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AXIOMATIC REDESIGN OF GUIDE FLEXURES: IMPROVING PRODUCT RELIABILITY AND REDUCING MANUFACTURING COST

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1. ABSTRACT

Axiomatic design was the guiding principle in the redesign of a set of motion constraint flexures used in the Micrascan photolithography tool. The original flexures experienced fatigue failure in the field and a replacement solution with infinite life was desired. In addition, the fabrication and inspection of the original flexures was difficult, resulting in high cost and quality fluctuations.

The Independence Axiom and system decomposition techniques were applied to derive functional requirements from the existing hardware, in order to guarantee that the redesign would yield a functionally identical solution that would not adversely affect system performance. The Information Axiom was used to select among the proposed solutions in order to improve manufacturability.

The chosen solution was tested and is being implemented. Test results show that the desired life expectancy improvement has been obtained with no impact on system performance and procurement cost reduction of about 60%.

2. INTRODUCTION

The need for pre-alignment

The Micrascan photolithography tool's function is to use ultraviolet (UV) light to project a reduced image of a pattern inscribed in a "mask" onto a photoresist coated wafer. UV Light alters the chemistry of the resist enabling etching of the features copied from the mask. The features thus reproduced on the wafer become the device and wiring layers of the chip. A typical chip requires many layers, and therefore many passes through one or more photolithography tools. Due to the microscopic dimensions in today's chips (less than 0.250 μm), connectivity between layers demands extremely fine alignment from one layer to the next. This is called "overlay". A sophisticated alignment system is needed to achieve satisfactory overlay. It comprises means of aligning the wafer and mask relative to each other, as well as coarse, intermediate and fine alignment means for the wafer itself. In order to fall within the narrow capture range of the intermediate level, a relatively high degree of accuracy is required from the coarse level. In addition, the first pattern printed on a wafer relies solely on the coarse alignment system, since there are no pre-existing features on the wafer to fine align to. The coarse level of wafer alignment is accomplished in a device called the pre-aligner. Its function is to determine the wafer's centroid location within a few tens of μm, and to re-orient the wafer to a predetermined angle within a few hundred microradians.

How the pre-aligner works

Micrascan's pre-aligner is shown in Figure 1. A servomotor spins the wafer under a radially mounted linear detector array that maps the wafer's edge. The edge data (instantaneous edge distance from the spin axis) is fed to a computer, which calculates the offset of the wafer's centroid from the spin axis, and the angular orientation of a notch at the wafer's perimeter. The computer then commands the motor to align the notch to a pre-determined orientation. The pre-aligner interfaces with a simple robot that loads the incoming wafer, and with a "flip arm", which transfers the coarse-aligned wafer onto the wafer stage. The flip arm operates between the horizontal plane of the pre-aligner and the vertical plane of the stage, hence its name. The stage performs the intermediate and fine alignment and exposes the wafer by scanning it under the actinic light.

3. PRE-ALIGNER ANALYSIS USING DECOMPOSITION AND THE INDEPENDENCE AXIOM

In order to prevent introducing undesirable side effects by arbitrarily altering the design, the pre-aligner was decomposed down to the flexure level. The decomposition and application of the independence axiom are discussed in the following sections. The accompanying figures show the relevant hardware as originally designed.

OPERATION

Since the transfer and the alignment occur at different planes, it is imperative that the motion between planes be extremely

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repeatable, or alignment accuracy is lost. For this reason, a set of two flexures, arranged in a parallelogram set-up was chosen. They constrain the motor-encoder section of the pre-aligner to a very repeatable, vertical arcuate path. Hard stops comprising adjustment screws that contact hard buttons located on the coupler link establish the upper and lower limits of travel. The pneumatic actuator powers the vertical motion via a dual swivel-jointed connecting rod. Air pressurization, moves the coupler link upward and venting allows gravity to move it downward. Wafer perimeter mapping is performed in the upper position, with the wafer very close to the detector array. Wafer loading/unloading is done in the lower position. The described arc is symmetric; thus there is nominally no horizontal displacement between the loading and the alignment positions. More importantly, the small horizontal displacement that would result from imperfect link geometry is constant from wafer to wafer and therefore easy to compensate by a one time adjustment of the pick-up position of the flip arm.

Decomposition of the wafer handler system

Index	Functional Requirement	Design Parameter					
2.1	Determine the wafer rotational orientation and centroid offsets prior to loading onto stage	Wafer pre-aligner					
2.2	{Automate wafer exchange with track machine}t_1 {Load/unload wafers into tool manually}t_2	Input/Output doors					
2.3	Provide additional buffer station for wafers	Park station					
2.4	Move wafer between stations and onto stage	Robotic mechanism					
2.5	Schedule and coordinate all system functions	Wafer handler Command and Control Algorithm					
2.6	Integrate wafer handler subassemblies	Wafer handler support framework					
Design Constraints			**Impacts FR 2.__**				
Index	Parent	Description	1	2	3	4	5	6
Critical Performance Specifications								
C2-1	C-2	Minimize wafer exchange time	Y	Y	Y	Y	Y	Y
Operational Constraints								
C2-2	C-6	Accommodate input position tolerance provided by track machines	Y	Y		Y	Y	Y
Global Constraints								
C2-3	C-6	Handle customer-provided 200mm wafers	Y	Y	Y	Y	Y	Y
C2-4	C-10 – C-20	Accommodate all higher level global constraints	Y	Y	Y	Y	Y	Y

Note: this paper only addresses the analysis of the constraints relating to FR 2a-1, which corresponds to the pre-aligner.

FIRST LEVEL DECOMPOSITION

Decomposition of the pre-aligner

x	Functional Requirement FR2.1.x	Design Parameter DP2.1.x	Design Matrix	
1	Map wafer perimeter	Motor-encoder and detector array	**X**	$t_{1,2}$
2	Accommodate wafer transfer	Vertical motion setup	**0**	**X**

$t_{1,2}$: Unintended horizontal wafer motion relative to the detector array reduces wafer perimeter mapping accuracy. Lack of horizontal positioning repeatability of the spinner section from wafer to wafer also compromises pre-alignment and transfer accuracy.

?	Parent	Constraint C2.1-?	Impacts FR2.1.x	
			1	2
1	C2a.1	Must be compatible with robot and flip arms		Y
2	C2a.1	Must cost less than present design	Y	Y

It is evident from the point of view of the independence axiom that the original vertical motion setup is a good solution. It is a de-coupled design and performs the intended function of moving the mo Copyright © #### by ICAD2000tor-encoder up and down introducing very minimal horizontal positioning uncertainty (it is nearly uncoupled). For example, if a linear slide were used to replace the linkage, play and wear in the linear bearing would diminish the pre-aligner's accuracy. Likewise, if the flexures in the linkage were replaced by rotary bearings, play and wear would also result in an increased degree of coupling.

Further analysis of other options lead us to conclude that the flexure linkage was more effective at minimizing $t_{1,2}$ than any other alternative we could envision. Since $t_{1,2}$ inversely affects pre-alignment accuracy, retaining the use of flexures was unquestionably justified on this basis. Narrowing down the alternatives early on allowed us to concentrate our effort on re-designing the flexures.

SECOND LEVEL DECOMPOSITION

Decomposition of the vertical motion setup

x	Functional Requirement	Design Parameter DP2.1.2.x	Design Matrix

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	FR2.1.2.x			
1	Guide motor-encoder motion	Parallel flexure linkage	**X**	**$u_{1,2}$=0**
2	Power vertical stroke	Pneumatic actuator with swiveling connecting rod	**$u_{2,1}$**	**X**

$u_{1,2}$=0: Since connecting rod has a dual swivel, the actuator piston does not apply sideways forces that may interfere with motion guidance by the flexure linkage.

$u_{2,1}$: This element represents the first mode stiffness of the flexure. A stiffer flexure would require a higher force actuator.

Figure 1 The Pre Aligner mounted on a test stand

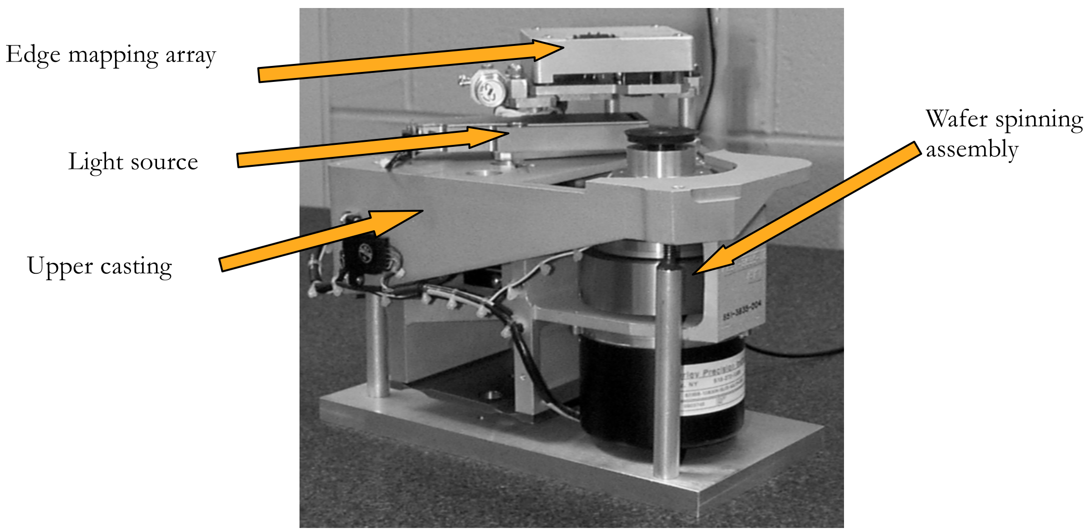

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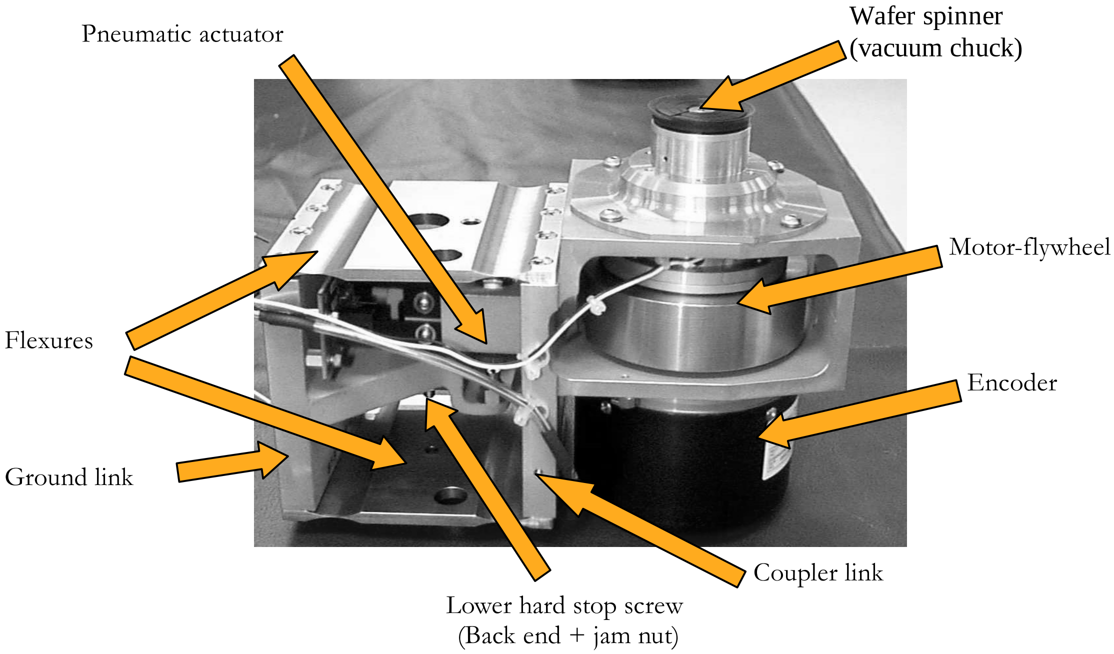
Figure 2 Pre-aligner: Wafer spinning and vertical tranlation subassembly



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?	Parent	Constraint C2.1.2-?	Impacts FR2.2.x	
			1	2
1	C2.1.1	Redesign hardware must be backward compatible	Y	Y

Once again, the original designers had produced a decoupled design. The decomposition made it evident that $u_{2,1}$ in the new design should be less than or equal to $u_{2,1}$ in the old design, in order to avoid having to replace the existing actuators in the field with more powerful ones capable of deflecting the stiffer flexures. It also became apparent that if the hinges in the new flexures were less stiff than the original, the resonant frequency of the guided assembly in the vertical mode of vibration would be lower. Although the actuator and the upper hard stop constrain the motor-encoder assembly fairly rigidly at the upper end of the stroke, the situation is worse at the lower end, where the motor-encoder assembly just rests on the lower hard stop. Wary of introducing a vertical vibration mode by decreasing the hinge stiffness, we decided to match it, leaving $u_{2,1}$ the same.

THIRD LEVEL DECOMPOSITION

<u>Decomposition of the new flexure</u>

x	Functional Requirement FR2.1.2.1.x	Design Parameter DP2.1.2.1.x	Design Matrix		
1	Match 1st mode (vertical) stiffness	Flex-hinges with combined stiffness equal to original design	**X**	$v_{1,2}=0$	$v_{1,3}=0$
2	Maximize longevity (choose hinge type with low maximum stress to stiffness ratio)	Beam-type flex-hinges	$v_{2,1}$	**X**	$v_{2,3}=0$
3	Maximize 2nd and higher resonant frequencies (space the beams widely along the flexure's width)	Stiffness concentrated at corners	$v_{3,1}$	$v_{3,2}$	**X**

$v_{1,2}=0$: By design. Only considering solutions with same 1st mode stiffness as the original.

$v_{1,3}=0$: Spacing of the beams along width does not affect 1st mode stiffness.

$v_{2,1}$: Stiffness matching limits the extent to which maximum stress can be lowered.

$v_{2,3}=0$: Spacing of the beams along width does not affect longevity (maximum stress).

$v_{3,1}$: Limiting the bending stiffness of the flex-hinges to match the vertical stiffness of the original flexure reduces the resonant frequency of the torsional mode about the longitudinal horizontal axis of the guide linkage.

$v_{3,2}$: Lower bending stress requires longer beam or smaller cross section. Both tend to elongate and buckle at lower resonant frequencies in the horizontal pendulum-like transverse mode and in the horizontal piston-like longitudinal mode.

?	Parent	Constraint C2.1.2.1-?	Impacts FR2.2.x		
			1	2	3
1	C2.1.2.1	Same overall dimensions as original design	Y	Y	Y
2	C2.1.2.1	Same mounting hole pattern as original design			Y
3	C2.1.2.1	Flex-hinge beam thickness (depth) must be at least .015”	Y	Y	Y

The first two constraints insure that the new flexure will fit the pre-aligner without the need to modify any other parts (a bolt-on replacement for the original design).

The third constraint limits the ratio of the tolerance (0.0005”) to thickness to one part in 30. Since the maximum stress is proportional to the thickness, the variation in stress is constrained to one part in 30 or less. Also, since the beam stiffness is proportional to the cube of the thickness, the variation in beam stiffness due to manufacturing tolerance cannot vary by more than about 10%. $(1+1/30)^3 = 1.103$. The magnitude of the variations determined by this constraint were deemed acceptable based on the overall sensitivity to change in resonant frequency and on the safety factor used to compute the maximum allowable operating stress.

The independence axiom analysis pointed out the difficulty in achieving very high resonant frequencies for the guide linkage due to the coupling elements $v_{3,1}$ and $v_{3,2}$. Fortunately, the original flexures, which also suffered from similar coupling, were wide enough for us to allow sufficient transverse separation between the beams to overcome the adverse effect of the coupling without having to increase the flexure width.

A set of dimensions for the flexures that met the three functional requirements was obtained as follows.

1. The longitudinal spacing between flex-hinges was set to the maximum allowable by the length of the original flexures and the location of the mounting screws, in order to minimize the angular deflection and bending stress.

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2. The beam thickness (depth) required to make the maximum operating stress equal to 0.7 of the fatigue limit of the material (the allowable operating stress for infinite life) was calculated to be 0.018". This also satisfied constraint # 3.

3. Since the design was under-determined, we somewhat arbitrarily set the span of the beam flex-hinges equal to ¼ inch. The length to thickness ratio was about 14, large enough for bending stresses to dominate and for the flex hinge to behave as a beam.

4. The beam width required to match the bending stiffness of the original flexures was calculated. This was a fraction of the total flexure width, allowing a cutout to separate each flex hinge in two segments as planned. (DP2a.1.2.1.3)

5. Finite Element vibration analysis of the original and redesigned pre-aligner mass-spring system was performed up to the fourth mode. It showed that the 2nd and higher resonant frequencies of the redesigned pre-aligner were above 250Hz, comparable to the original design and high enough.

4. FLEXURE ANALYSIS USING THE INFORMATION AXIOM

The independence axiom led to a solution that met all functional requirements. At that point, at least two ways to construct the beam flexures were envisioned. The level of manufacturing difficulty of either one would probably be lower than that of the original design.

The information axiom was used to choose objectively between the original and the two implementations of the redesigned beam flexure on the basis of lowest information content.

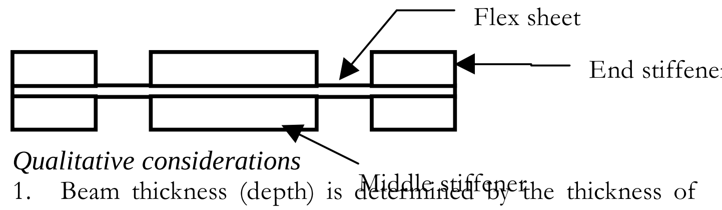
INFORMATION CONTENT ANALYSIS

1. The information content was computed from the number of dimensions required to produce the simplest possible implementation of a complete flexure of each design, having all the design parameters and meeting all the constraints.
2. Each dimension was classified as being either "critical" or "non-critical", based the sensitivity impact on the design, and the difficulty of achieving the manufacturing tolerance.
3. Each non-critical dimension was assigned a weight of 1 point, and each of the critical dimensions was given a weight of 10 points. The design with the lowest total point count would have the lowest information content.

TWO EMBODIMENTS OF THE REDESIGNED FLEXURE: BUILT-UP VS. MONOLITHIC

The beam flexure could be built up from a thin sheet of spring steel sandwiched between thicker aluminum stiffeners, using screws to hold it together. It could also be machined in one piece from a thick metal plate.
The information content of each of the proposed designs was assessed as shown below:

Built-up design



Qualitative considerations

1. Beam thickness (depth) is determined by the thickness of the flexing sheet, as rolled in the steel mill with excellent accuracy and repeatability, therefore involving only a low information content for the present design.
2. Clamping action between stiffener and flexing sheet is critical to the sheet's longevity, since relative motion at the microscopic level causes fretting. Fretting is a form of surface abrasion due to cyclic relative motion of extremely small amplitude and is usually accompanied by surface corrosion leading to loss of clamping hold and premature failure. Such a critical interface would be information rich. From a design perspective, a uniform clamping pressure requires many screws. In order to take advantage of the mounting screws also for clamping, geometry and stress analysis suggested the use of a total of 9 screws at each clamping edge.
3. Transition between flexing and rigid sections of the flexure requires a fillet to reduce stress concentration. This dimension was deemed non-critical

Part and dimension count for the built-up design

Part name	Qty Req'd	Size Dim's (L,W,T)	Number of Holes = # H	Dimensions for holes (D, X,Y pos'n) # H x 3	Number of Fillets = # F	Points per part	Total Points
Flex Sheet	1	3	36	108	0	111	111
End Stiffener	4	3	9	27	1	31	124
Middle Stiffener	2	3	18	54	2	59	118
Total	**7**						**353**

Maximum stress is determined by one stock dimension (flex sheet thickness)

Monolithic design

Qualitative considerations

1. Beam thickness has to be machined and is critical to the stiffness and stress. Since the beam has substantial width, its thickness should be measured at a minimum of two points.

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Therefore, high information content, and a factor of two are associated with the thickness dimension in the analysis.

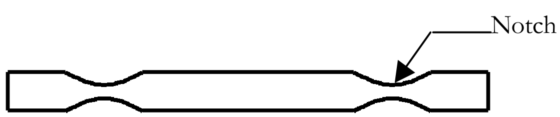
2. Surface roughness of the flex-hinge is critical to part life, since fatigue cracks will initiate at any scars left by machining and grinding operations. Therefore it also has high information content.
3. Transition between flexing and rigid sections of the flexure requires a fillet to reduce stress concentration. This dimension was deemed non-critical

Feature and dimension count for the monolithic design

Feature name	Qty	Dimensions for feature	D = # of dim's	N = # of meas't loc's	Non-critical or Critical K= 1 or 10	D x N x K	Total Points
Overall shape	1	L, W, T	3	1	1	3	3
Mounting hole	8	D, X,Y location	3	1	1	3	24
Cutout	2	L, W, 4 x cor.R, X,Y loc'n	8	1	1	8	16
Beam (trans. radius)	4	2 x R	2	1	1	2	8
Beam (thickness)		T	1	2	10	20	80
Beam (surf. finish)		SF	1	1	10	10	40
Total	**15**						**171**

THE MONOLITHIC BEAM FLEXURE VERSUS THE ORIGINAL DOUBLE NOTCH FLEXURE

Information content of the double notch flexure



Qualitative considerations

1. Hinge line must be machined, by notching both sides of the plate. There are three dimensions associated with each notch: arc radius, and x,y location of arc center. Therefore it takes a total of six dimensions to determine the thickness at the hinge line. The stress and stiffness sensitivity to the thickness is large, therefore the dimensions are critical.
2. The actual hinge line thickness varies along the width of the beam due to manufacturing error. From studies performed on actual parts it was established that the hinge line thickness should be measured at a minimum of five places (the ends, the middle and halfway between the ends and the middle). Therefore, in the tabulated calculation, we divided the dimensions for the notch into two groups
(Refer to Figure 4 for nomenclature)
Group A, measure at one place along the flex line: Two notch radii (R1, R2), Two "X" positions for the arc centers (CX1,CX2), One "Y" position of the upper arc center (CY1)
Group B, measure at five places along the flex line: Distance between the lower arc center and the upper arc center (CY1+CY2)
3. Surface roughness near the hinge line is critical to part life, since fatigue cracks will initiate at any scars left by machining and grinding operations. Therefore it has high information content.

Feature and dimension count for the original double notch flexure

Feature name	Qty	Dimensions for feature	D = # of dim's	N = # of meas't loc's	Non-critical or Critical K= 1 or 10	D x N x K	Total Points
Overall shape	1	L, W, T	3	1	1	3	3
Mounting hole	8	D, X,Y location	3	1	1	3	24
Flex line (group A)	2	R1, R2, CX1, CX2, CY1	5	1	10	50	100
Flex line (group B)	2	CY1+CY2	1	5	10	50	100
Total	**13**						**227**

Maximum stress is determined by six machining dimensions

FINAL CHOICE BETWEEN THE REDESIGNED FLEXURE AND THE ORIGINAL

Figure 3. Photograph of the monolithic beam flexure (left) and the dual notch flexure (right)

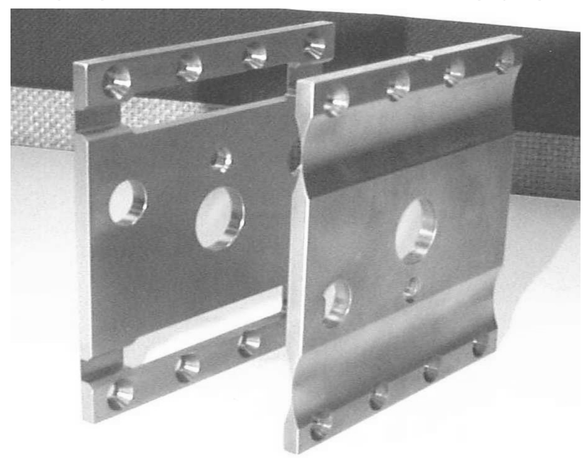
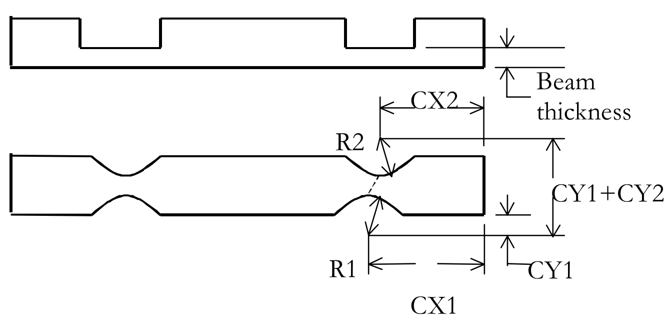


Figure 4. Critical dimensions and tolerances

Beam flexure's critical thickness is determined by just one dimension



A total of six dimensions are required to produce the notch thickness with the dual notch flexure

1. The information content of the original dual notch flexure is about 30% higher than that of the redesigned version (227 vs 171 points).
2. A deciding factor in favor of the beam flexure is the fact that only one dimension determines beam thickness, and therefore maximum bending stress, whereas it takes a

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combination of six dimensions to do the same in the dual notch design.

3. The beam flexure design allows tighter tolerancing of the hinge line thickness. In manufacturing a hinge line thickness tolerance of ±0.0005" was achieved, in comparison to ±0.001" best achievable for the dual notch design
4. As a result of all of the above, the part cost was reduced by 60%

5. CONCLUSION

Axiomatic analysis and design resulted in a robust, inexpensive solution to the flexure fatigue problem and guaranteed no negative impact to the function of the pre-aligner. In fact, thorough functional testing demonstrated that overall performance was indistinguishable between units equipped with the original and the re-designed flexures. The rigorous methodology of AD also helped us surface and quantify some of the finer aspects of the problem that may have been missed using a less structured approach. Most importantly, it provided a formal and conclusive venue to justifying every decision made along the redesign process.

6. REFERENCES

[1] Suh N.P., *The Principles of Design*, New York: Oxford University Press, 1990. ISBN 0-19-504345-6

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