SOME ASPECTS OF FIVE-AXIS MACHINE TOOL DESIGN, ASSEMBLY, AND TESTING

By

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In memory of Louis Arnold Bernhard and James Jacob Kremer for inspiring their grandson to use his mind and his hands to dream and build.
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**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>A, B, C</td>
<td>Rotary axes in spindle/tool</td>
</tr>
<tr>
<td>A', B', C'</td>
<td>Rotary axes in table</td>
</tr>
<tr>
<td>a</td>
<td>Radial depth of cut</td>
</tr>
<tr>
<td>b</td>
<td>Axial depth of cut</td>
</tr>
<tr>
<td>b&lt;sub&gt;lim&lt;/sub&gt;</td>
<td>Axial limit of stable cut</td>
</tr>
<tr>
<td>b&lt;sub&gt;lim-crit&lt;/sub&gt;</td>
<td>Axial critical limit of stable cut</td>
</tr>
<tr>
<td>c</td>
<td>Feed per tooth</td>
</tr>
<tr>
<td>CRAC</td>
<td>Chatter recognition and control</td>
</tr>
<tr>
<td>CNC or NC</td>
<td>Computer numerical control</td>
</tr>
<tr>
<td>CS</td>
<td>Coordinate System</td>
</tr>
<tr>
<td>CFRF</td>
<td>Cross frequency response function</td>
</tr>
<tr>
<td>DFRF</td>
<td>Direct frequency response function</td>
</tr>
<tr>
<td>DOF</td>
<td>Degrees of Freedom</td>
</tr>
<tr>
<td>f&lt;sub&gt;n&lt;/sub&gt;</td>
<td>Natural frequency</td>
</tr>
<tr>
<td>f&lt;sub&gt;t&lt;/sub&gt;</td>
<td>Tooth passing frequency</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational acceleration</td>
</tr>
<tr>
<td>HSM</td>
<td>High Speed Machining</td>
</tr>
<tr>
<td>ipm</td>
<td>Inches per minute</td>
</tr>
<tr>
<td>K&lt;sub&gt;s&lt;/sub&gt;</td>
<td>Specific power</td>
</tr>
<tr>
<td>kW</td>
<td>Kilowatts</td>
</tr>
<tr>
<td>m</td>
<td>Number of teeth on the cutter</td>
</tr>
<tr>
<td>MRR</td>
<td>Metal Removal Rate</td>
</tr>
<tr>
<td>MTRC</td>
<td>Machine Tool Research Center</td>
</tr>
<tr>
<td>n</td>
<td>Spindle Speed</td>
</tr>
<tr>
<td>N</td>
<td>Number full waves between teeth</td>
</tr>
<tr>
<td>rpm</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>TF</td>
<td>Transfer Function</td>
</tr>
<tr>
<td>X, Y, Z</td>
<td>Linear axes designators in spindle/tool</td>
</tr>
<tr>
<td>X', Y', Z'</td>
<td>Linear axes designators in table/work piece</td>
</tr>
<tr>
<td>e</td>
<td>Phase shift between old surface and new surface</td>
</tr>
<tr>
<td>μ</td>
<td>Orientation factor</td>
</tr>
</tbody>
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Abstract of Dissertation Presented to the Graduate School of the University of Florida in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy

SOME ASPECTS OF FIVE-AXIS MACHINE TOOL DESIGN, ASSEMBLY, AND TESTING

By

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Major Department: Mechanical Engineering

Five-axis machine tools are gaining broader usage throughout industry. The aircraft industry has always been a large user of five-axis machine tools for producing wing and landing gear components. These machine tools are generally large gantry type machines which operate at conventional feeds and spindle speeds. New trends in the industry are to use monolithic construction in the fabrication of aircraft components and to use high speed machining technology to produce them. The use of high speed machining has shown that there are significant improvements in the metal removal rates which can significantly reduce production times.
The following sections will present material to develop designs, point out many assembly details, and test new machine tools. The thrust of the work is to combine the technologies of five-axis machine tools with that of high speed machine. This will be realized in the design and construction of a five-axis machine tool. The machine tool will incorporates a number of new technologies in it design. One is the high speed/ high power high stiffness spindle design. Others include linear motors, open architecture controller, and chatter recognition and correction system.

This machine tool is to be used as a means to see how far the technology of producing thin walled and thin webbed structures can be taken. It will also be used to study tool paths for the cutting of five-axis parts.
CHAPTER 1
INTRODUCTION

It has always been the desire of aircraft manufacturers to reduce the time and cost required to produce their products. This has become of greater importance with the large reduction in military aircraft orders, which has occurred since the beginning of the 90's. It pushed the manufacturers of the defense aircraft sector into examining and using new technologies to improve their profitability. Competition in the commercial aircraft sector has given a similar incentive. One of the new technologies which is gaining wide acceptance in both sectors is high speed machining. High speed machining (HSM) offers a significant reduction of machining time. It has been found that implementing HSM in manufacturing of aircraft parts substantially reduces the production time [MDA94,SCH92].

In aircraft manufacturing it is necessary to produce large, light weight, integral structures which make up the airframe. At present most of these structures are fabricated from sheet metal parts which are held together with rivets. This method of manufacture requires a large quantity of tooling and is time intensive. The new technology finding wider acceptance as an alternative means of producing these parts has been the use of monolithic construction. Here the aircraft structural components, such as bulk heads, arches, doors,
avionics trays, are cut from solid billets of aluminum. This method of part manufacture has shown a significant reduction in the tooling and in labor costs, and production time. An additional benefit is an improved strength to weight ratio for the parts produced with this method. It is noted that this type of construction requires large quantities of material to be removed from the billet in production of the part. The percentage of material removed from the original billet can be as high as 80-90% in the production of some parts. To remove this quantity of material efficiently, high metal removal rates are required. This can be achieved by using high speed machining.

The geometries of these parts often have undercuts on the walls and floors as well as contoured surfaces. To produce these geometries on a milling machine would require either special fixturing or rotary axes, which can change the orientation of the cutter relative to the work piece. Note that special fixturing for work pieces is an added cost. The use of fixturing would increase the machine fixing time and the manufacturing costs. Further more, storage for the fixtures is required. The accuracy of the part is negatively impacted, because of its repositioning on the machine. These factors have given impetus to incorporating HSM technology into five-axis machine tools. This combination will achieve the level of metal removal rates (MRR) needed to produce the thin walled and floored structures. Additionally, the rotary axes in a five-axis machine tool will eliminate the need for special fixturing.

Several methodologies required to produce these monolithic structures have been developed [DVO96,RAO95,WIN95]. These methodologies make it
possible to produce thin walled and thin floor parts with thicknesses on the order of 0.5 mm (0.020"). The combining of HSM with five-axis machine tools will make it possible to have a flexible machine tool which can produce complex geometries with high MRR.

This work discusses the development, construction, and testing of the three-axis configuration of the five-axis machine tool. The dissertation is divided into the following sections: the review of the literature, kinematic combinations and parameters for design, machine tool assembly, testing and conclusions. The machine tool was designed and assembled in the Machine Tool Research Center at the University of Florida. This machine is a test stand for the combining and testing of machine tool technology and high speed machining technology.
CHAPTER 2
LITERATURE REVIEW

High Speed Machining

High speed machining, often referred to as high speed cutting, has been a topic of much discussion for many years. The application of HSM is seeing wider appeal throughout industry. It is being applied to the cutting of cast iron, hardened steel, titanium and aluminum alloys [TLU93]. HSM is considered as milling at spindle speeds such that the tooth passing frequency, $f_t$, approaches the dominant natural frequency, $f_n$, of the system. A general rule to determine high speed milling is when $f_t > 0.4 f_n$ [SMI88]. As an example, for a four fluted end mill with a dominant mode at 1100 Hz, $f_t = f_n$ means a corresponding spindle speed of 16,500 rpm. Many of the applications of HSM have an intrinsic lack of dynamic stiffness in the spindle/tool/work piece system. Examples of this would be the milling of deep pockets in aircraft structures, milling thin walls, or milling thin floors using long slender end mills. Thus it is important to have a good understanding of the dynamics and utilize methods to avoid chatter.

Schulz and Toshimichi indicate that HSM is a broad field. They define it as cutting at the highest possible speed for a given work piece/cutter material combination [SCH92]. Further more they point out the advantages of high speed
cutting. Such as increased machining accuracy and surface finish, reduction of burr formation, larger range of stable cuts, simplified tooling, and an increase of productivity. They go on to cover many of the requirements needed in modern machine tools so that the HSM may be fully realized. Research funded by McDonald Douglas Aircraft Company has show a two fold improvement of cutting time by utilizing HSM [MDA94]. Tlusty [TLU93] shows general agreement with Schulz and Toshimichi but makes more of an emphasis on the cutting dynamics and chatter. Tlusty shows that by using HSM it is possible to find spindle speeds where the axial depth of cut, \( b_{\text{ax}} \), can be increased several times beyond the critical depth of cut. To do this it is necessary to know the dynamic behavior of the cutting tool. He states that this may not always be practical. Another method to determine the best speed is to record the sound spectrum of the chatter when it occurs and then to extract its frequency. This frequency, \( f_n \), is used to avoid chatter by setting the tooth passing frequency, \( f_t \), equal to the chatter frequency or its integer fraction. This generally places the cutting process in a stable gap in the stability lobe diagram. This method has been fully realized in the chatter recognition and control system (CRAC) developed by Manufacturing Laboratories Inc. [MLI93]. There can be cases when two or more modes of similar magnitude exist which could lead to failure of the preceding method. This is known as competing modes. Smith et al. [SMI94] have developed algorithms which will avoid the problem of competing modes. Essentially, they find spindle speeds where the gaps for these competing modes coincide thus avoiding chatter. This algorithm works well for gaps below the \( N=0 \)
lobe of the lower frequency mode. There may be problems finding a coincident gap when looking at speeds in the region above the N=0 lobe for the lower frequency mode.

To implement HSM into a machine tool, many of the design issues must be taken into consideration. Tlusty points out a number of tasks which must be performed to fully to realize HSM in modern machine tool designs [TLU93]. He listed these tasks as design of high speed spindles, guideways, feed drives, controls, lightweight structures, and fast NC controllers. Schulz and Toshimichi listed similar design tasks and add the need to examine new designs for the tool/spindle interface, fixturing of the work piece, chip removal, and safety [SCH92]. The first task in both papers is the design of high speed spindles. They may employ integral motors, and hybrid angular bearing, hydrostatic bearings, or magnetic bearings. The spindle speeds should be in the range of 20,000 to 50,000 rpm. Available cutting power should be on the order of 1 kW per 1000 rpm. The spindle design has to be as dynamically stiff as possible. The dynamic stiffness should be constant over the operating speed range. The spindle-tool interface must be examined because of the high radial accelerations seen in the spindle.

The design of the guideways, drives, and structure must be examined because of the need for high accelerations and velocities in the axes. The CNC control system needs to meet the high demands imposed by these high feed rates. Furthermore, the handling of chips and coolant has to be taken into consideration because of the high MRR. Even safety precautions for protection
from chips, coolant spray, and cutter breakage must also be addressed.

One of the biggest demands of the design are the required velocities and accelerations. Tlusty [TLU93] and Heisel [HEI96] specify velocities in the range of 10-20 m/min (400 -1000 ipm) and accelerations of 2 g's. This high acceleration is required to secure the full feed rate in the shortest time. This is of great importance when producing small pockets. The structure of the machine must be able to handle velocities and accelerations and maintain its accuracy. It has been proposed by some authors that the beds of these high speed machines should be fabricated from polymer concrete because of its good damping characteristics and good rigidity. The moving structural components can be manufactured from composites, welded sheet metal, or honeycomb structures to reduce the weight that must be accelerated by the feed drives. This make it possible to utilize smaller drives.

**Multi-axis Cutting.**

Many parts in the aircraft structure implement three-dimensional surfaces. These parts can be cut using three or five-axis cutting operations. Three-axis cutting of sculpted surfaces requires the use of ball nose end mills. The use of a ball nose end mills is required to more closely approximate the three-dimensional surface. By using this type of tooling the surface of the work piece will show scalloping. These scallops are defined by their width and their depth [BAD94]. The scallop width depends on the step over and the cutter radius. The scallop depth is determined by the required tolerance. On another hand by five-axis
milling, the cutter can be maintained normal to the surface of the work piece. This gives a geometric advantage, allowing the use of standard end mills instead of ball nose end mills. The better geometrical approximation of the cutter to the surface produces wider scallops with the same tolerance as in three-axis milling, which improves the overall surface finish. This leads to a higher MRR and reduces the manufacturing cost.

Five-axis milling could be described as a general milling operation. All other milling operations are restricted by their lack of motions from the six degrees of freedom in the Cartesian coordinate system [DAM76]. These five axes are often realized by three translational and two rotational axes. Depending on the machine size and the work piece geometrical form, the rotational axes could be in the tool or in the table, or in both. This variation can be used to classified the five-axis milling machines by the number of the rotary axes which are found in the tool or in the work piece [EIS72, ESC72, SPU92]. There are three major groupings based on how the rotary motions are distributed in the machine. The first group contains machines where the two rotary motions are in the table. The second contains both rotary motions in the spindle/tool. The last has the rotary motions split between the table and the spindle. Each of these groupings has further sub-groupings which contain all the possible configurations.

Other possibilities for achieving the linear and rotary motions in the cutting operation have been shown in the form of hexapods [HEI96]. This alternative design is realized by utilizing struts and gimbal to produce the motions. This type
of machine has a small unconventional work volumes, which limits its applications when compared to a similarly sized five-axis machines. These machines are still in their infancy but are gaining wider interest.

The advantages of five-axis machining in cutting three dimensional surfaces does have an added complication in generating the tool paths. A number of authors have examined these complications. Most of the literature on five-axis machine tool kinematics is concerned with the development of the NC code generation for the cutting of three dimensional surfaces and contours. Jerard et al. [JER89] have examined methods to determine errors in sculptured surfaces. The objective of that work was to develop five-axis cutting strategies, which account for the offset of the tool and step-over size. Their method attempts to remove the errors due to gouging and undercutting. Rehsteiner [REH93] examined the milling of twisted ruled surfaces. This type of surface is well suited for five-axis flank milling. The author was interested in how these ruled surfaces intersect and how to fully realize the proper motion of the machine through the NC code. His work, like many of the others, looks at the tool path and geometry of the work piece. It does not address how the kinematics of the machine tool plays an important role in the motions of the machine. Takeuchi and Watanabe [TAK92] examined ways to generate collision-free tool paths for five axis-cutting. This is of great importance in order to prevent damaging of the machine tool and the work piece. They used solid models of the tool and the work piece to determine the interaction between the two during the cutting process.
All the above literature show the flexibility and benefits of five-axis machine tools. They also point out that generation of accurate tool paths is not an easy task.

**Testing and Evaluation of Machine Tools**

The standard utilized in the United States to evaluated the performance of machine tools is the ANSI ASME B5.54-1992 (Methods for the Performance Evaluation of Computer Numerically Controlled Machining Centers) [ASM92]. The Standard establishes requirements and methods for specifying and testing the performance of CNC machine centers. It is divide into six logical areas: general definition and machine classification, machine environmental requirements and responses, machine accuracy performance as a machine tool, machine performance as a measuring machine, machine cutting performance, and machining of test parts. This standard is used by manufacturers and end users as an acceptance test for machine tools.

Generally, acceptance tests of machine tools can be classified in two major groups. One of these groups is the direct evaluations of the machine accuracy. This evaluation test determines the systematic and random errors of the machine tool by using direct methods to determine the static, dynamic, and thermal behavior, the geometric accuracy of the axes and there positioning accuracy. The other group is the indirect evaluation of the machine accuracy. This group produces standardized test parts which are measured to determine the errors and general performance of the machine. These parts include the
errors which are caused by the machine, the cutting process, and the
environment. These parts can determine the positioning accuracy, parallelism
and orthogonality of the machine axes, linear interpolation behavior, circular
interpolation behavior, and the thermal drift. Besides the parts proposed in the
B5.54 there are other national and international recommended standard parts
like the NAS 930 [NAS91], the German VDI 2851 Blatt 3 [VDI86], and the
Russian GOST 26016-83 [ENI83]. All these standardized test parts may
determine, more or less, the same performance criteria as the B5.54. They are
useful for testing a machine with linear and incremental rotary axes, but they can
not be used to determine the behavior of the continuous motion rotary axes as
founded in a five-axis machine.

The foundations of the evaluation and acceptance testing starts with the
work of Schlesinger [SCL27]. He developed methods to determine the
gеometric accuracy of machine tools. Tlusty [TLU59] advanced methods to
quantify the abilities of machine tools. He lists five main qualities which
characterize a machine tool. These are accuracy, output, life, convenience,
safety, and economy. Each of these represents a complex list of information
about the machine tool’s performance. He points out that often manufacturers
disperse a limited amount of information on their machines. Generally this
information is in regard to the dimensions, weight, speeds, feeds, and power.
Much of the information related to actual ‘qualities’ of the machine is not
presented and for that matter is often not known by the manufacturers. He
points out the importance of testing the machine tool to see how its abilities
compare to others which have been similarly tested. The paper goes on to present a number of methods to be followed to test machine tools. Tlusty and Köenigsberger [TLU70] presented a very comprehensive report which covers the preceding material in much more detail. The UMIST report covers all the previously discussed categories explaining their importance and how to make the measurements in complete detail. Much of the material making up the testing methods can be found in the Technology of Machine Tools Report and supplements present by the Machine Tool Task Force [MTT80] as well as the UMIST report. These standardized tests give the manufacture and the end user a means to compare various machines in a given class.

The literature reviewed shows that many of the areas for implementing HSM into five-axis cutting have yet to be covered. This is especially true for the testing of five-axis machine tool performance. Further more in all the machine performance tests, no material was found to test and evaluate five-axis machine tools. Particularly, this is true for determining the accuracy of the machine’s rotary axes. The influences of five-axis kinematics on the dynamic behavior of the cutting process and the work piece accuracy also needs to be researched. The literature points out that there is a large need to develop interactive CAM systems to produce tool paths that incorporate collision avoidance and allow for dynamic feed control and technological issues. There has been work examining the accuracy of the NC codes but little or no cutting of test parts has been performed to determine which of these techniques provides the best solution. The machine tool described here will provide a test stand to examine how these
technologies work together and provide a means to develop the new technologies required to fully realize a production machine tool incorporating these technologies. The work presented in the following pages covers the development and construction of such a test machine.
CHAPTER 3  
KINEMATICS OF FIVE AXIS MACHINE TOOLS

Kinematic Combinations

A machine tool is an assemblage of prismatic and revolute joints which move in concert to bring the cutting tool to a desired location within the work volume. This is true for the smallest, manually controlled, bench top mill as well as the largest, computer numerically controlled (CNC), milling machine. A grouping of these kinematic elements form the machine tool kinematic loop. Actuation of these joints produces the machine motions. The kinematic loop is often described by the kinematic elements from the work piece moving along the structure and joints towards the cutting tool. The loop is closed by the cutting tool/work piece interface. The arrangement, type, and number of these joint elements has a significant impact on the types of parts which can be machined on a given machine tool.

This section focuses on the kinematics of five-axis machine tools. For the purpose of this chapter, a five degree of freedom (DOF) machine tool is a system having five single DOF joints connected in a serial chain. A serial linkage machine contains joint elements that are connected in a linear fashion where the end of one joint is connected to the start of the next in the chain. In this
configuration each of the individual joints produces a motion in its specific single DOF without any movement of the other joints. This differs from a parallel linkage machine which requires multiple joints elements to be actuated for motion in one DOF. A good example of a parallel machine tool is the Hexapod milling center being developed at Ingersoll Milling Machine Company, Rockford Illinois. This section focuses on serial rather then parallel machines.

On machine tools, a systematic method to label the direction and orientation is required in order to standardize programing of NC machines. To distinguish one axis of motion from another an internationally standardized system of letter addresses is employed. The letter address system which is used can found in the three following standards: EIA RE-267-B, issued by the Electronics Industries Association; the AIA NAS-938, issued by the Aerospace Industries Association; and the ISO/R 841, issued by the International Organization for Standards [MEC92]. Each of these three standards is in full agreement with the others.

The lettering system used for machine tools is an orthogonal “right-hand" Cartesian coordinate system (CS). This coordinate system describes the orientation as well as the direction of motion. The coordinate system could be attached at various points on the machine tool, such as the tool point or the work piece. The translation motions generally produced by prismatic joints, are designated as X, Y, and Z. Figure 3-1 shows the orientation of a Cartesian coordinate system which is attached to a typical table or pallet. The Z' axis is oriented such that it points normal to the plane formed by the surface of the
The X' and Y' are orthogonal to each other and are in the plane of the table. The arrows point in the positive direction of motion. The primes indicate that these letter addresses are attached to a coordinate system on the table/work piece. A coordinate system attached to the spindle/tool would not have these prime marks. The rotary motions are designated as A, B, and C. The rotary motions are orientated with respect to the translation axes of motions. The A rotary axis is about the X axis; the B rotary axis is about the Y axis; and the C rotary axis is about the Z axis. The positive direction of rotation follows the right-hand rule.

The three translation and the three rotation axes form the six degrees of freedom which can fully describe all motions of any rigid body. A five-axis machine tool utilizes five of these six DOF to position the tool relative to the work piece to obtain the desired geometry during the cutting operations. A base reference coordinate system can be located anywhere on the machine tool or in space in order to analyze the motions on the machine. Since the motion of the work piece with respect to the tool or vice versa is the concern here, reference coordinate systems are attached to the table/work piece and the spindle/tool. Either of these coordinate systems can be used to fully characterize the motion of the machine tool. It will be noted that the coordinate system attached to the table/work piece is used as the reference system for NC code generation. The two coordinate systems are ideally oriented so that the axes are parallel to each other. Figure 3-2 shows the general arrangement of these two coordinate systems. These two six DOF coordinate systems will be used to examine the
various possible kinematic configurations that can be incorporated into the design of a five-axis machine tool. The combinations of motions for a five-axis machine tool examined here will be limited to combinations with three prismatic joints and two revolute joints, since this is common arrangement for a five-axis machine tool.

It will first be necessary to determine the total possible number of combinations for a five-axis machine tool given the three prismatic and two revolute joints. It should be noted that the motion axes could be attached to the spindle/tool system or to the table/work piece system or divided between the two systems. The number of combinations which can be found in the most general terms is determined using basic combination mathematics. The fundamental equation is

\[ \binom{n}{k} = \frac{n!}{k!(n-k)!} \]  

where \( n \) is the number of items in the group to be selected from, and \( k \) is the number of items to be selected each a time. It can be shown that the total number of kinematic combinations is made up of a product of the minor combinational groups. These minor groups are the number of combinations for the two revolute joints, the number of combinations on how the revolute joints are distributed between the two coordinate systems, and the number of combinations of the three translation motions divided between the two coordinate systems.
Figure 3-1 Right hand cartesian coordinate system
Figure 3-2 General arrangement of the two coordinate systems
The first minor group of combinations is the number of revolute joint combinations that can be selected from the three possible axes of rotation. This one is simple to see by inspection, but will be a good example to see how the above equation is applied. The number of items, \( n \), in the group is 3. The number of items, \( k \), to be selected from the group is 2. The equation for the possible revolute joint combinations is thus

\[
C_{R1} = \binom{3}{2} = \frac{3!}{2!1!} = 3
\]  

(3-2)

The three possible rotary combinations are AB, AC, and BC. These three possible combinations of rotary axes can be in either of the two coordinate systems. The two rotations can be in the spindle/tool system or the table/work piece system or split between the two coordinate systems. Thus the minor group of combinations which is related to how the rotational axes are distributed between the two coordinate systems is the sum of possible combinations divided between the two coordinate systems. The size of the group is 2. The number of items to be selected varies from 0 to 2. This variation comes from distribution of the rotations from all the rotations are in one coordinate system; split between the two systems; and all rotations are in the other coordinate system. The resultant summation is

\[
C_{R2} = \sum_{k=0}^{2} \binom{2}{k} = 4
\]  

(3-3)
Now the combinations of the translation motions can be determined. Since all three translation motions will be selected, there is only one combination of the translation joints, X, Y, and Z. Next, the number of combinations due to distribution between the two coordinate systems needs to be determined. The same method is applied as was used to determine the number of combinations of the revolute joints distributed between the two coordinate systems. The group size is \( n = 3 \) and the number of items to be selected varies from \( k = 0 \) to 3. The resultant summation is

\[
C_T = \sum_{k=0}^{3} \binom{3}{k} = 8
\]  

(3-4)

The total number of possible combinations is determined by taking the product of the three combinational groups. The equation for the total is

\[
C = C_{R1} C_{R2} C_T = 96
\]  

(3-5)

These are the 96 combinations which are possible in the most general terms.

The next level analysis will show that 50% of the total number of combinations is either redundant or not functional for application in a machine tool.

The first examination will be for any similarities in the motions. Looking at the combined rotary motions AC and BC or A'C' and B'C' shows that they produce the same motions. This motion combination is often called a nutating motion. The A and B axes of rotation are in the same plane. The only difference is that these two axes of rotation are orthogonal. Though the orientation of the
tilting motions differs by 90° the resultant motion is the same. Figure 3-3 and 3-4 show how the motions in the two configurations are similar. This observation removes 32 combinations from the total number of theoretical possibilities. The next rotational combination to be checked is the AB rotary joint combinations where the two rotations are divided into each of the coordinate systems. This would be AB' or A'B rotary combinations. Similar motions can be produced if the A is in the tool/spindle system and the B is in the workpiece/table system or vice versa. Figure 3-5 shows how the two rotations distributed between the two coordinate systems produced the same resultant motions. This removes an additional eight combinations from the total. Next the case was examined where the C axis of rotation was the lone rotation on the tool/spindle side. This rotational motion has no net resultant effect to the tool orientation. This case simply rotates the spindle in the same axis of rotation as the tool. This removes an additional eight combinations from the total. Thus the total number of combinations which can practically be selected from are reduced from 96 to 48. Table 3-1 lists the possible rotational and translation combinations.

The combinations in Table 3-1 is in agreement with the generalized kinematic model presented by Rüegg [RUE92]. The work of Takeuchi and Watanabe is also in agreement with the above number of feasible structural arrangements for five axis machining centers. That paper goes on to further subdivide five axis machine tool configurations into three subdivisions.
Figure 3-3 AC & BC head configurations
Figure 3-4 A'C' & B'C' Rotating and Tilting table combinations
Figure 3-5 AB' & A'B Tilting spindle and Tilting Table Combinations
TABLE 3-1  MACHINE KINEMATIC COMBINATIONS:

<table>
<thead>
<tr>
<th>ROTATION COMBINATIONS</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>A'C'</td>
<td>AC'</td>
</tr>
<tr>
<td>AB</td>
<td>A'B'</td>
<td>AB'</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TRANSLATION COMBINATIONS</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>XYZ</td>
<td>XYZ'</td>
<td>XY'Z</td>
<td>XY'Z'</td>
<td>X'YZ</td>
</tr>
</tbody>
</table>

The three subdivisions are broken down according to how the 2 DOF of rotation distributed between the 2 CS. The first subdivision has both rotary motions in the table CS. There are two possible arrangements for this subdivision. Type 1 is an arrangement where a tilting table has a rotary table attached to it. Type 2 is a rotary table arrangement with an attached tilting table. Figure 3-6 shows these two types of table arrangements for this subdivision. Similarly, the second subdivision, having both rotary motions in the spindle CS, has two types of arrangements. Type 1 has two tilting axes, and is often called an AB head. Type 2 has one tilting axis and one rotating axis and is often called an AC head or nutating head. Figure 3-7 shows these two rotary head configurations. The third subdivision has the rotational motions divided between the 2 CS. There are two types in this subdivision as well. Type 1 is a tilting head spindle arrangement with a rotary table. Type 2 is a tilting spindle with a tilting table. It should be noted that the rotational joints are commonly located at the beginning or end of the kinematic loop.
The placement of the rotary motion in between translation joints would make the kinematic solution significantly more difficult to solve when developing the postprocessor to control the machine motions.

Parameters for the Selection of Combinations

The 48 combinations can all be used for the layout of a five-axis machine tool. To select an appropriate kinematic configuration, a number of different parameters need to be examined. These parameters will dictate which of the combinations is more practical to use than the others for a given application. These parameters were divided into four main groups. These groups are the work piece specifications, the types of cutting operations, the machine performance specifications, and the productivity of the machine tool specifications.

The first parameter to evaluate in the work piece specification group is the size of a typical work piece to be cut on the machine tool. The characteristic length, height, and width of typical parts to be cut on the machine are needed. For example, tilting and rotating table machines may not be practical for work pieces with large characteristic lengths and widths. It may be more practical to have the spindle move around the work piece. The next parameter, of equal importance, is the geometry of the work piece. The work piece geometry may be classified as either prismatic or rotary. An example of a prismatic work piece is an engine block.
Figure 3-6 Tilting table with rotary top and Rotating table with tilting top
Figure 3-7 AB Tilting Head and AC Nutating Head
Examples of a rotary work piece are turbine buckets or ship's propeller. A tilting table and tilting spindle arrangements are good for prismatic geometries. These motions can be used effectively on the planar surfaces seen in such parts. A rotary table configuration is better for rotary geometries where an axis curvature of the part could easily correspond to an axis of rotation on the machine. The form and the shape of the whole work piece will affect the size and shape of the work volume. Another geometric concern is that of the greatest angle of the under cut required in the work piece and the largest angle in contours. This will dictate the range of angular motion required in the rotary axes. The next parameter is the cutting strategies to be employed in the cutting operation. This is a function of cutter orientation and the type of tooling to be used. A parameter which is often not considered is the material of the work piece. The weight of the work piece is a vital parameter that has an effect on the design of the machine tool and must be considered.

The cutting operations of the machine tool are the next group of parameters to examine. The first parameter in this group is the desired power of the spindle. The power requirements of the spindle will govern the size of the drive and the type of drive to be used. The spindle may be of an integral motor design or utilize a drive train with any type of prime mover. The size of the drive and spindle will affect how it will be incorporated into the machine's structure and affect selection of a tilting head or tilting table configuration. The next subgroup is the types of cutting operations to be performed on the work piece. These cutting operations generally are milling, drilling, boring, tapping, and
grinding. This could affect the power requirements of the axes. The type of cutting technology to be employed by the machine also needs to be considered. The two technologies are high speed milling and conventional milling. This will dictate spindle configuration and dynamic stability considerations. The type of cutting tools to be employed and their geometry should be considered. The final parameter to be considered in this group is the type of cooling to be employed. These could vary from dry, mist spray, flood, or high pressure cooling. This will affect the amount of coolant that will be required and how the coolant and chips from the cutting process will be handled. The incorporation of the coolant and chip handling systems can affect kinematic selection.

The third group is that of the machine performance specifications. This general group covers the parameters desired in the machine tool motions and structure. The first parameter is the desire acceleration and velocities. The desire for higher production rates has pushed accelerations and velocities in the axes of motion higher. Higher accelerations require that the machine have high static and dynamic stiffness. The selection of the proper configuration to obtain the highest dynamic stiffness for the high speed, high acceleration cutting is a trade off. In general to achieve the high stiffness massive structures are required, but that gives rise to the obvious problem of having to accelerate these massive structures. Extensive analysis of the structures using finite element analysis is needed to obtain the optimal design. The last parameter in this group is the material to be used in the construction of the structural components making up the machine tool. Material such as cast iron and welded steel are
conventional materials used in many of today’s machine tools. Other materials which are finding more usage are epoxy concrete, granite, and composites materials such as carbon fiber. Cast iron is strong and has good inherent damping but requires large cross sections to obtain the same strength as steel. Granite is a good foundation material, but care must be taken to keep it in a dry environment. Moisture is absorbed into the granite causing it to swell which could cause distortion in the machine structure. Epoxy concrete is also a stiff material with good damping characteristics yet requires large sections to achieve the needed strength. Composites have high strength to weight ratios but there are may complications due to problems of creep and fiber orientation. The cost of composites is also high.

The last group relates to the productivity of the machine. The parameters in this group deal with how the machine tool would be integrated into production of the typical workpiece. The first parameter is the machine’s integration in the manufacturing facility. The machine tool could be part of a production line or it could be part of a flexible machining cell. The integration into the factory floor depends on the volume of parts to be produced and the desired duty cycle of the machine. The amount of automation incorporated into the machine may include an automatic tool changer or pallet changer. This will affect the layout of the machine and its structure so that these subsystems can be incorporated into the structure of the machine tool. The number of simultaneously operating axes should be considered. Some of the axes on the machine tool maybe strictly for positioning while others will be active during machining. This could require
braking or position locking devices to be incorporated. The number of spindles incorporated into the machine's design for cutting operations. Some machine tools utilize interchangeable spindles to obtain the best characteristics for different speed ranges. While others utilize multiple spindles in the cutting operation.

An example of how parameters are weighed and used in the selection of the most applicable kinematic arrangement of the machine motions will be shown in the next section. The example will be for the design of the five axis machine tool recently constructed in the machine tool laboratory.

MTRC's Five-Axis Machine Tool

The five-axis machine tool to be constructed in the laboratory is primarily a research machine. This machine will be used to test new technologies and new methods in high speed machining. It is noted that the analysis presented here is after the fact. The machine was well into its design at this writing. It is used as an example to show how these parameters may be used in the design process. The four groups of parameters were weighed given the primary purpose of the machine tool. Since the machine tool was not a production machine the group of parameters concerned with production was given secondary status to other design considerations. The work piece specifications were examined first. The work pieces to be cut on this machine tool are primarily
monolithic aircraft components milled from 4-6 inch thick aluminum billets. The parts will have a number of pockets with thin walls, and thin floors. The thicknesses of the thin walls and floors cut will be 0.5 mm (0.020 in). A high metal removal rates is of great importance because up to 80% of the billet will be removed in the production of these parts. The characteristic length of the work piece, height or width is generally up to five times or greater then the depth. Table 3-2 shows a listing of some typical aircraft parts used to select the machine configuration.

**Table 3-2 TYPICAL AIRCRAFT PARTS:**

<table>
<thead>
<tr>
<th>Part</th>
<th>Material</th>
<th>Dimension</th>
<th>Max. Depth</th>
<th>Method</th>
<th>Tool</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slide</td>
<td>7050-T7</td>
<td>45x17x3.5</td>
<td>1.5</td>
<td>5-axis</td>
<td>0.75x1.5</td>
</tr>
<tr>
<td>Housing</td>
<td>7050-T7</td>
<td>40x37x3.2</td>
<td>3.06</td>
<td>5-axis</td>
<td>0.5x2</td>
</tr>
<tr>
<td>Housing</td>
<td>7050-T7</td>
<td>40x40x3</td>
<td>1.55</td>
<td>5-axis</td>
<td>0.5x2</td>
</tr>
<tr>
<td>Rib</td>
<td>7050-T7</td>
<td>13x9.5x2.2</td>
<td>1.32</td>
<td>5-axis</td>
<td>0.75x1.5</td>
</tr>
<tr>
<td>Rib</td>
<td>7050-T7</td>
<td>24x13x4</td>
<td>1.84</td>
<td>5-axis</td>
<td>0.75x1.5</td>
</tr>
<tr>
<td>Floor</td>
<td>7050-T7</td>
<td>33.5x30x2</td>
<td>1.81</td>
<td>5-axis</td>
<td>0.75x1.5</td>
</tr>
<tr>
<td>Gear flap</td>
<td>7050-T7</td>
<td>29x17x2.5</td>
<td>1.4</td>
<td>3-axis</td>
<td>0.75x1.5</td>
</tr>
<tr>
<td>Door</td>
<td>7050-T7</td>
<td>34x29x4.5</td>
<td>1.73</td>
<td>5-axis</td>
<td>0.75x2.25</td>
</tr>
<tr>
<td>Door</td>
<td>7050-T7</td>
<td>30x10x1.2</td>
<td>1.08</td>
<td>5-axis</td>
<td>0.75x1.5</td>
</tr>
<tr>
<td>Guide</td>
<td>2219-T8</td>
<td>42x23x4</td>
<td>1.85</td>
<td>5-axis</td>
<td>1x2</td>
</tr>
</tbody>
</table>

Note: All dimensions are in inches.

These characteristic lengths make it desirable to have large ranges of motions in the X and Y axes. The geometry of the work pieces is prismatic. Some of the thin walls are angled requiring under cuts. Thus it will be possible to use either
tilting table or tilting spindles type configuration. The angular motion requirements are less than \( \pm 45^\circ \) from the normal. The materials of the work pieces will mostly be 7050-T7 and 7075-T6 aluminum, although consideration for the cutting hardened steel and titanium has been discussed. No geometric variations should be necessary to cut these two hard materials.

The cutting operations to be performed on the machine tool will be milling. The spindle to be employed is designed for milling operations only. The spindle is of a compact design with an integrated motor. A number of attachments for compressed air, lubrication, coolant, and power must be incorporated into the design. These many connections would add significant complexity to an AC type head design configuration. The spindle is designed for high power/high speed milling. The spindle is rated at 37 kW at 36,000 RPM. The design of the spindle supporting structure will require high dynamic stiffness. The cutting tools to be used on the machine will be primarily carbide end mills. Table 3-2 shows some of the tool diameters and lengths to be used in the milling of the work pieces. Due to the use of a high speed spindle, face mills will see very limited use. Due to the high metal removal rates to be achieved by the machine tool, an important design concern will be the removal of the hot chips from the work piece. High speed milling produces a large quantity of chips in a short period of time. The metal removal rates will be on the order of 200 cubic inches per minute, assuming 80% of full spindle power is utilized. The orientation of the table is an important consideration for the removal from the work piece. A horizontal table will have the problem that the chips will accumulate on the work piece. To
alleviate this problem a vertical orientation of the table is used. This allows gravity to remove the bulk of the chips from the work piece. Accommodations will needed to remove the high volume of chips from below the vertical table. The machine tool will utilize mist, flood and possible high pressure cooling systems. This will give the machine the added flexibility for performing various research projects.

The third group of parameters had a strong influence on the kinematics of the machine. The machine was to have linear accelerations on the order of 2 g's (19.62 m/s²) and linear cutting velocity of 25.4 m/min (1000 ipm). The high acceleration and velocity are needed to best utilize the positive aspects of high speed milling. Most of the parts have a large number of pockets requiring a large number of accelerations in and out of the corners of the pockets. The high linear acceleration will significantly reduce the time required for decelerate and accelerate in and out of the corners. Typical production machines on the market today have accelerations on the order of 0.5 g's (4.9 m/s²). The axes will achieve full speed in one quarter of the time. This enables the cutting operation to have a higher overall average feed rate and thus reduces the machining time. The higher accelerations seen in this machine required high-power servo drives. The X and Y motions are compounded and will be operated at high velocities and accelerations. The moving structural components were designed with the thought to minimizing the mass to minimize power requirements, and yet having sufficient stiffness to meet dynamic stiffness needs. The acceleration requirements made having a rotary or tilting table attached to the X and Y
motions less desirable due to the large mass associated with the components which make up the rotary axes. Attachment of the spindle to the X-Y motion was not desirable due to the weight of the spindle. The spindles mass was greater than that of a typical workpiece. This lead to the attachment of the work piece to the X and Y motions. This compounded axes arrangement minimized the mass to be accelerated, thus reducing the size of the drives and associated equipment required. This leaves the configuration of the rotary motions. Two possible designs were considered. One was a nutating head design and the other was a tilting head design. The tilting head design was selected for it could more readily be fitted with the high speed spindle. The many complexities of getting the existing spindle to fit into a nutating head made this design less attractive. The tilting head also had the added benefit of having a smaller footprint. The two rotary motions are attached to the Z axis. This makes the mass to be moved by the Z large. This is not a concern for the motions of the Z-axis do not require the same kind of performance needed in the X and Y axes. This comes from the fact that the work pieces are generally flat having a characteristic length of less than 102 mm (4").

The materials for the structure of the machine tool are limited to cast iron, steel and composites. The use of granite or epoxy concrete was considered. It would have been possible to use it for the bed of the machine, but facilities to handle the epoxy concrete and granite where not available. There where also structural limitation of the building which had to be considered. The moving structural components require dynamic high stiffness and low mass to minimize
power requirements in the drives. This would make for composites the first chosen in materials for the machine tools structural running gear. The cost associated with composites made use of these materials prohibitive. Estimates where that cost would be more then two times greater then conventional materials. This left cast iron and steel. Cast iron parts and castings would be difficult to obtain at a reasonable cost and had a higher mass the strength ratio then steel. The use of welded steel construction was decide upon because of the good stiffness characteristics and ease of manufacture.

The final configuration of the machine tool is seen in the figure 3-8. The machine has a compound translation X-Y axis. Each having 28 inches of motion. The table is mounted in a vertical position to use gravity to help remove the chips and coolant from the work piece during cutting operations. The Z-axis carries the tilting AB head spindle assembly. The AB head has angular motions of ±45° in both directions of rotation. The X and Y axes are driven by brush-less DC servo motors attached to ball screws. It should be noted that the X-axis has two drive assemblies associated with it. The Z-axis has two linear servo drives. The A and B rotary motions are performed by split-worm gear arrangements driven with brushed DC servo motors. The structural components are of welded steel. The bed is a 9 ½ ton weldment forming a rigid base for the machine. The moving structural components use tubular construction to reduce mass in the moving X-frame and the Y-table.
Figure 3-8 Final machine tool configuration
CHAPTER 4
PRACTICAL ASPECTS OF MACHINE TOOL ASSEMBLY

The assembly of machine tools is a trade which has traditionally been handled by craftsmen. Engineers are involved with the design but often have little contact with the machine during the assembly stage except when problems arise. From a technical writing point there is very little material discussing many of the practical aspects concerning the assembly of a machine tool. Much of that which is written is not in the public domain. The work presented in this chapter shows some of the practical points of the assembly process used in assembling the five axis machine tool. Note that it only goes through the assembly of the three translational axes because the design and manufacture of the AB head was not completed at this writing.

Installation and Leveling of the Machine Bed

The assembly of the machine tool began with the preparation of the machine bed. The bed arrived from the manufacture with dirt and rust on many of the ground surfaces. The bed was cleaned and painted. All of the rust and dirt was removed from the ground mating surfaces and a protective layer of
grease was applied. The bed was moved into position, using a set of rollers, called tanks. Once the bed was in position, the rollers were removed and the bed was placed on six Unisorb® Model LL-7 resilient mounts. Each mount has a load-bearing capacity of 3175 kg (7000 lbs.). The resilient mounts are sandwiched between Titan™ shock pads fitted between the foundation and the bed. These pads are a textile based laminated neoprene material which isolates the machine bed from the foundation. These pads were selected to give the machine sufficient support and to provide a means to absorb the energy produced during the high accelerations and deceleration of the axes and minimize its transmission into the foundation.

The leveling of the bed was done using a hydraulic jack and a set of Federal electronic levels to measure angular displacement. The angular measurements and profiling of the bed base was performed following the procedure laid out in the handbook supplied by the electronic level manufacture [FED01]. The angular measurements determine the angle the bed sits relative to the gravitation normal. The profiling of the bed determines if there is sagging, twisting, or other deformation of the machine bed. The bed was brought into level using the hydraulic jack and adjustment of the resilient mounts. The leveling of the bed was to minimize the amount of pitch and roll from the gravitational norm. This helps distribute the weight of the machine over the six resilient mounts and the foundation. The profiling measurements looked for any twisting or sagging of the bed. The measurements were made on the ground mating surfaces where the pedestals would be placed. No significant twist or
sag was measured in the structure. Measured value of twisting was 0.0005 mm (0.00002") variation in the longitudinal direction, and 0.0013 mm (0.00005") in the transverse direction. Similar magnitude of values was measured for the sag. These may not be sag or twist of the bed, but could also be attributed to minor errors in the final grinding of the mating surfaces. Since the values measure were so small it was considered unnecessary to look any further into their cause. Once the bed was set and leveled, the rail systems and moving components were attached.

Mounting of the X-Axis Rail Systems

The first components to be attached to the bed were the X-axis rails. The machine tool uses the AccuMax 55 rail-carriage system manufactured by Thomson for the X-axis. These rail systems were selected due to their high load carrying capability, high life cycle rating and compact size [THO95]. Three rails, each with two carriages, were used on the X-axis. The mounting of the rails, on the vertical wall of the bed, began with preparing the ground surfaces on the bed and the rails. All the protective coatings, dirt, and grease were removed to permit uniform contact between the two mating surfaces. The order of the installation of the rails began with the bottom rail, and moved up the bed’s vertical wall. All the threaded holes for fastening the rails to the bed where chased with a M14x2 tap to remove any burrs and debris. The tapped holes were blown clean of any debris. This helps in obtaining a uniform pre load on
each of the fasteners when they are tightened to their final torque. The fasteners used to mount the rails are M14x2 class 12.9 metric socket head cap screws. When mounting the rails care was taken to protect the precision surfaces against inadvertent impacts which could have damaged them. Note that the individual rails each weigh over 36 kilograms (80 lbs.). Sufficient manpower was used to safely handle the rails as they were attached to the bed wall. The socket head screws threads were given a coating of a nickel-based anti seize paste before being threaded into place. The anti seize was used to help prevent the possibility of the machine screws becoming seized in place with time due to corrosion or any galvanic action. This will make any disassembly in the future easier and will improve the uniformity of the preload. The cap screws have been put in place and tightened until they were finger tight. They were left finger tight until after the initial alignment of the rail was performed.

The primary concern during the alignment of the rails was to get the rails parallel to a reference edge, minimizing any bending or curvature. The reference edge was utilized as the datum for all the components on the X-axis. The reference edges used were the alignment edges machined into the bed wall for the alignment of ball screw bearing housings and pillow blocks. A precision beam was suspended on these alignment edges and the secured to the structure. This beam was used as the reference artifact for the alignment of the rail. Figure 4-1 shows the arrangement of the precision beam set on the alignment edges above the rail. Precision gage blocks are used to set the distance from the rail to the precision beam. The distance between the flank of
the rail and the beam was 49.53 mm (1.950"). A stack of precision gage blocks was used to produce the desired spacer. The blocks were wrung together to form the measuring stack. The rail was adjusted up or down to obtain the proper distance using a wood lever. The rail was considered to be in position when the stack of precision gage blocks fitted snugly between the precision beam and the rail. Note that the stack of blocks could still be moved freely yet some resistance could be felt. This was important point. To prevent any deformation or misalignment it was important not to apply so much force to the rail, that the gage blocks became pinched and could not be moved. The alignment procedure was started at one end of the rail. The stack of blocks was placed, and the rail is moved into position. The stack of blocks was moved regularly to check that they had not become bound between the rail and beam. Once the proper gap was set, the adjacent cap screw was tightened to hold the rail in position. The stack of blocks was then moved along the rail to a position two cap screws down from the preceding position. The rail was again moved into position until the stack was snug and the cap screw was tightened. This procedure was followed along the total length of the rail. Once this was completed, the direction was reversed and the rail was aligned as before at those locations which had been skipped. Once the rail was initially aligned, the first time, the stack of gage blocks was moved along the rail to check for any needed adjustments of the rail alignment. This procedure was repeated until the rail had obtained a nominal alignment.
Figure 4-1 X-axis rail alignment
The nominal alignment being that the gage block stack had a uniform snugness between the rail and the beam along the full length. Upon completing the alignment of the rail all of the socket head cap screws were torqued to 100 Nm. The cap screws were tightened starting from the middle of the rail, and working out to the ends alternating left to right. Every other cap screw was skipped. Once the ends where reached the direction was reversed and the cap screws which had been skipped where tightened to their proper torque. This method of tightening the socket cap screws was intended to give the rail/wall joint a uniform and balanced pre load along its length.

A final check was made of the alignment to make sure no movement occurred during the final tightening.

Once the bottom rail had been attached and aligned, the next two rails were attached and aligned. The same procedure was used to attach the two other rails, but the alignment procedure was different due to the distance between the rails and the desire to use the bottom rail as the datum. The selection of using the bottom rail as the datum would mean that each of the three rails would be parallel to the others. To align the upper two rails, a measuring jig was constructed. The jig consisted of a 25.4 mm (1") square rod 609 mm (24") in length attached to a 165.1 mm (6.5") by 139.7 mm (5.5") by 25.4 mm (1") block. This assembly was attached to one of the rail carriages which rides on the bottom rail. This jig gave a stiff attachment point to which a dial indicator with a 0.0127 mm (0.0005") resolution was attached. The same procedure was followed to align the middle rail to the bottom rail. The only difference was that
the measuring jig and dial indicator was used instead of the precision beam and the precision gage blocks. Figure 4-2 shows the arrangement of the jig and dial indicator on the bottom rail carriage. Once the alignment and final tightening of the cap screws was completed the measuring jig was moved to the middle rail so that the top rail could be attached and aligned using the same procedure.

Upon completion of the alignment of the rails a set of final measurements was made to examine the variation of the parallelism of the rails to each other. It was found that the largest overall variation between the rails was less than 0.05 mm (0.002") over the 2 m (78.74") run of the rails. The carriages were placed on the rails. To protect the rails and carriages a layer of oil was applied and a protective covering was placed over the rails and carriages to protect against corrosion and accidental impacts.

**X - Axis Ball Screw Assembly, Mounting and Alignment.**

The X-axis has two ball screws to drive the axis. The screws are situated between the three X-axis rails. The ball screws are 60 mm diameter with 20 mm pitch and were manufactured by Thomson Saginaw. The screws were removed from the shipping crate and all protective plastic and coatings were removed. Care was taken to protect the screw from any possible damage during assembly procedure. The installation of the ball screws began with the assembling the bearings and the housing on the driven end of the ball screw. The roller element bearings, seals, and spacers where all installed on the ball screw.
Figure 4-2 Alignment of middle X-axis rail
Note that the tapered-roller bearings are arranged to produce a cross to reduce axial displacement due to thrust loading. Figure 4-3 shows a cross section of the bearing housing displaying the component arrangement. A backing spacer was pressed into place by a bearing nut until the bearing assembly was snugly together. The nut was then tightened until 0.001" axial displacement was obtained. This applied the desired pre load to the taper-roller bearings of 1000 lbf. This pre load setting is equivalent to a bearing rolling torque of 58 in-lbf at 3.5 RPM. A locking nut was then spun up against the previous nut and tightened to prevent the pre load from backing off. The tapered-roller bearings were hand packed with an EP-II grease. The ball screw’s bearing assembly was then slid into the bearing housing. The grease port was aligned and the compression plate was set in place. The six socket head cap screws were taken up evenly around the compression plate to prevent any misalignment of the assembly in the housing. The cap screws were then tightened to a torque of 70 Nm. A feeler gauge was used to check the gap between the face of the housing and the compression plate to check for any misalignment. The ball screw housing assembly was now ready to be placed on the machine.

The screw assembly was lifted into place using a chain hoist. The assembly weighs approximately 90 kg. A lifting sling was placed around the screw where it exits the bearing housing. This was near the center of gravity allowing the assembly to be positioned horizontal with relative ease. The assembly was moved into position on the wall and the 5/8" fasteners were put in place to secure the assembly to the wall.
Figure 4-3 X-axis bearing housing assembly
The flank of the housing was placed against the alignment edge on the bed's mounting pad. The socket head cap screws were only tightened finger tight at this point. The pillow block was put in place at the free end. The alignment of the ball screw was begun now that all the components were in place.

To align the screw it was necessary to ascertain the longitudinal stiffness of the ball screw assembly. This information was needed to calculate the amount of sag at the free end, so the final placement of the pillow block could be established. This value was determined experimentally. The ball screw was modeled as a cantilevered beam which was fixed at one end and free at the other end and a point load applied at the free end. The equation for the displacement at the free end of this model is

$$\delta_{\text{point}} = \frac{PL^3}{3EI}$$  \hfill (4-1)

where $\delta_{\text{point}}$ is the displacement at the free end, $P$ is the point load at the free end, $L$ is the distance from the fixed end to the point load, $E$ is the modulus of elasticity, $I$ is the mass moment of inertia. An experiment was performed to determine the unknown value of the product of the modulus of elasticity and the mass moment of inertia (EI). This was be found using the equation (1) and a simple displacement test. A dial indicator on a magnetic base was attached to bed wall. The stylus of the dial indicator was set on the free end of the ball screw. A known mass was attached to the free end of the ball screw and the deflection due to its weight was measured. The amount of the deflection was used to
calculate the product EI. To calculate the sag the ball screw is now modeled as a cantilever beam with a distributed load. The equation for the calculation of the deflection is

\[ \delta_{\text{dist}} = \frac{wL^4}{8EI} \]  

(4-2)

where \( w \) is the distributed load. The distributed load \( w \) was the weight per unit length of the ball screw. Note that the ball nut was moved to the fixed end so that its impact on the deflection would be minimal. The diameter of the ball screw and the density of steel were used to calculate distributed load. This value and the calculated value of EI were used to find the sag. The value of the deflection was calculated to be 0.6604 mm (0.026") and was the amount which the center of the bore of the pillow block was offset from the center of the ball screw. The screw center was set low in the bore of the pillow block.

A post was screwed into the free end of the ball screw and a dial indicator was attached to the post so that the distance between the bore of the pillow block and the center of the screw could be measured. Figure 4-4 shows the post and dial indicator arrangement at the free end of the ball screw. This arrangement allowed for measurement of the vertical and horizontal offsets of the free end of the ball screw. The initial measurement of the bore position showed it to be 1.397 mm (0.055") offset in the horizontal and 1.346 mm (0.053") offset in the vertical. The horizontal offset was removed by placing steel shims between the pillow block and the wall.
Figure 4-4 X-axis ball screw alignment
The vertical was removed by adjusting the bearing housing's vertical position until the proper amount of vertical offset was read at the dial indicator. The vertical alignment of the pillow block was maintained using the ground locating edge on the mounting pad. When the pillow block was aligned and secured the ball bearing to support the free end was put into place. The backing ring was secured into position with the six cap screws. The bearing was then packed with grease. The ball screw assembly was rotated to check that it rotated freely.

The above procedure was repeated for the installation of the second X-axis ball screw. Both of the ball nuts were then given an initial charge of grease and spun back and forth along the length of the screws. This gave the screws a protective cover of grease and distributed the grease throughout the ball nuts. The couplings and servo motors were then attached.

**Placement of the Carriages and X-Axis Frame**

The X-axis frame is the structural component carried on the X-axis carriages which carries the running gear for the Y-axis. The X-axis frame is a weldment composed square tubes and plates. The X-axis frame was cleaned and painted in preparation for installation. The threaded holes on the mounting pads on the frame for connecting the X-axis carriages and Y-axis rails were chased and cleaned to remove any debris in the threads. The X-axis carriages and the mounting pads were wiped to remove any debris from the joint surfaces. The X-axis frame was hoisted into place using a chain hoist. The X-frame
weighs approximately 450 kg. (1000 lbs.). The structure was hoisted along the line of the center of gravity. Care was taken to prevent any damage to those components which were already attached to the structure. The X-axis frame was slowly moved toward the bed’s vertical wall and the carriages were positioned behind the mounting pads on the X-axis frame. Figure 4-5 shows the X-axis frame being hoisted into place. Once the X-axis frame was up against the carriages it was lowered down onto the alignment edges on the top set of mounting pads coming to rest on the top flanks of the top two carriages. M12x1.5 class 12.9 socket head cap screws were placed through the carriages into the X-axis frame mounting pads and tightened finger tight. The chain hoist was removed. A feeler gage was used to check that the alignment edges on the X-axis frame were up against the carriage flank faces. Then the cap screws were tightened starting from the top moving to the bottom. The X-axis carriages were given their initial charge of grease. The X-axis frame was then moved along the X-axis rails to see that there was no binding or areas of rough motion.

The ball nuts were connected to the X-axis frame at this point. A nut bracket serves as the intermediate piece between the X-axis frame and the ball screw’s nut. The brackets are attached to the flanges of the ball nuts. Care was taken to make sure that the attachment surfaces where clean of any debris. Spacer plates were fitted between the nut brackets and the mounting pads to fill the gap between the mounting pad and the bracket. This prevents any horizontal deflection of the screw.
Figure 4-5 Positioning the X-axis frame on to the X-axis carriages
The assembled X-axis frame, carriages and ball screws were moved back and forth along the length of travel by hand driving the ball screws. This was to check for any binding or misalignment of the assembly as it was moved through its range of travel. This completed assembly of the X-axis running gear.

**Y-Axis Rails Mounting and Alignment.**

The Y-axis is fitted with Accumax 45 rails and carriages. These carriage are similar to the ones used on the X-axis except they are of a smaller size. The rails were cleaned and prepared for mounting. A similar procedure was followed as was performed with the mounting of the X-axis rails. The rails were hung on the surfaces provided on the X-axis frame with socket head cap screws. The cap screws were tightened finger tight. The right-hand mounting pad on the X-axis frame has an alignment edge for positioning the rail. The flank of the rail was pressed firmly against this alignment edge using a clamp. Once the flank of the rail was snugly against the alignment edge the adjacent cap screw was tightened to hold the rail in place. This process was repeated moving down the rail until all the cap screws were tightened. The cap screw threads had been coated with anti seizing paste before being threaded. The cap screws were torqued to 100 Nm using the procedure as presented previously. The straightness of the rail was checked using a granite straight edge. No straightness errors could be seen. Once the rail was secured to the X-axis frame the squareness of the Y-axis to the X-axis rails was measured.
To make this squareness measurement a jig holding a precision granite square was placed the X-axis to act as the reference artifact. A dial indicator riding on a y-axis carriage was used to make the measurement. Figure 4-6 shows the arrangement of the square and the dial indicator on the X-axis frame. The square was positioned on the edge surface of middle rail’s right-hand carriage. The alignment surface on the carriage is approximately parallel to the X-axis rail. An adjustment screw, with a ball end, supports the square kinematically. This prevents any misalignment of the base of the square with the flank surface of the carriage. The dial indicator’s position was adjusted to bring the stylus into contact with the vertical surface of the precision granite square. The Y-axis carriage was moved along the Y-axis to check the squareness. The initial measurement showed a significant yaw, relative to a line perpendicular X-axis. The measurement showed a yaw of 0.00204 rads (7 arcmin). To remove this yaw the X-axis frame’s position was adjusted with shims placed between the alignment edges on the top mounting pads and the flanks of the two top X-axis carriages. This required that all of the cap screws securing the X-axis carriages and nut brackets be loosened to prevent any distortions or misalignments to the running gear. The amount of shim to be place was approximated by performing a simple trigonometric calculation of the rise over the run. The X-axis frame was lifted off the alignment edges using a hydraulic jack placed at its base and the shims were put in place. The hydraulic jack was then lowered and the bolts to the carriages were re-secured. The squareness was measured again. The process was repeated until the error was less then 3.57e-5 rads (0.123 arcmin).
Figure 4-6 Alignment of Y-axis rail and squaring to X-axis
This procedure brought the X and Y axis into the range of perpendicularity which could be measured with the equipment used.

The second Y-axis rail was mounted following the same procedure. The alignment of the second Y-axis rail to the preceding rail was accomplished using the same procedure as was used to align the upper two rails on the X-axis. Figure 4-7 shows the arrangement of the jig and the dial indicator. The measurement jig is placed on the Y-axis carriage of the aligned rail and the dial indicator stylus was brought into contact with the flank of the unaligned rail. Starting from the top the rail was moved into position and the adjacent socket screw was secured. The procedure was repeated moving down the Y-rail following the procedure used on the X-axis rails. Once the alignment was completed, a final check was made and then the fasteners were torqued to the proper level.

**Installation and Alignment of Y-Axis Ball Screw.**

The Y-axis ball screw is of similar design to the X-axis ball screws. A difference in the assembly of this ball screw is that it is unsupported away from the driven end. Since the Y-axis ball screw is in a vertical orientation there was no concern about any sag. The installation of the ball screw was straightforward. The bearing, ball screw, and housing were assembled in the same manner as the X-axis ball screws. Once assembled and greased the assembly was lifted into place with a chain hoist.
Figure 4-7 Alignment of the second Y-axis rail
Cap screws secured the bearing housing to the mounting pads on the X-axis frame. The ball screw was aligned to the rails using jig and a dial indicator. Figure 4-8 shows the arrangement of the jig and dial indicator. The screw's parallelism to the rails was checked by measuring the distance between the rail and the ball screw by moving the jig and indicator down the rail. The lack of parallelism showed up as a yaw in the ball screw relative to the rails. To remove the yaw the bearing housing's position was adjusted. This process was repeated until the yaw was removed. A gross check for pitch relative to the face of the X-axis frame was made on the ball screw. It was not possible to make a fine measurement due to the lack of a uniform surface to make measurements against. When the ball screw had been aligned, the free rotation was checked and the ball nut was charged with grease.

**Installation of the Y-Table**

The Y-table is the structure to which the work piece is attached. The table is a robust structure with a approximate mass of 700 kg. (1500 lbs.). Provisions were made in the Y-table for lifting eye bolts to be attached to the top edge which was close to the line of action for the center of gravity. The Y-axis ball screw nut bracket was fastened to the ball nut flange and the four Y-axis carriages were positioned on the rails. The table was hoisted and moved into position. Figure 4-9 shows the Y-table suspend in position.
Figure 4-8 alignment of the Y-axis ball screw
Figure 4-9 Positioning of the Y-axis table
The alignment edges on the Y-table’s carriage mounting pad were brought up against the flank surfaces on the carriages. The bolt holes on the carriages and pads were aligned and the socket head cap screws were put in place. The cap screws were tightened. The ball screw was turned by hand to bring the nut bracket into position aligning the bolt holes. The fasteners were threaded and tightened. Wood blocks were positioned under the Y-table and the table was lowered onto the blocks. These blocks prevented the table from moving off the end of the rails and the ball screw until the servo motor and the coupling were connected. The servo motor lifted the Y-table off the blocks and the wood blocks were removed. This completed assembly of the Y-axis running gear.

**Z-Axis Pedestals**

The Z-axis motion is set on two pedestals. Each of these pedestals are large steel weldments which have been machined to accept the drive gear for the Z-axis and support the AB rotary head. The weldments each have a mass of 900 kg. (2000 lbs.). To handle these pedestals, a lifting beam was fabricated. It was designed to span the length of the pedestals and attach to lifting bolts via wire slings. The lifting bolts were placed along the longitudinal line of action of the center of gravity. The chain hoist attached to the lifting beam at mid length. Each of the pedestals was lifted onto the bed and secured in place with socket head cap screws. Figure 4-10 shows the left pedestal being lifted into place.
Figure 4-10 Placement of Z-axis pedestal on to machine bed
Note that the mating surfaces were cleaned of any dirt or debris prior to assembly. Only a gross alignment was performed at this point of the Z-axis assembly.

Z-Axis Rail Installation and Alignment

The Z-axis rails are THK HSR-45 series linear ball guideways. One rail system was attached to each of the pedestals. The left pedestal has an alignment edge for the alignment of the rail. The flank of the rail was pressed against the alignment edge and the cap screws secured in the same fashion as the Y-axis rail, on the right side, was placed and aligned. The rail on the right pedestal was put in place and the cap screws were tightened. The straightness of the rails was checked using a granite strait edge. Due to the large span between the two rails it was not possible to use a jig riding on a z-axis carriage to measure the alignment between the two rails.

Rather the left pedestal was secured the right pedestal was allowed to be moved. The Z-axis table was set on the carriages. Figure 4-11 shows the Z-axis table being move into place to be set on the carriages. The Z-axis table has alignment edges to milled into the underside of the table. These two edges were used to align the Z-axis carriages on the two rails with one another. This would result in the two Z-axis rails being parallel to each other. The Z-axis table was positioned so the left alignment edge was pressed up against the flanks of the two carriages on the left rail. The Z-axis table was then secured to left rail's carriages with socket head cap screws.
Figure 4-11 Positioning of the Z-axis table
The right pedestal position was adjusted so that the right-hand alignment edge was brought up against the flanks of the carriages on the right pedestal’s rail. The pedestal was moved into position using a hydraulic jack. Care was taken not to jam or distort the carriages or rails. Once the pedestal was in place the carriages were fastened to the table and the pedestal was fastened to the bed. The Z-axis table was tested for freedom of motion. The Z-axis running gear assembly was completed with the attachment of the linear drives. The stationary magnets were attached to the pedestals and the coils where attached to the Z-axis table.

**Testing of the Motion Errors**

To check the accuracy of the assembly of the machine tool’s three linear axes squareness and straightness error were measured. The squareness and the straightness of the three axes were measured using the laser ball bar. The measurements showed that the assembly of the machine tool had been successful. The largest squareness error was between the X-Y axis. It was measured at 1.3245 mrad. The squareness errors for X-Z and Y-Z were both less than 1 mrad. The straightness errors on all the axes were less than 0.008 mm (0.0003”). A complete set of the measurements of the squareness errors and plots of the straightness errors is presented in the Appendix.

This completed the assembly of the three translational axes of motion. The next stage of assembly would be the two rotary axes of motion. Due to the
need to test the machine in its present configuration, the machine tool was set up as a three-axis machine tool. The spindle was attached to the Z-axis table on a stationary saddle. Figure 4-12 shows the spindle be put into position. Figure 4-13 displays a photograph of the machine tool in its three axis configuration as of January 1996. Presentation of the dynamic testing and cutting test is presented in the next section.
Figure 4-12 Placement of the Phase II high speed spindle
Figure 4-13 Photograph of the three axis configuration of the machine tool
CHAPTER 5
CUTTING PERFORMANCE TESTING

This section will cover the testing and evaluation of the cutting performance of the three axis configuration of the high speed machine tool. The cutting performance testing will follow procedures similar to those presented in Section 7 and appendix A of the ASME B5.54 [ASM92]. The spindle idle run losses test, the chatter limit tests, and full torque testing were performed. The testing of cutting force induced errors was not performed. The data found in these tests can be used to compare this machine tool with others in its class.

The fundamental concept of this high speed machine tool, from its initial inception, was to have high metal removal rates for the cutting of aluminum aircraft components. This requires that a majority of the available spindle power be utilized in the cutting the material. The power consumed in the cutting process can be determined from the following equation.

\[ P = K_s abc mn \]  

(5-1)

where \( K_s \) is the specific power of the material, \( a \) is the radial depth of cut, \( b \) is the axial depth of cut, \( c \) is the feed per tooth, \( m \) is the number of teeth on the cutter, and \( n \) is the spindle speed. \( K_s \) is a property of the material being cut. The value
of $K_s$ for aluminum is about 750 N/mm². The values of $c$, $m$, and $n$ are limited by cutter material, cutter geometry, and the maximum speed of the spindle. The axial and radial immersions are limited by the tool path and the dynamic stability of the tool/spindle/work piece system.

The MRR in HSM of aluminum is typically limited by the available torque of the spindle for cutting the material or by the onset of chatter. It would be ideal to reach the spindle’s torque limit at top spindle before the onset of chatter. This would mean that the cutting process is stable and drawing upon all the available power of the spindle, thus maximizing the metal removal rate (MRR). If on the other hand the limit of stability is reached first, the total available power cannot be used. This has the effect of reducing the maximum possible MRR.

The occurrence of chatter is due to insufficient dynamic stiffness of the tool/holder/spindle system and the supporting structure. Thus, the cutting performance is directly coupled to the dynamic stiffness of the spindle and the machine tool’s structure. To see how chatter is related to the lack of dynamic stiffness, a short discussion on chatter is presented in the following section. It will cover the fundamentals of chatter and the lobing diagram.

**Chatter**

Chatter arises from the regeneration of the waviness on the surface being cut. Each cutting edge removes material from the work piece producing a surface. Any vibration, at the time that the surface is being cut, generates a
wavy surface. This wavy surface leads to variable chip thickness, which produces unsteady cutting forces. The magnitude of the cutting force is proportional to the thickness of the chip. Depending on the conditions of the cutting process, the vibration of the tool either grows or diminishes. If the vibration diminishes, the cutting process is stable. If the vibration grows, the cutting process is unstable and chatter occurs. Chatter produces a rough surface on the work piece. Chatter is detrimental not only to the surface of the work piece, but also to the cutting tool and the spindle. The high dynamic forces due to chatter can break the cutter and damage the spindle.

The onset of chatter, as stated previously, limits the amount of power that can be utilized in the cutting process. The maximum axial and/or radial immersions become limited with chatter. This reduces the MRR causing and increase in the machining time for a given part. A complete discussion on Chatter is presented in the original work by Tlusty [TLU85]. The worst case for the onset of chatter is the cutting of a slot, 100% radial immersion. This case leaves only the axial depth of cut as the dominant variable for the onset of chatter. The chip load can also be varied but only affects the magnitude of the chatter. It does not affect the onset of chatter. The equation for determining the limit of the axial depth of cut is

$$b_{\text{lim}} = \frac{-1}{2K_{s}(\mu_{x}Re[G_{x}] + \mu_{y}Re[G_{y}])m_{\text{avg}}}$$

(5-2)

where $b_{\text{lim}}$ is axial depth of cut, $K_{s}$ is specific power of the material being cut, $\mu_{i}$ is
the directional orientation factor, \( m_{avg} \) is the average number of teeth in the cut. The orientation factor is a function of the radial immersion. It relates how the cutting force causes deflection in the direction of the measured FRF and how that deflection affects chip thickness. \( \text{Re}[G_x] \) is the negative real part of the FRF for each of the orthogonal directions in the plane of the cut. Figure 5-1 shows the relationship between the FRF of a cutter and axial limit of stability. There are two additional equations used to obtain the stability lobe diagram. These equations relate the calculated axial depth of the cut, \( b_{lim} \), to the spindle speed. Equation 5-3 relates the ratio of the frequency of chatter, \( f_c \), and the tooth passing frequency, \( f_t = \frac{n}{m} \), to the number of waves and phase shift between the previous surface and the new surface being cut. The equation is

\[
\frac{f}{nm} = N + \frac{e}{2\pi}
\]  

(5-3)

where \( f \) is the frequency of chatter, \( n \) is the spindle speed, \( m \) is the number of teeth on the cutter, \( e \) is the phase shift between the vibration of the cutter tooth and the wavy surface left by the previous tooth, \( N \) is an integer such that \( e/2\pi < 1 \). Equation 5-4 determines the phase shift between the vibration of the cutter tooth and the surface left by the previous tooth as

\[
e = 2\pi - 2\tan^{-1}\left[\frac{\mu_x \text{Re}[G_x] + \mu_y \text{Re}[G_y]}{\mu_x \text{Im}[G_x] + \mu_y \text{Im}[G_y]}\right]
\]  

(5-4)
Figure 5-1 Relationship between frequency response function and stability lobes
The preceding equations define the relationship between the spindle speed and the permissible depth of cut which is commonly referred to as the “stability lobe diagram.” Figure 5-2 shows the stability lobe diagram for a simple single DOF system.

To calculate the stability lobe diagrams for a given cutting tool/machine tool configuration, the FRF of the cutting tool relative to the work piece must be measured. The modal parameters can them be extracted and used to produce the stability lobe diagrams. These diagrams can be used to determine the maximum stable depth of cut for a given spindle speed. Stability lobe diagrams for varying radial immersions can also be produced. It is noted again that the feed per tooth is not a contributing factor to the onset of chatter. It will only affect the magnitude of the forces.

**Measurement of the Frequency Response Functions**

The modal parameters of the cutting tools were extracted from measurements made using the impulse excitation method. Direct frequency response functions (DFRF) were measured between the tool and the workpiece in the X and Y directions perpendicular to the axis of rotation of the spindle. The FRF’s, \( G_x \) and \( G_y \), are the ratios \( X(\omega)/F_1(\omega) \) and \( Y(\omega)/F_2(\omega) \). \( F_1(\omega) \) and \( F_2(\omega) \) are the variable forces acting in the two axes, respectively. The DFRFs are obtained using an instrumented hammer, accelerometer and a dynamic analyzer.
Figure 5-2 Lobing diagram for simple single degree of freedom system
The hammer used was a PCB model 086C80 micro impulse hammer, and the accelerometer is a PCB model U352A10 micro accelerometer. The dynamic analyzer and software used in the measurements were developed by Manufacturing Laboratories Inc. The measurement of the DFRF was made at the free end of the cutting tool. The micro accelerometer was attached to the tip of the end mills using a petroleum-based wax. The end mill/spindle was rotated to bring the accelerometer into alignment with the direction that is being measured. Figure 5-3 shows a diagram of the accelerometer positioned at the tip of the end mill, and the location where the impulse was applied. The impulse is applied to the tool tip on the opposite side from the accelerometer. The analyzer measures and records the accelerometer's response and the magnitude of the impulse and performs the fourier transform by dividing the former by the latter. Five measurements were taken and averaged in the X and Y directions. Figure 5-4 shows the frequency response function of a 19 mm (0.75") diameter carbide end mill with 44 mm (1.73") overhang from the end of the tool holder installed in the spindle. The tool was held by a shrink fit 40 taper type holder. Examination of the FRF's shows that there is a significant difference between the two directions. In the X-direction it can be seen that there are four significant modes, while in the Y-direction there are six significant modes. Table 5-1 presents the modal data extracted from the 2 FRFs. It can be seen that there are modes at 747 Hz, 944 Hz, 1075 Hz and 1693 Hz in both the X and Y direction. The Y-direction has two additional modes at 875 Hz and 1149 Hz.
Figure 5-3 Diagram of impulse excitation measurements
FREQUENCY RESPONSE FUNCTIONS
4 FLUTE CARBIDE END MILL
19 mm DIA.  44 mm OVERHANG

**X-DIRECTION**

**REAL**

**IMAG**

**Y-DIRECTION**

**REAL**

**IMAG**

Figure 5-4 Frequency response functions of carbide end mill
# MEASURED MODAL DATA

## 4 FLUTED CARBIDE END MILL

### 44 mm OVERHANG

## MODAL DATA X-DIRECTION

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Table 5-1 Modal data of carbide end mill
It can also be observed that there is a difference in the magnitudes of the modes. Examining the most compliant modes in the two directions also shows an interesting difference. In the X-direction the most flexible mode is at 1075 Hz with a modal stiffness is 1.586e+7 N/m. The corresponding mode in the Y-direction is two times as stiff. The most flexible mode in the Y-direction is at 944 Hz. It has a modal stiffness of 4.452e+7 N/m. The corresponding modal stiffness in the X-direction differs by less then 1%. A calculation of the critical depth of cut, b_{lim,crit}, can be made for the most flexible modes in both directions. The smaller of the two values will be a stable depth of cut for any spindle speeds (safe side estimate). It will be calculated for the cutting of a slot in each of the directions, maximizing the radial immersion. The equation for this calculation is

\[
b_{\text{lim,crit}} = \frac{1}{2 K_s \mu \frac{H m}{2}}
\]

(5-5)

where \( b_{\text{lim,crit}} \) is the maximum axial depth of cut, \( K_s \) is the specific power of the material being cut, \( \mu \) is the orientation factor, \( H \) is the magnitude of the mode, \( m \) is the number cutting edges on the end mill. Taking the most flexible modes from Table 5-1 for the two directions, the \( b_{\text{lim,crit}} \) is calculated as 1.114 mm (0.044") for the X-direction and 2.329 mm (0.092") for the Y-direction. This calculation shows a two fold greater depth of cut is possible in the Y-direction then in the X-direction. It is noted that this calculation only gives an estimate of the difference in the depth of cut. It assumes that the modal stiffnesses in the two directions act independently of each other which is not the case.
This lack of symmetry between the 2 DFRFs was found with all the end mills tested. It is believed that this variation in the FRF's for these two directions is due to the structural arrangement of the Z-axis. The structure supporting the spindle consists of the spindle housing, saddle, Z-axis table, and the two pedestals. The pedestals were placed wide apart to make room for the AB head's drive components. To examine the difference between the two directions' DFRFs, a systematic analysis was performed. The analysis consists of dynamic modeling of the spindle and taking modal measurements of the spindle and the structure.

**Spindle Model Analysis**

The spindle model analysis was performed using a spindle analysis program called SPA. This program was developed by Manufacturing Laboratories Inc. The program utilizes 4 DOF beam elements to build a spindle model. The four DOF for these elements are divided into one translational DOF and one rotational DOF at each end of the beam element. The model is assumed to be symmetric about the axis of spindle rotation. The stiffnesses will be considered to be equal in all planes passing through this axis. Spring and mass elements are added to the model to account for the bearings, tool holder, motor elements, and other components. Listings of the two spindle models used are presented in the appendices. The two spindle models incorporate a shrink fit
tool holder and carbide end mills with overhangs of 54 mm and 123 mm. Plots of the mode shapes with their corresponding natural frequencies, and modal stiffnesses at the tool tip are presented in figure 5-5.

Examination of the model modes shapes and stiffnesses shows that the tool length has a strong effect on the mode shapes and the stiffness at the tool tip. This is consistent with what has been seen by others [TLU96, WIN95]. The short tool model shows that the most flexible modes are the first, second, and third modes of the spindle. The stiffness at the tool tip is approximately $6 \times 10^7$ N/m at 826 Hz, 1042 Hz, and 1373 Hz. The tool mode is stiffer, having an approximate stiffness of $8.6 \times 10^7$ N/m at a frequency of 3568 Hz. Note that all four of these modes have a similar magnitude of stiffness. The second model with the long tool shows the tool mode to be the most flexible. It has a modal stiffness of $3.4 \times 10^6$ N/m at a frequency of 648 Hz. The spindle modes are all at least an order of magnitude greater in stiffness. The comparison of the mode shapes and stiffness from the model to experimental data will be conducted in the next section.

**Modal Measurements of the Spindle**

A modal analysis of the spindle was performed examining the direct and cross FRF's of the spindle in two perpendicular directions. These measurements were made on two tool/spindle combinations. Two micro-grain carbide end mills were used. Both end mills are 19 mm (0.75") in diameter.
SPINDLE MODELS OF HIGH SPEED SPINDLE FITTED WITH CARBIDE END MILLS

Figure 5-5 Spindle model analysis of mode shapes
One was a four fluted end mill with an overall length of 101.6 mm (4 inches).
The other was a two fluted end mill with an overall length 152.4 mm (6 inches).
Both tools were held by shrink fit tool holders manufactured by Tooling Innovations. The tool holder has a standard 40 taper tool holder, for the spindle/tool interface.

The measurements made are a combination of one DFRF taken at the tool tip and a number of cross transfer functions (CFRF’s) measured along the tool/holder/spindle system. The frequency range measured was 5000 Hz.

Figures 5-6 and 5-7 show the placement of the accelerometer for the measurements. The impulse excitation was always applied at location #1, the tool tip. The measurements were made in two perpendicular directions. These were made along the lines formed by the XZ and the YZ planes as it passes through the rotational axis of the spindle. The line formed by the XZ plane is parallel to X-axis of motion of the machine and is considered the X-direction. Similarly the line formed by the YZ plane is parallel to the Y-axis of motion and is considered the Y-direction. It is noted that any orientation of the two perpendicular directions could be used as long as they remain perpendicular to each other and the Z axis. Figure 5-6 and 5-7 shows the resultant mode shapes, natural frequencies, and stiffnesses at the tool tip, which has been extracted from the measured DFRF’s. It can be readily seen from the mode shapes that there is a difference in the two directions. There are more significant modes in the Y-direction for both cases. This is consistent with the initial tool measurements.
MODE SHAPES
EXTRACTED FROM MODAL MEASUREMENTS OF THE
HIGH SPEED SPINDLE FITTED WITH A 4 FLUTED
CARBIDE END MILL
19 mm DIA  54 mm OVERHANG

MEASURED MODE SHAPES
OF 19 mm CARBIDE END MILL
54 mm OVERHANG
X-DIRECTION

797 Hz
2.928e+007 N/m

984 Hz
2.905e+007 N/m

1244 Hz
3.358e+007 N/m

2271 Hz
7.814e+007 N/m

2448 Hz
5.893e+007 N/m

3291 Hz
2.512e+007 N/m

MEASURED MODE SHAPES
OF 19 mm CARBIDE END MILL
54 mm OVERHANG
Y-DIRECTION

443 Hz
7.448e+008 N/m

754 Hz
8.426e+007 N/m

902 Hz
2.097e+008 N/m

1001 Hz
6.582e+007 N/m

1122 Hz
1.334e+008 N/m

1196 Hz
8.714e+007 N/m

1331 Hz
5.053e+007 N/m

2264 Hz
9.877e+007 N/m

2440 Hz
7.864e+007 N/m

3293 Hz
2.996e+007 N/m

Figure 5-6 Measured mode shapes of 54 mm length end mill
MODE SHAPES
EXTRACTED FROM MODAL MEASUREMENTS OF THE HIGH SPEED SPINDLE FITTED WITH A 2 FLUTED CARBIDE END MILL
19 mm DIA 123 mm OVERHANG

Figure 5-7 Measured mode shapes of 123 mm length end mill
Closer examination of the mode shapes shows that there are corresponding modes in the two directions for both tools. There are some differences in the frequencies. This maybe due to the failure of capturing the peak values of the FRF, because of the discrete data acquisition and non-linearities of the system. Comparison of the mode shapes and stiffnesses measured on the 54 mm tool and those of the 54 mm model shows reasonable agreement. The frequency variation is less the 10% at its greatest variance. The stiffnesses are of the same magnitude. It can be seen that there are three individual modes which show up in the Y-direction that are not found in the model or measured in the X-direction. These modes occur at 443 Hz, 1001 Hz, and 1122 Hz. There is also a mode at 2270 Hz that shows up in both X and Y directions, but is not found in the spindle model. Similarly, examination of the measured values of the 123 mm tool show good correspondence in the two directions and with the model. However, there are modes in the Y-direction which do not correspond to the X-direction or to the model. These modes are at 439 Hz, 858 Hz, 1147 Hz and 1227 Hz. There is a mode at 3111 Hz in the X-direction which does not match up with model or the Y-direction measurements. It is noted that the models assumed a value of 3% for the damping ratio. Measured values of the damping ratio varied from less than 1% to more than 7%. The stiffnesses found at the tool tips for both cases shows that the X-direction is always more compliant.

The difference in the number of significant modes between the two directions is probably a result of the influence of the structure supporting the spindle. The supporting structure consists of the spindle housing, spindle
bracket, saddle, Z-axis table, and the two support pedestals. The spindle housing and bracket support the rotating elements of the spindle. The bracket sits on a saddle placed on the Z-axis table. The table rides on two linear rail systems which set upon the two steel weldments that make up the pedestals. To see how the structure combines with the spindle to produce the observed modes and stiffnesses, modal measurements were collected on the structure.

**Examination of Modal Measurements on Spindle Housing**

Modal measurements were made along the spindle housing in the X and Y directions. The impulse method was used to make the measurements. Since the measurements were to be made on the structure of the machine tool, lower frequencies were expected. A larger impulse hammer and accelerometer were used to take this into account. The hammer imparted more energy to excite the structure and the larger accelerometer has a higher sensitivity for the lower frequencies. The hammer was a PCB model 086B03 instrumented hammer and the accelerometer was a PCB model 308B09. The frequency range measured was 2500 Hz. Figure 5-8 shows the locations of accelerometer placement to make the measurements for the X and Y directions. The excitation impulse was applied at the spindle nose, location number #1. The resulting mode shapes from the DFRF and CFRF's are presented in figure 5-8. The stiffnesses shown, are those seen at the nose of the spindle. The X-direction shows predominately rigid body motions until about 2000 Hz.
MODE SHAPEs
EXTRACTED FROM MODAL MEASUREMENTS OF THE
HIGH SPEED SPINDLE HOUSING

Figure 5-8 Measured mode shapes of the high speed spindle housing
These lower modes are rocking and twisting of the whole housing saddle and bracket. The higher modes show the spindle housing bending modes. The nose of the housing can be seen to deflect in the last five mode shapes. There are corresponding modes in the Y-direction. The Y-directions also has mostly rigid body motions until above 2000 Hz. Note that the 437 Hz mode seen in both tools tested is seen here as well. It can be seen that it is a rigid body mode with respect to the housing. This could be the spindle housing bracket rocking or the Z-axis table deflecting. There are also modes at 1000 Hz and 1100 Hz which were previously seen in the spindle modal measurements. The twisting and rocking seen in these modes must be due to the structure supporting the housing. Examination of the supporting structure is now presented to see how the rigid body modes seen here fit together with the structure.

Modal Analysis of the Z-Axis Table

Three sets of modal measurements were made on the Z-axis table. One set of measurements was made along center line of the table in line with the YZ plane passing through the spindle axis. The second set was made along a line parallel to the X-axis on the top of the table just behind the saddle. The last set of measurements was made along the flank of the table in the X-direction. Figures 5-9, 5-10, and 5-11 show the locations of the measurement points and the point of excitation. The resultant measured mode shapes, natural frequencies, and stiffnesses are also presented in these figures. It can be seen from the mode shapes presented in the three figures that the Z-axis table and the spindle housing bracket are the compliant components of the structure.
Figure 5-9 Measured mode shapes of the Z-axis table along the axis of the spindle
Figure 5-10 Measured mode shapes of the Z-axis table along X in Y direction
MODE SHAPES
EXTRACTED FROM MODAL MEASUREMENTS OF THE
Z-AXIS TABLE ALONG Z IN X DIRECTION

Figure 5-11 Measured mode shapes of the Z-axis table along Z in X direction
Corresponding mode shapes and natural frequencies can be seen in the four figures.

**Discussion of the Results**

The figures show a number of mode shapes which can be discussed. To reduce the number modes to be examined the frequency range of concern is addressed. In the cutting process the tooth passing frequency, \( f_t \), is the frequency of the excitation force. The ranges of frequencies of concern here are from 333 Hz to 2400 Hz. The lower limit is set by a two fluted cutter rotating at 10,000 RPM (166.67 Hz), which gives a tooth passing frequency of \( f_t = 333.34 \) Hz. The upper limit is set by a four fluted rotating at 36,000 RPM (600 Hz), which gives a tooth passing frequency of \( f_t = 2400 \) Hz. The number of cutting edges used for these limits is typical to what will be seen in the operation of this machine tool. The spindle speed range is the lower and upper limits for cutting operations with this spindle. It is noted that some of the lower frequency modes may be excited by the servo motors.

Examination of the Y-direction in the above frequency range show that the most flexible mode lies at 1000 Hz. It is a rigid body rocking mode as seen in figure 5-8. Figure 5-9 show the table and the spindle housing tilting in opposite directions to each other, as well as a bowing mode in the Z-direction of the table. Figure 5-10 shows a mode shape of the third plate mode in the table in the X direction of the table. This mode also showed up in the tool measurements. A
second mode at 1130 Hz, has a similar, but inverted mode shape. The mode shapes in the X direction are shown in figure 5-11. The mode shapes are all rigid body modes until the last one. The resultants from these measurements show where a number of the variations seen in the X and Y tool transfer functions are related to the structure. It is noted that the stiffnesses are an order of magnitude greater then seen at the tool.

Cutting Simulations

Prior to the performance of cutting test cutting simulations of the cutting tools were performed. These simulations were used to determine the starting point initial depths of cut and spindle speeds. The cutting simulations were performed using the Milsim 2.0 cutting simulation program developed by Manufacture Laboratories Inc. The simulation program utilizes a regenerative force, dynamic deflection model as described by Smith and Tlusty [SMI91, SMI93]. The program performs a number of simulation runs stepping thru a range of spindle speed and axial depths of cut. The results of the many simulations are summarized in the form of a PTP (Peak-to-Peak) graph [SMI93]. The PTP graph plots lines of constant axial depth of cut on axes of force or displacement verse spindle speed. This format readily shows the regions where stable cutting can be performed and what is the maximum depth of cut in that region. The dynamic parameters used in the simulations are extracted from the measured transfer function of the given cutting tool.
Figures 5-12 and 5-13 show the PTP plots of the 54 mm tool and 123 mm tools. The spindle speed was stepping by 130 RPM thru the range of 10,000 RPM to 36,000 RPM in the simulation. The range of axial depths of cut for the 54 mm tool was 0.2 mm to 1.1 mm in steps of 0.1 mm. The axial range for the 123 mm tool was 0.2 mm to 1.3 mm in steps of 0.1 mm.

Examination of the PTP plots for the two tools shows the best speeds and depths of cuts for each of the two tools. The plot for the 54 mm tool shows two regions where stable cuts can be performed. These are at approximately 17,000 RPM and 24,000 RPM. The maximum depth of cut was determined to be approximately 0.8 mm for both gaps. It can be seen that the two gaps close-up as the depth is increased and the cut will become unstable. Similarly the 123 mm tool has two gaps over the operational speed range. The lower speed gap is down around 10,000 rpm and the upper gap is around 20,000 rpm. The maximum depth of cut is 1 mm. It can be seen that the longer tool has a greater axial depth limit then the shorter tool. This seems counter intuitive. It would often be thought that the shorter tool would have the greater axial depth of cut. The reason this occurs is that the tool/spindle system is continuous system. The tool spindle combination in this case produces a system with poor dynamic stiffness. Changing of the tool length can produce a more desirable result. Tlusty et al. [TLU96] shows that by changing the length of the tool can have a positive effect on the dynamics of the tool/spindle system. Winfough had similar findings in his research [WIN95]. This idea has been given the name tool tuning.
Figure 5-12 PTP plot of 54 mm carbide end mill cutting a slot
PTP GRAPH OF CARBIDE END MILL
2 FLUTED 19 mm DIA 123 mm OVERHANG

Figure 5-13 PTP plot of 123 mm carbide end mill cutting a slot
The idea is that by adjusting the tool’s length it would be possible adjust the modal parameters and thus bring the largest gap between the lobes to the top spindle speed. This would give the greatest MRR. Some times the tool’s length is increased to bring the lobe into place.

**Cutting Tests**

Initial cutting tests were performed on the machine tool. The cutting tests consisted of cutting of slots in the X & Y direction, facing of the test piece, and production of thin walled, and thin floored test pieces. A wide selection of 19 mm diameter carbide end mills was used in the tests. The end mills had two, three, and four flutes. They varied in length of overhang from 41 mm (1.61") to 127 mm (5"). The end mills were held by shrink fit and hydraulic type 40 taper tool holders. Dynamic measurements were made for each tool. Milling simulations were made for some of the tools to determine the best operating speeds. The chatter recognition and control (CRAC) system was also employed to rapidly determine the best spindle speeds. The feed per tooth was kept constant at 0.127 mm per tooth (0.005").

The slotting tests were used to determine the greatest stable axial depth of cut at the best spindle speed. These tests were compared to the simulations and where found to be in good agreement. The CRAC system worked well in determining the best spindle speeds. There were cases where the CRAC system found competing modes.
In these cases the CRAC system would jump back and forth between two spindle speeds. This generally occurred when the limit of stability was reached at the best speed. It was also observed that the best speed did not follow the general rule where \( f_i \) is set to \( f_n \) of the most compliant mode. This arises when the gaps between lobes for a given mode are filled by the lobes of additional modes of the system. Smith et al. point this out in their discussion about competing modes. The net effect of this occurrence is a diminishing of the maximum depth of cut that can be achieved.

A full-power cutting test was used to check that the spindle delivered its full torque. This test consisted of facing a test piece at a 90% radial immersion and maximum axial depth of cut for the given end mill. The maximum MRR which could be achieved was 133 cubic inches per minute. Higher metal removal rates caused the spindle speed to drop off significantly. This was the onset of the stall condition. The power consumed in this cut was 27.4 kW. The available power for the cut was 28.8 kW. The spindle idle run loss test showed that there is a 20% loss in the operating range. This means that at full 36,000 rpm there is only 28.8 kW available for cutting.

The thin-wall and thin-floor test parts were used to demonstrate the machine tool's capability. The first part was a thin walled pocket cut from a 177.8 mm (7") long by 101.6 mm (4") wide that is 101.6 mm (4") thick. The pocket is 101.6 mm (4") by 76.2 mm (3") cut to a depth of 76.2 mm (3") with a wall thickness of 0.5 mm (0.020"). The part was cut using two and four fluted relived cutters. The time required for fabrication of the part was one hour for the
two fluted end mill and one half hour for the four fluted end mill. The parts cut with the four fluted end mill had poor surface finishes. A problem with built up edge seemed to be the cause. Multiple attempts were made to get around this problem, but no solution has been found as of this writing. The end mills used to cut these parts could take a maximum of 1.5 mm (0.060") depth of cut. Feed rates were 7.62 m/min (300 ipm) for the two fluted cutter and 15.24 m/min (600 ipm) for the four fluted end mill. The thin floored test part is a 101.6 mm (4") by 76.2 mm (3") pocket. The pocket is cut into both sides of the test part to producing a thin floor of 0.5 mm (0.020") thickness. A four fluted end mill is used to produce the part. The feed rate was 12.7 m/min (500 ipm).

The machine tool performed well in the test cuts and producing the test pieces. Some minor problems were encountered during the testing. These were generally minor control problems. A number of each of the test parts was produced during testing of the machine tool.

**Double Hexagon Test Part**

To test the machine more completely and produce a test part which shows many of the advantages of HSM a double hexagon thin walled thin floored part was milled. The double hexagon test part is shown in figure 5-14. This is a monolithic part cut from a 10" by 7" by 4" block of 7075 T6 aluminum. The test parts have thin walls that are 3.75" in height. It has a base which is a thin floor. The wall and floor thickness is 0.020". The test part requires tooling which can
reach into the pockets 3.75". End mills of 19 mm (3/4") diameter with overhang lengths of 5" and 6" where used. Two and four fluted end mills where also used. Prior to cutting, dynamic measurements of the cutting tools were made. Cutting simulations of the cutting tools were performed to find the best spindle speeds and the maximum stable cutting depth.

Examination of the frequency response functions for a pair of the end mills shows that the assumption of using the most compliant mode to find the best speed does not hold. An example is the 100 mm two-fluted end mill presented in figure 5-15. Examining the X-direction FRF shows that the most compliant mode is at approximately 1136 Hz. This would correspond to a spindle speed of 34080 rpm. This would be great from the point of view of obtaining highest MRR. But examination of the milling simulation shows that this spindle speed is in the middle of a highly unstable region as seen in figure 5-16. The simulations' points to a spindle speed of 27750 rpm. Closer examination of the FRF of the end mill shows that there are two additional modes at 778 Hz and 1731 Hz, which have magnitudes on the same order as the most compliant mode. In such cases the idea of using only the most compliant mode to determine the best spindle speed fails. Such cases are known to exist and are known as competing modes [SMI94]. In such cases by moving to the best speed of one mode puts the cutting conditions in the unstable region of the other mode. Similar results were found for other end mills. Cutting tests were performed to see if this was in agreement to the physical system. The results of the cutting tests corresponded well with the cutting simulations.
Figure 5-14 Double Hexagon Test Part
Transfer Functions
2 Fluted Carbide End Mill
19 mm Dia. 100 mm Extension

Figure 5-15 Frequency Response Function
The initial cutting simulations did not give the full picture due to its lack of resolution. The initial simulations were rather coarse in hopes of minimizing the computation time. This is seen in the PTP of the cutting tool, figure 5-16. It only showed that best speed and the regions of instability. The simulations were refined to take a smaller step in the depth of the cut and finer spindle speed steps. PTP plots for the cutting of slots in the X and Y direction and 45° angle between the two directions. This was performed due to the fact that during some of the test cuts the maximum axial depth of cut differed in some directions. Figures 5-17, 5-18 and 5-19 shows the PTP plots for the three directions. The forces for the X and Y directions are presented for each case. Examination of the PTPs shows the two lobes from different modes. The best speed for this tool is 27,500 rpm.

The cutting of the test part was performed using two and four fluted end mills. The axial depth of cut was 1.5 mm (0.060"). The two fluted cutter performed well. The four fluted cutter performed well on the outer walls of the double-hexagon but had problems with built-up edge with cutting the pockets. The built-up edge problem was attributed to possible failure of the mist lubricant/coolant to reach into the full depth of the pocket. It is felt that once flood cooling is implemented that the built-up edge problem can be controlled. Figure 5-14 Displays a photo of a finished tests part. It was cut using a two fluted end mill at a feed rate of 400 ipm.
PTP of 2 Fluted Carbide End Mill 100 mm Extension

Figure 5-16 PTP Plot of 100 mm End Mill
Cutting simulation of 2-flute carbide 100 mm End Mill Slot Y-direction

Figure 5-17 PTP for slot in Y-direction
Cutting simulation of 2-fluted 100 mm End Mill
Slot X-direction

Figure 5-18 PTP for slot in X-direction
Cutting simulation of 2-fluted carbide 100 mm End Mill
Slot cut at 45 angle

Figure 5-19 PTP for slot at 45° angle
Conclusions and Continuing Work

The three-axis configuration of the high speed machine tool has performed up to expectations. It has been able to cut aluminum parts utilizing the high feed rates. It has performed cutting operations at cutting feed rates of 22.9 m/min (900 ipm). The machine tool has also been able to fully utilize the available cutting speed and power of the spindle. Power cuts at 32,000 rpm and an axial depth of cut of 7 mm (0.276") performed, achieving metal removal rates on the order of 0.00164 m³/min (100 inches³/min). The squareness of the X and Y axes to each other, and alignment of the spindle to the Z-table can be improved. Completion of the enclosure would be beneficial. Work is continuing on the design and construction of the AB head. It should be added to the machine tool in the next year.
APPENDIX
ERROR MEASUREMENTS OF THREE-AXIS MACHINE TOOL

The following material presents error measurements of the translation motions of the three-axis configuration of the machine tool. The measurements were taken using a Laser Ball Bar.

Squareness Errors:

<table>
<thead>
<tr>
<th>Direction</th>
<th>Average</th>
<th>Error (mrad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>X-Y</td>
<td>90.07592°</td>
<td>1.324967</td>
</tr>
<tr>
<td>X-Z</td>
<td>89.98621°</td>
<td>-0.24077</td>
</tr>
<tr>
<td>Y-Z</td>
<td>89.97874°</td>
<td>-0.037106</td>
</tr>
</tbody>
</table>

Straightness Errors:

X-axis:
Y-axis:

![Graph for ystx error](image1.png)

![Graph for ystz error](image2.png)

Z-axis:

![Graph for zstx error](image3.png)

![Graph for zsty error](image4.png)
References


BIOGRAPHICAL SKETCH

The author was born on the tenth of May 1961 in Milwaukee, Wisconsin, to Dr. and Mrs. Victor M. Bernhard. He spent the next 19 years growing up in Milwaukee always dreaming of becoming an engineer. His first steps to achieving this goal was his entrance into the United States Merchant Marine Academy. Here, the author studies mechanical engineering with its application to the maritime industry. He also had the chance to travel and see much of the world by ship. He finished his training at the academy in 1984. He left the Kings Point, New York, and headed to New Orleans, Louisania, to work for Avondale Shipyards Inc. Here he gained experience in the construction of large ships. In the spring of 1985 he left the shipyard to take a shipboard engineering position with Exxon Shipping Company. The author gained much experience in the operation and maintenance of steam and diesel systems during his five years sailing for Exxon. In 1989 the author began his graduate studies at the University of Florida with the hopes of achieving a doctorate. For the next seven years the author toiled feverishly to achieve this goal. The author is presently working at Boeing Aircraft Company in the Manufacturing Research & Development Division. The author wishes to thank all his friends and the Divine spirit for all the blessings that have come the author’s way.
I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.

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I certify that I have read this study and that in my opinion it conforms to acceptable standards of scholarly presentation and is fully adequate, in scope and quality, as a dissertation for the degree of Doctor of Philosophy.

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