*ISSN 1392 - 1207. MECHANIKA. 2013. Volume 19(6): 676-680*

# Elastic vibrations of the peg during part alignment

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crossref http://dx.doi.org/10.5755/j01.mech.19.6.6014

## 1. Introduction

Since 1961, when the first industrial robot was introduced in GM (General Motors) Ternstedt plant in Trenton, NJ (New Jersey), annual demand on robot systems continuously increases and consisted of 166028 units in 2011. The main consumers of robotic systems remains automotive, electronics, chemical/rubber/plastic industries, where robots used in various parts assembly, welding, painting pick-and-place tasks [1].

To ensure product quality and reduce production costs the use of robotic systems is effective but expensive solution. More over robots/manipulators requires periodical maintenance and replacement of wear and tear parts to ensure high performance accuracy which is very important in automated assembly. However wearing parts not the only source of the assembly errors. Tolerances of assembled parts, positioning errors, vibrations in manufacturing area reduces robots ability for successful task performance and simple peg-to-hole assembly operation might be impossible.

Here are number of inventions that help to increase accuracy and efficiency of robots or manipulators. Device with remote center of compliance (RCC) introduced by D. E. Whitney in 1981 is probably the best known invention helping to accomplish assembly and engaging tasks [2]. Different modifications of RCC devices are still used now days. It is very effective mechanical tool with no requirement of auxiliary energy source or sophisticated software and acting on forces that rises in a contact point between mating parts. However drawback of this device is that it can be used only for the chamfered parts.

Active compliant devices are systems mainly with position, force/momentum sensors feedback or vision based systems. Position corrections of the mating parts mainly done by robot/manipulator servo actuators in correspond to error signal from the feedback sensors [3-5]. It is advanced but expensive systems requiring additional computing power for feedback signal processing. And not always possible to get correct signal from the sensors or because of the wrong view angle machine vision systems fails.

Vibrational part alignment classifies as passive compliant method. Whilst auxiliary energy source is used to generate vibrations were is no feedback from the sensors and no position correction signals formed to the robot/manipulator actuators.

Kaunas University of Technology is one of the leading institutions in vibratory technologies as well as in vibratory alignment research. Their scientists have introduced alignment method when RCC device is used together with axial vibrations of one of the mating parts (peg or bushing) [6, 7]. System excitation and compliant device stiffness parameters were defined for steady and stable performance. This method prevents key insertion problem jamming and wedging.

Mating part alignment on a vibratory plane is described in paper [8]. In this case chamferless parts can be used since every point of interwinding helix path is checked in a search field. But this is more complex system then previous described as bushing is movable based on a plane vibrating in a two perpendicular directions. Also forces of dry friction between plane-bushing and bushing-peg have to be taken to account when setting parameters for stable and reliable performance.

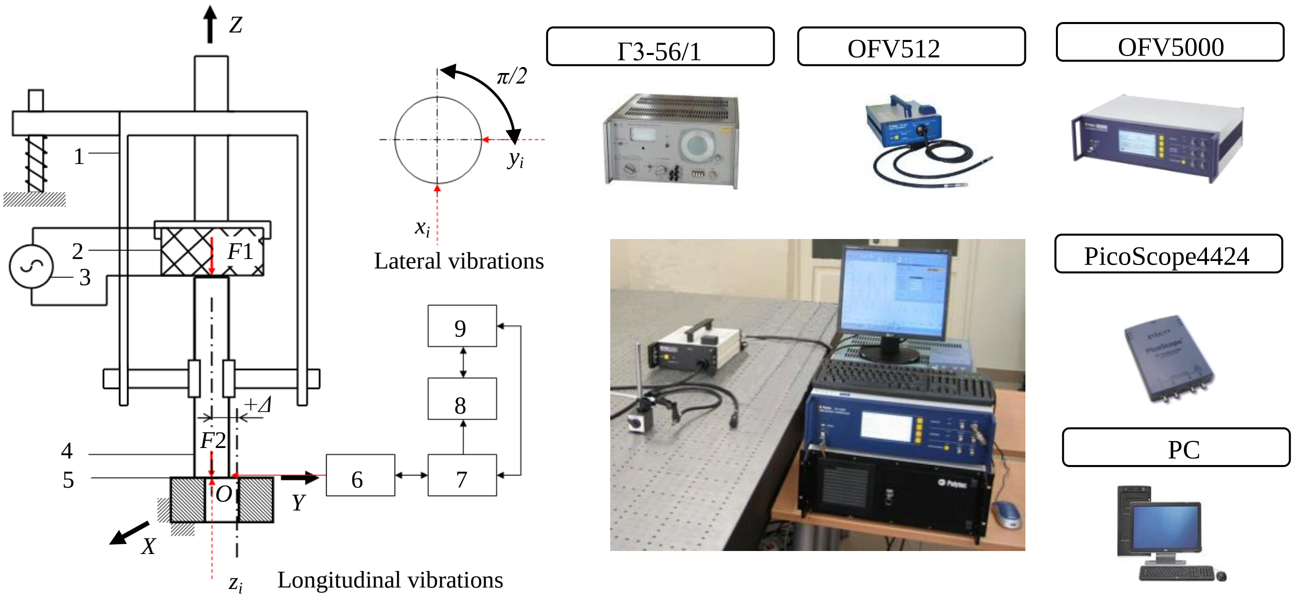
All vibratory methods mentioned above uses low frequency vibrations (tenth to hundreds of Herz). Paper [9] describes alignment method when one of the mating parts (particular peg) is excited with the high frequency vibrations thus creating elastic deformations in it. In a mean time bushing is movable based on a plane. The alignment goes due friction forces when the peg is in a contact with bushing. It was experimentally set what influence of excitation frequency and amplitude, initial peg-bushing pressing force has to the alignment time. It was also showed the ability to use this method for mutual part alignment made from different materials and with different geometrical shape. However there is still lack of research data on a vibration character of the peg while in a contact with bushing.

The aim of this paper is to explain nature of the peg's tip vibrations when it's in contact with bushing. As well as to answer how bushing is pointed to the correct alignment direction.

## 2. Experimental setup for vibratory part investigation

To investigate peg's vibrations while it's in a contact with bushing the following experimental equipment has been used (Fig. 1). Peg 4 is fixed in a middle cross-section in a gripper 1. Piezoelectric vibrator 2 is pressed to the upper end of the peg with pressing force $F1$ and excites peg in axial direction. Excitation signal to the vibrator provided by signal generator 3. The lower end of the peg is pressed to the bushing 5 with initial pressing force $F2$ while axis misalignment $\Delta$.

To register peg's vibrations, one axis laser dopler vibrometer (LDV) is used. The interferometer head OFV512 measures vibrations and controller OFV5000 coverts signal from interferometer to the voltage signal corresponded to vibration amplitude. Further signal captured with oscilloscope PicoScope 4424 and displayed on a computer screen.




a b

Fig. 1 Experimental setup: *a* – measurement scheme: *1* – gripper; *2* – piezoelectric vibrator; *3* – excitation signal generator Г3-56/1; *4* – peg; *5* – bushing; *6* – fiber interferometer OFV512; *7* – vibrometer controller OFV5000; *8* – oscilloscope PicoScope4424; *9* – personal computer PC; *b* – measurement equipment

Vibrometer measurements were taken in a three directions *X*, *Y*, *Z*. Where *X*, *Y* corresponds to lateral vibrations in a two perpendicular directions and *Z* are longitudinal vibrations. Axial misalignment of the parts $-\Delta$ and $+\Delta$ lies on *OY* axis. Thus mutual part alignment occurs when bushing center coincides with coordinate axes center.

Peg's tip vibration magnitude was investigated under different pressing forces *F*1 (vibrator to the peg) and *F*2 (peg to the bushing) when excitation frequency $\nu$ varies from 6523 to 6723 Hz, and tip movement trajectory was defined in relation with misalignment position $\Delta$.

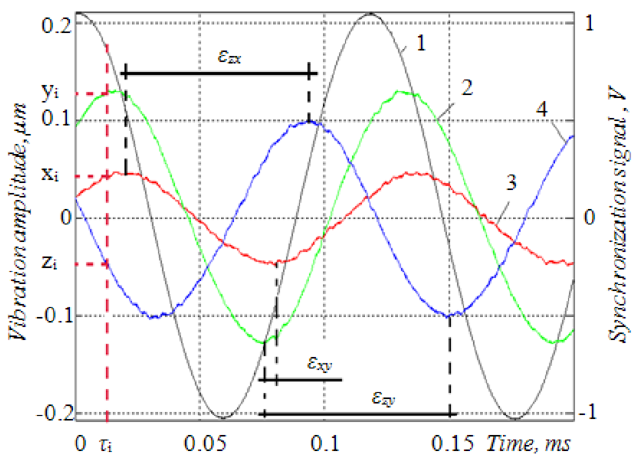



Fig. 2 Vibration signals and phase difference $\varepsilon$: *1* – synchronization signal; *2* – *Y* vibrations; *3* – *X* vibrations; *4* – *Z* vibrations

In order to find movement trajectory of the peg's end tip, measurements of two perpendicular axes (*X-Y*, *Z-Y*, *Z-X*) were taken. Synchronization signal related to the excitation signal was used to synchronize measurement process. As long as vibrations are periodic and steady, vibrations magnitude ($x_i$, $y_i$, $z_i$) of each axis is defined at the same periodic time $\tau_i$ according to the synchronization signal (Fig. 2). Plotting those values in a Cartesian coordinate system peg's path in all three planes found.

Time interval between two vibration signals at the same instantaneous phase gives us phase difference $\varepsilon$ between those signals.

$$\varepsilon = 2\pi f \times \Delta t$$

where $f$ – vibration frequency; $\Delta t$ – time interval between two signals.

Excitation parameters and objects of experiments presented in Table 1.

Table 1

Characteristics of excitation signal and aligned parts

| No. | I | II | III |
|---|---|---|---|
| Peg | Steel S235JR | | |
| Diameter, mm | 10 | 10 | 10 |
| Length, mm | 59.8 | 79.65 | 99.75 |
| Chamfers | No | | |
| Bush | Steel S235JR | | |
| Hole diameter, mm | 10.1 | 10.1 | 10.1 |
| Excitation signal parameters | | | |
| Frequency, Hz | 8475 | 6711 | 6623 |
| Amplitude, V | 132 | | |

## 3. Results of experiments

Influence of forces *F*1 and *F*2 to vibration amplitude was investigated on peg No. III. Pressing force of piezoelectric vibrator to the peg was gradually increased every 14 N and corresponding measurements of vibration magnitudes on all three axes were taken. The results are shown in Fig. 3.

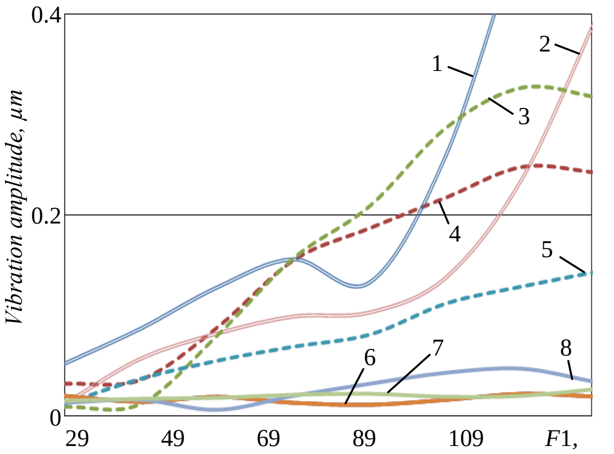



Fig. 3 Vibration amplitude versus force $F1$: longitudinal $Z$ vibrations: *1* – $v$ = 6523 Hz; *2* – $v$ = 6623 Hz; lateral $X$ vibrations: *3* – $v$ = 6523 Hz; *4* – $v$ = 6623 Hz; *5* – $v$ = 6723 Hz; lateral $Y$ vibrations: *6* – $v$ = 6523 Hz; *7* – $v$ = 6623 Hz; *8* – $v$ = 6723 Hz

Force $F1$ and excitation frequency has no impact on vibration magnitude along axis *OY*. Vibration amplitude in *OX* axis direction gradually increases when pressing force reaches 49 N and later stabilizes at 115 N. Meanwhile overall *OX* vibration magnitude decreases as excitation frequency increases. Amplitude of longitudinal vibrations increases more rapidly after $F1$ exceeds 90 N until that growth relatively small. Such character of amplitude increment related with contact area changes between peg and piezoelectric vibrator. More force is applied bigger micro deformations between peg and vibrator thus bigger contact area and more excitation energy transferred to the peg. Since excitation of the peg is done with high frequency and small amplitudes contact area between peg and vibrator plays vital role. This could be seen from graph 1 and 2, as excitation frequency increases pressing force $F2$ also has to be increased to keep same longitudinal vibration amplitude. It was also experimentally set that mutual part alignment starts when force $F2$ exceeds 90 N, until that process of part alignment is not stable or it doesn't work at all.

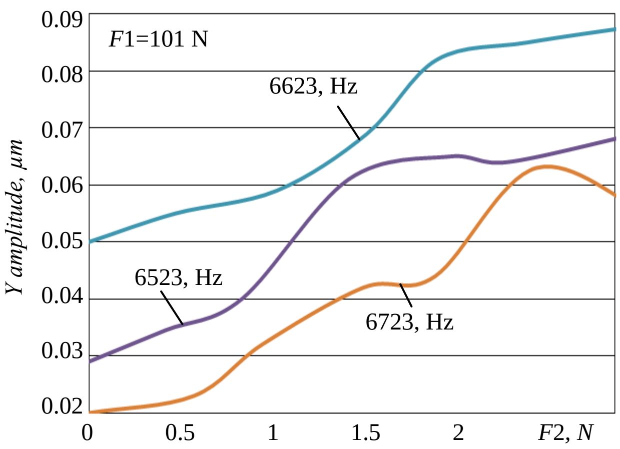
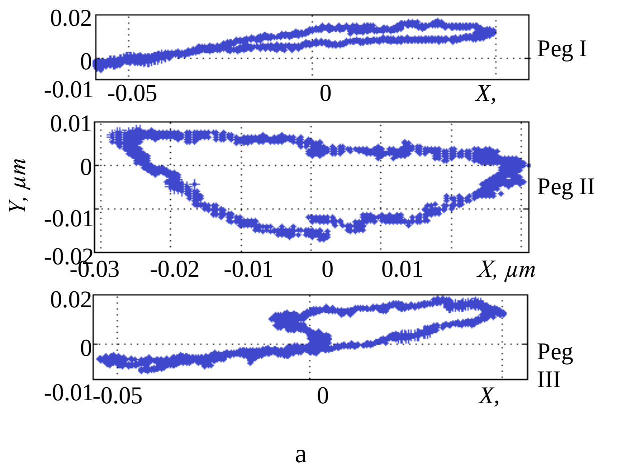



Fig. 4 Vibration amplitude versus force $F2$

If force $F1$ had no impact on vibrations in *OY* axis, totally different impact had force $F2$. As the peg is pressed to the bushing with axis misalignment $\Delta$ = +1.5 mm, vibration amplitude gradually increases as force $F2$ increases. The same tendency is kept even if excitation frequency changes in range from 6523 to 6723 Hz (Fig. 4). In our case part alignment is most rapid when peg is excited at 6623 Hz frequency and vibrations in *OY* axis is the biggest.

Experiment results mentioned above in generally shows what influence for the vibrations amplitude has mounting conditions of the peg, and also that excited peg vibrates in three directions perpendicular each other. But there still no answer why bushing is slides toward coordinate axes centre. To find out what factors in charge of this effect, motion trajectory and direction of peg's tip was determined.

After excitation frequency for stable and steady part alignment was experimentally set to all pegs (Table 1), motion trajectory of the tip was taken in all three coordinate planes. Excitation frequency mainly depends from the peg's natural frequency, design of the gripper and force $F2$. Thus for the grippers with different design or made from different material excitation frequency for steady and stable part alignment will be different. In our case excitation frequency for stable and steady part alignment have lied between second and third natural bending mode of the peg. Fig. 5 shows peg's tip path while forces $F1$ = 101 N, $F2$ = 0 N.



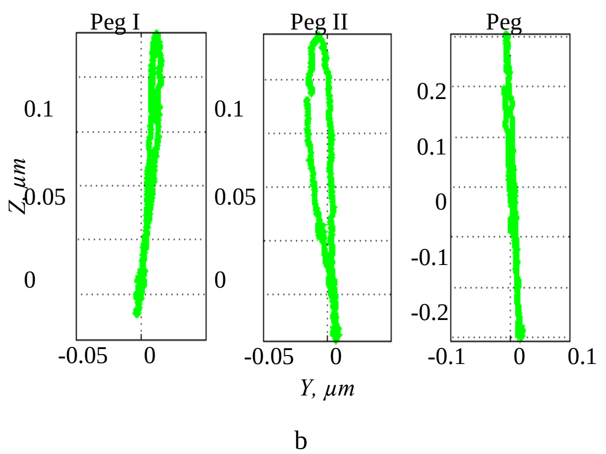



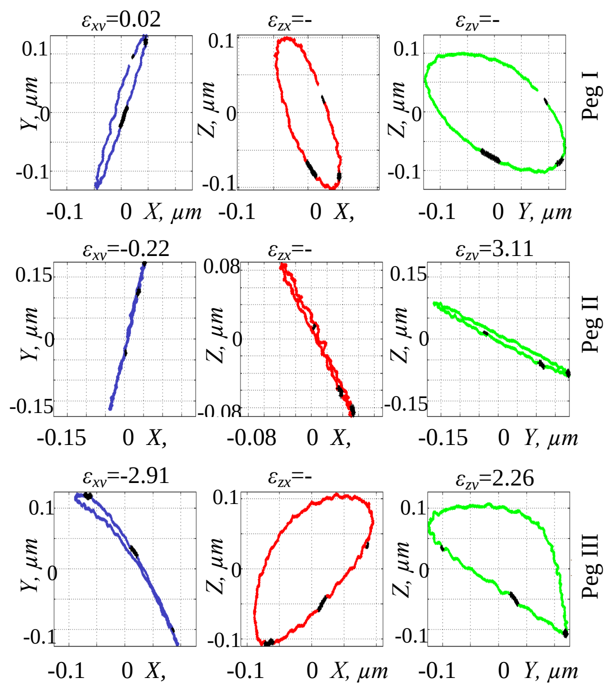

Fig. 5 Path trajectory of unloaded peg: *a* – in *YOX* plane; *b* – in *ZOY* plane

Longitudinal vibrations are dominant in all cases and are twice as high as transverse ones. While in *YOX* plane they are polarized in *OX* direction since peg's vibrations in *OY* direction are negligible.

When the peg is pressed to the bushing with the force $F2 = 2.2$ N and axis misalignment $\Delta = -1.5$ mm, lateral vibrations on *OY* axis increases significant and peg's end moves in elliptical shape trajectory in all three coordinate planes(Fig. 6). Black dots on the path indicate its direction. For the different pegs, direction of rotation is different and that depends from excitation frequency and natural mode gripper–peg system vibrates.

In order the alignment of the parts could occur, bushing has to slide along positive *OY* direction. There are two ways how bushing is aligned. First is so called direct alignment (Peg II and III). In this case peg's tip moves counter-clockwise in *ZOY* plane (Peg II and III, c), thus direction of the normal force in the contact point lies on the positive *OY* direction and bushing is directly pushed toward coordinate axes center. Vibrations along *OX* axis has little effect since their amplitude smaller than *OY*, and overall vibrations are more polarized along *OY* axis (Peg II and III, a). Normal peg to bushing pressing force is bigger when longitudinal vibrations amplitude is negative. Thus propellant force is bigger when peg vibrates along positive *OY* axis rather than negative.

Second way of part alignment is indirect alignment (Peg I). Here peg's motion is clockwise in *ZOY* plane (Peg I, c) and bushing is pushed from the coordinate axes center. But because of the peg's tip elliptical movement in *YOX* plane (Peg I, a), bushing is turned by the angle so the alignment trajectory lie on the major axis of the ellipse and then pushed towards coordinate axes center.



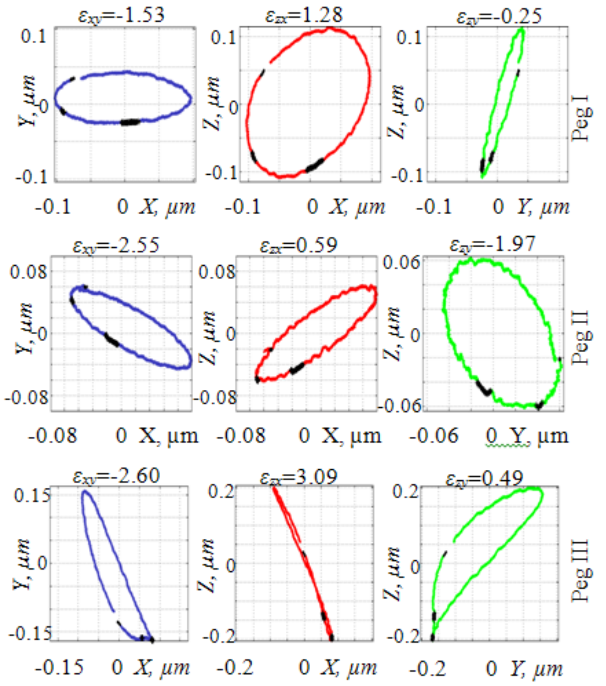

a b c

Fig. 6 Path trajectory of loaded peg when $\Delta = -1.5$ mm: *a* – in *YOX* plane; *b* – in *ZOX* plane; c – in *ZOY* plane

Results of peg's tip trajectory while $\Delta = +1.5$ mm presented in Fig. 7. As contact conditions between peg and bushing has changed (contact area crescent now faced to opposite side), phases between vibrations also changed. In this case for the bushing to align with the peg, bushing has to slide along negative *OY* direction. Peg's I and II tip moves clockwise in a *ZOY* plane (Peg I and II, c) thus direct alignment is going.

Bushing with the Peg III is aligned during indirect alignment. The bush is propelled along negative *OY* direction, but because of rotation effect in *YOX* plane (Peg III, a) bushing is turned and pointed to the coordinate axes center.

It's clear that during direct alignment peg's motion in *ZOY* plane plays key role, meanwhile during indirect alignment there is combination of peg's movement in *ZOY* and *YOX* planes.




a b c

Fig. 7 Path trajectory of loaded peg when $\Delta = +1.5$ mm: *a* – in *YOX* plane; *b* – in *ZOX* plane; *c* – in *ZOY* plane

## 4. Conclusions

1. It is necessary that the entire rear surface of the peg is in contact with piezoelectric vibrator, so the maximum vibration energy passes from vibrator to peg. However there is a limit of maximum vibrator to peg pressing force and further increase of $F1$ has no effect to magnitude of lateral vibrations.

2. Under proper excitation frequency peg has longitudinal and lateral *OX* vibrations while lateral *OY* vibrations are negligible. Excitation frequency depends on peg's natural frequency, mechanical design of the gripper and pressing force $F1$.

3. Lateral *OY* vibrations increases when the peg to bushing pressing force increases.

4. When initial relative position between peg and bush changes, contact area crescent between parts also changes its orientation. That leads to the phase changes between vibrations and direction of propellant force.

5. If propellant force directed to the coordinate axes center, direct part alignment is going. If propellant force directed from the coordinate axes center, indirect part alignment is going.

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VELENO TAMPRIEJI VIRPESIAI CENTRUOJANT DETALES

R e z i u m ė

Straipsnyje eksperimentiškai nagrinėjami velenėlio, kontaktuojančio su įvore automatinio rinkimo metu, tamprieji virpesiai. Pateiktas eksperimentinis stendas ir tyrimo metodika. Nustatyta, kokią įtaką skersinių ir išilginių virpesių amplitudėms turi žadinimo dažnis ir velenėlio tvirtinimo sąlygos. Sudarytos velenėlio galo judesio trajektorijos trijose statmenose plokštumose, kai velenėlis su įvore liečiasi skirtingose vietose. Atlikta eksperimentinių duomenų analizė, nustatyta, dėl kokių priežasčių įvorė slenka koordinačių ašių centro link nepriklausomai nuo veleno ir įvorės pradinės tarpusavio padėties.

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ELASTIC VIBRATIONS OF THE PEG DURING PART ALIGNMENT

S u m m a r y

In this paper experimentaly analyzed elastic vibrations of the peg while in a contact with bushing in automated assembly. Experimental setup and methodology of investigation are explained. Influence of excitation frequency and mounting conditions of the peg to the peg‘ tip lateral and longitudinal vibrations amplitude was determined. Peg‘s tip movement trajectory in three perpendicular planes were concluded while contact position between peg and bush is different. Analysis of experimental data explaines reasons why bushing always slides toward coordinate axes center despite its original position.

**Keywords:** elastic vibrations, peg-in-hole alignment, vibrometer.

Received January 09, 2013
Accepted December 10, 2013