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(RESEARCH ARTICLE)

Study of inspection technique for precision component on indexing table

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Abstract

This work presents inspection method for component of indexing table. The indexing table employs a vee and 3 balls kinematic location system. The 6 points contact kinematic design concept of ball and vee groove was used, because the ball and vee groove system can attain good repeatability and has good rigidity. The vee plays an important role as the indexing angle and locking mechanism. Also eccentricity is an important factor as alignment between the rotation center of the rotary table. So, the concentricity between the center of the vee plate and the center of the rotary table is required to be as close as possible to make an accurate vee plate. The indexing table uses the kinematic design concept of vees and balls. But contact stress in the vee is often very high, so the plastic deformation could occur on the vee due to the loaded balls. The elastic deformation of cam shaft is described in this paper.

Keywords: Eccentricity; Elastic Deformation; Indexing Table; Shaft Deflection; Squareness; Parallelism.

1. Introduction

The indexing tables have recently become an increasing necessity for improving the precision and accuracy of angle measurement, and circle dividing in various fields of science and technology. Most indexing tables use a worm wheel mechanism or a grating disc. The manufacturing process together with the requirements for precision is very difficult. Consequently, these indexing tables are expensive and usually used for manufacturing purpose of rotating [1].

The indexing table using a ball and pin was developed [2]. This performed very well in a laboratory environment, but it was not considered robust enough to reproduce as a commercial design. The problem with this indexing table is the balls which are located on the circumference of the body could move when the top plate was located, or any force was applied to the top plate. Consequently, a new mechanism should be considered for the indexing table. Thus 6 points contact kinematic design concept of ball and vee groove was used, because the ball and vee groove system can attain 0.1 μm repeatability [3] and has good rigidity [4].This paper introduce inspection method for component of indexing table.

2. Mechanism of indexing table

The indexing table employs a vee and 3 balls kinematic location system. Each ball is contact with both side of vee surface so each ball has 2 contact points on the vee. Therefore, three balls have 6 kinematics contact points. The incremental indexing angle is 5 degree because 5 degrees incremental angle is considered to be the smallest increment used for the practical calibration of 360° indexing tables. Generally, commercially available indexing tables contain two step operations for indexing (i.e. lift up and rotate), but this indexing table employs one step operation, lifting up and rotating simultaneously by using a camshaft and motor. Fig. 1 shows the mechanism of the indexing table. To hold the top disc on the body and lock the top disc on the vee, the indexing table employs a compression spring between the body and pin disc which is connected to the shaft by screws.

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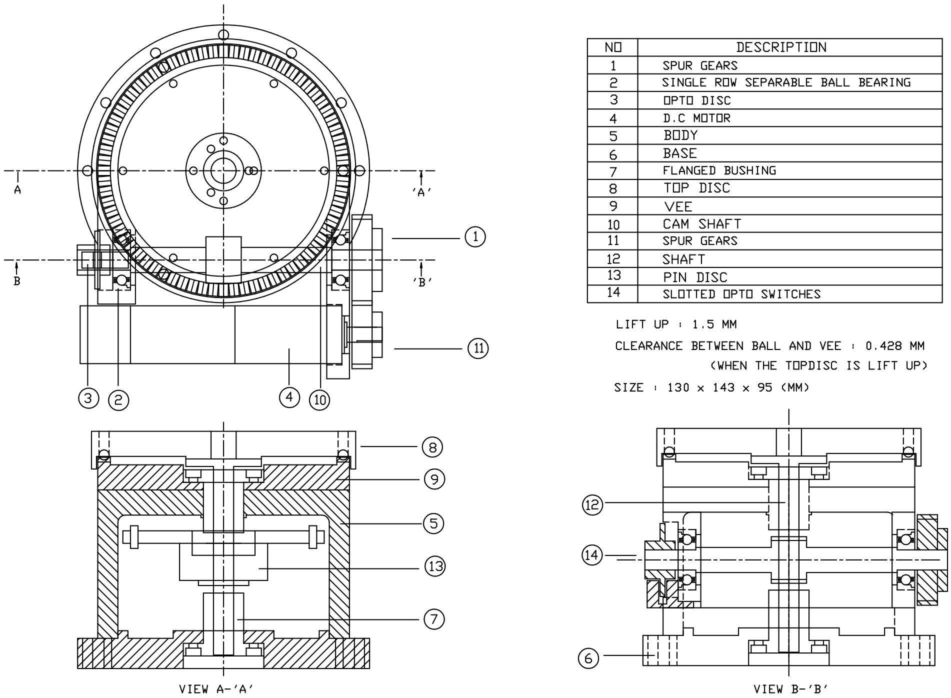
A needle bearing is fitted between the spring and pin disc to prevent torsion and minimize friction. The vee and ball type of indexing table is designed to minimize the working space and gives good repeatability because the vee and ball contact with 6 points kinematic concept. (i.e. each ball contact with vee surface with 2 contact points). Spur gears are employed to translate motor motion into the camshaft and to increase rotation force (i.e. torque).



NO	DESCRIPTION
1	SPUR GEARS
2	SINGLE ROW SEPARABLE BALL BEARING
3	OPTO DISC
4	D.C MOTOR
5	BODY
6	BASE
7	FLANGED BUSHING
8	TOP DISC
9	VEE
10	CAM SHAFT
11	SPUR GEARS
12	SHAFT
13	PIN DISC
14	SLOTTED OPTO SWITCHES

LIFT UP : 1.5 MM

CLEARANCE BETWEEN BALL AND VEE : 0.428 MM (WHEN THE TOPDISC IS LIFT UP)

SIZE : 130 x 143 x 95 (MM)

Figure 1 Mechanism of indexing table

3. Assembly of indexing table

3.1. Squareness and parallelism of body and base

All parts were manufactured according to the drawings. The body and base are the foundation part of the indexing table so that the top surface of body and bearing housing should be square. The base was assembled on the body and two dowel pins used to locate the assembly. The mandrel was inserted through bearing housing of the body and base to check the squareness of body and base when mounted on the bench center.

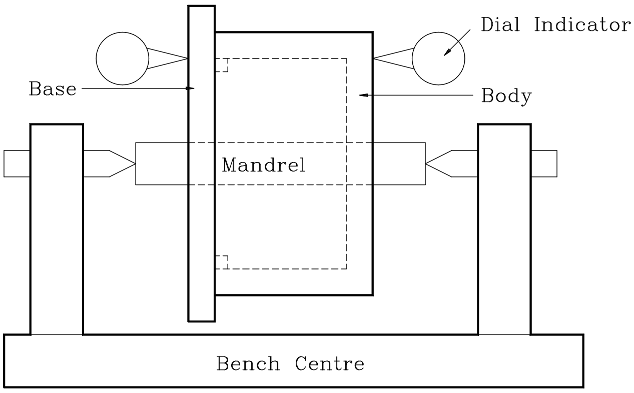


Figure 2 Checking the squareness of body and base

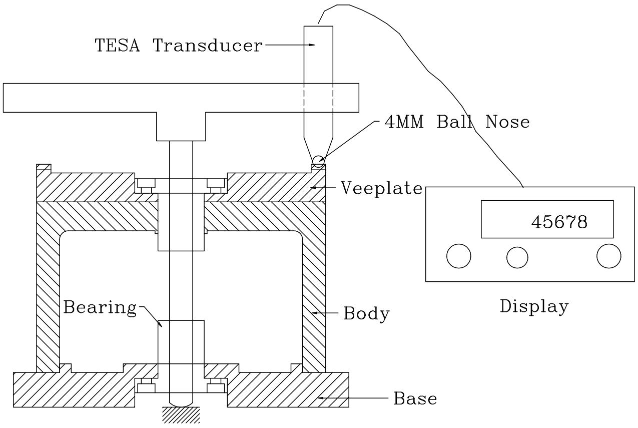


Figure 3 Turn round method for checking squareness

The dial indicator was set against the surface of the assembly as shown in fig. 2. If the reading of the dial indicator changes when the assembly is rotated, it shows the assembly is out of square with the housing of linear bearing. A 0.125mm out of squareness was measured on the top surface of the body, and also 0.102mm out of squareness measured on the bottom surface of the base with respect to be housing of the linear bearing, which indicated that both sides of the assembly were not square to the bearing housing.

In order to rectify this squareness error, the assembly was mounted on a sine tilt-table on a surface grinder and the faces re-ground. After rectification the squareness error was reduced to 10 μm.

The vee plate was assembled on the body together with the linear bearing. The bearing housing and the vee plate should be square to prevent the main shaft touching the bearing, when the top plate is located in the vees. Otherwise this would result in poor repeatability. The device shown in fig. 3 checks the squareness between the bearing and the vee plate.

The transducer with a 4mm ball nose fixed in the device, is located on the vee plate, and the display of the transducer reset to zero. The change of each vee depth is then read around 360 degrees. The difference between the maximum and minimum readings of depth of the vee is the value of out of squareness between the bearing and the main assembly.

To minimize this value, the linear bearing on both sides and the vee plate were rotated one by one using trial and error until the out of squareness was minimized. Because of the variation in the depth of the vees, we could not expect to achieve a squareness error of less than 12μm. This value was confirmed by the measuring method shown in fig. 3.

3.2. Eccentricity between the top plate assembly and body assembly

The hardened main shaft was inserted into the bore of the linear bearing, the top plate is put on the vee plate, and then the hardened main shaft and top plate were then fastened with three bolts with equal force. Although the top plate and hardened main shaft were manufactured square according to the drawing, the top plate assembly which includes the top plate and main shaft was not necessarily square. This feature was however assessed by placing the assembly between center and checking, see fig. 4. This result indicated that these were within 10μm.

If two assembly parts, the top plate assembly and the body assembly, are exactly square and the center between the top plate assembly and the body assembly is concentric, the main shaft won't touch the bore of the linear bearing as the top plate is located on the vee plate. However, if the body assembly and top disc assembly are not square, the hardened main shaft will be touched by the bore of the linear bearing due to the out of squareness and eccentricity between the center of vee plate and the center of linear bearing. This cause poor repeatability and if this occurs then the indexing table would not achieve the 6 points kinematic design concept. If the main shaft is touched in the bearing. The hardened main shaft diameter must therefore be reduced so as not to touch the bore of the linear bearing.

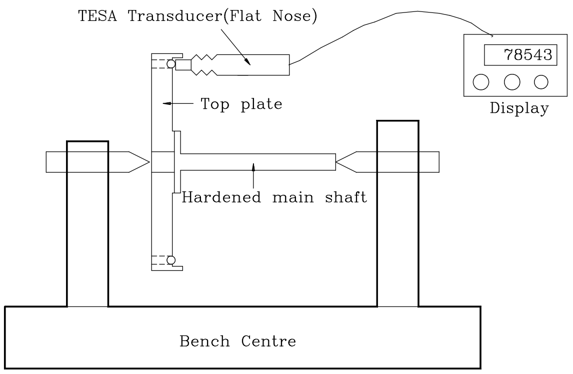


Figure 4 Checking the squareness of the top plate assembly

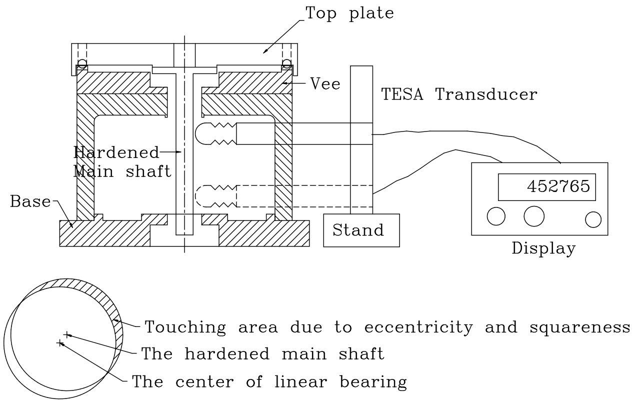


Figure 5 The method to check the eccentricity

3.3. Minimum reduction of the hardened main shaft diameter

The top plate was located on the vee plate after removing the linear bearings on the body assembly. The transducer is set against the hardened main shaft to check the center eccentricity between the vee plate and top plate. The top plate was rotated 5 degrees, and then the reading change measured at 5 degrees incremental angle around 360 degree. The physical set up for the minimizing for the reduction of the hardened shaft is sketched as shown in fig. 5.

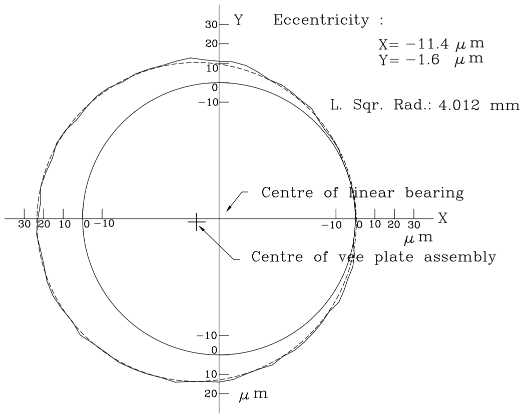


Figure 6 The results of eccentricity near the top bearing

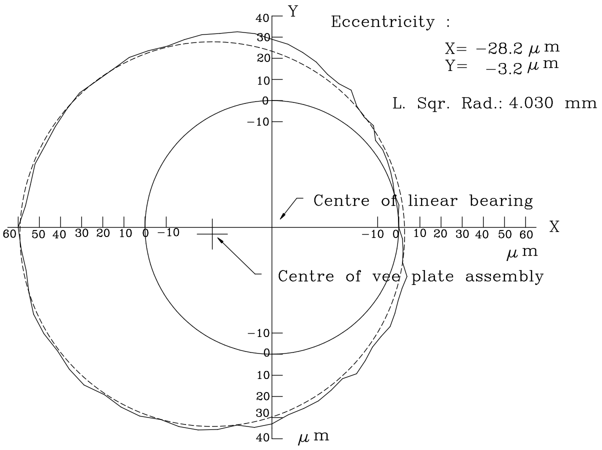


Figure 7 The results of eccentricity near the bottom bearing

The eccentricity offset was found to be 10.5μm near the top bearing and 25.5μm near the base linear bearing. This is shown in fig. 6 and 7. The hardened main shaft was reduced in diameter by an amount equivalent to the eccentricity. This prevented interference between the shaft and the bearing. With clearance established between the shaft and the bearing, the top plate assembly should only make contact with the vees in the vee plate.

To check the repeatability, the transducer was put against the vee plate, and reset to zero. The transducer reading was measured when force by the finger was applied at the center of the top plate, and then removed. The transducer reading always returned back to the same reading(zero). Which means that the hardened main shaft does not touch at the bore of the linear bearing. The bottom linear bearing was then fastened on the base, and same method was applied to check the repeatability, which produced similar results. The indexing table should therefore give good measurement repeatability when used with the laser interferometer.

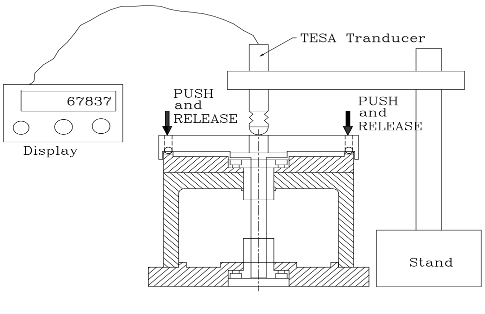


Figure. 8 Method used to check interference

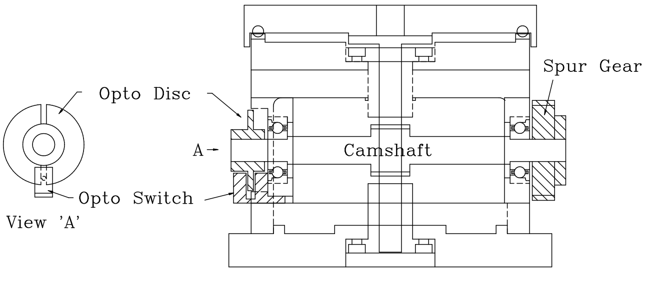


Figure 9 Opto-switch and opto-disc

3.4. Installation of the indexing table and function of parts

The manufactured parts were assembled to the assembly drawing. After removing the assembly problems, i.e. eccentricity and squareness, an opto-switch was fitted on the body and connected to the motor, to allow revolution control.

The opto-disc was fixed on the end of the camshaft, also the spur gears were fixed to the other side of the camshaft, to transmit motor torque to the camshaft. The motor was installed parallel to the camshaft to reduce the working space of the indexing table.

The functions of the parts are described below.

Opto-disc and opto-switch: These are employed for control of the revolution of the camshaft. Which is designed to stop after the camshaft has rotated 180 degrees. The opto-switch is connected on the motor and body, and the opto-disc fixed on the end of the camshaft.

Spur gears: The camshaft and motor are connected parallel in the main body assembly so that the spur gears can transmit motor torque to the camshaft torque.

Camshaft and pin disc: The camshaft translate rotational movement into lift up, rotation and lower down movement of the top plate through the pin disc. The top plate, hardened main shaft and pin disc are assembled to one solid body to obey the 6 points kinematic design concept. This assembly should not interference with the camshaft and bearing when the indexing table is locked on the vee. The camshaft and pin disc are therefore assembled with 0.01inch clearance between them when the table is in the locked position.

Spring and needle roller bearing: These parts are sit between body and the bore of the pin disc. The spring force is required when the table is used in the horizontal and upside down applications. The needle roller bearing which is located between the pin disc and the spring, prevents torsion when the indexing table is operated.

Linear bearing: Two linear bearings are utilized to support the hardened main shaft and prevent the wobble of the top plate during table rotation.

Spur gear: These are connected to the end of the camshaft and to the end of the motor shaft. These translate the rotary motion of the motor to the camshaft. The gear ratio was 1.36 to 1, and increased the available motor torque to the camshaft.

4. Performance evaluation of indexing table

4.1. The deformation of the vees by the balls

The indexing table uses the kinematic design concept of vees and balls. When correctly designed, the kinematic concept is deterministic; i.e. It only makes contact at a number of points equal to the number of degrees of freedom being restrained. Being deterministic gives good repeatability. On the other hand, contact stress in the vee is often very high, so the plastic deformation could occur on the vee due to the loaded balls.

When two surfaces represented by a sphere and a plane surface are loaded, the inertial contact condition will be elastic. When the loading is increased, the stresses between the surfaces will reach the yield stress of the material and at this condition plastic deformation will occur. The elastic deformation is described by the classical Hertz equations which are covered in the book by Tabor [6] and are stated below.

Consider the deformation of an ideal plastic metal vee, of yield stress Y, by a ball of radius r. The friction between the ball and the vee surface is assumed to be negligible small. When the load is applied to the ball, the vee surface and ball will both deform elastically according to the classical radius a as a given by the equation.

$$a = 1.109\left\{\frac{Pr}{2}\left(\frac{1}{E_1}+\frac{1}{E_1}\right)\right\}^{\frac{1}{3}} \qquad (1)$$

Where E1 : Young’s moduli of the ball

E2 : Young's moduli of the plane

P : The load applied

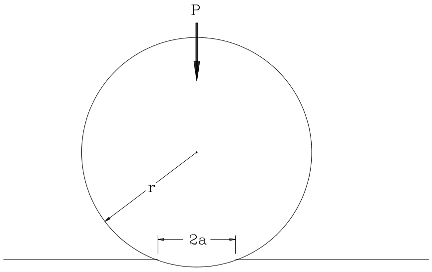


Figure 10 Deformation of a flat surface by hard sphere

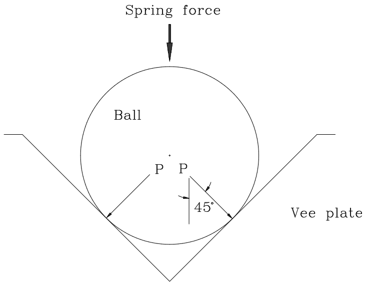


Figure 11 Deformation of a flat surface by hard sphere

The projected area A of the indentation is proportional to $P^{2/3}$.

The maximum pressure is given by the equation.

$$q_0 = \frac{3}{2}\cdot\frac{P}{\pi a^2} = 0.388\left\{\frac{4P}{r^2}\left(\frac{E_1\cdot E_1}{E_1+E_1}\right)\right\}^{\frac{1}{3}} \tag{2}$$

Also mean pressure is given by the equation.

$$q_0 = 1.5P_m \tag{3}$$

i.e. The maximum pressure is 1.5 times the average pressure on the surface of contact.

When the maximum contact shear stress excesses the yield stress for the material, plastic deformation will occur. i.e. plastic deformation will commence when Pm=1.06Y, where Y is yield stress. Provided the mean pressure is less than that causing deformation, and ball and vee, remain completely elastic, i.e. the surface and the ball will return to their original shape.

Loaded force on the vee (P) $p = \frac{Spring\ Force}{3} \times Sin45 = \frac{84.7}{3} \times Sin45 = 19.96N$

Suppose that young modules of vee and balls is $E = 200 \times 10^3\ N/mm^2$, radius of ball is 2mm.

From the eq. (1) $a = 1.109\sqrt[3]{\frac{1}{E}Pr} = 64.8\mu m$

Therefore, maximum pressure is $q_0 = \frac{3}{2}\cdot\frac{P}{\pi a^2} = 0.388\sqrt[3]{P\cdot\frac{E^2}{r^2}} = 2267$

The mean pressure is $P_m = \frac{q_0}{1.5} = 1511.68\ N/mm^2$

Also, plastic deformation begins in the region when P_m=1.1Y, the yield stress of the vee is $2.00 \times 10^3 N/mm^2$. From the results, the mean pressure is less than the yield stress, so that the plastic deformation will not occur.

4.2. Limited load of the indexing table for horizontal application

When the indexing table is used in the horizontal direction, the new indexing table is locked by the spring force during operation. If a heavy load is applied on the table during horizontal application, the spring force can not be sustained against, a heavy load and will therefore adversely affect repeatability.

To calculate the limited load of the new indexing table in the horizontal application.

Spring force (F_s) at each contact is $F_s = \frac{S}{3}$,Where S is the loaded spring force.

The normal force F_n

The vertical direction of normal force (F_{nv}) is $F_{nv} = F_n Cos30 \cdot Cos45$

The horizontal direction of normal force (F_{nh}) is $F_{nh} = F_n Cos60 \cdot Cos45$

The friction force (F_f) and suppose that the coefficient of friction is 0.3. $F_f = \mu \cdot F_n$

The vertical direction of friction force (F_{fv}) $F_{fv} = F_f Cos45 = \mu \cdot F_n \cdot Cos45$

The horizontal direction of friction force (F_{fh}) $F_{fh} = F_f Cos45 = \mu \cdot F_n Cos45$

The equivalent force of in the vertical direction is $\frac{W}{2} + F_{fv} = F_{nv}$

$$\frac{W}{2} + \mu \cdot F_n \cdot Cos45 = F_n Cos30 \cdot Cos45 \qquad \text{---(4)}$$

The equivalent force of in the horizontal direction is $F_s = F_{nh} + F_{fn}$

$$F_s = F_n Cos60 \cdot Cos45 + \mu \cdot F_n \cdot Cos45 \ \rightarrow F_n = 0.589S \qquad \text{---(5)}$$

From the eq. (4), W=0.471S

Thus the new indexing table sustains 47% of spring force in the horizontal application.

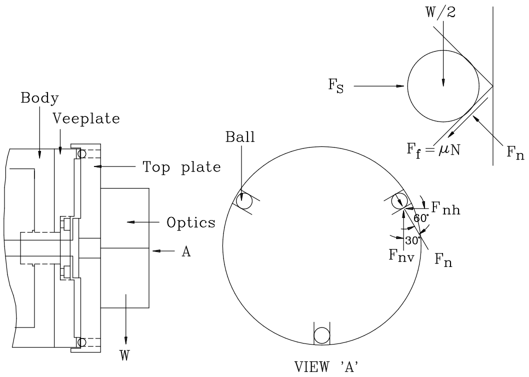


Figure 12 Horizontal application of the indexing table

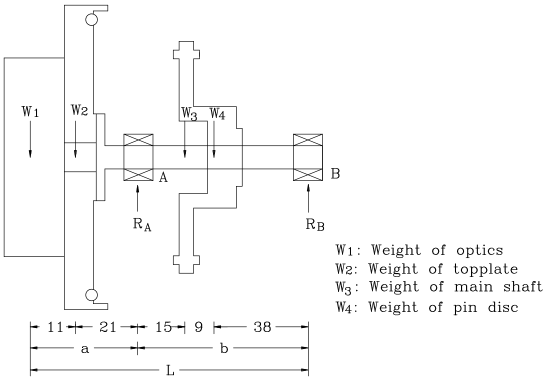


Figure 13 Elastic deflection of hardened main shaft

4.3. Elastic deflection of the hardened main shaft

When the indexing table is used for horizontal applications, the hardened main shaft will be deflected at the end of the main shaft, due to the weight of the optics and top plate and clearance the shaft and bearing. If the deflection is bigger than the clearance between the bottom of the ball and the top of the vee plate when the automatic indexing table lifts, the balls will still touch the vee plate when indexing. In this case damage to the vee plate by the balls will occur.

From fig. 13

The bending moment equation in the left overhanging region is

$$M = -W_1 x - W_2(x - a) \qquad \text{for } 0<x<a \qquad \text{--- (6)}$$

Consequently, the differential equation of the camshaft in that region is

$$\mathrm{EI}\left(\frac{\mathrm{d}^2\mathrm{y}}{\mathrm{dx}^2}\right) = -\mathrm{W}_1\mathrm{x} - \mathrm{W}_2(\mathrm{x} - \mathrm{a}) \qquad \text{for } 0<x<a \qquad \text{--- (7)}$$

Two successive integration yield

$$\mathrm{EI}\frac{\mathrm{dy}}{\mathrm{dx}} = -\mathrm{W}_1\frac{\mathrm{x}^2}{2} - \mathrm{W}_2\left(\frac{\mathrm{x}^2}{2} - 11\mathrm{x}\right) + \mathrm{C}_1 \qquad \text{--- (8)}$$

$$\mathrm{EIy} = -\mathrm{W}_1\frac{\mathrm{x}^3}{6} - \mathrm{W}_2\left(\frac{\mathrm{x}^3}{6} - \frac{11}{2}\mathrm{x}^2\right) + \mathrm{C}_1\mathrm{x} + \mathrm{C}_2 \qquad \text{--- (9)}$$

The bending moment equation in the region between bearing support is

$$\mathrm{M} = -\mathrm{W}_1\mathrm{x} - \mathrm{W}_2(\mathrm{x} - 11) + \mathrm{R}_\mathrm{A}(\mathrm{x} - \mathrm{a}) + \mathrm{W}_3(\mathrm{x} - 47) + \mathrm{W}_4(\mathrm{x} - 56) \qquad \text{--- (10)}$$

The differential equation of the shaft in that region is

$$\mathrm{EI}\left(\frac{\mathrm{d}^2\mathrm{y}}{\mathrm{dx}^2}\right) = -\mathrm{W}_1\mathrm{x} - \mathrm{W}_2(\mathrm{x} - 11) + \mathrm{R}_\mathrm{A}(\mathrm{x} - \mathrm{a}) + \mathrm{W}_3(\mathrm{x} - 47) + \mathrm{W}_4(\mathrm{x} - 56) \qquad \text{--- (11)}$$

Two integration of this equation yield

$$\mathrm{EI}\frac{dy}{dx} = -\mathrm{W}_1\frac{x^2}{2} - \mathrm{W}_2\left(\frac{x^2}{2} - 11\mathrm{x}\right) + \mathrm{R}_\mathrm{A}\left(\frac{x^2}{2} - \mathrm{ax}\right) + \mathrm{W}_3\left(\frac{x^2}{2} - 47\mathrm{x}\right) + \mathrm{W}_4\left(\frac{x^2}{2} - 56\mathrm{x}\right) + C_3 \qquad \text{---(12)}$$

$$\mathrm{EI}y = -\mathrm{W}_1\frac{x^3}{6} - \mathrm{W}_2\left(\frac{x^3}{6} - \frac{\mathrm{x}^2}{2}11\right) + \mathrm{R}_\mathrm{A}\left(\frac{x^3}{6} - \frac{\mathrm{a}}{2}\mathrm{x}^2\right) + \mathrm{W}_3\left(\frac{x^3}{6} - \frac{47}{2}\mathrm{x}^2\right) + \mathrm{W}_4\left(\frac{x^3}{6} - \frac{56}{2}\mathrm{x}^2\right) + C_3x + C_4 \qquad \text{---(13)}$$

Since we started with two second-order differential equation, (7) and (10), and two constants of integration arose from each, we have four constants C_1, C_2, C_3 and C_4 to use to evaluate conditions for slope and deflection.

These conditions are

(a) when x=a, y=0 in the overhanging region

(b) when x=a, y=0 in the region between bearing supports

(c) when x=L, y=0 in the region between bearing supports

(d) when x=a, the slope given by (8) must be equal to that given by (12); consequently the right side of these equations must be equal when x=a

C_1, C_2, C_3 and C_4 are found with using the above conditions. The deflection of the shaft is calculated by

$$\mathrm{EI}y = -\mathrm{W}_1\frac{x^3}{6} - \mathrm{W}_2\left(\frac{x^3}{6} - \frac{11}{2}x^2\right) + 23427.57\mathrm{x} - 671909.44 \quad 0<x<a=32 \qquad (6\text{-}19)$$

Where E=200,000Nmm^{-2}, I=2010.062mm^4, When x=0, y=-0.0167μm i.e. 16.7μm downward

The 16.7μm deflection is calculated at the end of main shaft so the deflection of the vee plate is 25.6μm. The bearing clearance will cause the shaft to deflect during lift. Assuming a designed clearance of 25μm, the equivalent displacement at the ball is 50.6μm. The clearance between the ball and vee plate is 428μm when the top plate is lifted so the ball does not touch the vee plate during the new indexing table operation.

4.4. Repeatability test of the indexing table

As the system is used for calibration, one of the most important things is that a new indexing table should give good repeatability. Also, British Standard 5233: defines repeatability as follows, “The quality which characterized the ability of a measuring instrument to give identical indications, or response, for repeated application of the same value of the measured quantity under same condition of use.” The accurate measuring device and technique is developed for this purpose. The repeatability of the new indexing table was measured as shown below

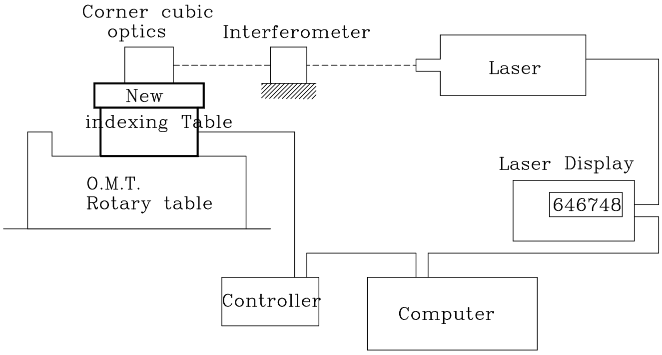


Figure 14 Schematic of the equipment used to check the repeatability of the new indexing table

The laser optics were mounted on the automatic indexing table which was clamped on the O.M.T. table. The laser display and motor control box were connected to a computer. The P.C. is programmed to control the rotation and direction of automatic indexing table, (i.e. clockwise and anticlockwise), and takes the laser display reading. The value is read at each target when the indexing table is rotated clockwise and then back to same position. The reading is taken 3 times in the clockwise direction and 3 times in the anticlockwise direction with a 5 degrees incremental angle during a 360 degrees rotation. The bi-directional standard deviation (σ) at each position of the table was evaluated using the equation $\sigma = \left(\sqrt{\frac{1}{n-1}\sum_{i=1}^{n}(x_i + \bar{x})^2}\right)$ The 2σ(95% confidence) is shown in the figure. The repeatability result of the automatic indexing table in the horizontal and vertical directions is shown in fig. 15 and 16.

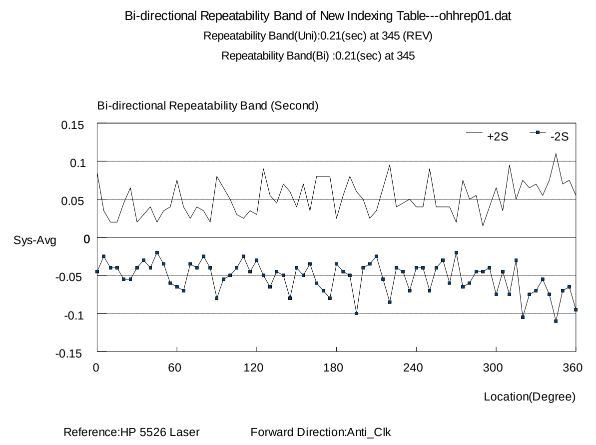


Figure. 15 Repeatability of indexing table (Vertical)

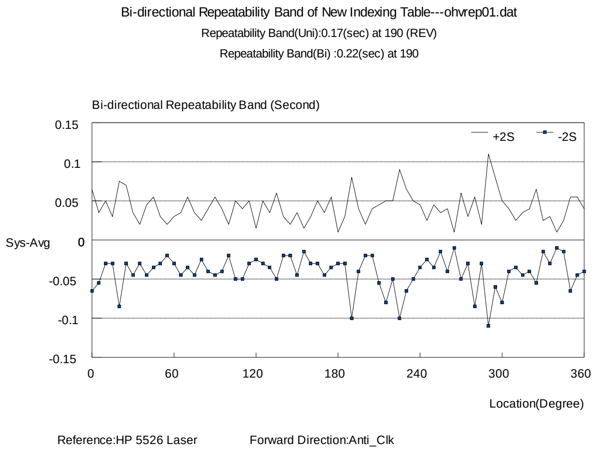


Figure 16 Repeatability of indexing table (Horizontal)

5. Conclusion

Many techniques and methods for calibrating angular measurement exist in engineering industry. The cost-effective systems are generally available using general purpose hardware. The kinematic design of the indexing table using the vee plate and the ball location gives good repeatability. Compact indexing table was achieved by inspection technique.

Also, eccentricity is an important factor as alignment between the rotation center of the rotary table. So, the concentricity between the center of the vee plate and the center of the rotary table is required to be as close as possible to make an accurate vee plate. The indexing table uses the kinematic design concept of vees and balls. But contact stress in the vee is often very high, so the plastic deformation could occur on the vee due to the loaded balls. The elastic deformation of main shaft is described. For component assembly, squareness and parallelism were measured using a simple measuring instrument and the component was assembled on the indexing table.

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