ISSN 1392−1207. MECHANIKA. 2014 Volume 20(2): 165−170

Experimental research of vibratory alignment using passive compliance devices

B. Bakšys*, J. Baskutienė, J. Vėžys*****
**Kaunas University of Technology, Kęstučio 27, LT-44312 Kaunas, Lithuania, E-mail: bronius.baksys@ktu.lt*
***Kaunas University of Technology, Kęstučio 27, LT-44312 Kaunas, Lithuania, E-mail: jbask@ktu.lt*
****Kaunas University of Technology, Kęstučio 27, LT-44312 Kaunas, Lithuania, E-mail: joris.vezys@stud.ktu.lt*

crossref http://dx.doi.org/10.5755/j01.mech.20.2.6945

1. Introduction

To compensate the errors of the mutual location of the components in assembly position, two basically different methods, i.e. active and passive compliance, are used [1, 2]. When active method is used, the assembly devices and systems are equipped with the means of adaptation. The sensors are mounted in the feedback circuits to ensure the control of the mutual position of the components, forces and moments arising during the joining and to form the control signals for the additional displacement of the robot links or components fixing devices. To implement the active method, it is necessary to use the expensive and complex means of adaptation, the devices are slowly reacting, consequently, the efficiency is relatively low. Passive method is based on the compliance of the assembly devices or kinematic elements, which may be mounted at the end of the manipulator or locating device. Due to elastic constraints of the device at least one of the connective components is able to displace within the limited space. During the assembly the interaction forces that arise between the components, result the mutual alignment of the components. The devices based on this method have characteristic of high reaction, they are relatively inexpensive and simple, because there is no need to use sensors, feedback systems, positioning actuators and complex control algorithms. By using the devices of passive compliance, it is possible to ensure only the automated assembly of the components with chamfers, which predetermine the allowable part-to-part misalignment error.

The passive compliance assembly devices of different construction are currently used, which ensure the angular and linear displacements aiming to compensate the misalignment errors between the components [3, 4]. The common characteristic of passive devices is that under the influence of the external forces and moments their elastic structures get deformed and so the necessary displacement and turn of the movable component is obtained. During the automated assembly, external forces and moments arise at the contact place between the part or other component and a chamfer. The range of the elastic deformations, as well as the displacement, is limited and dependent on the construction of the elastic elements and properties of the materials.

The remote center compliance devices are developed, when in this centre the interaction force between the components results a pure displacement, whereas the moment about this centre results the pure turn [5].

The promising is assembly method, when both the passive compliance device and vibratory excitation are used [6, 7]. Using this method, one of the components is movably based in the device, ensuring the displacement within the particular limited space, while the other is based immovably in locating device and components are pressed to each other applying the particular force. One of the components in assembly position is provided with vibratory excitation of predefined frequency and amplitude along the particular direction. Due to influence of vibrations, the movably based component, being in contact with the connective component, is able to move and turn in respect of the other. In such a way the components in assembly position are mutually aligned so, that their connective surfaces are matched and unhindered assembly is possible. The parameters of the excitation and stiffness of the compliance device which depend on the construction of the elastic components of the device, have high influence on the reliability and duration of the vibratory alignment,

The experimental analysis of vibratory alignment of the shaft and bushing, when the shaft is movably fixed using the bellows type elastic element and remote compliance device with springs, is considered in the publication [8]. The linear and also angular displacement of the fixed shaft is ensured by the same elastic elements, therefore, the alignment takes place within the smaller range of the mutual misalignment of the parts. Having aim to increase the efficiency of the alignment, different elastic elements should be used to ensure the linear and angular displacement of the shaft.

The aim of the work is to develop the passive compliance devices with elastic elements ensuring the linear and angular displacement and to carry out the experimental analysis of the shaft–busing type parts alignment using the mentioned devices, determine both the mechanical system and excitation parameters influence on the duration and reliability of the alignment.

2. The devices of passive compliance

For the experimental analysis of the shaft and bushing alignment, devices of passive compliance have been made using different elastic elements. The elastic elements *4* of the first device are of cylinder shape (Fig. 1), made of porous rubber, are located by 120 degrees angle each from the other and designed to ensure the linear compliance. Top ends of them are attached to the plate *3*, whereas the bottoms are attached to the bottom cover *5*. The body *7* is mounted between the top cover *2* and bottom cover *5*. Both the covers are fastened to the body using the fastening bolts. On the top cover a holder *1* is mounted, which provides the possibility to move the device along the horizontal direction and in such a way to change the axial misalignment of the being aligned components. The central elastic element *8*, which holds the shaft *6* and ensures its angular compliance, also is fixed to the metallic plate *3*. To avoid the magnetization of the being aligned parts, the shaft and also the bushing *9* are made of stainless steel. The stiffness of the device is measured at the end of the shaft and axial stiffness is 4.1 N/mm, transverse is 0.2 N/mm.

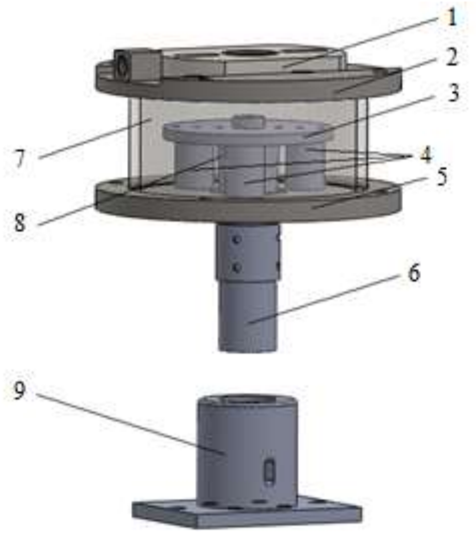


Fig. 1 The passive compliance device with porous rubber elastic elements: *1* – device holder, *2* – top cover, *3* – metallic plate, *4* – elastic rubber elements, *5* – bottom cover, *6* – shaft, *7* – body, *8* – central elastic element of angular compliance, *9* – bushing

The construction of the second type compliance device comprises spring type elastic elements *1*, which are located by 120 degree each from the other (Fig. 2) and aimed to ensure the linear compliance. The central elastic element *2*, the same as in the first type device is made of the rubber. The stiffness of the device is measured at the end of the shaft and axial stiffness is equal to 1.8 N/mm, transverse is 0.15 N/mm.

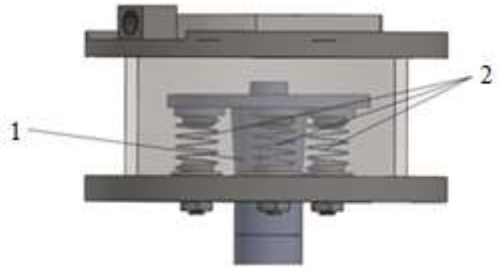


Fig. 2 Passive compliance device with springs: *1* – springs, *2* – central elastic element

3. The experimental setup and technique of experiments

To carry out the experimental analysis of the vibratory alignment and assembly of the shaft and bushing an experimental setup was mounted (Fig. 3). The experimental setup comprises the mount *11*, to which the device *6* of passive compliance is attached. The shaft *5* is attached to the central elastic element of the device and needs to get aligned in respect of the bushing *4*, which is fixed on the platform of the vibrator *3* .While moving the holder of the mount down, the shaft is pressed towards the bushing by the predefined force and displacement is measured by means of micrometer *7*. The axial misalignment of the parts is adjusted by displacing the holder, which is fixed to the top cover of the device, along the horizontal direction, in respect to the mount, The amplitude and frequency of vibrations are adjusted by the signal generator *1*, which via the amplifier *2* transfers the signal to the vibrator. The accelerometer is attached to the vibrator; the accelerometer signal, which is proportional to the amplitude of vibrations, is transferred to the oscilloscope *8* and from the electrical measurement circuit 9 the signal is obtained. That data is processed by the computer *10* and so the alignment duration is determined (Fig. 4). The signal *3* comes from the accelerometer, whereas signal *4* is obtained from the electrical measurement circuit, which supplies the voltage to the oscilloscope at the same moment as the shaft contacts the element, which is mounted within the hole of the bushing. In such a way the end moment of the alignment is defined. In Fig. 4, the interval *1* between the vertical measurement lines indicates the duration of the alignment, whereas the interval between the horizontal lines *2* shows the value of the input voltage.

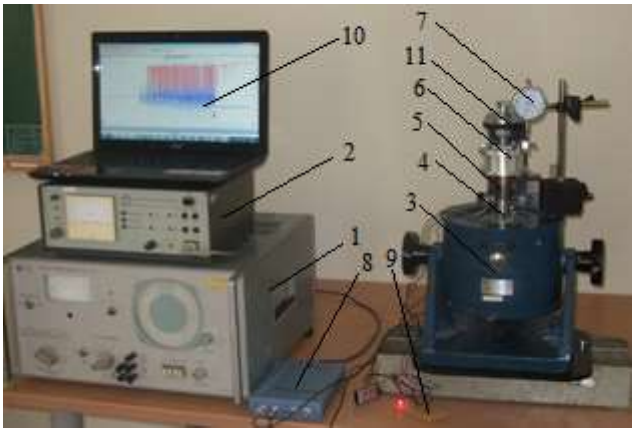


Fig. 3 The general view of the experimental setup for vibratory alignment: *1* – signal generator,*2* – amplifier, *3* – vibrator, *4* – bushing, *5* – shaft, *6* – passive compliance device, *7* – micrometer for measuring of the pressing force, *8* – oscilloscope, *9* – electrical system used to state the insertion time of the shaft, *10* – computer, *11* – mount

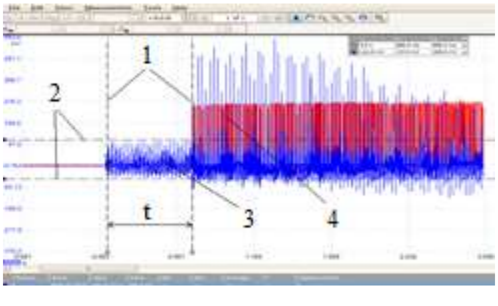


Fig. 4 The schema for alignment duration measurement

The shafts and bushings of different cross-section have been used for the experiments. During the experimental analysis the circular cross-section shaft of 16 mm diameter was aligned in respect of 16.1 mm hole of the bushing, as well as the rectangular cross-section shaft, having dimensions 16x16 mm, was aligned in respect of the 16.1x16.1 mm hole. The parts of rectangular cross section have been aligned only along the one axial direction.

4. Experimental results

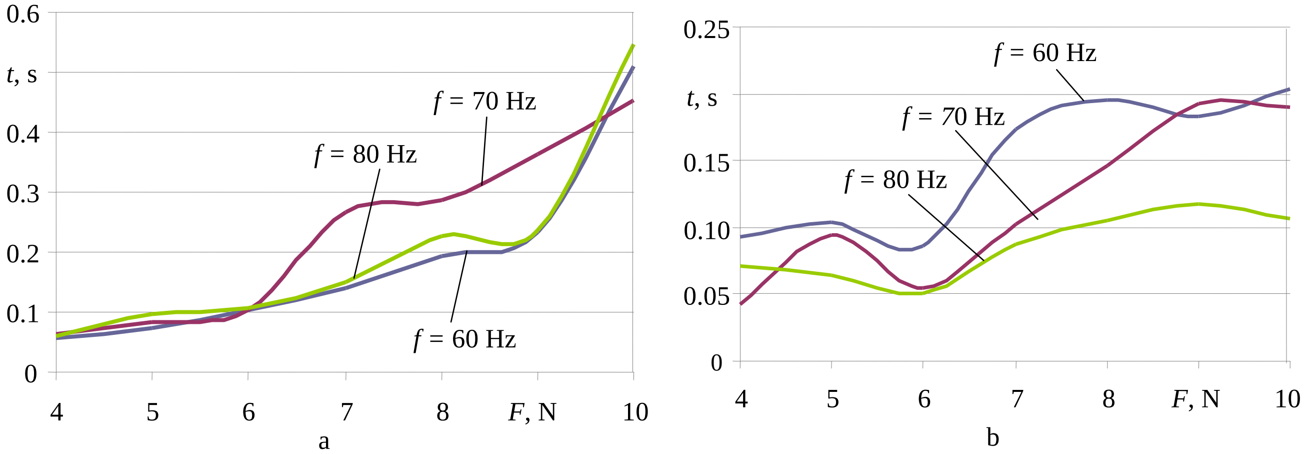
By changing the main parameters, i.e. the axial misalignment between the shaft and the bushing, pressing force, vibration frequency, vibration amplitude, the alignment duration dependencies on the particular parameter were obtained. The ranges of the parameters values are presented in Table.

y adjusting the pressing force within the indicated range and keeping constant both the axial misalignment $(\Delta = 1\ \mathrm{mm})$ and also the amplitude of excitation $(A = 1.2\ \mathrm{mm})$, the alignment test is carried out. The graphical dependences are made under different values of the bushing's excitation frequency. Fig. 5 shows dependences for the rectangular parts, Fig. 6 dependences are for the circular parts.

Table

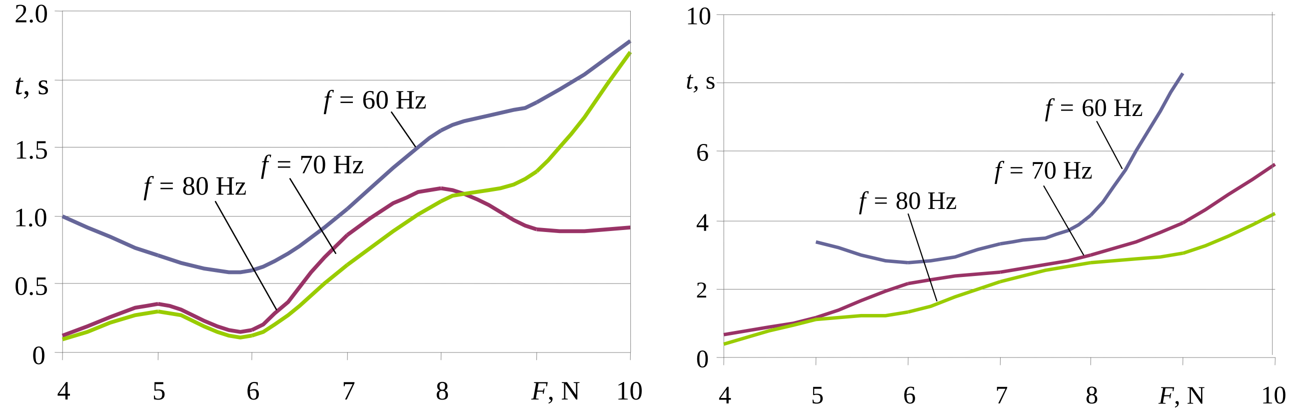
Range of the parameters

Parameter	Range
Axial misalignment Δ	0.5 – 2 mm
Pressing force F	4 – 10 N
Frequency of vibration f	60 – 90 Hz
Amplitude of vibration A	0.8 – 1.6 mm



a b

Fig. 5 The alignment duration dependences on pressing force under different excitation frequency for the rectangular parts: a – using elastic elements made of rubber, b – using spring type elastic element



a b

Fig. 6 The alignment duration dependences on pressing force under different excitation frequency for the circular parts: a – using elastic elements made of rubber, b – using spring type elastic element

If the pressing force F and also the frequency of the bushing's excitation are increased, the duration of the alignment also increases. When elastic elements of the device are made of rubber, the alignment goes significantly faster, if compared to that when spring type elements are used.

The cross-section shape of the parts has high influence on the duration of the alignment. When both the shaft and the bushing are of rectangular cross-section, the alignment goes significantly faster than the circular parts alignment. Under relatively small excitation frequency (60 Hz), by using the spring type elastic elements, the alignment of the circular cross-section parts is not taking place under pressing force less than 5 N and higher than 9 N (Fig. 6, b).

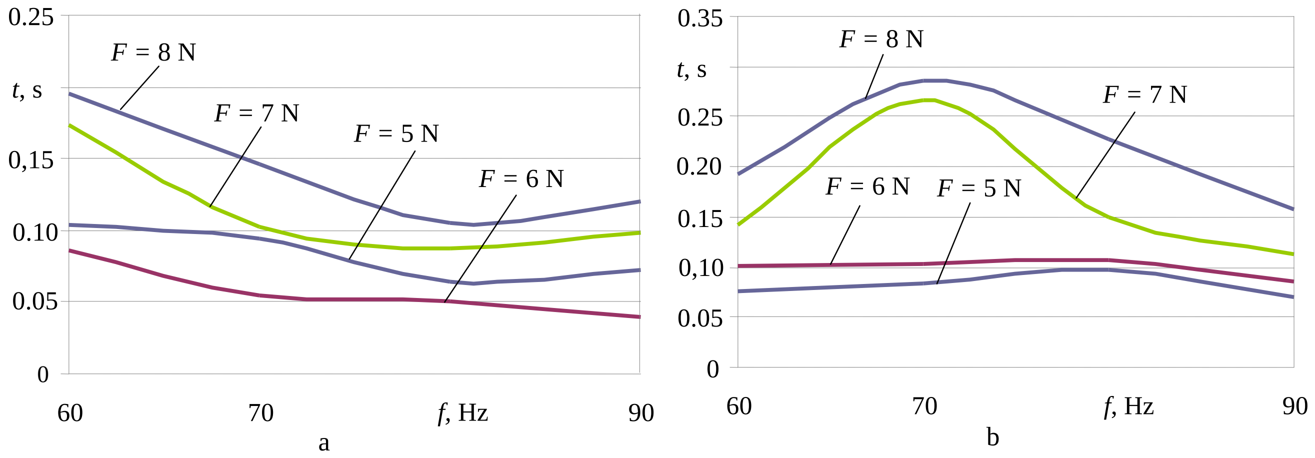


Fig. 7 The alignment duration dependences on excitation frequency under different pressing force for the rectangular parts: a – using elastic elements made of rubber, b – using spring type elastic elements

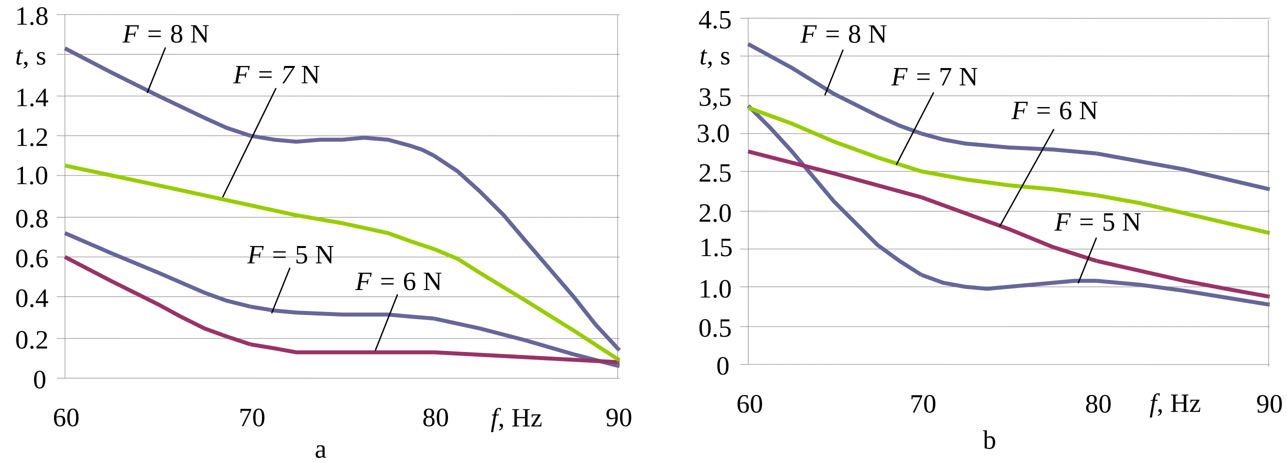


Fig. 8 The alignment duration dependences on frequency under different pressing force for the circular parts: a – using elastic elements made of rubber, b – using spring type elastic elements

The rectangular parts alignment duration dependences on excitation frequency under different pressing force are presented in Fig. 7, and the same for circular parts are shown in Fig. 8, as axial misalignment $\varDelta = 1\,\text{mm}$, the amplitude of excitation is $A = 1$ mm. The dependences show, that increase in excitation frequency f, results decrease in alignment duration, whereas increase in pressing force results increase in alignment duration. The alignment goes significantly faster, when parts are of rectangular cross-section and when rubber elastic elements are used.

The alignment duration dependences on excitation amplitude under different frequencies, considering the rectangular parts, are presented in Fig. 9, the same for the circular parts are given in Fig. 10, under axial misalignment $\varDelta = 1\,\text{mm}$, and pressing force $F = 7\,\text{N}$.

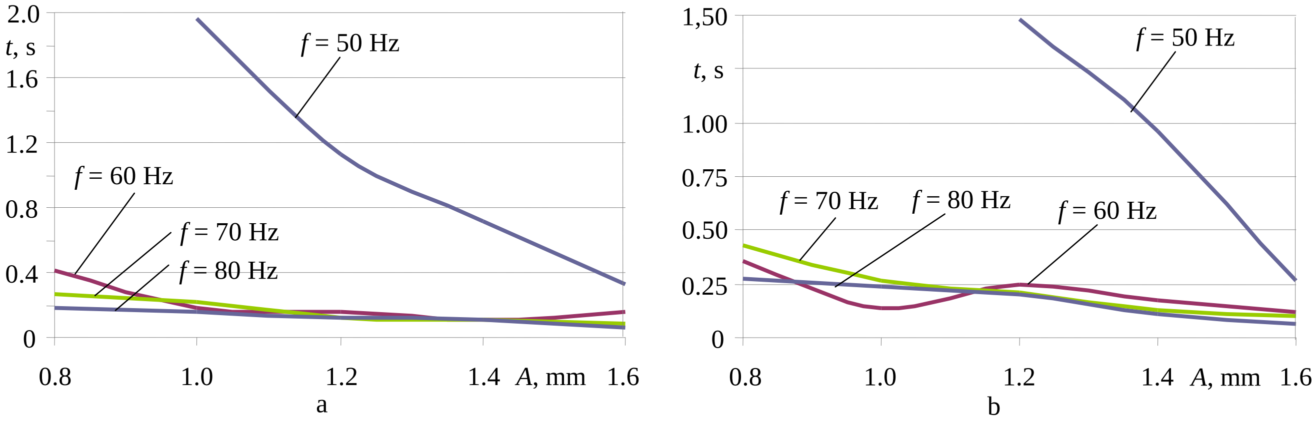


Fig. 9 The alignment duration dependences on vibration amplitude under different excitation frequency for the rectangular parts: a – using elastic elements made of rubber, b – using spring type elastic elements

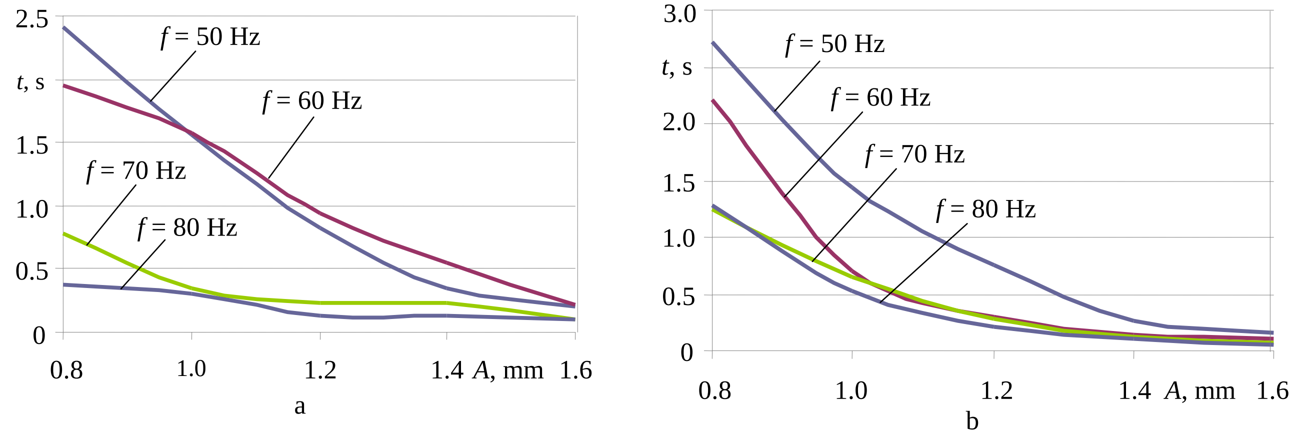


Fig. 10 The alignment duration dependences on vibration amplitude under different excitation frequency for the circular parts: a – using the device with rubber elastic elements, b – using the device with spring type elastic elements

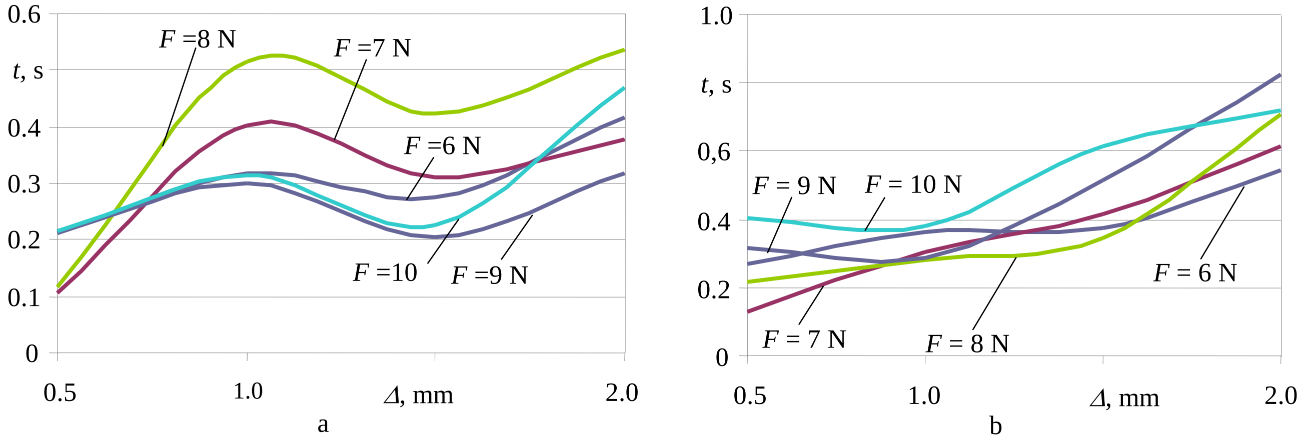


Fig. 11 The alignment duration dependences on axial misalignment under different pressing force for the rectangular parts: a – using the device with rubber elastic elements, b – using the device with spring type elastic elements

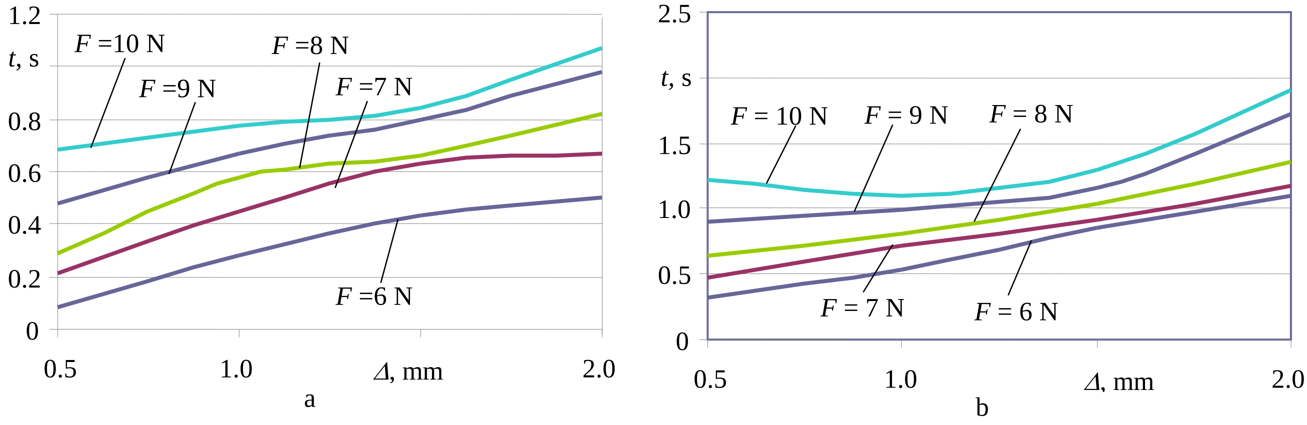


Fig. 12 The alignment duration dependences on axial misalignment under different pressing force for the circular parts: a – using the device with rubber elastic elements, b – using the device with spring type elastic elements

As excitation frequency f and amplitude A are increased, the duration of the alignment decreases. The alignment goes faster, when parts are of rectangular cross-section, as well as when elastic elements are made of rubber.

Rectangular parts alignment is not taking place under relatively small excitation frequency and amplitude ($f = 50$ Hz, $A < 1$ mm) and when elastic elements made of rubber are used (Fig. 9, a). When spring type elastic elements are used, the alignment is not taking place when the amplitude is $A \le 1.2$ mm (Fig. 9, b).

The alignment duration dependences on the axial misalignment Δ under different pressing force for the parts of rectangular cross-section are given in Fig. 11 and for circular parts is presented in Fig. 12, under excitation frequency $f = 60$ Hz and amplitude $A = 1$ mm. While the axial misalignment Δ and pressing force F are increased, the duration of the alignment is increasing. The rectangular parts alignment goes faster and as porous rubber elastic elements are used.

The experimental analysis showed that using the higher stiffness device with the rubber elastic elements, the duration of alignment generally is significantly shorter if compared to that of device with spring elastic elements. Moreover, the sets of the considered parameters exist, under which the alignment is not taking place.

6. Conclusions

1. The alignment duration dependence on the axial misalignment, pressing force, as well as on frequency and amplitude of vibratory excitation was analyzed, considering the parts of circular and rectangular cross-section.
2. It was defined, that increase in frequency and amplitude of excitation, results decrease in the duration of the parts alignment, whereas an increase both in axial misalignment and pressing force, results an increase in alignment duration.
3. When the shaft and the bushing are of rectangular cross-section, the alignment along one axis takes place significantly faster, if compared to that of the circular cross-section parts alignment.
4. If the frequency of excitation is not high enough (50 Hz), the alignment of the parts occurs only under higher excitation amplitude ($A > 1.2$ mm) of the busing.
5. At 60 Hz excitation of the circular cross-section bushing, the alignment of the parts is not taking place if pressing force is less than 5 N or more than 9 N, when springs are used as elastic elements.

References

1. **Lefebvre, T.; Xiao, J.; Bruyninckx, H.; Gersem, G**. 2005. Active compliant motion: a survey. Advanced Robotics 19(5): 479-499. http://dx.doi.org/10.1163/156855305323383767.
2. **Wang, W.; Loh, R.; Gu, E**. 1998. Passive compliance versus active compliance in robot-based automated assembly systems. Industrial Robot: An International Journal 25(1): 48-57. http://dx.doi.org/10.1108/01439919810196964.
3. **Havlik, Š.** 2011. Passive compliant mechanisms for robotic devices.13th World Congress in Mechanism and Machine Science, Guanajuato, Mexico, 19–25 June: 7. http://somim.org.mx/conference_proceedings/pdfs/A12/A12_273.pdf.
4. **Havlik, Š.; Carbone, G.** 2006. Design of compliant robotic microdevices. In Proc.15th Int. Workshop on Robotics, Hungary, June 15 – 17. http://www.meccanicadellemacchine.it/uk/ricerca/Prin2005/it/main%20frame_files/pubblicazioni%20correlate/2%20Design%20of%20Compliant%20Robotic%20Micro-Devices.pdf.
5. **Ciblak, N.; Lipkin, H.** 2003. Design and analysis of remote center of compliance structures, Journal of robotic systems 20(8): 415-427. http://dx.doi.org/10.1002/rob.10096.
6. **Bakšys, B.; Baskutienė, J.** 2012. The directional motion of the compliant body under vibratory excitation. Int. Journal of Non-Linear Mechanics 47: 129-136. http://dx.doi.org/10.1016/j.ijnonlinmec.2012.01.008.
7. **Kilikevičius, S.; Bakšys, B.** 2011. Dynamic analysis of vibratory insertion process. Assembly Automation 31(3): 275-283. http://dx.doi.org/10.1108/01445151111150613.
8. **Bakšys, B.; Baskutienė, J.; Kilikevičius, S.; Chadarovičius, A.** 2010. Experimental analysis of the robotized assembly applying vibrations, Journal of Vibroengineering 12(4): 572-581.

B. Bakšys, J. Baskutienė, J. Vėžys

EKSPERIMENTINIS VIBRACINIO CENTRAVIMO TYRIMAS NAUDOJANT PASYVAUS PASLANKUMO ĮTAISUS

Reziumė

Atlikti stačiakampio ir apvalaus skerspjūvio veleno ir įvorės tipo detalių vibracinio tarpusavio centravimo eksperimentiniai tyrimai, panaudojant skirtingų konstrukcijų pasyvaus paslankumo įtaisus. Sumontuotas vibracinio rinkimo eksperimentinis stendas, kai velenas tvirtinamas pasyvaus paslankumo įtaise ir nustatyta jėga prispaudžiamas prie įvorės, kuri bazuojama ant vibratoriaus platformos ir žadinama ašine kryptimi. Sudarytos centravimo trukmės priklausomybės nuo ašių nesutapimo, vibracinio žadinimo dažnio ir amplitudes bei pradinės prispaudimo jėgos.

B. Bakšys, J. Baskutienė, J. Vėžys

EXPERIMENTAL RESEARCH OF VIBRATORY ALIGNMENT USING PASSIVE COMPLIANCE DEVICES

Summary

The experimental analysis of interdependent vibratory alignment of the shaft and bushing type parts of rectangular and circular cross-section is carried out, using the passive compliance devices of different construction. An experimental setup for vibratory assembly was mounted, when the shaft is fixed in the passive compliance device and pressed by predetermined force towards the bushing, which is based on the vibratory platform and subjected to vibratory excitation along the axial direction. The alignment duration dependencies on axial misalignment, frequency and amplitude of vibrations and initial pressing force of the device were made.

Keywords: vibratory alignment, passive compliance, automated assembly.

Received January 15, 2014
Accepted April 04, 2014