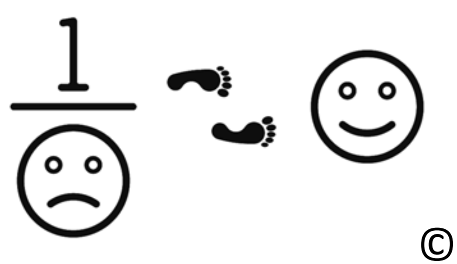
Precision Product Design

Case Study 3

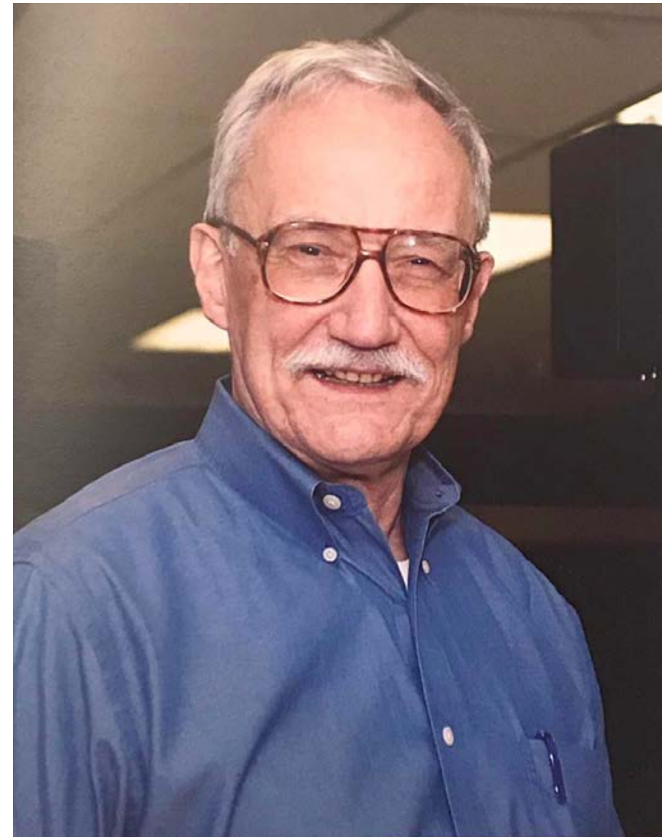
OMAX Waterjet

Alexander H. Slocum
Walter M. May and A. Hazel May Professor of Mechanical Engineering
Massachusetts Institute of Technology
slocum@mit.edu
http://pergatory.mit.edu
@ahslocum



In the beginning…

- 1993: A long long time ago…
- At the faculty club far far away…
- At a rubber chicken alumni lunch…
- “Hi, my name is Dr. John Olsen and have you ever heard of abrasive waterjet machining? What do you do?”
- “Hi, my name is Alex Slocum and I design precision machine tools”
- …



JOHN HENRY OLSEN
August 18, 1939 to February 14, 2019

To infinity and beyond…

- “I started Flow Corporation which makes big machines, and I think there can be a market for thousands of small Bridgeport mill size waterjets…”
- “Let me show you my design…”
- ”That’s cute”
- “WHAT DO YOU MEAN CUTE? What would you do?”
- ”To make a lot, need to minimize materials but lets let the math do the talking…”

US005472367A

United States Patent [19]

Slocum et al.

[11] **Patent Number: 5,472,367**

[45] **Date of Patent: Dec. 5, 1995**

[54] **MACHINE TOOL APPARATUS AND LINEAR MOTION TRACK THEREFOR**

[75] Inventors: **Alexander H. Slocum**, Concord, N.H.; **John H. Olsen**, Vashon, Wash.

[73] Assignee: **OMAX Corporation**, Auburn, Wash.

[21] Appl. No.: **134,524**

[22] Filed: **Oct. 7, 1993**

[51] **Int. Cl.**[6] **B24B 47/22**

[52] **U.S. Cl.** **451/5**; 451/14; 451/75; 451/150; 451/152

[58] **Field of Search** .... 451/5, 11, 14, 451/75, 119, 150, 152, 360; 409/80, 210, 211, 212; 82/172; 83/113; 74/89.15, 428.81 B; 29/563, 27 R, 34 B

[56] **References Cited**

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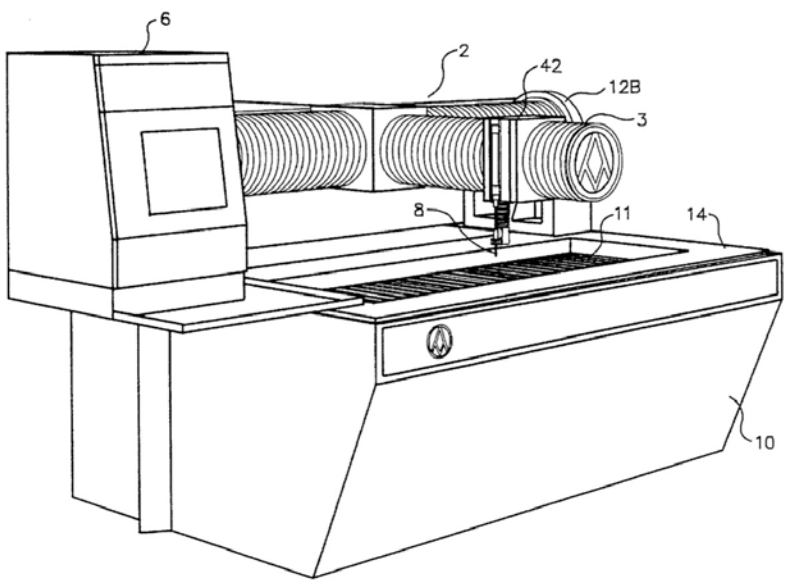
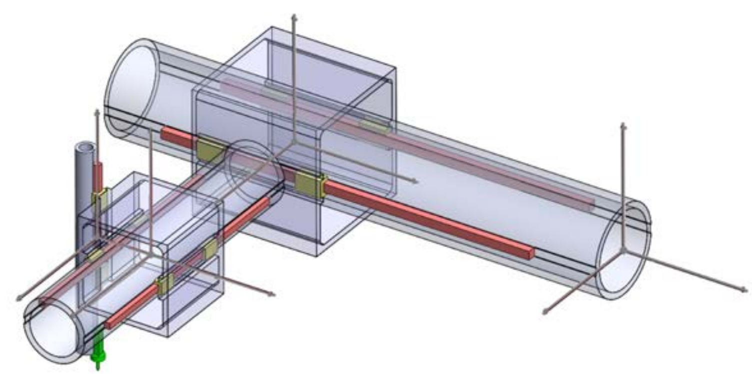
Primary Examiner—Maurina T. Rachuba
Attorney, Agent, or Firm—Graybeal Jackson Haley & Johnson

[57] **ABSTRACT**

A machine tool apparatus, and a linear motion track therefore, are disclosed where the linear motion track is comprised of a tube having a circular cross-section with at least two bearings longitudinally disposed on opposite exterior sides of the tube such that the tube forms a pair of arches joined by the two bearings. The tube arches minimize localized bending moments when radial or moment loads are applied to the tube arches through the bearings and from the longitudinally slidable carriage on the tube. In addition to the linear motion track, the machine tool apparatus also includes a table, for supporting a workpiece, to which the linear motion track is attached, and a machine tool attached to the linear motion track.

26 Claims, 5 Drawing Sheets





is essentially the same as that of the X axis **2**, owing to the modularity of the design concept of this invention. As shown in FIG. **7**, a structural tube **41** has a plate **63** welded to it which is bolted to the X axis carriage **22**. Bellows **46**B and **46**A are each located on a side of tube **21** and are each connected to an end of the tube **41** at plates **60** and **63** and to the carriage **42** with a thin short bellows section such as **46**C shown in FIG. **9**. These round bellows **46**A and **46**B effectively seal the Y axis **3**, and abrasive dirt tends to fall off the Y axis **3** from between the folds as the bellows **46**A and **46**B expand and contract.

As shown in FIG. **7**, the Y axis carriage **42** has T-slots **42**A cut in its outer surface to facilitate mounting of tools and an optional Z motion axis. The Y motion carriage **42** is longitudinally slidable on the tube **41** by linear motion bearing blocks **45**C and **45**D, which are slidably attached to linear bearing rails **45**A and **45**B that are anchored in grooves that are cut in the tube **41**. As shown in FIG. **7**, the carriage Y **42** is moved by a force from the ballscrew nut **54**A, which is attached to the Y axis carriage **42**. Force on the ballscrew nut **54**A is generated when the ballscrew **54**, which is supported in modular bearing units **59**A and **59**B, is turned by a timing belt **57** that engages teeth on pulleys **58** and **59**. Pulleys **58** and **59** are attached to the ballscrew **54** and motor **50**, respectively.

As with the X axis motor, as shown in FIG. **8**, the Y axis motor **50** is attached to a plate **64** that is inside and perpendicular to the tube **41**. This plate's radial position is adjusted by screws inside the tube which adjusts the timing belt **57** tension.

The use of a round tube maximizes the torsional rigidity of the beam, and also increases the efficiency of load transfer between the linear bearings and the tube. Specifically, the round tube acts as an arch locally where the bearing rails are attached to the tube. Because arches are structurally efficient, this minimizes local bending effects of the tube's walls when radial or moment loads are applied to the carriage which are transferred to the tube via the bearings. In addition, the bearings are diametrically opposed to each other on the tube. When a moment load is applied to the carriage, the reaction forces of the bearing blocks are opposed to each other and are tangent to the tube walls. Thus there is no force in a radial direction along the tube wall, and the reaction forces are most efficiently transferred to the structure of the tube. The round tube resists these loads with a circular distribution of shear stresses in the tube's walls, which, because there are no corners, produce no stress concentrations. This minimizes deflection and maximizes efficiency of load transfer between the carriage to the bearings to the tube to the frame. From Rourke[1]:

[1] R. Rourke and W. Young *Formulas for Stress and Strain*, Fifth Edition, McGraw-Hill Book Company, NY, page 292–293.

$$\theta_{twist}=T_{(torque)}L_{(length)}/(K_{(torsional\ stiffness)}G_{(shear\ modulus)});$$

$$K_{(torsional\ stiffness\ hollow\ round\ shaft)}=\pi(D_o^4-D_i^4)/32;\text{ and}$$

$$K_{(torsional\ stiffness\ hollow\ square\ shaft)}=t(h-t)^3.$$

where D_o and D_i are the inner and outer diameters respectively of the round shaft, and h and t are the outside dimension and wall thickness, respectively, of a square tube. Table 1 shows a numerical comparison which is also documented in Tables 3 and 4, below:

TABLE 1

Square verses Round Tube

	round	square
Tube OD	254	254
tube wall thickness	25.4	25.4
Bending Inertia	7.03E–05	0.00012
Torsion factor K	0.000141	0.0003
Torsion ratio square/round defl.	—	2.16
Deflection ratio round/square defl.	—	1.70

The round shaft bends more than a square shaft (by a factor of 1.7), but twists even less (by a factor of 2.16). Thus, since angular deflections are worse because they are amplified by the distance to the tool, a round shaft is a superior choice for this type of design.

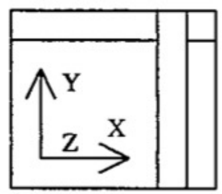
The design of a modular system incorporating the objects of this invention is aided by the use of a spreadsheet to calculate deflections of components and the resulting error motions at the tool tip of the system. The technique of error budgeting is discussed in detail in the textbook *Precision Machine Design* by Alexander H. Slocum (®1992 by Prentice-Hall, Inc. ISBN 0-13-719972-4). As applied to the preferred embodiment of the subject invention, spreadsheet-based calculations of errors in the system is shown in Table 2 as well as in Tables 3 and 4:

TABLE 2

Property	X axis	Y axis
Linear guide size	20	15
Linear guide block length	70	55
width of axis (Nx lg block)	4	3
Width of carriage	280	165
Axis stroke	1219	610
Bellows Lmax/Lmin	8	8
Collapsed bellows length	305	152
End flanges for bellows	20	20
Axis length	1824	947
Load applied at tip along a direction (N)		250 250
Tube OD (mm)	254	203.2
Tube ID (mm)	203.2	177.8
Wall	25.4	12.7
Tube I	1.21E–04	3.46E–05
Density	2.70E+03	2.70E+03
Mass	89.84	19.43
Cost/kilo	$5.50	$5.50
Cost	$494.10	$106.89
Modulus	6.9E+10	6.9E+10
Beam length (mm)	1824	947
X Deflection (μm) at jet	5	30
Y Deflection (μm) at jet	8	1
Z Deflection (μm) at jet	8	30
Z Deflection (μm) own weight	8	8
Torsional moment (N-m)	237	
twist angle (rad)	1.69E–05	
Z torsion @ x error (μm) at jet	16	
Estimated tool tip error from load applied along X, Y, or Z axis		
Total est. error equals 2*Sum defl. mm		
Total estimated machine error X (mm, in)		0.069
Total estimated machine error Y (mm, in)		0.018
Total estimated machine error Z (mm, in)		0.140

TABLE 3

Enter numbers in bold Property	X axis	Y axis
Linear guide size (mm)	20	15
Linear guide block length (mm)	70	55
Width of axis (Nx lg block) (mm)	4	3
Width of carriage (mm)	280	165
Axis stroke (mm)	1219	610
Bellows Lmax/Lmin	8	8
Collapsed bellows length (mm)	305	152
End flanges for bellows (mm)	20	20
Axis length (mm)	1824	947
Load applied at tip along a direction (N)	250	250
Tube OD (mm)	254	203.2
Tube ID (mm)	203.2	177.8
Wall (mm)	25.4	12.7
Tube Inertia (m^4)	1.21E–04	3.46E–05
Density (kg/m^3)	2.70E+03	2.70E+03
Mass (kg)	89.84	19.43
Cost/kilo	$5.50	$5.50
Cost	$494.10	$106.89
Modulus (Pa)	6.9E+10	6.9E+10
	1824	947
Beam Length (mm)	5	30
X Deflecton (μm) at jet	8	1
Y Deflection (μm) at jet	8	30
Z Deflection (μm) at jet		



Z deflection (μM) own weight	8	8
Torsional moment (N-m)	237	
twist angle (rad)	1.69E–05	
Z torsion @ x error (μm) at jet	16	
Estimated took tip error from load applied X, Y, or Z axis		
Total est. error equals 2*Sum defl.		
Total estimated machine error X (mm)	0.069	
Total estimated machine error Y (mm)	0.018	
Total estimated machine error Z (mm)	0.140	
Square verses round tube	round	square
Tube OD	254	254
tube wall thickness	25.4	25.4
Bending inertia	7.026E–05	0.000119
Torsion factor K	0.0001405	0.000303
Torsion ratio square/round defl.	2.16	
Deflection ratio square/round defl.	1.70	

TABLE 4

Enter numbers in bold Property	X axis	Y axis
Linear guide size (mm)	20	15
Linear guide block length (mm)	70	55
Width of axis (Nx lg block) (mm)	4	3
Width of carriage (mm)	=B5*B4	=C5*C4
Axis stroke (mm)	=48*25.4	=24*25.4
Bellows Lmax/Lmin	8	8
Collapsed bellows length (mm)	=2*B7/B8	=2*C7/C8
End flanges for bellows (mm)	20	20
Axis length (mm)	=B10+B9+B7+B6	=C10+C9+C7+C6
Load applied at tip along a direction (N)	250	=B12
Tube OD (mm)	=10*25.4	=8*25.4
Tube ID (mm)	=B15–2*25.4*1	=C15–2*25.4*0.5
Wall (mm)	=(B15–B16)/2	=(C15–C16)/2
Tube Inertia I (m^4)	=PI()*((B15/1000)^4–(B16/1000)^4)/64	=PI()*((C15/1000)^4–(C16/1000)^4)/64
Density (kg/m^3)	2700	=B19
Mass (kg)	=B19*PI()*((B15/2000)^2–(B16/2000)^2)*(B11/1000)	=C19*PI()*((C15/2000)^2–(C16/2000)^2)*(C11/1000)
Cost/kilo	=2.5*2.2	=B21

TABLE 4-continued

Enter numbers in bold Property	X axis	Y axis
Cost	=B21*B20	=C21*C20
Modulus (Pa)	=6900*10^7	=6900*10^7
Beam length (mm)	=B11	=C11
X Deflection (μm) at jet	=1000000*((B12*B24/2000)/(B23*((B15/2000)^2–(B16/2000)^2))+(C24/1000)*B30*(B24/1000)/(12*B23*B18))	=1000000*C12*(C24/1000)^3/(3*C18*C23)
Y Deflection (μm) at jet	=1000000*B12*(B24/1000)^3/(24*B18*B23)	=1000000*(C12*C24/1000)/(C23*((C15/2000)^2–(C16/2000)^2)
Z Deflection (μm) at jet	=1000000*B12*(B24/1000)^3/(24*B18*B23)	=1000000*(C12*C24/1000)^3/(3*C18*C23)
Z deflection (μM) own weight	=1000000*5*(B20*9.8)*(B24/1000)^3/(384*B23*B18)	=1000000*(C20*9.8)*(C24/1000)^3/(8*C23*C18)
Torsional moment (N-m)	=C24*C12/1000	
twist angle (rad)	=(B30*B24/4000)/((B23/2.6)*PI()*((B15/1000)^4–(B16/1000)^4)/32)	
Z torsion @ x error (μm) at jet	=1000000*B31*C24/1000	
Estimated tool tip error from load	applied along X, Y, or Z axis	
Total est. error equals 2*Sum defl.		
Total estimated machine error X (mm)	=2*(B25+C25)/1000	
Total estimated machine error Y (mm)	=2*(B27+C27)/1000	
Total estimated machine error Z (mm)	=2*(B28+C28+B32+B29+C29)/1000	
Square verses round tube	round	square
Tube OD	=B15	=B41
tube wall thickness	=(B15–B16)/2	=B42
Bending inertia	=PI()*((B41/1000)^4–((B41–B42)/1000)^4)/64	=((C41/1000)^4–((C41–C42)/1000)^4)/12
Torsion factor K	=PI()*((B41/1000)^4–((B41–B42)/1000)^4)/32	=C42*(C41–C42)^3/1000)^4
Torsion ratio square/round defl.	=C44/B44	
Deflection ratio round/square defl	=(C43/B43)	

Throughout the class, I will present many of the “secret” details key to creating a robust precision product that can evolve from a core idea into a family of products and a very successful company

Epilog

Three decades have passed
Much knowledge amassed
Many machines been sold
By thinking big & **BOLD!**