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Feasibility study of compliant assembly device applications to robotic assembly /

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FEASIBILITY STUDY OF COMPLIANT ASSEMBLY DEVICE APPLICATIONS TO ROBOTIC ASSEMBLY

by

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12/13/84
Date

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Acknowledgment

I wish to thank my major advisor, Dr. Emory W. Zimmers Jr., for his support throughout this thesis. I extend special thanks to Mr. Frank L. Bracken for his continued interest and encouragement concerning the intergrated design.

I dedicate this thesis to my parents for their support and patience during my years at Lehigh.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

ABSTRACT

This thesis presents a feasibility study for the development of a compliant assembly device. Compliant, in this context, means a yielding by displacement due to an external force. Human dexterity during manual assembly operations inherently possesses a compliant action. The ability to allow an object to yield relative to another object during assembly is essential for an efficient assembly operation. Several examples are presented that illustrate the need for compliance during automated assembly. The concept of including computer controlled compliance into the assembly machine is demonstrated with a preliminary device design for a particular robot with an overhead gantry cartesian configuration.

The device is a computer controlled pneumatic compliant-locking coupling that can be used for many automated operations. It is designed to give an assembly robot the capability for selective compliance and provide accurate feedback as to the amount and direction of the compliance.

The device allows the robot to perform more complex assembly operations with greater speed and less programming effort. The selectively passive compliant function facilitates the use of design features on the parts, pallets or end effecter to locate workparts relative to their mating parts or the end effecter. The accurate feedback function is used for calibration and inspection.

The integrated design considers the device's functionality as defined by its applications and its automated fabrication and assembly in a flexible manufacturing environment. Features are included on the components to allow automated handling and orientation during fabrication and assembly. The design contains a minimal number of complex and nonfunctional parts. All the fabricated components can be manufactured with standard CNC equipment and assembled with a simple four-degree-of-freedom robot. The design allows for certain device characteristics to be modified, in order to customize it for specific applications. These changes will not require new tooling to fabricate and assemble the devices through a flexible manufacturing system. A possible automated fabrication and assembly process is also suggested.

INTRODUCTION

Automated assembly has proven itself as an economically viable alternative to manual assembly for many simple operations.[1] With proper component design and more versatile assembling equipment and techniques , many more complex assembly operations may also be economically accomplished. It is believed by many that designing the components for automated assembly is the most effective way to achieve a successful automated assembly.[2] As an example, a before and after design for automated assembly of an electro-mechanical device is shown in Figure 1.

Clearly the redesigned device would be easier to assemble with automated equipment than the old design. If, however, the assembling equipment is not sufficiently flexible and sophisticated, the efficiency of the assembly operation

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1 G. Boothroyd, "Design for Producibility - The Road to Higher Productivity", Assembly Engineering, March, 1982, p. 1.

2 Frank Bracken, Design of Data Processing Equipment for Automated Assembly, (Endicott, NY: IBM, 1983), p. 1.

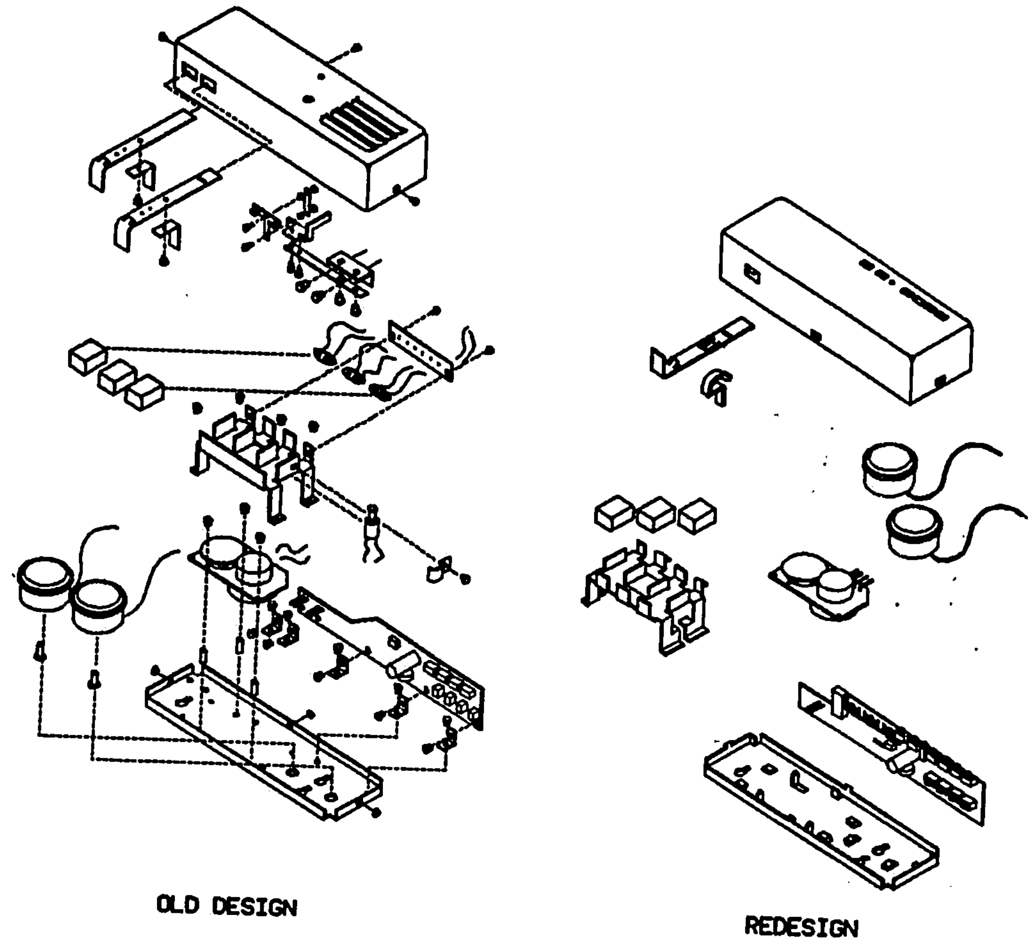

Figure 1. Design for Automated Assembly Comparison: Source: M. A. Ritland, "Redesign of a Home Security Device." Case study, Lehigh University, Bethlehem, PA , May 1984.

may be compromised. The assembling equipment's capabilities must work with component design features to facilitate the most efficient assembly. On the other hand, in a flexible manufacturing environment, assembling equipment must be versatile and not product specific. Therefore special jigs, fixtures or complicated programming should be avoided. If possible the automated assembly equipment should provide basic and widely used capabilities as standard features. This would eliminate the need for many special jigs and fixtures.

An important capability would be to allow the assembly machine's manipulator to selectively comply to a slight external force. This function is extremely useful when combined with a chamfered design feature (See Figure 2). The forces on the mating parts tend to align them, provided relative motion (compliance) between the two parts is allowed.[3] Thus, with combined design features and the compliant manipulator capability, a successful assembly is accomplished.

This thesis descibes, through specific examples , the need for selective compliancy in the automated assembly environment. It then presents the basic design of a device that provides this required selective compliancy.

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3 D. Whitney et al., "Designing Chamfers," The International Journal of Robotic Research Vol. 2, No 4, (Winter 1983).

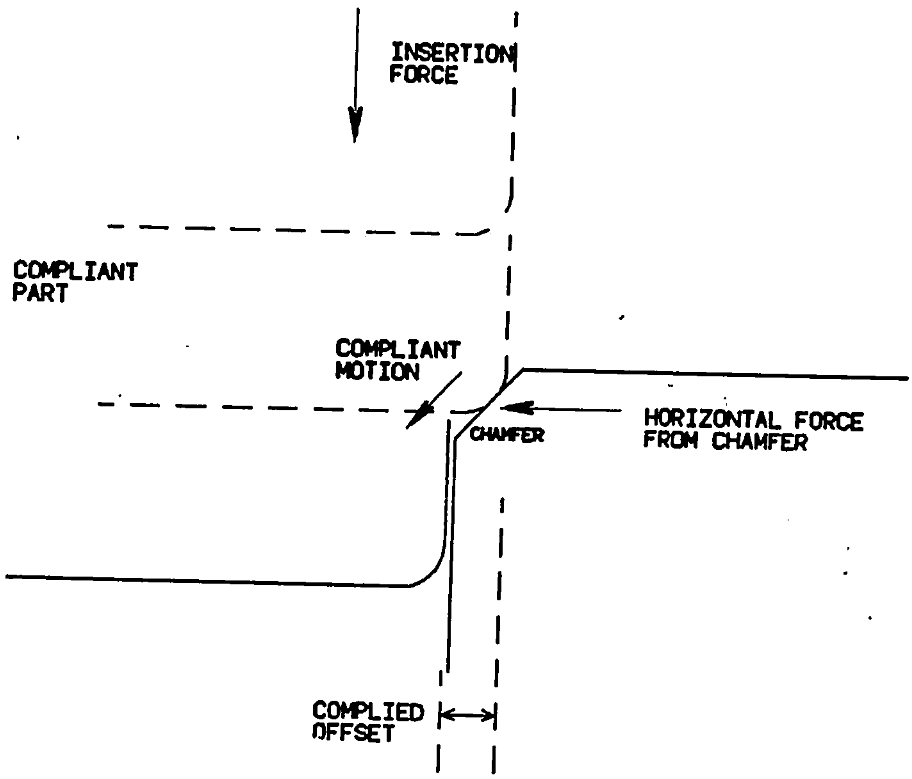

Figure 2. Chamfered Insertion

Although this device is used to help assemble products in a flexible manufacturing system, it in itself is a manufactured product. It is only fitting that the device be designed, fabricated, and assembled in an automated environment. This requires an intergrated design approach that considers fabrication, assembly and other factors in addition to device functionality.

BACKGROUND INFORMATION

For selected applications it has been shown that if the components are properly designed, ninety-five percent of assembly operations can be accomplished from one side and with the aid of gravity, most assembly operations can be performed from the top down with a three or four degree of freedom assembly machine.[4] When possible the design engineer should use the force of gravity to aid in the assembly operation, since this is a free form of compliance.

The effect of gravity acts as a compliant force in the vertical direction. All assembly machines have some degree of mechanical play due to tolerance and wear in their couplings and joints.[5] This play allows compliance, but it may be limited, irregular, or require excessive external force to be realized. The mechanical compliance may also be the

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4 Interview with Frank Bracken, IBM, Endicott, NY, January 25, 1984.

5 E. George, "Evaluating Bearings for Robots," Machine Design, April 7, 1983, p. 81.

wrong type. For instance, external forces may cause tilted compliance at the manipulator end, which may not be desired. For compliance to work properly with design features to assist assembly, the compliance must be selective (compliant/rigid) and also limited in degree and direction.

If we consider top down assemblies with a three or four degree of freedom assembly machine; that is, where the end effecter's orientation remains parallel to the vertical and horizontal working planes, then the most effective form of compliance would be one that allows motion parallel to the X Y plane (See Figure 3). Tilted and rotational compliance with reference to the Z axis would tend to cause jamming and should be avoided.

At certain times compliant motion may be invaluable, but at other times it may be a nuisance. Therefore, compliant capabilities should be selectively controlled. The controlling program should be capable of turning the compliance on or off, and returning the device to the centered position. Later in this discussion application examples will be presented that illustrate the uses and benefits of such a compliant function.

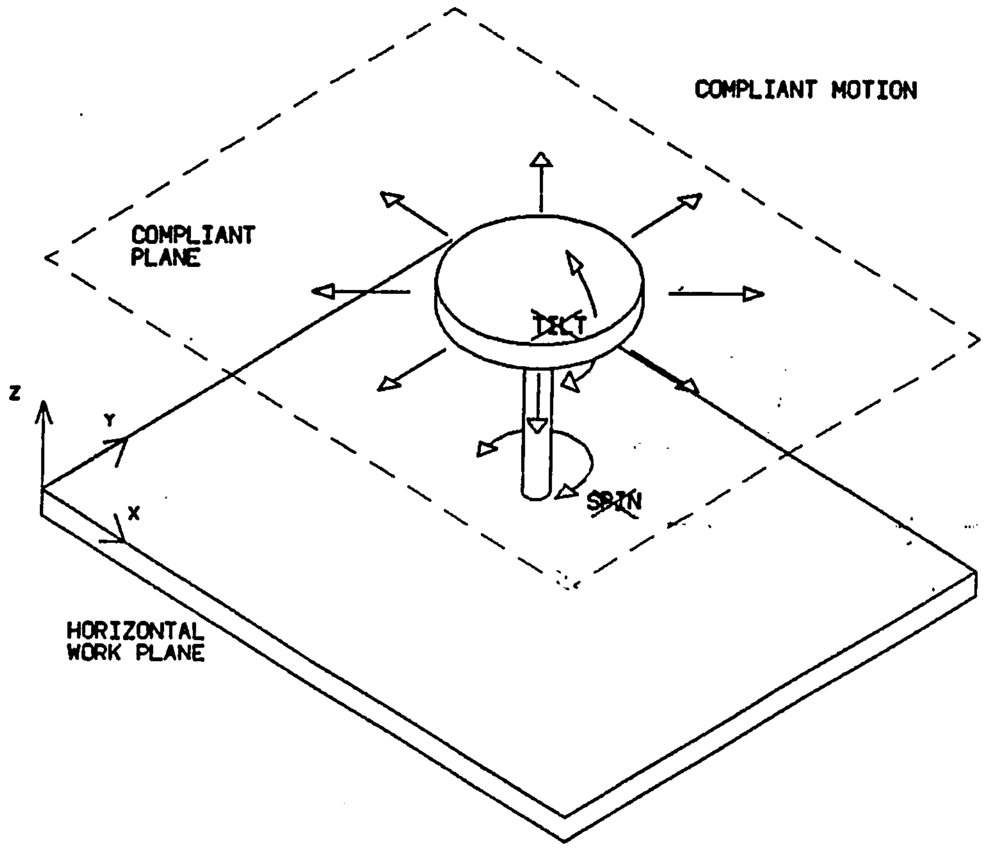

Figure 3. Compliant Motion

DEVICE DESCRIPTION

A device has been conceived that will provide selective and limited compliance in a plane parallel to the horizontal X Y plane. In addition the device is equipped to provide accurate feedback as to the amount and distance of the complied offset. An exploded isometric view of the device prototype is presented in Figure 4.

This device functions as a coupling between the end of the automated manipulator and its end effecter. The location of the device on the RS-1 robot is shown in Figure 5. It functions in the following three modes.

1. Centered and locked

2. Compliant

3. Locked offset

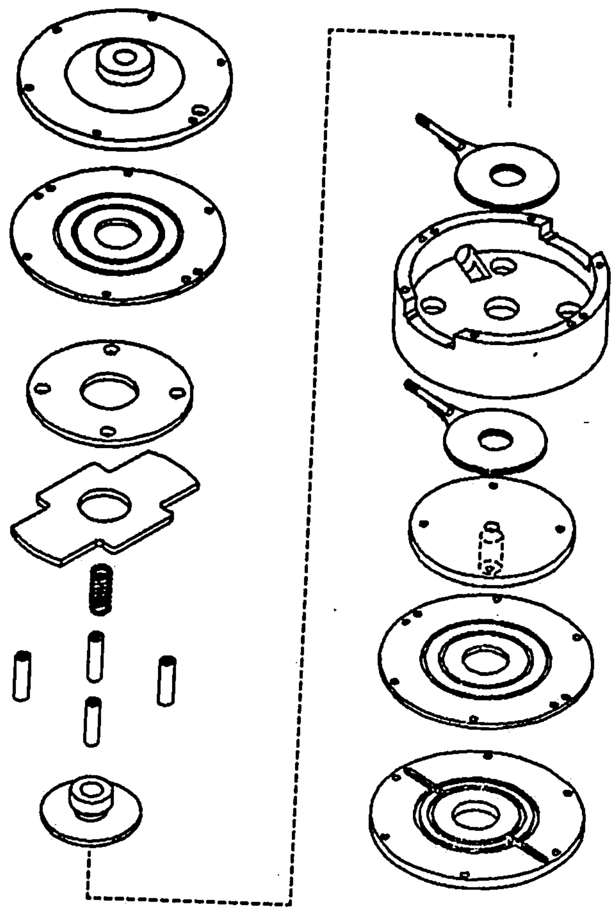

Figure 4. Compliant Device Exploded Isometric View

These modes are computer selected and pneumatically actuated and digital feedback is available as to the component complied offset in the X and Y directions.

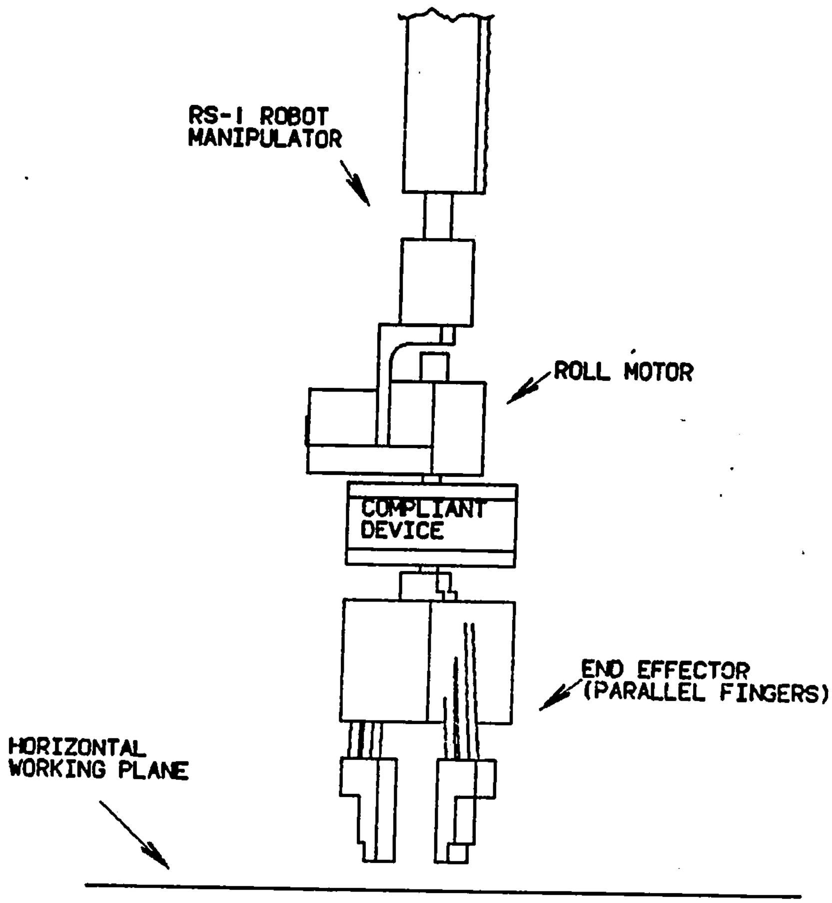

Figure 5. Compliant Device on the RS-1

Mode 1 is the device's passive state. It is locked in the centered position. Without externally supplied air pressure it acts like a solid coupling as long as field loads are within specified maximums. Being centered and locked without air pressure is advantageous , because if compliance is not needed the device does not have to be supplied with air. In addition if the air supply system malfunctions the device will be centered and rigid.

Returning and locking to the centered position is an absolutely essential function, if automated accuracy is to be maintained. This concept is explained more thoroughly in a later section. The X and Y component offsets from the feedback system are at zero in this mode.

Mode 2 is the compliant mode. In this mode a limited low resistive compliant motion is allowed in any direction within a plane parallel to the horizontal plane. Any forces acting parallel to the compliant plane will cause motion in the direction of the force. Motion tilted out of the compliant plane or rotation about the Z axis is prevented. The feedback system continuously provides the X and Y components of the offset from the centered position.

In mode 3 the device again acts like a solid coupling except it is locked in an offset position (See Figure 6). This allows the end effecter to be locked in a new frame of

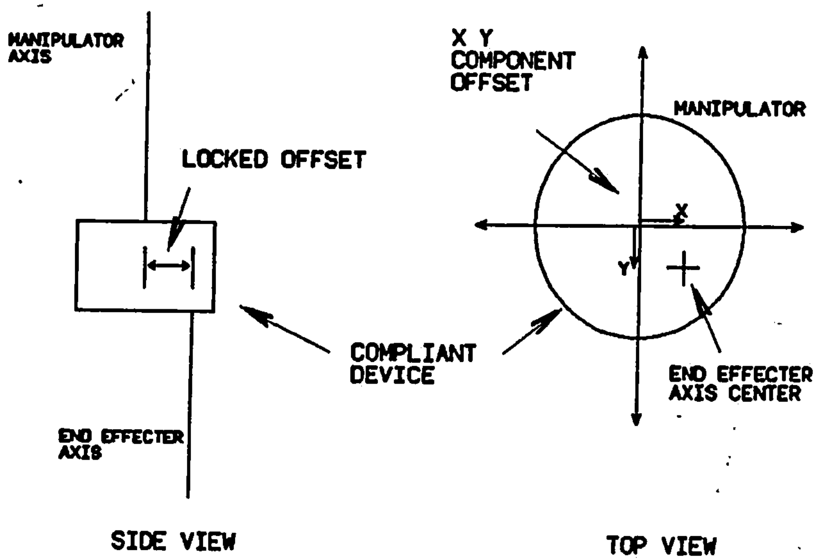

Figure 6. Locked Offset Mode

reference that was determined during the compliant mode. The X and Y component offsets from the feedback system measure the complied offset from center.

Several application examples will now be discussed that illustrate some assembly problems and demonstrate how the device modes would function to solve these problems.

APPLICATION EXAMPLES

The insertion of a cylindrical peg into a round hole may be one of the most common and basic assembly operations yet it still can be one of the most challenging. The insertion is dependent on the relative location and parallel orientation of the peg and hole centers.

An example is the inspection operation of a robot testing a through hole with a "go no go" gauge (See Figure 7). Assume the hole is located in the work volume accurately. Further assume the worst case clearance between the gauge and hole is 0.025mm. Thus, to insure insertion the centers must be aligned at least within 0.025mm. However, the end of the robot's manipulator is only repeatable to 0.1mm with an inherent mechanical compliance of 0.05mm. If both the gauge and hole corners are square, direct insertion cannot be guaranteed (See Figure 8 (a)). The design solution is to chamfer the gauge, the hole, or both. Now if the gauge is chamfered 0.125mm at a 45 degree angle, then the centers can be offset as much as 0.125mm (Figure 8 (b)), because the

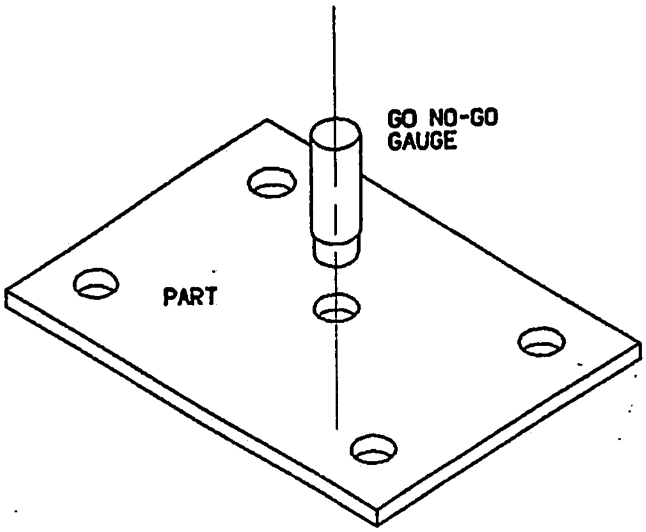

Figure 7. Hole Inspection

chamfer will align the centers during insertion. The robots repeatability can achieve this positioning with consistency. The gauge must now move relative to the hole to allow the chamfer to guide the gauge in. The manipulator's inherent mechanical compliance will allow only 0.05mm and, when forced, will allow undesired tilt. Again, proper insertion can't be guaranteed (Figure 8 (c)).

A solution is to allow the gauge to comply at least 0.125mm with the slightest horizontal force from the chamfer. The chamfer force will move the gauge in that direction

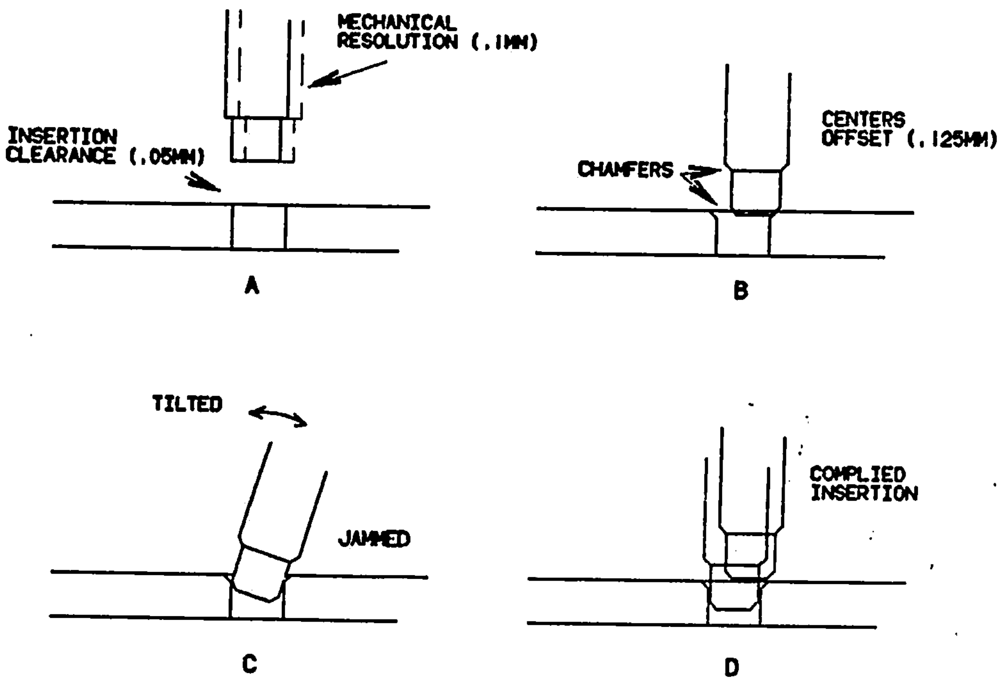

Figure 8. Gauge Insertion

(See Figure 8 (d)). If the compliant friction is small, then a minimal torque develops and tilting is prevented.

Another problem that occurs with peg insertion is when the hole is not located accurately on the workpart. Even if the insertion clearance is within the manipulator's repeatability (See Figure 9) , the hole's location is still not known. Here again , by chamfering the gauge and allowing

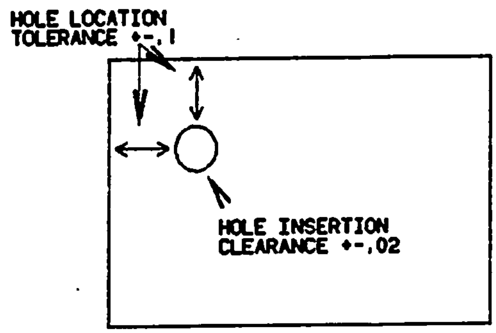

Figure 9. Hole Location

compliant motion, direct insertion is possible , eliminating the need for time consuming software searching and minimizing the possibility of a jammed insertion.

Chamfering of parts may not be possible or practical in a design. Instead, it may be possible to add a chamfered feature to the manipulator's fingers, which then hold the part directly. This technique is illustrated in Figure 10. It shows a high precision assembly operation-the insertion of twelve hammers into a pivot block for a printer hammer block component, one at a time.

A chamfered leader feature on the left finger enters a vacant hammer slot and aligns the fingers and hammer with

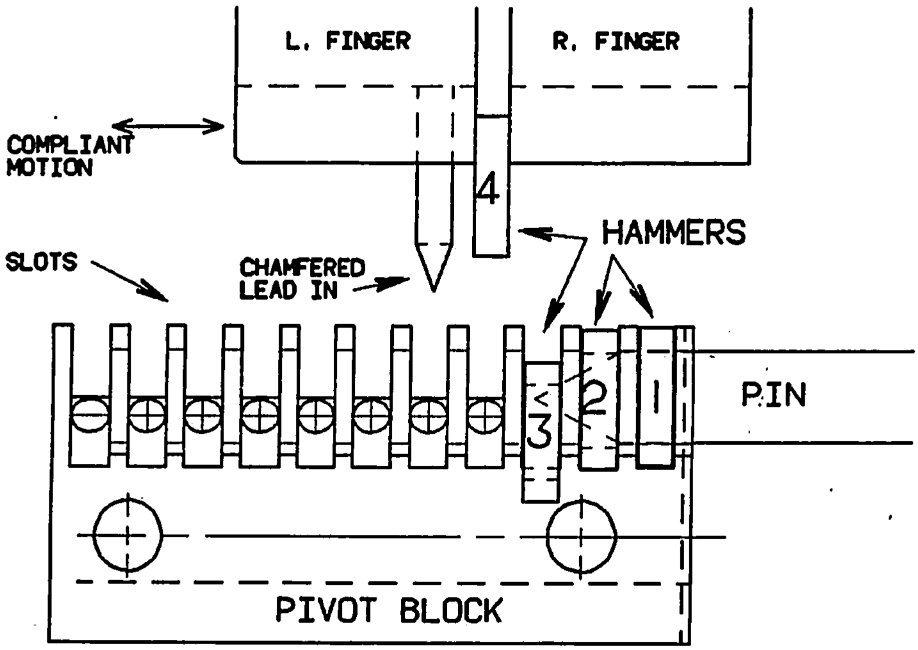

Figure 10. Hammer Insertion: Source: M. A. Ritland, "The Automated Assembly of a Precision Component." Case study performed at Lehigh University , Bethlehem, PA.

the pivot block. Compliance of either the hammer or the block is required. Presently a special fixture is used that allows the block to slide relative to the fingers. This is an effective method except that it requires the special fixture. An alternative would be to have the fingers comply to the pivot block. The compliant device allows the fingers to comply without any special fixturing.

The problem of inserting and threading bolts and screws into threaded holes is a common one. In the opinion of this writer the best solution to this problem would be to eliminate the threaded fastener altogether. However this is not always possible. To minimize the chance of cross threading, the bolt and hole axes should be parallel and located as close as possible to each other. This alignment becomes even more critical with very small bolts.

By using a powered screwdriver and a compliant device that prevents spin and tilt and takes advantage of the

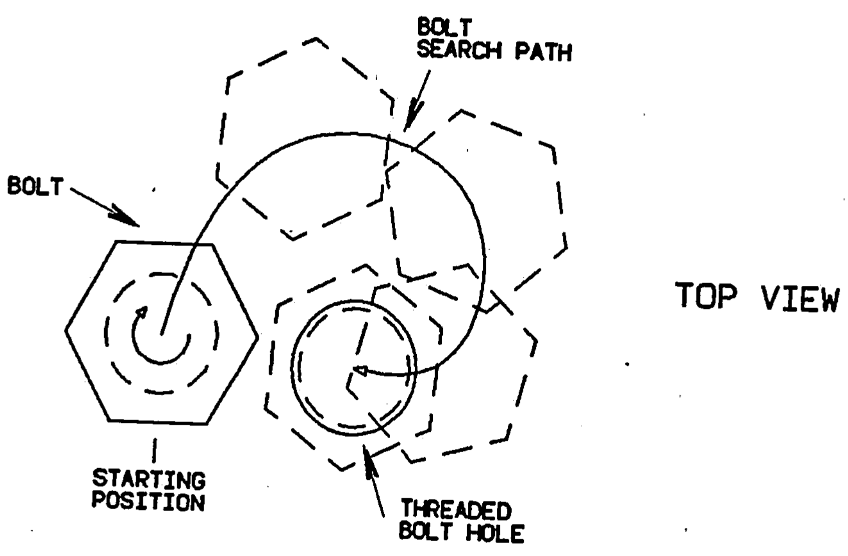

Figure 1.1. Bolt Hole Search

natural tendency of a rotating bolt to walk around a flat surface, successful bolt threading is possible (See Figure 11). The bolt will search for the hole automatically within the compliant range and then thread in without cross threading due to the confined straight approach governed by the device.

Motion in the compliant plane should be caused by very little force even when the insertion force is large. An example would be the assembly of circuit board connectors. These sometime require a high insertion force due to the number of individual connections in one board connector and the tight fit required for a good electrical connection. Despite the high insertion force, low resistive compliance is still desired so the delicate tapered connector ends will align with the mating connectors. Here again tilting is not desired, because the insertion force is directed straight down through the board to prevent buckling.

This low resistive compliance under high insertion force will also allow for use of less than ideal chamfers

(See Figure 12). As the chamfer angle decreases, the ratio of the insertion force to the horizontal force increases.[6] The chamfer will still align the parts because only a slight force is needed to move the end effector in the compliant plane.

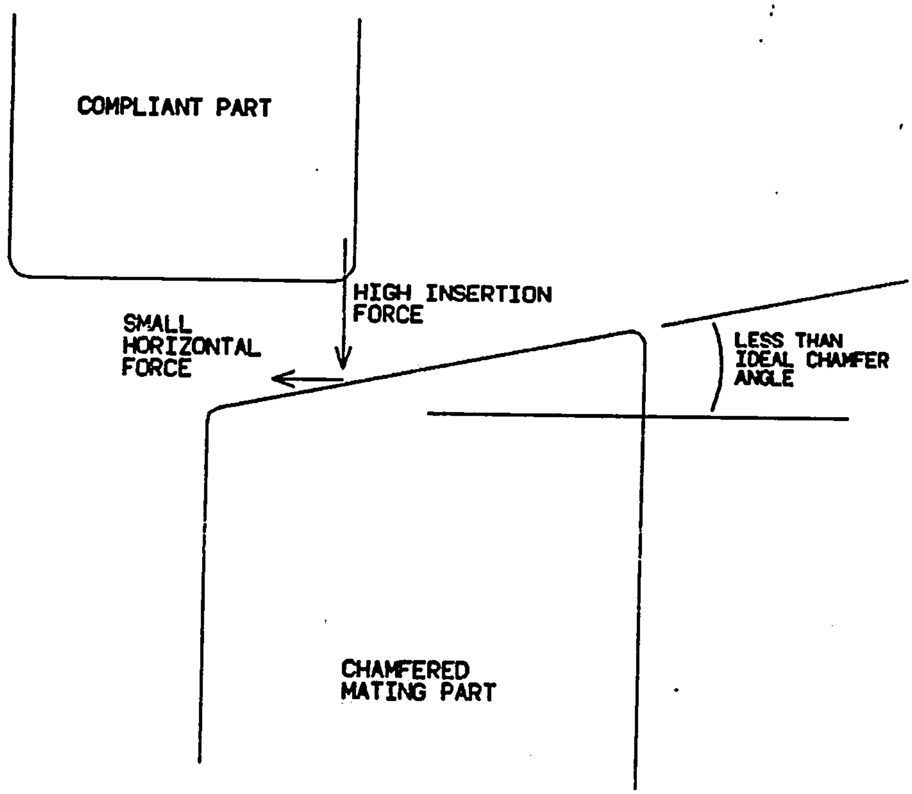

Figure 12. Not Optimal Chamfer

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6 D. Whitney et al. Robotic Research

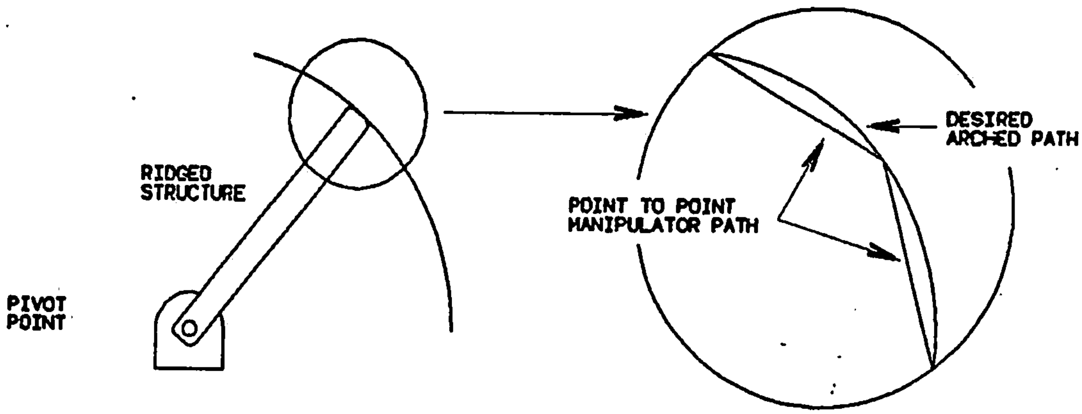


Figure 13. Curved Path Smoothing

Most assembly machines move in a point to point fashion. Continuous curves can be approximated at the expense of calculating many points. Even then the curved motion may be jerky and stepped. If the parts being assembled are rigid, then this erratic curve may cause problems. The use of a limited compliant device can smooth out the curved path and let the rigid parts create their own curve (See Figure 13). This assembling technique is further illustrated in Figure 14. It shows the assembly of a rigid support structure by hooking the tab features over a part and swinging the structure about the tabs , snapping it into

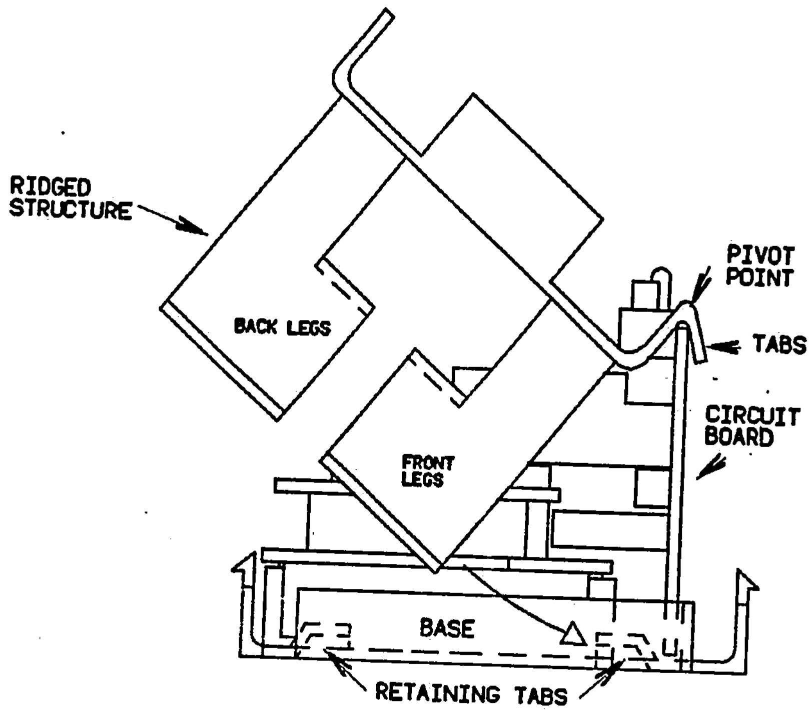


Figure 14. Rigid Structure Assembly

place on the base. This operation is accomplished with a minimum of intermediate points along the curve. The compliant motion allows the structure to move in a smooth arc even though the manipulator is moving along a stepped, point-to-point path. If compliant motion is not allowed, then the rigid structure is forced to follow the stepped path, which might stress the main circuit board during the assembly.

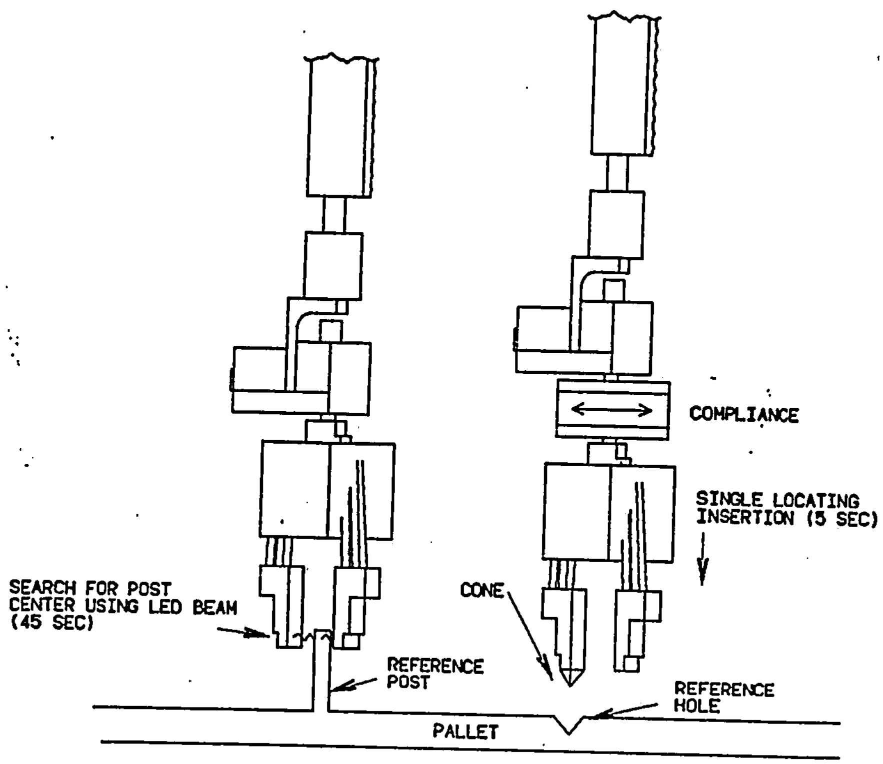
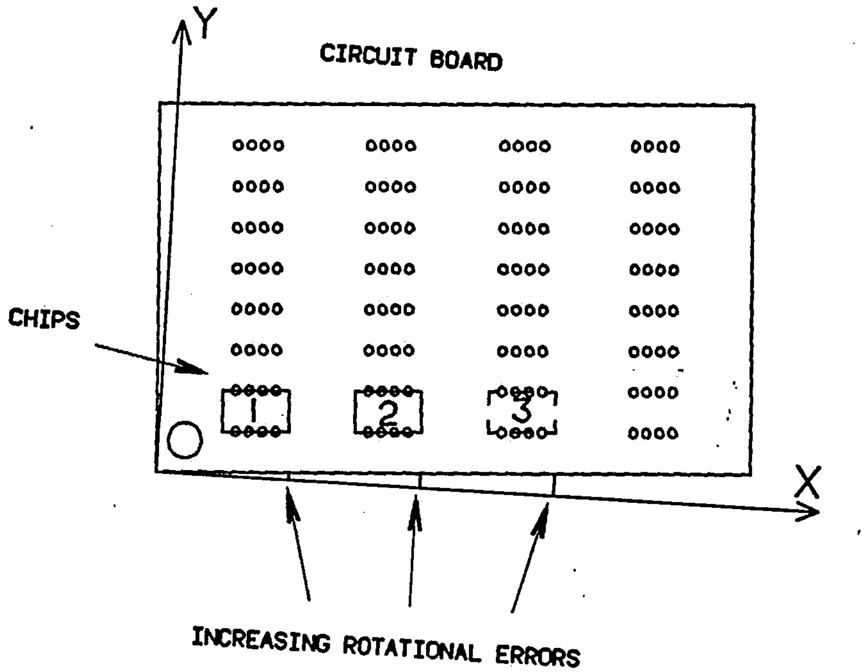


Figure 15. Pallet Calibration

The compliant mode allows design features and gravity to align the end effector to a particular frame of reference. This is essentially calibrating the end effector frame of reference to that of the workpart or pallet. This is necessary to adapt the manipulator to changes encountered in the assembly environment, such as a slightly displaced pallet. Calibrating is usually accomplished by searching for a reference post on the pallet or a feature on the part, using sensory feedback and a software search routine. This method requires extra time for finding the posts and for recalibrating production points throughout the assembly program. In addition the calibrating accuracy is limited by sensory capabilities, calculation errors and manipulator accuracy. Instead a coned hole feature on the pallet combined with a coned feature on the fingers can be used to directly align the manipulator with the pallet (See Figure 15). Then, locking the compliant device will preserve the calibrated frame of reference. This, obviously, only calibrates the local X,Y and Z coordinates. Rotational errors cannot be compensated for with a single offset. These errors increase as they get farther from the calibrating feature.

An example of this problem is shown in Figure 16. The first integrated chip will be inserted correctly after calibrating because rotational errors are minimal close to the calibration hole. The second chip is more affected by rotational errors but the chamfered pins on the chip will still guide them into their receptacle using the compliant mode. Locking the compliant device after this second insertion acts to recalibrate again. This process can be continued



Fiqure 16. Interqrated Chip Insertion

for each successive chip. Rotational errors are negated by this progressive calibrating technique.

Using the locked offset mode allows for calibrating of only one frame of reference. Then it prohibits the use of compliant mode if the original calibration is to be maintained. Many frames of reference may be desired in a single operation. For instance, a reference frame may be described for each of the feeders, the pallets, the work part, and the end effector. All of these frames are described by separate transformations. These transformations are found, as described before, by determining the location of reference points within the particular frame and then calculating the appropriate transform. Searching for the reference points in each desired frame is time-consuming. A faster method would be to use reference points to align the end effector directly. Then using the offset feedback, the complied offset can be obtained for each reference point. The component offsets are then used to calculate the appropriate transform. A substantial time savings is realized because only one move is required to locate each reference point using the offset feedback feature in the device, instead of using separate search routines for each of the three reference points.

DEVICE DESIGN

The need for, and specific requirements of, a compliant device have been described above. This section will now present design details of a prototype. The design of this device is an intergrated one, and considers many factors, such as functionality, fabrication, assembly, and model changes. An overall approach is taken that blends these considerations into a single design, compromising and adjusting where necessary. A totally integrated design requires such an iterative approach. A first step is the design of a feasible prototype-feasible in the sense that all parameters and constraints are considered but a detailed final mechanical design was not performed. If the prototype functions as planned, a more detailed analysis and precise design would be justified. For example, a finite element analysis of the structure subject to related forces would give more exact information that could modify the design and produce a lighter and stronger design. A further reason for a prototype is that the performance of the aerostatic bearings used in this device is difficult to predict accurately due to the turbulent flow in the air film. The standard method for air bearing design is to construct a prototype from analytical calculations and then adjust by trial and error until the design performance is obtained.[7]

The prototype is designed to work on the RS-1 robot. It is larger than ultimately necessary in order to accommodate design alterations and to facilitate the initial fabrication of the prototype. The design is structured such that particular functional characteristics can specified . This, however, is usually at the expense of another characteristic. For instance, if a very ridged, locked, and centered characteristic is required by the application, a larger locking spring would be used. This would require more room for the locking spring itself and for the retracting mechanism so the device's size and weight might be compromised in order to accommodate the altered components.

The basic component design is preceded by a general analysis of the field loading the device is expected to undergo during operation.

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[7] Interview with J. Hero, Dover Instument Co., Westboro, PA, April, 1984.

Field Loads: The field loads are established for performance at the maximum operating conditions defined for the RS-1 robot.[8] The forces exerted on the device during its three modes of operation are calculated and then exaggerated by a specified safety factor. These calculated field forces are then used to set the minimum requirements for specific device components.

In Mode 1 the main primary function of the device is to act as a solid coupling, aligning the end effecter axis with the robot's manipulator axis. Since the device is comprised of separate mechanical components, absolute rigidity and alignment is not possible. This problem-however, is not critical due to the compliant nature of the device. Rigidity is desired in the device, even when it is shut off or disconnected. This rigidity is affected by the locking spring. The force generated by the spring must be great enough to resist displacement when the device is functioning under maximum conditions.

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[8] IBM Corporation, A Manufacturing Language Concepts and User's Guide, (Boca Raton, Fla.: IBM Corporation, 1982).

Maximum conditions are assumed to occur when the RS-1 is decelerating with maximum payload from maximum velocity to zero velocity. Assume constant acceleration. The maximum field forces are calculated as follows:

Fx= max. field load force (N)

Mp= max. payload (kg)

Me= end effecter mass (kg)

V0= max. velocity (m/sec)

V1= min. velocity (m/sec)

Va=average velocity (m/sec)

D= deceleration distance (m)

A=acceleration (m/sec**2)

t= time to deceleration

Va = (V0-V1)/2 Va = (1-0)/2 = 0.5 m/sec

t = D/Va t = 0.25/0.5 = 0.5 sec

A = (V1-V0)/t A = (0-1)/0.5 = 2 m/sec**2

F = MA

Fx = (Me+Mp)*A Fx = (10+10)*2 = 40 kg m/sec**2

This force is consolidated and applied a certain distance down the ridged lever of the end effecter. This is assumed to be the center of mass of the payload and end

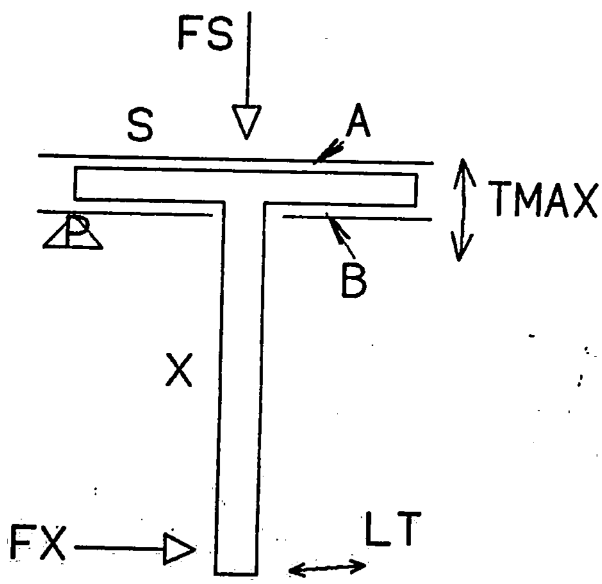

Figure 17. Field Forces

effecter. Figure 17 shows a schematic of the device structure and end effecter and the related forces and motions.

It is assumed that mass of the decelerating payload and end effecter will cause two types of motion. Tilting, or pivoting about the circumference of the lower disk (point P), and sliding in the compliant plane. Neither of these motions are desired. Tilting is limited by the clearances A and B. These clearances are required by the air bearing for the air film and are approximately (.0127mm). The maximum tilt in degrees about the pivot point P is:

Tmax = maximum tilt (degrees)

A = upper air film clearance (mm)

B = lower air film clearance (mm)

Dia = diameter of disk (mm)

Lt = linear tilt (mm)

L = length of lever arm (mm)

pi = 3.14

Tmax = Sin-1 (2*(A+B)/Dia)

= Sin-1 (2*.01875)/100 = .0214 degrees

This translates to a linear distance at the end of the fingers:

Lt = (Tmax*Sq((Dia/2)**2*L**2)*pi)/180

= 0.0214*SQ((100/2)**2*200**2) * pi/ 180 = .077mm

Although this amount of tilt is minimal, it is undesirable. The smooth air bearing surface may be damaged if tilting occurs. No tilt will occur if a great enough force is applied by the spring to the disk.

Fs= min. spring force (N)

S = applied distance of Fs from pivot (mm)

X = applied distance of Fx from pivot (mm)

Sum of torques = 0 (Static)

$$= Fx*X+Fs*S$$

$$Fs = Fx*X/S$$

$$= 40*.2/.05 = 160 \text{ N}$$

Sliding motion is limited positively by the clearance between the centering pin and the support guide hole because the centering pin acts as a shear pin between the bottom disk and the middle structure.

If air pressure is available, an additional force may be added to Fs. This additional force is generated by an air actuator and can be consolidated and applied with Fs.[9]

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9 See "Locking Air Actuator" on page 43 for details.

Ft = total force (N)

Fa = air actuator force (N)

Ft = Fs+Fa

Using air pressure, the device stiffens up by a factor of 10, which makes it essentially solid.

Mode 2, the compliant mode calls for a minimum resistance motion in the compliant plane, even under high insertion forces or heavy loading. This requirement suggest the use of externally pressurized air bearings which have low frictional resistance under heavy loading. The frictional resistance is calculated as follows.[10]

Fr = frictional resistance force (N)

H = film thickness (m)

Ab = area of bearing (m**2)

U = velocity of gas (m/sec)

m = gas viscosity (microP)

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10 J. W. Powell, Design of Aerostatic Bearings, (Brighton , England: The Machinery Publishing Co. Ltd. , 1970) , p. 63.

Fr = -m*Ab*U/H

= (181**(10-6))*0.0723*1.0/(18.7**(10-6)) = .7 N

Even under maximum axial loading the air bearing is virtually frictionless.

The forces acting on the device during the compliant mode are:

1. axial-due to the weight of the end effecter and payload, and insertion forces
2. compliant-due to any horizontal forces from the chamfer during insertion.

The maximum force on the lower air bearing is assumed to be twice that of the weight of the end effecter and payload. This is done to account for acceleration and deceleration in the vertical direction. The maximum force affecting the top air bearing is to be the same.

Fd = downward force (N)

A = acceleration due to gravity (m/sec**2)

Fd = 2*(Me+Mp)*A

= 2*(10+10)*9.8 = 392 (N)

The forces applied to the device during mode 3, locked offset, are the same as those during mode 1. However the spring and centering pin are not used in mode 3. The active force to resist tilting and sliding is due to the locking air actuator only (Fa). This force is many times greater than the force due to the spring so rigidity in this mode is assured as well.[11]

Estimates of forces on the device due to its operations have been approximated. The following sections will use these estimates to design the specific device components to meet or exceed the field requirements.

Spring Design: A minimum force of Fs must be provided by the spring. This analysis will verify that such a force is obtainable within the space requirements. The approximate constraining dimensions are as follows:

OD = outside diameter (13mm)

WL = min. working length (29mm)

LUF = length under force (35mm)

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11 See "Locking Air Actuator" on page 43 for details.

A closed end (squared) unground spring made of oil-tempered MB steel will be used. Such a spring has a safe stress under light duty of appproximately 700 MPa.[12] The feasible spring dimensions are:

d = wire diameter (2mm)

FL = free length (47mm)

F = deflection (12mm)

SH = solid height (must be less than WL)

This feasible spring is shown capable of generating the required force as follows.[13]

Fs = desired spring force (N)

S = safe stress (MPa)

Dm = mean diameter (mm)

G = modulus of elasticity (MPa)

N = number of active coils

TC = total number of coils

Si = spring index

K = curvature correction factor

S' = calculated actual stress (MPa)

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12 Harold Carlson, <u>Spring Designers Handbook</u>, (New York, NY: Marcel Dekker, Inc., 1978), p.146.

13 Ibid., p. 160.

The actual stress of the feasible spring is:

S' = Fs*Dia/.393*d**3

= 160*10/.393*2**3 = 508.9 MPa

This is below safe stress of 700MPa.

N = G*d*Fs/pi*S*Dm**2

= 77200*2*12/pi*509*100 = 11.5 coils

TC = N + 3 (ground,etc)

= 11.5+3 = 14.5

SH = 12 * 2 + 24mm (which is less than WL)

These calculations are now corrected for spring curvature:

Si = OD/d = 12/2 = 6

The correction factor with this spring index is:

K = 1.26[14]

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[14] Ibid., p. 163

The total corrected spring stress for force Fs is:

$$509*1.26 = 641 \text{ MPa}$$

This is still less than safe stress of 700MPa, so such a spring is feasible and will meet the requirements.

Spring Retraction Air Actuator: The air actuator is essentially an air bag in the shape of a torrid (See Figure 4 on page 11). The minimal hole diameter is set by the centering pin outer diameter, approximately 21mm. The maximum outside diameter is limited by the clearance needed for compliant motion by the four support posts, which from Figure 18 is:

ID = inside diameter (centering pin OD)

OD = outside diameter

ODa =2*((disk)-(post inset)-(post diameter) -(offset))

= 2*(50-3-8-3) = 72mm

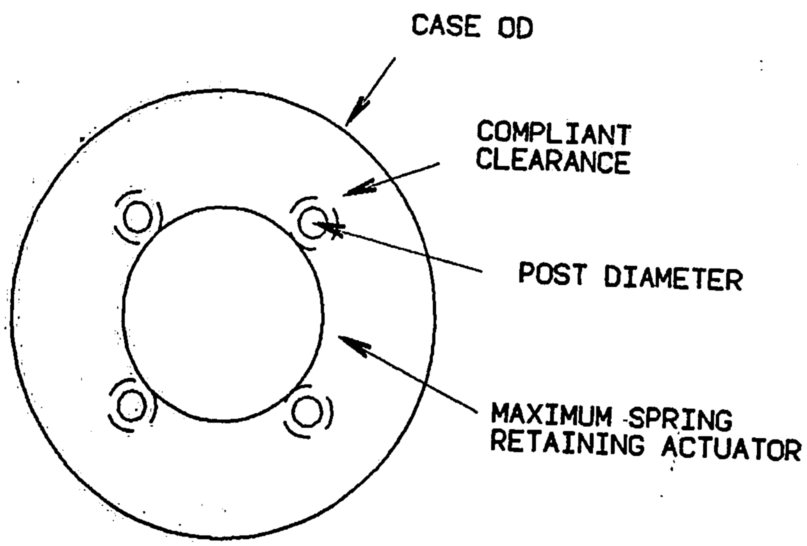

Figure 18. Compliant Clearance

The spring-retracting air actuator must develop a force greater than Fs over a distance of 6mm. The 6mm stroke assures, using a 45-degree chamfer, that the device can be centered from offset.[15] The maximum force developed by the actuator is found.

Fa = air actuator force (N)

Aa = area of actuator (mm**2)

Pa = air pressure (MPa)

Fa = Aa * Pa

---

[15] See "Centering" on page 44 for details.

Aa = (ODa/2)**2*pie-(IDa/2)**2*pie

= (36 2pie-(21/2)2*pie) = 3700 mm**2

Pa = 60(note)psi*.006895MPa = .437 MPa

Fa = 3700mm2 * .437MPa = 1530 N

This generated force is much greater than required. Air actuators of this type are available from industrial vendors.[16]

Locking Air Actuator: This is of similar design as the spring retracting actuator except the stroke required is less than 1mm. The same force can be developed which exceeds the locking requirement. If we assume the air actuator generates a 750 N force (half that possible), the static friction force is:

Fs = static friction force (N)

Mu = coefficient of friction(.8 for steel on steel)[17]

Fn = normal force (750N from air actuator)

Fs = Mu*Fn

= .8*750 = 600 N

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16 Interview with Robert Blackert, Aero Tech Laboratories, Ramsey, NJ, June 1984.

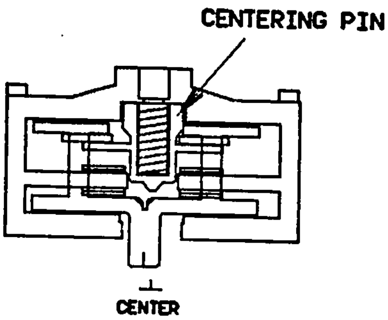
17 Robert Parr, Principles of Mechanical Design, (McGraw-Hill Book Company, 1970).

This force far exceeds the 40N field force causing disk sliding.

Centering: Centering the device is accomplished while the air bearings are activated. This assures that the load bearing disks will move easily from offset to center.
The spring retaining air actuator is slowly deflated allowing the centering pin to extend (refer to Figure 19). The chamfered surface (Figure 19 (a)) on the pin comes in contact with the upper disk. The pin extension continues and forces the upper disk 3mm, at the most, towards the center (Figure 19 (b)). At this point the tapered end of the pin contacts the hole in the lower disk. The pin extension continues and the device centers the remaining 3mm. It then forces the disk against the lower air film (Figure 19(c)). When the air pressure to the bearings is turned off the air film dissipates and the two bearing surfaces are pressed together by the force from the spring. The centering accuracy is dependent only on the clearance between the pin circumference and the support guide hole.

Anti Spin Coupling: Rotation about the vertical axis is prevented by an X Y coupling shown in

A



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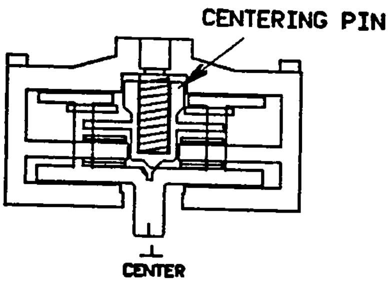

C

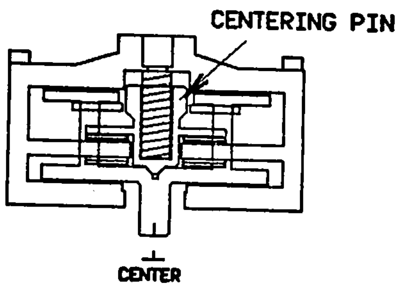

Figure 19. Centering

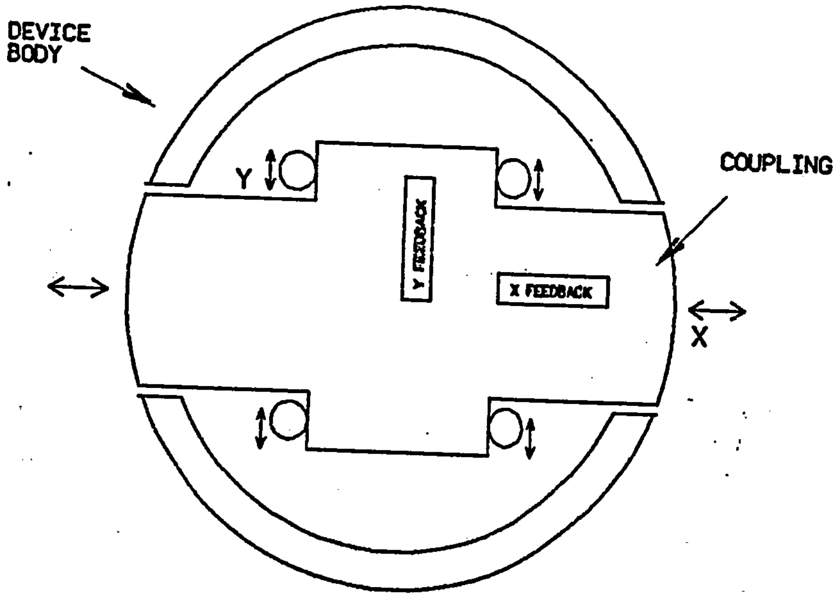

Figure 20. X Y Compliant Coupling

Figure 20. The posts are limited to motion along the coupling's Y axis while the coupling is limited to move along the manipulator's X axis. The combination of X and Y motion allows total freedom within the compliant range for the two disks and the attached end effecter. The bearing is under light field loads and its frictional resistance is consid-

ered negligible.[18] The wide stance of the coupling minimizes rotational effects due to sliding clearance provided in the coupling. It also reduces the bearing pressure about the vertical axis.

Air bearings: Two externally pressurized aerostatic thrust bearings loaded against each other are used to provide low frictional resistance under high loads. The lower bearing supports the load and the upper bearing resists high axial insertion forces. In addition, the maximum axial stiffness is doubled and the load-carrying capability is increased by 25% for each bearing by loading against each other.[19]

Both bearings are ring bearings, to allow the centering pin and end effecter shaft to pass through the center. They are inherently compensating with two narrow grooved annulars

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18 F. T. Barwell, Bearing Systems, Principles and Practices (East Kilbride, Great Britain: Oxford University Press, 1979).

19 Powell, Design of Aerostatic Bearings, p. 106.

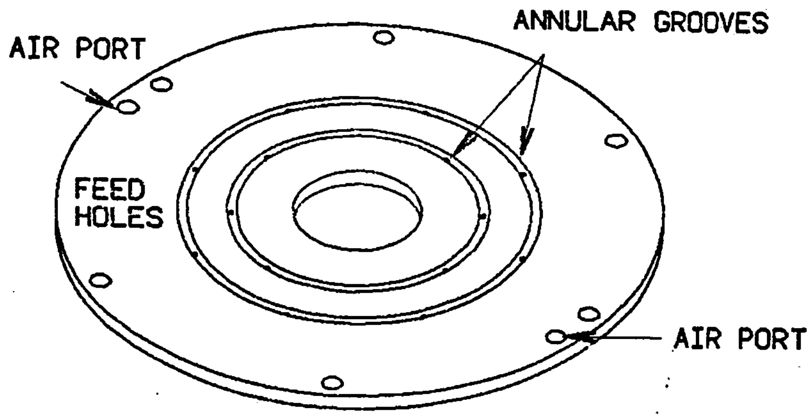

Figure 21. Double Grooved Annular Air Bearing Surface

and multiple feed holes (See Figure 21). This type of bearing is the most resistant to tilting.[20]

The field loads require each bearing to carry up to 392 N axial force and maintain axial stiffness of 55250 N/m. The load bearing capability for this bearing is determined.

a = bearing inside diameter (25mm)

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20 E. Dudgeon, " A Simplified Method for the Design of Externally Pressurized Gas Lubricated Thrust Bearings, " Transactions of the Canadian Society of Mechanical Engineering , August, 1973, p.207.

b = bearing outside diameter (100mm)

Pa = atmospheric pressure (.1MPa)

Po = supply pressure (.689MPa)

h = film thickness = .0015 (mm)

Kg = max. axial stiffness (.69)

A maximum load of 1332.6 N is found to be obtainable.[21] This is well above the field load requirements.

The next step is to determine the feed hole characteristics. Tilt stiffness increases as the number of feed holes increases. This is traded off against the difficulty in drilling very small holes. A certain mass flow is required for the bearing to function properly. This can be calculated for a simple single feed hole using the ratio of a to b (100mm/25mm=4). A single .2mm diameter hole is found.[22] If eight feed holes per groove are choosen then the new diameter for the sixteen feed holes is:

d = multi-hole diameter (mm)

dd = single feed hole diameter (mm)

n = number of holes

---

21 Powell, Design of Aerostatic Bearings, p. 103.

22 Ibid., p. 108.

d = dd*2/Sq(n)
  = .2*2/4 = .1mm

The optimal placement for the multiple holes is on the circumference of a circle with a radius:

c = optimal radius (mm)

c = Sq(a*b)
  = Sq(12.5*50) = 25mm

Using two annular grooves on the bearing increases resistance to tilt. The two annular feedhole rings are placed on either side of the optimal radius.

The air is supplied to the bearing surfaces through a single inlet into the body of the device and distributed top and bottom through ports and grooves. The air flow through the device is shown in Figure 22. The air exhausts out where the coupling is guided in the body.

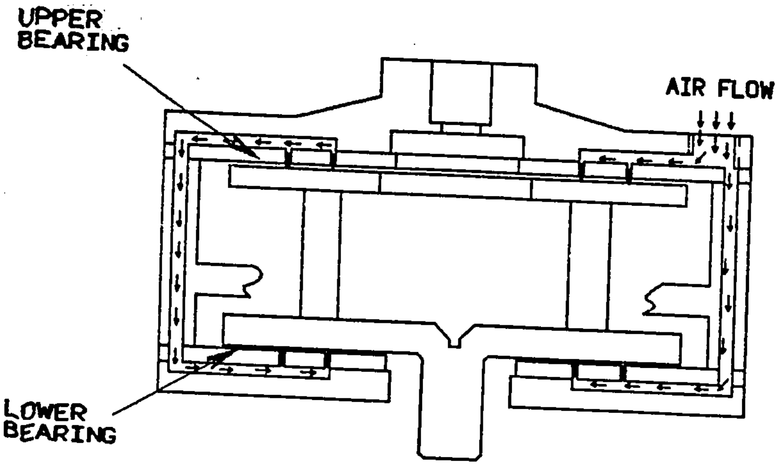

Figure 22. Air Flow

FABRICATION

Presented in Figure 23 is an expanded parts list for this device. The list indicates whether parts are fabricated or purchased. All of the fabricated parts can be made on standard CNC machines and the purchased parts are readily available.

The parts that require orientation about the center axis during handling and assembly have a notch machined into their outer circumferences (See Figure 4 on page 11). This additional feature allows for a single end effecter to handle all of the component parts. The single end effecter shown in Figure 24 has a spring loaded ball pin that will engage the notch and define the part's orientation. The part's rough orientation is maintained by removing it from its orientated position in the machining workspace to a set position on a pallet. When the part is retrieved from the pallet for assembly or further machining, its general position and orientation is known. The end effecter automatically locates the part accurately by slow parallel closing

Part	Oriented	Process
top	yes	turned
bearing plate (2)	yes	turned, milled, ground
top mid section	yes	turned, milled, ground
coupling	yes	milled
spring	no	purchased
posts (4)	no	turned
centering pin	no	turned
air actuator (2)	yes	purchased
middle section	yes	turned, milled, ground
bottom mid section	yes	turned, milled, ground
bottom	yes	turned, milled
bolts, body (6)	no	purchased
bolts, posts (4)	no	purchased

Figure 23. Expanded Parts List

and turning about the vertical axis. The "vee" features locate the axes of the circular part and as the ball rolls along the circumference it engages the notch and defines the rotational orientation. The combined design of the single end effecter with the notch feature in the part allows the orientation and location to be defined without a complicated search or extensive fixturing.

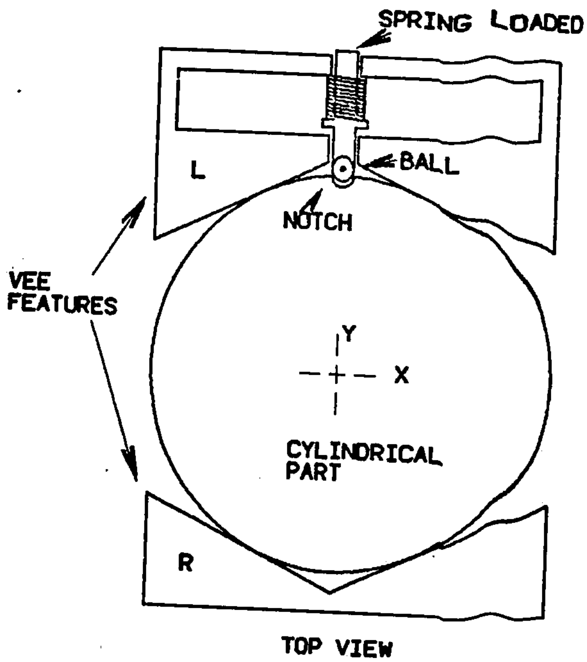

Figure 24. Single End Effecter

ASSEMBLY

The compliant device was designed to facilitate assembly. All the components are assembled from one side, top down. A four degree of freedom robot, an air screwdriver, and a single end effecter is all that are required for assembly. Features have been included during fabrication to assist the assembly operation, (such as chamfers to guide insertion and orientation notches to define part location accurately.)

The middle section with the two air actuators would be supplied to the assembly area as a subassembly. The air actuators are epoxied to the middle section and require time to dry, so this operation is performed prior to the main assembly.

The same end effecter is used for both fabrication and assembly. Two special fixtures are used to assist the assembly and are shown in Figure 25. The tapered pins are spring loaded and act as guides during assembly, aligning

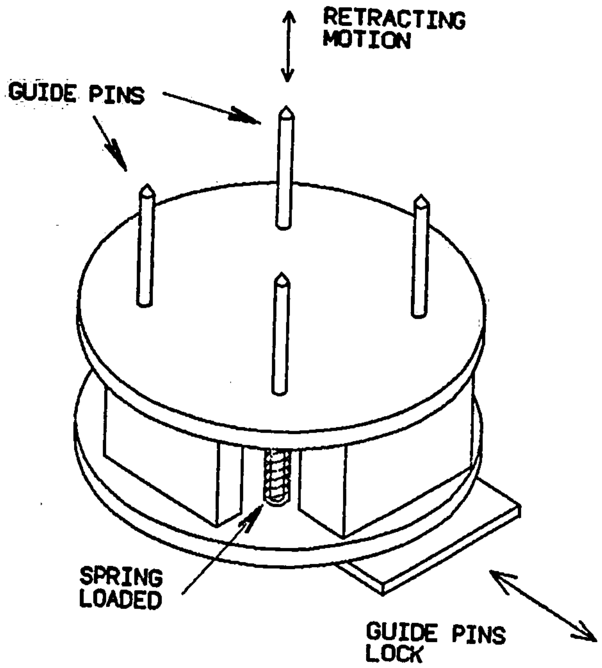

Figure 25. Assembly Fixtures

and restraining the stacked components. Finally when the bolts are assembled the guide pin lock is released and the bolts push the guide pins down.

SUMMARY

In summary, this thesis presented a feasible design of a device that would give an overhead gantry, four degree of freedom robot the ability of selective compliance. Several applications examples were presented that demonstrated the benefits of a compliant capability during automated assembly.

A particular range of compliance was suggested as being the most useful. The device design suggested achieves this range of compliance under a certain control scheme. Approximate calculations were performed to establish the feasibility of the prototype for the RS-1. It uses two aerostatic bearings loaded against each other that provide a stiff compliant motion. Two air actuators act as selective locks and a spring loaded centering pin serves to center the device.

The design was carried out such that the fabrication and assembly of the device were considered as well as it is functionality. The integrated approach to the design is demonstrated to be feasible,requiring only a four degree of freedom robot, assembling from one direction. In addition all of the components during fabrication and assembly can be handled by a single end effecter.

CONCLUSION

In conclusion the feasibility of applications for a compliant assembly device was demonstrated for a typical robotic device The device was shown to give the RS-1 the capability of selectively controlled compliance in a plane parallel to the horizontal working plane. The device would function in one of three modes; locked and centered, compliant, and locked offset.

The application examples presented why and how such a device could be used to increase automated assembly opportunities and efficiencies. They demonstrated that the type and range of compliance provided by the device is beneficial to an efficient automated assembly system.

The integrated design approach taken was shown to result in a device that not only functioned as required but also lent itself to automated fabrication and assembly. The integrated design allowed a family of devices with varied characteristics to be constructed from a base model and then manufactured and assembled in a flexible manufacturing environment.

FUTURE WORK

The integrated design discussed in this thesis has only been shown to be feasible. Due to the unpredictable behavior of the aerostatic bearings, a prototype must be constructed and tested. The design should then be refined and retested.

The design suggested allows a limited type and range of compliance. The ideal compliant wrist would not be restricted to motion in a parallel plane, but would mimic the total compliant range possessed by humans. Much more research and development is required in this area.

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VITA

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