INVESTIGATION OF A SPECIAL 6-6 PARALLEL PLATFORM FOR CONTOUR MILLING

Waheed A. Abbasi, Shannon C. Ridgeway, Phillip D. Adsit, Carl D. Crane, and Joseph Duffy

Center for Intelligent Machines & Robotics
300 MEB, University of Florida
Gainesville, FL 32611
waa@cimar.me.ufl.edu

ABSTRACT

Recently, companies have been experimenting with parallel-mechanism based approaches for milling machines. This research presents an investigation into the development of a special 6-6 parallel mechanism for application to contour milling. The idea behind this approach is that existing non-CNC milling equipment can be augmented to increase its capability at a lower cost than purchasing traditional 5-axis machining centers. This paper presents the phase of research associated with developing a parametric kinestatic design methodology for a special 6-6 parallel mechanism (Kinestatic Platform, KP). This methodology was applied to the design specifications associated with 5-axis contour milling. The resulting kinestatic design's dynamics were evaluated to determine the actuation requirements of each connector. A prototype connector was built to allow the evaluation of actuator response under simulated loading conditions. Joint stiffness and control strategy were of primary concern in evaluating the performance of the prototype connector. The parametric kinestatic design and control strategy results are presented. Several observations are evidenced from the research. Joint deflection is an obvious critical issue and the most difficult to quantify. A scheme is proposed detailing the concept of using a separate metrology frame to overcome difficulties associated with accurate connector length determination.

1. Introduction

A Kinestatic Platform (KP) is being developed to augment a non-CNC three axis mill to perform five axis contour milling. Two implementations of the concept are shown in Figure 1. On the left side of the figure, the milling head of the three axis mill is rotated to the side

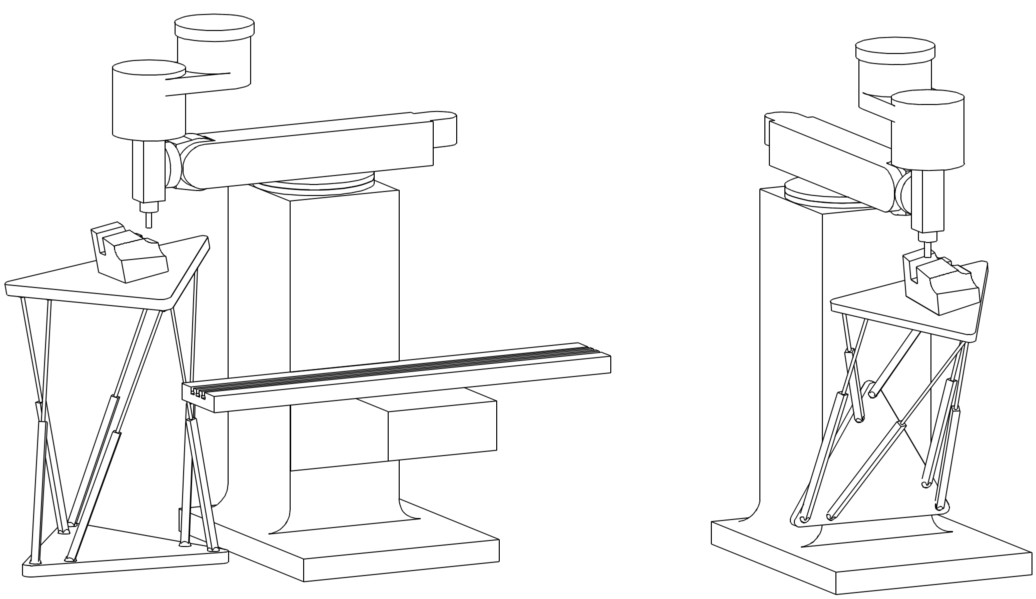

Figure 1: Two implementations of the Kinestatic Platform as applied to contour milling

and the KP is placed beneath the rotating spindle. The work piece to be milled is mounted to the top platform of the KP. The KP moves the workpiece into contact with the stationary (but spinning) tool to perform the milling operation.

On the right side of Figure 1, the translating carriage of the mill has been removed and the KP mounted in its place. In typical mills, the translating carriage is connected to the mill "iron", or base using a dovetailed connection. The base plate of the KP can be mounted to the base of the mill using the same dovetailed interface.

An advantage of the first implementation is that the KP can be moved to another machine or another facility and the mill can quickly be put back into standard service. The second implementation offers the advantage of not using additional floor space. Additionally, the mill and KP are rigidly connected so that vibrations of the floor will not affect the accuracy of the milling operation.

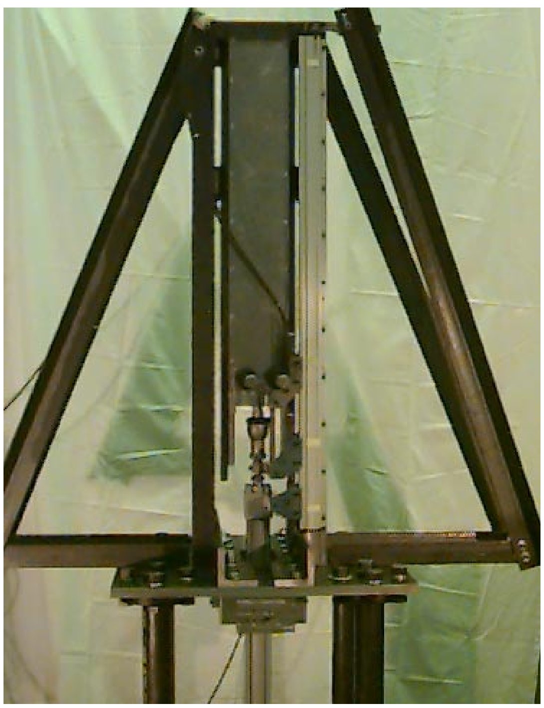

Figure 2: KP Test bed

A design of the KP was created based upon specifications derived from the milling operation. One actuator was fabricated (see Figure 2) and control algorithms were implemented and tested to evaluate whether the necessary accuracy could be obtained under varying loading conditions. Based on the results obtained, it seems feasible to fabricate a prototype KP and to test the system in a manufacturing environment.

The rationale for this endeavor is that existing milling equipment can be augmented to greatly increase its capability at a lower cost than purchasing traditional five axis machining centers. The approach offers the benefits of high stiffness, strength, and accuracy due to the parallel nature of the KP mechanism. An accuracy of 0.001 inches is to be achieved throughout the cutting operation.

Recently, companies such as Ingersoll, Inc. and Giddings and Lewis as well as international milling companies have been experimenting with new parallel-mechanism based approaches [1,2]. The distinction between their approach and ours is the utilization of the KP which incorporates a new and novel patented simplified geometry. Additionally, the KP has the capability of simultaneous precision control of contact force and displacement at the milling cutter.

2. Background: Kinestatic Platform Development

A platform or parallel mechanism is defined as any mechanical device that has six legs that connect a moving platform to a base. This kind of mechanism possesses the desirable characteristics of high accuracy, high payload-to-weight ratio, and good static stability. To apply Kinestatic Control to these mechanisms it is necessary to first obtain accurate compliance models. These models can be readily determined for parallel mechanisms, provided that the position and orientation of the moving platform is known relative to the base. Therefore, the key and central task is to determine the position and orientation of the moving platform relative to the base given the sensed lengths of the six legs. This task is referred to as the forward kinematic analysis for the system and, for these kinds of mechanisms, the simplest solution involves solving an eighth degree polynomial in a single variable.

The geometrically simplest parallel mechanism has the structure of an octahedron and it is designated as a "3-3 platform" since there are three connecting points on the base and three on the moving platform. The double connection points shown in Figure 3 produce a very simple geometry. However, there is a very serious mechanical disadvantage. It is very difficult to design the necessary concentric ball and socket joints

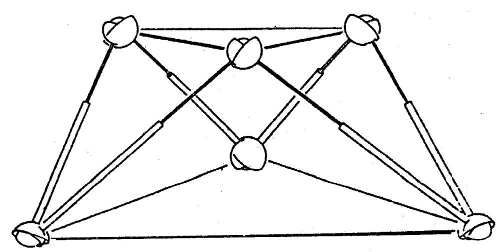

Figure 3: 3-3 Platform

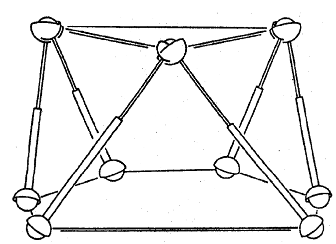

Figure 4: 6-3 Platform

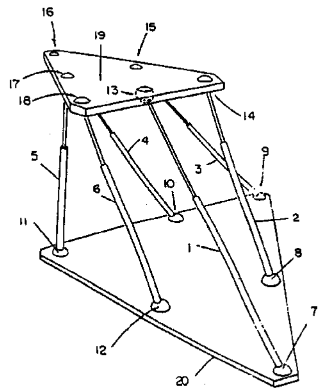

Figure 6: Special 6-6 Platform

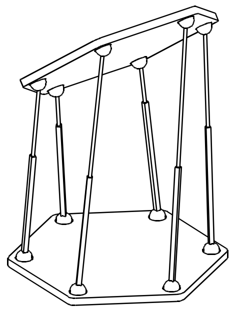

Figure 5: 6-6 Platform

at each of the double connection points without mechanical interference. It is necessary to separate the double connection points to overcome mechanical design problems.

When the double connecting points are separated, the complexity of the forward kinematic analysis for the platform increases. It should be noted that there are multiple solutions. This means that there are multiple closures of the mechanism and that there exists multiple ways it can be assembled. Each assembly yields a different position and orientation of the platform while possessing the same set of six leg lengths.

It is, of course, possible to perform numerical iterations (an optimization using six independent variables) to obtain the position and orientation of the platform. However, it is well known that such iterative solutions have a tendency to “jump” from one closure to another. From an implementation viewpoint, this is undesirable. It is far more desirable to derive a single polynomial in a single variable, the solution of which yields all possible locations of the moving top platform. The desired solution can then be extracted from this finite set of all solutions. Such a solution is said to be in “closed-form”.

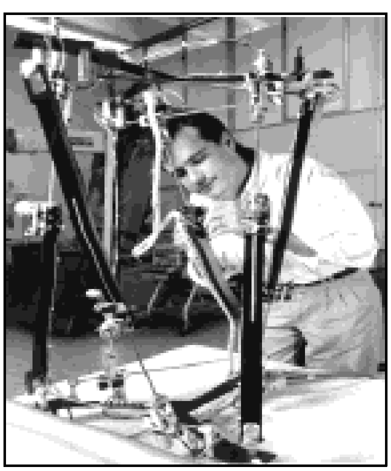

Figure 7: Table Mounted SKIP

It was only recently that the closed-form forward analysis for the geometrically simplest 3-3 platform was solved by Griffis and Duffy [3]. Briefly, an eighth degree polynomial solution was derived, and this has been extended to a 6-3 platform (Stewart’s original platform [4], see Figure 4). It would be desirable to perform the forward analysis for a general 6-6 device as shown in Figure 5, however this is unrealistic. The closed form equation is a 40th degree polynomial [5] and this is computationally impractical for real time control. Griffis and Duffy have invented two platforms [6] which provide the benefits of both the 3-3 and general 6-6 platforms. These platforms allow for the simple analysis of the 3-3 with an eighth degree polynomial, and allow for the mechanical benefits of the general 6-6 by eliminating mechanical interference. One of these new platforms is shown in Figure 6. A table mounted model SKIP has been fabricated and is shown in Figure 7.

As previously stated, the necessity for a simplified closed-form forward kinematic analysis (specialized geometry) manifests itself whenever the mechanism is to control force and position simultaneously. The requirements of specialized geometry and good mechanical design (no mechanical interference) are satisfied by the platforms that have been patented by the University of Florida. It is the union of the theory of Kinestatic Control [7] [8], developed under an NSF grant and these platforms that yields the KP concept.

3. Design Specifications

The Kinestatic Platform design specifications are established by the milling application. Cutting speeds are dictated by the available spindle power. Workspace mobility, accuracy, and payload are established by the expected geometry and material of parts to be fabricated. The machine volume is established by the space available for augmenting a typical 3 hp to 5 hp knee mill. The resulting design specifications are listed as follows:

Workpiece Speeds
- Cutting: 20 in./min (508 mm/min)
- Rapid Traverse: 100 in./min (2540 mm/min)

Work Space Mobility
- 8 inch (203.2 mm) cube:
  - translation: ±4" in Cartesian axes x, y, z about a null position
  - rotation about a line in or parallel-to xy plane: ±20°

Accuracy:
- Path following within a normal envelope of radius < 0.001 in. (0.0254 mm)

Payload:
- 50 lbs (22.68 kg)

Machine Volume:
- Width × Height × Depth: 60×28×48" (1524.0×711.2×1219.2 mm)

The goal of this research is to develop a low cost augmentation of an existing 3 to 5 hp knee mill, allowing 5 axis computer controlled milling within the cutting capacities of the existing spindle. The reported design specifications are to be valid over the quoted work space volume. The proposed geometry must meet these requirements but can exceed them in some areas of the workspace. It should be noted that the description of workspace mobility for a platform based machine tool in conventional nomenclature is cumbersome. The machine has only one obvious linear axis (typically noted “z”) and one rotational axis about the linear axis. Assigning two more linear axes “x” and “y” orthogonal to “z” leads to useful information about the rectilinear capabilities of the machine. It is proposed that “roll”, “pitch”, and “yaw” are not as useful in evaluating a platform for a milling application. Instead, the range of rotation about any line in a plane parallel to the x-y plane is quoted. This measure is more appropriate in evaluating the application of the proposed machine to a specific contour.

4. Kinestatic Design (Parametric Geometric Design)

Generally, when designing a platform, the anticipated application must be considered. The application defines the loading conditions, workspace geometry, and dexterity. Based on the specifications that evolve for a particular application, the platform parameters are optimized.

The number of design variables for a general kinestatic platform is many. To examine various designs in a proficient manner, a methodology has been developed that reduces the number of design variables to a manageable quantity and facilitates the kinestatic design of the platform. The kinestatic design, coupled with the application specifications (cutting force, frequency) and mechanism dynamics, provides the specifications for the actuators (connectors).

The kinestatic platform’s kinematics can be described parametrically by defining two triangles, the base and the top, and six side pivot connections. Since the expected loading is arbitrarily distributed in the workspace, the platform should exhibit as far as possible uniform

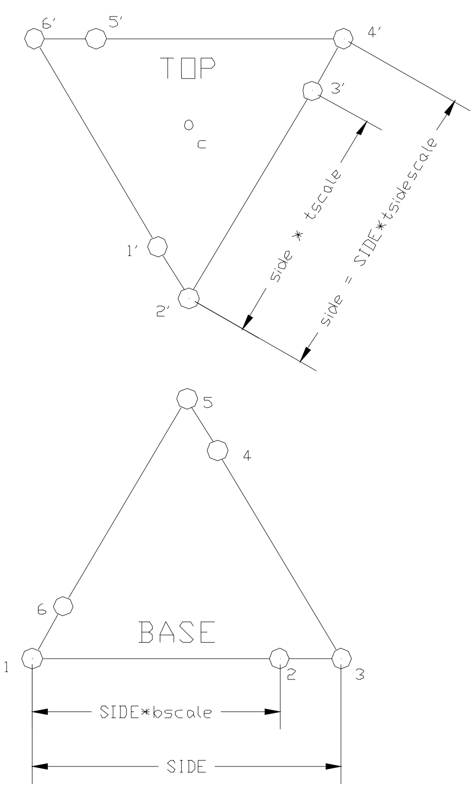

Figure 8: Geometric configuration of the base and top of Platform.

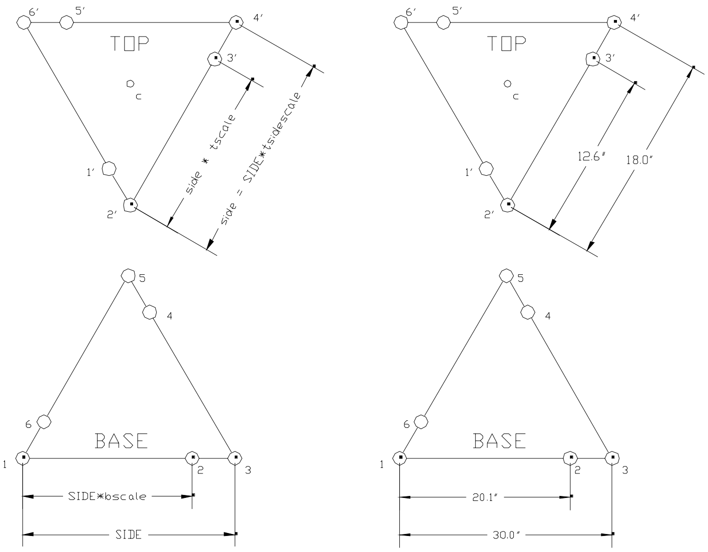

Figure 9: Design parameters and associated values for the platform.

stiffness and response over the workspace. This design requirement is best met by using equilateral triangles for the base and the top. The position of each connecting pivot should be symmetric on each triangle.

An equilateral triangle can be defined by one parameter, the side of the triangle. Figure 8 shows the parametrization of the top and the base of the platform, respectively. The base triangle is defined by the parameter SIDE. The top triangle is related to the base triangle by the parameter *tsidescale*. The base and top side pivot points are defined by the parameters *bscale* and *tscale*, respectively. Another parameter, *hscale*, is used to define the distance between the top and the base of the platform. This parameter is also a function of SIDE; i.e., if *hscale* is equal to 1, the distance between the top and the base of the platform would be equal to SIDE, and any variation of this parameter would vary the height of the platform accordingly.

The special 6-6 parallel mechanism requires that the pivot connection lie on the line that contains the two vertex connections. For example, in Figure 8, side pivot connection 2 must lie on the same line that connects 1 and 3. The parameters *bscale* and *tscale*, along with the parameters SIDE and *tsidescale*, are used to establish the position of the side pivots as shown in Figure 8. On the base, if *bscale* is less than 1, then the side pivot lies between the two vertex connections and vice versa. The same is valid for *tscale* with respect to the top.

A given kinestatic platform can have a large kinematic workspace, but its stability over the workspace may vary enough to have significant impact on the top platform’s performance as it traverses the workspace. The parameter SIDE allows the mechanism to scaled up or down. The other parameters define the kinematic workspace and variation of static stability across the workspace. This methodology entails finding suitable parameters that define a kinestatic platform that has a well behaved (desired) static stability over the workspace that meets the design specifications.

It is relatively simple to compute a unique set of connector lengths when the platform’s location is specified. This is known as the inverse displacement analysis and is useful in developing the expressions for the dynamic state of the system. The platform’s location must be specified when it is used, for example, to present a workpiece to a machine tool. The displacement and orientation of each of the connectors that actuate the platform can be determined, once the platform's motion time history has been specified. From the motion specification of the platform, the velocity and acceleration state of the platform is computed and subsequently, the actuator speeds and accelerations are determined.

A parameterized dynamic analysis of the platform was also performed which allowed the investigation of platform dynamics for various design configurations throughout the mechanisms' workspace. This analysis was based on a dynamic analysis model by Baiges and Duffy [9], which yielded the dynamic and static loading conditions on each actuator of the platform. These results, along with payload (200 lb_m (90.27 kg)) and cutting force (200 lb_f (889.6 N)) estimates, allow the actuator power requirements to be established.

The generation of a particular platform design entails the specification of the mechanism design parameters discussed above. The selection of these parameters is based not only on the inherent characteristics of the platform and desired performance specifications, but also on the work-space to be traversed, the physical location of the platform with respect to the milling head, and the type of milling operations to be performed. Thus, the design parameters selected represent a realistic combination of parameters that meet the above kinestatic, dynamic, and work space requirements.

Based on the physical design constraints and the performance specification requirements, a platform was designed with the following parameters, schematically shown in Figure 9:

- **SIDE** 30.0" (762 mm)
- *tsidescale* 0.60
- *bscale* 0.67
- *tscale* 0.70
- *hscale* 0.60

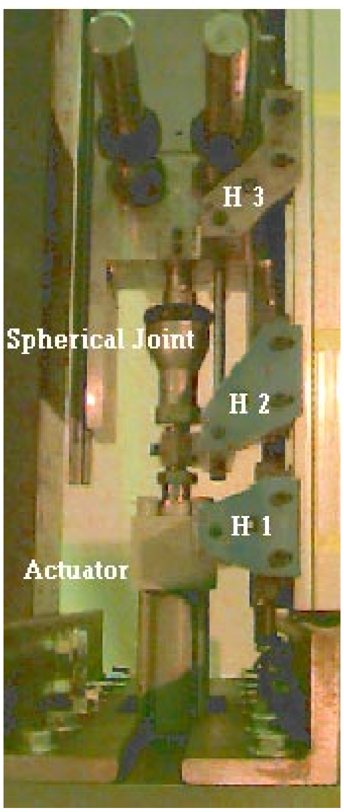

Figure 10 View of the Joint and Sensor attachments.

Tests were conducted on the actuator designed and built to determine its behavior and evaluate the feasibility of its use as a self-contained actuating and measuring device in the Kinestatic Platform. A summary of these results is discussed in Section 7. The results from the tests conducted of the project have lead to several significant inferences as to the design of future platforms.

5. Hardware Implementation

After finalizing the design of the platform, a test apparatus was required to test the capabilities of the actuators. A test bed was designed so that actual loading conditions could be simulated as well as joint behavior could be studied. The final design which has been fabricated and assembled was shown in Figure 2. Figure 2 shows the test fixture in its entirety. The fixture consists of three distinct sub-systems. The first part is the support structure, which provides the mounting surfaces for all the components. It consists of a flat plate supported by four heavy pipe columns, two vertical steel C-channels to provide the structure for mounting the sensor; and angled steel L-sections to support the vertical channels and provide rigidity to the structure in two planes. The second major component was the moving block which is mounted to the end of the actuator and was used to vary the loading conditions on the leg. The third and final component is the hydraulic actuator.

The actuator is mounted on a base plate with a specially designed Hooke joint (2 degrees-of-freedom) which attaches underneath the base plate. The actuator connects to the joint via mid mounted trunnions which fit the joint. The vertical C channels are used to mount the guide rails. In this particular instance, the guide rail blocks were mounted on the C-channel face (fixed) and the rail was attached to the moving block, so that the during a motion, the rail would move instead of the blocks. This was again necessitated by the constraints referred to earlier. The actuator is connected to the moving block by an off-the-shelf ball-and-socket (spherical, 3 degrees-of-freedom) joint. The moving block consists of three compartments which provide space for lead blocks to be used for pre-loading the actuator. The moving block also houses two one inch diameter steel rods, mounted parallel to each other, which extend equally in either direction to provide balanced loading on the actuator. These rods act as support posts for loading the actuator for various testing loads. The loads can be varied by using specially designed steel plates. A linear scale (Heidenhain LS-106C) is mounted on one side of a C-channel. Three sensor heads are used with the scale to determine the displacement of the actuator as well as deflections in the joints (Figure 10). The first head (H 1) is attached to the actuator frame for determining the deflection in the Hooke joint since the actuator frame is rigidly attached to the trunnion mounts. The second head (H 2) is attached to the actuator rod just below the top spherical joint. This head measures the actual displacement of the actuator. The third (H 3) head is mounted on the top end of the spherical joint and is used to determine the behavior (deflection) in the spherical joint. All three heads are collinear and use the same linear grated optical scale which provides a resolution of 3 microns (0.003 mm, 0.00012 in.).

6. Testing & Evaluation

Tests were conducted on the connector to determine its behavior and evaluate the feasibility of its use as a self-contained actuating and measuring device in the Kinestatic Platform. These tests encompassed the evaluation of the displacement accuracy of the actuator under various loading conditions, the determination of the joint deflections and their effects, testing of various control laws for control of the actuator, and from the results of these tests, the development of a control strategy that produces a desired motion within milling specifications. Such a control algorithm is needed to reduce the effects of overshoot, lag, and harmonics, while providing effective disturbance rejection.

Tests were conducted using Proportional (P), Proportional-Integral-Derivative (PID), and feedforward control strategies. Under these control strategies, step response tests were used to determine the behavior of the system. Also, experimental determination of position accuracy (position deadband) and deflection of the joints were performed. These tests were performed under various loading conditions. Only a small subset of the test results are presented in the next section.

Control Algorithm Development

Control algorithm development for the platform is driven by performance specifications of the top platform. Positional accuracy and repeatability are primary concerns when performing milling operations. The development of the control algorithm for the platform connectors is not a simple undertaking. Requirements for connector performance are greater than that for the top platform and often meeting these specifications conflict with each other. An example is meeting positional accuracy with minimal overshoot while having a “fast” response to movement commands. The specifications required for the performance of the top platform were discussed in Section 3.

The actuation device chosen to drive the platform connector was a hydraulically actuated cylinder. Factors affecting accuracy of the system include the transfer functions of all the components in the loop including the feedback portion of the loop. Components in the control loop are the cylinder, the servo valve, the servo amplifier, the load, the sensor used to quantify connector length, and digital to analog conversion hardware. Using parameters for components selected for the control system implemented for the platform connector, a repeatable error range of 0.0004" (0.0102 mm) to 0.001" (0.0254 mm) was calculated. While the parameters used in the calculations do range depending on a number of system parameters (i.e. the mass of the system), lenient values were chosen in the above calculated values. Another parameter calculated was a tracking error. The tracking error defines how the system will lag the commanded position during maximum speed. This is also know as the velocity error and is a positional error due to velocity. The calculated value for the tracking error was 0.026" (0.6604 mm). Note that variations in the gain constant, the hydraulic natural frequency, and the damping ratio occur at different operating points and cause considerable frequency response shift. This should be kept in mind while viewing test data plots.

Development of a control strategy to meet all the platform performance specifications will require a combination of feedback techniques to improve actual system performance from that estimated from theory. While a straight proportional feedback control law can give satisfactory results to some of the performance specifications, it can not meet all the requirements. Problems with proportional feedback control include the presence of steady state error and position overshoot if the system damping ratio is less than unity. Also, changes in the load on the cylinder will alter system parameters that affect system stability. To investigate system dynamics for the connector, step tests were performed using a proportional controller with low and high gain values. Plots for a 0.040" (0.0102 mm) step are shown in Figure 11. From these plots, it can be seen that for the high proportional gain, a considerable overshoot occurred. For the low proportional gain response, no overshoot was observed but stiction in the cylinder seals was apparent. The step of 0.040" (0.0102 mm) was selected to ensure that the servo valve opening was not saturated. However, for a brief period of time, the valve was almost open to capacity. Steady state error for the step responses were on the order of magnitude of 0.0002" (0.00508 mm) to 0.0008" (0.0203 mm).

Step responses were also performed using a PID controller. Results for a 0.040" (0.0102 mm) step are shown in Figure 12. For these tests, the PID controller gains were selected using Ziegler-Nichols Rules as starting values. Overshoot and steady state errors were reduced using this control scheme as expected. However, response time increased. While a PID controller helped meet some of the requirements, PID control alone will not be satisfactory. Development of an adaptive control strategy will have to be made to increase closed loop stiffness and may include estimating system statics and dynamics, possibly inclusion of a minor velocity feedback loop, and prediction of external loads.

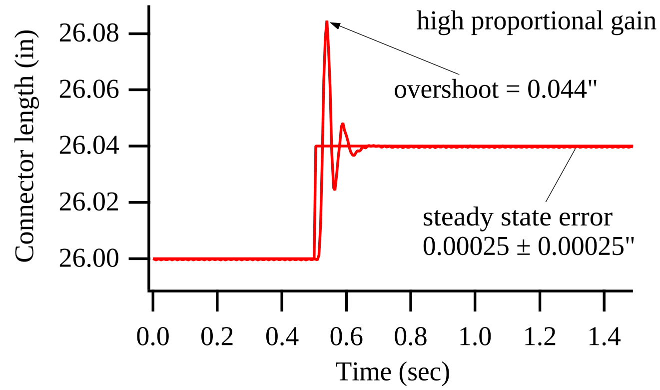

Figure 11a: Step response using a Proportional controller with high feedback gain.

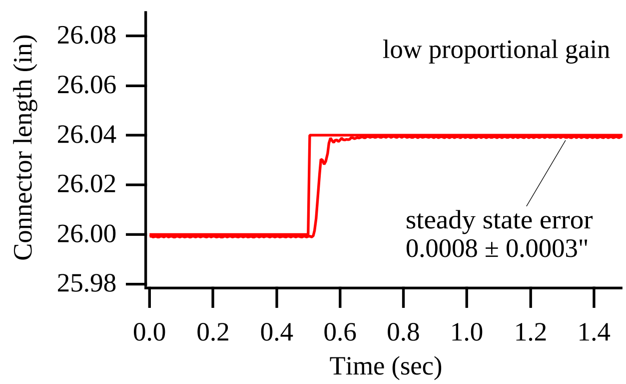

Figure 11b: Step response using a Proportional controller with low feedback gain.

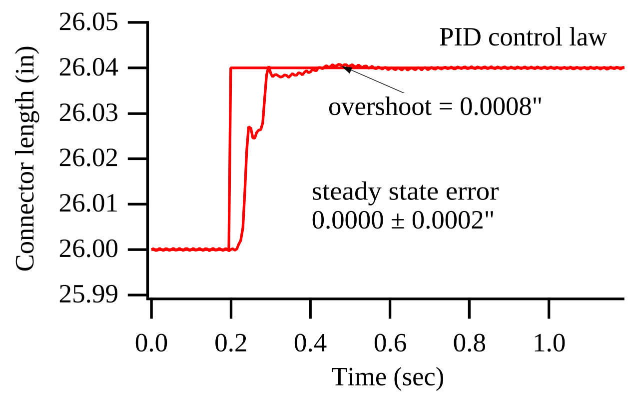

Figure 12: Step response using a PID controller.

After a sufficiently high static position accuracy was obtained, the next step was to investigate the performance of the system while moving the platform through a planned path simulating a milling operation. In order to reduce the velocity error (commonly know as lag) inherent in any controlled system, a simple feed forward lead scheme was implemented. The graph in Figure 13 shows results from a constant velocity extension path of 0.5"/sec (12.7 mm/sec) for the connector.

Since the position and position set point are very close together on the plot, the position error is shown with the respective axis on the right side of the graph. The graph shows the results from two tests. The first test represents the use of a position set point from the planned path without any lead. The second test shows the results from supplying a position set point 3 time steps in advance of the planned path. By leading the position set point, the error between the desired position set point and the actual position was reduced. Maximum errors inside the region of interest (0.75 to 1.75 sec) were 0.00055 in (0.014 mm).

The implementation of this scheme demonstrated that the desired position accuracy at low velocities could be achieved with a simple feed forward lead strategy. However, a model of system components will be required to identify and characterize control parameters so that performance within the desired criteria can be achieved. This would facilitate the compensation/correction for various system behaviors such as the presence of stiction in the hydraulic cylinder and loading/unloading effects during the milling operation.

Joint Deflections

Joint deflections for both Hooke and spherical type joints were investigated under loading conditions similar to those expected in contour milling. The Hooke joint exhibited deflections with a magnitude of 0.0004" (0.01 mm) and the spherical joint exhibited deflections with a magnitude of 0.0008" (0.02 mm). Both joint deflections magnitudes are significant with respect to tool position (top platform) error. From investigation of accuracy requirements for this geometry, it was determined that leg length accuracy must be three times the desired tool point (top platform) accuracy. These results lead to several observations concerning the viability of measuring actuator lengths to prescribe tool position.

7. Results and Conclusions

7.1 Summary

The major accomplishments of this research were as follows:

- Performance specifications were developed for the milling task.
- A geometric design was developed based on dynamic modeling results.
- One connector was fabricated and tested under load conditions.

The results of this undertaking can be summarized as follows:

- It is apparent that the control strategy significantly affects the position accuracy. Thus, a control strategy needs to be developed that would adapt to changing conditions in real time and be able to compensate for disturbances to maintain the required accuracy deadband.
- Measured joint deflections were of significant magnitude. Joint deflections must be accounted for by measurement, estimation, or revision of the design so that the joint deflection effect on the tool point position is minimized. Separation of actuation and metrology frames leads to the minimization of this effect.
- The results of the PID controller show that sufficient positional accuracy can be achieved with the current actuator/sensor configuration.
- From the simulated platform motion data, the PID controller exhibited that it is possible to accurately follow a commanded path with a lag. In the next phase of the project, work will be done on the development of a control strategy that compensates for this lag.
- Hydraulic actuation was used in the Test bed as the most cost effective means of identifying control and joint deflection accuracy issues. Electric actuators with ball-screw drives can be used in the future prototype with very similar results.
- A separate metrology frame will allow for accurate determination of the position and orientation of the top platform. The influence of the joint stiffness and thermal and load deflection on the top platform position and orientation are reduced with this approach. Milling specifications require accurate location of the top platform and control of the actuator length. Test results from the Test bed apparatus leads to the conclusion that the actuator can be controlled to meet these milling specifications. A brief discussion on the merits of using separate frames follows.

7.2 Observations and Future Work

The accurate measurement of the actuator lengths is of prime importance in determining the overall accuracy of the platform. There are several disadvantages in measuring connectors that incorporate actuators. These disadvantages include increased connector size, and the inclusion of joint compliance, actuator compliance, and thermal deformation. These disadvantages require a carefully designed actuation connector which leads to increased cost and complexity. The order of the accuracy error in connector length determination, when considering deflections due to varying mechanical and thermal loads, is significant with respect to desired accuracy of the moving platform location as

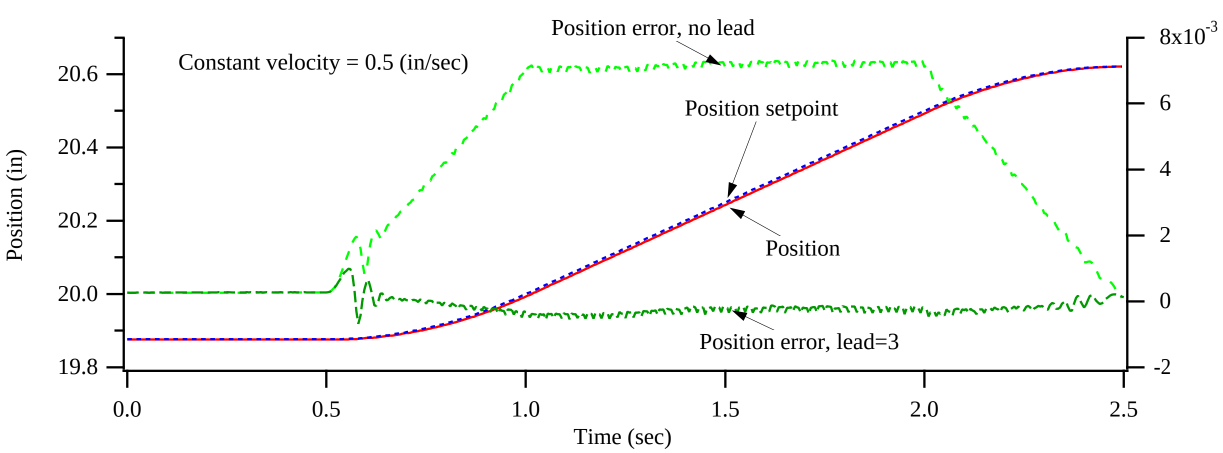

Figure 13: Tracking errors for a constant velocity extension of 0.5 in./sec.

demonstrated by the results of this research.

These observations lead to an investigation of an alternative platform design. At the outset, it would appear that the separation of the measurement system from the actuation system is desirable. This would remove load and thermal deformation from the top platform location determination. The separation of actuation connectors and measurement connectors is applicable to a platform configuration in which at least the measurement connectors are arranged in one of the configurations for which a simple forward analysis exists, i.e., in the special 6-6 configuration for which all analyses are known. The measurement connectors can be considered as a “metrology frame”, responsible for determining the location of the work piece with respect to a global frame. This information can be used to generate control commands for the actuation connectors, or "actuation frame”. The accuracy of the metrology frame would be dependent on the same variables as the initial design except that it will not incorporate the undesired effects that the actuation connectors undergo such as loading resulting from connector inertia, moving platform inertia, and external loading. The actuators are power generating devices with a significant amount of unwanted heat generation. The load deflections and thermal deflections that a passive connector undergoes are based on joint friction, gravity effects, and inertia of the passive connector, effects whose influence on accuracy can be estimated and whose magnitude is expected to be negligible. This leaves as accuracy issues for a passive connector the accurate location of connector joints and the measurement system resolution. It allows the direct evaluation of system accuracy and provides a machine tool in which the work piece location is directly quantifiable.

The performance requirements of the intended application are expected to prove difficult to achieve. It can be seen that individual actuator lengths can be controlled within requirements but other sources of error are not as easily compensated. From our work, we see the necessity of an off-line command generator along with dynamic simulation as a starting point for meeting performance requirements. The “real time” control system will include error compensation based on the accurate determination of the tool point position. It is also expected that a minor loop around each actuator will be incorporated in hardware. Figure 14 is a simple block diagram of a possible system.

The incorporation of the metrology frame eliminates the condition that the actuator frame needs to be in a special 6-6 configuration since it is not involved in the direct measurement of platform position. Thus the actuator frame can be a general 6-6 platform without the need for a closed-form forward and reverse solution. This allows for the placing the actuators in such a manner that they provide maximum effective thrust in the direction(s) of interest while minimizing the possibility of affecting the dexterity of the platform due to actuator collision. Another freedom that this geometry allows is the location of actuator pivots according to the workspace available under the milling head. It is obvious that the separation of the metrology and actuator frame provides greater freedom in designing a platform that meets the desired performance specifications.

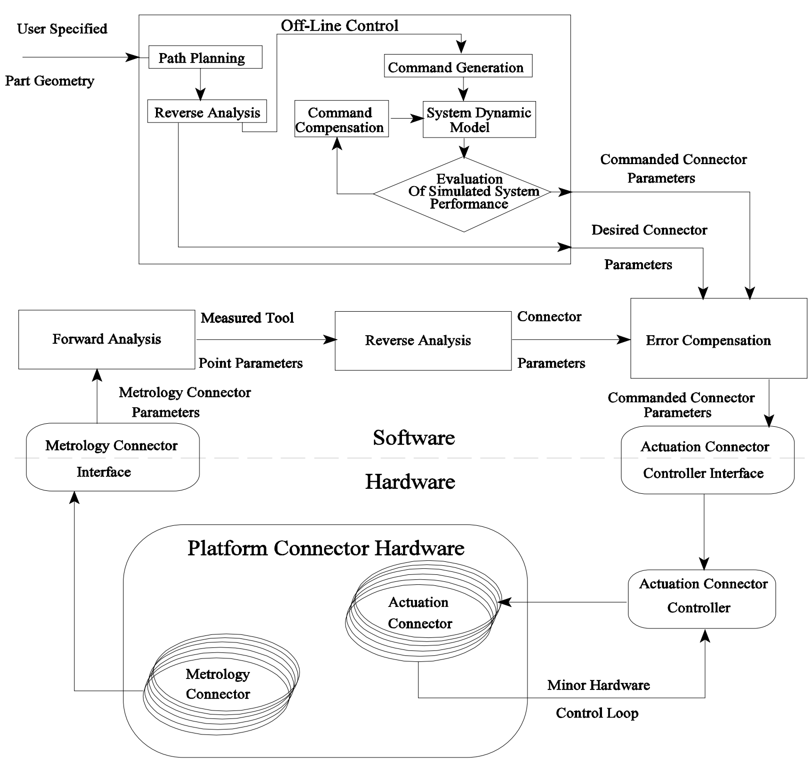

Figure 14: System Block Diagram

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