MODULAR SEMICONDUCTOR TEST, ASSEMBLY & PACKAGING MANUFACTURING EQUIPMENT DESIGN

by

Luis Antonio Muller

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Abstract

Semiconductor equipment manufactures need to be innovative in their designs to follow rapid changes in the semiconductor business. This thesis introduces a new concept for semiconductor Test, Assembly and Packaging (TAP) floor layout through the development of a modular machine design strategy. Careful economical analysis of TAP manufacturing line, through cost-of-ownership concepts, highlights the need for innovation in the design of semiconductor equipment. A modular machine design strategy is proposed with the main objective to provide functional interchangeability features to semiconductor equipment, thus creating the ability to change the functionality of an equipment by replacing modular units. The ability to change equipment functionality defines a new paradigm in semiconductor equipment design. Equipment design can be modularly built to improve upon semiconductor manufacturer capability to follow fast moving market changes.

Another section of this thesis presents two enabling technologies for modular machine design. KinFlex™ is a flexural kinematic coupling that enables precision alignment to occur, and FlexTray™ is a six degree of freedom alignment fixture.

The tangible result of this thesis is a production semiconductor equipment, the Apollo Sorter, which is being used by a major memory manufacturer in the United States and is also being evaluated by several other semiconductor manufacturers for implementation in their production lines.

Thesis Supervisor: Prof. Alexander H. Slocum
Title: Alex & Britt D'Arbeloff Professor

Acknowledgments

My graduate studies have been an enjoyable period of my life because of support and input of many people. I have made many friends at MIT and learned much from them. The classes at MIT were fundamental to my personal development and I am thankful to the professors I had.

Prof. Alex Slocum has been a supportive advisor throughout this thesis. Alex helped me achieve my goals when I came to MIT, to do significant research for industry as well as academia. Alex helped develop me as an engineer and to transform ideas into reality. Alex, with his always enthusiastic attitude forced me forward to explore new fields of research and always keeping a realistic perspective of how and where to apply the results.

The Brazilian government through CNPq (Conselho Nacional de Pesquisa) sponsored my classes and research at MIT. To CNPq I am thankful for the four years I stayed at MIT with the freedom to develop my research and to give my best to it.

Teradyne, a Boston based company supported the development of the material objectives of my research, providing me with necessary equipment and support to develop an equipment for the semiconductor industry that proved the concepts I proposed in this research.

I thank my wife Cristina for the patience. My son Lucas, who started his first two years of life with minimum support from his father. To Lucas I owe dedication and love to recover the time I did not spend with him. My family is my soul.

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Desiderata

Go placidly among the noise and haste and remember what peace there may be in silence. As far as possible without surrender be on good terms with all persons. Speak your truth quietly and clearly; and listen to others, even the dull and ignorant; they too have their story. Avoid loud and aggressive persons; they are vexations to the spirit. If you compare yourself with others, you may come vain and bitter; for always there will be greater and lesser persons than yourself. Enjoy your achievements as well as your plans. Keep interested in your own career, however humble, it is a real possession the changing fortune of time. Exercise caution in your business affairs; for virtue there is; many persons strive for high ideals; and everywhere life is full of heroism. Be yourself. Especially, do not feign affection. Neither be cynical about love; for in the face of all aridity and disenchantment it is perennial as the grass. Take kindly the counsel of the years, gracefully surrendering the things of youth. Nurture strength of spirit to shield you in sudden misfortune. But do not distress yourself with imaginings. Many fears are born of fatigue and loneliness. Beyond a wholesome discipline, be gentle with yourself. You are a child of the universe, no less than trees and the stars; you have a right to be there. And whether or not it is clear to you, no doubt the universe is unfolding as it should. Therefore be at peace with God, whatever you conceive Him to be, and whatever your labors and aspirations, in the noisy confusion of life, keep peace with your soul. With all its sham, drudgery and broken dreams, it is still a beautiful world. Be careful. Strive to be happy.

Unknown Author
St. Paul's Cathedral - Baltimore - USA

Chapter 1

INTRODUCTION AND THESIS OVERVIEW

1.1 Introduction

Since the development of the first integrated circuit by Jack Kilby of Texas Instruments and Robert Noyce of Fairchild Camera in 1959, the semiconductor industry has shown rapid development. The $101.8 billion world-wide semiconductor industry will exceed the $200 billion barrier in 1999, according to the Semiconductor Industry Association. A forecast prepared by World Semiconductor Trade Statistics reports that the North American market concluded 1995 at $41.2 billion, a 35.1 percent increase over 1994, and will be valued at $69.6 billion by 1998.

The semiconductor equipment manufacturing industry is responsible for producing electromechanical systems for handling, processing and testing semiconductors, circuit boards, and other electronic components. The back-end of semiconductor manufacturing is responsible for testing, trimming, lead forming, laser marking and a few other operations on the already packaged integrated circuit (IC). The equipment responsible for transferring and processing ICs at these back-end manufacturing stages are increasingly being automated. However, often this automation is in the form of stitching different machines together, as opposed to carefully coordinating design to optimize machines for an automation floor layout.

Fundamental challenges faced by these back-end process mechanical equipment are: machine functionality, change-over flexibility, ability to scale the design for different applications, ability to easily implement customer specific functions, high throughput, small footprint, machine and handling reliability (issues related to handling jams, machine scheduled and unscheduled maintenance). The importance of these parameters are better understood with an analytical process that relates the dollar value with the non-productive time of the system. SEMATECH proposes a Cost-of-Ownership model (CoO) that evaluates the dollars per unit time of a processing system according to manufacturers' specific data.

Accordingly, this thesis presents a framework of modular machine design and component technology to improve performance, flexibility and reliability of back-end automated equipment; a set of new patented technologies that push forward the state-of-the-art of IC handling; and the design, prototyping and testing of an IC Sorter that together with an in-tray test handler introduces a new concept for test floor layout.

1.2 The Semiconductor Industry

The initial resistance to the new IC technology gave way to enormous popularity. By the end of the 1960s, nearly 90% of all the components manufactured by the semiconductor industry were integrated circuits.

1.2.1 Historical Perspective

Semiconductor materials were studied in laboratories as early as 1830. The first materials studied where a group of elements and compounds that were usually poor conductors if heated. Shining light on some of them would generate an electrical current that could pass through them in one direction only.

By 1874, electricity was being used not only to carry power, but to carry information. The telegraph, telephone, and later the radio were the earliest devices in an industry that would eventually be called *electronics*.

In December 1947, John Bardeen, William Shockley, and Walter Brattain of AT&T Bell Telephone Laboratory developed the transfer resistor which amplified electric current using a soil semiconductor material. This made it possible to selectively regulate the flow of electricity through silicon, thus giving name to Semiconductors.

Until 1958, all electronic components were discrete: that is, they performed only one function, and many of them had to be wired together to create a functional circuit. Although a great number of identical discrete transistors could be fabricated on a single wafer, they then had to be cut up and individually packaged in tiny cans.

In 1959, Jean Hoerni and Robert Noyce developed a new process called planar technology at Fairchild Semiconductor which enabled them to diffuse various layers onto the surface of a silicon wafer to make a transistor, leaving a layer of protective oxide on the junctions. This process allowed metal interconnections to be evaporated onto the flat transistor surface and replaced the hand wiring. The new process used silicon instead of germanium, and made commercial production of ICs possible.

1.2.2 The Semiconductor Manufacturing Equipment Industry

During the infancy of semiconductor manufacturing, prior to the production and marketing of ICs, the companies producing semiconductors generally developed the equipment required to manufacture and test these new electronic components. The production of ICs, however, changed this pattern.

The development of a separate Semiconductor Equipment Manufacturing Industry is a result of the extreme complexity and diversity of the manufacturing processes involved in the production of semiconductors. These are customarily divided into "front end" and "back end" processes steps [U.S. Department of Commerce '85].

The front-end group involves several exacting steps. First, a light-sensitive chemical coating known as "resist" is laid down on the surface of a silicon or non-silicon wafer. The microscopic image of a pattern is then projected onto the resist; where it is struck by light, the resist becomes much more susceptible to removal by means of a chemical reaction . The image is then etched into the resist by means of wet chemicals or chemically reactive gases. The resulting pattern can be processed in many different ways, depending on the layer in question. The underlying silicon may be doped with impurities to create semiconductor regions; metal may be deposited to create circuit interconnections, the silicon may be oxidized to create insulation.

The back-end consists of testing and assembly of the completed semiconductor device. First, the electrical properties of the individual chips on the wafer are examined. The wafer is then sliced and the chips that passed the electrical test are assembled into stable, convenient to use packages. Finally, the finished product is subjected to heat and humidity tests to assure its continued reliability over a wide range of environmental conditions.

A typical semiconductor manufacturing line is represented in Figure 1.1 [U.S. Department of Commerce '85].

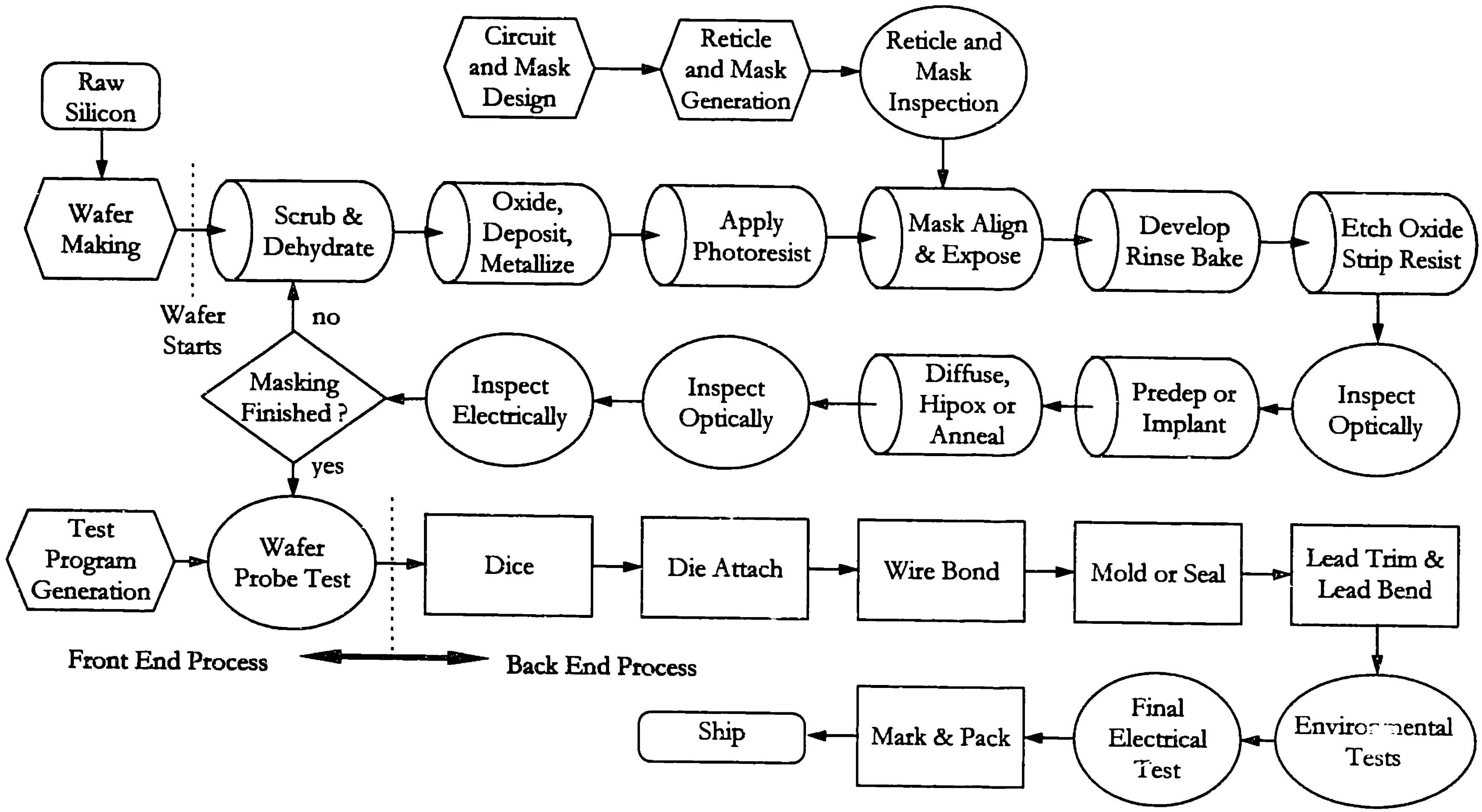


Figure 1. 1 - Typical Semiconductor Manufacturing Line

Semiconductor device technologies are directly dependent on the equipment used in their production. The present high degree of miniaturization and circuit integration became possible only with the development of machinery capable of meeting increasingly demand requirements. Similar considerations will determine the pace and degree of change in semiconductor device design in the future.

1.3 The Back-End of the Semiconductor Manufacturing

The back-end of the semiconductor industry is responsible for all the processes required to transform a silicon wafer into tested ICs. Initially the die (cut from the silicon wafer) goes through wire bonding the leads and molding into a plastic enclosure. Several processes than take place: ambient testing, lead trimming, lead forming, burn-in testing, laser marking, tri-temp testing (at temperatures between -60^{0}C and +160^{0}C), lead inspection (coplanarity).

This thesis covers in more detail a subsection of the back-end production line, named Test, Assembly & Packaging (TAP). Figure 1.2 illustrates the existing arrangement for testing with a duo-headed tester; two manipulators/test heads; two handlers (existing configuration allows for sorting).

The process flow on this test floor layout is reasonably simple. An operator manually loads a stack of trays on each handler. The handler automatically picks-up the IC from the tray, cycles it through a thermal chamber, and presents it to the test head; testing takes place and the IC is sorted into one of the output bins according to the test result. Whenever an output bin is full the handler stops the process and a light tower alerts the operator that the machine requires human assist to continue testing and sorting. The elapsed time that the machine is inoperable significantly affects the productivity of the test cell.

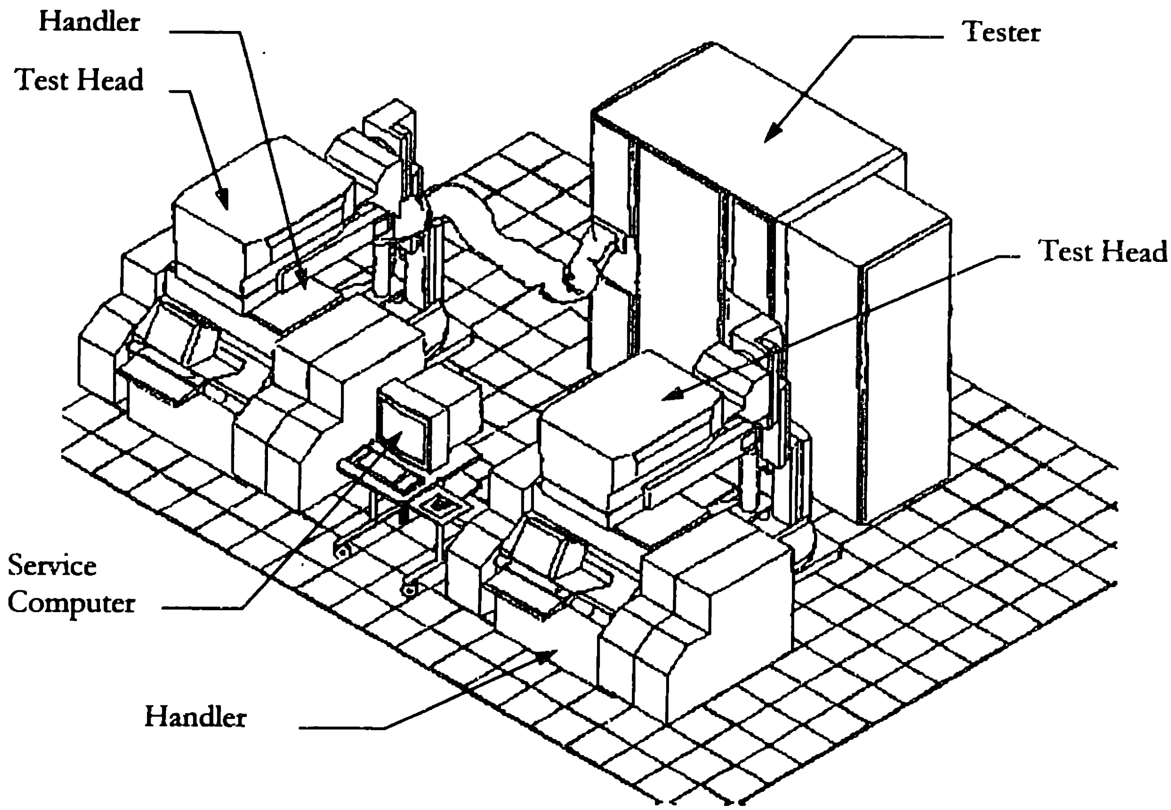


Figure 1. 2 - Testing at the back-end

The reliability of the system is dependent on the individual reliability of the tester plus test head, and the reliability of the handler. If the handler requires service, half of the test cell would be inoperable for the time required to maintain the handler or to replace it.

The industry trend is not to fully automate the back-end processes but to reduce its cost of operation (especially true for testing) by achieving an adequate level of equipment integration. An adequate level of integration is a difficult to measure parameter but with some economical back ground it is possible to understand what factors play a major influence in the cost of operation of a TAP facility.

1.4 TAP Cost-of-Ownership Analysis

SEMATECH has developed a standard calculation method for estimating the total life-cycle cost of owning a system used to perform a specific semiconductor process step, Figure 1.3. This modeling tool can be used to support the efforts of equipment engineers, manufacturing engineers and capital equipment buyers [SEMATECH '92], [SEMATECH '93].

COST OF OWNERSHIP CALCULATION			
# of Systems	251	Total Cost per Year	98,651,005
		Actual Equipment Util.	95%
INPUTS			
Volume and Throughput Data			**Metric**
Time Not Available (PER SYSTEM)	Scheduled Maintenance	1	hrs/wk
	Engineering Usage	1	hrs/wk
	Standby/Waiting/Warmup-dwn	2	hrs/wk
	Production Tests Scheduled		#/wk
	MTTT		hrs
	MTBF	1000	hrs (R)
	Avg Response Time to Failure	0.08	hrs
	MTTR	0.33	hrs
	MTBA	10	hrs (R)
	MTTA	15	minutes
Throughput @ Capacity per System		750	devices/hr (R)
Production Requirements		30,000,000	device starts/wk (R)
Lot Size		4000	devices/lot (R)
Redo Rate			% as decimal
Throughput Yield		99.00%	% as decimal (R)
# of Systems 1 Operator can run		5	number (R)
Equipment Data			**Metric**
Original Cost per System (all are deprcble)	Equipment Cost	1.10E+06	dollars
	Transportation Cost		dollars
	Installation Cost		dollars

Figure 1. 3 - A Cost-of-Ownership Spreadsheet

Historically, purchase decisions have been based on initial purchase and installation costs. However, purchase costs do not consider the effect of equipment reliability, utilization, and yield. Over the life of the system, these factors may have a greater impact on cost of ownership than initial purchase costs.

1.4.1 Test and Assembly Models

Equipment design engineers should thoroughly understand the sensitivity of the different input parameters to this CoO model. There is already a lot of effort being done on non-relevant design functions of semiconductor equipment. The economical analyses clearly point the direction for which design engineers should concentrate effort; one should never go blind about a design or misguided by individuals that lack the analytical power to describe well behaved economical systems.

The basic cost-of-ownership algorithm is described by the following equation:

$$C_{IC} = \frac{C_F + C_V + C_Y}{TPT \cdot Y_{TPT} \cdot U} \tag{1.1}$$

where:

C_{IC} = Cost per IC
C_F = Fixed cost
C_V = Variable cost
C_Y = Cost from yield loss
TPT = Throughput
Y_{TPT} = Mechanical throughput yield
U = Utilization

Fixed costs include costs for such things as equipment purchase, installation, and facilities support that are normally amortized over the life of the equipment. Variable costs (such as material, labor, repair, utility, and overhead expenses) are incurred during equipment operation.

The cost of discarding a good device and the cost of shipping a bad device must also be considered at test. Minimizing the total cost of shipping bad devices is one purpose of testing. However, if either the sampling plan of the test methods are ineffective, bad devices will be shipped. However, if the test specifications are too restrictive, then good devices may be rejected. These costs also must be included in the test CoO.

The proposed test systems involve both the tester and one or more handlers, also integrated with different process machines. Thus, the test system is similar to a work cell. The MTBFp of the test system is the combined MTBFp of the tester and the handlers, and the MTTR of the test system is the weighted average of the tester and handler MTTRs.

Another characteristic of testing is the relationship between test yield and throughput rate. Testing a good circuit may take much longer than failing a bad circuit since production test methods are frequently designed to stop after the first failure and are organized to test the most common failure modes first. Test throughput rate may be estimated by:

$$TPT_t = \frac{n}{Max\left[T_p \cdot Y + T_f \cdot (1 - Y) + T_h, T_{sort}\right] + T_i} \qquad (1.2)$$

where:

Y = Test yield

n = Number of devices tested in parallel

T_{sort} = Average handler sorting time (singulation)

T_p = Average test pass time

T_f = Average test fail time

T_h = Handler index time

T_t = Tester & handler overhead time

This equation assumes a single headed tester and a single handler. A more common configuration consists of a dual-headed tester and two handlers. In this configuration, a device is tested on one head while the handler on the other head is indexing to another device. Test throughput rate for this configuration may be estimated by:

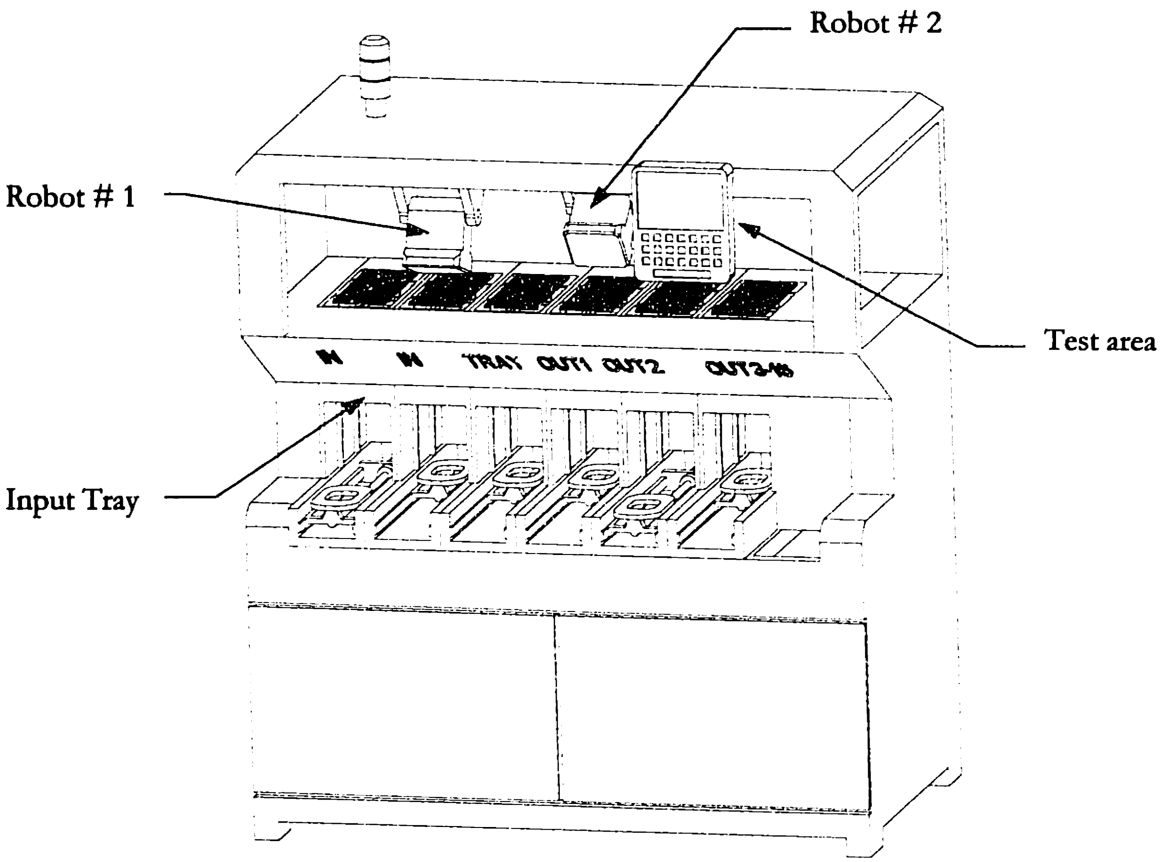
$$TPT_t = \frac{2 * n}{Max\left[T_{h2} + (T_{sort2} - T_{test2}), T_{test1}\right] + Max\left[T_{h1} + (T_{sort1} - T_{test1}), T_{test2}\right] + T_t} \qquad (1.3)$$

where,

$$T_{test} = T_p \cdot Y + T_f \cdot (1 - Y)$$

More complex arrangements would require different equations to estimate test TPT. As an example, a comparison is presented between three different handler manufacturers and the design proposed on Figure 1.4 (TwinHandler).

This proposed Handler design (TwinHandler - Kinetrix, Inc.), has two robots parallel tasking in a work envelope, one robot presents devices to test while the second robot sorts devices already tested. This handler was rejected still on concept phase due to thermal impracticalities and also due to the fact that two functions (testing and sorting) would still be integrated in the same machine, even though their throughput dependencies are different. The concept, though, serves the purpose of demonstrating the economics of testing. The handler concept is shown on Figure 1.4.



* U.S.& International Patent Pending

Figure 1. 4 - TwinHandler Concept

Analyzing the TwinHandler throughput versus existing handlers on the market shows the necessity of isolating the factors that influence testing and sorting, and by consequence separating the two functions into independent machines. Having both factors influencing the throughput of the same machine limits the overall throughput of the system at realistic test times (above 60 seconds for memory devices - the case being analyzed). In this scenario, the TwinHandler would become economically viable (higher throughput) over the other manufacturers' handlers for test times bellow 10 seconds, as shown on figure 1.5, which is not the reality for the market being analyzed (memory devices).

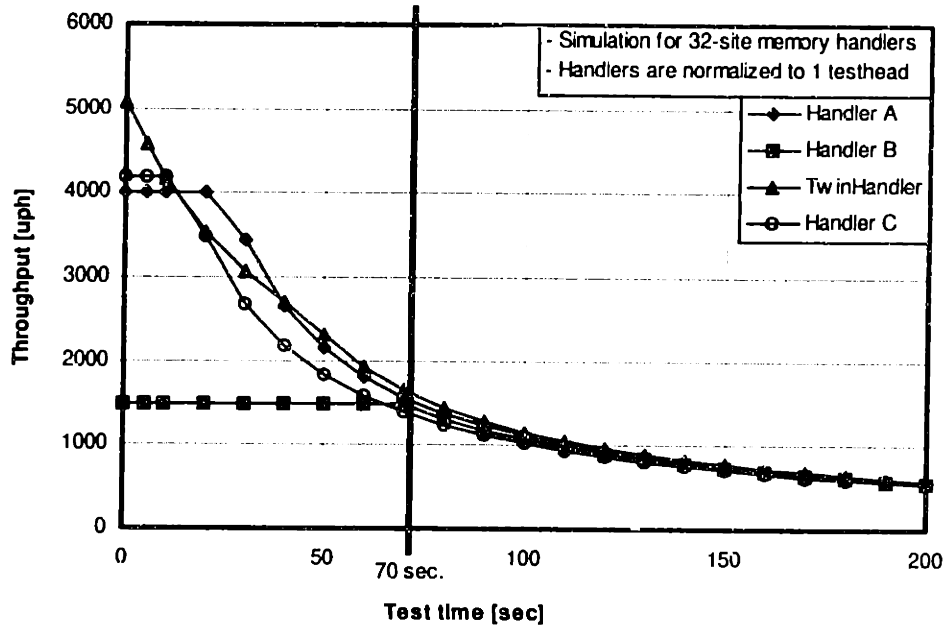


Figure 1. 5 - Throughput Benchmarking

From Figure 1.5 it is interesting to notice that all handlers but the TwinHandler reach a plateau on the throughput vs. test time curve. This is the point where throughput of the machine is sort limited. The TwinHandler has two mechanisms working in parallel for testing and sorting (not in series like in existing handlers), so for each test time there is a unique throughput. It is also curious that above approximately 70 seconds test time all handlers have the same throughput. This suggests that for memory devices, where test times may be on the order of 100 seconds, all handlers are equivalent, and their sorting mechanism is probably idle for a good portion of the test time (idle capital equipment). Equation 1.2 can be rewritten for the TwinHandler as:

$$TPT_t = \frac{2\ n}{T_p \cdot Y + T_f \cdot (1 - Y) + T_h + T_{sort} + T_t} \tag{1.4}$$

An alternative way to view these curves is to represent the dollar value of a certain throughput. Figure 1.6 shows curves for the same manufacturers based on their reliability data and on their price. Also on this same chart, the TwinHandler is priced accordingly to its reliability.

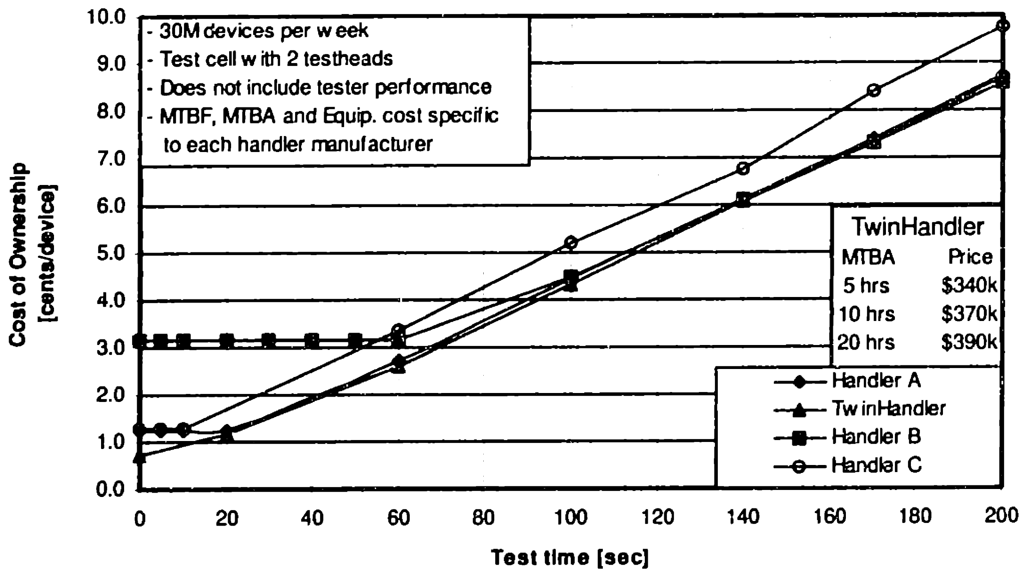


Figure 1. 6 - Cost-of-Ownership Benchmarking

Based on Figure 1.6 all manufacturers' handler but C cost the same to their customers with test times greater then 70 seconds. By adjusting the curve of the TwinHandler to be right bellow the minimum of all manufacturers it is possible to figure out the maximum selling price of this machine as a function of Mean Time Between Assist. This chart allows the engineer to quantify reliability and actually price the machine based on reliability performance of a certain design.

These charts are based on handler manufacturer information and on the existing test floor layout, general bench marking data is shown on Figure 1.7.

Comparative Benchmarking													
Company	Model number	Devices Handled Per Hour @ Zero Test Time	Sort Categories	Temperature Range [Celsius]	Sort Time [sec]	Index Time [sec]	MTBF [hrs]	MTTR [min]	MTBA [hrs]	MTTA [min]	Number of Test Sites	Price
Handler B (P&P)	M6861A	1500	9	-30 C to +125 C	69.0	7.8	1000	20	23	15	32	$ 450k*
Handler C (P&P)	M3200	4200	11	-60 C to +160 C	14.4	13.0	500	20	4	15	32	$ 400k
Handler A (Gravity)	3287	4000	22	-55 C to +150 C	25.3	3.5	500	20	3.9	15	32	$ 300k
Twin (P&P)	KMX 100	7021	16	-60 C to +160 C	29.6	3.3	1000	20	10	15	32	$ 370k

* Normalized to 1 testhead and 32 sites

Figure 1. 7 - Comparative Benchmarking

1.4.2 Cell Arrangement

Estimating the cost of ownership of a cell arrangement requires considering additional factors. Just considering the CoO of each component is not sufficient. The interactions among the components of the cell must also be considered. By building on the basic CoO concepts introduced in the previous section, the CoO model can be extended to cell arrangements.

A Cell Testing System (cell arrangement) is an integrated system consisting of handler, tester, sorter and transport mechanically linked together. The machines may not come from the same supplier. The main considerations of a cell analysis are: mean time between failure, mean time to repair, throughput rate, yield.

An integrated system (cell) can be configured in several ways, as shown in Figure 1.8:

• Serial: several different machines, each performing a different operation in sequence;

• Parallel: a group of similar machines, each performing the same operation;

• Combination: a combination of machines that are used in both serial and parallel operations.

Each configuration requires a different approach for estimating critical CoO factors.

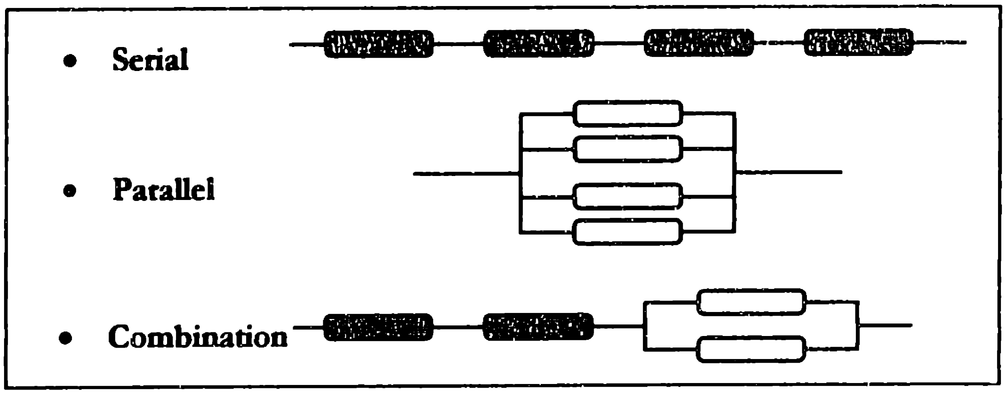


Figure 1. 8 - Integrated system configurations

1.4.2.1 Mean Time Between Failures

By assuming an exponential distribution of time to failure, the failure rate (Fr) can be estimated as:

$$F_r = \frac{1}{MTBFp} \qquad (1.5)$$

For a simple serial system, component failure rates may be added to estimate system failure rate. However, an integrated system is not a simple serial system. If assumed a four handler system with a single transport mechanism, the transport is serial with respect to each handler. Each handler is independent of the other handlers unless handler repair requires interruption of the transport or common process supplies for safety reasons. Thus, each handler has two failure modes:

• Serial (interrupts other handlers);

• Parallel (no interruption of other handlers).

In a completely serial system (that has a serial failure mode):

$$MTBF_A = \frac{1}{F_{r1} + F_{r2} + \ldots + F_{rn}} = \frac{1}{\sum_{i=1}^{n} \frac{1}{MTBF_{Ai}}} \quad (1.6)$$

If the parallel failure mode dominates, then estimating the system MTBFp becomes more complex. In addition to the exponential distribution of times to failure, one must assume that the MTBFp of the parallel components are equal, so:

$$MTBF_B = \left(1 + \frac{1}{2} + \frac{1}{3} + \ldots + \frac{1}{n}\right) MTBF_{Bi} \quad (1.7)$$

In this specific case, all transport failures will interrupt other handlers, but the four handlers may the thought of as operating in parallel. Thus, equations 1.6 and 1.7 must be combined to estimate system MTBFp. First the parallel failure mode MTBFp will be estimated for the four handlers, and that estimate will be combined with the transport MTBFp and the serial failure mode MTBFp for the four handlers to estimate system MTBFp [SEMATECH '93]:

$$MTBF_{sys} = \frac{1}{\frac{1}{MTBF_{Ar}} + \frac{1}{MTBF_{Ah1}} + \frac{1}{MTBF_{Ah2}} + \frac{1}{MTBF_{Ah3}} + \frac{1}{MTBF_{Ah4}} + \frac{1}{\left(1 + \frac{1}{2} + \frac{1}{3} + \frac{1}{4}\right) MTBF_{Bh}}} \quad (1.8)$$

The results of a combined MTBFp such as describe in equation 1.8 are not distributed exponentially, thus invalidating the assumption in equation 1.5. However, these simple equations will provide an initial estimate of system MTBFp for an integrated system. For better simulation, considering that MTBFp has a statistical distribution with a certain dispersion, there are several commercial software that can deal with discrete-event simulation and dynamic simulation (not really necessary in this case).

1.4.2.2 Mean Time To Repair

Using the component MTBFp, one can estimate the annual number of failures by component. The system MTTR is the average of the component MTTRs weighted by estimated number of annual failures per component.

1.4.2.3 Throughput Rate

Like MTBFp, system throughput rate depends on whether the components of the system are configured in serial or in parallel. If four handlers of an integrated system were configured serially, then products would move from handler A, to handler B, to handler C, and finally to handler D (completing the sequence). In this case, the system throughput rate is that of the bottle-neck handler:

$$TPT_{sys} = MIN(TPT_A, TPT_B, TPT_C, TPT_D) \tag{1.9}$$

If four handlers of a system were configured in parallel such that each tool performed the same task, the system throughput rate would be defined as the sum of the throughput rates:

$$TPT_{sys} = (TPT_A + TPT_B + TPT_C + TPT_D) \tag{1.10}$$

Integrated systems are the combinations of serial and parallel configurations. Equations 1.9 and 1.10 may be combined to provide an initial estimate of system throughput rate. However, machine interactions that may limit throughput are neglected in this estimate.

1.4.2.4 Yield and Other Factors

Throughput yield and defect-limited yield for parallel systems are the average of the component yields weighted by component throughput rate. Throughput yield and defect-limited yield for serial systems are the product of the component yields.

1.5 Thesis Contributions

This thesis has three major areas of contributions:

- The introduction of a new modular test floor layout for TAP. The modularity of test floor design is intimately dependent on the modular design of the machines used.

- The design of fundamentally new mechanical elements that enable high parallelism of device testing, sorting and handling in different processes;

- The modular design strategy and proof of concept by design and implementation of a modularly built Sorter for semiconductor TAP application.

Several authors propose methodologies for modular machine design [Kusiak '96, Chen ' 94, Ulrich '91, Yan '94, Ouyang '95]. These are based on the creation of a database of mechanical elements (bearings, couplings, actuators, metrology, etc). The modular design framework developed for this thesis focuses on the freedom to select whatever component the design engineer can choose at that moment in time. This is not limited to a universe of components loaded into a database. The constraints are in a higher level, where the problem is described from a deterministic stand point of geometry (overall dimensions, physical interfaces, utilities, data transfer), static and dynamic performance, reliability and kinematics. This way this thesis presents a set of guidelines to work out the design issues to achieve the set of deterministic goals for an individual machine module. Modularity is fundamentally important for machine reliability. The cost-of-ownership analysis described in previous section is used, on chapter one, to highlight the importance of reducing Mean-Time-To-Assist of a back-end equipment. Physical modules can be designed to be quickly interchangeable, minimizing MTTA.

Another contribution is the development of new hardware to facilitate device handling. The duality stiffness and compliance is difficult to consider in design. Several situations arise where stiffness is required in some degrees-of-freedom and compliance is necessary in the others. For this purpose the best solutions are based on the design of flexural bearings. This thesis presents the design of a Flexible Mount Kinematic Coupling (KinFlex™), shown in Figure 1.9.

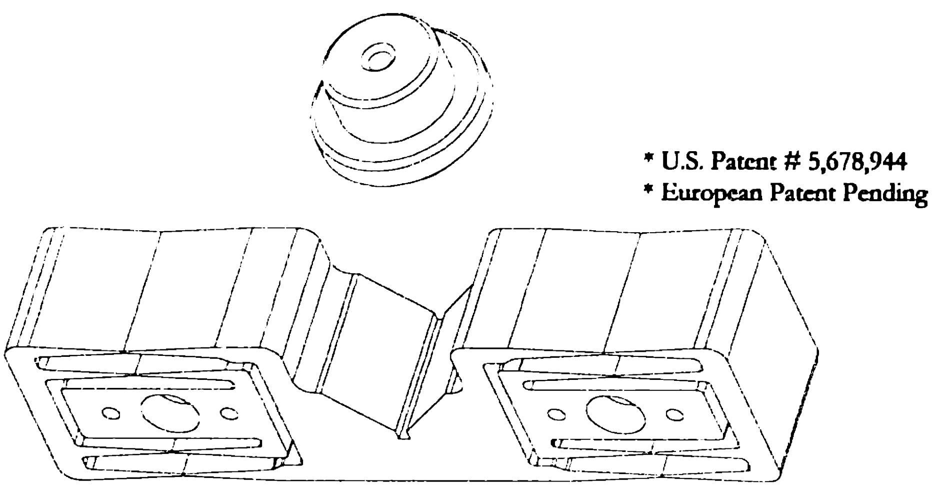


Figure 1. 9 - Flexible Mount Kinematic Coupling

Another application for flexural bearings is the design of a Flexible Mount Package Insert (FlexInsert™), shown in Figure 1.10 and further discussed on ***Enabling Technology*** chapter, for package trays (JEDEC standards). The test industry is moving towards massive parallel testing of devices to maximize throughput. Machine reliability becomes an even more important issue as well as handling reliability (known as jamming).

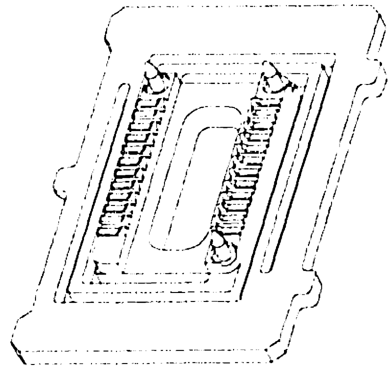


Figure 1. 10 - Flexible Mount Package Insert
* U.S. Patent pending

In addition to the modular machine design framework and the deterministic *enabling technology* developed, the design of a unique stand-alone Sorter is presented as a first step towards semiconductor back-end automation, concept is shown in Figure 1.11. This machine was developed with support from Kinetrix, Inc., a Teradyne subsidiary.

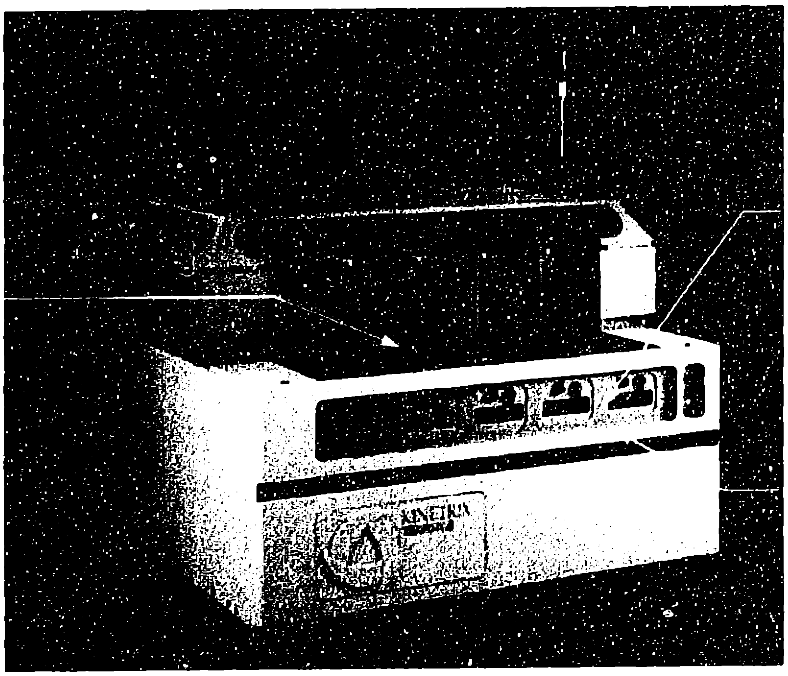


Figure 1. 11 - Apollo Sorter Concept

This stand-alone Sorter breaks the traditional concept used by semiconductor equipment manufacturers. The project proceeds from the initial product conceptualization, concept selection, product planning, application of modular design guidelines, design detailing, fabrication and testing of a prototype machine. Therefore, this machine not just sets the cornerstone for a new test floor layout but also shows the application of the techniques proposed for the modular machine design framework.

1.6 Thesis Structure

This thesis contains three basic parts. The first part of is covered on Chapter 2. A new layout configuration for TAP of the semiconductor manufacturing process is proposed. The first step towards this new configuration is the implementation of modular machine design and also on a greater scale to treat the machines as modules on a Modular Testing System (MTS).

The second part is covered on Chapter 3 where the guidelines of Modular Machine Design are presented. This chapter will not focus on modular hardware construction, but on the basics of modularly analyzing the design. Deterministic design makes use of several tools to accurately predict performance of a machine. The main focus of these guidelines is the design of machines for the TAP of the semiconductor manufacturing. TAP still lacks mechanical optimization of equipment design and a higher degree of automation.

Chapter 3 also introduces guidelines and a analytical tool for Time Budgeting. One of the major concerns on designing automated TAP equipment is correctly budgeting the time across all tasks to obtain the desired throughput performance of the machine. Time Budgeting also helps evaluate different machine concepts. A few parameters that dominate the equations of time are discussed and placed in the framework of design.

The third part of this thesis, presented on Chapter 4, describes analytically the enabling technology that create the necessary chemistry for modularity and parallel processing of devices.

Finally Chapters 5 and 6 present the Apollo Sorter project and an evaluation of the designed machine in light of the methodology described on Chapter 3.

Chapter 2

MODULAR TEST FLOOR DESIGN

2.1 Test, Assembly & Packaging Automation

Several authors proposed automation and integration of the Test, Assembly and Packaging of semiconductor manufacturing, [Mackenzie '97], [Hartman '97], [Grant '97], [Seggern '97], [Pothoven '97]. A total integration of Test, Assembly and Packaging is a big one step towards automation. This thesis approaches the automation and integration of TAP from a more basic perspective. The first proposed step is to develop flexible TAP equipment. Flexibility in this case means modularity, ease of interchangeability and easy configuration for different applications.

There are so many different types of packages with such a wide spread of test cycle time that it becomes virtually impossible to have one universal machine being able to handle all of them. The ability to adapt to each individual package is a design flexibility of the machine. Design flexibility is better achieved by modularization. Taking, for example, the case of IC handlers: a specific handler can adapt to a certain family of packages, but none can adequately adapt to various test times. A handler can be usually described as a two process machine:

1. First Process: It presents devices to the Test Head and hold them during test;

2. Second Process: It sorts the devices to the output bins corresponding to the results of testing.

This is simple enough, but still each process has its dependencies. The first process is limited by the test time for the device and/or the mechanical index time. The second process is dependent on how fast the mechanism can sort devices and how many are binned (yield dependency). For any package there will always be waiting time, either for the mechanism that perform process one or the mechanism that perform process two. This thesis proposes an evolution in the design of handlers by separating the two process functions into two distinct machines and integrating them and the tester in a Modular Testing System.

It is possible to elaborate a cost-of-ownership comparison between the old test floor layout and this new modular system approach, although it is difficult to describe the performance for all different combinations of devices being tested at one time in a test facility. This thesis is not going to focus on the design of an optimized test floor based on this new layout, but on the modularization of the machine design to achieve even higher levels of flexibility and reliability.

2.2 The Modular Test Floor Design

2.2.1 Introduction

Automation is long term strategic investment and therefore the design should accommodate flexibility. Most people perceive that equipment integration on the test floor is about fixed functional mechanization, this view has to be changed by the practical implementation of integrated systems which are modular and flexible [MacKenzie'97].

2.2.2 The Modular Testing System

The proposed test system has the same basic functionality as the existing one. The difference resides in the flexibility to adapt to new production scenarios without the need to buy new capital equipment.

Flexibility is achieved by adding modularity to the equipment. Maudsley in the beginning of the century introduced the term modularity, in the context of interchangeable parts in guns. This facilitated mass production. In this thesis, modularity permits that a complete functional unit within an equipment can be replaced by an identical unit or by another unit that perform a different function but that would make use of the same platform from which the equipment was built (facilitating customization).

This modularity is possible by analyzing the functions performed in the production floor and by grouping machines into families that could be designed to be modularly interchangeable.

A manufacturer of semiconductor could design a production floor for today's needs and still be capable of adapting the equipment on the floor for future needs without having to buy completely new equipment. The design of the equipment need to be thought up front to include all the features required to perform different functions.

Figure 2.1 shows an example of back-end test & packaging processes as it is today. Note that every machine can perform only the function it is designed for.

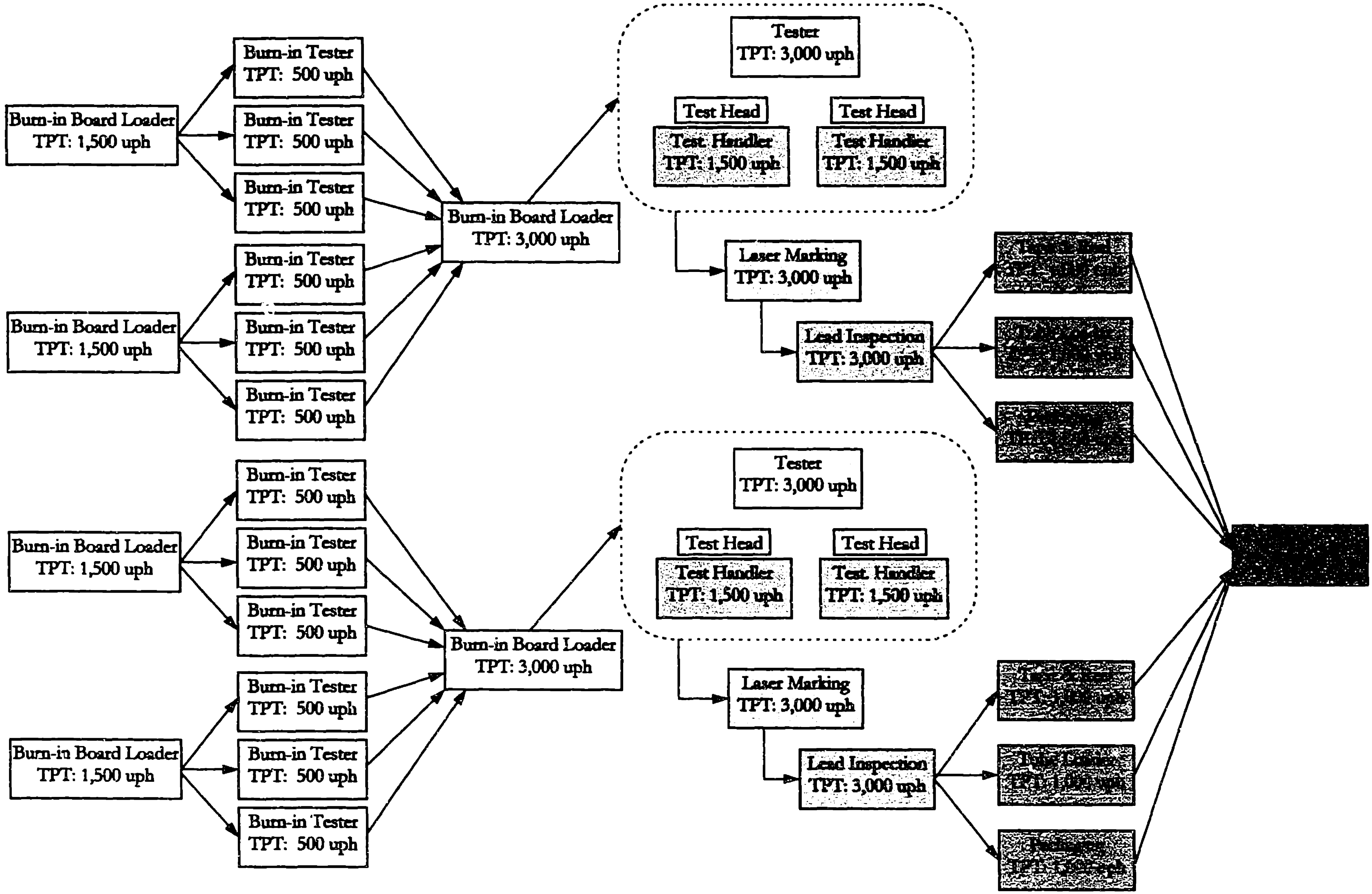


Figure 2. 1 - Existing back-end test & assembly production layout

Figure 2.2 shows the same production floor with modular equipment. This time the equipment is implemented to the floor to execute the same functions as in the previous layout.

Observe that the state of the art layout utilizes 9 machines between burn-in board loading and shipping product. Using modular machines only 4 systems (including testers) are needed. Due to machine modularity, the same system platform can perform a variety of different functions with add-on modules. This also enhances the manufacturers ability to rapidly adapt to changes in the market place

It is noticeable that a flexible layout due to the existence of modularly built machines facilitate the change from one production need to another. This suggests that one can make a significant contribution to the state of the art of semiconductor manufacturing by changing the way TAP back-end equipment is designed implementing modular machine design concepts. This is the main focus of this thesis, to propose a method of modular design and to implement it into a successful product line.

Flexibility comes both from the ability to shuffle machines around based on production needs but mostly from the ability to change the machine functionality. This is the type of modularity that helps reliability by easily replacing modules when preventive maintenance occurs but also allows the customer to replace modules by others that have different functions. So the layout of the production floor is modular because a system's platform can now perform a variety of functions like sort devices into output trays (if using a Tray Module), tubes (if using a Tube Loader Module), tape (if using a Tape & Reel Module), it can perform lead inspection (if using a Lead Inspection Module), it can function as a burn-in board loader (if using a Burn-in Board Module), etc.

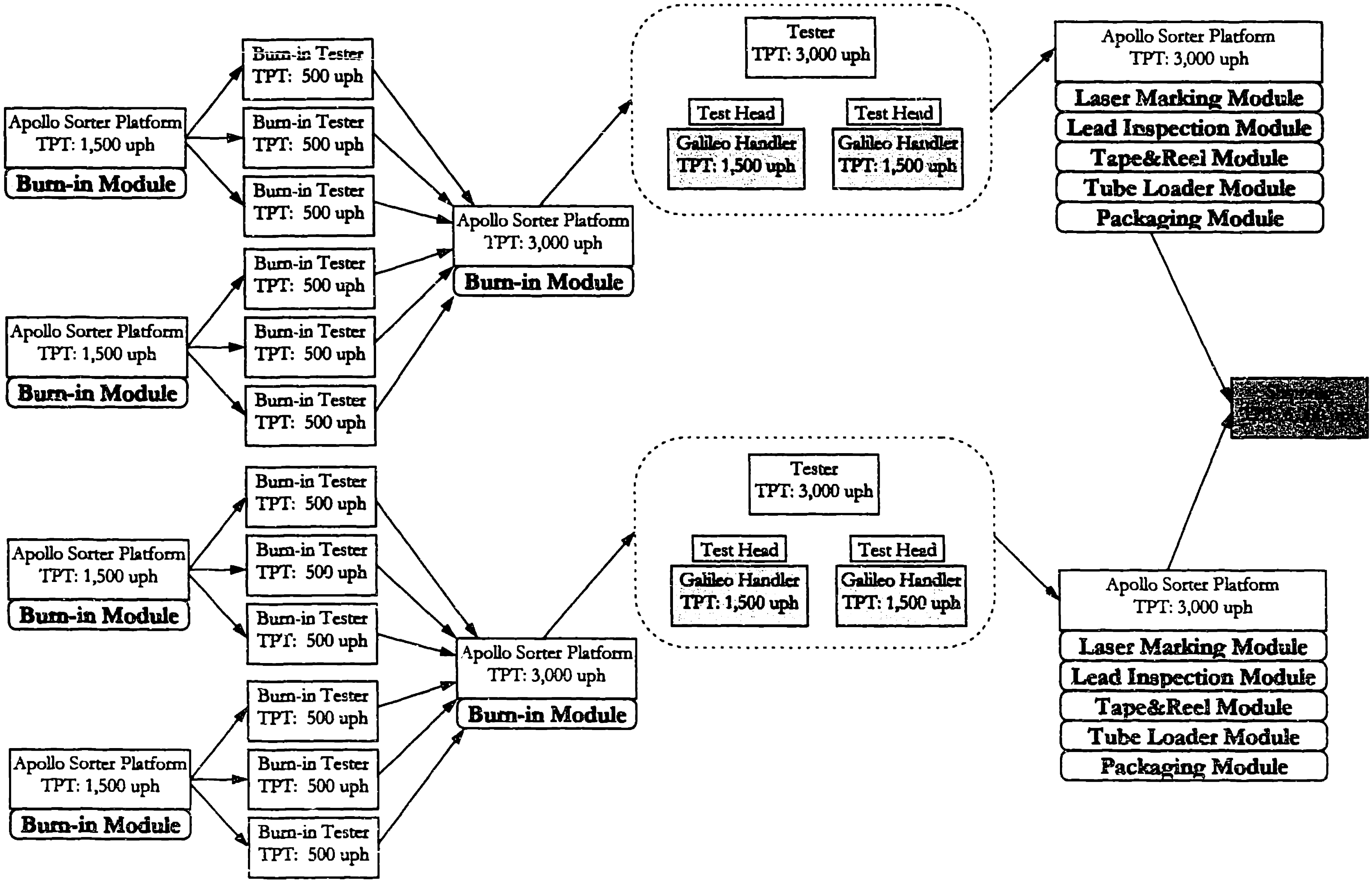


Figure 2. 2 - Flexible back-end production layout

The success of the Modular Testing System is tied to the ability to design modular machines. The testing system is chosen as the first step into TAP production layout because test represents a gross fraction of TAP's cost of a device.

2.2.3 Test Floor Economics

It is extremely difficult to accurately represent the cost of producing devices in a flexible floor layout. The inherent difficulty comes from the lack of knowledge of future market changes and manufacturing needs as well as its cost implications to semiconductor manufacturers, which could require new equipment on a standard production layout and new modules for existing platform machines on a modular floor layout.

For the sake of this thesis one example was chosen to represent the Cost-of-Ownership of both floor layouts. A SEMATECH '93 based model is used to equate all the factors into a common understandable denominator, money or dollars of cost per device produced. The comparison is then reduced to existing floor layout with a double headed tester and two handlers attached to it versus a double headed tester with two In-Tray Handlers plus a Sorter. The model is balanced such that only the percentage of the Sorter utilization is calculated into the Cost-of-Ownership. Figure 2.3 presents what is called a *Management Report*, which represents the final results of viability of each test floor model. The last line of this spreadsheet is the total cost per good devices shipped. Even though the correct input numbers are difficult to obtain from chip manufacturers, the relationship between the numbers stays valid (within ± 5%). In this case, for a memory device tested 32 in parallel the ratio between the two proposed floor layouts (handler vs. Handler+sorter) is:

$$RATIO = \frac{TotalCost\_GoodDUTOut_{MODULAR\_LAYOUT}}{TotalCost\_GoodDUTOut_{STANDARD\_LAYOUT}} * 100\% = 72.7\% \qquad (2.1)$$

which means that the Modular Test Floor Layout costs 72.7% of the cost of the Standard Test Floor Layout, so it is 27.3% cheaper.

MANAGEMENT REPORT

PRINT SECTION

	Martin/J396 STANDARD	Martin/J396 MODULAR		DEFINITIONS
Cost Per Station	$1,050,000	$1,200,832	Dollars	Original purchase price of system
Number Of Stations Required	173	117	Stations	Production requirements/max. DUTs/week/system rounded up
Total Depreciable Costs	$194,273,700	$145,061,179	Dollars	Sum of all capitalized expenses/system * number of systems
Equipment Dependent Utilization	31.03%	33.67%	%	% hours/week system not undergoing any maintenance
Production Utilization	30.25%	32.89%	%	% hours/week sys. not undergoing any maint. or production tests
Composite Yield	94.06%	94.02%	%	(Tool yield)*(test yield)*(1-alpha error)
DUTs Out Per Week (At Composite Yield)	942,628	946,285	DUTs	Production requirements * composite yield
LIFE OF EQUIPMENT				
Equipment Costs	$196,991,069	$150,616,403	Dollars	Sum of equipment costs over life of equipment
Cost Per Good DUT Out	$0.84	$0.64	Dollars	Sum of equipment costs/year/(number of good DUTs out/year)
Production Costs	$278,400,476	$194,203,650	Dollars	Sum of production costs over life of equipment
Cost Per Good DUT Out	$1.16	$0.83	Dollars	Sum of production costs/year/(number of good DUTs out/year)
Total Costs	$475,391,545	$344,820,053	Dollars	Sum of equipment and production costs over life of equipment
Total Cost Per Good DUT Out	$2.02	$1.47	Dollars	Sum of all costs/year/(number of good DUTs out/year)

Figure 2. 3 - Cost-of-Ownership comparison between Standard and Modular Test Floor Layout

Due to the complexity of a billion dollar fab, creating a complete model of the production and test processes is a daunting task. Figure 2.4 shows cost items that go into a cost of ownership model. Some of these costs are both easy to determine and measure, while others are more difficult.

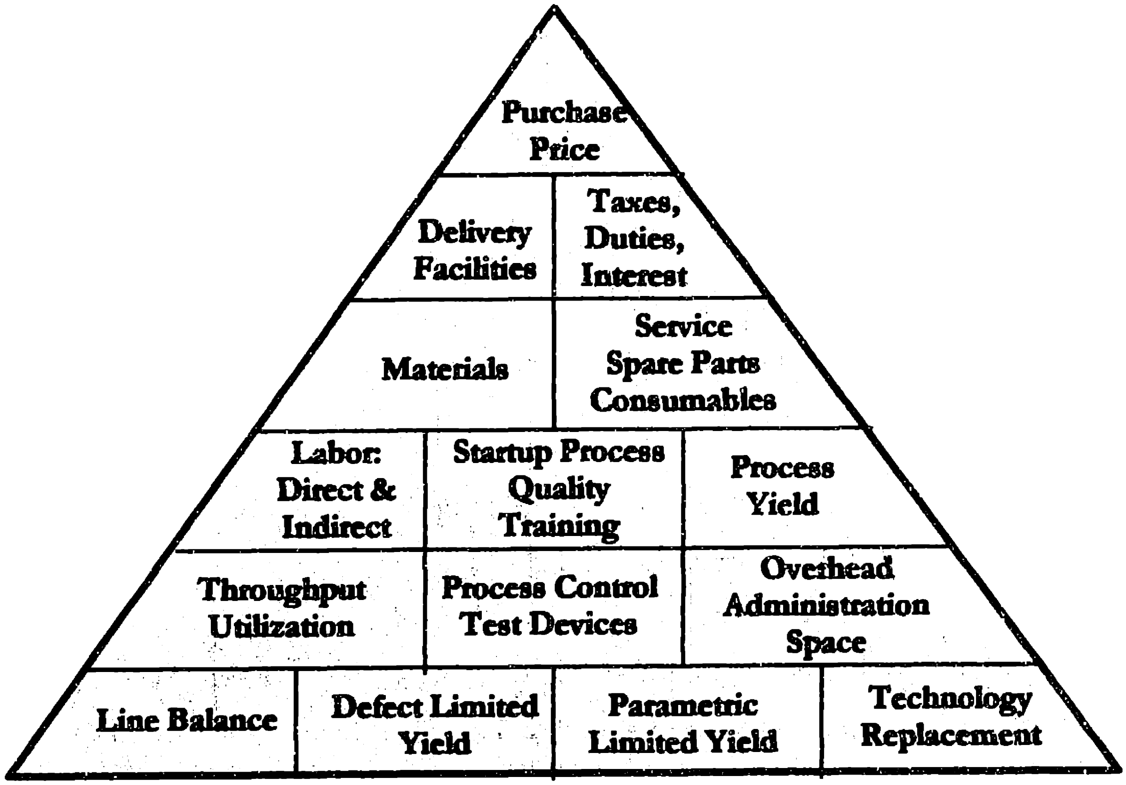


Figure 2. 4 - Factors in Cost-of-Ownership

Chapter 3

MODULAR DESIGN FRAMEWORK

3.1 Introduction

The semiconductor industry is undergoing a major paradigm shift that is taking it from traditional manufacturing into a world of agile manufacturing. Agile corporations should be able to rapidly respond to all changes in the market environment and present flexible products to their customers.

In the previous chapter a concept for flexible semiconductor TAP production floor was presented. The advantages of that layout are on the flexibility of the machines to perform a different variety of functions. If customer requirements change, a flexible product structure can be readily adapted by changing a small number of components or modules. To achieve high flexibility, a modular machine design strategy is needed. A lot of effort has been put into Design for Manufacturability (DFM) and Design for Assembly (DFA), but little has been done for what some call Design for Modularity.

Several authors, Kusiak '96, Chen ' 94, Ulrich '91, Yan '94, Ouyang '95 and others, proposed different methodologies for modular product and machine design. One can find a complete list of advantages and disadvantages for modular design amongst the authors sited.

Modular machine design has a different set of boundaries then commercial product design. Those who proposed modular machine design methodologies tend to focus on creation of databases of components and on the power of computer-aided design tools. This proposed framework of design does not intend to limit the designers choice to a pool of 3D-modeled components but to brake the design problem into isolated sub-designs that are simpler to analyze.

3.2 The Concept of Design Control Volume

When trying to describe fluid flows engineers use control volumes to help describe effects of the fluid on a certain region of space. In the design of mechanical elements there are finite elements that can be tracked individually, so the Design Control Volume, DCV, in this case has no similarities to the one used for Fluid Mechanics.

The concept though is to define a geometrical portion of space, where the geometry of the DCV matters and work the design inside this region. Without going into the details of how the module of the machine is defined as a subsystem there is a volume wherein the design of a electro-mechanical system takes place, and where the boundaries or surfaces of the module are defined, see figure 3.1.

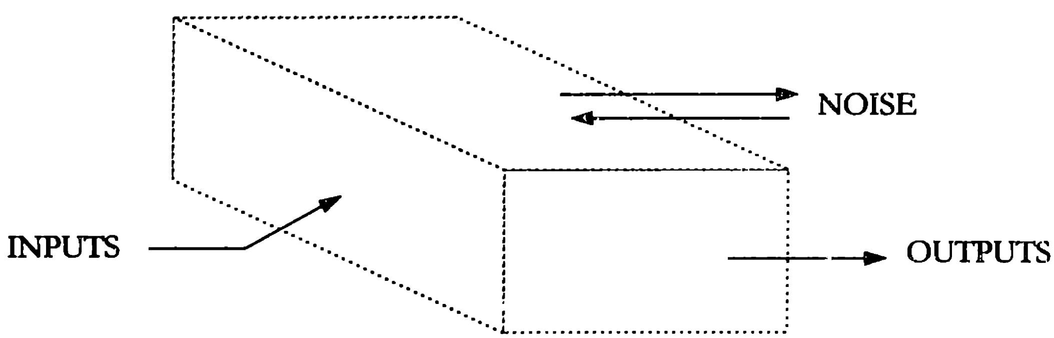


Figure 3. 1 - Design Control Volume

The intent is to have a well prescribed set of iterations between the module, designed inside the DCV, and the rest of the system (machine), outside of the DCV. The elements that cross the boundaries of the DCV would be defined as being either:

• Force/Energy type (mechanical support, heat transfer, dynamic force, fluid, power for motors and sensors, etc);

• Material (for the semiconductor industry the processing material could be a silicon wafer, chips, cassette or tray of chips);

• Signal (input/output signals from an external control unit that define when/what functionality should the module perform at an instant of time).

There are potentially other forms of defining elements that cross the boundaries of a DCV. These three types of elements can be either controlled (Inputs and Outputs) or non-controlled (Noise). The objective is obviously to minimize non-controlled transfers.

It is mandatory that a module specification is written prior to design conceptualization phase for modularity to follow a smooth process. This specification lists in detail:

• The desired functionality of the module. The DCV can be defined without knowledge of functionality. On a modular machine as presented in this thesis several functions (different modules) would fit in the DCV described by this specification;

• The material transfer through the surface boundaries of the DCV. Note that there is no restriction about material cycling inside the volume;

• The geometry and how it will interact with the rest of the system so that force/energy flows through accordingly; e.g. structural integrity, surface resistance for heat transfer, positioning accuracy of the module with respect to the rest of the system in accordance with the systems error budget, location and type of connectors for energy/signal inputs and outputs;

- The energy and signal specifications; e.g. utility inputs, asynchronous communications, servo motor power inputs, stepper motor power inputs, encoder outputs, digital outputs, digital inputs, analog inputs, analog outputs, etc;

- Other specifications to bound the design, including maximum allowed cost.

Overall, the Design Control Volume limits the flow of force/energy, material and signal through the surface boundaries of the DCV but does not describe or limit what happens inside the boundaries. Instead of creating a database of components for the design engineer, this creates a space for the designer to be creative. This sets a well define limit and a well defined problem, but does not limit creativity. This concept also helps break the system level complexity of a machine into subsystems that can be worked out independently and in parallel to each other with the assurance that the final outcome fits and work together like Lego™ blocks.

3.3 Modular Design Strategy

Problem definition: "Given that markets for semiconductor equipment is moving toward customer expectation of a flexible production layout, how can one support the design decision on structuring a modularly built family of machines ?"

The strategy is to architect the product before designing it, so that plug-in modules can deliver the required functionality. The key research question that arises is: "What are the underlying design principles that guide design for modularity?".

Suh's axiomatic design represents one of the more recent approaches in the development of design principles [Suh '90]. He describes a design equation which links functional requirements for a specific design to the design parameters.

$$\{FR\} = [A]\{DP\} \tag{3.1}$$

where {FR} is the vector of functional requirements;

{DP} is the design parameter vector.

[A] is the design matrix which specifies the mapping from the functional requirements to the design parameters. The elements of matrix [A] can be represented as:

$$A_{ij} = \frac{\partial FR_i}{\partial DP_j} \tag{3.2}$$

It is though not always clear to the designer how to generated these matrices, specially when the complexity of the design increases. In this thesis a graphical strategy is proposed where the designer can visually identify functions and constructional boundaries for the product.

The manufacturing equipment in the semiconductor Test, Assembly & Packaging (TAP) industry is used to transport and process ICs in a variety of forms. They have in common the need to transfer material between stations (input, process, output, etc). Because TAP industry is focused on Cost-of-Ownership (CoO), as explained in previous chapters, the desired outcome of the machine is processing ICs with minimum cycle time and maximum reliability. For that matter cycle time turns out to be one of the major functional requirements in designing these systems together with system reliability.

The strategy is to define up front all possible functional modules that could be incorporated in the system platform. This involves customer inquires and market investigation. Bearing in mind all possible combinations of modules (different machines) create a block diagram that shows the material transfer in the system. Take into account geometrical incompatibilities and constrains (e.g. two independent processes can not happen in the same location for a given material, so each requires a material block). It is also important to maintain a well balanced production cycle, so make sure the modules processing time match or that there is a multiplying integer that matches them. This is important to avoid idle mechanisms in a system, like what was happening with existing test handler. If such situation appears it is possible that the best solution is indeed to separate the functions into different machines (e.g.: state-of-the-art test handler = in-tray handler + sorter).

With several diagrams (one for each type of machine) one can compare them and identify repeating patterns in the material transfer. Isolate them outside of the diagram as a potential module. As the next step add force/energy transfers and continue to compare diagram and isolate common ones. Figure 3.2 is a representation of the above strategy.

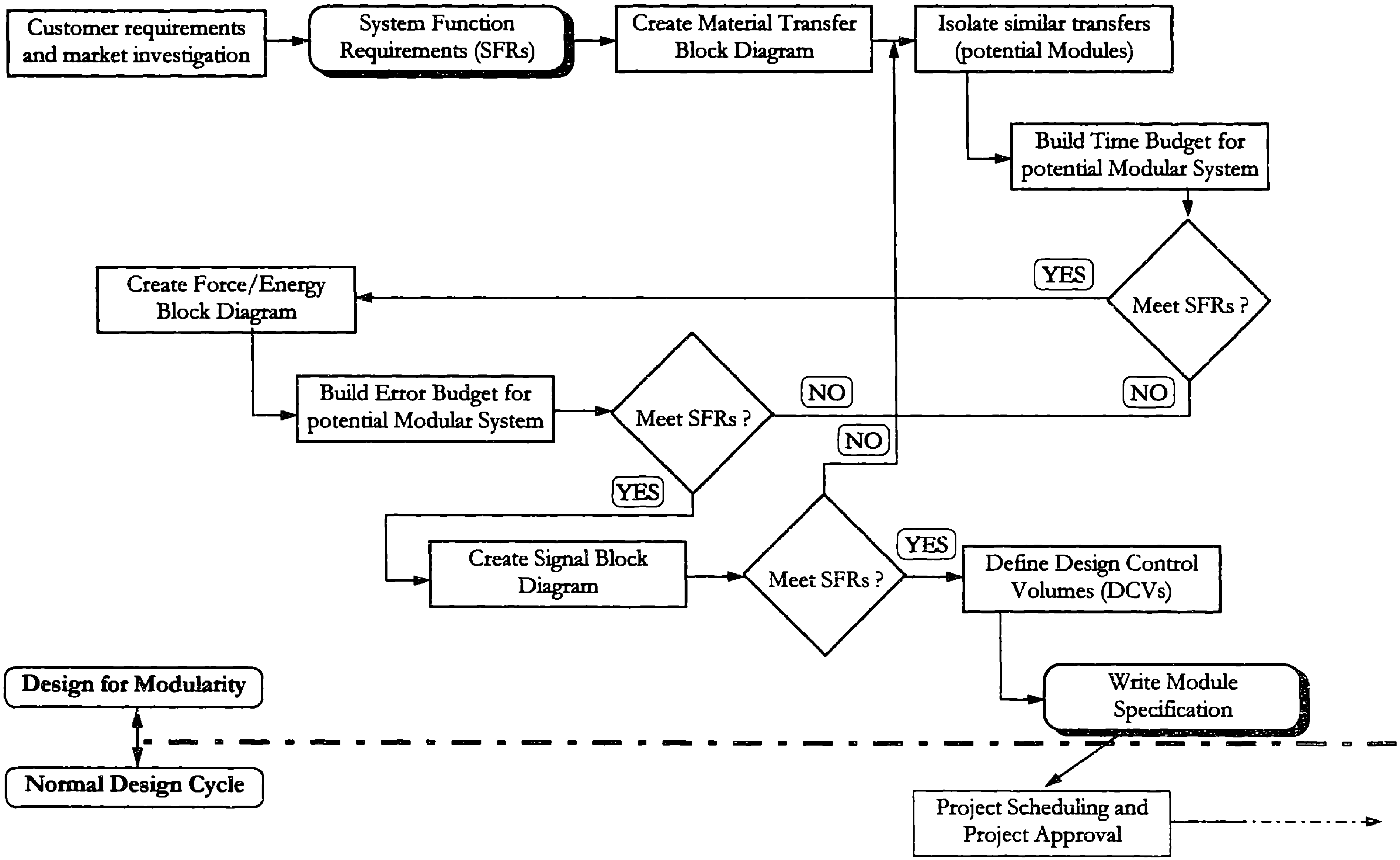


Figure 3. 2 - Problem-solving strategy for Design Modularization

The above strategy is classified as a scheme process, because it makes use of illustration and empirical data to represent the strategy. The author does not believe in the existence of a universal solution for design but in proposed guidelines that help better describe a problem. Chapter 5 should serve also as an example of application of the proposed strategy, where the success is measured by customer acceptance of the modular family of machines that the Apollo platform introduces.

3.4 Time Analysis for Semiconductor TAP Manufacturing Equipment

Semiconductor Test, Assembly & Packaging manufacturing equipment has in common the need to perform material transfers and processing in the least possible time. These processes do not add value to the device so time is fundamentally important. Since the process is mostly defined by the associated physics, material transfer time appears to be the neglected element in the design of existing equipment.

A Time Budget spreadsheet is designed to support design engineers when building an estimated table of move times and by consequence throughput of the equipment being conceptualized. The intent is to facilitate concept selection by supplying a very important piece of information about the design, cycle time.

Shown on figure 3.3 are five basic motion graphs for mechanism movements. The motion is basically defined by:

$$\begin{aligned} x &= x_0 + v_0 t + a\frac{t^2}{2} + j\frac{t^3}{6} \\ v &= \frac{dx}{dt} = v_0 + at + j\frac{t^2}{2} \\ a &= \frac{d^2x}{dt^2} = a_0 + jt \\ j &= \frac{d^3x}{dt^3} = j \end{aligned} \tag{3.3}$$

where:

x - displacement

v - velocity

a - acceleration

j - jerk

t - time

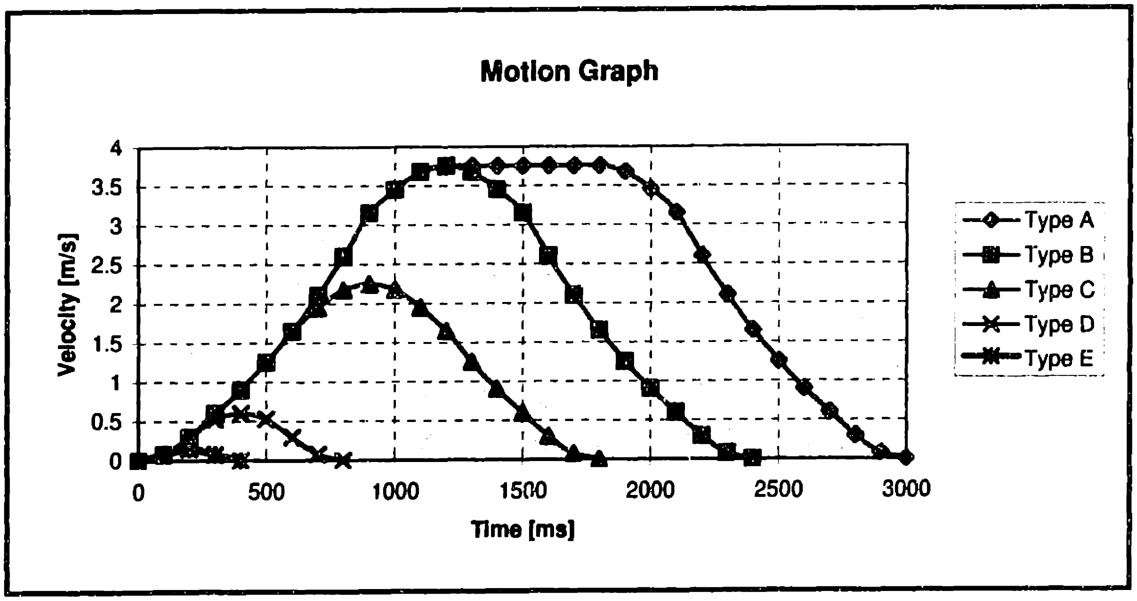


Figure 3. 3 - Motion Graph

Using the nomenclature,

s - subindex for the S portion of the curve

r - subindex for the ramp with constant acceleration

a - subindex for total acceleration up to top speed (including S portion and ramp portion)

X - is the motion displacement

V - is the maximum velocity of motion

a - is the maximum acceleration

Each graph can be described by a set of equations such as:

<u>Types A & B:</u>

$$
\begin{aligned}
t_s &= \frac{a}{j} \\
x_s &= \frac{2a^3}{3j^2} \\
v_s &= \frac{3a^2}{2j} \\
t_{ramp} &= \frac{V}{a} - \frac{3a}{j} \\
x_{ramp} &= \frac{V^2}{2a} - \frac{3Va}{j} + \frac{9a^3}{2j^2}
\end{aligned}
\qquad (3.4)
$$

and combining all that,

$$
\begin{aligned}
x_a &= \frac{V^2}{2a} - \frac{3Va}{j} + \frac{35a^3}{6j^2} \\
t_a &= \frac{V}{a} - \frac{a}{j}
\end{aligned}
\qquad (3.5)
$$

<u>Types C & D:</u>

$$
\begin{aligned}
x_{ramp} &= \frac{a t_{ramp}^2}{2} \\
t_{ramp} &= \sqrt{\frac{X}{a} - \frac{8a^2}{3j^2}}
\end{aligned}
\qquad (3.6)
$$

which leads to:

$$x_a = \frac{X}{2}$$

$$t_a = \sqrt{\frac{X}{2a} - \frac{4a^2}{3j^3}} + \frac{2a}{j} \quad (3.7)$$

Type E:

$$x_a = \frac{1}{2} j t_a{}^3$$

$$t_a = \sqrt[3]{\frac{X}{j}} \quad (3.8)$$

As a simplification to the equations above one could use the standard trapezoidal move graphic and estimate the settling time to be equal to the system's time constant:

$$\tau_{\text{settling}} = \tau_{\text{mech.}} = 2\pi\sqrt{\frac{\mathrm{M}}{\mathrm{K}}} \quad (3.9)$$

where, M - system's mass

K - system's stiffness

Combining the above equations into a spreadsheet simplifies the process of collecting and processing the data. Figure 3.4 shows a portion of the Time Budget spreadsheet, which is intended to be of general use for computing cycle time of a system based on its individual movements. The real cycle time is dependent on random factors (like process optimization, machine reliability, etc.), so this spreadsheet intends to be a first level estimate of the cycle time, which has major influence on early design decisions as well as creates a budgetary time for individual module cycle time. The budgetary time for a module is part of the Module Specification document described before, and so it basically defines, together with physical data about the mechanisms, important design parameters: e.g. motor data (power, peak torque, utilization, etc).

Axis Parameters

	AxisName	MaxVelocity	MaxAccelaration	Jerk	LumpedTime	SetTime
1	Gantry	1000	9810	5	0.000	0.050
3	GripperSiteUp	0	0	0	0.600	0.050
4	GripperSiteDown	0	0	0	0.600	0.050
5	GripperSiteSuction	0	0	0	0.100	0.050
6	GripperSiteBlow	0	0	0	0.100	0.050
7	GripperTrayOpen	0	0	0	0.100	0.050

Media Information

Input Tray Information		Output Tray Information	
TrayNumber	2	TrayNumber	1
Columns	12	Columns	12
Rows	16	Rows	17
Xpitch	10.5	Xpitch	10.5
Ypitch	19.2	Ypitch	17.9
DeviceCount	192	DeviceCount	204

TrayNumber	Package	Body Size (inches)				Tray Matrix		Packages/Tray	Lead count	Column pitch	Row pitch	JEDEC Spec
			X	Y	Z	*Columns*	*Rows*			X	Y	
1		.300"x.625"	0.335	0.625	0.138	12	17	204	24	10.500	17.900	CO-028
2		.300"x.675"	0.335	0.675	0.138	12	16	192	20, 24	10.500	19.200	CO-028
3	SOJ 300 mil	.300"x.725"	0.335	0.725	0.138	12	15	180	28	10.500	20.400	CO-028
4		.300"x.825"	0.335	0.825	0.138	12	13	156	32	10.500	22.950	CO-028
5		.300"x1.075"	0.335	1.075	0.138	12	10	120	42	10.500	29.300	CO-028

Media Transfer, Move 6 Devices from Automation Tray (32-site tray) to Shipping Tray (135-site tray)
6-Sites Fixed Gripper

Gripper Sites	6	
Gripper Type	0	(1 for Expanding Gripper; 0 for Fixed Pitch Gripper)
Input Tray, # of Sites	32	
Ouput Tray, # of Sites	135	

Throughtput	3245	uph
Total Sort Time	35.50	sec
Total PickUp Chips Time	23.45	sec
Total PickUp Tray Time	5.78	sec
Total Gantry Motion Time	6.27	sec

	Motion Description	Axis	Motion Frequency	X Dist (in)	Y Dist (in)	Move Time	Lumped Time	Settle Time (sec)	Total Move Time (sec)
1	Load Gripper (6 devices)	GripperSiteDown	1.00	0.000	0.000	0.000	0.600	0.050	0.65

Figure 3. 4 - Time Budget spreadsheet

Using the Time Budget spreadsheet the design engineer can selectively analyze the motion on each axis or in each module and develop a better understanding of his/her design.

Chapter 4

ENABLING TECHNOLOGIES

4.1 Introduction

The Enabling Technologies are the elements of design that inhibit modular construction of semiconductor TAP machines. They are not necessary in every design but they improve the chances of a modular machine design to achieve its desired performance.

Modularity has several advantages but one disadvantage to note is the difficulty to geometrically locate modules with respect to each other. Kinematic couplings solve this difficulty in a very elegant way. The introduction of Flexible Kinematic Couplings allows for compliance on the coupling direction, introducing though another feature to the well known kinematic couplings.

Parallel testing of devices is a problem because of the accuracy required between a matrix of chips and a matrix of contactors. The problem could be simplified by breaking the error chain and building the mechanisms such that this error loop (chain) is reduced to the device to contactor level for every single device individually.

4.2 Flexible Kinematic Coupling

There are many systems where two parts mate with a very high degree of planar repeatability, while allowing the part surfaces to come into intimate contact. This is traditionally accomplished with the use of two pins that mate with a circular hole and a slot respectively. The problem with this concept is that there must be some clearance between the pins and the features into which they are inserted. Practically, this results in a repeatability on the order of 20 micrometers.

Kinematic couplings have long been know to be able to provide sub micrometer repeatability by mating three spherical surfaces (e.g., hemispheres), anchored to one body, to three centrally pointing grooves in another body. This provides six points of contact which thus mathematically defines the six degrees of freedom needed to constrain the position and orientation of a body with respect to another body.

Note, however, that this design will not allow the bodies to move towards each other to also make intimate contact between two flat surfaces such as would be required for a lid to seal a vacuum chamber, or for a stencil frame to mate with a solder paste dispensing machine or for utilities (power, signal, air and vacuum) connection between a electromechanical module to the machine frame. What is needed is a method for allowing a kinematic coupling to constrain three degrees of freedom, X, Y, and yaw, while allowing for three degrees of freedom to be unrestrained, Z, pitch and roll.

4.2.1 System requirements

To position and fixture two surfaces with micrometric repeatability and stiffness determined by intimate surface contact, having vacuum as pre-load, it is necessary that the kinematic couplings have a linear displacement with enough length to sustain the weight of the lid without suffering influence of the sealing system (e.g. O-ring) plus the required displacement to squeeze the O-ring for sealing. This type of situation leads to millimetric displacement requirement (1.0 to 3.0 mm) with micrometric repeatability.

So this system would have high repeatability when mounting the two surfaces and also high stiffness because of the intimate contact of the surfaces (area contact). This is possible by the use of a kinematic coupling groove or ball mounted on a linear displacement bearing that has:

- low stiffness on the coupling mounting direction,
- high stiffness on the coupling planar directions (radial stiffness and lateral stiffness),
- low linearity errors (on the order of 1 to 4 μm).

Flexural bearings are ideal for the thought system because they are very predictable and present very good linearity for small displacements (in the elastic region). The linearity is directly associated with the quality of manufacturing the flexures.

4.2.2 Flexural bearing with V-groove

A kinematic coupling V-groove mounted on a flexural bearing is an adequate solution to the problem stated. Flexural bearings rely on the stretching of atomic bonds during elastic deflection to attain smooth motion. For monolithic flexural bearings, the ratio of range of motion to bearing size is on the order of 1/100 to guarantee just elastic deflection. Flexural bearings have high repeatability and excellent resolution. One of the greatest advantages of flexural bearings is its low cost and no need for maintenance.

There are probably hundreds of different ways to design a flexural bearing, but restraining the case to monolithic four bar flexures, five types are analyzed, see Figure 4.1.

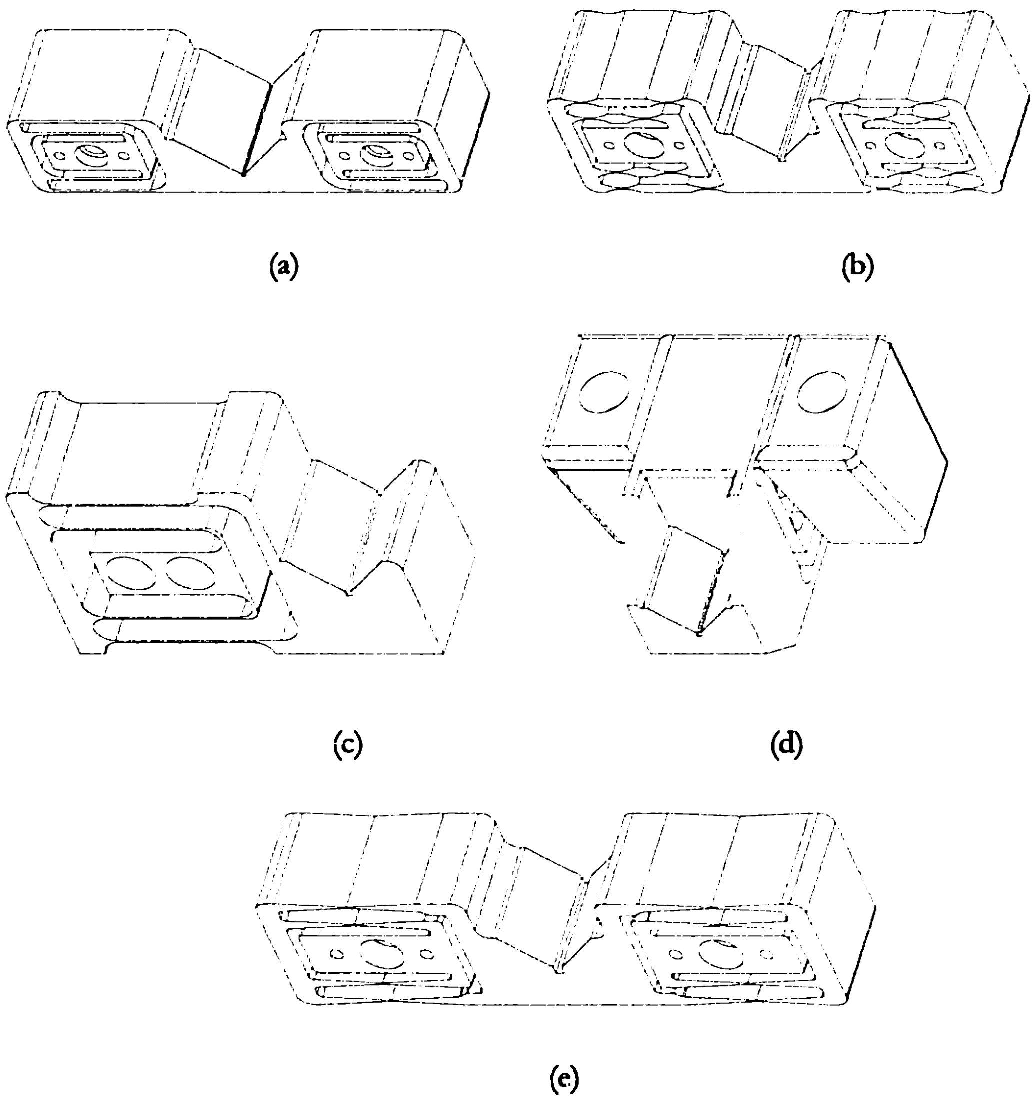
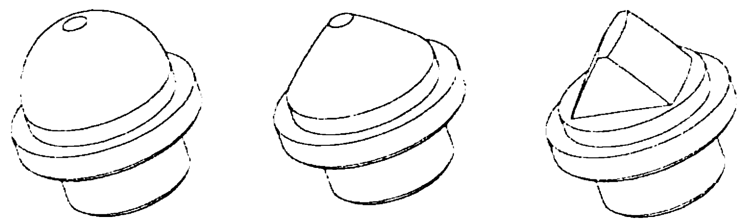


Figure 4. 1 - Types of flexures with V-groove

The suggested solution is a composition of flexural bearing/V-groove unit with a ball type element, as seen if Figure 4.2. These last elements can be: (a) a half sphere, (b) a cone shape or (c) a gothic arc shape on a cone structure; depending on the desired stiffness and error of the kinematic coupling itself.



(a) (b) (c)

Figure 4. 2 - Ball type elements

The system is herein called KinFlex™, stating for the sum of a kinematic coupling with a flexural bearing, see Figure 4.3.

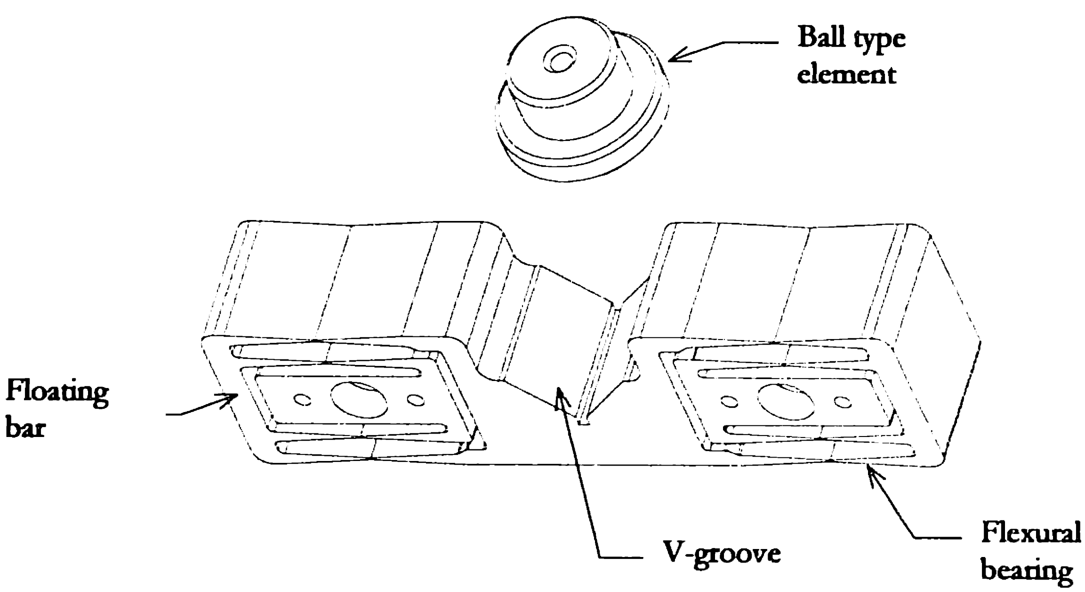


Figure 4. 3 - Taper beam type KinFlex {e.g. (e) of Fig. 4.1}

4.2.3 Flexural bearing general solution

The flexural bearing can be analyzed as four sets of two independent flexures (or two sets of flexures for cases (c) and (d)). This way the force acting on each flexure, F, is one fourth of the preload on each ball (cases (a),(b) and (e)) or one half (cases (c) and (d)). This is true considering that there are four or two reaction forces which are equal because of the symmetry of the structure.

The solution is then worked out independently for each flexure. Making the flexures with the same geometry, the total displacement can be computed as:

$$d_z = 2 \cdot v_{max} \tag{4.1}$$

Where v_{max} is the maximum deflection per flexure. The total deflection d_z is two times the deflection of each flexure because there are two flexures serially connected by the floating bar, which is assumed to be much stiffer than the flexures.

The advantage of using two flexures in series and a floating bar is that each flexure compensate for the consequent lateral displacement caused by the z-axis deflection of the flexures. Without this, each flexure would have an elongation, e, equivalent to:

$$e = L \cdot \left\{ 1 - \cos\left[\arcsin\left(\frac{v_{max}}{L} \right) \right] \right\} \tag{4.2}$$

Where L is the length of the flexure. This would create a tensile stress field proportional to the elongation, which is bigger for increasing values of v_{max}. By using two flexures in series and in opposite direction, each flexure compensates for the elongation of the other, while at the same time increasing two times the total deflection, d_z, for the same stress level.

Each flexure can be modeled as a beam with one fixed end and one end restrained to a slope $\theta = 0$, with no elongation (free on a direction perpendicular to the original displacement d_z), see Figure 4.4.

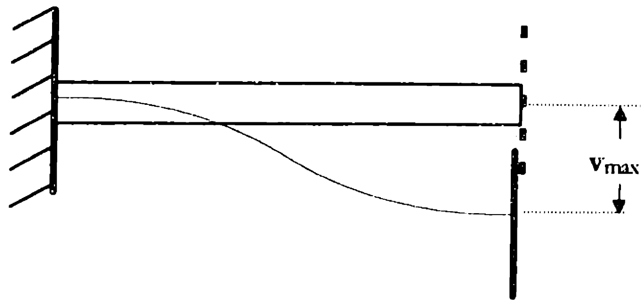


Figure 4. 4 - Flexure deflection diagram

4.2.4 KinFlex type (a) analytical solution

This type, together with types (c) and (d) are probably the easiest forms of the KinFlex to analyze. Calculating the minimum thickness, t_{min}, for the flexure based on the maximum moment, M:

$$t_{\min} = \sqrt{\frac{6 \cdot n \cdot M}{w \cdot \sigma_{yield}}} \tag{4.3}$$

where,

$$M = F \cdot \frac{L_{1st}}{2} \tag{4.4}$$

and,

w - flexure's width

σ_{yield} - yield strength of the material

n - safety factor. This number should come from a S-N diagram where the limiting stress is a function of the number of cycles that the flexure is designed for. For this example n = 2, which means S = 50 % σ_{yield} (common safety factor for low cycle metallic flexures).

Also knowing that:

$$v_{max} = \frac{F \cdot L^3}{12 \cdot E \cdot I} \tag{4.5}$$

This way a first approximation for the length of the flexure can be made by:

$$L_{1st} = \sqrt[3]{\left(\frac{v_{max} \cdot F^{1/2} \cdot E \cdot 6^{1/3}}{w^{1/2} \cdot \sigma_{yield}{}^{3/2}} \right)^2} \tag{4.6}$$

where,

E - Young's modulus of the material

From this first approximation of the flexure's length it is possible to calculate its minimum thickness by using equations *(4.3)* and *(4.4)*. A spreadsheet was written to calculate the flexures and in all cases it is given the designer the ability to set the desired thickness. This way, setting a thickness, $t > t_{min}$, the final length can be calculated as:

$$L = \sqrt[3]{\frac{E \cdot w \cdot t^3 \cdot v_{max}}{F}} \tag{4.7}$$

Total deflection on z-axis is then computed as:

$$d_z = 2 \cdot v_{max} \tag{4.8}$$

Besides being the simplest form of flexures, this solution presents good results not just by giving the desired z-axis compliance but also by allowing for good radial and lateral stiffness on the coupling plane.

4.2.5 KinFlex type (b) analytical solution

KinFlex type (b) analytical equations describe the deflection of two hourglass type beams connected by a rigid straight beam. Displacement is the sum of the hourglass maximum deflection, v_{max} ,and the rigid beam angular offset:

$$d_z = 4 \cdot v_{max} + 2 \cdot L_{one} \cdot \sin(\theta_{max}) \qquad (4.9)$$

where L_{one} is the rigid beam length. Although this seams to be a simple and straight forward solution the mathematics involved in getting to a final applicable solution is not trivial, once the hourglass is constructed to have an initial curvature angle, γ, as shown in Figure 4.5.

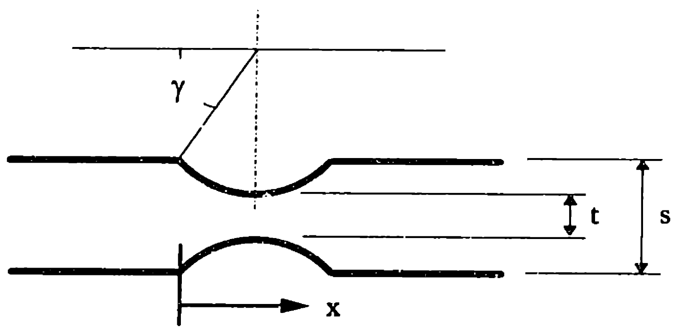


Figure 4. 5 - Hourglass type beam

Assuming no deflection on the rigid beam, its existence is explained by the necessity to amplify the angular motion of the hourglass into a linear z direction displacement while maintaining a good lateral stiffness. From geometry:

$$x = R \cdot (\cos\gamma - \cos\psi) \qquad (4.10)$$

$$dx = R \cdot \sin\psi d\psi \qquad (4.11)$$

The solution is based on:

$$\theta = \int_0^L \frac{M}{E \cdot I} dx \tag{4.12}$$

So, representing I as a function of the angle γ:

$$\gamma = \arcsin\left(1 - \frac{s}{2 \cdot R} + \frac{t}{2 \cdot R}\right) \tag{4.13}$$

$$I(\psi) = \frac{w}{2} \cdot \left[t + 2 \cdot R \cdot (1 - \sin\psi)\right]^3 \tag{4.14}$$

$$\frac{\mathrm{d}v}{\mathrm{d}z} = \theta = \frac{12 \cdot R}{w} \int_{\gamma}^{\pi-\gamma} \frac{M(\psi) \cdot \sin(\psi)}{\left[t + 2 \cdot R \cdot (1 - \sin\psi)\right]^3} \mathrm{d}\psi \tag{4.15}$$

Where $I(\psi)$ is the moment of inertia of the hourglass as a function of the angle, ψ.

This way, the solution of equation *(4.9)* depends strictly on finding v_{max} and θ_{max}. Equations *(4.16)* to *(4.25)* presents the desired solutions (v_{max} and θ_{max})as composition of the integrals, In_i, and the design parameters:

for θ_{max}:

$$\theta = \frac{12 \cdot F \cdot L \cdot R}{2 \cdot w \cdot E} \int_{\gamma}^{\pi-\gamma} \frac{\sin\psi}{\left[t + 2 \cdot R \cdot (1 - \sin\psi)\right]^3} d\psi + \frac{12 \cdot F \cdot R^2}{w \cdot E} \cos\gamma \int_{\gamma}^{\pi-\gamma} \frac{\sin\psi}{\left[t + 2 \cdot R \cdot (1 - \sin\psi)\right]^3} d\psi + \frac{12 \cdot F \cdot R^2}{w \cdot E} \int_{\gamma}^{\pi-\gamma} \frac{\sin\psi \cdot \cos\psi}{\left[t + 2 \cdot R \cdot (1 - \sin\psi)\right]^3} d\psi$$

(4.16)

$$\text{In}_1 = \int_{\gamma}^{\pi-\gamma} \frac{\sin\psi}{\left[t+2\cdot R-2\cdot R\cdot\sin\psi\right]^3}\,d\psi = \frac{1}{2\cdot t^2+8\cdot t\cdot R}\cdot\left\{A+B+C\cdot\left[D-E\right]\right\}$$

(4.17)

with,

$$A = \frac{(2\cdot t+4\cdot R)\cdot\cos\gamma}{\left(t+2\cdot R-2\cdot R\cdot\sin\gamma\right)^2}$$

$$B = \frac{\left(2\cdot t^2+8\cdot t\cdot R+24\cdot R^2\right)\cdot\cos\gamma}{\left(t^2+4\cdot t\cdot R\right)\cdot\left(t+2\cdot R-2\cdot R\cdot\sin\gamma\right)}$$

$$C = \frac{12\cdot t\cdot R+24\cdot R^2}{\left(t^2+4\cdot t\cdot R\right)\cdot\sqrt{t^2+4\cdot t\cdot R}}$$

$$D = \arctan\left[\frac{(t+2\cdot R)\cdot\tan\left(\frac{\pi-\gamma}{2}\right)-2\cdot R}{\sqrt{t^2+4\cdot t\cdot R}}\right]$$

$$E = \arctan\left[\frac{(t+2\cdot R)\cdot\tan\left(\frac{\gamma}{2}\right)-2\cdot R}{\sqrt{t^2+4\cdot t\cdot R}}\right]$$

and,

$$\text{In}_2 = \int_{\gamma}^{\pi-\gamma} \frac{\sin\psi\cdot\cos\psi}{\left[t+2\cdot R-2\cdot R\cdot\sin\psi\right]^3}\,d\psi = 0 \qquad (4.18)$$

so,

$$\theta_{max} = \frac{12 \cdot R}{w \cdot E} \cdot \left(F \cdot R \cdot \cos\gamma + \frac{F \cdot L_{one}}{2} \right) \cdot \mathrm{In}_1 \tag{4.19}$$

for v_{max}:

$$\mathrm{In}_3 = \int \frac{\cos\psi}{\left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right)^2} d\psi = \frac{1}{2 \cdot R \cdot \left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right)} \tag{4.20}$$

$$\mathrm{In}_4 = \int \frac{\cos\psi}{\left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right)} d\psi = -\frac{1}{2 \cdot R} \cdot \mathrm{Ln}\left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right) \tag{4.21}$$

$$\mathrm{In}_5 = \int \arctan\left[\frac{(t + 2 \cdot R) \cdot \tan\frac{\psi}{2} - 2 \cdot R}{\sqrt{t^2 + 4 \cdot t \cdot R}} \right] d\psi$$

$$\mathrm{In}_5 = \psi \arctan\left[\mathrm{H} \cdot \tan\left(\frac{\psi}{2}\right) + \mathrm{J} \right] - \mathrm{H} \int \frac{\psi}{1 + \mathrm{H}^2 + \mathrm{J}^2 + \cos\psi + 2 \cdot \mathrm{H} \cdot \mathrm{J} \cdot \sin\psi} \mathrm{d}\psi \tag{4.22}$$

$$\mathrm{In}_6 = \int \frac{1}{\left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right)^2} d\psi$$

$$\mathrm{In}_6 = \frac{1}{t^2 + 4 \cdot T \cdot R} \cdot \left\{ \frac{-2 \cdot R \cdot \cos\psi}{\left(\left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right)\right.} + 2 \cdot \mathrm{H} \cdot \arctan\left[\mathrm{H} \cdot \tan\left(\frac{\psi}{2}\right) + \mathrm{J} \right] \right\} \tag{4.23}$$

$$\mathrm{In}_7 = \int \frac{1}{\left(t + 2 \cdot R - 2 \cdot R \cdot \sin\psi\right)} d\psi = \frac{2}{\sqrt{t^2 + 4 \cdot t \cdot R}} \cdot \arctan\left[\mathrm{H} \cdot \tan\left(\frac{\psi}{2}\right) + \mathrm{J} \right] \tag{4.24}$$

with,

$$H = \frac{t + 2 \cdot R}{\sqrt{t^2 + 4 \cdot t \cdot R}}$$

$$J = -\frac{2 \cdot R}{\sqrt{t^2 + 4 \cdot t \cdot R}}$$

say,

$$P_1 = \frac{1}{2 \cdot (t^2 + 4 \cdot t \cdot R)} \cdot \left\{ (2 \cdot R - t) \cdot \mathrm{In}_3 + \frac{4 \cdot R}{(t^2 + 4 \cdot t \cdot R)} \cdot \left[-2 \cdot R \cdot \mathrm{In}_4 + \frac{2 \cdot (t + 2 \cdot R)}{\sqrt{t^2 + 4 \cdot t \cdot R}} \cdot \mathrm{In}_5 \right] \right\}$$

$$P_2 = \frac{(t + 2 \cdot R)}{2 \cdot (t^2 + 4 \cdot R)^2} \cdot \left[(-t + 2 \cdot R) \cdot \mathrm{In}_4 + \frac{4 \cdot R}{\sqrt{t^2 + 4 \cdot t \cdot R}} \cdot \mathrm{In}_5 \right]$$

$$P_3 = \frac{(t + 2 \cdot R)}{4 \cdot R^2} \cdot \mathrm{In}_6 - \frac{1}{4 \cdot R^2} \cdot \mathrm{In}_7$$

$$v_{max} = \frac{12 \cdot R}{w \cdot E} \cdot \left(F \cdot R \cdot \cos\gamma + \frac{F \cdot L_{one}}{2} + F \cdot R \right) \cdot (P_1 + P_2 + P_3)_{\pi}^{\pi - \gamma} \tag{4.25}$$

This solution is relatively complicated to implement, depending on the design parameter that will be calculated as a function of the geometrical constrains. The main problem observed with this solution is that the hourglass causes a stress concentration, because the deformation is not distributed along the flexure, as in the previous case, but rather it is concentrated on the hourglass.

4.2.6 KinFlex types (c) and (d) analytical solution

There should be no surprise to observe that solutions (c) and (d) are the same as solution (a), except by the fact that the force per flexure is now half of the force applied on each ball-groove kinematic coupling, and not one fourth as in that case.

The radial and lateral stiffness on the plane of coupling of these flexures are smaller than that of type (a). The only possible advantage of this solution is on cases where there is no possibility to accommodate the length of the flexure-groove unit on the mechanical system.

4.2.7 KinFlex type (e) analytical solution

A moment diagram of the flexures indicates that the moment reaches its maximum value at both ends of the flexure and zero at the center. From equation *(3)*, where thickness is determined by a moment restriction, it is demonstrated that there is no need for thickness at the center of the flexures. This is not quite true because of the shear force and also the requirement of good lateral and radial stiffness on the plane of coupling.

This investigation suggests that instead of using two hourglass per flexure, which causes a reduction in thickness exactly where the thickness should be maximum, it should be interesting to assume the flexure as one entire hourglass. This would eliminate the calculation for θ_{max} and with modifications on the equations for v_{max} the solution could be easily obtained. It is not very practical to manufacture an hourglass like this because of its big radius. Straight lines, forming a V should do the same function in this case, with the advantages of easy of manufacturing and easy of calculation.

It is not too much to remind that:

$$d_z = 4 \cdot v_{\max} \tag{4.26}$$

and for this flexure:

$$v_{\max} = \frac{12 \cdot F}{w \cdot E \cdot K^2} \cdot \left[-\frac{1}{K} \cdot Ln(s) - \frac{1}{2 \cdot K} + \frac{1}{K} \cdot Ln(t) + \frac{s}{2 \cdot K \cdot t} - \frac{L}{2 \cdot t} + \frac{s \cdot L}{4 \cdot t^2} \right] \tag{4.27}$$

with,

$$K = \frac{2}{L} \cdot (t - s)$$

where:

L - length of the flexure

s - thickness at the center of the flexure

Equation *(4.27)* should be initially evaluated substituting t by equation *(4.3)*. With the first result of length, t_{min} can be evaluated, which is an important geometric constrain to the design. Choosing a thickness, $t > t_{min}$ for the design, a new value of length is calculated and used for the construction of the flexures.

The designer have to understand that reducing the thickness at the center of the flexure, s, will cause a reduction of the lateral stiffness but also promote a reduction of the length of the flexures, which increases the radial stiffness. There is though a trade off on how much the ratio of radial to lateral stiffness is desired on the project.

4.2.8 Design example

A vacuum chamber lid is designed with an array of optical sensors for a wafer processing tool. The lid is open and closed for maintenance, and the time required to re-calibrate the sensors implies in several thousands of dollars. Using KinFlex, high repeatability is achieved on repositioning of the lid on the chamber, so eliminating the re-calibration time of the sensors. Sealing is achieved by moving the lid towards the face of the chamber, squezing an Oring and holding the lid against the chamber's top surface, see Figure 4.6.

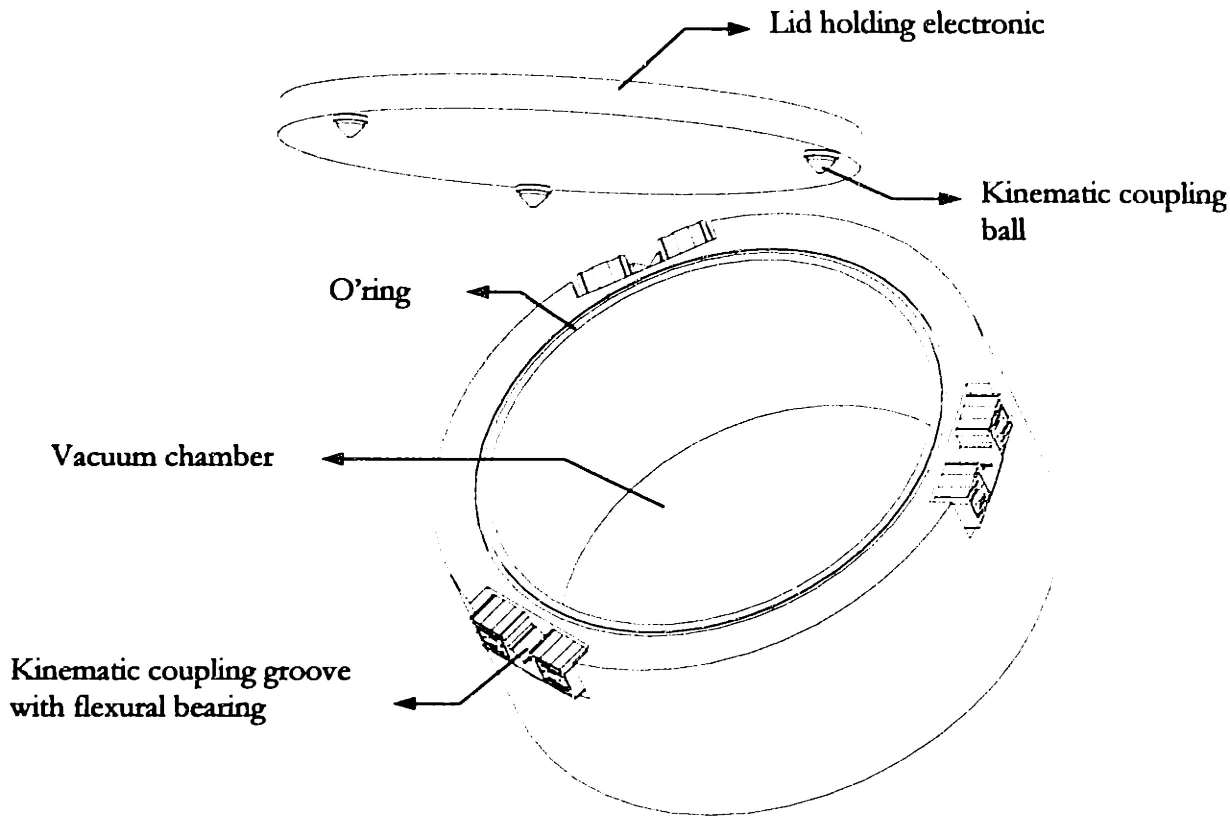


Figure 4. 6 - Vacuum chamber with KinFlex

Considering the O'ring thickness, a 1.5 mm displacement on the flexural bearings is enough to achieve the desired positioning and final linear displacement to seal the chamber with the lid. Also, the electronics on the lid will be accurately repositioned to their previous location. Utilizing the analysis implemented in a user friendly spreadsheet, one can quickly obtain the set of design parameters presented on Figure 4.7.

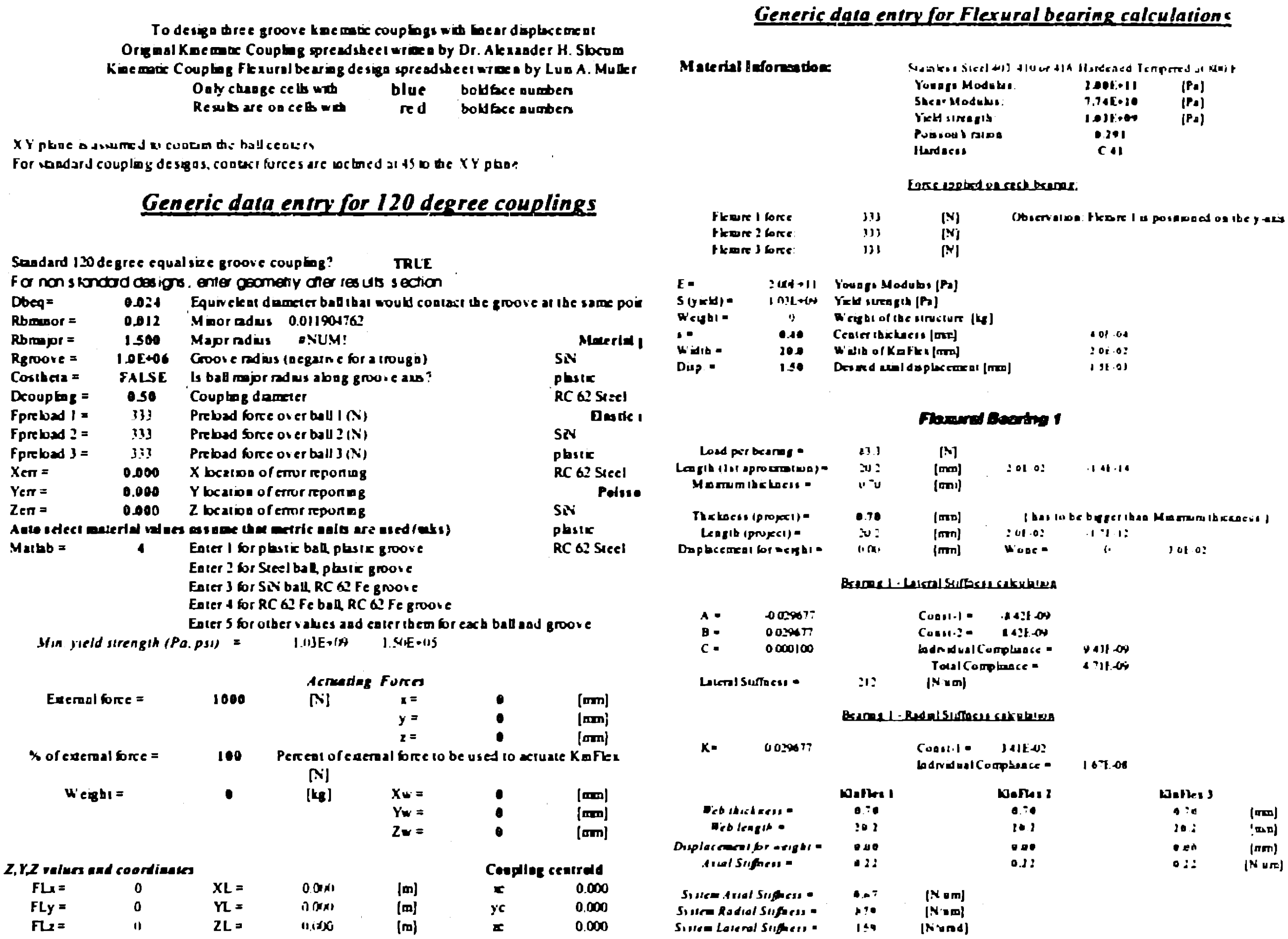


Figure 4. 7 - Tapered beam analytical solution

The finite element analysis done on the same design, presented on Figure 4.8, results in a 3% discrepancy with the analytical solution previously described, which is an acceptable error for flexural bearing design.

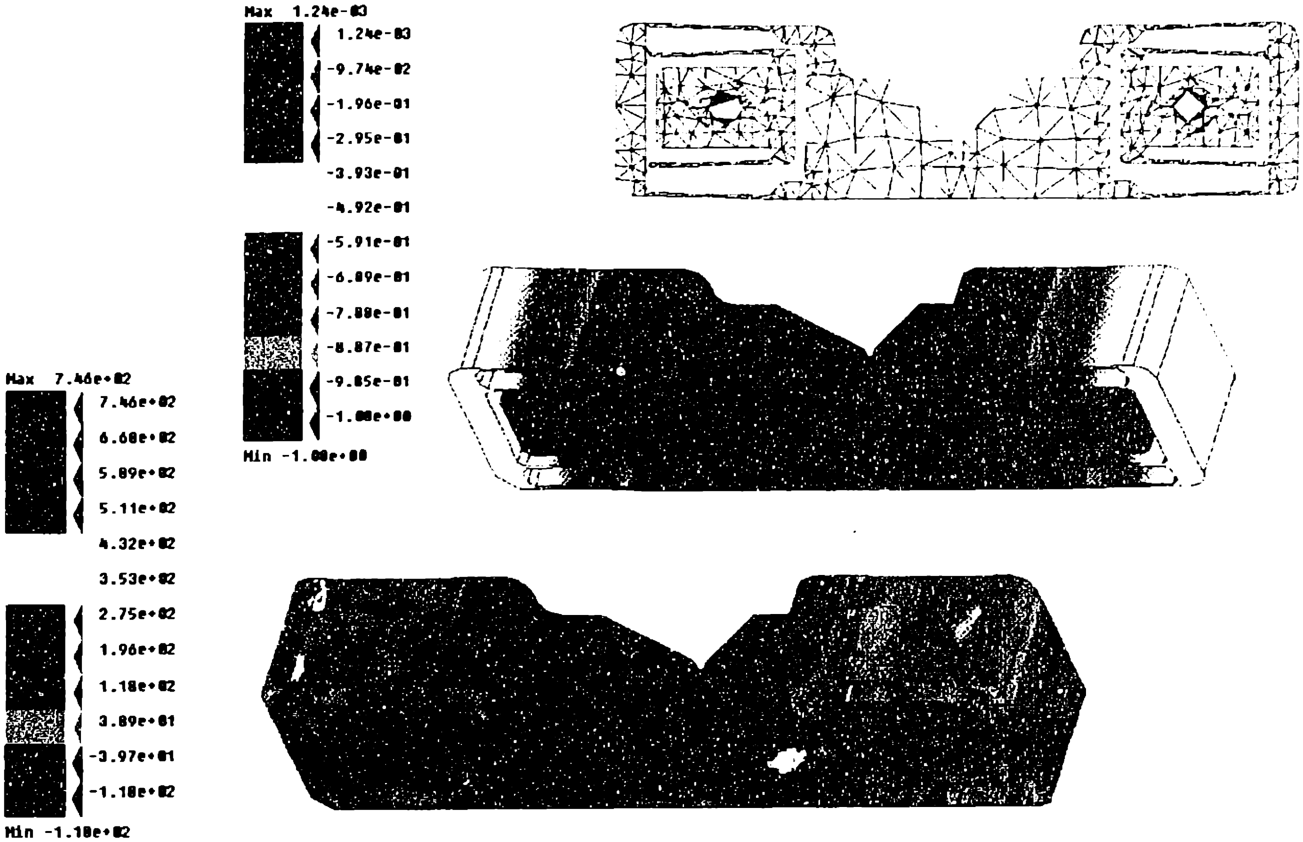


Figure 4. 8 - Finite element analysis on KinFlex

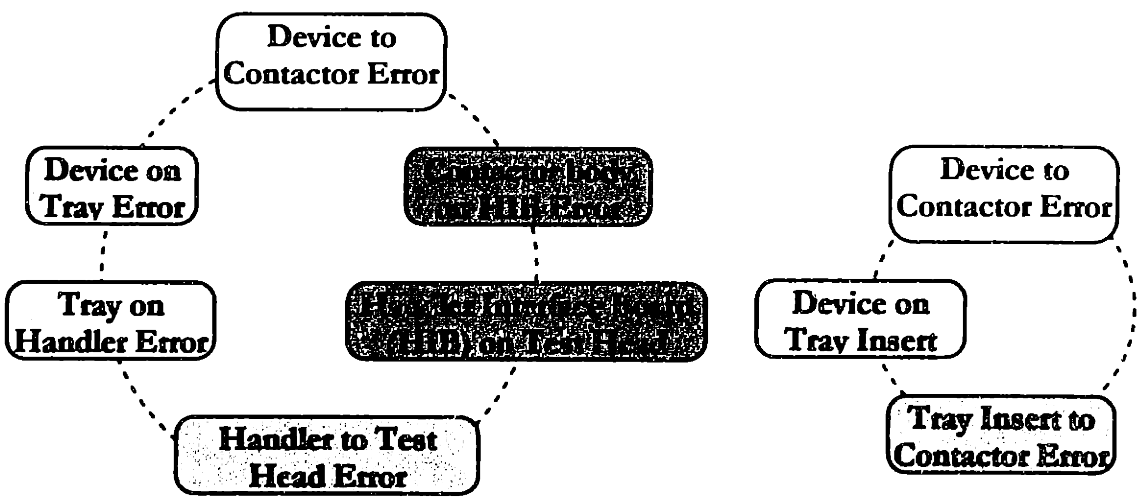
4.2.9 Conclusion

The KinFlex analytical solutions proved to be within less then 8% discrepancy with the finite element analysis of the flexures. Specifically, solution type (e) with tapered flexures proved to be advantageous by providing close to constant strain through the entire beam. The tapered beam solution maximizes deflection with minimum beam length.

KinFlex proved to be an easy solution to the problem of accurately locating a module or vacuum chamber lid to the mating part allowing for millimeter displacement on coupling direction and micrometer repeatability.

4.3 Flexible Site Chip Tray, FlexTray™

The semiconductor test industry is moving towards parallel testing of devices. There is an inherent mechanical difficulty in testing several devices at the same time, it is extremely difficult to control the location of each chip with respect to alignment features to a Handler Interface Board (HIB). The location of a matrix of packages to the contactors located at the Handler Interface Board (HIB) relies on the accuracy of a long mechanical chain (error loop), shown in Figure 4.9. The Error-loop between device and contactor can be relatively big if each device is located to a gripper and the gripper then located to a structure which also locates the HIB with its contactors, as shown schematically on Figure 4.9-A. A much smaller Error-loop can be obtained by locating each device to its contactor by allowing enough individual locating features, as shown in Figure 4.9-B. The bigger the error loop the more difficult it will be to reliably perform the alignment. The FlexTray concept allows a package to "float" in six degrees-of-freedom when presented to the contactor while some features on both provide the necessary alignment of each individual device.



A. Without FlexTray **B.** With Flextray

Figure 4. 9 - Test Error Budgeting

4.3.1 System Requirements

An engineered tray that conforms to JEDEC standards for tray manufacturing and achieve alignment requirements for in-tray testing of many devices in parallel (first design for 32 memory devices in parallel test). Parallel testing of devices is being the goal for most of the packages handled by the major manufacturers. One of the biggest limitations for parallel testing is the capability of accurate locating each package relative to its contactor. Memory chips have the highest parallelism in the market today, up to 32 devices simultaneously being tested.

A potential solution would allow individual location of each device to its contactor, so alignment features on a complaint tray would ideally perform the required function. Another important issue is that this new tray has to comply with the JEDEC standards that determine general tray outline design for handling purposes. This new tray would cycle through the handler, changing the paradigm from testing devices to testing trays (devices never leave the trays). These trays would have to survive the temperatures inside the handler, which can vary from - 60 Celsius to 160 Celsius.

Existing chip trays have no engineering requirements, they merely support devices at ambient temperature that are carried inside a semiconductor manufacturing facility and/or are shipped to the customer.

4.3.2 Proposed Solution

FlexTray is a 6 degree-of-freedom compliant insert for holding chips in a JEDEC Solid State Product Outline for Thin Matrix Tray, CS-005. The tray incorporates FlexSert flexurally compliant inserts to individually support each device under test (DUT).

The FlexSerts each support a DUT, in some cases (such as the 44 lead TSOPII) the insert will support both formed and unformed parts, though not simultaneously. The degrees of freedom are obtained by means of flexural bearings connecting a surrounding structure to a middle structure that supports the chip, as shown on Figure 4.10.

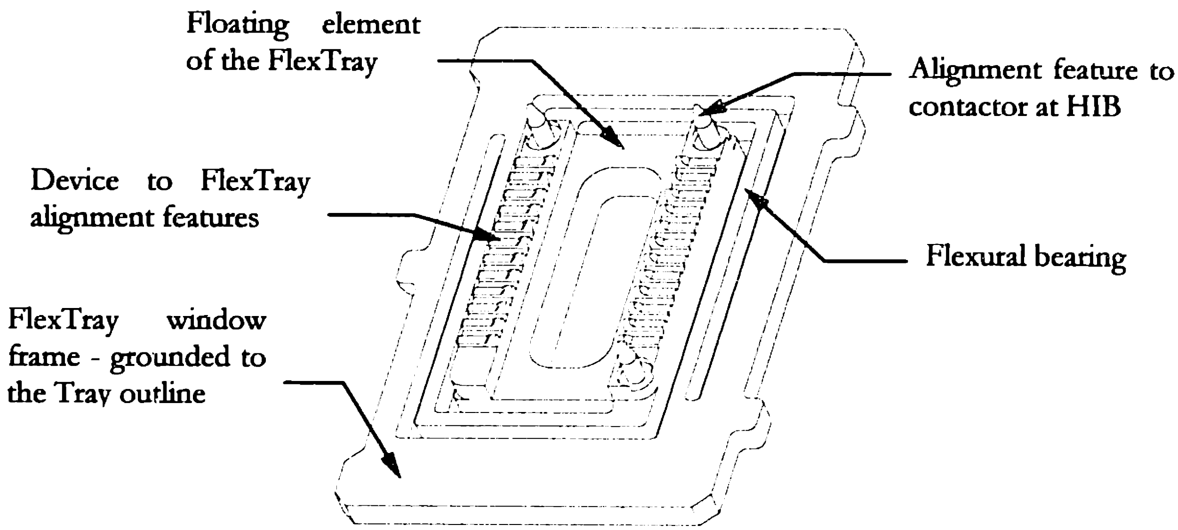


Figure 4. 10 - FlexTray concept

The insert holds the DUT and provides self centering compliance to accommodate location errors. The centering capability is exploited by the handler to permit proper contactor/lead alignment in a low cost (not precision machined) tray.

The inserts are device specific, may accommodate a variety of configurations in each tray frame pattern and may include such features as device retention, flippability etc., as needed, while being economic to manufacture. The back of the insert has drafted and chamfered features which can be used for coarse alignment from behind, it also has kinematic grooves for a KinFlex equipped pusher.

4.3.3 Solution Variations

A FlexTray concept for a PLCC device (which has leads at all four sides of the package) is presented on Figure 4.11. This package is supported by a small tab on its four corners and held in place with a positive retention mechanism which also self centers the device to the floating element of the FlexTray.

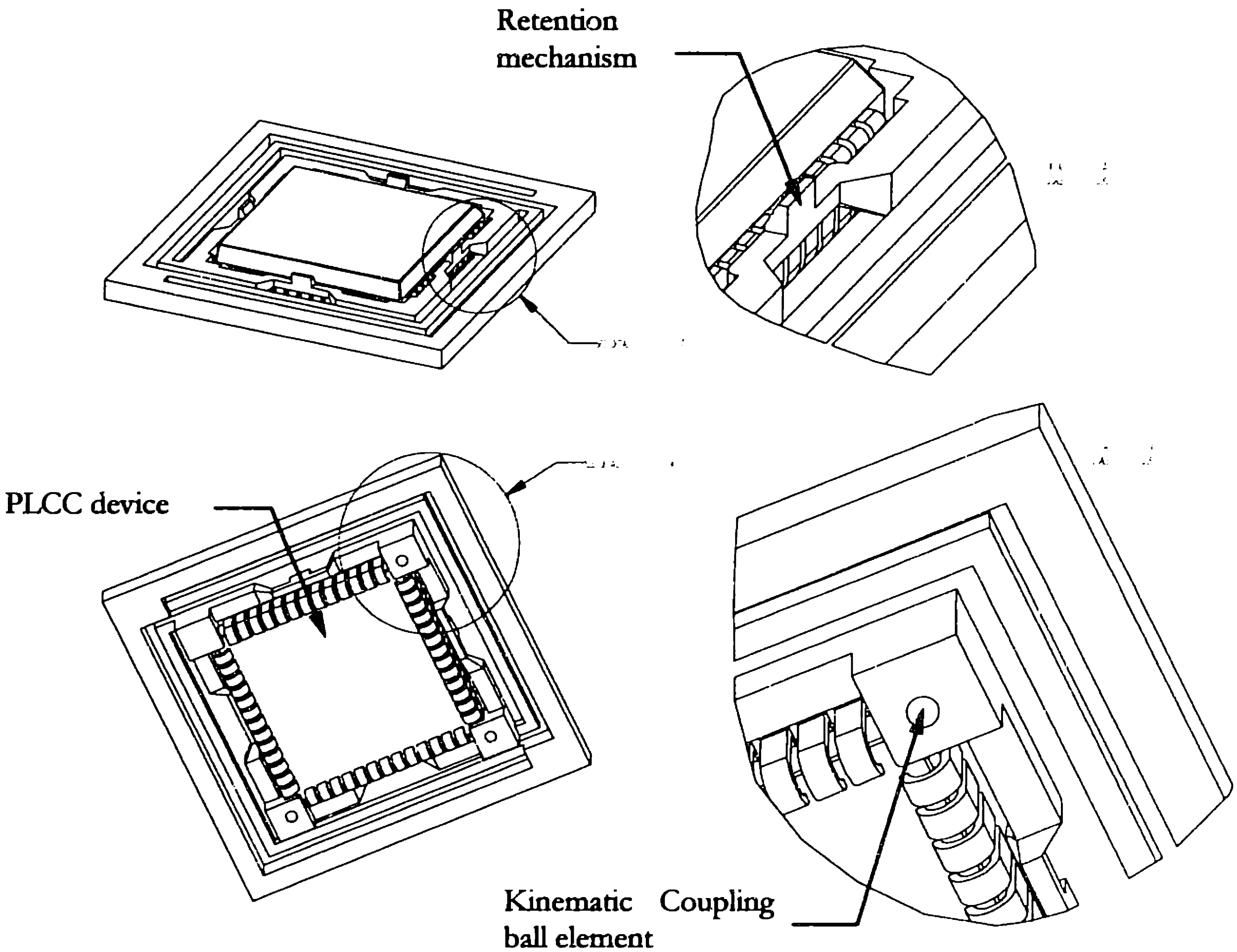
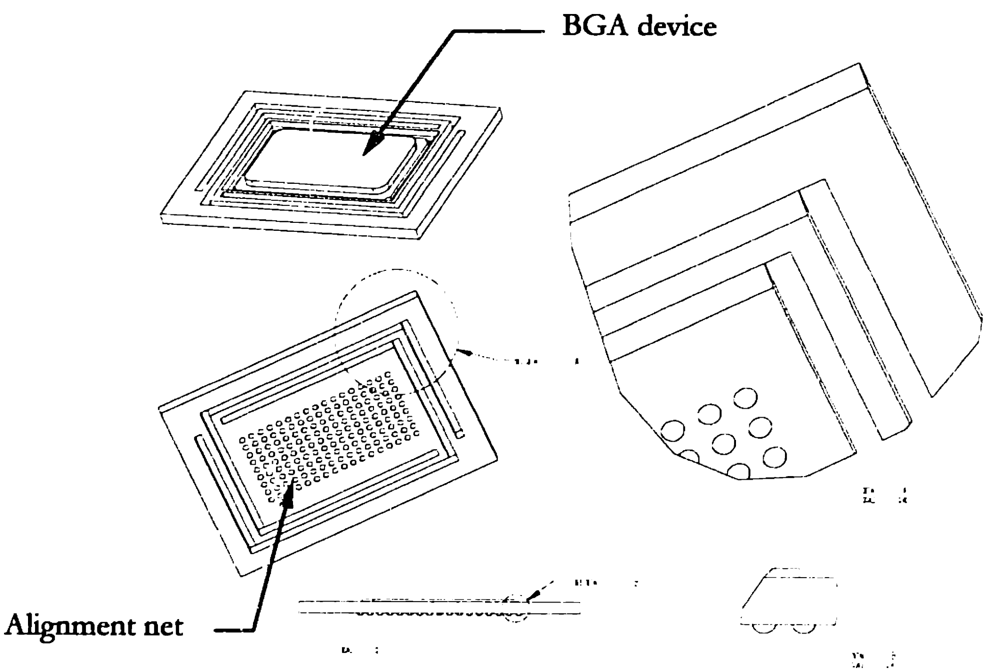
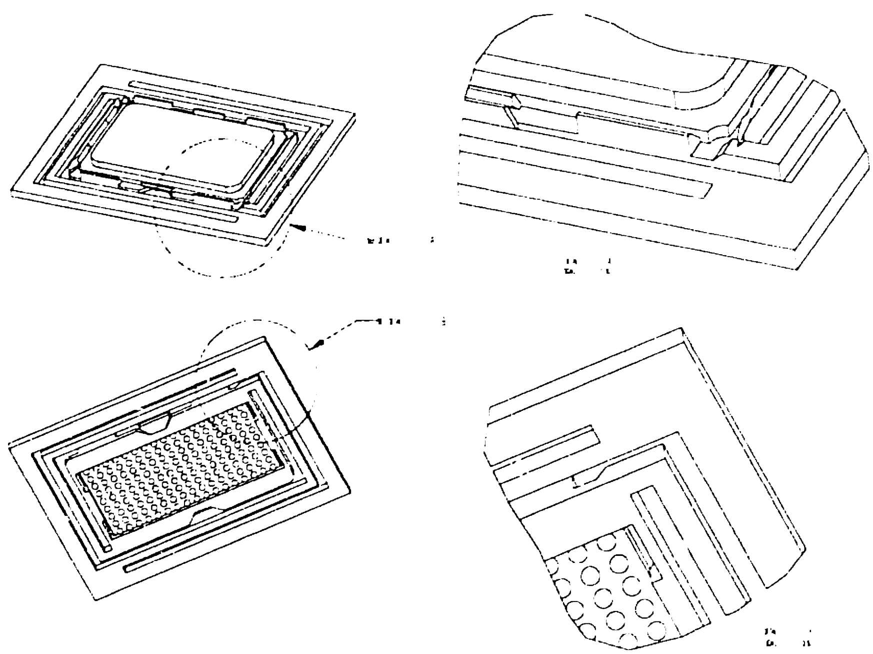


Figure 4. 11 - PLCC type FlexTray

Figure 4.12 presents two concepts for BGA devices. The first one allows location of the package on the floating element of the FlexTray by nesting the contact balls through a net. The second concept holds the package by two edges and references it by the plastic body.



Concept A



Concept B

Figure 4. 12 - BGA type FlexTray

4.3.4 Flexural Bearing Analytical Solution

There are different ways to design the flexures to minimize stress and/or maximize deflection. The straight beams design approach provides very good results for this application. Consider the "L" shape beam from figure 4.13.

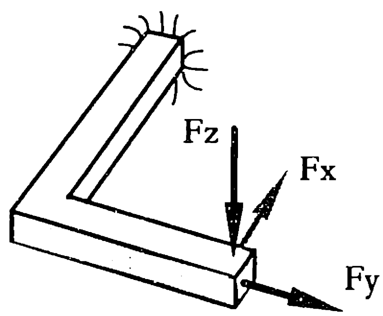


Figure 4. 13 - "L" shape flexure for FlexTray design

Based on the coordinate system shown in this figure, the compliance can be derived from solid mechanics equations as:

- Compliance due to force acting on X-axis:

$$\frac{1}{k_x} = \frac{\frac{F_x}{EI}\left[\frac{l_2^{\,3}}{3} - \frac{l_2^{\,4}}{4(l_1 + l_2)}\right]}{F_x} \tag{4.28}$$

- Compliance due to force acting on Y-axis:

$$\frac{1}{k_y} = \frac{\frac{F_x}{EI}\left[\frac{l_1^{\,3}}{3} - \frac{l_1^{\,4}}{4(l_1 + l_2)}\right]}{F_y} \tag{4.29}$$

- Compliance due to force acting on Z-axis:

$$\frac{1}{k_z} = \frac{\frac{F_z l_2{}^3}{EI}\left[\frac{1}{3} - \frac{l_2}{4EI\left(\frac{l_1}{Gbh^3C} + \frac{l_2}{EI}\right)}\right]}{F_z} \tag{4.30}$$

and,

$$C = 0.031 + 0.11\left(\frac{b}{h}\right)$$

where,

l_1, $l2$ - are beam lengths represented on Figure 3.4;

Fx, Fy, Fz - are the forces on each respective axis shown on Figure 3.4;

b - base length of the rectangular beam cross-section;

h - high length of the rectangular beam cross-section;

C - Torsion coefficient for rectangular shafts.

Utilizing the analysis implemented in a user friendly spreadsheet, one can quickly obtain the set of design parameters presented, see Figure 4.14 for part of the referred spreadsheet, where only the X-axis displacements are analyzed.

FLEXTRAY

To design a compliant chip tray with hourglass type flexures
Hourglass Flexural bearing design spreadsheet written by Luis A. Muller

Only change cells with	**blue**	boldface numbers
Results are on cells with	**red**	boldface numbers

MATERIAL PROPERTIES

Material:	**Policarbonate**	
Modulus of Elasticity =	**3.00E+05**	
Yield Strength =	**7.00E+03**	
Number of stress cycles desired =	**1.00E+06**	Life
Fatigue Strength =	**5.00E+03**	For rotating-beam specimen, from S-N diagram
Surface factor =	**0.90**	
Size factor =	**1.00**	
Reliability factor =	**0.81**	
Temperature factor =	**0.80**	
Design Fatigue Strength =	**2.92E+03**	For the flexure

BOUNDARY CONDITION

Applied Force =	**0.050**	[lbf]	Force applied

X-AXIS COMPLIANCE

Flex thick. =	**0.040**	[in]	Thickness of rigid flex beam
Length =	**0.505**	[in]	Lenght of rigid flex beam
Width =	**0.070**	[in]	Width of rigid flex beam
Hourglass radius =	**0.000**	[in]	Radius of the hourglass
Hourglass Design thickness =	**0.000**	[in]	Designed hourglass thickness
Hourglass Min. thickness =	**0.019**	[in]	Minimum thickness of hourglass
Hourglass Gama =	**0.00**	[degrees]	Angle of hourglass start point
Theta =	**0.000**	[degrees]	Maximum angular deflection
X-axis Displacement =	**6.99E-03**	[in]	Maximum total deflection
Reflected y-displacement =	**8.16E-03**	[in]	
X FlexInsert Stiffness =	**7.15E+00**	[lbf/in]	
X FlexInsert Compliance =	**1.40E-01**	[in/lbf]	
Hourglass Stress =	**0.00E+00**	[psi]	
Beam Stress =	**6.76E+02**	[psi]	*TRUE*

Figure 4. 14 - Straight-beam type flexural bearing

The finite element analysis done on the same design, presented on Figure 4.10, results in a 3.3 % discrepancy with the analytical solution previously described, which is an acceptable error for flexural bearing design, shown on Figure 4.15.

Analytical Model Evaluation

Width =	0.04	in	Thickness=	0.07	in			
X-beam =	0.505	in	Y-beam =	0.895	in			
Force [lbf]	X-Deflection [in]			Y-Reflected Deflection [in]			Stress [psi]		
	Analysis	FEA	Error (%)	Analysis	FEA	Error (%)	Analysis	FEA	Error (%)
0.05	6.99E-03	7.09E-03	1.4	8.16E-03	8.39E-03	2.7	6.76E+02	6.68E+02	1.2
0.10	1.40E-02	1.42E-02	1.4	1.63E-02	1.64E-02	0.6	1.35E+03	1.34E+03	0.7
0.15	2.10E-02	2.13E-02	1.4	2.45E-02	2.51E-02	2.4	2.03E+03	2.01E+03	1.0
0.20	2.80E-02	2.83E-02	1.1	3.26E-02	3.37E-02	3.3	2.71E+03	2.67E+03	1.5

Figure 4. 15 - Finite element analysis on FlexTray

The behavior of the flexural beam for different ratios of width to thickness of the flexure and for different Elastic Modulus is presented on Figure 4.16.

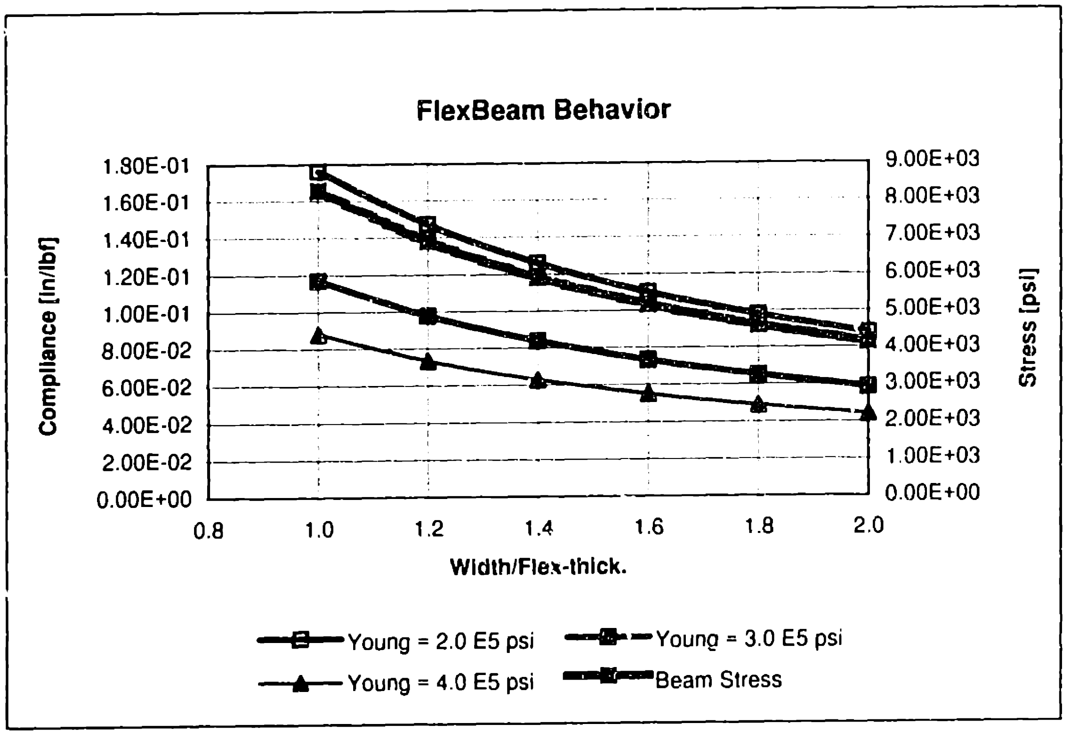


Figure 4. 16 - Flexural beam behavior

When designing a flexure one will probably want to minimize maximum stresses and obtain the desired compliance with the smallest possible flexure.

Chapter 5

DESIGN OF THE APOLLO SORTER

5.1 Introduction

The Apollo Sorter is a project developed with support from Kinetrix, Inc. This machine serves as the case study for the modular machine design framework proposed on chapter 3. This machine, together with a tray based handler, described in chapter 2, are the equipment necessary to achieve the first step towards a proposed back-end automation.

The Apollo is a dedicated sorting machine that takes as input either thin or thick, identifiable by bar code, JEDEC trays containing packages, and sorts the contents into 5, 7 or 9 different categories without tray mapping; or up to 256 categories if doing tray mapping. The configuration build is defined specifically for each customer (machine models KMX 516, KMX 716 and KMX 916; meaning KMX ABC: A-number of Output Bins, B-number of Grippers working in parallel, C-number of sites per Gripper. The Apollo Sorter project configured is a KMX 716 type machine.

The Apollo Sorter is the first machine of a family. The same Frame, potentially the same Gantry and Gripper could be used with different modules in place of Tray Modules to perform different functions. This constitutes the principle of modularity on the machine design that allows the same basic machine to perform different functions and adapt function and morphology to more adequately balance a back-end production line.

Semiconductor manufacturers do not need to buy spare machines for serviceability, they can swap modules instead (Tray Modules, Grippers and Gantries). Also semiconductor manufacturers do not need to buy different machines for performing different functions on the production floor, they do buy different modules to change function and morphology of an existing machine. For example:

- Tape & Reel Loader Module could be used to sort chips into tape;

- A Tube Loader Module would allow shipping the devices in plastic tubes;

- A Lead Inspection Module would allow for scanning, coplanarity and lead inspection on devices prior to shipment;

- An Expanding Gripper module allows transport of devices from a tray of pitch r to a different tray of pitch q with a gripper that adapts to in X and Y coordinates to each pitch. This allows extremely fast pick & place as the high parallelism of the gripper is maintained for both types of trays. For example: a six sites Expanding Gripper would be able to pick up six devices at once on a 32-site automation tray (pitch between devices r) used on the tray-based handler, this Expanding Gripper also allows the same six devices to be placed in one step on a shipping tray with 220 devices (pitch between devices q).

The first Apollo machine derives from the goal of developing a machine for sorting devices between trays of the same pitch. This machine have provisions for other modules, saved restrictions on dimensions, size of data lines (e.g.: number of bits on device net), drivers, amplifiers, pneumatic restrictions. In general the restrictions are based on the type and quantity of power, signal and material that has to flow through the control volume of each module.

5.2 Design Specifications

The Sorter is designed for a set of specifications determined by customer interviews and customer buy specs. The following lists the design specifications:

Physical Dimensions: Width = 1651 mm Depth = 1422 mm Height = 1270 mm	**Clearances:** Front panel opening = 762 mm Rear panel opening = 500 mm
Weight: Total weight = 450 kg	**Power Requirements:** 208 VAC single phase, 50-60 Hz 30 Amps
Compressed Air: Input Air 100psi, regulated to 50 psi	**Package Type:** SOJ .300, .400, 16 mm, Type I TSOP Type II TSOP QFP, SOP, TQFP, PQFP, and BGA
Tray Module Capacity: Primary Stack (input): 1 @ 50 trays Primary Stack (output): 3 @ 50 trays Primary Stack (empty tray buffer): 1 @ 50 trays Secondary Stack (output): 4 @ 5 trays Secondary Stack (back fill buffer): 1 @ 5 trays	**Reliability:** 1 Jam per 3000 picks 1 Jam per 1000 trays MTBF: 1000 hours MTTR: < 30 minutes MTBA: > 2 hours
Throughput: 10,000 uph (>96% Yield on 32 device/tray) 3,600 uph (media transfer only mode)	**Software:** Windows NT platform Interface GEM SECS
Communications: RS-232 IEEE-488 Ethernet RJ-45 Ethernet RJ-12	**Safety Standards:** Meets or exceeds requirements specified in: SEMI S2-93 SEMI S8-95 EN60204-1 1993
Software: Windows NT platform Interface GEM SECS	**Service Life:** 5-7 years

Figure 5. 1 - Apollo Sorter Specifications

5.3 Conceptual Design of the Machine

5.3.1 The Back Fill Methodology

Test results usually present a very high percentage of devices classified in one category for output (test yield usually above 95%). The number of different categories needed is dependent on each semiconductor manufacturer and device under test, but it usually goes from two to sixteen.

To reduce device movement and provide maximum throughput, only bad devices are sorted from the input tray. The input tray is back filled with known good devices (e.g. Bin 1) located in the Back Fill Buffer Tray. The replaced device is sorted to another category other than bin 1, using 32 site trays. This procedure is specially useful when test yields are high (e.g. >95% test yield). One possible sorter logic is shown in figure 5.2:

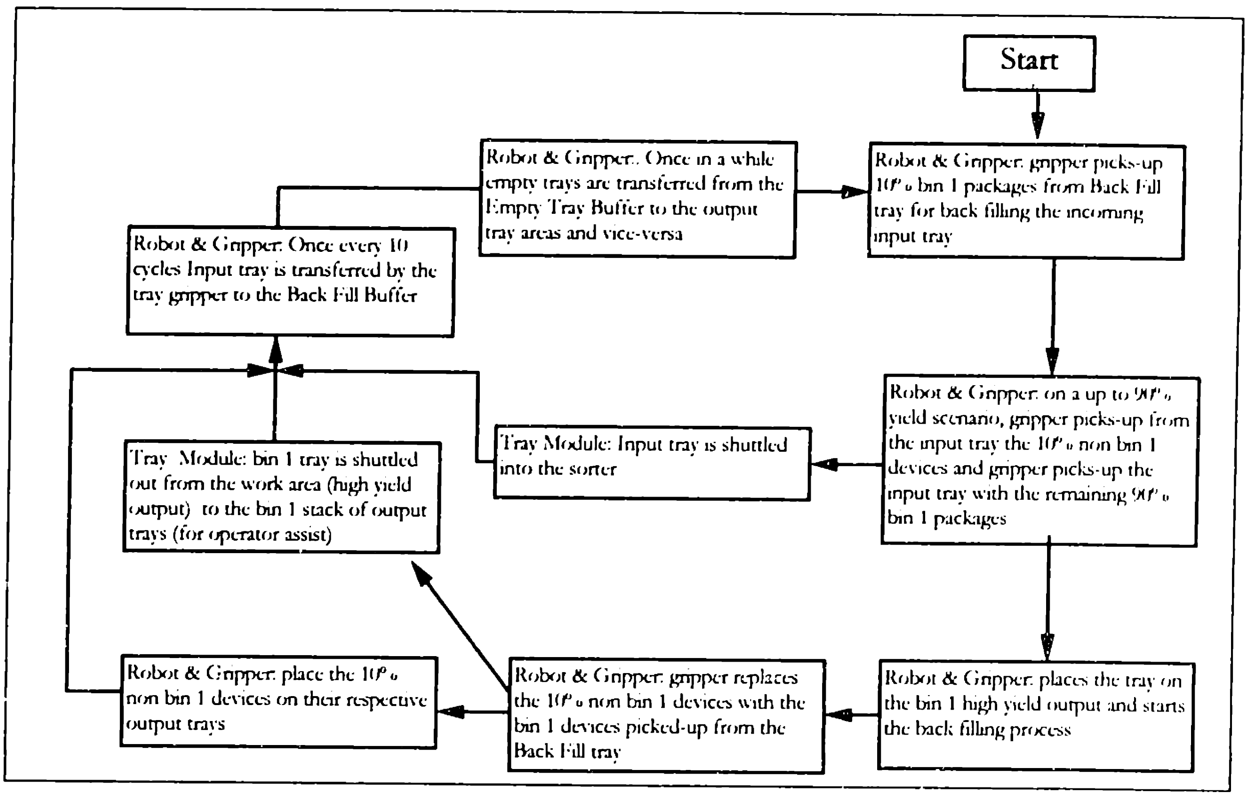


Figure 5. 2 - Functional Description of the Apollo Sorter

Based on this logic a set of concepts were generated for the Sorter project. The processed included an Analytical Hierarchy Process (AHP) that helped select and optimize the concept pursued as the Apollo. The steps of generating, selecting and optimizing concepts are well known by machine design engineers and there are several authors that describe it.

5.3.2 Functional Description of the Apollo Sorter

The Apollo is a dedicated sorting machine. Thin or thick (identified by bar code) JEDEC outline trays containing devices are input to the sorter, and the contents are sorted into 7 different tray positions.

The Sorter has a PC-based computer using the Windows NT operating system and Pentium processor. A separate CD ROM and Floppy Drive are installed in the Control Cabinet accessible through an access panel in the front of the unit. The input data is transferred to the sorter via a TCP/IP network connection in real time using a database. The output data is then transferred from the sorter via the same network connection.

The Sorter is composed of a Gantry which moves devices and trays around the work area using a Gripper head. Trays are transferred between the work area (process chamber) and the operator assist area by way of Tray Modules (5 available). All components are modularly assembled to a frame. The machine is capable of sorting up to 10,000 uph (units per hour) on Back Fill mode.

The Sorter is also capable of Tray Bin Mapping, shown on figure 5.3. This function allows the user to sort several types of devices in one tray. This function could be used if the user has many levels of device failures.

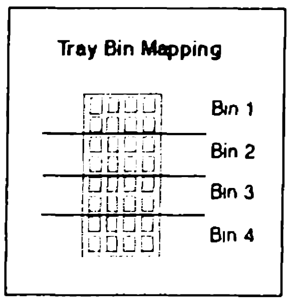


Figure 5. 3 - Tray Bin Mapping diagram

Cascade Binning, shown on configuration 5.4, takes the Tray Bin Mapping one step further. If the user has multiple types of failures (e.g.: 24), the devices may be sorted using Bin Mapping and each tray may be resorted onto separated trays. One type of failure is now on each tray.

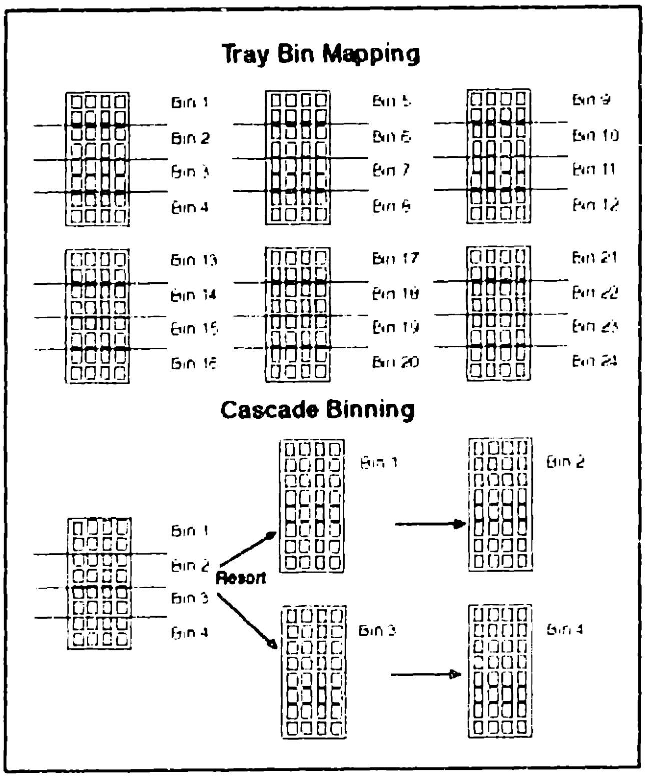


Figure 5. 4 - Cascade Binning diagram

The Scrter also features a Media Transfer function, shown on figure 5.5. Devices may be sorted in three modes for added versatility.

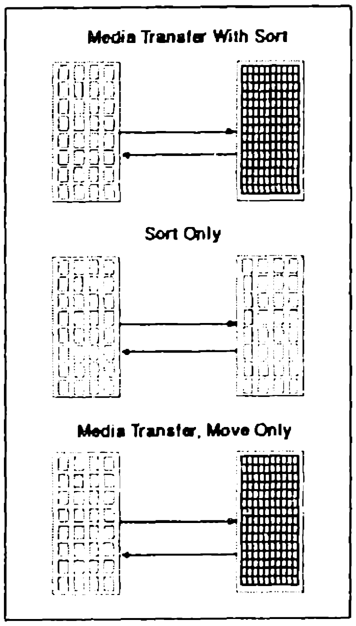


Figure 5. 5 - Media Transfer diagram

Media Transfer with Sort: sorting a specific number of device tray (e.g. 32 devices) with an unlike numbered tray. Example: sorting the tray which just came off the handler (32 devices for memory chips) and placing those devices on a different pitch shipping tray.

Sort Only: sorting a specific number of device tray with a tray of like number (e.g. sorting devices into different categories of quality but using the same type of trays - same tray pitch).

Media Transfer, Move Only: moving devices in a low density tray to a high density tray or vice versa (e.g. testing in a 32 device tray and shipping in a 145 device tray without doing any sorting)

Figure 5.6 shows a top view of the machine with the nomenclature associated with each tray/stack position. The concept proposed is based on the optimization of throughput by back filling high yield input trays and using them as output trays.

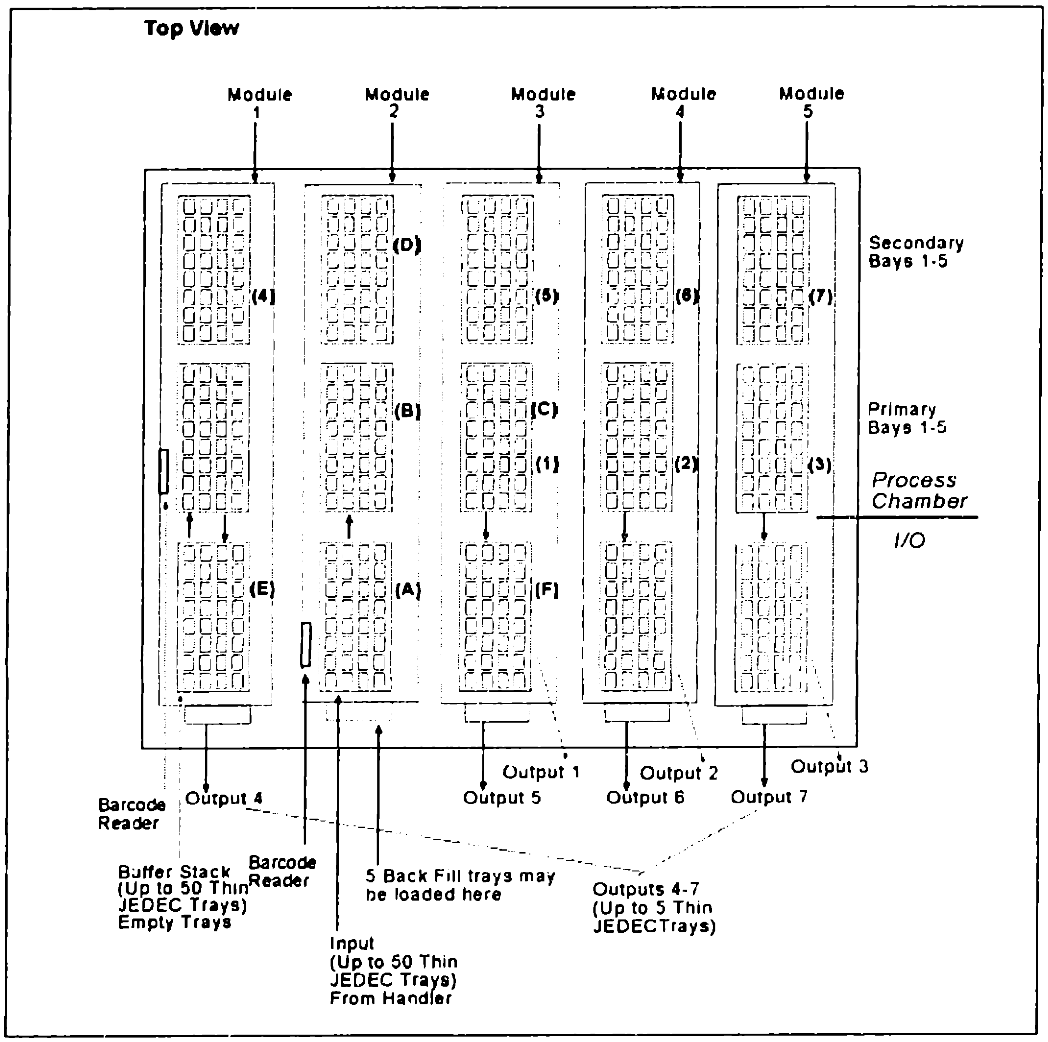


Figure 5. 6 - Top view of the Apollo work area

• At "Position A"; Input up to 50 thin JEDEC trays to be sorted. Elevator mechanism lifts trays and the bottom tray is released on to a shuttle mechanism. Stack fingers holds the remaining trays on the stack. Bar code reader scans bar code to determine tray status for sorting.

• The tray in "Position A" is shuttled to "Position B". The tray is aligned along 3 cam followers by an air actuated plunger mechanism (the precisor), as shown on figure 5.7.

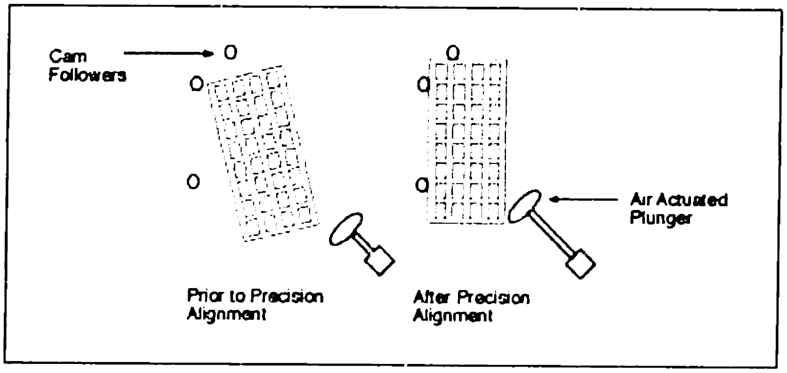


Figure 5. 7 - The precising mechanism in the Tray Module

• Gripper removes non-type 1 chips from the tray in "Position B" and distributes them to the respective trays holding those types in primary bays 1-3 or secondary bays 4-7.

• Gripper lifts tray in "Position B" and places tray in "Position C". Tray in "Position C" is re-populated (back filled) using type 1 devices located in "Position D" (Back Fill Buffer Tray).

• Tray in "Position C" shuttles to "Position F" and is stored building up to a 50 thin trays stack, a primary output stack.

• Empty trays are loaded by the operator in "Position E".

5.3.3 System's Modularization

Based on the functional description of the Apollo Sorter the designer can make a functional chart of the machine and integrate similar functional groups in subsystems. The subsystems than become modules of the machine. One must observe the requirements of serviceability and human factors to replace/service these modules (size, weight, shape, etc). Figure 5.8 shows the Sorter as a system where power (electrical and air), data transfer and trays (with and without devices) cross the boundaries of the system. Inside the system Sorter there are 5 Tray Modules, 1 Gantry, 1 Gripper, 1

Frame and 1 Controls. Each of these subsystems constitute a module of the machine. Within each module there are still some subdivisions that can be grouped together as sub-subsystems.

For assembly purposes the sub-subsystems can be put together independently of the modules, then a set of them constitute a module which is completely independent of the rest of the machine to perform its function, except by power, data and support structure (alignment). This way, modules can be developed and optimized based on the most recent advances on engineering without affecting the functionality of other modules. The designer must only observe the physical limitations imposed up front on to his design module or design control volume (DCV) and what elements cross the boundaries of this DCV.

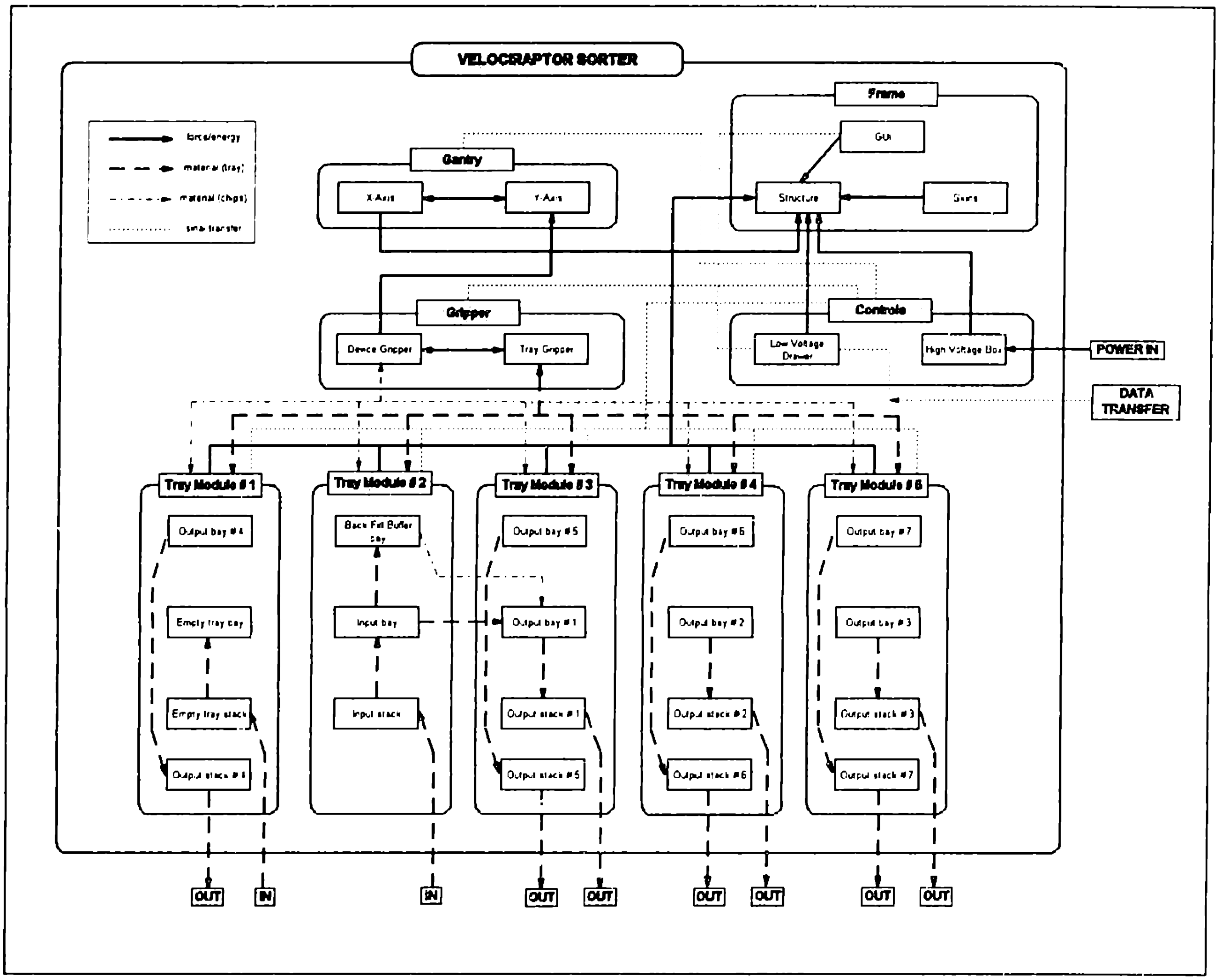


Figure 5. 8 - Apollo Sorter functional diagram

5.3.4 System Error Budget

After defining the modularity of the system it is important to evaluate the geometrical errors of the machine and statistically accumulate them. Figure 5.9 is a spreadsheet representation of the Apollo sorter error budget.

		Ideal Error between tool tip and workpiece	0.0000	0.0000	0.0000			
		These MUST be zero when you start						
Number of tool coordinate systems			**10**	**ntcs**				
Number of work coordinate systems			**6**	**nwcs**				
Always start at the tooltip and work back to the reference frame, and then on to the workpiece!								
Index #	System	Description	X	Y	Z	ThetaX	ThetaY	ThetaZ
1	T10	**Suction Cup**	**0.0000**	**0.0000**	**-2.5079**	**0.0000**	**0.0000**	**0.0000**
2	T9	**Vacuum Piston**	**0.0000**	**0.0000**	**-0.2756**	**0.0000**	**0.0000**	**0.0000**
3	T8	**Vacuum Cylinder**	**0.0000**	**0.7638**	**-5.9301**	**0.0000**	**0.0000**	**0.0000**
4	T7	**Gripper**	**0.0000**	**0.0000**	**-0.6250**	**0.0000**	**0.0000**	**0.0000**
5	T6	**Y-Carriage**	**0.0000**	**-8.7579**	**0.0000**	**0.0000**	**0.0000**	**0.0000**
6	T5	**Y-Rail**	**0.0000**	**0.0000**	**-1.3750**	**0.0000**	**0.0000**	**0.0000**
7	T4	**Y-Extrusion mount to X-Carriage**	**0.0000**	**0.0000**	**-0.6250**	**0.0000**	**0.0000**	**0.0000**
8	T3	**X-Carriage**	**0.0000**	**0.0000**	**0.0000**	**0.0000**	**0.0000**	**0.0000**
9	T2	**X-Rail**	**0.0000**	**0.0000**	**-1.3750**	**0.0000**	**0.0000**	**0.0000**
10	T1	**X-Extrusion mount to Frame**	**25.0000**	**32.0000**	**44.3810**	**0.0000**	**0.0000**	**0.0000**
11	W1	**Frame**	**0.0000**	**0.0000**	**0.0000**	**0.0000**	**0.0000**	**0.0000**
12	W2	**Tray I/O Module**	**25.0000**	**24.0000**	**20.0000**	**0.0000**	**0.0000**	**0.0000**
13	W3	**Module-Tray Reference Edge**	**-2.7000**	**6.2500**	**11.4670**	**0.0000**	**0.0000**	**0.0000**
14	W4	**Tray**	**2.7000**	**-6.2500**	**0.0000**	**0.0000**	**0.0000**	**0.0000**
15	W5	**Site**	**0.0000**	**0.0059**	**0.1004**	**0.0000**	**0.0000**	**0.0000**
16	W6	**Package**	**0.0000**	**0.0000**	**0.1000**	**0.0000**	**0.0000**	**0.0000**

Results	Ideal error between tool & work in the reference frame		Actual error between tool & work in the reference frame	
dX (m)	0.0000000		**0.0112104**	
dY (m)	0.0000000		**-0.0010101**	
dZ (m)	0.0000000		**-0.0049886**	
dPHIX (rad, deg	0.0000000	0.0000000	**0.0013379**	**0.1677019**
dPHIY (rad, deg	0.0000000	0.0000000	**0.0003998**	**0.1764009**
dPHIZ (rad, deg	0.0000000	0.0000000	**0.0029291**	**0.2568695**

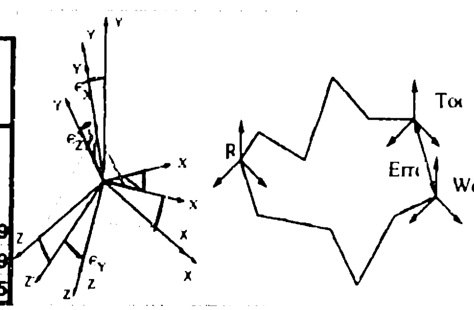


Figure 5. 9 - Apollo sorter error budget

5.4 Embodiment of the Apollo Sorter Design

Before the analysis of design for each module there is a last system level procedure to be observed. The mechanical interface of each module is defined as well as the error budget of the machine. This way each module can be designed keeping in mind a statistical tolerance stack up that can not surpass the budget error of the individual module, guarantying the integrity of the positional accuracy of the machine as a system. This phase on the design is fundamental to assure deterministically the performance of the machine.

5.4.1 Robot Module

The Gantry on the Sorter is the robot mechanism responsible for moving the Gripper around the workspace to the different sort locations. The Gantry provides the quick move times necessary to meet the throughput specifications of the Sorter. It has a quasi-kinematic coupling interface with the Gripper, allowing easy changeover of Grippers.

The linear motors provide the acceleration and maximum velocity required for the system throughput. Each axis is conceived as a compact, simple, and stiff aluminum extrusion that minimizes system weight with a pair of linear bearings and a centered linear motor.

The X-axis structure mounts to the Frame of the Sorter, the Y-axis structure is centered to the X-axis carriage and the Gripper mounts to the Y-axis carriage, as shown in Figure 5.10. The proposed arrangement has the Y-axis mounted centered relative to the X-zxis carriage, so reducing Y-direction footprint of the machine (no need for overhanging) and minimizing gravity moments on the X-axis bearings.

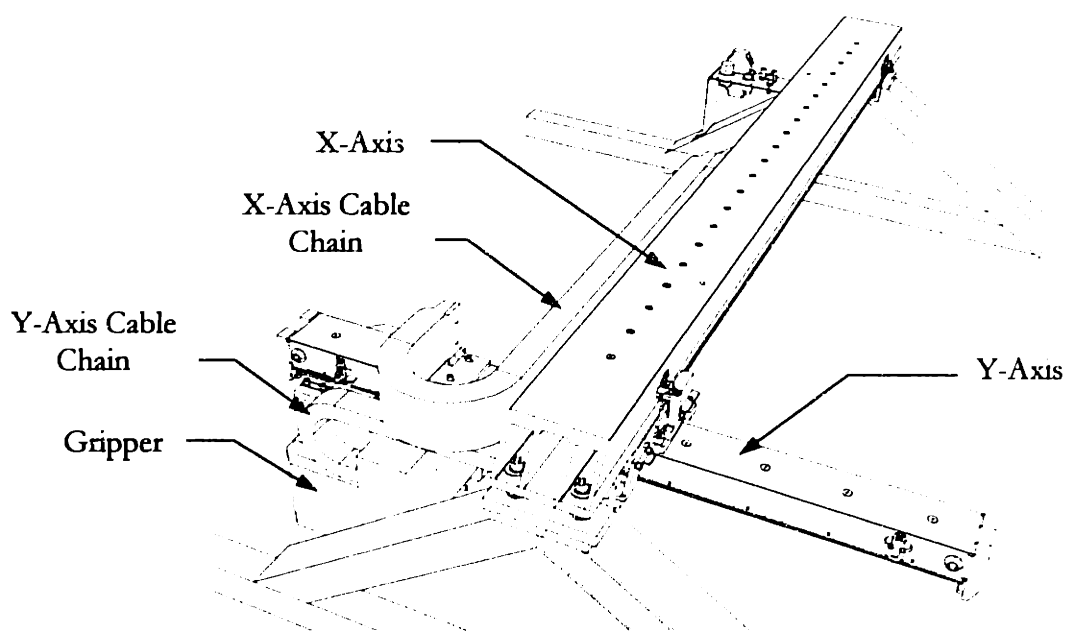


Figure 5. 10 - Gantry design

The physical dimensions of the of the Gantry for the KMX 716 are deterministically defined based on device dimensions, pitch on tray, size of Gripper head, number of sites on Gripper head, etc. Figure 5.11 shows a portion of the spreadsheet created to define axis and travel lengths.

JEDEC Carriages Outline - Formed Packages

Package	Body Size (inches)	X	Y	Z	Tray Matrix Columns	Rows	Column pitch X	Row pitch Y	X-Magnet Track	Y-Magnet Track	X-Axis Size	Y-Axis Size
	.300"x.625"	0.335	0.625	0.138	12	17	0.413	0.705	50.400	34.800	52.192	34.800
	.300"x.675"	0.335	0.675	0.138	12	16	0.413	0.756	50.400	34.800	52.192	34.900
SOJ 300 mi	.300"x.725"	0.335	0.725	0.138	12	15	0.413	0.803	50.400	34.800	52.192	34.900
	.300"x.825"	0.335	0.825	0.138	12	13	0.413	0.904	50.400	33.600	52.192	34.699
	.300"x1.075"	0.335	1.075	0.138	12	10	0.413	1.154	50.400	33.600	52.192	34.738
											52.192	34.900
	.400"x.725"	0.440	0.725	0.138	9	15	0.520	0.803	50.400	34.800	52.228	34.900
	.400"x.825"	0.440	0.825	0.138	9	13	0.520	0.904	50.400	33.600	52.228	34.699
	.400"x.875"	0.440	0.875	0.138	9	12	0.520	0.957	50.400	33.600	52.228	34.486
SOJ 400 mi	.400"x.925"	0.440	0.925	0.138	9	12	0.520	1.004	50.400	34.800	52.228	35.100
	.400"x1.025"	0.440	1.025	0.138	9	11	0.520	1.102	50.400	34.800	52.228	35.278
	.400"x1.075"	0.440	1.075	0.138	9	10	0.520	1.161	50.400	33.600	52.228	34.825
	.400"x1.125"	0.440	1.125	0.138	9	10	0.520	1.205	50.400	34.800	52.228	35.301
											52.228	35.301
Micron	Ring 2				4	8			45.600	22.800	45.992	22.800
											45.992	22.800

Figure 5. 11 - Deterministic definition of axis dimensions

After inputting the data on the spreadsheet for the devices handled by this machine one can determine that each axis specification is:

- X-axis: 1194 mm Travel, 1372 mm extrusion length;
- Y-axis: 813 mm Travel, 965 mm extrusion length.

To meet the throughput requirements in the Sorter specification, the following motion characteristics are designed:

- Acceleration: 1 g (386.2 in/sec^2);
- Max Velocity: 1.0 m/s (40 in/sec);
- Settling Time: 50-100 msec.

5.4.1.1 Control System

The Gantry is commanded to do individual point-to-point moves by the central control. The Gantry provides basic status information back to the central control (i.e. Actual Position, Desired Position, Target Position, and Moving/Settling/Settled/Killed Status).

Each axis has Linear Motor Command Signal, Encoder Feedback, Positive Limit, Negative Limit, and Home Sensor. Pass-through for all Gripper Utilities. Gantry specific utilities:

- 220 VAC Single-Phase for Linear Motor Amplifiers;
- +24 VDC for Limit Switches;
- + 5 VDC for Encoders;
- total of 130 Watts continuous and about 260 W peak..

5.4.2 Tray Module

The Tray Module manages empty and full tray transport and location in the Sorter. The Tray Module handles either input or output functions (programmable). Each module has the following components.

- A primary stack capable of handling a minimum of 50 thin JEDEC trays. An operator is able to safely load/unload 50 thin JEDEC trays in a single motion;
- A secondary stack capable of handling a minimum of 3 stacks or 5 thin JEDEC trays. An operator is able to safely unload 5 thin JEDEC trays in a single motion;
- A means for holding a single tray in the high throughput bay (primary bay) location, accurately enough for successful package pick/place (the precisor); and a means for shuttling a single tray at a time between the primary bay location and the primary stack;
- A means for holding a tray in the low throughput bay (secondary bay) location accurately enough for successful package pick/place (all five secondary trays could be held in this location so long as the top tray is correctly located)(the precisor); and a means for shuttling up to 5 thin JEDEC trays from the secondary bay location to the secondary stack location in a single motion.

There are [(A+3)/2] Tray Modules designed to create A sort locations (A=7 for this version of the Apollo Sorter). The sort locations are addressed as 1 Primary Bay for inputting, [(A-1)/2] = 3 @

5.4.1.1 Control System

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Each axis has Linear Motor Command Signal, Encoder Feedback, Positive Limit, Negative Limit, and Home Sensor. Pass-through for all Gripper Utilities. Gantry specific utilities:

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There are [(A+3)/2] Tray Modules designed to create A sort locations (A=7 for this version of the Apollo Sorter). The sort locations are addressed as 1 Primary Bay for inputting, [(A-1)/2] = 3 @

Primary Bays for outputting, [(A+1)/2] = 4 @ Secondary Bays for outputting, 1 Primary Bay for empty tray buffering and 1 Secondary Bay for back fill buffering.

The Input and the Empty Tray Buffer have means for identifying trays by a bar code posted on the tray. At the Empty Tray Buffer location the bar code reader is placed at the primary bay inside the sort area (reachable by the Robotics) allowing the Gantry & Gripper to cycle trays through for identification in case of loss of data.

The motion characteristics for the shuttle motion are:

- Travel: Full depth of the module. (1320 mm);
- Type: Conveyor belt (timing belt) driven by stepper motor;
- Positioning Accuracy: +/- 50 μm for service tray on top of module;
- Settle Time: 200 msec.

The Tray Module can be further subdivided into two subsystems: Upper Shuttle mechanism and Lower Shuttle mechanism. The design is presented on figure 5.12.

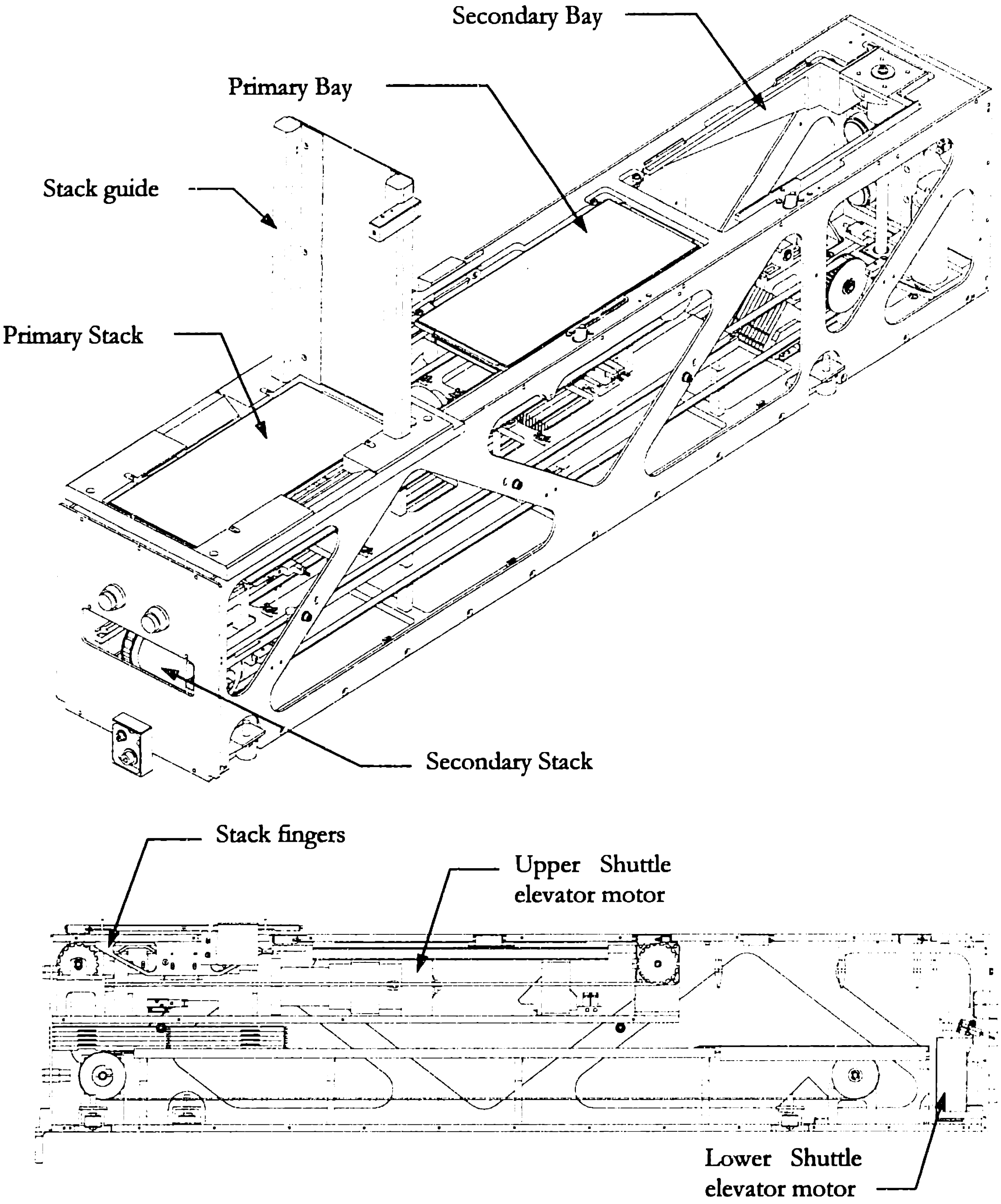

Figure 5. 12 - Tray Module design

5.4.2.1 Upper Shuttle

The Upper Shuttle is the subsystem of the Tray Module that handles trays between the primary bay and the primary stack. Trays shuttle back and forth between the bay and the stack location on a conveyor belt driven by a stepper motor. At the stack location, an elevator servo actuated carries the tray up until it reaches the bottom of the stack. A set of pneumatic driven fingers holds the primary stack. To load or download a tray from the bottom of the stack there is a set of coordinated motions between the elevator and the stack fingers that successfully allows the performance of the task. Figure 5.13 shows a representation of the Tray Module where only the Upper Shuttle mechanism is visible. The Upper Shuttle functional characteristics are:

• tray location within +/- 50 μm to the reference edges on the top master plate on primary bay;

• tray moves in or out of the primary bay in less then 2.5 seconds;

• it handles a stack of up to 50 thin JEDEC trays on the operator assist area (outside of the robot work envelope).

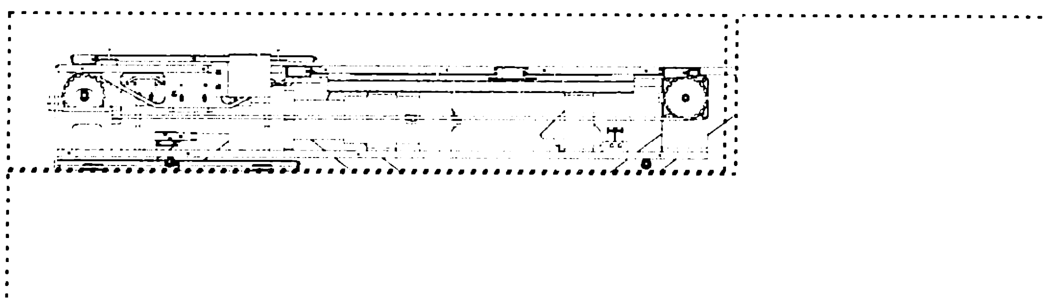

Figure 5. 13 - Upper Shuttle mechanism

The Upper Shuttle can be assigned both input or output functions. A functionality flowchart for each function (input and output) is presented on figure 5.14.

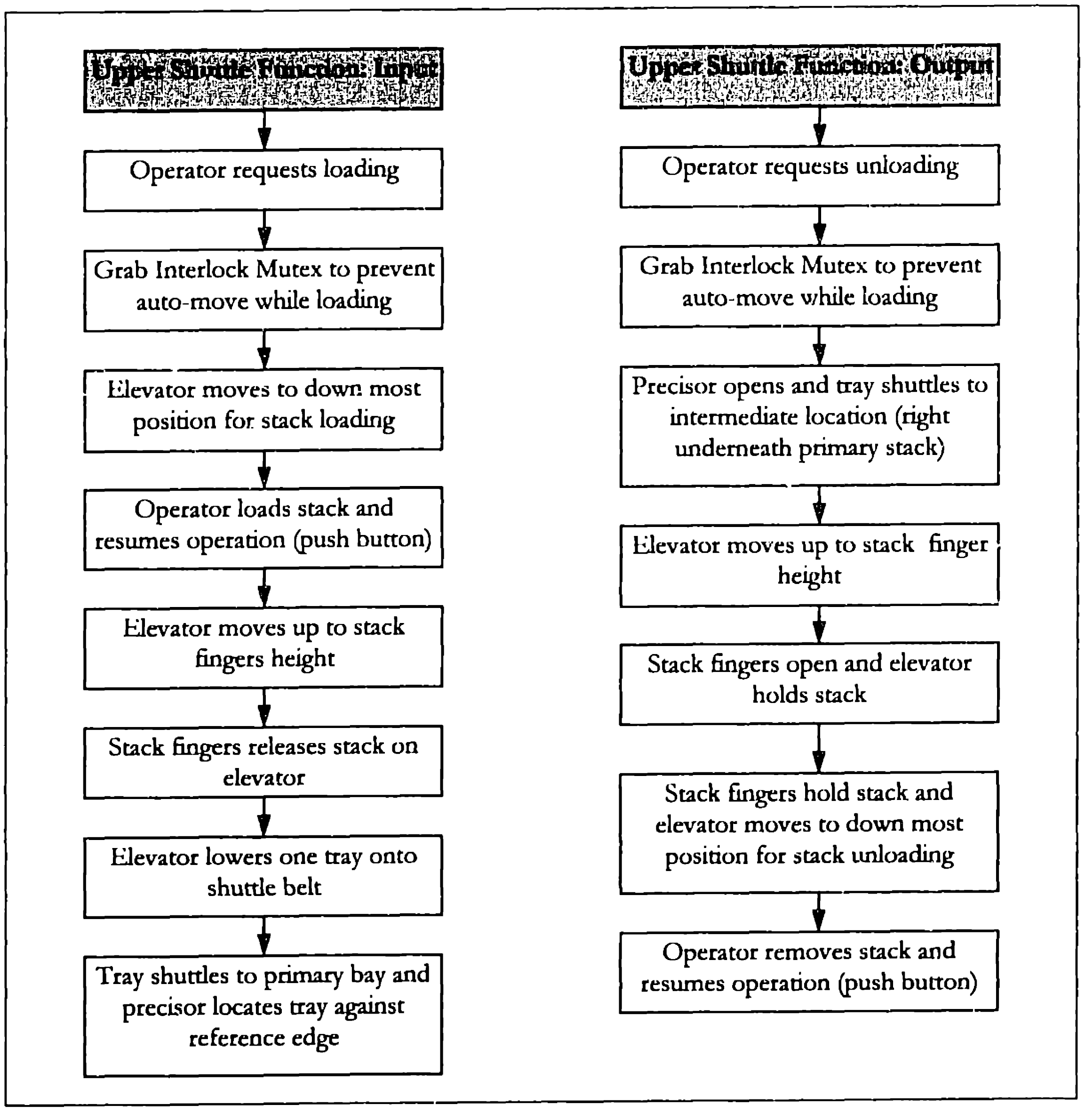

Figure 5. 14 - Upper Shuttle functionality flowchart

The front elevator is designed having the budget move time as a motion constrain, which helps define deterministically the motor and power train specifications, as shown in figure 5.15.

Gear Strength Calculation

Face width of gear =	0.38	[in]
Material Yield Strenght =	20	[kpsi]
Velocity, at pitch diam. =	197	[FPM]
Outline factor =	0.408	
Diametral pitch =	24	
Pitch diameter =	1	[in]
Allowable stress =	15.1	[kpsi]
Max. tangential load =	96.0	[lbf]
Static torque capacity =	48.0	[in-lbf]
Mass of one full tray =	283.5	[g]
Number of trays on stack =	50	
Mass of trays =	14.2	[kg]
	31.3	[lb]
Mechanism mass =	3.0	[kg]
Safety factor =	2	
Total load =	168.5	[lb]
Number of gears =	2	
Static Load per Gear =	168.5	[N]
	37.9	[lbf]
Static Torque =	18.9	[in-lbf]
	TRUE	
Acceleration/Deceleration =	1.0	[g]
Dynamic Load per Gear =	337.0	[N]
	75.8	[lbf]
Dynamic Torque =	37.9	[in-lbf]

Front Elevator Actuator

Tray stack mass =	14.2	[kg]
Mechanism mass =	3.0	[kg]
Inertial mass =	4.0	[kg]
Safety factor =	2.0	[kg]
Total inertia =	42.4	[kg]
Travel distance =	33.02	[mm]
Time budgeted =	0.5	[sec]
Gear pitch diameter	1.0	[in]
	25.4	[mm]
Gear box efficiency =	70	%
Gear box ratio =	50	: 1
Motor velocity =	3000	[rpm]
Max. linear velocity =	80	[mm/s]
Min. linear acceleration =	0.93	[m/s^2]
Driving torque =	577.5	[Ncm]
Motor torque =	16.5	[Ncm]

Figure 5. 15 - Front Elevator deterministic design

5.4.2.2 Lower Shuttle

The Lower Shuttle handles 3 stacks of 5 thin JEDEC trays (secondary stack). It is primarily used as a low yield output system. Trays are placed on the secondary bay until a stack of up to 5 thin JEDEC trays is built up. The servo driven rear elevator moves the full stack down on to a conveyor belt (driven by a stepper motor) which brings the stack to a mid position inside the module. When a second 5-trays stack is lowered on to the conveyor belt, this stack and the previous one are shuttled forward. The first stack lowered appears in the front window for operator assist and the second one waits in the mid position inside the module for another stack to come on to the conveyor system. Whenever necessary an operator can give the machine a command to remove the stack in the mid point inside the module without having to wait for another stack to be lowered by the back elevator (e.g. when purging the machine). Figure 5.16 shows a representation of the Tray Module where only the Lower Shuttle mechanism is visible. The Lower Shuttle functional characteristics are:

• tray location within +/- 50 μm to the reference edges on the top master plate on secondary bay;

• tray moves up or down on the secondary bay (back elevator) in less then 0.5 seconds when moving just one tray width and in less then 2 seconds when moving the stack to or from the conveyor belt.

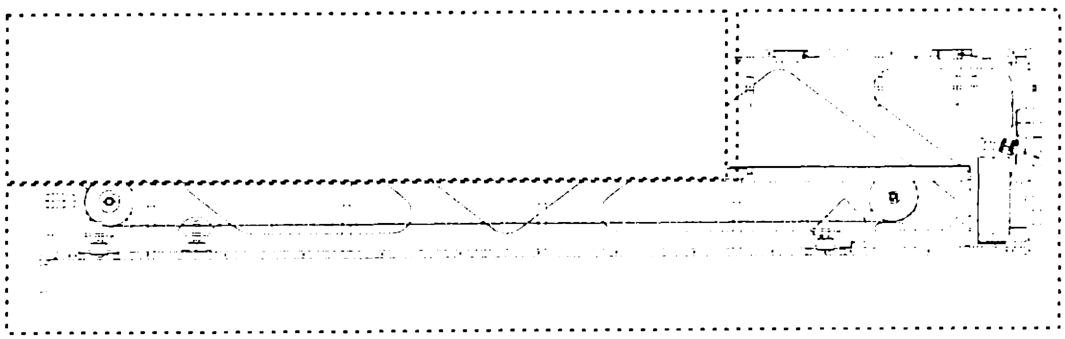

Figure 5. 16 - Lower Shuttle mechanism

The Lower Shuttle is assigned with output function. A functionality flowchart is presented on figure 5.17.

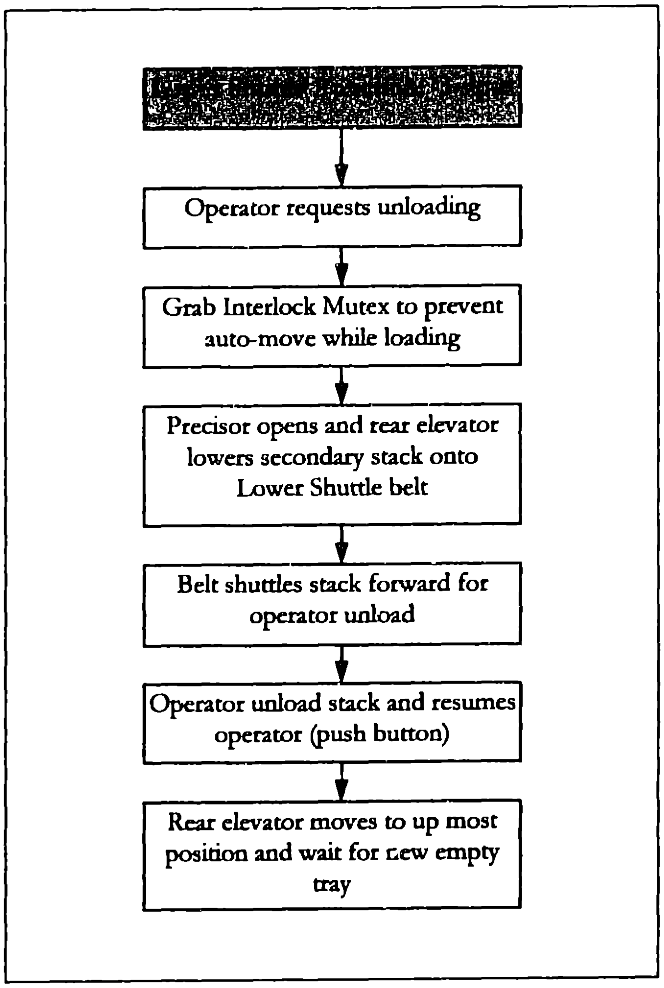

Figure 5. 17 - Lower Shuttle functionality flowchart

Deterministically calculating the motor torque requirements, bearing load capacity and life and ball screw parameters for the motion to happen within the budget time. Figure 5.18 shows the spreadsheet analysis performed for the Rear Elevator design.

Linear Bearing Calculation		
Travel under load per cycle =	160.0	[mm]
Cycle time =	150.0	[sec]
5 years Travel =	168.2	[km]
10 years Travel =	336.4	[km]
Mass of one full tray =	283.5	[g]
Number of trays on stack =	5	
Safety factor =	2	
Mass of trays =	1.4	[kg]
Mechanisms mass =	2.5	[kg]
Acceleration/Deceleration =	0.8	[g]
Maximum Load =	103.8	[N]
Nominal Load =	85.0	[N]
Distance from CG to Bearing =	203.0	[mm]
Distance from Screw to Bearing =	41.0	[mm]
Maximum Moment on Bearing =	16.8	[Nm]
Nominal Moment on Bearing =	13.8	[Nm]
Moment used for Life Calculation =	14.7	[Nm]
For 5 year life, Dynamic Moment =	17.2	[Nm]
For 10 year life, Dynamic Moment =	21.2	[Nm]

Rear Elevator Ball Screw		
Ball-screw pitch =	10	[mm]
Accelerating Load =	61	[N]
Proportion of Stroke a	0.087	
Constant Speed Load	58	[N]
Proportion of Stroke a	0.913	
Equivalent Operating L	58	[N]
Driving torque =	10.9	[Ncm]
Backdrive torque =	8.8	[Ncm]
Ball circle diameter =	12.7	[mm]
Length of the screw =	150	[mm]
Safety factor =	2	
Critical speed =	488	[mm/s]
Rated dynamic load ca	720	[lb]
	3203	[N]
Travel Life =	1.68E+11	[in]
Days/year =	365	[days]
Ball screw operating h	24	[hrs]
Ball-screw life =	407.0	[years]

Rear Elevator Actuator		
Travel =	160	[mm]
Tray stack mass =	1.4	[kg]
Mechanism mass =	2.5	[kg]
Total inertia (w/ safety) =	5.9	[kg]
Time budgeted =	1.00	[sec]
Ball-screw pitch =	10	[mm]
Gear box efficiency =	95	%
Gear box ratio =	1.5	: 1
Motor velocity =	3000	[rpm]
Max. linear velocity =	333	[mm/s]
Min. linear acceleration =	0.64	[m/s^2]
Driving torque =	10.9	[Ncm]
Backdrive torque =	8.8	[Ncm]
Motor torque =	7.6	[Ncm]
Customized:		
Ball-screw pitch =	10	[mm]
Designed Motor Torque =	10.0	[Ncm]
Motor velocity =	3000	[rpm]
Max. linear velocity =	333	[mm/s]
Acceleration =	3.89	[m/s^2]
TimeLinear =	0.57	[sec]
# of line counts =	500	[counts]
resolution =	13.33333	[um]

Figure 5. 18 - Rear Elevator deterministic design

The same can be done for the shuttle mechanism (upper and lower) on the tray module, figure 5.19.

Shuttle Motor Analysis		
Mass of one full tray =	283.5	[g]
Number of trays on stack =	5	
Belt high =	4.5	[mm]
Belt width =	16.0	[mm]
Belt length =	3000.0	[mm]
Belt material density =	2.7	[mm]
Belt mass =	0.6	[mm]
Mass of trays =	1.4	[kg]
Static friction coeficient =	0.45	
Dynamic friction coeficient =	0.30	
Pulley Diameter =	57.29	[mm]
Acceleration/Deceleration =	0.60	[g]
Normal sliding force =	12.7	[N]
Static friction force =	5.7	[N]
Dynamic friction force =	3.8	[N]
Kinematic force =	11.8	[N]
Total static force =	17.5	[N]
Total kinematic force =	15.6	[N]
Safety factor =	1.3	
Motor starting torque =	130.5	[Ncm]
Motor running torque =	116.3	[Ncm]

Figure 5. 19 - Shuttle mechanism analysis

5.4.2.3 Control System

The drivers and amplifiers for the motors are located on the Frame on a main control unit. The Tray Module carries the pneumatic manifolds for the stack finger and precisor operations. The Tray Module requires air supply (50 psi), 24 VDC for the motor, 5 VDC for the sensors.

Information from sensors is carried out to a connector at the back of the module that matches with a connector from the Frame. All sensor information is then processed on the control unit in the Frame of the machine.

5.4.3 Frame

The Frame is designed to support and allow manual alignment of the Tray Modules, Gantry & Gripper. The Frame is built out of tubular stainless steel, which provides good corrosion resistance and also good grounding, necessary on semiconductor equipment. The Frame is designed to 35 Hz first mode natural frequency, which as a first approximation allows the design to use 70 msec for controls settling time. The overall dimensions of the frame with skins are:

- Frame X-dimension: 1422 mm (width);
- Frame Y-dimension: 1220 mm (depth);
- Frame Z-dimension: 1168 mm (height);
- Clearance to the floor: 150 mm.

The Frame performs the following functions:

- it supports and allow for manual alignment (Z-axis, Theta-X, Theta-Y and Theta-Z) of the Tray Modules;
- it supports and allow for manual alignment (Z-axis, Theta-X and Theta-Y) of the Gantry & Gripper System;

Figure 5.20 presents the designed frame where the Gantry is supported at the top of the "A" section at the ends of the tubular structure.

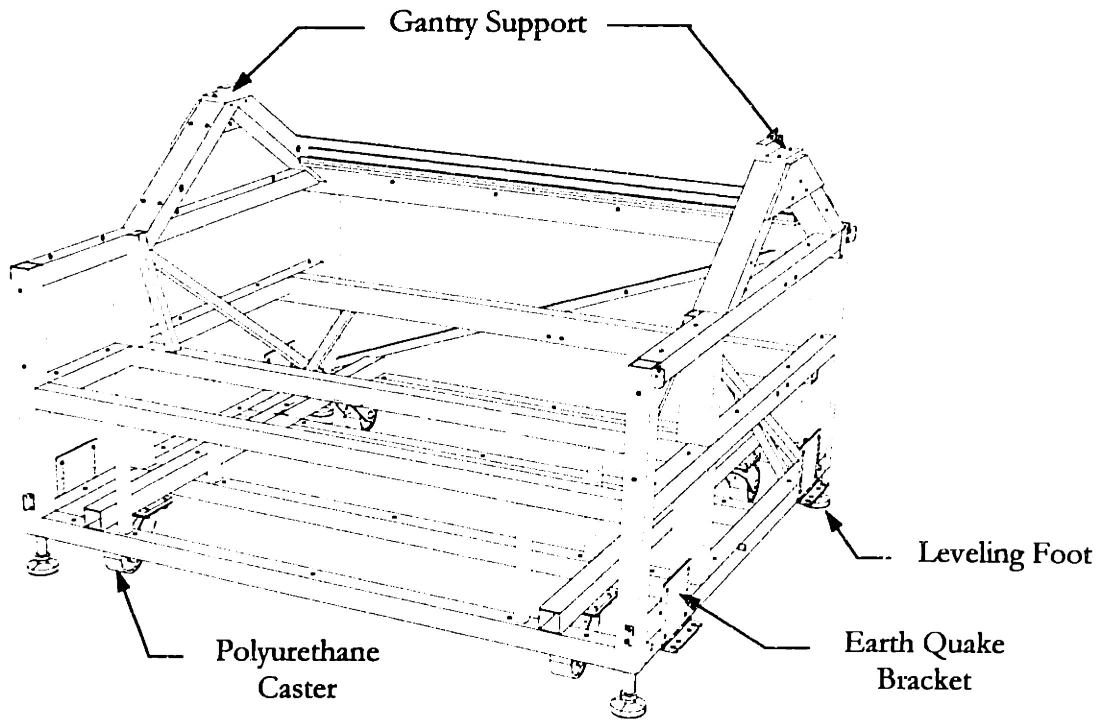

Figure 5. 20 - Stainless steel tubular Frame

5.4.5 Gripper Module

The Gripper system of the Apollo Sorter is designed to hold up to six devices (Device Gripper) simultaneously with a thin or thick JEDEC tray (Tray Gripper). The Gripper system is composed of the Tray Gripper which attaches to the side of the Device Gripper. Trays can be picked-up with or without devices on the Device Gripper, increasing productivity by allowing simultaneous transport of trays and devices. The structure of the Gripper system is made of machined aluminum having all the pneumatic components (regulator, valves, etc.) and controls assembled to it. There is only one line for pressurized air, one line for vacuum and a cable bundle as utility interface between the Gripper and the adjacent Gantry. The concept designed for the Apollo Sorter is presented on figure 5.21.

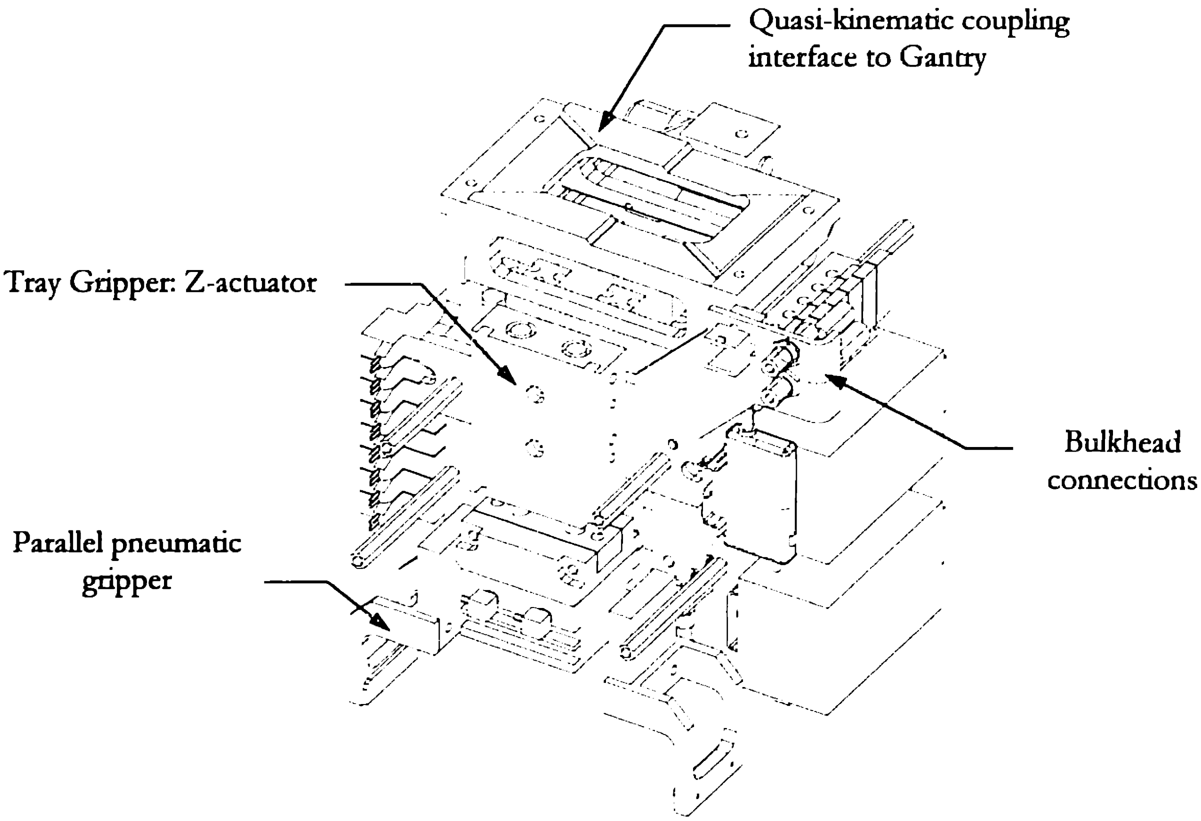

Figure 5. 21 - The Gripper head

5.4.5.1 Device Gripper

The Device Gripper is an N-Site system (design is for N=6 for the KMX 716). Each site is individually controllable in both Z up/down and device gripper pressure/vacuum. Each site can sense the presence or absence of a package. The Z motion is a 30 mm pneumatic actuated cylinder. The pressure/vacuum control is either suck or blow, normally open to pressure to keep the system clean from debris. The sites are located at tray pitch (for a 32-site automation tray).

The motion characteristics for the Delta-Z motion of each site are:

- Actuator: dual-acting air cylinder, 30 mm to allow for tray transport simultaneously with devices;
- Settling Time: 100 msec;
- Vacuum characteristics: two-position valve. NC=>Vacuum, NO=>Exhaust or low pressure (2 psi);
- Sensor: digital sensor to measure presence/absence of a device when under vacuum. The same sensor is used with a two threshold logic to determine not only presence/absence of devices but also initial clogging of the vacuum lines. This allows, for the first time on this type of equipment, for predictive maintenance (as is possible nowadays in the machine tool industry).

All pneumatics and controls for the Device Gripper are assembled to the structure of the Gripper System.

5.4.5.2 Tray Gripper

The Tray Gripper is capable of picking up a single JEDEC tray (full size, thin or thick). The tray can be picked-up with or without packages on it. The Tray Gripper is composed of a short stroke (15 mm) guided cylinder (2 positions) with a pneumatic parallel gripper mounted at the end. The Tray Gripper will not interfere with Device Gripper function of carrying packages as long as the Device Gripper is in the Up position.

5.4.5.3 Control System

A single DeviceNet drop cable is used to control all valves and read all sensors. The Gripper is configured as two DeviceNet nodes. One is all the control outputs. The other is all the sensor inputs. A DeviceNet scanner card is installed in the central computer. It is configured during system-startup as desired for the application.

The controllability of each axis on the Gripper is as following:

- Each Site-Z can be individually controlled to be Up (NC) or Down (NO);
- Each Site-Vacuum can be individually controlled to be Suck (NC) or Blow (NO), (<5psi);
- The Tray-Z can be controlled to be Up (NC) or Down (NO);
- The Tray Gripper can be controlled to be Closed (NC) or Open (NO);
- Each Site-Z has one sensor to tell if the cylinder is Up or NotUp;
- Each Site-Vacuum has a sensor to report if a chip is Present or Absent. The Site-Vacuum sensor only reports valid information if the Site-Vacuum control is Suck;
- The Tray-Z has one sensor to tell if the cylinder is Up or NotUp;
- The Tray-Gripper has two sensors to report if the gripper is Closed or Open.

5.5 System Integration

The integration of the modules previously described into a machine is presented on figure 5.22. Because of the modularity, this machine can be configured with more or less modules or even with modules that perform different functions (lead inspection, tape & reel, etc). The functionality change of the machine allows users to maintain a minimum stock of machines and modules and still be able to balance their production lines with the necessary equipment. Also serviceability is much improved, where customers used to maintain back up machines now it is only necessary to keep spare modules, so improving on cost-of-ownership of these systems.

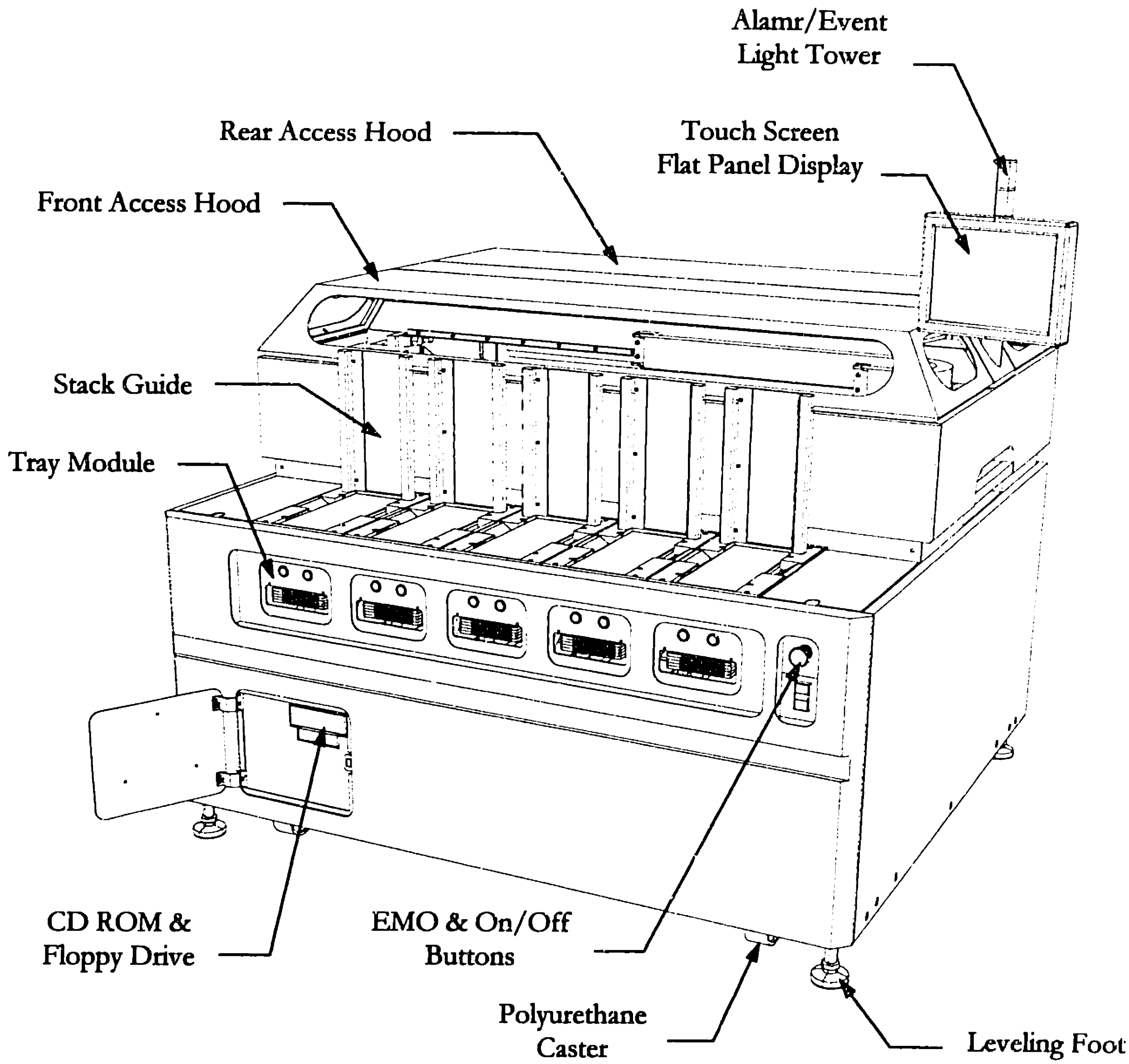

Figure 5. 22 - The Apollo Sorter

Chapter 6

CHARACTERIZATION OF THE APOLLO SORTER

6.1 Introduction

This chapter provides the method for testing and certifying the performance of the Apollo Sorter, together with the results obtained from the first prototype. The specifications presented in this chapter also facilitate matching equipment to production requirements.

The experimentation is used to validate the modular machine methodology and the time budget simulation tool introduced on chapter 3. The test bench on this case is the β-prototype. There is no need for special software, but there is a need for tracking time stamps and saving them into a file for throughput characterization.

6.2 Equipment States Stack Chart

The Sorter operation sequence can be classified either as Setup, Production or Down Time. Within this classification there are nine states of operation. To obtain a realistic measure of throughput the software issues time stamps at end of each state of operation, such that throughput can be computed as the total time intervals associated to a set of states. This way setup times, or load/unload times which are operator dependent can be factored out of the process. Figure 6.1 presents a stack chart of the basic equipment states.

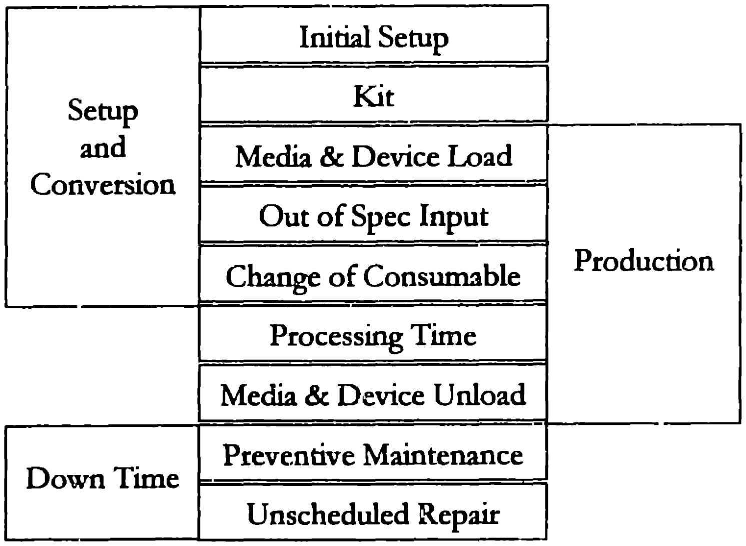

Figure 6. 1 - Sorter operation states stack chart

6.3 Test Methods

Figure 6.2 presents the sequence of tests to be performed and objected results, based on the Time Budget spreadsheet.

Test	Name	Description	Objective (w/ Parallelism)	Objective (w/o Parallelism)
1	Tray Load Sequence	Prepare the work surface for sorting	40 seconds	60 seconds
2	97% Good Sort (1/tray) Backfill Mode	Sort 1 device per tray to 1 output bin Backfill with good device	100%	62%
3	81% Good Sort (6/tray) Backfill Mode	Sort 6 devices per tray to six output bins Backfill with 6 good devices	48%	36%
4	Media Transfer Mode	Transfer all 32 devices to an adjacent tray	27%	22%
5	Purge	Remove all trays from the work surface	12 seconds	30 seconds

Figure 6. 2 - Test Methods (normalized to Test 2 Throughput)

6.3.1 Preparation of Apparatus

- Equipment - Sorter is cleared of all units;

- Trays - 50 trays of 32 devices each are required as well as 10 empty trays. Trays of devices may be recycled from the output to the input to achieve the desired number of test cycles;

- Tray Placement - The full trays are placed in the input primary stack and the empty trays in the empty tray primary stack.

6.3.2 Test: Tray Load Sequence

This test measures the time required to prepare the sorter to begin productive work. With the inputs loaded as above, the sorter is programmed to: (All positions numbers are row, column.)

- Move a full tray of devices from input tray to output bin 1; the sorting site;

- Move six empty trays from empty tray location to all the primary and secondary output bins but output bin 1. That leaves one secondary bay open for the back fill tray and output bin 1 location for the sorted input tray.

The time is measured from the operator starting the operation to when the first device is ready to be removed from the tray. The qualification is made from the average of five load sequences.

6.3.3 Test: 97% Good Sort

The Sorter will be programmed to simulate a 97% Good, 3% Bad. (one device per 32 device tray). The prime yield is sorted to the primary bay assigned bin 1; the non bin 1 devices are sorted to the other output bins; bin 1 tray is back filled with category 1 devices taken from a back fill buffer tray. The test sequence is as follows:

- Sort out non bin 1 devices from the input tray and move it to the back fill secondary bay position;
- Cycle the next input tray from input primary stack to primary bay;
- Sort the non bin 1 devices from the input tray into the output bins other than 1;
- Transfer the input tray with the remaining bin 1 only devices to the output bin 1 primary bay location;
- Back fill the output bin 1 tray with devices (bin 1 only) from the back fill buffer tray;
- Repeat 2-5 until 5000 trays have been processed. (10,000 devices moved);
- The output stack may be recycled to the input by the operator as long as at least 50 trays are used;
- Whenever the back fill tray is empty it is replenished by moving the empty tray to empty tray primary bay position and moving the next input tray to be sorted to the back fill position.

6.3.4 Test: 81 % Good Sort

The Sorter is programmed to simulate an 81% Good, 19% Bad. (six devices per 32 device tray). The prime yield is sorted to bin 1 primary bay; the other non bin 1 devices are sorted to the other primary and secondary output trays. The test sequence is as follows:

- Move the first input tray to the back fill position after sorting out the non bin 1 devices;

- Cycle the next input tray from input primary stack to primary bay;

- Sort the six bad parts from the input tray out to the other outputs (e.g. 2, 3, 4, 5, 6 and 7);

- Transfer the input tray with the remaining bin 1 only devices to the output bin 1 primary bay location;

- Back fill the output bin 1 tray with devices (bin 1 only) from the back fill buffer tray;

- Repeat 2-5 until 1000 trays have been processed. (12,000 devices);

- The output stacks will be recycled to the input by the operator. At least 50 trays should be used for the test;

- Whenever the back fill tray is empty it is replenished by moving the empty tray to empty tray primary bay position and moving the next input tray to be sorted to the back fill position.

6.3.5 Test: Tray to Tray Transfer (32 position to 108 position)

The Sorter is programmed to transfer all of the devices from one tray to another. The tray positions for the start of this test are identical to the end of the test for 97% yield.

- Move all of the devices from the input tray to the adjacent output tray (e.g. bin 1 output tray);

- Transfer output tray bin 1 out to primary stack 1;

- Transfer empty input tray to primary bay empty tray stack;

- Cycle next input tray into the machine (input primary bay) from the input primary stack;

- Repeat 1-4 until 200 trays have been processed. (6400 devices moved);

6.3.6 Test: Purge

The Sorter is programmed to clear the work surface of all trays and devices. The tray positions for the start of this test are identical to the end of the test for 97% yield.

- Move the full input tray from input primary bay position to output bin 1 primary bay with the tray gripper and cycle the tray out to bin 1 primary stack;

- Transfer all primary bay trays to their respective primary stacks;

- Transfer all secondary bay trays to their respective secondary stacks;

- Signal the operator to remove the primary and secondary stacks;

- Check all sensors to confirm machine has been purged.

6.4 Throughput Characterization

The characterization was done on the first production unit to verify the initial predictions made using the Time Budget spreadsheet. Figure 6.3 presents the results for each individual test.

Test	Name	Objective (w/ Parallelism)	Objective (w/o Parallelism)	Measurement (w/o Parallelism)
1	Tray Load Sequence	40 seconds	60 seconds	71 seconds
2	97% Good Sort (1/tray) Backfill Mode	100%	62%	51%
3	81% Good Sort (6/tray) Backfill Mode	48%	36%	31%
4	Media Transfer Mode	27%	22%	23%
5	Purge	12 seconds	30 seconds	36 seconds

Figure 6. 3 - Test Results (normalized to Test 2 Throughput)

The worst case throughput result is within 18% of the measurements. This is an acceptable condition for evaluation done in early stages of the design, were different concepts can be early evaluated.

The error in predicting the throughput of the system gets worse as the proportion of non-controlled motion time to total cycle time increases. Pneumatic actuated axis are considered non-controlled axis and their motion is very difficult to predict or measure. On this machine pneumatic actuated motions are very common and they account for the majority of the time discrepancies encountered.

The parallelism pointed out in this figure reflects a future capability of the machine (software capability) to perform a variety of motions in parallel if satisfying a set of constraints. The machine has the capability to control up to 24-axis of motion and parallelism is certainly one major performance enhancement to the system. The calculated performance numbers are left in the table for future evaluation, once software is brought up to that level.

Chapter 7

SUMMARY AND CONCLUSIONS

7.1 Conclusions on the Modular Testing System

Several semiconductor manufacturers showed great interest in the new floor arrangement and specially on the capability of modifying machine functionality with one basic platform, in other words, one platform can be adapted to perform functions that required several machines on the standard layout. Two major US semiconductor manufacturers are in process of implementing the said system.

7.2 Conclusions on the Modular Design Framework

Similarly to an IBM laptop, where one can use an extra battery pack or an extra hard disk or a floppy drive or a CD ROM drive, the modularly built semiconductor machines have multiple functionality.

The Appolo Sorter can perform functions of a tape&reel machine, a tube loader machine, a laser marking machine, a lead inspection machine, a burn-in board loader/unloader machine, etc. The same way in the beginning of the century Maudsley set the premises for parts interchangeability, this work tries to define the basics of module interchangeability within a machine.

The customer acceptance of this proposed designs is the best proof of success of this concept.

7.3 Final Remarks

The enabling technology are a fundamental design tool for implementing parallel and massive parallel testing of devices and to facilitate modules interchangeability. With FlexTray™ the ideas for parallel testing of devices in a tray became a reality through the introduction of the Galileo In-Tray Test System, another modularly designed machine, see figure 7.1.

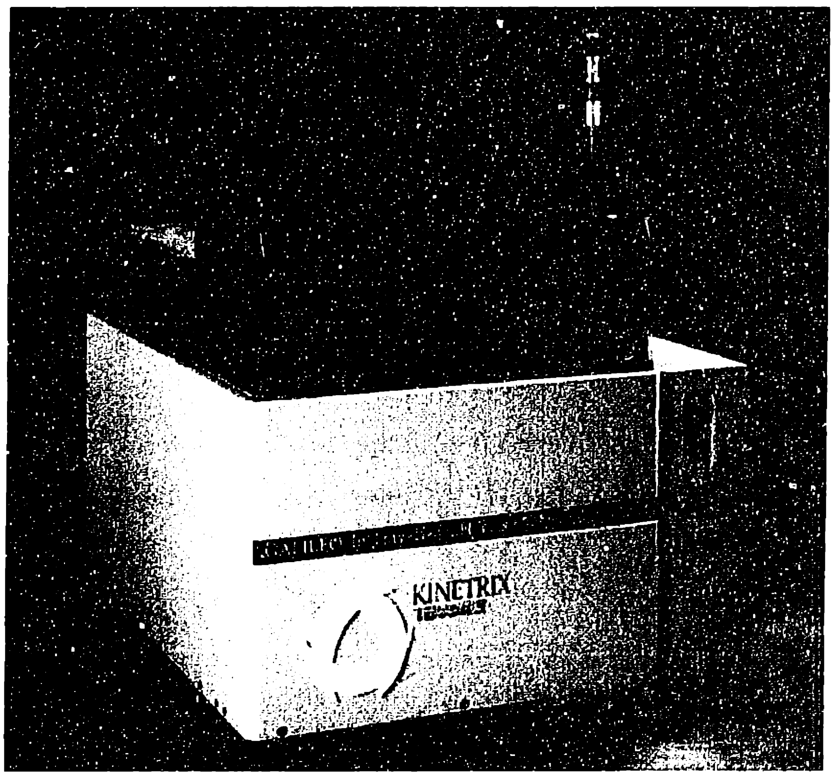

Figure 7. 1 - Galileo In-Tray Test System

7.4 Recommendation for Future Work

There is still further work to be done on implementing modularly built machines (with functional interchangeability capability) in a semiconductor production floor. Since this is a modular architecture it would be interesting to develop multiple modules to these machines and to determine an accurate cost-of-ownership model and properly evaluate the resultant cost of this configuration of production floor against the current state of the art. This design architecture opens up a new avenue for operations research on semiconductor facilities; where dynamic simulation models would help evaluate the production capability of different production floor layouts and different equipment. This would open the possibility to evaluate different machine structures and concepts for TAP.

FlexTray™ needs to be further developed for different types of devices, and more specifically for those that need under package contact for testing such as BGA, μBGA, and CSP. It would also be interesting to add programmable memory capability to the trays, which could avoid the existing need of gigantic database systems for tray mapping information. These databases will eventually slow down the production line due to the time required to access and retrieve data in parallel with other machine performance data.

KinFlex™ technology proved the significance of flexural beam design for system's compliant kinematic coupling. Another step on this would be to implement it for accurate location of the chip inserts (FlexTray™) to a contactor with three balls and three grooves.

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Appendix A

TERMINOLOGY AND DEFINITIONS

ambient environment (test method) - environmental conditions consisting of temperature: 25 degrees Celsius (+/- 2 degrees Celsius) relative humidity: 45-60%

assist - An unplanned interruption that occurs during an equipment cycle where all three of the following conditions apply:

- The interrupted equipment cycle is resumed through external intervention (e.g., by an operator or user, either human or host computer).
- There is no replacement of a part, other than specified consumables.
- There is no further variation from specifications of equipment operation.

change of consumables - The unscheduled interruption of operation to replenish the empty trays, tubes, or tapes.

coplanarity - a special case of the profile of a group of features as defined in ANSI Y14.5. The coplanarity of an I/O on a surface mount semiconductor unit is the distance between the lowest point on the under surface of that I/O and the actual seating plane of the unit. The actual seating plane is determined by allowing the unit to rest on a flat level surface. NOTE: There are three methods for determining unit I/O coplanarity in common use.

actual coplanarity - Unit is measured with the unit I/O on a flat level reference surface.

theoretical coplanarity - Unit is measured without the unit weight on the unit I/O. The theoretical (weight-off-I/0) seating plane is calculated from the spatial coordinates (X,Y,Z) of the individual I/O low points measured with the unit I/O in free space. The actual and theoretical seating planes are the same for units with I/O that do not change when bearing the weight of the unit.

regression plane coplanarity - Unit is measured without the unit weight on the unit I/O. The "best fit" seating plane is calculated from the spatial coordinates (X,Y,Z) of the individual I/O low points measured with the unit I/O in free space. This "best fit" seating plane is commonly called a regression plane and is calculated using a least squares technique.

"Per unit" coplanarity is the coplanarity of the one I/O on the unit which has the worst case deviation from the seating plane.

cycle - (equipment cycle) —- One complete operational sequence (including product load and unload) of processing, manufacturing, or testing steps for an equipment system or subsystem. In single unit processing systems, the number of cycles equals the number of units processed. In batch systems, the number of cycles equals the number of batches processed. For the sorter, a cycle is the processing of one tray of devices in and out of the sorter.

downtime (equipment downtime) The time when the equipment is not in a condition, or is not available, to perform its intended function. It does not include any portion of non–scheduled time.

failure (equipment failure) - Any unplanned interruption or variance from the specifications of equipment operation other than assists.
NOTE: It may be important to categorize and qualify failures in ways that facilitate the resolution of problems and improve overall equipment performance.

flush time - elapsed time to remove (clear out) all units from the equipment. Flush time includes the transfer of all remaining units in partially full input media to the output media.

jam - vernacular; special case of assist that occurs during handling. Commonly associated with mechanical interference between devices or media and the equipment, mechanical interference between the equipment's components, or misplacement of media, devices or piece parts by the equipment.

load time (unit load time) - elapsed time from the first automated contact with the first tray to the sending of the first "start sort signal" to the processing unit.

maintenance - The act of sustaining equipment in or restoring it to a condition to perform its intended function. In this document, maintenance refers to function, not organization; it includes adjustments, change of consumables, repair, preventive maintenance, etc., no matter who performs the task.

media - carrier (tubes, trays, coin stack tubes) for devices. Distinct from support tools in that media are considered a consumable item with limited reuse potential.

$MTBA_P$- Mean (productive) time between assists; the average time the equipment performed its intended function between assists; productive time divided by the number of assists during that time. Only productive time is included in this calculation. Using MTBA, therefore, requires that the user not only have the capability of capturing assist information, but also of tracking and categorizing total time accurately. Reliability can be measured using various factors such as equipment times, cycles, or states.

<u>MCBA</u> - Mean cycles between assists

<u>MCBF</u> - Mean cycles between failures; the average number of equipment cycles between failures; total equipment cycles divided by the number of failures during those cycles (include both product and non-product cycles) This calculation transcends equipment states to include all cycles that the system or sub-system, being considered experiences. It does not require tracking equipment states, only equipment cycles and equipment failures.

<u>$MTBF_P$</u> - Mean (productive) time between failures; the average time the equipment performed its intended function between failures; productive time divided by the number of failures during that time. Only productive time is included in this calculation. Failures that occur when an attempt is made to change from any state to a productive state are included in this calculation. Using MTBF p , therefore, requires that the user not only have the capability of capturing failure information, but also tracking and categorizing total time accurately.

<u>MTTR</u> - Mean time to repair; the average time to correct a failure and return the equipment to a condition where it can perform its intended function; the sum of all repair time (elapsed time not necessarily total man hours) incurred during a specified time period (including equipment and process test time, but not including maintenancedelay), divided by the number of failures during that period.

<u>operator</u> - Any person who communicates with the equipment through the equipment's control panel.

<u>out-of-spec input</u> - A period when the equipment cannot perform its intended function solely as a result of problems created by out of specification or faulty inputs. Those inputs include:

- Support tools: (e.g., warped trays, backward trays, or faulty bar code labels)
- Product: (e.g., upstream process/product problems, wrong product, warped devices, contaminated product, warped lead frames, bent leads, pin one orientation, etc.)
- Test data: (e.g., tray matrix, binning data incorrect, etc.)
- Consumables: (e.g., gripper suction cups , etc.)

Any downtime created by the items listed above shall be included in out of specification input downtime. For example, if, as a result of a bad tray or devices, the sorter is down for repair, all downtime incurred prior to identifying the problem is re-categorized as (out of specification) input down-time.

output media load time - elapsed time from the issue of a "Bin / Tray Full" command to issue of a "Output Media Change Complete" command.

preventive maintenance - The sum of:

- Preventive action: A predefined maintenance procedure (including equipment ramp–down and ramp–up), at scheduled intervals, designed to reduce the likelihood of equipment failure during operation. Scheduled intervals may be based upon time, equipment cycles, or equipment conditions.
- Equipment test: The operation of equipment to demonstrate equipment functionality; (e.g., system loads trays, transfers devices, reads bar code labels, etc.).
- Verification run: The processing and evaluation of devices after preventive action to establish that the equipment is performing its intended function within specifications.

note: Equipment suppliers are responsible for specifying a preventive maintenance program to achieve a predetermined equipment performance level. Users are obligated to identify any deviation from the recommended program if they expect the supplier to meet or improve that performance level.

process site - the locations within the equipment at which processing (sorting/testing/verification/function) is actually performed. For test handling equipment this is normally the contactor or test site ; for visual/mechanical inspection systems it is the vision or inspection station, for lead conditioners the forming station, for sorters the primary sort bay, etc.

processing time - elapsed time from "start function" signal to "end of function" signal. For sorting equipment this is the elapsed time from receipt of "Start Tray Sort" to the issue of a " Tray Sort Complete"

product - Any unit which is intended to become a functional semiconductor device including multichip devices. This includes functional engineering devices.

setup (conversion) - The sum of:

- Conversion: The time required to complete an equipment alteration necessary to accommodate a change in process, product, package configuration, etc. (excluding modifications, rebuilds, and upgrades).
- Equipment test: The operation of equipment to demonstrate equipment functionality; (e.g., system loads trays, transfers devices, reads bar code labels, etc.).
- Verification run: The processing and evaluation of devices after conversion to establish that equipment is performing its intended function within specifications.

- with the understanding that the kit or process had been successfully installed (initial setup) on the piece of equipment and the times for conversion, equipment test, and process test in the definition are the times for change from running one installed setup to another.

note : Equipment suppliers are responsible for providing procedures which achieve setup conversion and testing within predetermined specifications. Users are obligated to identify any deviation from the procedures if they expect the supplier to make setups fall within those specifications.

setup (initial) - The time required to install a kit or process for the first time on a piece of equipment in the field, with the understanding that the times for conversion, equipment test, and process test presume a previously designed/developed and proven kit and or process have been initially installed.

shutdown - The time required to put the equipment in a safe condition when entering a nonscheduled state. It includes any procedures necessary to reach a safe condition. Shutdown is only included in nonscheduled time.

size (device) - the length, width and height (or diameter and height) of an individual device in millimeters. Unit size specifications are applicable to product, piece parts and media.

skew - a special case (for devices with peripheral input/output) of the true position tolerance defined in ANSI Y14.5. skew is the worst case variation of an individual device I/O foot/tip centerline from the ideal or theoretically perfect location. "Per device" skew is the skew of the one I/O on the device which has the worst case deviation from the theoretically perfect location.

sort time - The elapsed time from receipt of a tray at the processing site to when that tray is moved out of that site, not including any time to move trays to the sorting site. It is only the actual sort transport time. The overhead time to prepare the sorting site would be contained in processing time.

standoff - the distance between the actual unit I/O seating plane and lowest point on the unit body measured perpendicular to the actual seating plane. In actual practice standoff is often measure relative to the unit theoretical or regression plane seating plane (see coplanarity definition.) The means by which the lowest point on the unit body is determined also vary depending on the type of measuring equipment used.

start up - The time required for equipment to achieve a condition where it can perform its intended function, when leaving a non–scheduled state. It includes pump down, warm up, cool down, stabilization periods, initialization routines, etc. Start up is only included in nonscheduled time.

support tool - A mechanical device that, although not part of a piece of equipment, is required by, and becomes integral with it during the course of normal operation (e.g., trays, cassettes, wafer carriers, probe cards).

sweep - variation of skew tolerance that references the center of the unit body or the center of opposing side I/0 (lead) forms as datum(s) in determining the ideal unit I/O centerlines. "Per unit" sweep is the sweep of the one I/O on the unit which has the worst case deviation from the theoretically perfect location.

time - all time measurements, calculations and results will be stated in hours: minutes: seconds format unless otherwise specifically defined in the individual test methods and calculations.

throughput - number of units that pass through the processing system expressed in units per hour with the processing conditions at specified values.

total time - All time (at the rate of 24 hours./day, 7 days/week) during the period being measured. In order to have a valid representation of total time, all three basic equipment states must be accounted for and tracked accurately.

Training (off-line) - The instruction of personnel in the operation and/or maintenance of equipment done outside of operations time. Off–line training is only included in non–scheduled time.

Training (on-the-job) - The instruction of personnel in the operation and/or maintenance of equipment done during the course of normal work functions. On–the–job training typically does not interrupt operation or maintenance activities and can therefore be included in any equipment state (except standby and non–scheduled) without special categorization.

unit (device) - Any wafer, die, packaged device, multichip module, or piece part thereof (includes product and non–product units).

unit input / output (deviceI/0) - Inclusive term for the physical feature(s) on the unit which permit electrical contact/connection with circuits external to the unit. Includes "leads", "balls", "columns", "tabs".

unload time - elapsed time from the sorter receiving an "end of sort" signal from the processing system to the point where the equipment places that tray in its appropriate output bin.

uptime (equipment uptime) - the hours when the equipment is in a condition to perform its intended function. It includes productive, standby, and engineering time, and does not include any portion of non–scheduled time

verification run - any unit or units (product or non–product) processed by the equipment to establish that it is performing its intended function within specifications.

Appendix B

KINFLEX SPREADSHEET

The following pages are a copy of KinFlex.xls spreadsheet generated to calculate individual compliance of a set of flexural bearings used on the Flexural Kinematic Coupling design.

The first part of the spreadsheet was designed by Prof. Alexander Slocum (Mechanical Engineering - MIT) to calculate stiffness of a set of kinematic couplings (ball and groove arrangement) through Hertzian contact stress analysis. The continuation of the spreadsheet implements the equations described on "***Enabling Technology***" chapter of this thesis.

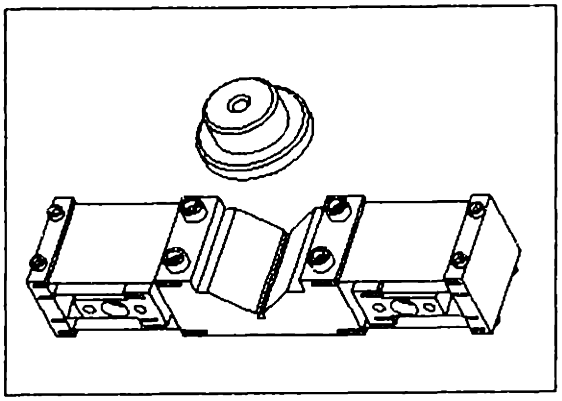
The equations for a set of serial and parallel flexural bearings can be very cumbersome to calculate by hand, also spreadsheet allows for simpler optimization of the design given the input parameters. The serial flexural bearings allows for bigger displacement for a given applied force and limiting stress, and the parallel bearings permit increased rotational stiffness along the symmetry axis of the groove and also linear displacement of the groove without beam axial load (combination of parallel flexures and floating bar, see chapter for explanation).

KINFLEX

To design three groove kinematic couplings with linear displacement
Original Kinematic Coupling spreadsheet written by Dr. Alexander H. Slocum
Kinematic Coupling Flexural bearing design spreadsheet written by Luis A. Muller

Only change cells with	**blue**	boldface numbers
Results are on cells with	**red**	boldface numbers

XY plane is assumed to contain the ball centers
For standard coupling designs, contact forces are inclined at 45 to the XY plane



Generic data entry for 120 degree couplings

Standard 120 degree equal size groove coupling? **TRUE**
For non standard designs, enter geometry after results section

Dbeq=	**0.024**	Equivelent diameter ball that would contact the groove at the same points
Rbminor =	**0.012**	Minor radius 0.011904762
Rbmajor =	**1.500**	Major radius #NUM!
Rgroove =	**1.0E+06**	Groove radius (negative for a trough)
Costheta =	**FALSE**	Is ball major radius along groove axis?
Dcoupling =	**0.50**	Coupling diameter
Fpreload 1 =	333	Preload force over ball 1 (N)
Fpreload 2 =	333	Preload force over ball 2 (N)
Fpreload 3 =	333	Preload force over ball 3 (N)
Xerr =	**0.000**	X location of error reporting
Yerr =	**0.000**	Y location of error reporting
Zerr =	**0.000**	Z location of error reporting

Auto select material values assume that metric units are used (mks)

Matlab =	4	Enter 1 for plastic ball, plastic groove
		Enter 2 for Steel ball, plastic groove

Material properties	
SiN	6.8E+09
plastic	1.0E+08
RC 62 Steel	1.8E+09
Elastic modulus	
SiN	3.1E+11
plastic	1.0E+10
RC 62 Steel	2.0E+11
Poisson ratio	
SiN	0.27
plastic	0.2
RC 62 Steel	0.29

Enter 3 for SiN ball, RC 62 Fe groove
Enter 4 for RC 62 Fe ball, RC 62 Fe groove
Enter 5 for other values and enter them for each ball and groove

Min. yield strength (Pa, psi) =	1.03E+09	1.50E+05

Actuating Forces

External force =	**1000**	[N]	x =	**0**	[mm]	0.00
			y =	**0**	[mm]	0.00
			z =	**0**	[mm]	0.00
% of external force =	**100**	Percent of external force to be used to actuate KinFlex [N]				1000
Weight =	**0**	[kg]	Xw =	**0**	[mm]	0.00
			Yw =	**0**	[mm]	0.00
			Zw =	**0**	[mm]	0.00

Z,Y,Z values and coordinates					**Coupling centroid**	
FLx =	0	XL =	0.000	[m]	xc	0.000
FLy =	0	YL =	0.000	[m]	yc	0.000
FLz =	0	ZL =	0.000	[m]	zc	0.000

Results: Hertz stresses and deformations

Error displacements at the point of interest (micron)

DeltaX	0.00E+00	DeltaY	0.00E+00	DeltaZ	0.00E+00	0.00E+00

			Ball-Groove 1				*Contact ellipse*	
Groove normal forces		*Contact stress*		*Stress/Allow*	*Deflection (+into ball)*		*Major radius*	*Minor radius*
Fbnone	***-2E+02***	sigone	***#NUM!***	***#NUM!***	delone	***0E+00***	***#NUM!***	***#NUM!***
Fbntwo	***-2E+02***	sigtwo	***#NUM!***	***#NUM!***	deltwo	***0E+00***	***#NUM!***	***#NUM!***
			Ball-Groove 2				*Contact ellipse*	
Groove normal forces		*Contact stress*			*Deflection*		*Major radius*	*Minor radius*
Fbnthree	***-2E+02***	sigthree	***#NUM!***	***#NUM!***	delthree	***0E+00***	***#NUM!***	***#NUM!***
Fbnfour	***-2E+02***	sigfour	***#NUM!***	***#NUM!***	delfour	***0E+00***	***#NUM!***	***#NUM!***
			Ball-Groove 3				*Contact ellipse*	
Groove normal forces		*Contact stress*			*Deflection*		*Major radius*	*Minor radius*
Fbnfive	***-2E+02***	sigfive	***#NUM!***	***#NUM!***	delfive	***0E+00***	***#NUM!***	***#NUM!***
Fbnsix	***-2E+02***	sigsix	***#NUM!***	***#NUM!***	delsix	***0E+00***	***#NUM!***	***#NUM!***

Results: Error motions

Error motions are at X,Y,Z coordinates:				**0.000**	**0.000**	**0.000**
deltaX	**0.00**	[um]				
deltaY	**0.00**	[um]	*Homogenous Transformation Matrix:*			
deltaZ	**0.00**	[um]	**1.0E+00**	**0.0E+00**	**0.0E+00**	**0.0E+00**
EpsX	**0.00**	[um]	**0.0E+00**	**1.0E+00**	**0.0E+00**	**0.0E+00**
EpsY	**0.00**	[um]	**0.0E+00**	**0.0E+00**	**1.0E+00**	**0.0E+00**
EpsZ	**0.00**	[um]	**0.0E+00**	**0.0E+00**	**0.0E+00**	**1.0E+00**

Generic data entry for non-120 degree couplings

NOTE! For calculation of angular errors, the coupling is assumed to lie in the XY plane.
Ball 1 must lie in quadrants 1 Or 2, and Balls 2 & 3 must lie in quadrants 3 and 4

Enter X,Y,Z coordinates and alpha, beta, gamma direction cosines for Ball 1

Contact point 1		Contact point 2	
Xba =	**0.00849**	Xbb =	**-0.00849**
Yba =	**0.25000**	Ybb =	**0.25000**
Zba =	**-0.00849**	Zbb =	**-0.00849**
Aba =	**-0.70711**	Abb =	**0.70711**
Bba =	**0.00000**	Bbb =	**0.00000**
Gba =	**0.70711**	Gbb =	**0.70711**

Enter characteristics for groove 1 and ball 1

Egone =	**2.0E+11**	Groove material elastic modulus
vgone =	**0.29**	Groove material Poisson ratio
Rgone =	**1000000**	Groove radius of curvature
Ebone =	**2.0E+11**	Pin material elastic modulus
vbone =	**0.29**	Pin material Poisson ratio
Eeone =	1.11E+11	Equivelent modulus
Dbone =	**0.024**	Equivelent diameter ball that would contact the groove at the same points
Rpone =	**1.500**	"Ball" major radius of curvature
Raone =	**0.012**	"Ball" minor radius of curvture
Reone =	0.012	Equivelent radius
ctone =	0.984126972	Cos(theta)
theta_1 =	0.178	
alpha_1 =	6.497	
beta_1 =	0.311	
lambda_1 =	0.324	
Sone =	**1.8E+09**	Allowable Hertz stress

Enter X,Y,Z coordinates and alpha, beta, gamma direction cosines for Ball 2

Contact point 3		Contact point 4	
Xbc =	**-0.22075**	Xbd =	**-0.21226**
Ybc =	**-0.11765**	Ybd =	**-0.13235**
Zbc =	**-0.00849**	Zbd =	**-0.00849**
Abc =	**0.35355**	Abd =	**-0.35355**
Bbc =	**-0.61237**	Bbd =	**0.61237**
Gbc =	**0.70711**	Gbd =	**0.70711**

Enter characteristics for groove 2 and pin 2

Egtwo =	**2.0E+11**	Groove material elastic modulus
vgtwo =	**0.29**	Groove material Poisson ratio
Rgtwo =	**1000000**	Groove radius of curvature
Ebtwo =	**2.0E+11**	Pin material elastic modulus

vbtwo =	0.29	Pin material Poisson ratio
Eetwo =	1.11E+11	Equivelent modulus
Dbtwo =	0.024	Equivelent diameter ball that would contact the groove at the same points
Rptwo =	1.500	"Ball" major radius of curvature
Ratwo =	0.012	"Ball" minor radius of curvture
Retwo =	0.012	Equivelent radius
cttwo =	0.984126972	Cos(theta)
theta_2 =	0.178	
alpha_2 =	6.497	
beta_2 =	0.311	
lambda_2 =	0.324	
Stwo =	1.8E+09	Allowable Hertz stress

Enter X,Y,Z coordinates and alpha, beta, gamma direction cosines for Ball 3

Contact point 5		Contact point 6	
Xbe =	0.21226	Xbf =	0.22075
Ybe =	-0.13235	Ybf =	-0.11765
Zbe =	-0.00849	Zbf =	-0.00849
Abe =	0.35355	Abf =	-0.35355
Bbe =	0.61237	Bbf =	-0.61237
Gbe =	0.70711	Gbf =	0.70711

Enter characteristics for groove 3 and ball 3

Egthree =	2.0E+11	Groove material elastic modulus
vgthree =	0.29	Groove material Poisson ratio
Rgthree =	1000000	Groove radius of curvature
Ebthree =	2.0E+11	Pin material elastic modulus
vbthree =	0.29	Pin material Poisson ratio
Eethree =	1.11E+11	Equivelent modulus
Dbthree =	0.024	Equivelent diameter ball that would contact the groove at the same points
Rpthree =	1.500	"Ball" major radius of curvature
Rathree =	0.012	"Ball" minor radius of curvture
Rethree =	0.012	Equivelent radius
ctthree =	0.984126972	Cos(theta)
theta_3 =	0.178	
alpha_3 =	6.497	
beta_3 =	0.311	
lambda_3 =	0.324	
Sthree =	1.8E+09	Allowable Hertz stress

Preload forces' X,Y,Z components and coordinates

Fpxone =	0	Fpxtwo =	0	Fpxthree =	0
Fpyone =	0	Fpytwo =	0	Fpythree =	0
Fpzone =	333.3333333	Fpztwo =	333.3	Fpzthree =	333.3
Xpone =	0.000	Xptwo =	-0.217	Xpthree =	0.217
Ypone =	0.250	Yptwo =	-0.125	Ypthree =	-0.125
Zpone =	0.048	Zptwo =	0.048	Zpthree =	0.048

Calculations

Build Force Moment equilibrium matrices: AF = B (Equations 1-6)

Matrix A

Fbn1	Fbn2	Fbn3	Fbn4	Fbn5	Fbn6
-7.07E-01	7.07E-01	3.54E-01	-3.54E-01	3.54E-01	-3.54E-01
0.00E+00	0.00E+00	-6.12E-01	6.12E-01	6.12E-01	-6.12E-01
7.07E-01	7.07E-01	7.07E-01	7.07E-01	7.07E-01	7.07E-01
1.77E-01	1.77E-01	-8.84E-02	-8.84E-02	-8.84E-02	-8.84E-02
0.00E+00	0.00E+00	1.53E-01	1.53E-01	-1.53E-01	-1.53E-01
1.77E-01	-1.77E-01	1.77E-01	-1.77E-01	1.77E-01	-1.77E-01

Matrix F

	B with loads	B w/o loads	
Fbn1	0.00E+00	0.00E+00	Sum forces X direction
Fbn2	0.00E+00	0.00E+00	Sum forces Y direction
Fbn3	-1.00E+03	-1.00E+03	Sum forces Z direction
Fbn4	-2.84E-14	-2.84E-14	Sum moments X axis
Fbn5	0.00E+00	0.00E+00	Sum moments Y axis
Fbn6	0.00E+00	0.00E+00	Sum moments Z axis

Res. Forces with applied loads		Res forces with preload only	
fbnone	-235.70	fone	-235.70
fbntwo	-235.70	ftwo	-235.70
fbnthree	-235.70	fthree	-235.70
fbnfour	-235.70	ffour	-235.70
fbnfive	-235.70	ffive	-235.70
fbnsix	-235.70	fsix	-235.70

Original ball corrdinates

xboneO	0.0000000	xbtwoO	-0.216506	xbthreeO	0.2165064
yboneO	0.2500000	ybtwoO	-0.125000	ybthreeO	-0.1250000
zboneO	0.0000000	zbtwoO	0.000000	zbthreeO	0.0000000

New ball coordinates (=original + ball deflection*direction cosines)

xboneN	0.000000	xbtwoN	-0.216506	xbthreeN	0.2165064
yboneN	0.2500000	ybtwoN	-0.125000	ybthreeN	-0.1250000
zboneN	0.0000000	zbtwoN	0.000000	zbthreeN	0.0000000

Ball centers' deflections

dxone	0.00E+00	dxtwo	0.00E+00	dxthree	0.00E+00	0	1.3194E-07
dyone	0.00E+00	dytwo	0.00E+00	dythree	0.00E+00	0	4.3509E-08
dzone	0.00E+00	dztwo	0.00E+00	dzthree	0.00E+00	0	4.3509E-08

Theory applicability check

Initial dist. between balls		Final dist. between balls		Difference	
LotI	0.433013	LotN	0.433013	DLotI	0.00E+00

LttI	0.433013	LttN	0.433013	DLttI	0.00E+00
LtoI	0.433013	LtoN	0.433013	DLtoI	0.00E+00

Change in length/distance between balls	Deflection/ball radius	Ratio (should be >5)
0.00E+00	0.00E+00	#DIV/0!
0.00E+00	0.00E+00	#DIV/0!
0.00E+00	0.00E+00	#DIV/0!

Coupling centroid is assumed to be at intersection of coupling triangle's angle bisectors

Initial centroid		Dist. from ball to centroid		Error motion at centroid from weighted ball motions	
xci	0.0000000	Dcone	0.2500000	dxc	0.00E+00
yci	0.0000000	Dctwo	0.2500000	dyc	0.00E+00
zci	0.0000000	Dcthree	0.2500000	dzc	0.00E+00

Original angles between balls			Original altitude lengths		
Angone	60.0000	angle at ball 1	Aone	0.3750	Ball 1 to side 2 3
Angtwo	60.0000	angle at ball 2	Atwo	0.3750	Ball 2 to side 1 3
Angthree	60.0000	angle at ball 3	Athree	0.3750	Ball 3 to side 2 1

New angles between balls			Original sides' angle with X axis		
AngoneN	60.0000	angle at ball 1	Aot	60	Side opposite ball 3
AngtwoN	60.0000	angle at ball 2	Att	0	Side opposite ball 1
AngthreeN	60.0000	angle at ball 3	Ato	120	Side opposite ball 2

New sides' angle with X axis

AotN	60	Side opposite ball 3	0
AttN	0	Side opposite ball 1	0
AtoN	120	Side opposite ball 2	

Original altitudes' slope angles and Y intercepts

AmtwoO	30	AbtwoO	-1.38778E-17
AmthreeO	150	AbthreeO	1.11022E-16

Rotation about opposite side (radians)

Ttt	0.00E+00	rotation about side 23 due to Z motion at ball 1
Tto	0.00E+00	rotation about side 13 due to Z motion at ball 2
Tot	0.00E+00	rotation about side 12 due to Z motion at ball 3

Coupling error rotations

EpsX	0.00E+00	EpsZ1	0.00E+00	Z rot from ball 1
EpsY	0.00E+00	EpsZ2	0.00E+00	Z rot from ball 2
EpsZ	0.00E+00	EpsZ3	0.00E+00	Z rot from ball 3

Coupling HTM					Point of interest
1.00E+00	0.00E+00	0.00E+00	0.00E+00	Xerr	5.96354E-17
0.00E+00	1.00E+00	0.00E+00	0.00E+00	Yerr	4.51028E-17
0.00E+00	0.00E+00	1.00E+00	0.00E+00	Zerr	0
0.00E+00	0.00E+00	0.00E+00	1.00E+00	1	1

Error displacements at the point of interest

DeltaX	0.00E+00
DeltaY	0.00E+00
DeltaZ	0.00E+00

Generic data entry for Flexural bearing calculations

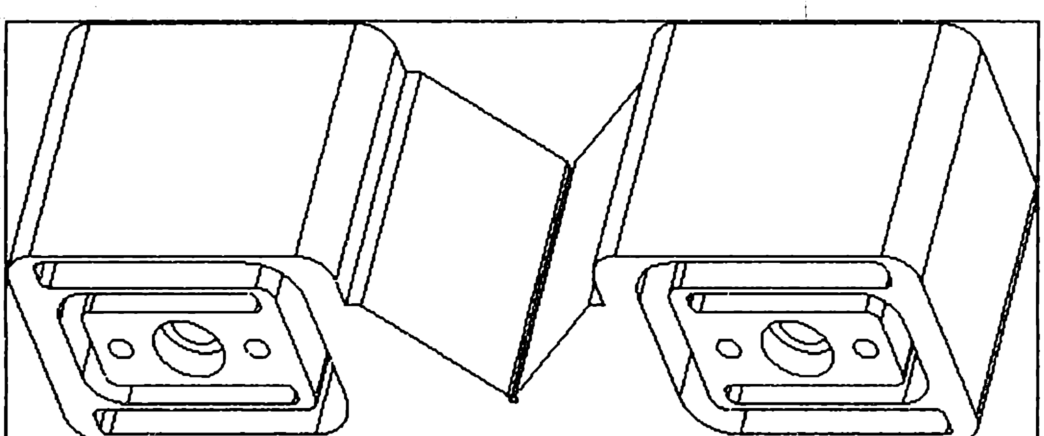
Material Information: Stainless Steel 403, 410 or 416 Hardened Tempered at 800 F

Youngs Modulus:	**2.00E+11**	[Pa]
Shear Modulus:	**7.74E+10**	[Pa]
Yield strength:	**1.03E+09**	[Pa]
Poisson's ration:	**0.291**	
Hardness:	**C 41**	

Force applied on each bearing:

Flexure 1 force:	333	[N]	Observation: Flexure 1 is positioned on the y-axis
Flexure 2 force:	333	[N]	
Flexure 3 force:	333	[N]	

TYPE I



E =	2.00E+11	Youngs Modulus [Pa]		
S (yield) =	1.03E+09	Yield strength [Pa]	1.03E+09	
Weight =	0	Weight of the structure [kg]		
Width =	**20.0**	Width of KinFlex [mm]		2.0E-02
Disp. =	**1.50**	Desired axial displacement [mm]		1.5E-03

Flexural Bearing 1

Load per bearing =	83.3	[N]
Length (1st aproximation) =	26.4	[mm]

Minimum thickness =	0.80	[mm]		
Thickness (project) =	**0.80**	[mm]	{ has to be bigger than Minimum thickness }	
Length (project) =	26.4	[mm]	2.6E-02	
Displacement for weight =	0.00	[mm]	Wone =	0

Bearing 1 - Lateral Stiffness calculation

Individual Compliance =	8.25E-09	
Total Compliance =	4.13E-09	
Lateral Stiffness =	242	[N/um]

Bearing 1 - Radial Stiffness calculation

Individual Compliance =	2.88E-08	
Total Compliance =	7.20E-09	
Radial Stiffness =	139	[N/um]

Flexural Bearing 2

Load per bearing =	83.3	[N]
Length (1st aproximation) =	26.4	[mm]
Minimum thickness =	0.80	[mm]

Thickness (project) =	**0.80**	[mm]	{ has to be bigger than Minimum thickness }	
Length (project) =	26.4	[mm]	2.5E-02	
Displacement for weight =	0.00	[mm]	Wtwo =	0

Bearing 2 - Lateral Stiffness calculation

Individual Compliance =	8.25E-09	
Total Compliance =	4.13E-09	
Lateral Stiffness =	242	[N/um]

Bearing 2 - Radial Stiffness calculation

Individual Compliance =	2.88E-08	
Total Compliance =	7.20E-09	
Radial Stiffness =	139	[N/um]

Flexural Bearing 3

Load per bearing =	83.3	[N]
Length (1st aproximation) =	26.4	[mm]
Minimum thickness =	0.80	[mm]

Thickness (project) =	0.80	[mm]	{ has to be bigger than Minimum thickness }	
Length (project) =	26.4	[mm]	2.6E-02	
Displacement for weight =	0.00	[mm]	Wthree =	0

Bearing 3 - Lateral Stiffness calculation

Individual Compliance =	8.25E-09	
Total Compliance =	4.13E-09	
Lateral Stiffness =	242	[N/um]

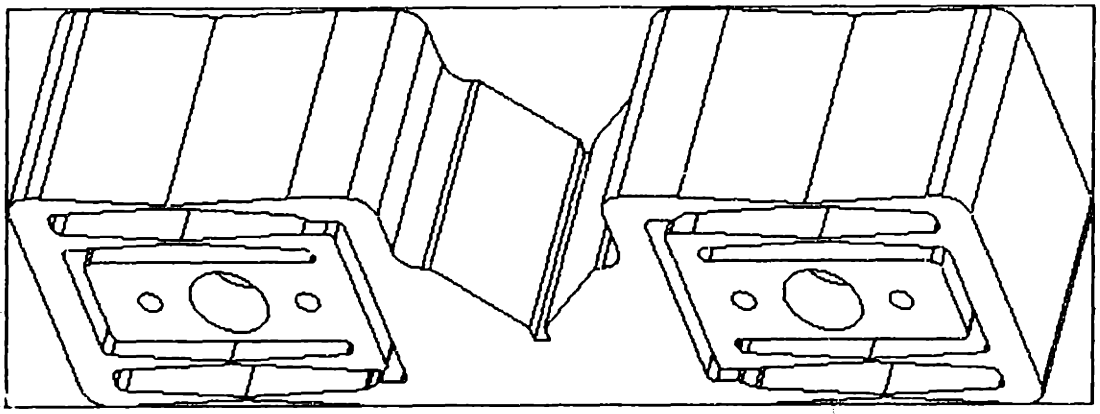
Bearing 2 - Radial Stiffness calculation

Individual Compliance =	2.88E-08	
Total Compliance =	7.20E-09	
Radial Stiffness =	139	[N/um]

Design Results:

	KinFlex 1		KinFlex 2	KinFlex 3	
Web thickness =	0.80		0.80	0.80	[mm]
Web length =	26.4		26.4	26.4	[mm]
Displacement for weight =	0.00		0.00	0.00	[mm]
Z-Stiffness =	0.22		0.22	0.22	[N/um]
System Axial Stiffness =	0.67	[N/um]			
System Radial Stiffness =	678	[N/um]			
System Lateral Stiffness =	182	[N/urad]			

<u>*TYPE II*</u>



E =	2.00E+11	Youngs Modulus [Pa]	
S (yield) =	1.03E+09	Yield strength [Pa]	
Weight =	0	Weight of the structure [kg]	
s =	**0.40**	Center thickness [mm]	4.0E-04
Width =	**20.0**	Width of KinFlex [mm]	2.0E-02
Disp. =	**1.50**	Desired axial displacement [mm]	1.5E-03

Flexural Bearing 1

Load per bearing =	83.3	[N]			
Length (1st aproximation) =	20.2	[mm]	2.0E-02	-1.4E-14	
Minimum thickness =	0.70	[mm]			
Thickness (project) =	**0.70**	[mm]	{ has to be bigger than Minimum thickness }		
Length (project) =	20.2	[mm]	2.0E-02	-1.7E-12	
Displacement for weight =	0.00	[mm]	Wone =	0	3.0E-02

<u>Bearing 1 - Lateral Stiffness calculation</u>

A =	-0.029677	Const-1 =	-8.42E-09	
B =	0.029677	Const-2 =	8.42E-09	
C =	0.000100	Individual Compliance =		9.43E-09
		Total Compliance =		4.71E-09
Lateral Stiffness =	212	[N/um]		

<u>Bearing 1 - Radial Stiffness calculation</u>

K=	0.029677		Const-1 =	3.41E-02	
			Individual Compliance =		1.67E-08
			Total Compliance =		4.17E-09
Radial Stiffness =		240	[N/um]		

Flexural Bearing 2

Load per bearing =	83.3	[N]			
Length (1st aproximation) =	20.2	[mm]	2.0E-02	-1.4E-14	
Minimum thickness =	0.70	[mm]			
Thickness (project) =	**0.70**	[mm]	(has to be bigger than Minimum thickness)		
Length (project) =	20.2	[mm]	2.0E-02	-1.9E-13	
Displacement for weight =	0.00	[mm]	Wone =	0	3.0E-02

<u>Bearing 2 - Lateral Stiffness calculation</u>

A =	-0.029677		Const-1 =	-8.42E-09	
B =	0.029677		Const-2 =	8.42E-09	
C =	0.000100		Individual Compliance =		9.43E-09
			Total Compliance =		4.71E-09
Lateral Stiffness =		212	[N/um]		

<u>Bearing 2 - Radial Stiffness calculation</u>

K=	0.029677		Const-1 =	3.41E-02	
			Individual Compliance =		1.67E-08
			Total Compliance =		4.17E-09
Radial Stiffness =		240	[N/um]		

Flexural Bearing 3

Load per bearing =	83.3	[N]			
Length (1st aproximation) =	20.2	[mm]	2.0E-02	-1.4E-14	
Minimum thickness =	0.70	[mm]			
Thickness (project) =	**0.70**	[mm]	(has to be bigger than Minimum thickness)		
Length (project) =	20.2	[mm]	2.0E-02	-1.9E-13	
Displacement for weight =	0.00	[mm]	Wone =	0	3.0E-02

<u>Bearing 3 - Lateral Stiffness calculation</u>

A =	-0.029677	Const-1 =	-8.42E-09

B = 0.029677		Const-2 = 8.42E-09	
C = 0.000100		Individual Compliance =	9.43E-09
		Total Compliance =	4.71E-09
Lateral Stiffness =	212	[N/um]	

Bearing 3 - Radial Stiffness calculation

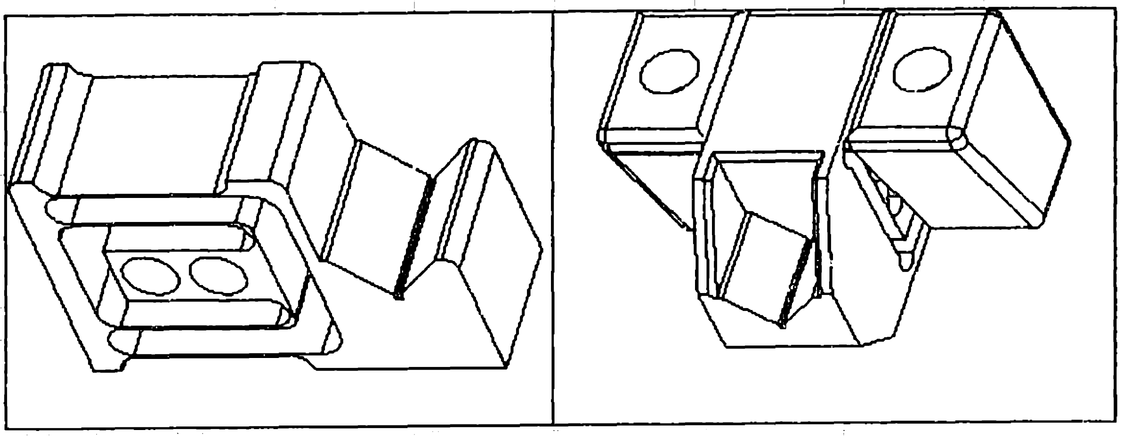
K= 0.029677		Const-1 = 3.41E-02	
		Individual Compliance =	1.67E-08
		Total Compliance =	4.17E-09
Radial Stiffness =	240	[N/um]	

Design Results:

	KinFlex 1	KinFlex 2	KinFlex 3	
Web thickness =	0.70	0.70	0.70	[mm]
Web length =	20.2	20.2	20.2	[mm]
Displacement for weight =	0.00	0.00	0.00	[mm]
Axial Stiffness =	0.22	0.22	0.22	[N/um]

System Axial Stiffness =	0.67	[N/um]
System Radial Stiffness =	879	[N/um]
System Lateral Stiffness =	159	[N/urad]

TYPE III



E =	2.00E+11	Youngs Modulus [Pa]	
S (yield) =	1.03E+09	Yield strength [Pa]	
Weight =	0	Weight of the structure [kg]	
Width =	**20.0**	Width of KinFlex [mm]	2.0E-02
Disp. =	**1.50**	Desired axial displacement [mm]	1.5E-03

Flexural Bearing 1

Load per bearing =	166.7	[N]	
Length (1st aproximation) =	33.2	[mm]	
Minimum thickness =	1.27	[mm]	
Thickness (project) =	**1.30**	[mm]	{ has to be bigger than Minimum thickness }
Length (project) =	34.1	[mm]	3.4E-02
Displacement for weight =	0.00	[mm]	

<u>Bearing 1 - Lateral Stiffness calculation</u>

Individual Compliance =	6.55E-09	
Total Compliance =	6.55E-09	
Lateral Stiffness =	153	[N/um]

<u>Bearing 1 - Radial Stiffness calculation</u>

Individual Compliance =	3.80E-08	
Total Compliance =	1.90E-08	
Radial Stiffness =	53	[N/um]

Flexural Bearing 2

Load per bearing =	166.7	[N]	
Length (1st eproximation) =	33.2	[mm]	
Minimum thickness =	1.27	[mm]	
Thickness (project) =	**1.30**	[mm]	{ has to be bigger than Minimum thickness }
Length (project) =	34.1	[mm]	3.4E-02
Displacement for weight =	0.00	[mm]	

<u>Bearing 2 - Lateral Stiffness calculation</u>

Individual Compliance =	6.55E-09	
Total Compliance =	6.55E-09	
Lateral Stiffness =	153	[N/um]

<u>Bearing 2 - Radial Stiffness calculation</u>

Individual Compliance =	3.80E-08	
Total Compliance =	1.90E-08	
Radial Stiffness =	53	[N/um]

Flexural Bearing 3

Load per bearing =	166.7	[N]	
Length (1st eproximation) =	33.2	[mm]	
Minimum thickness =	1.27	[mm]	
Thickness (project) =	**1.30**	[mm]	{ has to be bigger than Minimum thickness }
Length (project) =	34.1	[mm]	3.4E-02
Displacement for weight =	0.00	[mm]	

<u>Bearing 3 - Lateral Stiffness calculation</u>

Individual Compliance =	6.55E-09	
Total Compliance =	6.55E-09	
Lateral Stiffness =	153	[N/um]

<u>Bearing 3 - Radial Stiffness calculation</u>

Individual Compliance =	3.80E-08	
Total Compliance =	1.90E-08	
Radial Stiffness =	53	[N/um]

Design Results:

	KinFlex 1		KinFlex 2	KinFlex 3	
Web thickness =	1.30		1.30	1.30	[mm]
Web length =	34.1		34.1	34.1	[mm]
Displacement for weight =	0.00		0.00	0.00	[mm]
Z-Stiffness =	0.22		0.22	0.22	[N/um]
System Axial Stiffness =	0.67	[N/um]			
System Radial Stiffness =	343	[N/um]			
System Lateral Stiffness =	114	[N/urad]			

Appendix C

STEWARD PLATFORM INVERSE KINEMATICS

The InverseKinematics.xls spreadsheet was developed to calculate inverse kinematics of a Steward platform type parallel manipulator and together with another spreadsheet MotionAnalysis.xls (not presented here) they were used to perform a time analysis of each motion of the TwinHandler concept introduced on "***Introduction and Thesis Overview***" chapter of this thesis.

This information was fundamental in evaluating the concept and its capabilities as well as determining the potential market value of the design given its performance characteristics.

The Steward platform which is constructed by connecting two plates to six adjustable legs and is a six-degree-of-freedom 6-SPS platform mechanism (S and P denote spherical and prismatic joints, respectively), was originally designed as an aircraft simulator, and was also suggested for the applications of machine tool, space behicle simulator, transfer machine, etc. [Yang '84].

C.1 Displacement Analysis

Initially some definitions of controllable elements and desired outputs of the mechanism are needed to describe the equations used for the spreadsheet presented (see figure C.1)

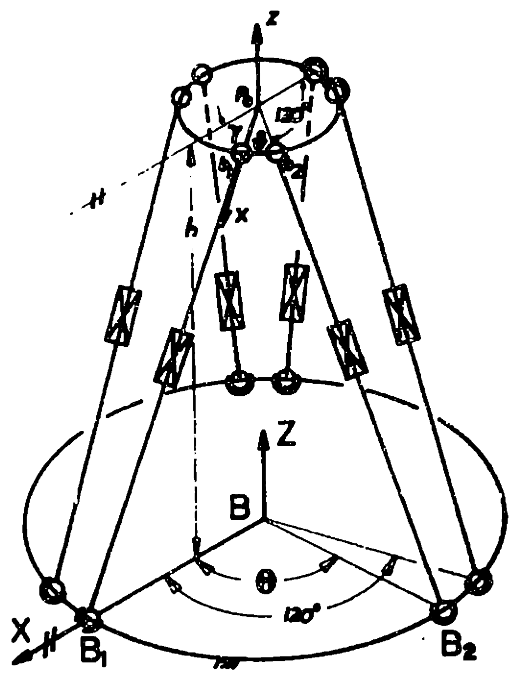


Figure C. 1 - 6-SPS platform mechanism and its design parameters

First, let the controlling of the position of the center point, p, of the top plate be the desired objective. Both top and bottom plates are circular in shape with the top one assumed to be movable and the bottom one fixed and having its center at B. Furthermore, let the distribution of each pair of ball joints on the bottom plate be symmetrical with respect to each of the tree radii located at 120 deg. Apart from one another on the plate. The ball joints on the top plate have similar arrangements. The platform mechanism is then an octahedron. Denoting R and r as the radii of the bottom and the top plates, respectively; l_1 through l_6 as the leg lengths; l_0 as the nominal length for all legs; and h as the distance between the two plates when all leg lengths are equal to the nominal length l_0.

To give an analytical representation of the kinematics of the 6-SPS mechanism, the fixed Cartesian coordinate frame is selected at point B with the Z axis pointing vertically upward and the X axis passing through point B_1. The location of the ball joints on the fixed bottom plate can be established.

$$B_1 = [R,0,0]^T$$

$$B_2 = [RC_\theta, RS_\theta,0]^T$$

$$B_3 = [RC_{120}, RS_{120},0]^T$$

$$B_4 = [RC_{120+\theta}, RS_{120+\theta},0]^T$$

$$B_5 = [RC_{240}, RS_{240},0]^T$$

$$B_6 = [RC_{240+\theta}, RS_{240+\theta},0]^T \quad \text{(C1)}$$

where θ represents the angle between ball joints B_1 and B_2, B_3,and B_4, and B_5,and B_6; and C_θ and S_θ represent the *cos* θ and *sin* θ, respectively.

Similarly, let a moving Cartesian coordinate frame $[T]_P$ be associated with the top plate, having its origin at the point p, its z axis normal to the plate, and the x axis passing through the point b_1. Therefore, the relative locations of the ball joints, b_i's with respect to the moving frame are:

$$b_1 = [r,0,0]^T$$

$$b_2 = [rC_\phi, rS_\phi,0]^T$$

$$b_3 = [rC_{120}, rS_{120},0]^T$$

$$b_4 = [rC_{120+\phi}, rS_{120+\phi},0]^T$$

$$b_5 = [rC_{240}, rS_{240},0]^T$$

$$b_6 = \left[rC_{240+\phi}, rS_{240+\phi}, 0\right]^T \tag{C2}$$

where ϕ denotes the angle between ball joints b_1 and b_2, b_3, and b_4, and b_5, and b_6.

Let the point p_0 and the Cartesian coordinate $[T]p_0$ be the position of p when the top plate is at its original (the nominal). The geometrical relationship between the coordinate frame $[T]p_0$ and the fixed coordinate frame [B] can be represented by a 4x4 homogeneous transformation:

$$[T]p_0 = \begin{bmatrix} \cos\gamma & -\sin\gamma & 0 & 0 \\ \sin\gamma & \cos\gamma & 0 & 0 \\ 0 & 0 & 1 & h \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{C3}$$

where $\gamma = \theta - \phi/2$.

Assuming that the center of the top plate moves away from its nominal positon p_0 to a new desired position p and have the transformation $[T]p$ with respect to the fixed coordinate system [B]. Let

$$[T]p_0 = \begin{bmatrix} d_{11} & d_{12} & d_{13} & x_p \\ d_{21} & d_{22} & d_{23} & y_p \\ d_{31} & d_{32} & d_{33} & z_p \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{C4}$$

Where the (x_p, y_p, z_p) are the Cartesian locations of the point p, and (d_{11}, d_{21}, d_{31}), (d_{12}, d_{22}, d_{32}), (d_{13}, d_{23}, d_{33}) are the direction cosines of the axes x, y and z with respect to the fixed coordinate [B], correspondingly. Therefore, the location of each ball joint on the moving top plate with respect to the coordinate frame [B] can be obtained from equations (C2) and (C4), and they are:

$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}_{b1} = [T]_p \begin{bmatrix} r \\ 0 \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} d_{11}r + x_p \\ d_{21}r + y_p \\ d_{31}r + z_p \\ 1 \end{bmatrix}$$

$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}_{b2} = [T]_p \begin{bmatrix} rC_\phi \\ rS_\phi \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} d_{11}rC_\phi + d_{12}rS_\phi + x_p \\ d_{21}rC_\phi + d_{22}rS_\phi + y_p \\ d_{31}rC_\phi + d_{32}rS_\phi + z_p \\ 1 \end{bmatrix}$$

$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}_{b3} = [T]_p \begin{bmatrix} rC_{120} \\ rS_{120} \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} d_{11}rC_{120} + d_{12}rS_{120} + x_p \\ d_{21}rC_{120} + d_{22}rS_{120} + y_p \\ d_{31}rC_{120} + d_{32}rS_{120} + z_p \\ 1 \end{bmatrix}$$

$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}_{b4} = [T]_p \begin{bmatrix} rC_{\phi+120} \\ rS_{\phi+120} \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} d_{11}rC_{\phi+120} + d_{12}rS_{\phi+120} + x_p \\ d_{21}rC_{\phi+120} + d_{22}rS_{\phi+120} + y_p \\ d_{31}rC_{\phi+120} + d_{32}rS_{\phi+120} + z_p \\ 1 \end{bmatrix}$$

$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}_{b5} = [T]_p \begin{bmatrix} rC_{240} \\ rS_{240} \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} d_{11}rC_{240} + d_{12}rS_{240} + x_p \\ d_{21}rC_{240} + d_{22}rS_{240} + y_p \\ d_{31}rC_{240} + d_{32}rS_{240} + z_p \\ 1 \end{bmatrix}$$

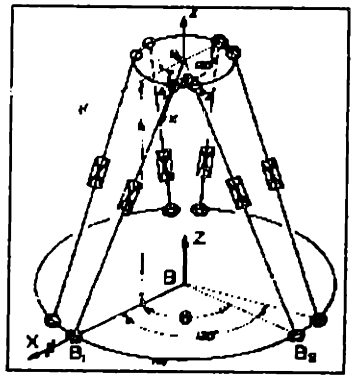
$$\begin{bmatrix} x \\ y \\ z \\ 1 \end{bmatrix}_{b6} = [T]_p \begin{bmatrix} rC_{\phi+240} \\ rS_{\phi+240} \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} d_{11}rC_{\phi+240} + d_{12}rS_{\phi+240} + x_p \\ d_{21}rC_{\phi+240} + d_{22}rS_{\phi+240} + y_p \\ d_{31}rC_{\phi+240} + d_{32}rS_{\phi+240} + z_p \\ 1 \end{bmatrix}$$

INVERSE KINEMATICS

A solution for the inverse kinematics of a Hexapod
Inverse kinematic spreadsheet written by Luis A. Muller

Only change cells with	**blue**	boldface numbers
Results are on cells with	**red**	boldface numbers

Rest frame is assumed to be fixed at the center of and with XY plane parallel to the base plate
Coordinate system is assumed to be fixed at the center of and with xy plane parallel to the top plate
Assume Euler description of orientation angles



Collapsed height calculation for a Hexapod

Construction data

R =	**10.0**	[inches]	Base plate radius
r =	**7.0**	[inches]	Top plate radius
theta =	**90.0**	[inches]	Angle between adjacent U-joints on base plate
qsi =	**40.0**	[inches]	Angle between adjacent U-joints on top plate

Cartesian coordinate of farthest position:

X_max. =	0.0	**0.0**	[inches]	X-coordinate of end-effector
Y_max. =	0.0	**0.0**	[inches]	Y-coordinate of end-effector
(Z-H) =	0.0	**0.0**	[inches]	(Z-H)-coordinate of end-effector
alpha =	0.0	**0.0**	[degrees]	rotation angle about x-axis
beta =	0.0	**0.0**	[degrees]	rotation angle about y-axis
gama =	0.0	**0.0**	[degrees]	rotation angle about z-axis
gamaR =	25.0	[degrees]	rotation angle about Z-axis when centered	
Ld =	**4.6**	[inches]	dead lenght of the linear actuator	

Direction cossines of end-effector coordinate system relative to rest frame

d11 =	0.91	d21 =	0.42	d31 =	0.00
d12 =	-0.42	d22 =	0.91	d32 =	0.00
d13 =	0.00	d23 =	0.00	d33 =	1.00

Lengths of the hexapod legs for the especified spatial position & orientation

L1 =	**4.7**	[inches]
L2 =	**4.7**	[inches]
L3 =	**4.7**	[inches]
L4 =	**4.7**	[inches]
L5 =	**4.7**	[inches]
L6 =	**4.7**	[inches]
Height =	**0.0**	[inches]
Stroke =	**-0.5**	[inches]

Design personification of a Hexapod

Construction data

R =	**10.0**	[inches]	Base plate radius
r =	**7.0**	[inches]	Top plate radius
theta =	**90.0**	[inches]	Angle between adjacent U-joints on base plate
qsi =	**40.0**	[inches]	Angle between adjacent U-joints on top plate
H =	**11.0**	[inches]	Collapsed height of the hexapod

Spatial position & orientation of the end-effector

Xp =	**3.5**	[inches]	X-coordinate of end-effector
Yp =	**9.0**	[inches]	Y-coordinate of end-effector
(Z-H)p =	**2.0**	[inches]	Z-coordinate of end-effector
alpha =	**0.0**	[degrees]	rotation angle about x-axis
beta =	**0.0**	[degrees]	rotation angle about y-axis
gama =	**0.0**	[degrees]	rotation angle about z-axis
gamaR =	25.0	[degrees]	rotation angle about Z-axis when centered

Direction cossines of end-effector coordinate system relative to rest frame

d11 =	0.91	d21 =	0.42	d31 =	0.00
d12 =	-0.42	d22 =	0.91	d32 =	0.00
d13 =	0.00	d23 =	0.00	d33 =	1.00

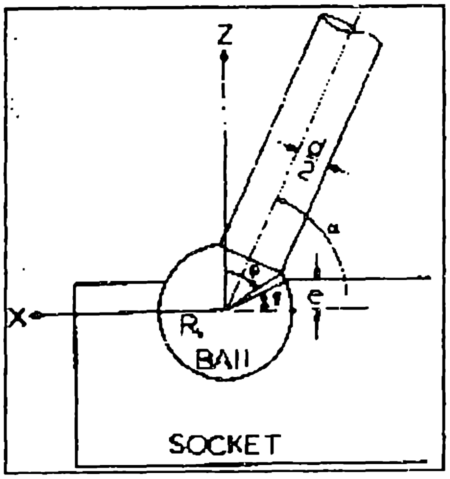
Lengths of the hexapod legs for the especified spatial position & orientation

	Collapsed	Extended	Stroke	
L1 =	**12.5**	**17.7**	**5.2**	[inches]
L2 =	**11.6**	**15.5**	**3.9**	[inches]
L3 =	**12.5**	**14.0**	**1.5**	[inches]
L4 =	**11.6**	**19.4**	**7.8**	[inches]
L5 =	**12.5**	**18.6**	**6.1**	[inches]
L6 =	**11.6**	**15.4**	**3.9**	[inches]

A solution for the rotatibility of Ball-and-Socket Joint of a Hexapod

Written by Luis A. Muller

Only change cells with	**blue**	boldface numbers
Results are on cells with	**red**	boldface numbers



Design inputs for the Ball-and-Socket Joint

Rb =	**50**	[mm]	Ball radius
d =	**20**	[mm]	Connecting leg diameter
e =	**4**	[mm]	Holding width

163.9 **< alpha <** **16.1**

Function Leg1(d11, rp, x, rb, d21, y, d31, z, H)

'Calculates leg length #1 when on the extended position

Leg1 = ((d11 * rp + x - rb) ^ 2 + (d21 * rp + y) ^ 2 + (d315 * rp + z + H)^2) ^ 0.5

End Function

Function Leg2(d11, rp, qsi, d12, x, rb, theta, d21, d22, y, d31, d32, z, H)

'Calculates leg length #2 when on the extended position

```
Leg2 = ((d11 * rp * Cos(qsi * 3.1416 / 180) + d12 * rp * Sin(qsi * 3.1416 / 180) + x - rb * Cos(theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(qsi * 3.1416 / 180) + d22 * rp * Sin(qsi * 3.1416 / 180) + y - rb * Sin(theta * 3.1416 / 180)) ^ 2 + (d31 * rp * Cos(qsi * 3.1416 / 180) + d32 * rp * Sin(qsi * 3.1416 / 180) + z + H) ^ 2) ^ 0.5
```

End Function

Function Leg3(d11, rp, d12, x, rb, d21, d22, y, d31, d32, z, H)

'Calculates leg length #3 when on the extended position

```
Leg3 = ((d11 * rp * Cos(120 * 3.1416 / 180) + d12 * rp * Sin(120 * 3.1416 / 180) + x - rb * Cos(120 * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(120 * 3.1416 / 180) + d22 * rp * Sin(120 * 3.1416 / 180) + y - rb * Sin(120 * 3.1416 / 180)) ^ 2 + (d31 * rp * Cos(120 * 3.1416 / 180) + d32 * rp * Sin(120 * 3.1416 / 180) + z + H) ^ 2) ^ 0.5
```

End Function

Function Leg4(d11, rp, qsi, d12, x, rb, theta, d21, d22, y, d31, d32, z, H)

'Calculates leg length #4 when on the extended position

```
Leg4 = ((d11 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + d12 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + x - rb * Cos(120 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + d22 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + y - rb * Sin(120 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 +(d31 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + d32 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + z + H) ^ 2) ^ 0.5
```

End Function

Function Leg5(d11, rp, d12, x, rb, d21, d22, y, d31, d32, z, H)

'Calculates leg length #5 when on the extended position

Leg5 = ((d11 * rp * Cos(240 * 3.1416 / 180) + d12 * rp * Sin(240 * 3.1416 / 180) + x - rb * Cos(240 * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(240 * 3.1416 / 180) + d22 * rp * Sin(240 * 3.1416 / 180) + y - rb * Sin(240 * 3.1416 / 180)) ^ 2 +(d31 * rp * Cos(240 * 3.1416 / 180) + d32 * rp * Sin(240 * 3.1416 / 180) + z + H) ^ 2) ^ 0.5

End Function

Function Leg6(d11, rp, qsi, d12, x, rb, theta, d21, d22, y, d31, d32, z, H)

'Calculates leg length #6 when on the extended position

Leg6 = ((d11 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + d12 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + x - rb * Cos(240 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 +(d21 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + d22 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + y - rb * Sin(240 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 +(d31 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + d32 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + z + H) ^ 2) ^ 0.5

End Function

Function Height(L1, L2, L3, L4, L5, L6, rb, rp, theta, qsi, x, y, z, alpha, beta, gama, gamaR, Ld, d11, d12, d13, d21, d22, d23, d31, d32, d33

*'Calculates the heights for which the extended_leg = (2*collapsed_leg - dead length of the actuator)*

'It initially calculates the leg lenghts for H=0 and then increments H until desired heights

'The heights are independently calculated for each leg and then the biggest one is chosen at the end

H = 0

Collapsed_leg = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (H) ^ 2) ^ 0.5

```
'Does the calculation for leg #1

Extended_leg1 = ((d11 * rp + x - rb) ^ 2 + (d21 * rp + y) ^ 2 + (d31 * rp + z + H) ^ 2) ^ 0.5

If Extended_leg1 > (2 * Collapsed_leg - Ld) Then

  For a = 1 To 1000 Step 1

    Collapsed_leg1 = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (a) ^ 2) ^ 0.5

    Extended_leg1 = ((d11 * rp + x - rb) ^ 2 + (d21 * rp + y) ^ 2 + (d31 * rp + z + a) ^ 2) ^ 0.5

    If Extended_leg1 <= (2 * Collapsed_leg1 - Ld) Then

      Height1 = a

      Exit For

    End If

  Next

Else Height1 = 0

End If

'Does the calculation for leg #2

Extended_leg2 = ((d11 * rp * Cos(qsi * 3.1416 / 180) + d12 * rp * Sin(qsi * 3.1416 / 180) + x - rb * Cos(theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(qsi * 3.1416 / 180) + d22 * rp * Sin(qsi * 3.1416 / 180) +y - rb * Sin(theta * 3.1416 / 180)) ^ 2 + (d31 * rp * Cos(qsi * 3.1416 / 180) + d32 * rp * Sin(qsi * 3.1416 / 180) + z + H) ^ 2) ^ 0.5

If Extended_leg2 > (2 * Collapsed_leg - Ld) Then

  For b = 1 To 1000 Step 1

    Collapsed_leg2 = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (b) ^ 2) ^ 0.5

    Extended_leg2 = ((d11 * rp * Cos(qsi * 3.1416 / 180) + d12 * rp * Sin(qsi * 3.1416 / 180) + x - rb * Cos(theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(qsi * 3.1416 / 180) + d22 * rp * Sin(qsi * 3.1416 / 180) + y - rb * Sin(theta * 3.1416 / 180)) ^ 2
```

```
+ (d31 * rp * Cos(qsi * 3.1416 / 180) + d32 * rp * Sin(qsi * 3.1416 / 180) + z + b) ^ 2) ^
0.5

            If Extended_leg2 <= (2 * Collapsed_leg2 - Ld) Then

              Height2 = b

              Exit For

            End If

          Next

        Else Height2 = 0

        End If

        'Does the calculation for leg #3

        Extended_leg3 = ((d11 * rp * Cos(qsi * 3.1416 / 180) + d12 * rp * Sin(qsi
* 3.1416 / 180) + x - rb * Cos(theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(qsi * 3.1416 /
180) +d22 * rp * Sin(qsi * 3.1416 / 180) + y - rb * Sin(theta * 3.1416 / 180)) ^ 2 +
(d31 * rp * Cos(qsi * 3.1416 / 180) + d32 * rp * Sin(qsi * 3.1416 / 180) + z + H) ^ 2) ^
0.5

        If Extended_leg3 > (2 * Collapsed_leg - Ld) Then

          For c = 1 To 1000 Step 1

            Collapsed_leg3 = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (c) ^ 2) ^ 0.5

            Extended_leg3 = ((d11 * rp * Cos(qsi * 3.1416 / 180) + d12 * rp *
Sin(qsi * 3.1416 / 180) + x - rb * Cos(theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(qsi *
3.1416 / 180) +d22 * rp * Sin(qsi * 3.1416 / 180) + y - rb * Sin(theta * 3.1416 / 180)) ^ 2
+ (d31 * rp * Cos(qsi * 3.1416 / 180) + d32 * rp * Sin(qsi * 3.1416 / 180) + z + c) ^ 2) ^
0.5

            If Extended_leg3 <= (2 * Collapsed_leg3 - Ld) Then

              Height3 = c

              Exit For

            End If
```

```
Next

Else Height3 = 0

End If

'Does the calculation for leg #4

Extended_leg4 = ((d11 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + d12 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + x - rb * Cos(120 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) +d22 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + y - rb * Sin(120 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 + (d31 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) +  d32 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + z + H) ^ 2) ^ 0.5

If Extended_leg4 > (2 * Collapsed_leg - Ld) Then

 For d = 1 To 1000 Step 1

  Collapsed_leg4 = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (d) ^ 2) ^ 0.5

  Extended_leg4 = ((d11 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) +d12 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + x - rb* Cos(120 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 +(d21 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) +d22 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + y - rb * Sin(120 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 + (d31 * rp * Cos(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + d32 * rp * Sin(120 * 3.1416 / 180 + qsi * 3.1416 / 180) + z + d) ^ 2) ^ 0.5

  If Extended_leg4 <= (2 * Collapsed_leg4 - Ld) Then

   Height4 = d

   Exit For

  End If

 Next

Else Height4 = 0

End If
```

'Does the calculation for leg #5

Extended_leg5 = ((d11 * rp * Cos(240 * 3.1416 / 180) + d12 * rp * Sin(240 * 3.1416 / 180) + x - rb * Cos(240 * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(240 * 3.1416 / 180) + d22 * rp * Sin(240 * 3.1416 / 180) + y - rb * Sin(240 * 3.1416 / 180)) ^ 2 +(d31 * rp * Cos(240 * 3.1416 / 180) + d32 * rp * Sin(240 * 3.1416 / 180) + z + H) ^ 2) ^ 0.5

If Extended_leg5 > (2 * Collapsed_leg - Ld) Then

For e = 1 To 1000 Step 1

Collapsed_leg5 = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (e) ^ 2) ^ 0.5

Extended_leg5 = ((d11 * rp * Cos(240 * 3.1416 / 180) + d12 * rp * Sin(240 * 3.1416 / 180) + x - rb * Cos(240 * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(240 * 3.1416 / 180) + d22 * rp * Sin(240 * 3.1416 / 180) + y - rb * Sin(240 * 3.1416 / 180)) ^ 2 +(d31 * rp * Cos(240 * 3.1416 / 180) + d32 * rp * Sin(240 * 3.1416 / 180) + z + e) ^ 2) ^ 0.5

If Extended_leg5 <= (2 * Collapsed_leg5 - Ld) Then

Height5 = e

Exit For

End If

Next

Else Height5 = 0

End If

'Does the calculation for leg #6

Extended_leg6 = ((d11 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) +d12 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + x - rb * Cos(240 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 + (d21 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) +d22 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + y - rb * Sin(240 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 +(d31 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + d32 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + z + H) ^ 2) ^ 0.5

```
If Extended_leg6 > (2 * Collapsed_leg - Ld) Then

   For f = 1 To 1000 Step 1

      Collapsed_leg6 = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (f) ^ 2) ^ 0.5

      Extended_leg6 = ((d11 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) +d12 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + x - rb * Cos(240 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 +(d21 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) +d22 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + y - rb * Sin(240 * 3.1416 / 180 + theta * 3.1416 / 180)) ^ 2 + (d31 * rp * Cos(240 * 3.1416 / 180 + qsi * 3.1416 / 180) +d32 * rp * Sin(240 * 3.1416 / 180 + qsi * 3.1416 / 180) + z + f) ^ 2) ^ 0.5

      If Extended_leg6 <= (2 * Collapsed_leg6 - Ld) Then

         Height6 = f

         Exit For

      End If

   Next

Else Height6 = 0

End If

'Calculates the biggest height

Height = Height1

For y = 1 To 6 Step 1

   If Height2 > Height Then

      Height = Height2

      ElseIf Height3 > Height Then

         Height = Height3

      ElseIf Height4 > Height Then
```

```
            Height = Height4
        ElseIf Height5 > Height Then
            Height = Height5
        ElseIf Height6 > Height Then
            Height = Height6
        End If
    Next
End Function
```

Function Stroke(rp, rb, H, L1, L2, L3, L4, L5, L6)

'Calculates the maximum stroke among the six strokes of each leg for the maximum reach position

```
    Collapsed = ((0.87 * rp - rb) ^ 2 + (0.5 * rp) ^ 2 + (H) ^ 2) ^ 0.5
    'Calculates stroke for leg #1
        Stroke1 = L1 - Collapsed
    'Calculates stroke for leg #2
        Stroke2 = L2 - Collapsed
    'Calculates stroke for leg #3
        Stroke3 = L3 - Collapsed
    'Calculates stroke for leg #4
        Stroke4 = L4 - Collapsed
    'Calculates stroke for leg #5
        Stroke5 = L5 - Collapsed
```

```
'Calculates stroke for leg #6
    Stroke6 = L6 - Collapsed
'Calculates the biggest stroke
    Stroke = Stroke1
For u = 1 To 6 Step 1
   If Stroke2 > Stroke Then
      Stroke = Stroke2
      ElseIf Stroke3 > Stroke Then
         Stroke = Stroke3
      ElseIf Stroke4 > Stroke Then
         Stroke = Stroke4
      ElseIf Stroke5 > Stroke Then
         Stroke = Stroke5
      ElseIf Height6 > Stroke Then
         Stroke = Stroke6
   End If
Next
End Function
```