tool can be more easily reproduced and will benefit from more uniform performance.
- There are several mechanical improvements that the tool can undergo. The center pivot section should be redesigned so that it is of greater stiffness. A change in the geometry (make it shorter to reduce bending, as the sensors employed are now known to have sufficient resolution) as well as a change in material will allow for less bending of the tool and increased performance. A

shorter, smaller air spindle and smaller FESTO cylinders should also be incorporated to make the tool more compact and versatile. The gimbals that facilitated the rotation required for the radial motion should also be improved. This can be enhanced either by a higher quality (more expensive) 2 axis gimbal system or a change to a grooved ball joint like that found in the FlexDeburr (Figure 2-6 and accompanying discussion) to reduce slack and friction.

- The control system for this tool has yet to be developed. Modeling and controlling the entire air system volume is an immense undertaking in and of itself. Much of the groundwork for this was accomplished in (Liao, 2008) and the future works of that thesis can be referenced for further guidance on that front.
- Once the above mentioned future works are accomplished, a means by which to test the tool on line and eventually incorporate the tool into potential manufacturing processes should be developed. A method of signal transfer and processing (there are wireless possibilities here as well as on board processing), a means of incorporating the tool with a positioning system, whether it be CNC or Robotic and software to generate tool paths that incorporate the geometry and behavior of the tool are just some of the avenues of research that can be pursued.

Appendix A – Electronic Sensors

HFC Deburring Tool: Electronics Design and Development

The following sections refer to the HFC Deburring Tool Electronics schematic (v0.25)

v0.10 - Initial schematic and PCB layout

v0.15 - Modified sensor traces on PCB layout

v0.20 - Added buffer op-amp for M150 sensor

v0.25 - Modified components spacing and traces (sent out for manufacturing)

Section A: Power Circuitry

Power is supplied to the HFCDT's electronics with an 18 VDC, 350 mA wall wart through an 8-pin PREH connector located on the tool cap.

The supply is fed to two parallel linear voltage regulators to obtain +10 VDC and +12 VDC supplies. The 12 volt regulator supplies the two op amps (TLC072 and LF412). The 10 volt regulator supplies the HMC1501 position sensors.

The LM2937 requires an output bypass capacitor for stability. A 10 μ F tantalum chip capacitor with an equivalent series resistance (ESR) < 3 was chosen. In addition a 0.1 μ F ceramic capacitor was added to the input side of the regulators (and also on the voltage supply side of each IC).

Other notable features of the LM2937 regulator are:

- low voltage dropout
- short circuit protection
- reverse battery protection
- thermal shutdown protection

Section B: Position Sensing

The sensor chosen for position sensing was the Honeywell HMC1501. This sensor has the following beneficial features:

- Small size and low cost as compared to other non-contact measurement systems such as laser
- Typical resolution of up to 0.002" for linear position sensing
- Accuracy up to 0.1%
- +/- 45° angular sensing range

The sensor is comprised of four anisotropic magnetoresistive (AMR) sensing elements configured in a Wheatstone bridge configuration (see Figure 2: AMR Bridge [2]). An AMR material has an impedance that varies with an applied magnetic field. Specifically, it is dependent on the angle between the direction of electric current and the orientation of the magnetic field.

The sensor operates in saturation mode, meaning that the sensor only responds to changes in the orientation of the field and not the magnitude. The minimum strength of the field to saturate the sensor is 80 gauss, with no specified upper limit.

The maximum resistance of the sensing element occurs when the direction of the current is parallel to the applied magnetic field.

Signal Output

The HMC1501 contains one active sensing bridge. The output of the bridge can be obtained from the following equation:

$$\Delta V = -V_s S \sin(2 \theta)$$

where ΔV is the differential output voltage

V_s is the bridge supply voltage

S is the material constant (12 mV/V)

θ is the reference to the magnetic field value (degrees)

Taking the bridge supply voltage as 10V and a magnetic field value of +/- 45° provides a differential output $\Delta V = +/- 120$ mV (biased at 5V).

Note: The Wheatstone bridge performs as a rail splitter to create two near +5 volt sources that are driven apart by ΔV . A differential amplifier is then used to amplify ΔV to a more suitable level for analog to digital conversion (ADC).

The 10 V bridge supply was chosen since most data acquisition systems have 0-10 volt ADCs. A 5V or 3.3V supply would be more appropriate if the tool was to be made "smart" by embedding a microcontroller.

Section C: HMC1501 Signal Conditioning

The outputs of the HMC1501 are fed to an instrumentation amplifier. An instrumentation amplifier is a differential amplifier with buffered inputs; this eliminates the need for impedance matching of the source and input electronics. The instrumentation amplifier typically also offers very high common mode rejection; the common noise on the inputs is greatly attenuated.

The instrumentation amplifier chosen for the signal conditioning of the HMC1501 outputs was the TLC072 as recommended in the sensor's Application Note. The TLC072 has dual amplifiers thus both sensors can be fed into one unit.

The TLC072 has the following specifications (using a 12V supply):

- Gain Bandwidth Product = 10 MHz
- Slew Rate = 16 V/ μ s
- CMRR = 100 dB

The HMC1501 sensors have a bandwidth of approximately 5 MHz. This far supersedes the requirements of the application. The signal conditioning circuit was designed with a LP filter with a cutoff frequency of about 100 Hz. The bandwidth of the system can easily be increased by simply swapping out the feedback capacitor in the TLC072's feedback loop. The cutoff frequency f_c is simply calculated as:

$$f_c = (2\pi RC) = [(2\pi)(390\,000)(4 \times 10^{-9})]^{-1} = 102.2 \text{ Hz}$$

Since the valve used to actuate the HFCDT's actuator is only capable of 5 to 10 Hz, the 100 Hz bandwidth of the sensors was considered to be more than adequate.

The amplifier gain is determined by the ratio of the external resistors as:

$$A_g = R_f/R_i = 390\,000 / 10\,000 = 39 \text{ or about } 32 \text{ dB}$$

Where R_f is the feedback resistor and

R_i is the resistor to the op amp input.

The 39x amplification of the ± 120 mV output from the HMC1501 sensors provides a full scale range of almost 10 volts (0.32 V to 9.68 V) output to the data acquisition system.

The signal conditioning circuit also has the capability to perform offset trimming using a multiturn 1 k Ω trimpot. This allows any offset voltages from sensor manufacturing, temperature effects, sensor misalignment, or tool assembly to be compensated for. The trimming circuit acts as a voltage divider and biases the positive input on the op amp. The 1 k Ω resistors on the trimpot leads allow for greater fine tuning by minimizing the adjustment range. In this case the positive input can be biased between 3.33 and 6.67 volts. This can be tightened further by replacing the 1 k Ω resistors with 2 k Ω resistors (changing the bias range to between 4 and 6 volts). The trimpots can be accessed by removing the 8-pin female PREH connector on the HFCDT cap and thus can be adjusted while the tool is powered using a small screwdriver. The 1 k Ω resistors attached to the trim pot create a 5 volt bias using the voltage divider principle. The output signal must be biased so that positive and negative displacements can be measured since the bridge and op amps are only powered by a single positive voltage supply.

Section D: Axial Displacement Measurement

The axial displacement of the tool bit is measured using the Celeco M150 cable extension position transducer. This sensor was primarily chosen because of its small size (the manufacturer claims it to be the world's smallest stringpot) and ease of use. The sensor provides an output signal proportional to the input signal. The sensor is supplied by the 12 V source. The output of the sensor is sent to a JFET op amp (LF412) which is configured in a voltage follower setup; again this is to simply eliminate loading effects due to impedance mismatch.

No analog filtering circuitry was added due to the lack of space on the PCB; however, in future revisions of the PCB, this could be added.

Section E: PCB Design

A 4-layer board was designed using the Express PCB board service. A 4-layer board provides greater noise immunity and can be designed more compactly and efficiently since ground and power traces don't have to be run to every component since the two inner board layers provide a power and ground plane.

The board layers are copper and the laminate is FR-4. The total thickness of these boards is 0.067". The board thickness had to be accounted for in order to properly position the sensors to be in-line with the magnets mounted on the pivot rod.

Two circular boards were designed to stack and fit within the tool cap. The PCBs are segregated by function: 1) bottom PCB houses the sensors; and 2) the top PCB contains the signal conditioning and power circuitry (see PCB board layout). The bottom PCB has a circular cutout so that the PCB fits over the top of the pivot rod bringing the sensors in line with the magnets. Sections of the inner copper layers were removed in the design so that when the cutout was performed there would be no chance of bridging the ground and power planes resulting in a short; and also for when the aluminum pivot rod comes in contact with the PCB. After the centre hole was cut in the bottom PCB, the inner surface was coated with M-Coat A (an oil modified polyurethane) that would ensure that no copper planes were exposed. The centre cutout provided about a 0.125" clearance between the pivot rod and the edge of the PCB. This allows burr sizes of up to 20 mils ($0.125/6$... where 6 is the displacement amplification of the pivot arm) to be detected. Larger burr sizes can be accommodated by changing the top diameter of the of the pivot rod.

The bottom PCB also constrains the top of the HFCDT actuator.

The PCBs are connected through a 20-pin surface mount connector (DF40 series - 0.4 mm contact pitch). This type of connector is specifically designed for board to board connections. Four aluminum standoffs were also epoxied to the bottom PCB for board spacing and to allow for attachment to the top of the HFCDT cap. The PCB's were only fastened to the cap for easier assembly and maintenance or repair. The bottom layer of the sensor PCB was left free from any electronic parts or other hardware (i.e. machine screws) so that the inflatable actuator would not damage or be damaged (i.e. punctured).

The side of the HFCDT has additional ports so that the exhaust from the actuator could be fed back into the tool to cool down the electronics (if necessary). An additional temperature sensor may be an appropriate future consideration so that temperature offsets could be adjusted for (monitoring and active cooling).

The HFCDT was made from a non-ferrous metal in order to test the capability of the sensors. The magnets used are particularly strong (NdFeB rated at 10800 gauss) and it was thought that the magnets could potentially magnetize the tool body and affect the sensor readings. Making the tool from stainless steel was considered, but the cost was prohibitive.

Section F: 8-pin PREH Pin-Out

Female			Male				DAQ
Pin No.	Signal	Colour	Pin No.	Signal	Colour	Pin No.	Signal
1	+18V	RED	1	+18V	B/W		NC
2	X _{out}	WHT	2	X _{out}	WHT	1	X _{out}
3	Z _{out} (M150)	BLU	3	Z _{out} (M150)	BLU	3	Z _{out} (M150)
4	NC	--	4	NC	--		NC
5	Y _{out}	GRY	5	Y _{out}	BRN	2	Y _{out}
6	GND	BLK	6	GND	BLK		NC
7	GND	GRN	7	GND	BLK	?	GND
8	NC/Shield		8	NC/Shield	BARE		NC

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