Saitama University
Graduate School of Science and Engineering

DEVELOPMENT OF FIXING SYSTEM WITH 2-AXIS LOW FREQUENCY VIBRATION FOR MICRO DEEP DRILLING

(微細深穴あけ加工のための 2 次元低周波振動付与工作物保持システムの開発**)**

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Ph.D. dissertation

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Sincerely,

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ABSTRACT

In recent years, the miniaturization of the nozzles in the gas turbines for the thermal power stations or in the marine diesel engines has been required to cope with the technological innovations of the high-pressure fuel injection. Therefore, it is considered that diameter of future holes in the nozzles must be reduced to 0.2 mm or below from the current value of 1.0 mm. The aim of the previous study was improvement of micro deep drilling at hard materials (such as Stellite, Hastelloy or SUS304), which hole diameter is 0.2 mm or below and an aspect ratio (L_h/d_h) of 20 or above. The micro deep drilling of hard materials is required including of step feed in the process that increases machining time. To decrease the machining time by increasing the step feed while maintaining tool life, two methods had been proposed. Increasing of the step feed is providing by reducing of cutting force. The first method reduces the cutting force by applying of the ultrasonic vibration in the micro drilling by oscillation of the tool (frequency ~40 kHz, amplitude ~1 μm). The second method also reduces the cutting force by applying of low frequency vibration in the micro drilling by oscillation of workpiece (frequency ~150..250 Hz, amplitude ~10 μm). Each method can improve the drilling process independently because of different vibration parameters by frequency (~40 kHz and ~200 Hz) and amplitude (1 μm and 10 μm). Presence of two types of vibrations makes requirement of different way including its to the drilling process (oscillating of the tool and the workpiece).

The nozzle has a curved shape in place for the drilling that means processing of the 2-axis drilling. To cope with the drilling process assisted by low-frequency vibration of curved surface, we have developed fixing system for 2-dimensional vibration. The fixing system for 2-dimensional vibration (FS2DV) consists of horizontal and vertical vibration sources with spring systems along directions of the sources. We select voice coil actuator (VCA) for using as the vibration source to supply enough of power for oscillation the workpiece with one spring system and connecting parts. The direction and value of the 2-dimensional vibration is passively controlled by amplitude ratio and values of the vibrations from the vertical and horizontal sources. The spring systems have variable rigidities for using of the FS2DV with different vibration frequency. The vertical spring system consists of number of plates (made of spring steel), which are fixed between internal and external collars. Rigidity of the vertical spring system is varied by changing number and shape of the spring plates. The horizontal spring system consists of three sets with belleville springs located at 120º relatively to each other, which are fastened on the external collar of the vertical spring system.

Rigidity of the horizontal spring system is varied by changing number and assembling combination of the belleville springs.

Thrust force (6…10N) from drilling process induces unintended displacement of the workpiece. Depending on unbalancing between rigidities of the vertical and horizontal systems, unintended displacement may create hole diameter error. Setting of the spring systems with balanced rigidities can minimize influence of the unintended displacement on the hole diameter error. Analysis of the vertical and horizontal spring systems was made for successful using of the FS2DV during 2-axis micro deep drilling process assisted by low frequency vibration. Based on this analysis, the setting requirements of the FS2DV have been proposed for particulars vibration frequencies.

The FS2DV have been verified for controlling vibration under loading force (which imitate the axial force of drilling) with different angles (15°, 45°, 75°) or frequency (149.5 Hz, 198.0 Hz, 252.3 Hz). Rigidities of the spring systems were adjusted with values that they satisfy to mention above requirements for each frequency (800 kN/m, 1050 kN/m, 1600 kN/m). Resultant vibration was measured by laser sensor of displacement in vertical and horizontal directions. The analysis of the experimental results has been made. The total hole diameter error can be 2.1…3.7 μm (depending on vibration frequency and direction), but it is not big compering with 6 μm, which is a standard hole's tolerance with diameter 0.2 mm.

At the final, we observe the deviation of the thrust force during vibration process, because the thrust force is important parameter of the drilling. The trust force may enlarge the temperature of drill's edge or initiate breakage of the drill. So the thrust force measuring is made during vibration process with frequency 198.0 Hz, amplitude 10 μm and direction 45º. The thrust force deviation is present during the vibration process. It is up to 1.3 N with the frequency of workpiec's vibration and 0.4 N with frequency about 1200 Hz. The thrust force is not big relatively to the value of thrust force of the drilling at the study of Dr. Nanbu (9 N).

From experimental and analytical study, we obtain the following conclusions:

It was proposed concept for passive controlling of the vibration. It consists in passive controlling of two the independent vibrations in the vertical and horizontal direction.

The fixing system of 2-dimensional vibration (FS2DV) has been designed for fulfilling this concept. The FS2DV consist of the vertical and horizontal voice coil actuators with spring systems along action of each actuator.

We have the developed FS2DV with usage range frequency of 150…250 Hz, amplitude 10 μm. The system can be properly used with cutting force (up to 10 N) and any angle (from 0 to 90°) by selecting appropriate rigidities for the vertical and horizontal spring systems.

Finally, we observe the deviation of the thrust force. The trust force is deviating during the vibration process with amplitude up to 1.3 N and frequency of the workpiece's vibration.

Contents

Chapter 1

Background

1.1. Micro-deep drilling of hard materials

In recent years, the miniaturization of nozzles in gas turbines for thermal power stations or in marine diesel engines has been required to cope with the technological innovations of high-pressure fuel injection(1). Therefore, it is considered that the diameter of future holes in nozzles must be reduced to 0.2 mm or below from the current value of 1.0 mm. The aim of the previous study was improvement of micro deep drilling at hard materials (such as Stellite, Hastelloy or SUS304), which hole diameter is 0.2 mm or below and an aspect ratio (L_h/d_h) of 20 or above. From the viewpoint of considering practical utility, the machining efficiency must be improved and the machining time must be reduced. To short the machining time while maintaining the tool life, the addition of ultrasonic vibration and low-frequency vibration to increase the step feed was examined.

In micro deep drilling, it is difficult for cutting fluid to reach the cutting point, heat cannot easily be dissipated, and the cutting edge tends to have a high temperature. Furthermore, since the size of the flute with respect to the diameter of the drill is small for a drill having a regular diameter, it is difficult to smoothly dispose the chips. These tendencies become more prominent as the aspect ratio increases. Thus, in micro deep drilling, such step feeding as depicted in Figure 1.1 is conducted

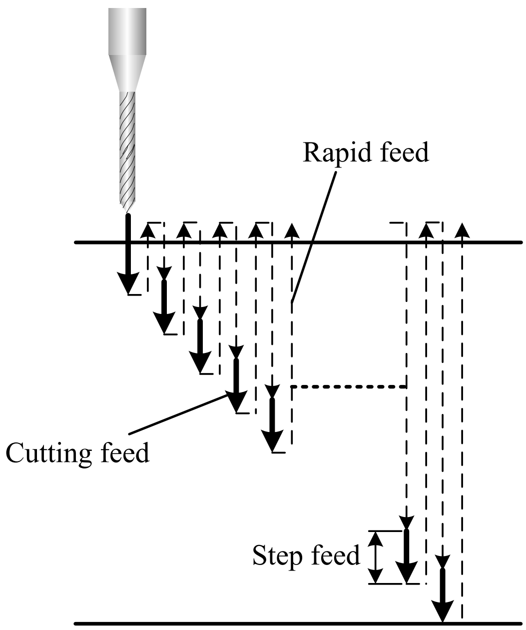


Figure 1.1. Step feed drilling

to cool the cutting edge and dispose the chips. However, since the step feed must be greatly reduced in a micro deep hole with a large aspect ratio, the number of steps increases, and the non-cutting time devoted to reciprocating the rapid feed is increased. Therefore, a serious practical problem, i.e., a marked increase in machining time, arises. As a typical example of conducting micro drilling to obtain a hole with a diameter of 0.2 mm, Figure 1.2 shows the relationship between the step feed and the machining time of a single hole when performing drilling to obtain a hole with a depth of 4 mm at a rapid feed speed of 200 mm/min and cutting feed speeds of 10 to 40 mm/min. The machining time decreases rapidly as the step feed increases.

Effects such as a reduction in the cutting force due to the addition of ultrasonic vibrations in micro drilling have been reported(3)~(6). The authors have suggested conditions that extend tool life upon adding ultrasonic vibrations when drilling a hole of diameter 0.2 mm or below with an L/D of 20 or above (3).

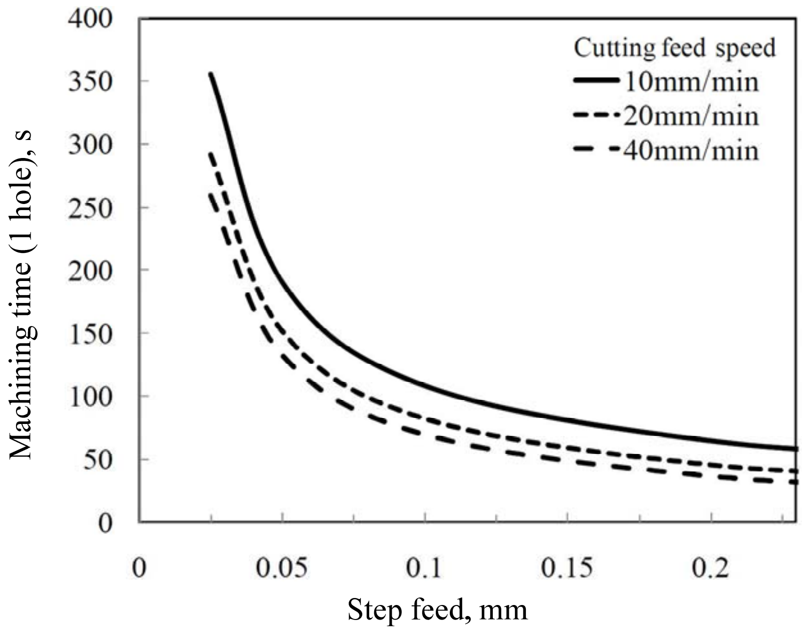


Figure 1.2. Relation between the step feed and the machining time

1.2. Method of drilling assisted by vibrations(3), (7)

Two methods had been proposed to short the machining time by increasing the step feed while maintaining a long tool life (Figure 1.3). Increasing of the step feed is providing by reducing of cutting force. The first method reduces the cutting force due to addition of the ultrasonic vibration in micro drilling (frequency ~40 kHz, amplitude ~1.2 μm). The vibration is applied to drilling by oscillation of the tool. This method is widely used. The second method reduces the cutting force due to the addition of low frequency vibration in the micro drilling by oscillation of the workpiece (frequency ~190 kHz, amplitude ~10 μm) [(7)~(11)]. The vibration is involved to drilling by oscillation of the workpiece.

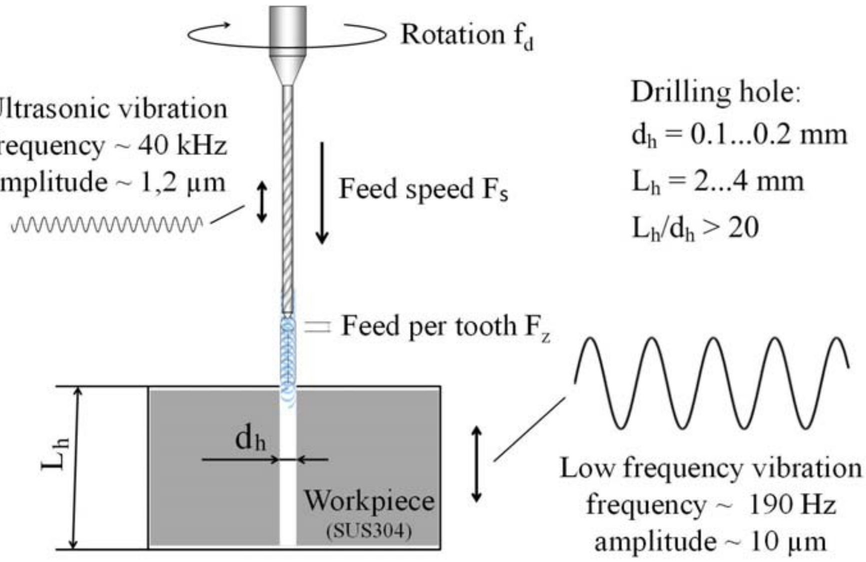


Figure 1.3. Drilling assisted by vibrations

Both methods can be applied to drilling process concurrently. Each method can decrease the cutting force of the drilling process independently because of different vibration parameters by frequency, amplitude [(11)]. The workpiece is chosen as a oscillated purpose of the low frequency vibration because it will be hard to oscillate the tool with large parts of oscillating spindle (for the ultrasonic vibration).

1.3. Method assisted by low frequency vibration[(7)]

The method of the drilling assisted by the low frequency vibration (Figure 1.4) is corresponded to this study research and it is used not as wide as the method of the drilling assisted by the ultrasonic vibration. So, below, it is explained about this method more in detail.

Figure 1.5 shows axial displacements of drill's edge relatively to the workpiece during the drilling process assisted by the low frequency vibration. Last cutting line closes volume of chip-making.

Succession from involving of the low frequency vibration can be predicted by two parameters cutting time ratio R_c (quotient of cutting time T_c to vibration period T_v) or positive velocity ratio R_v (quotient of time when drill's edge moving to workpiece T_p to cutting time T_c). The cutting time ratio is smaller, the better cooling of drill's edge and drilling process will be. The positive velocity ratio is bigger, the smaller cutting resistance and better drilling process will be.

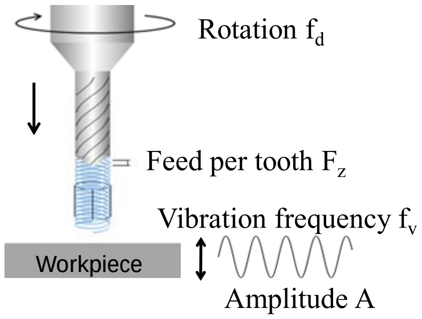


Figure 1.4. Drilling assisted by low frequency vibration

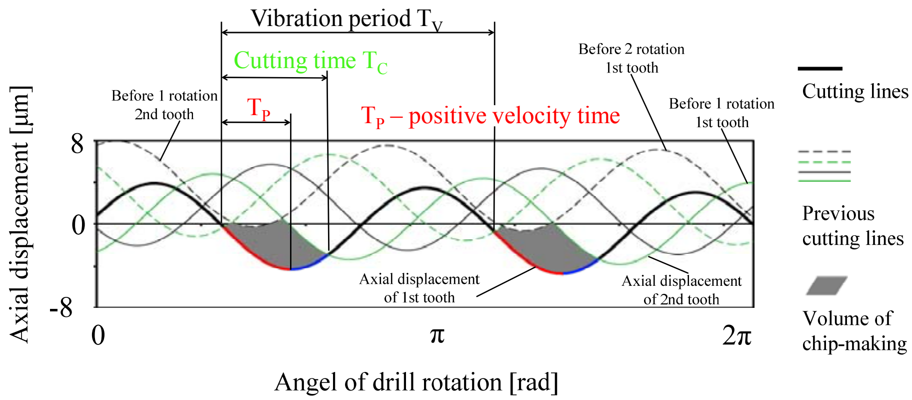


Figure 1.5. Axial displacement of the drill's edge

Depend on amplitude ratio (quotient of vibration amplitude to feed per tooth A/F_z) and frequency ratio (quotient of vibration frequency to spindle speed of the f_v/f_d), it is possible creation a discontinuous contact of the drill with the workpiece. Figure 1.6 shows variations of drilling assisted by low frequency vibration for different values of amplitude or frequency ratios.

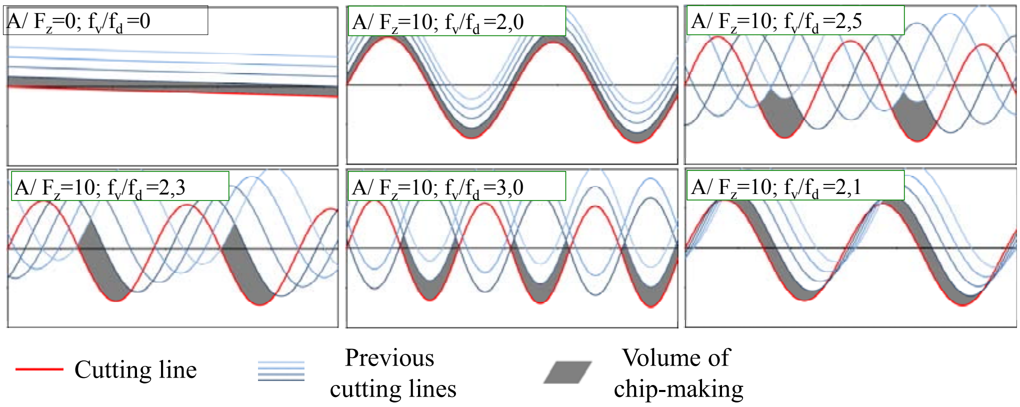


Figure 1.6. Variations of the drilling assisted by the low frequency vibration

Parameters the cutting time ratio and the positive velocity ratio with measuring of tool wear have been observed. The best drilling process corresponds to applied vibration with values of amplitude ratio A/F_z = 10 and frequency ratio f_v/f_d = 2.3 (if feed per tooth and speed spindle of drilling are 1 μm/tooth and 5040 min^{-1}, then vibration frequency and amplitude are 10 μm and 193 Hz respectively). Exact values of amplitude and frequency ratios are important. Because, if frequency ratio is changed from 2.3 (193 Hz) to 2.2 (181 Hz), then the tool wear are increased in 5 times. Also the value of vibration frequency should be changed in some range for case of changing spindle speed of the drilling.

Chapter 2

Objective of study

The previous study consisted in developing of the drilling process. All drilling were made for the 1-axis process. The workpiec's vibration (plate) is produced by piezo electrical actuator. Micro drilling of holes is required for the manufacturing of the nozzle at the gas turbine. But the nozzle's surface has curved shape in place for the drilling that means processing of the 2-axis drilling. To involve the method of drilling process assisted by low frequency vibration (independently from ultrasonic vibration of tool), the workpiece should be oscillate with controlled variable angle, which would correspond to drilling direction. So the workpiece should be oscillated with controlled variable 2-dimensional low frequency vibration to correspond with drill's direction (Figure 2.1).

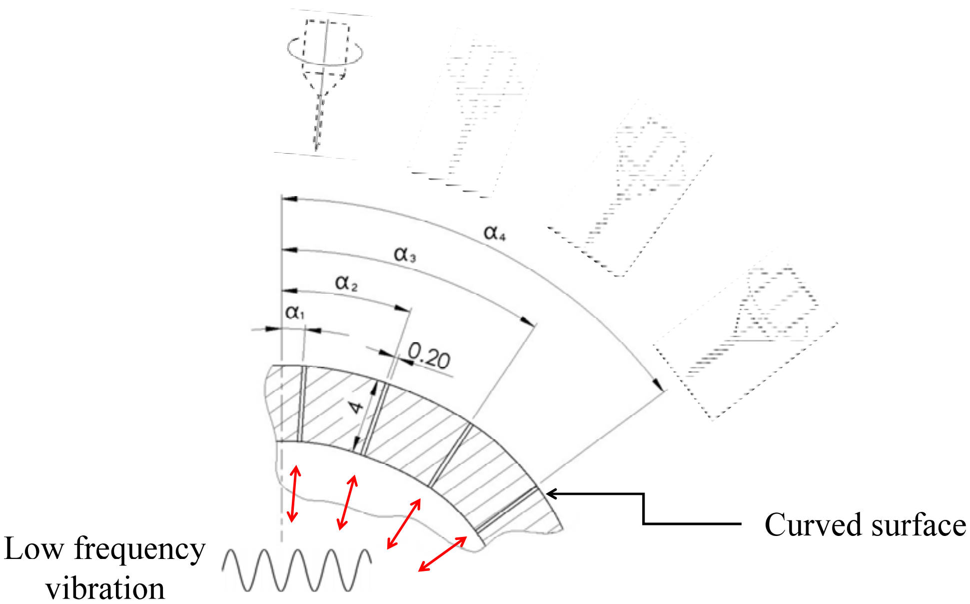


Figure 2.1. 2-axis drilling assisted by the low frequency vibration

For fulfilment of this objective it is proposed fixing system with the 2-dimentionally passively controlled low frequency vibration. Main requirement to the fixing system are next:

- vibration frequency should have exact value for valid applying of the low frequency vibration to the drilling process;
- vibration frequency should be regulated in range of 150…250 Hz in case of changing speed spindle of the drilling;

- the workpiece should be oscillated with vibration amplitude of 10 μm;
- the direction of the vibration should be regulated with angle from 0 to 90°;
- The system should oscillate the workpiece with mass up to 250 g (5 times of the typical nozzle's mass)

Chapter 3

Fixing system for 2-dimensional vibration (FS2DV)

For satisfying mentioned above requirements, it is designed the fixing system for 2-dimensional vibration (FS2DV). The FS2DV consist of two independent vibration sources with spring systems along action of each vibration source (Figure 3.1). The controlling is passive and provided by the vibration sources in horizontal and vertical directions. The resultant vibration equals vector summing of the vertical and horizontal vibrations. For success summing of vibrations, the vertical and horizontal vibrations should be synchronized by frequency and phase, otherwise the resultant vibration becomes non-harmonic or elliptical.

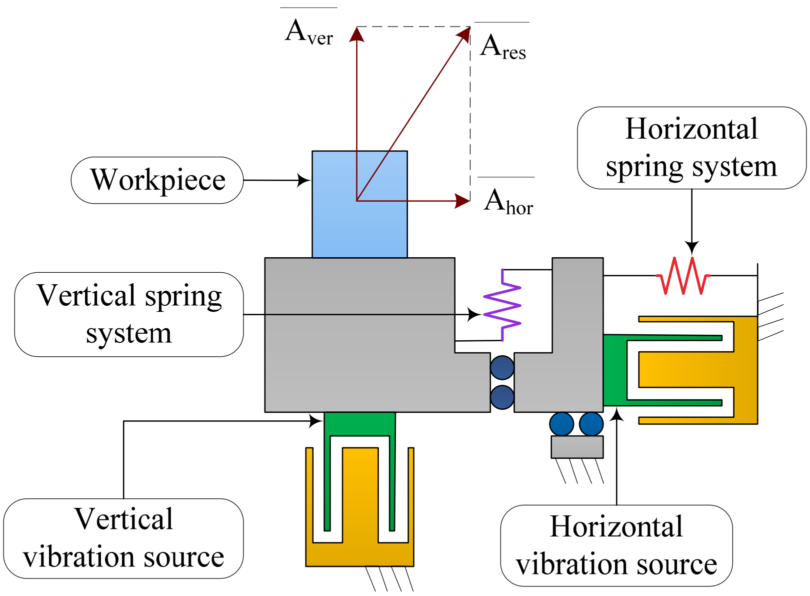


Figure 3.1. Scheme of the FS2DV

Equation (3.1) shows solving of vibration amplitude A from oscillated force with frequency f_v and amplitude F_o for spring system with rigidity k and oscillated mass m.

$$A = \frac{F_o}{\sqrt{(4\pi^2 f_v^2 - \frac{k}{m})^2}}. \qquad (3.1)$$

3.1. Oscillating by voice coil actuator

Constructional connection of the workpiece with two vibration sources makes additional mass to the oscillated part that does not allow using piezo electrical actuator as the vibration source.

For that reason we have tried to use voice coil actuators with spring system. In our research, we have introduced two type of voice coil actuator (Akribis AVM30-15 and AVM35-HF-7) for oscillation in the vertical and horizontal directions. Both actuators are executed by driven sinusoidal current to produce sinusoidal force with amplitude up to 14.7 N and 28.8 N respectively.

Figure 3.2 shows electrical schematic for driving of the actuators. The vertical and horizontal voice coil actuators are driven by one input current signal. This fact gives opportunity to provide horizontal and vertical oscillated forces which are synchronized by frequency and phase. Amplitude of signal for each actuator is managed by resistance of potentiometers R_1 or R_2 (Equation 3.2).

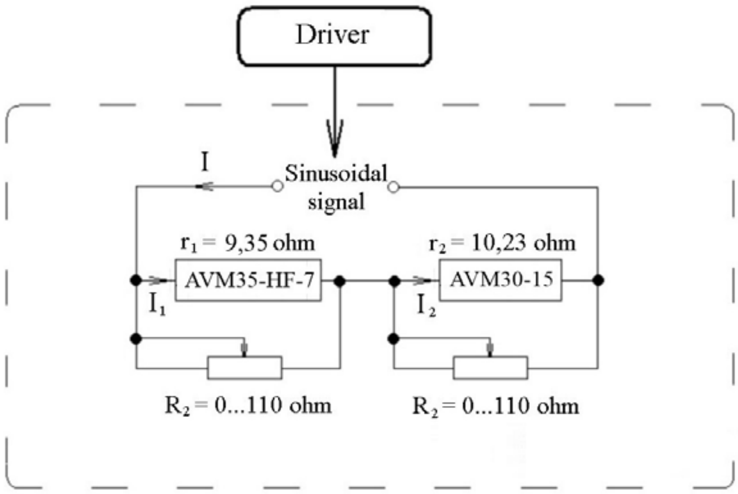


Figure 3.2. Current driving of the voice coil actuators

$$I_{1,2} = I \frac{R_{1,2}}{R_{1,2}+r_{1,2}} \quad (3.2)$$

3.2. Design of FS2DV

Figure 3.3 show design of the FS2DV. The fixing system contains of vertical horizontal spring system (bellow it will be explained about them more in detail), two voice coil actuators, bearing sliders and three-jaw chuck. Rigidity of each spring system can be adjusted for using of the FS2DV with different vibration frequency. The workpiece is fastened in the vertical spring system by clamping bolts. The horizontal spring system is fastened to base of the three-jaw chuck through bearing sliders with allowing translational displacement in horizontal plan. Jaws of the three-jaw chuck make initial spring compression in the horizontal spring system for properly it’s working.

Two voice coil actuators are fastened in the vertical or horizontal directions. Rotors of voice coil actuators are fastened to the workpiece. Stators of voice coil actuators are fastened to the base of the three-jaw chuck. Between stator and rotor is fastened corresponded the vertical or horizontal spring systems and bearing slider for allowing necessary amplitude and smooth of the vibration.

The detailed drawings of the FS2DV's elements are given in Appendix A.

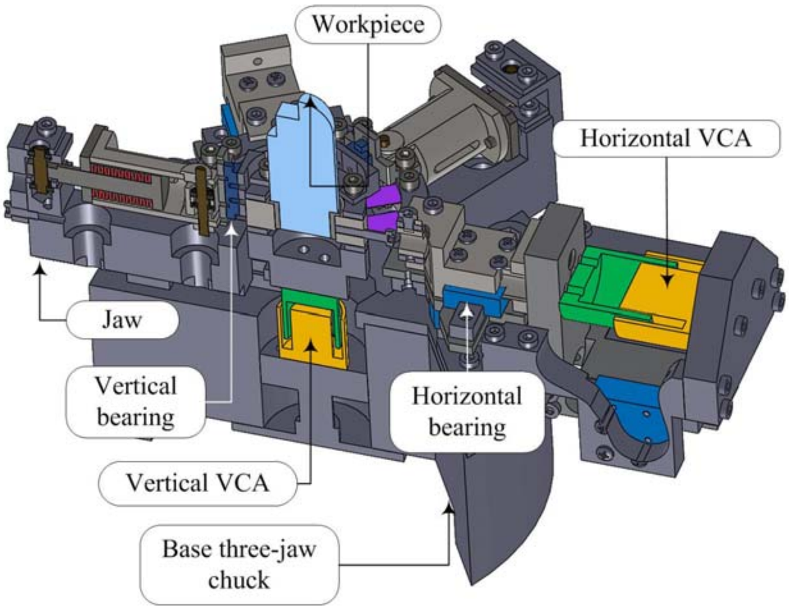


Figure 3.3. Fixing system for 2-dimenstional vibration

3.3. Vertical and horizontal spring systems

The vertical spring system consists of number of plates which are fixed between internal and external collars (Figure 3.4).

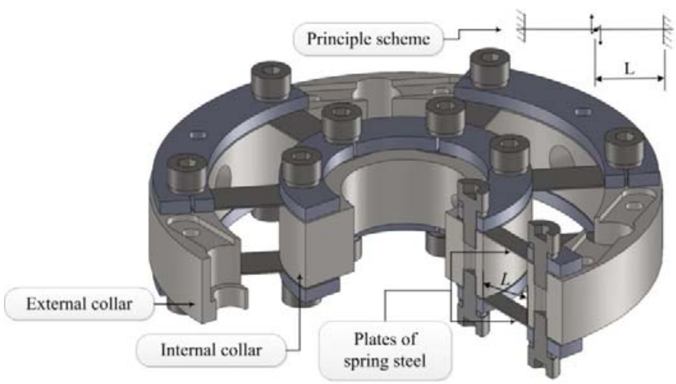


Figure 3.4 Vertical spring system

The plates are made of special spring steel with thickness of 0.3 or 0.6 mm. Rigidity of the vertical spring system k_{ver} is solved by Equation 3.3 and it can be adjusted by number of the plates N_p, and shape sizes *b, h, L* of the plates (width, thickness and length, Fig. 3.5)[12].

$$k_{\text{ver}} = \frac{\text{N}_p \cdot \text{E} \cdot \text{b} \cdot \text{h}^3}{L^3} \tag{3.3}$$

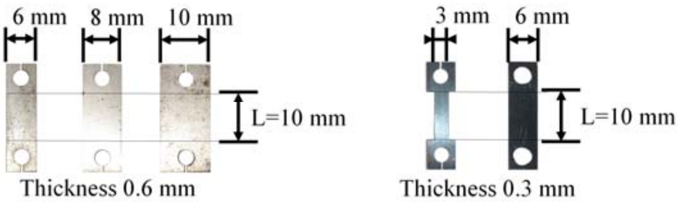


Figure 3.5. Types of used spring plates

The horizontal spring system consists of three sets of belleville springs located at 120º relatively to each other, which are fastened on the external collar of the vertical spring system (Figure 3.6).

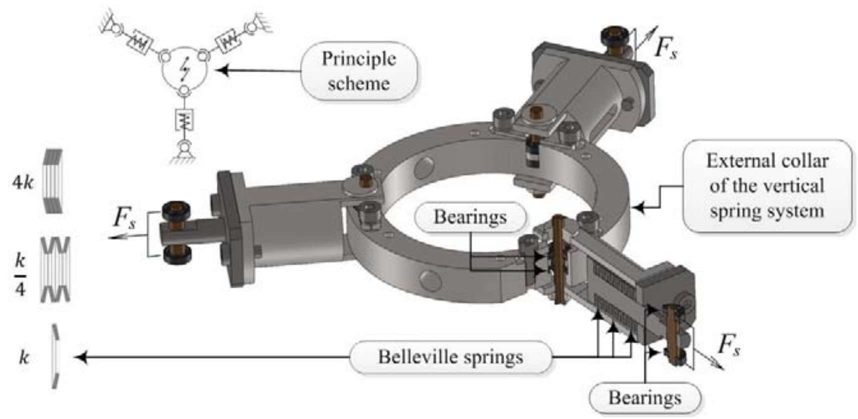


Figure 3.6. Horizontal spring system

The detailed drawings of the vertical or horizontal spring's elements are given in Appendix A.

For easier solving rigidity of horizontal spring system, it is possible to simplify its scheme (Figure 3.7)

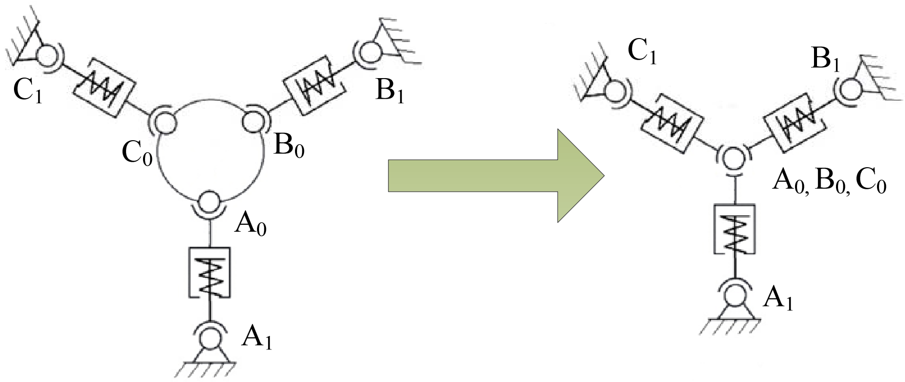


Figure 3.7. Simplifying of horizontal spring system's scheme

Figure 3.8 shows forces, which is appeared in the horizontal spring system after displacement $x_d = OO_1$ with angle γ. Resultant rebalance force F_{ABC} have angle β from direction of the displacement OO_1. In case of small displacement (up to 0.1 mm), rigidity of the spring system is varied from $1.51k_{set}$ to $1.55k_{set}$ depending on the direction of application of radial force, where k_{set} is rigidity of the one set with the belleville springs. In case of displacement x_d = 0.1 mm or less, angle β is varied from 0 to 0.8° depend on angle γ (direction of the displacement OO_1). These facts mean possibility of the horizontal spring system using in the vibration process in the proper way for amplitude up to 100 μm (our objective vibration amplitude is 10 μm).

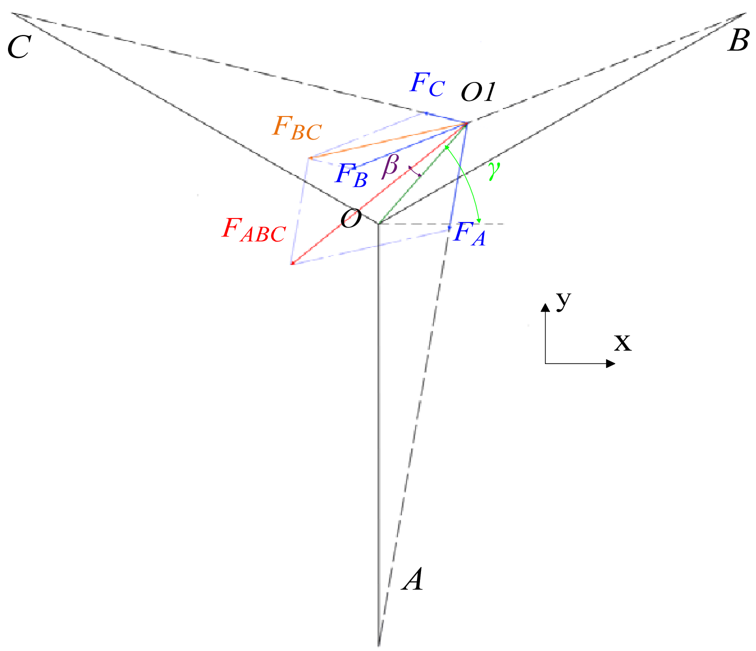


Figure 3.8. Forces in horizontal spring system

The horizontal spring system has almost same rigidity in any radial direction that gives opportunity to improve to the fixing system with 3-dimensional low frequency vibration by adding rotary axis to the horizontal vibration source or applying of the third voice coil actuator.

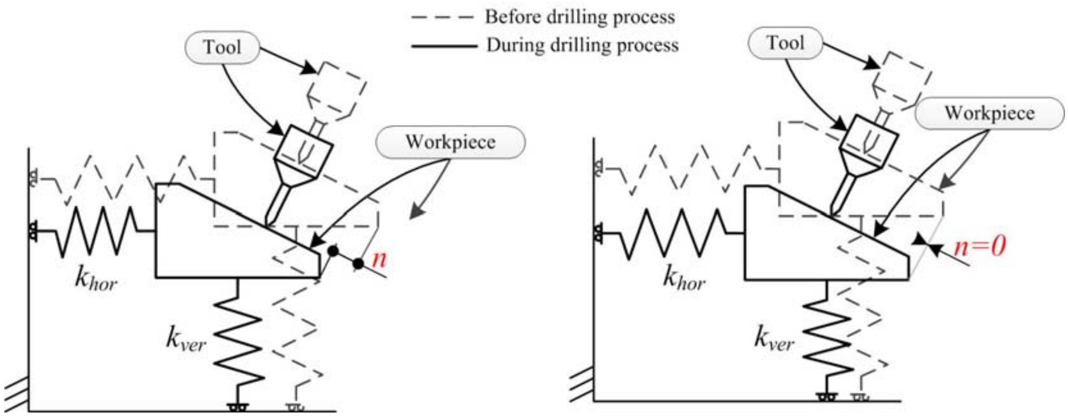
Rigidity of the horizontal spring system can be adjusted by assembly order and number of the belleville springs. The belleville springs should be compressed for correct it's working in both direction, that corresponds to our case of vibration. For that reason initial force is applied to each set of belleville spring F_s. The initial force almost does not effect on the radial rigidity of the horizontal spring system. The horizontal spring system has small rigidity to tangential load. This problem is determined with connection method of the vibration source by excluding of rotary displacement.

3.4. Setting of the spring systems

The vertical and horizontal spring systems allow for vibration process, but they create unintended displacements of the workpiece from thrust force during the drilling process. In this report, analysis of the horizontal and vertical spring systems was made to decrease influence of the unintended displacement of the workpiece on the drilling process.

The FS2DV has the vertical and horizontal spring systems with variable rigidity. If the vertical and horizontal spring systems have different rigidities each other, the workpiece is shifting by an amount n relative to the tool (Figure 3.9a). In this case, radial force in the tool is increased during the drilling process that may create hole diameter error. If the vertical and horizontal spring systems have equal rigidities, the workpiece displaces along the drilling (Figure 3.9b). In this case, the hole diameter error caused by using spring systems is minimized. Problem of the unintended displacement of the workpiece during drilling process should be minimized by adjusting of the vertical and horizontal spring systems with equal rigidity for the vibration process.

Main parameters of workpiece's oscillating by the FS2DV are amplitude vibration of 10 μm, range frequency vibration of 150…250 Hz and variable angle of direction from 0 to 90°. For intended frequency vibration (in range 150…250 Hz), the spring systems must be set with appropriate rigidities to provide amplitude vibration of 10 μm and controlling angle of direction (from 0 to 90°). Setting of appropriate rigidities can be based on frequency-amplitude characteristics of the vertical or horizontal spring system with different rigidities.



a) rigidities of horizontal and vertical springs are different ($k_{hor}< k_{ver}$)

(b) rigidities of horizontal and vertical springs are equal ($k_{hor}= k_{ver}$)

Figure 3.9. Displacement of the workpiece during the drilling process

Frequency response analysis is made by laser sensor measuring of amplitude vibration A of the workpiece for successive values of frequency vibrations f (Fig. 3.10). The workpiece is oscillated by the voice coil actuator with same amplitude F_o of a vibration force for whole experiment. To prevent the extremely big vibration amplitude at a nature frequency mode, the force F is applied with amplitude F_o of 20% (5.8 N and 3.0 N for the horizontal and vertical actuators respectively) from maximum possible value.

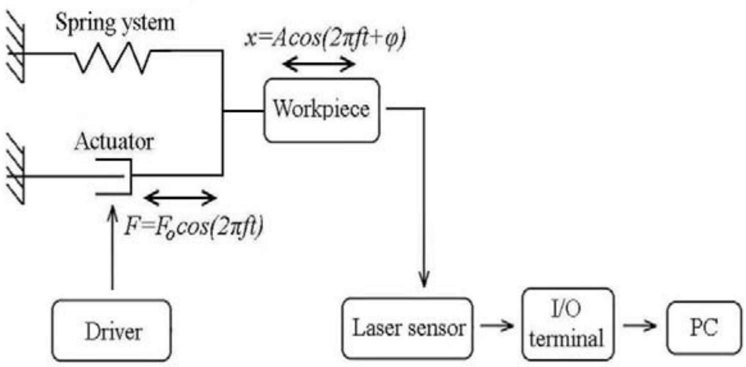
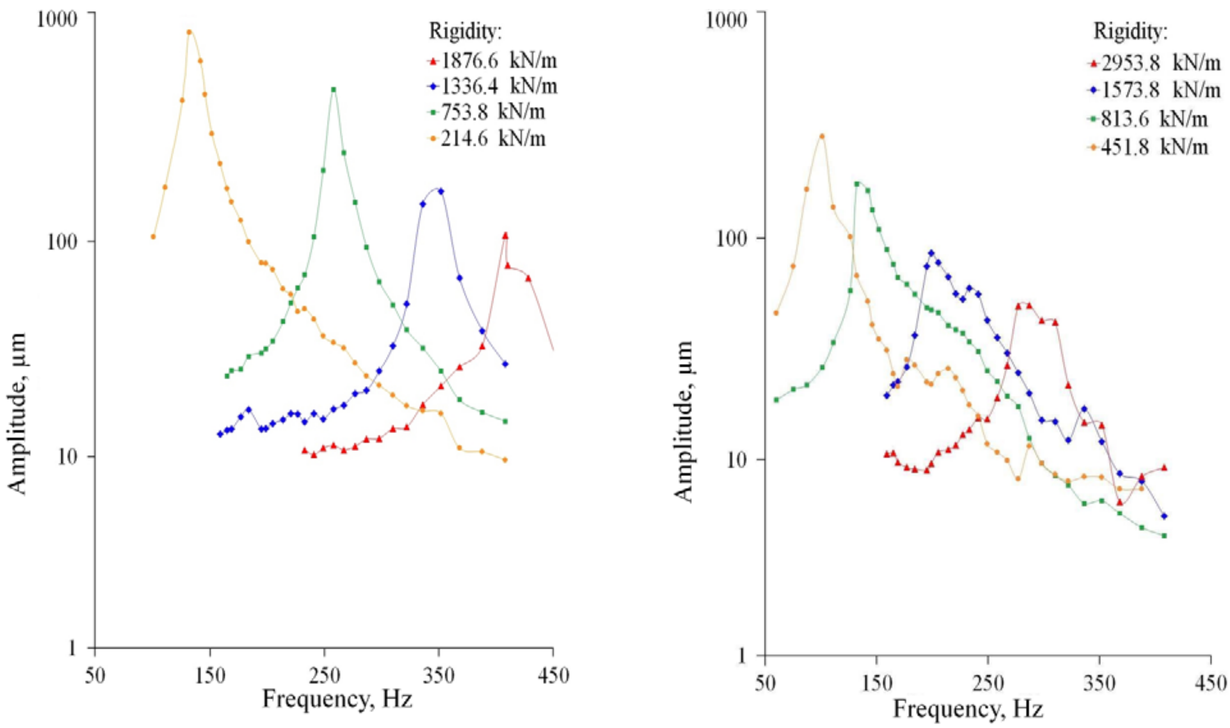


Figure 3.10. Scheme of amplitude vibration measuring

Figure 3.11 shows frequency-amplitude characteristics of the horizontal or vertical spring system on several rigidities. This experiment was made for the horizontal and vertical spring systems with various rigidities (from 215 kN/m to 1876 kN/m for the vertical spring system and from 416 kN/m to 2909 kN/m for the horizontal spring system). The lowest mechanical stiffness between the actuators and the workpiece was measured by setting of the vertical and horizontal spring system with the biggest possible values of rigidities. The lowest mechanical stiffness in the horizontal direction is 5570 kN/m, in the vertical direction is 4640 kN/m. So, the mechanical stiffness of the FS2DV is significantly high and its influence on frequency-amplitude characteristics can be ignored. To adjust the FS2DV for the particular frequency, the vertical and horizontal spring systems should be set with rigidities according to these frequency-amplitude characteristics.



(a) vertical spring system (b) horizontal spring system

Figure 3.11. Frequency-amplitude characteristics

Setting of rigidities of the horizontal and vertical systems for particular frequency vibration should be satisfied to next conditions:

- rigidities of the vertical and horizontal spring systems should be equal (for providing displacement of the workpiece along line of loading force with any direction from 0 to 90°, that is why this requirement is the most important);
- rigidity of the spring system should be not in resonance (for preventing of the non-steady vibration under the force loading);

- rigidity of the spring system should be not far from the resonance mode (for avoiding fluctuation of vibration and supplying enough of power by vibration source for providing necessary amplitude value);
- in case of phase difference between the vertical and horizontal vibrations, it should be corrected by applied additional force F_s (Figure 3.5) to the sets with the belleville springs (phase of horizontal vibration is shifted and amplitude of vibration is decreased almost without changing of spring system's rigidity).

3.5. Phase difference correction

Figure 3.12 shows frequency-response characteristics of the vertical and horizontal spring systems for vibration frequency 250 Hz. It is satisfied to all mentioned requirements. The rigidities are nearly same and equal about 1520 kN/m, not in resonance mode and too far from resonance mode.

Resonance modes of the vertical and horizontal spring systems with rigidities about 1520 kN/m are situated by different sides (left and right) from defined vibration frequency 250 Hz. This circumstance means that phase difference between vertical and horizontal vibrations will be 180º. But this effect is not a problem for the our fixing system because it can be solved by using different connection methods of the actuators. In this case, actuator will be driven by signals with phase difference 180º.

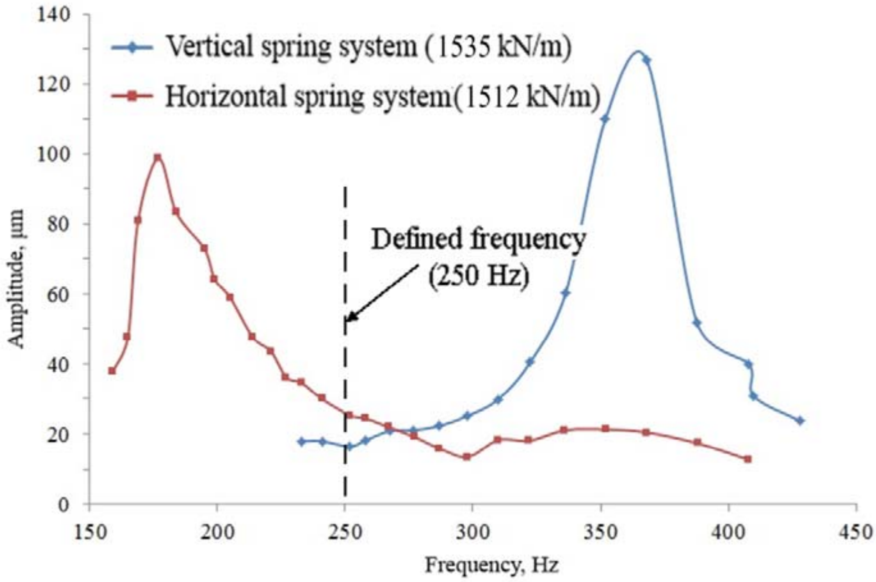


Figure 3.12 Frequency-response characteristic of the vertical and horizontal spring systems for vibration frequency 250 Hz

Chapter 4

Vibration controlling

4.1. Experimental setup

Figures 4.1, 4.2 show a scheme of experimental setup and its photograph. The workpiece is oscillated by the FS2DV, which is set by rigidities of the vertical and horizontal systems for defined frequency. The vibration is verified by measurement of displacement in vertical and horizontal

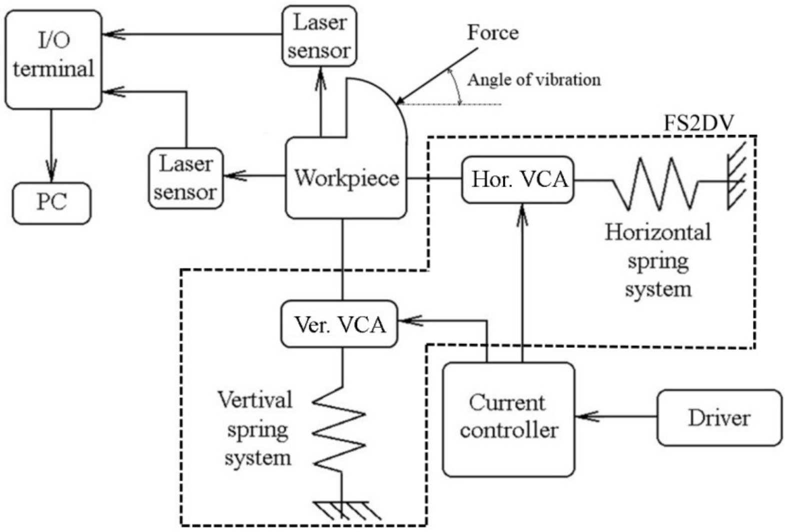


Figure 4.1. Scheme of experimental setup

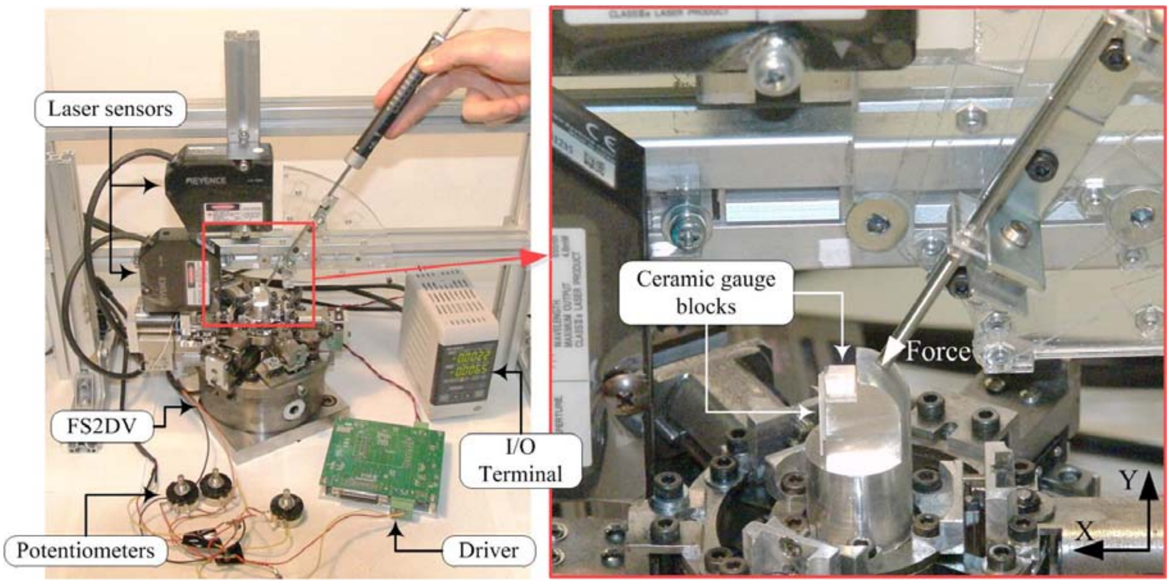


Figure 4.2. Photograph of experimental setup

directions by laser sensors. Ceramic gauge blocks (with surface roughness up to 0.05 μm) are used in places of sensor's applying to prevent measurement error during 2-axis movement of the workpiece. The workpiece has a spherical surface for ability of applying force with angle from 0 to 90°, which direction would correspond to angle of vibration and imitate the axial force of drilling.

4.2. Controlling of vibration

Results of controlled angle vibration (15°, 45°, 75°) of the workpiece with frequency 149.5 Hz, 198.0 Hz, 252.3 Hz are shown in Appendix B. It shows track lines of the workpiece before loading force and after loading force with values 3 N, 7 N and 10 N. The vertical and horizontal spring systems of the FS2DV are adjusted by rigidity for each vibration frequency (k_{ver} = 785 kN/m, k_{hor} = 835 kN/m for vibration frequency 149.5 Hz; k_{ver} = 1076 kN/m, k_{hor} = 1027 kN/m for 198.0 Hz; k_{ver} = 1547 kN/m, k_{hor} = 1620 kN/m for 252.3 Hz) to allow properly vibration during loading force.

For evaluating of vibration's performance under force loading, it is proposed tree parameters: shifting and deviation of the vibration trace, trace of the workpiece. Figure 4.3 shows measurement example of these parameters.

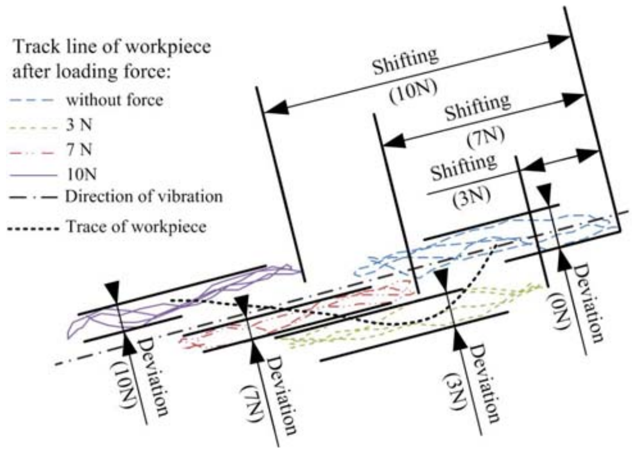
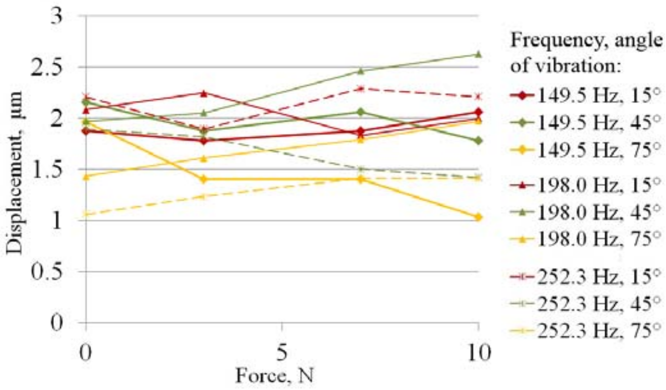
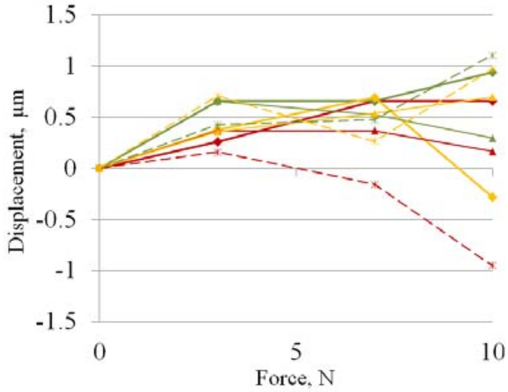


Figure 4.3 Example of evaluating succession of vibration

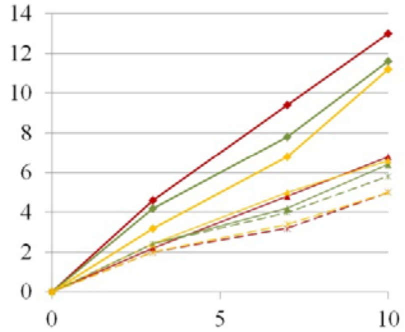
Figure 4.4 shows analysis of the experimental results with controlled vibrations. The deviation of the workpiece is gradually increasing with loading force but by small value 0.2…1.1 μm depending on vibration frequency or vibration direction. The deviation of the vibration trace is stable in range from 1.4 to 2.6 μm depending on vibration frequency or vibration direction. Also, the shifting of the workpiece is gradually increasing with loading force from 5 μm to 14 μm depending on vibration frequency or vibration direction



(a) deviation of the vibration



(b) deviation of the workpiece



(c) shifting of the workpiece

Figure 4.4 Analysis of experimental results

The deviation of the vibration and the deviation of the workpiece will create the diameter hole error because of it shows the displacement of the workpiece in direction, which is normally to line of the drilling. The deviation of the vibration is a consequence of a phase difference between the horizontal and vertical vibrations (our system has a passive controlling and can’t change the phase of the vibration). The deviation of the workpiece is a consequence of spring system’s using but it value is small (up 1 µm). The total hole diameter error can be 2.1…3.7 µm (depending on vibration frequency or direction of the vibration), but it is not big compering with 6 µm (a standard tolerance of the hole with diameter 0.2 mm).

The shifting of the workpiece almost will not influence on the diameter hole error in case of through type of the hole, because of it shows displacement of the workpiece along line of drilling. And in case of blind type of the hole, the shifting of the workpiece will influence of hole’s depth and may initiate error up to 12 µm. But this error can be taken in a account and minimized by decreasing of drilling depth. But our purposed holes are thought type.

Chapter 5

Deviation of thrust force during vibration process

5.1. Experimental setup

Figure 5.1 shows a scheme of the experimental setup for measuring thrust force. Force is applied to the workpiece through screwed guideway with direction of the vibration, by small turning of the guideway. The values of measured force (by dynamometer Kistler type 9601A31) and workpiece displacement (by laser sensors Keyence LK-G80 and LK-G30) are observed by the oscilloscope (Tektronix 2004). . Ceramic gauge are used in places of sensor's applying. The workpiece has a spherical surface for ability of applying force with angle from 0 to 90°, which direction would correspond to angle of vibration and imitate the axial force of drilling. Figure 5.2 shows the picture of the experimental setup. The dynamometer is fastened between the workpiece and the FS2DV.

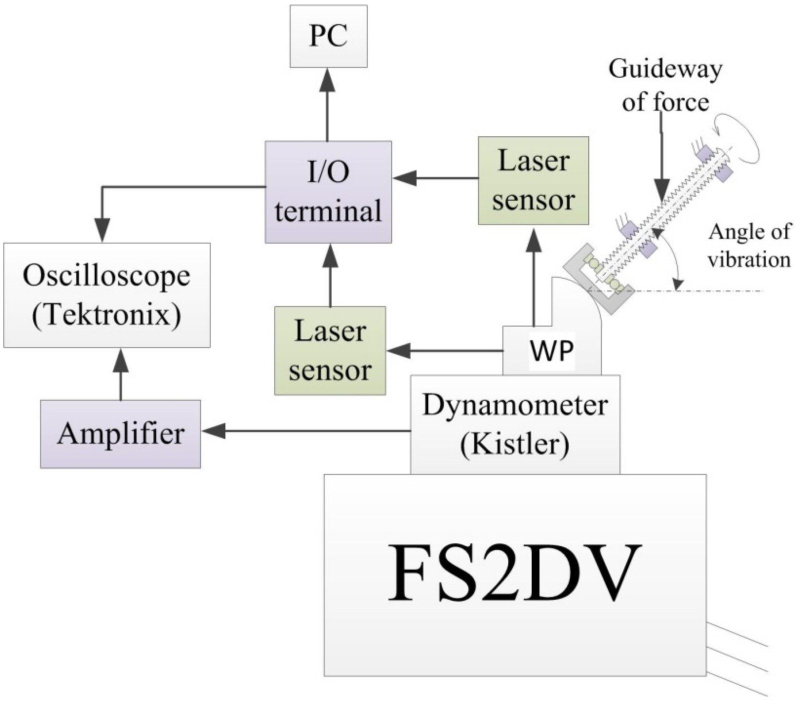


Figure 5.1 Scheme of the experimental setup

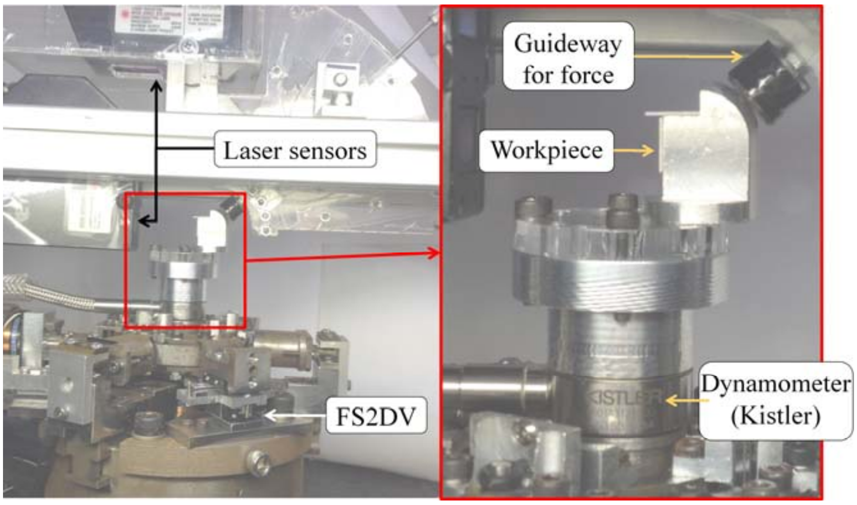


Figure 5.2 Picture of the experimental setup

5.2. Observing of the thrust force deviation

Figure 5.3 shows measuring of the force and workpiece's displacement during vibration process with frequency 198 Hz, amplitude 10 mkm and direction 45 degrees.. Measuring is dividing on three stages: without vibration and loading force; 2nd stage is with vibration, but without loading of force; and 3rd stage is graduate loading with steps of the thrust force during vibration process. The thrust force is applied to the workpiece by small stepped turning of the guideway and the workpiece start displacing in the vertical and horizontal directions after each step of force loading.

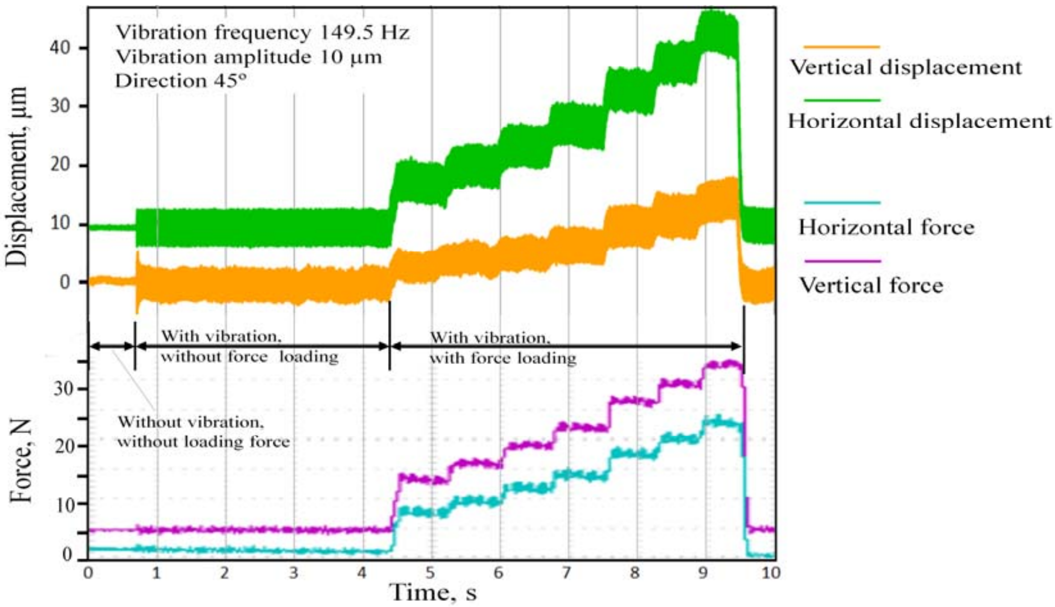


Figure 5.3 Measuring of force and workpiece's displacement
during the vibration process

Figures 5.4, 5.5 and 5.6 show narrow behavior of force measuring and workpiece's displacing for each stage (without force loading and without vibration process, without force loading and with vibration process, with vibration process and with force loading).

During measuring of force and workpiece's displacing without force loading and without vibration process, it is seen the oscillating of the force with frequency 50 Hz and amplitude 0.25N. Because it is no loading of force and no vibration process, it means this is a noise at the dynamometer.

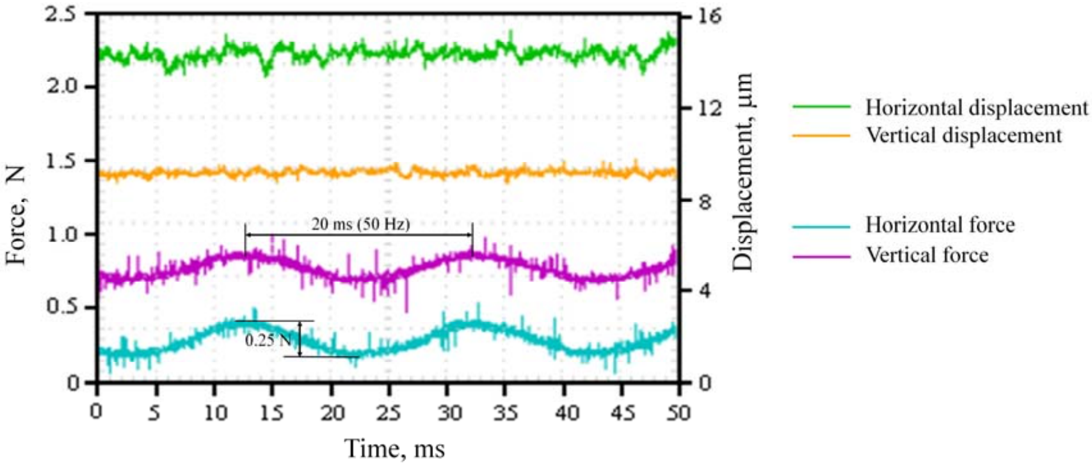


Figure 5.4 Measuring without force loading and without vibration process

During measuring of force and workpiece's displacing without force loading and with vibration process, it is seen oscillation of measured force with amplitude 1.0 N and with frequency of wotkpiece's vibration, and by frequency about 1200 Hz with amplitude 0.4 N

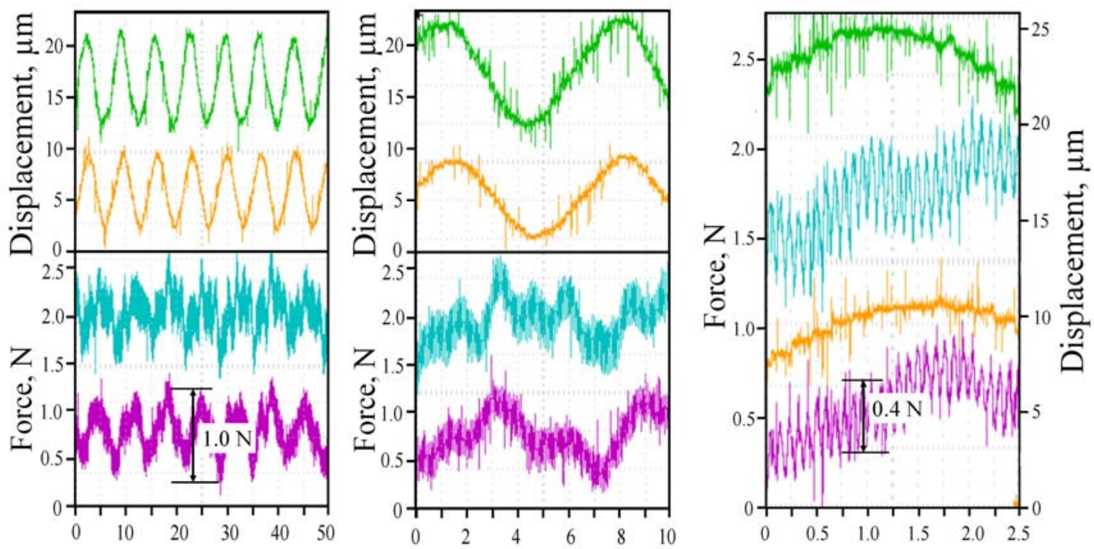


Figure 5.5 Measuring without force loading and with vibration process

During measuring of force and workpiece's displacing with force loading and with vibration process, it is seen that the deviation of thrust force is present. It is by frequency of vibration with amplitude 1.3 N, and by frequency about 1200 Hz with amplitude 0.4 N.

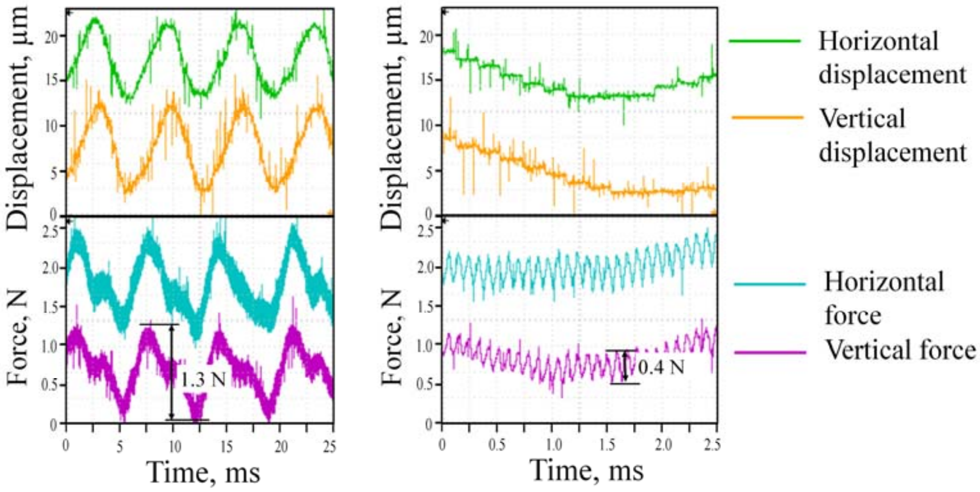


Figure 5.6 Measuring with force loading and with vibration process

So the thrust force measuring is made during vibration process with frequency 198.0 Hz, amplitude 10 µm and direction 45º. The thrust force deviation is present during the vibration process. It is up to 1.3 N with the frequency of workpiec‘s vibration. The thrust force is not big relevantly to the value of thrust force of the drilling at the study of Dr. Nanbu (9 N). The deviation of the thrust force with amplitude 0.4 N with frequency 1200 Hz may be induced by stepped driven signal on the actuators. But frequency of the signal stepping is far from the resonance mode of the spring systems that is why it is not influence on behavior of the workpiece's displacing.

Conclusion

- To cope micro deep drilling assisted by low-frequency frequency of curved surface, concept of controlling direction vibration of the workpiece have been proposed. Regulation of the direction is made by changing of two independent vibrations in vertical and horizontal axis.
- Piezo electrical actuator was replaced for voice coil actuator to supply enough power for oscillating of large part at new fixing system.
- The fixing system for 2-dimensional vibration (FS2DV) is designed. To achieve this purpose the FS2DV is equipped by vibration sources (voice coil actuators) in horizontal and vertical directions. For success using of the sources, the FS2DV has spring systems along action of the vibration sources.
- Amplitude-frequency analysis of vertical and horizontal spring systems was made for conception of balanced rigidities of spring systems. Requirements of selecting rigidities of the spring systems are determined for defined frequency vibration of the workpiece.
- The FS2DV is verified during loading force 3 N, 7 N and 10 N. This experiment was made for vibration with frequencies 149.5 Hz, 198.0 Hz, 252.3 Hz and angles 15°, 45°, 75° for each frequency. Experimental results show that the FS2DV properly oscillates the workpiece during loading force with steady amplitude and direction of the vibration for range vibration frequency of 150…250 Hz and with any angle of direction from 0 to 90°. Caused by balanced rigidities of spring systems, the workpiece is being displaced along line of drilling with small deviation of vibration trace (up to 1.5 μm) that will minimize error of hole shape during drilling process.
- As result, we have developed FS2DV with usage range frequency of 150…250 Hz, amplitude 10 μm. The system can be properly used with cutting force (up to 10 N) and any angle (from 0 to 90°) by selecting appropriate rigidities for vertical and horizontal spring systems.

Nomenclature

d_h	diameter of the drilling hole
L_h	longitude of the drilling hole
F_s	Feed speed of drilling
F_z	feed per tooth of drilling
f_d	speed of rotation spindle
f_v	vibration frequency
A	vibration amplitude
R_c	cutting time ratio
T_c	cutting time
T_v	vibration period
R_v	positive velocity ratio
T_p	positive velocity time
F_o	amplitude of oscillated force
R_1, R_2	resistance of potentiometers
r_1, r_2	resistance of voice coil actuators
I -	current of controlled signal
I_1, I_2	current of signal at voice coil actuators
k_{ver}, k_{hor}	rigidities of vertical and horizontal spring systems
b, h, L	width, thickness and length offspring plate
N_p	number of plates
E	module of elasticity
F_{ABC}	rebalanced force
γ	angle of vibration at horizontal spring system
β	angle of rebalanced force
F_o	forces at sets with belleville springs
x_d	displacement at horizontal spring system
FS2DV	fixing system for 2-dimenetional vibration
VCA	voice coil actuator

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12. Den Hartog J.P., Mechanical vibrations, Dover Publications Inc., New York, 1985, pp. 23-47

Appendix A. Design of fixing system for 2-dimensional vibration

Fixing system for 2-dimentional vibration

Position	Name	Quantity	Figure
	Fixing system for 2-dimentional vibration		A. 1, A. 2
1	Vertical spring system (assemble)	1	A. 14
2	Horizontal spring system (assemble)	1	A. 20
3	Vertical bearing slider (assemble)	1	A. 25
4	Horizontal bearing slider (assemble)	3	A. 28
5	Horizontal bearing slider – 01 (assemble)	2	A. 28
6	Three-jaw chuck	1	A. 3
7	Voice coil actuator "Akribis AVM-30"	1	Purchased
8	Voice coil actuator "Akribis AVM-35-HF"	1	Purchased
9			
10	Bearing "NSK 983A-MC1-U1-AF2-Q"	6	Purchased
11	Workpiece	1	A. 4a
12	Vertical console	1	A. 5
13	Horizontal console	1	A. 6
14	Horizontal console – 01	2	A. 6
15	Axis	3	A. 4b
16	Bearing plate	6	A. 7a
17	Plug	1	A. 2b
18	Connector	1	A. 8
19	Connector	1	A. 9a
20			
21	Base plate	1	A. 11
22	Corner	3	A. 9b
23	Bracket	2	A. 12
24	Plate	1	A. 13

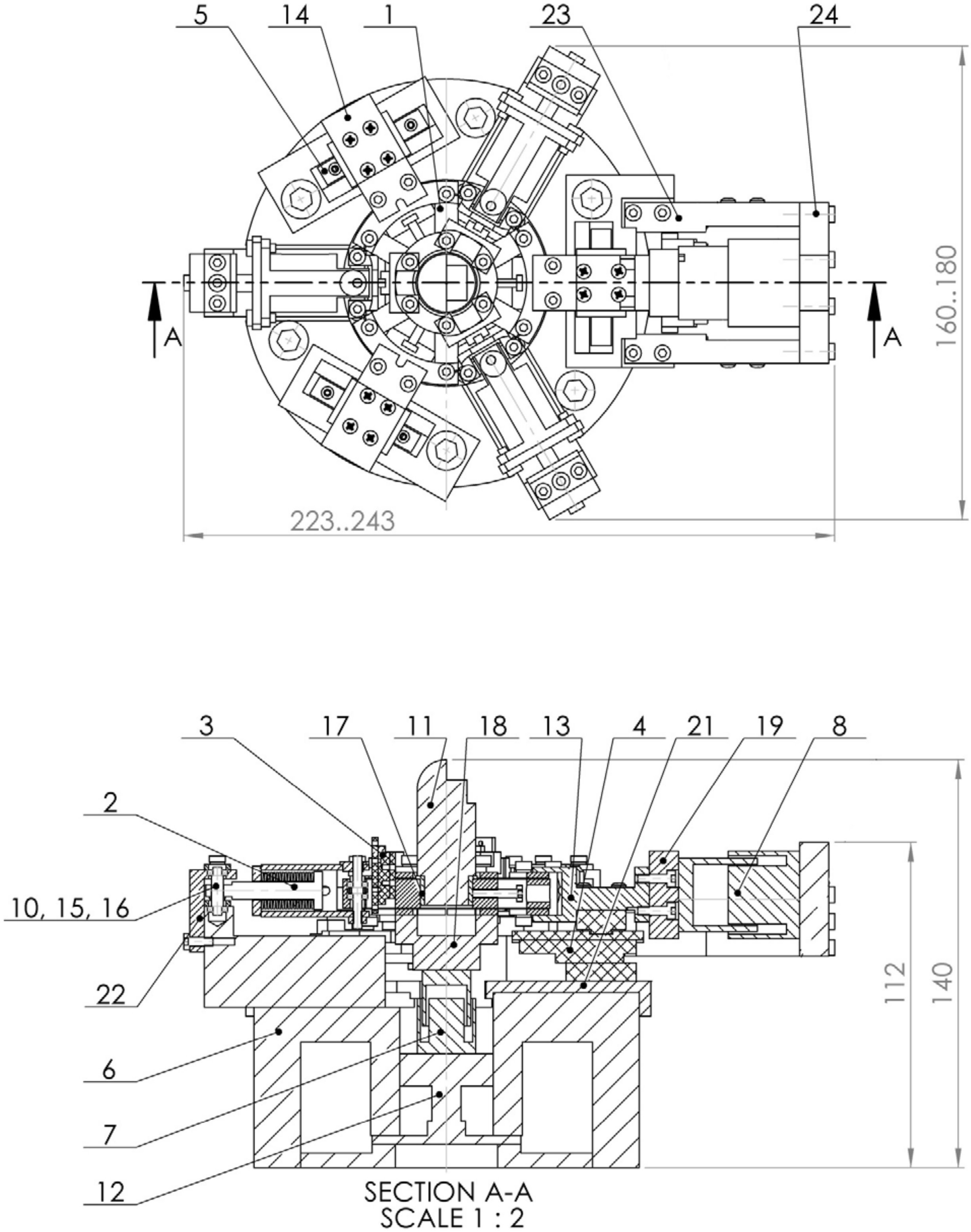


Figure A. 1. Fixing system for 2-dimensional vibration (FS2DV)

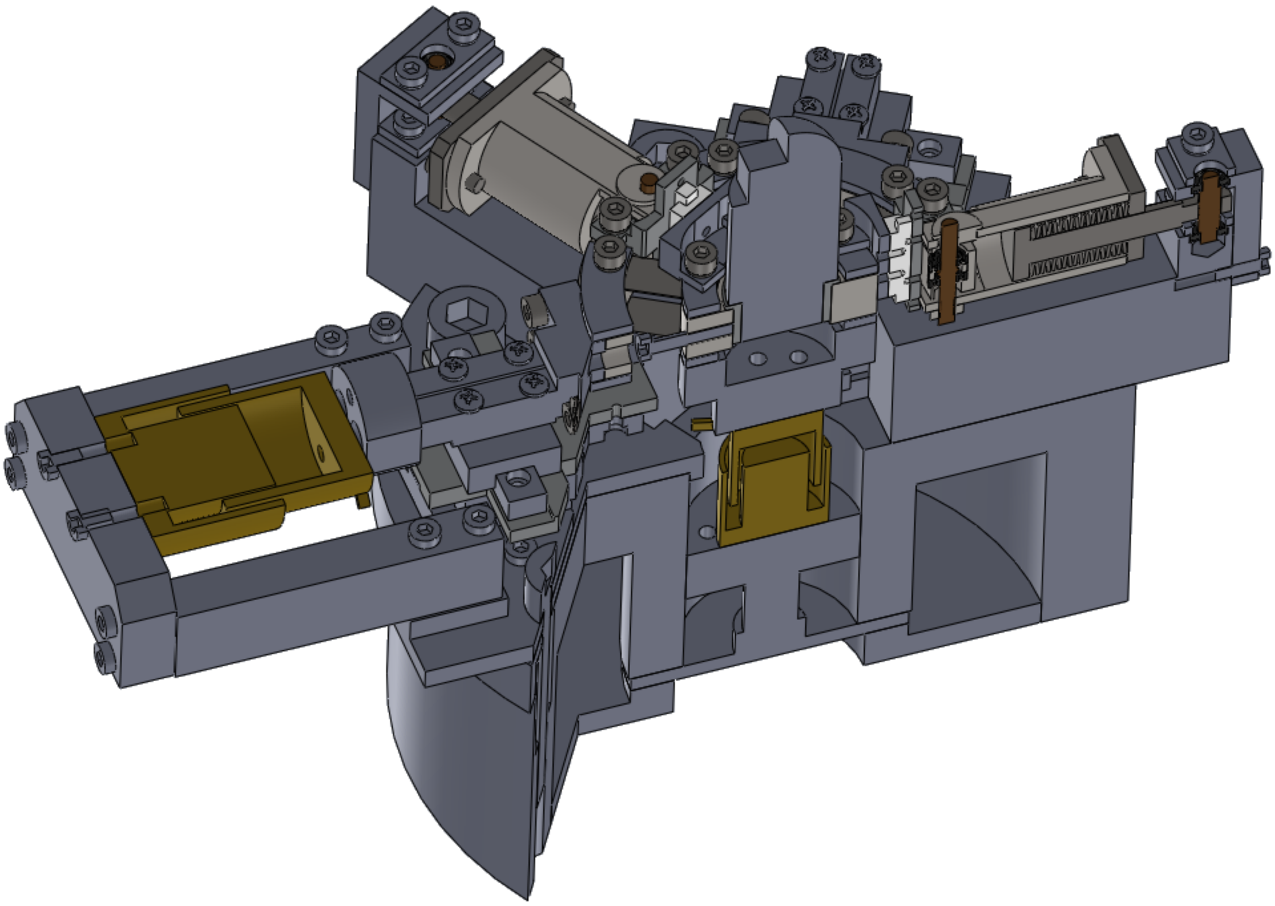
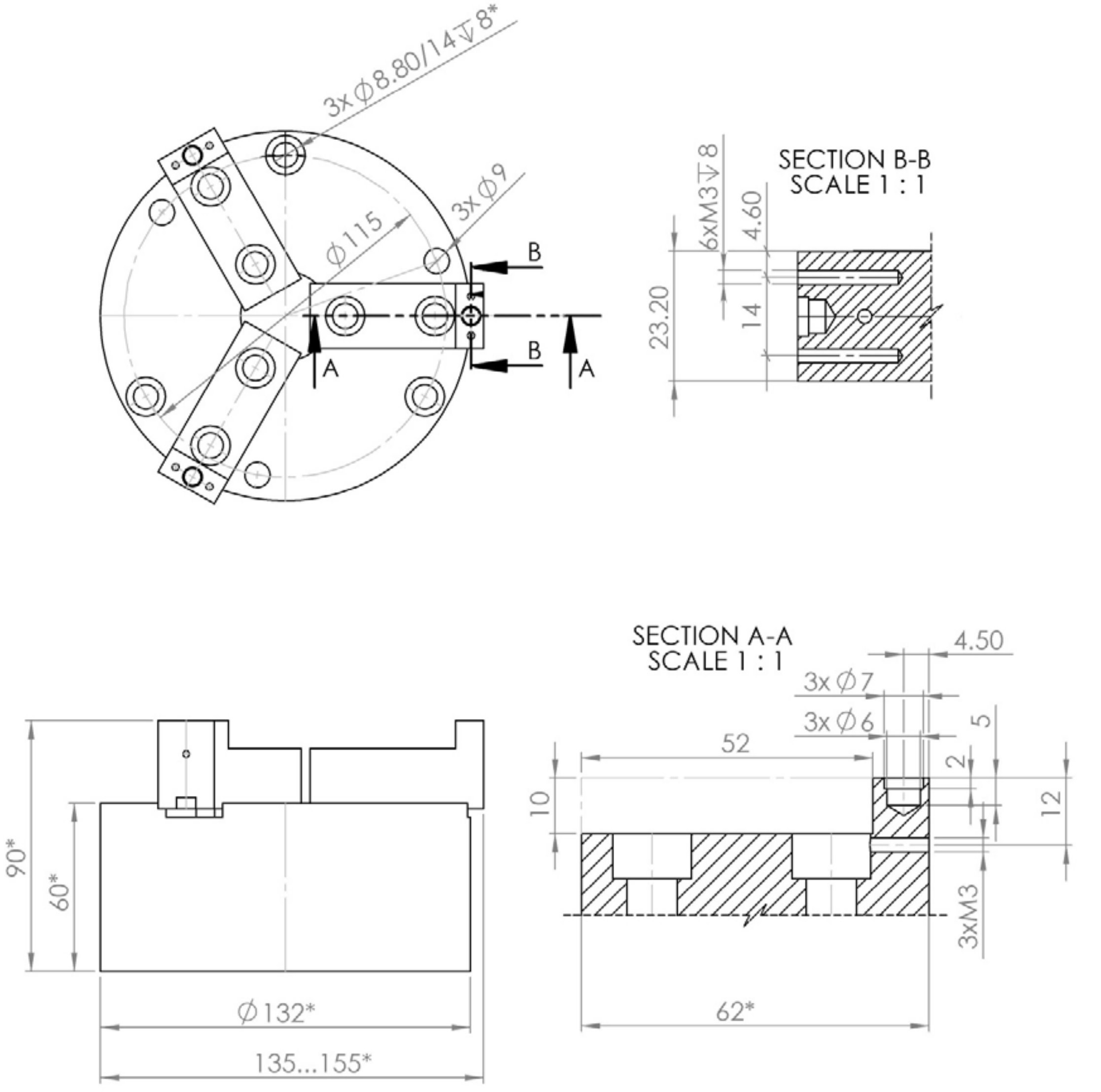
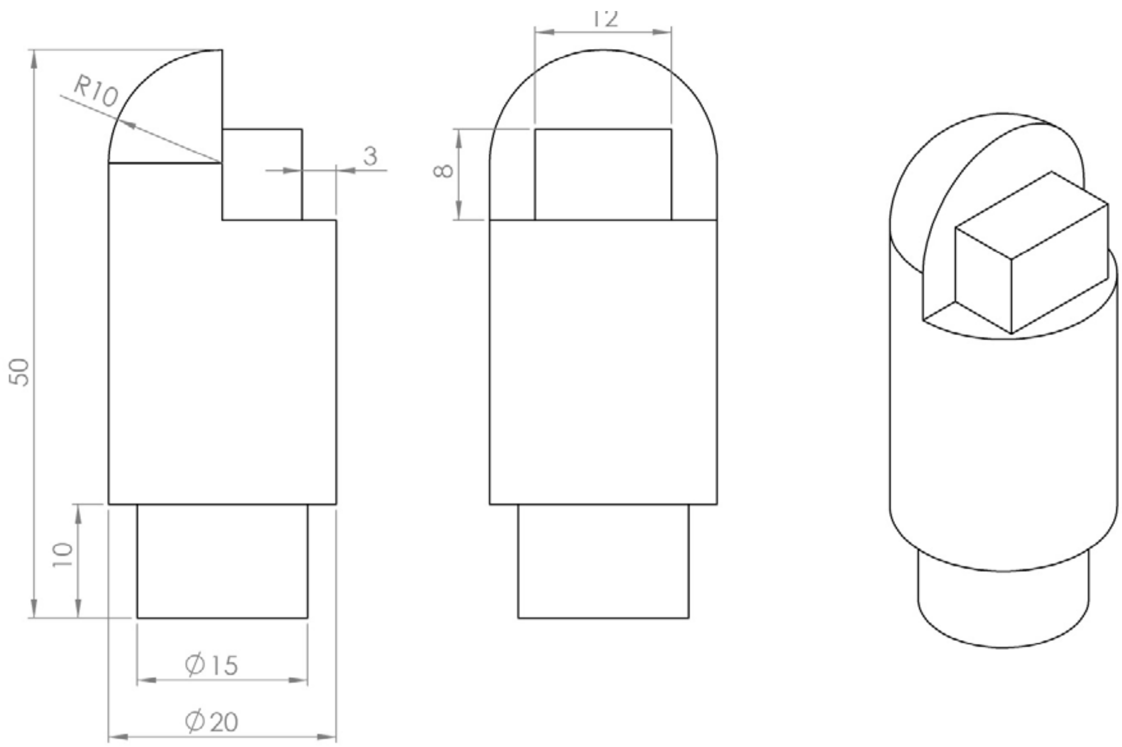


Figure A. 2. FS2DV

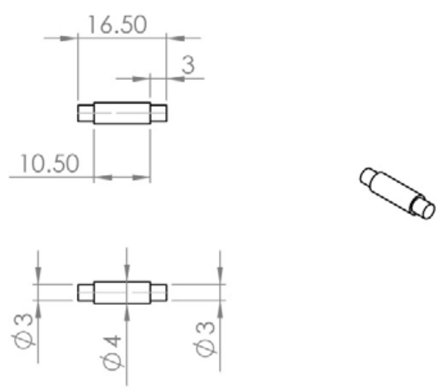


*Sizes for refernce.

Figure A. 3. Three-jaw chuck



(a) Workpiece



(b) Axis

Figure A. 4. Parts of FS2DV

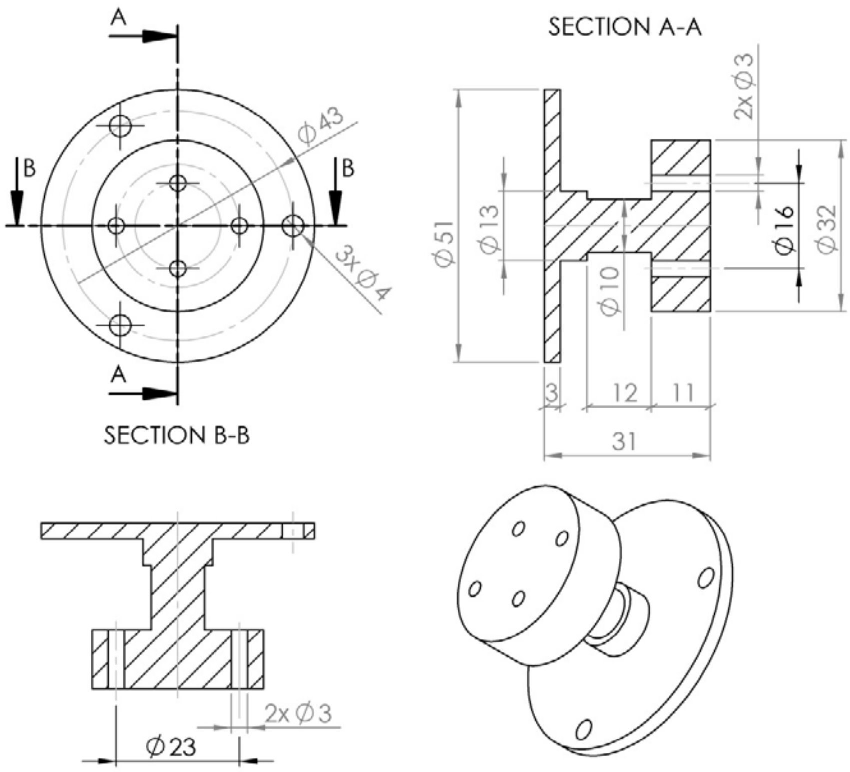
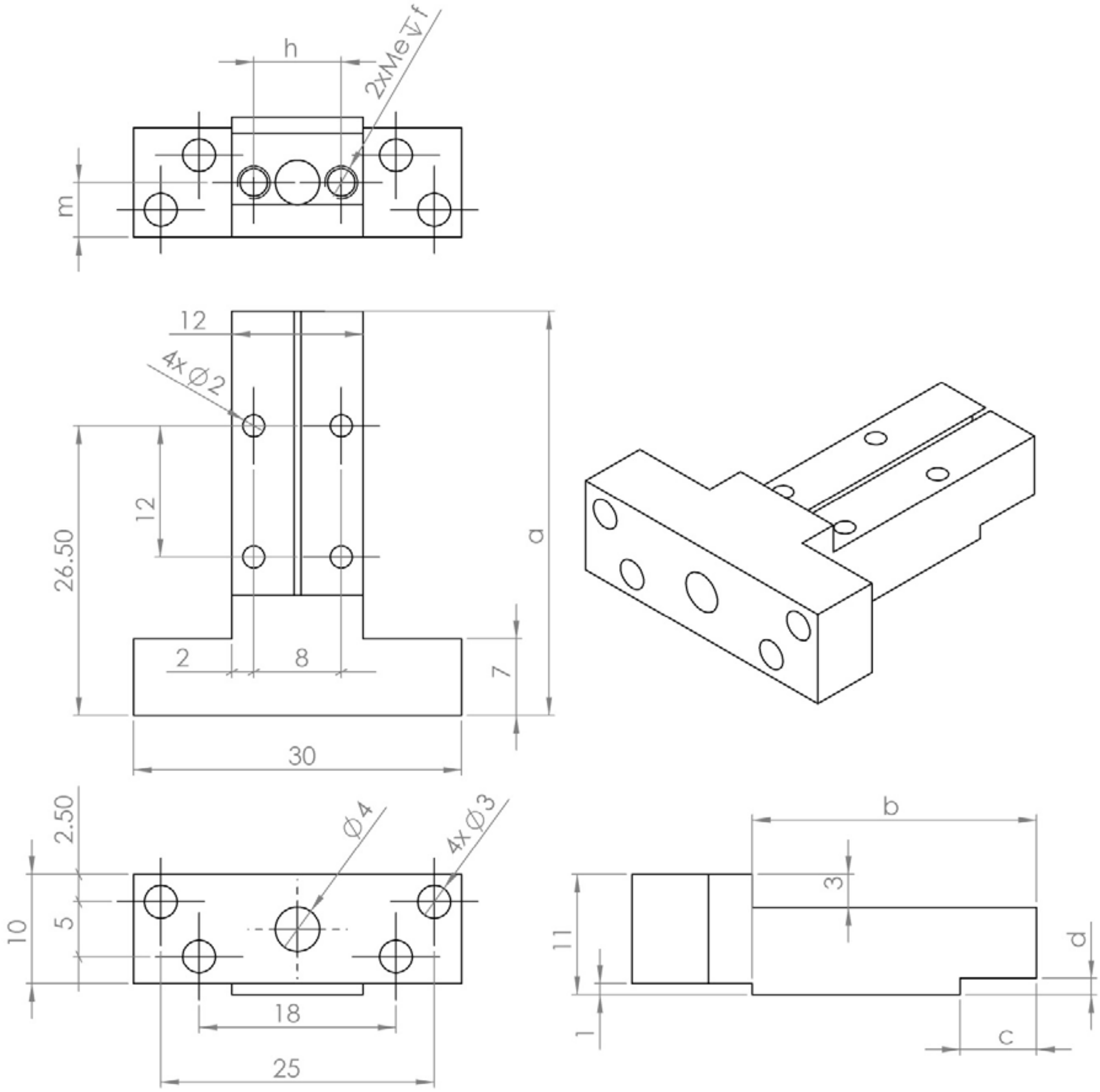
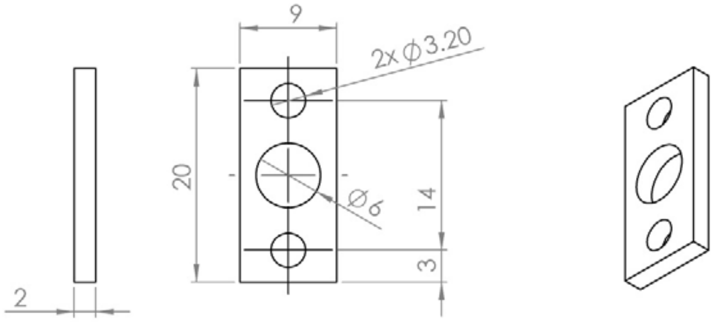


Figure A. 5. Vertical console

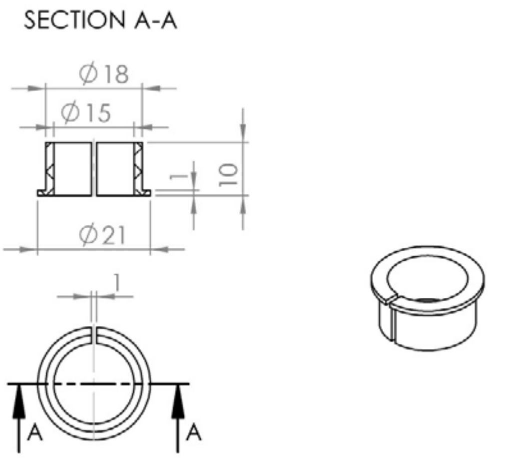


Name	a	b	c	d	e	f	h	m	
Horizontal consol	37	26	7	1.5	M2	7	8	5	
Horizontal consol-01	29	18	-	-	-	-	-	-	

Figure A. 6. Horizontal console



(a) Bearing plate



(b) Plug

Figure A. 7. Part of FS2DV

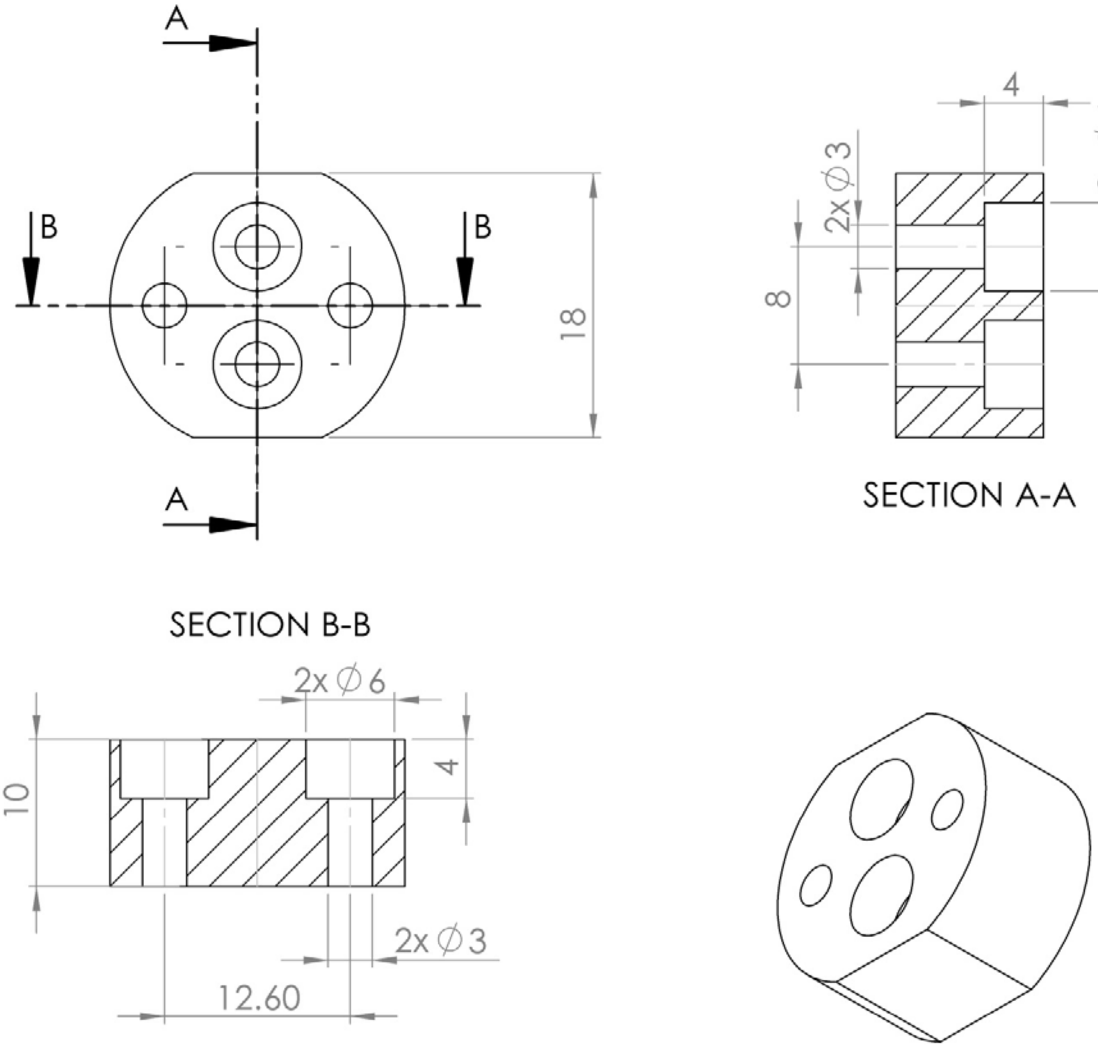
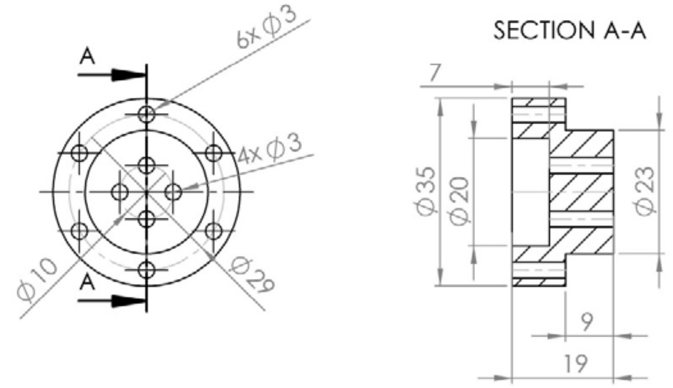
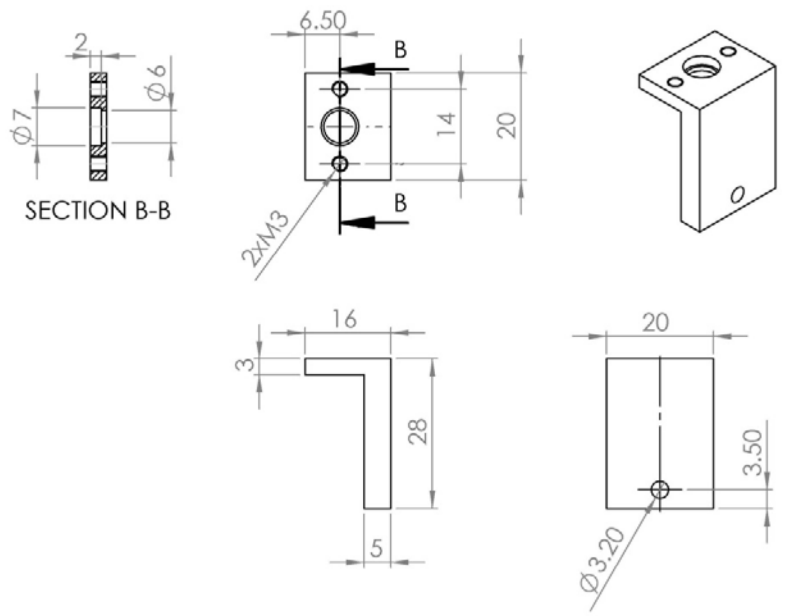


Figure A. 8. Connector



(a) Connector



(b) Corner

Figure A. 9. Parts of FS2DV

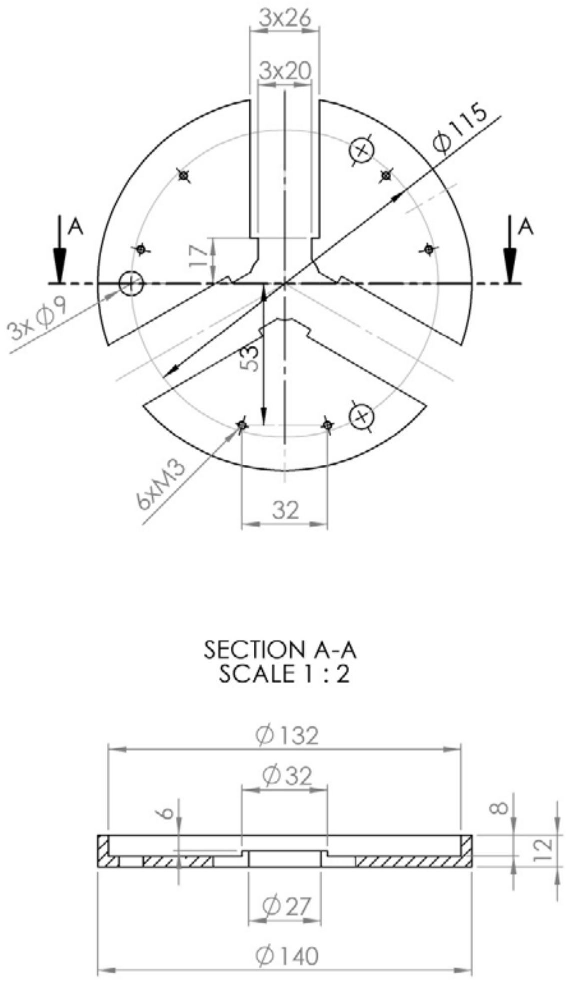


Figure A. 11. Base plate

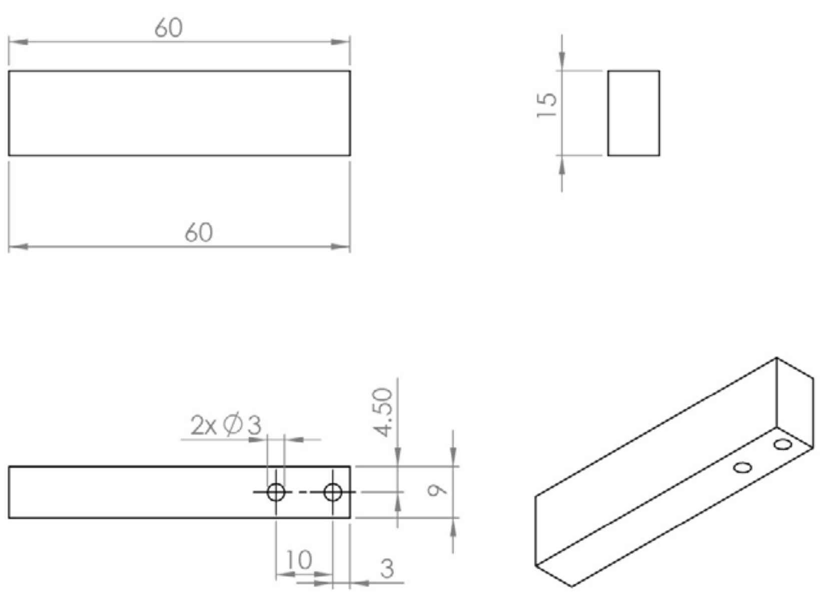


Figure A. 12. Bracket

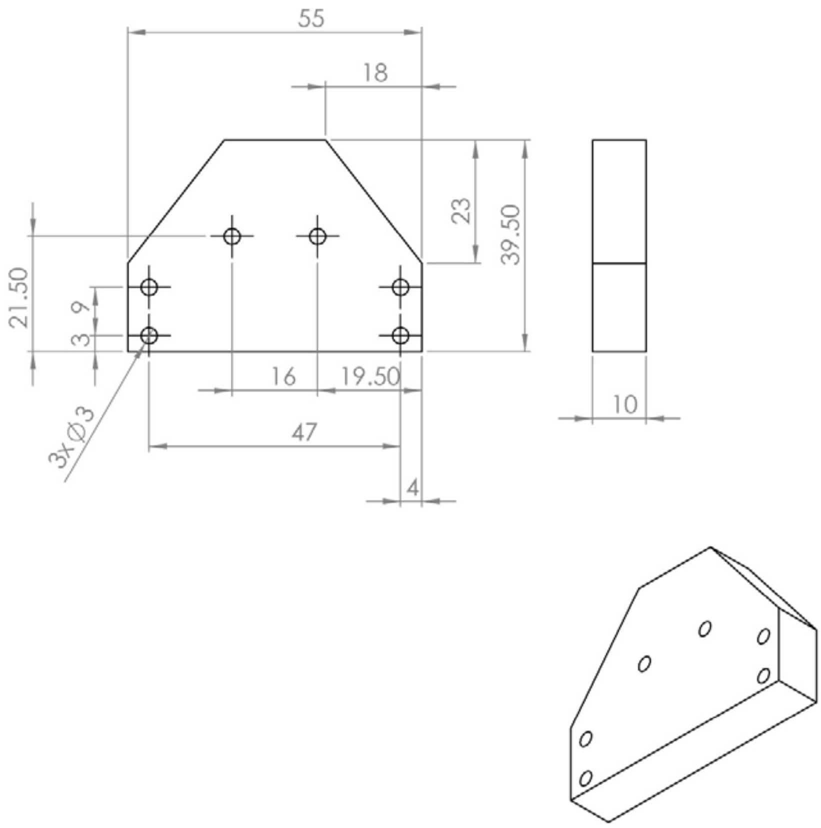


Figure A. 13. Plate

Vertical spring system

Position	Name	Quantity	Figure
	Vertical spring system		A. 14
1	External collar	1	A. 15
2	Internal collar	1	A. 16
3	Cover	6	A. 17
4	Cover	2	A. 18
5	Spring	12	A. 19

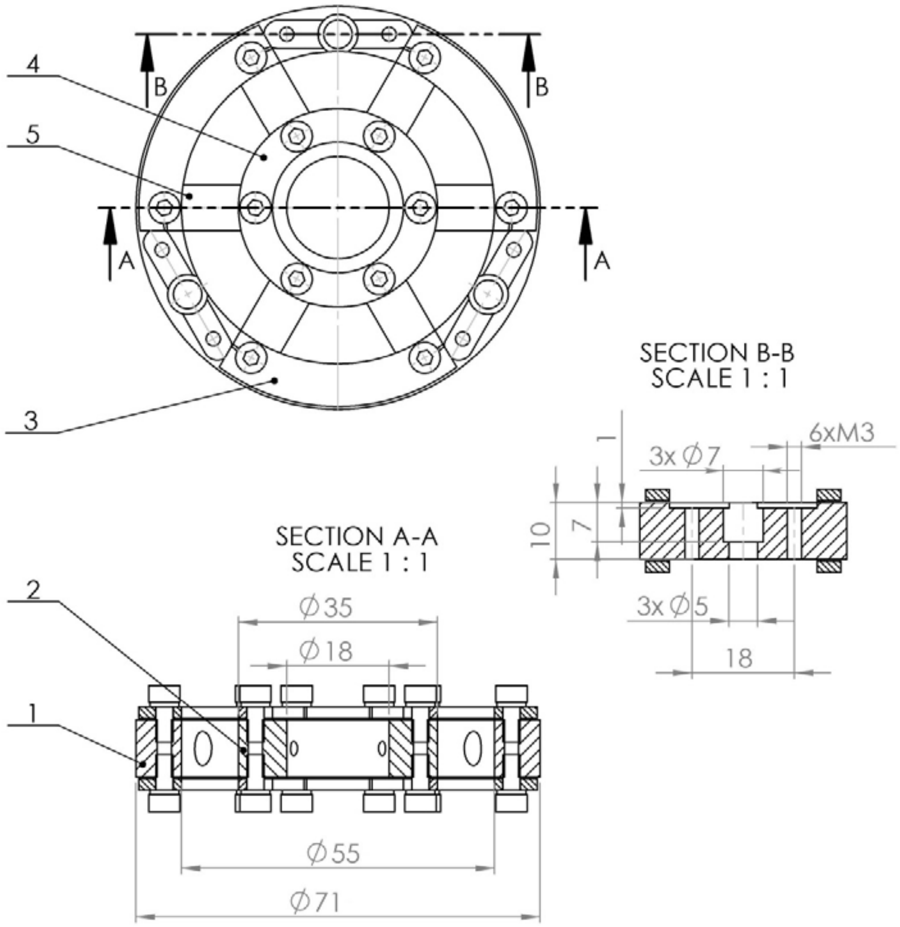


Figure A. 14. Vertical spring system

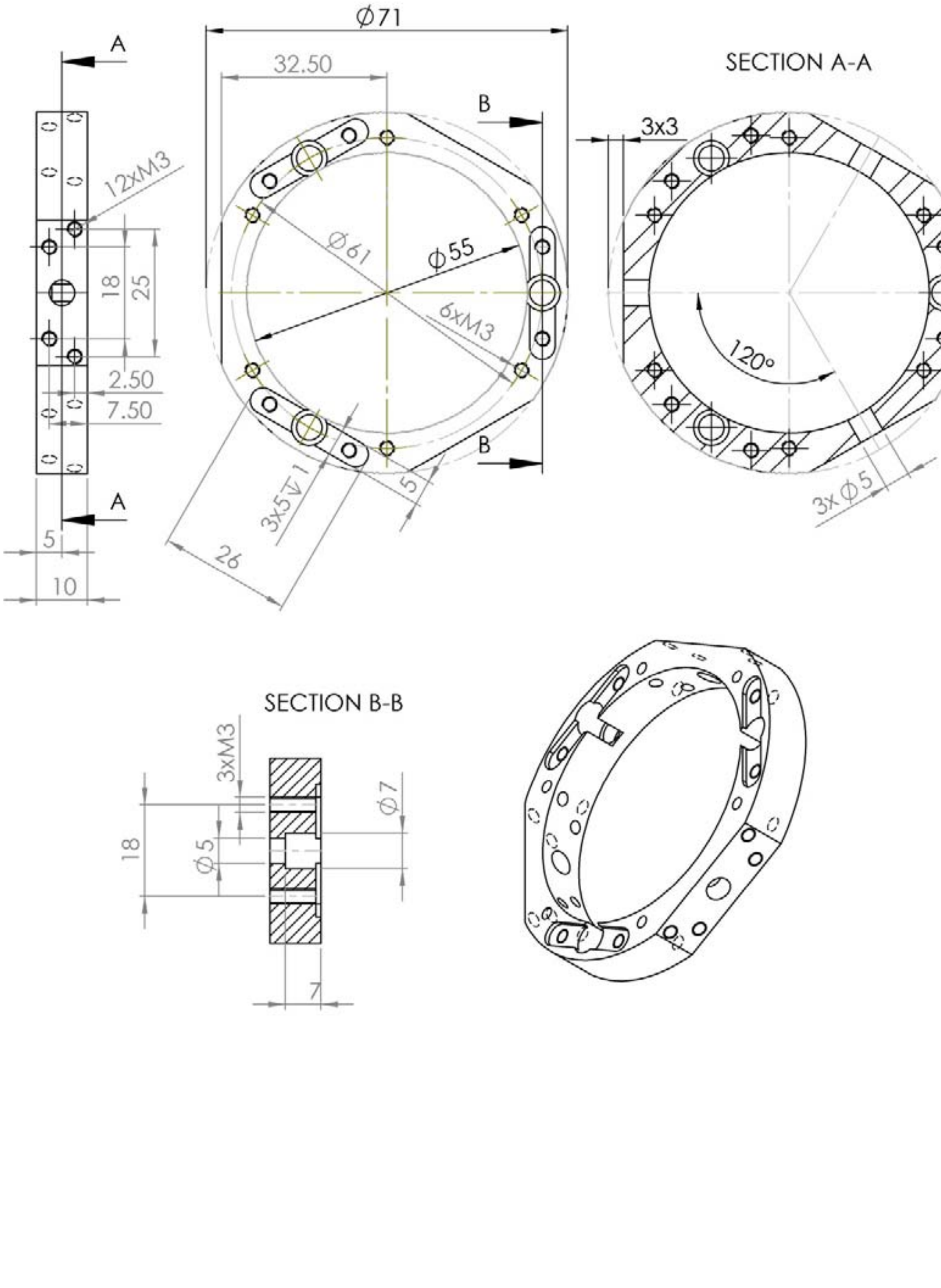


Figure A. 15. External collar

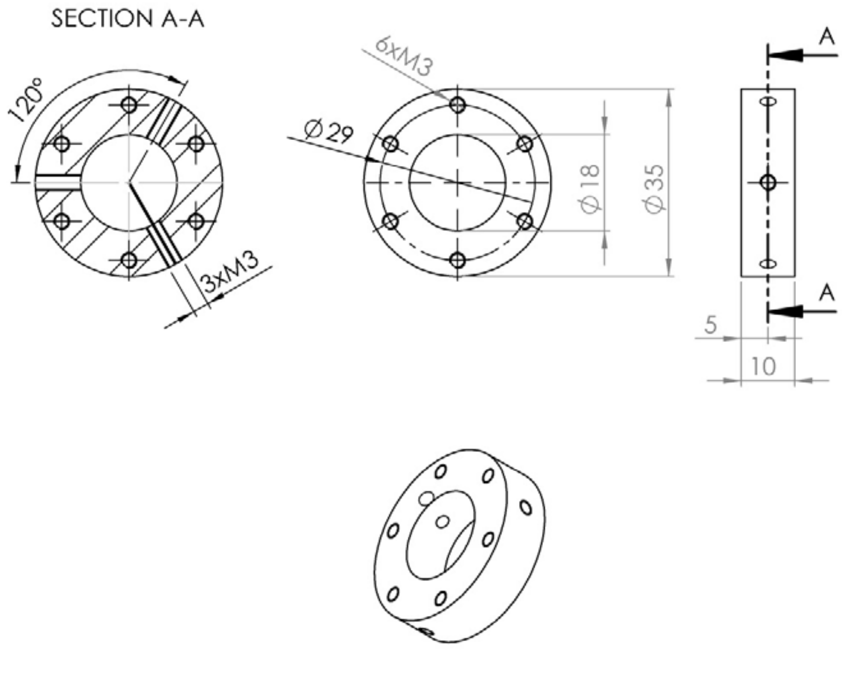


Figure A. 16. Internal collar

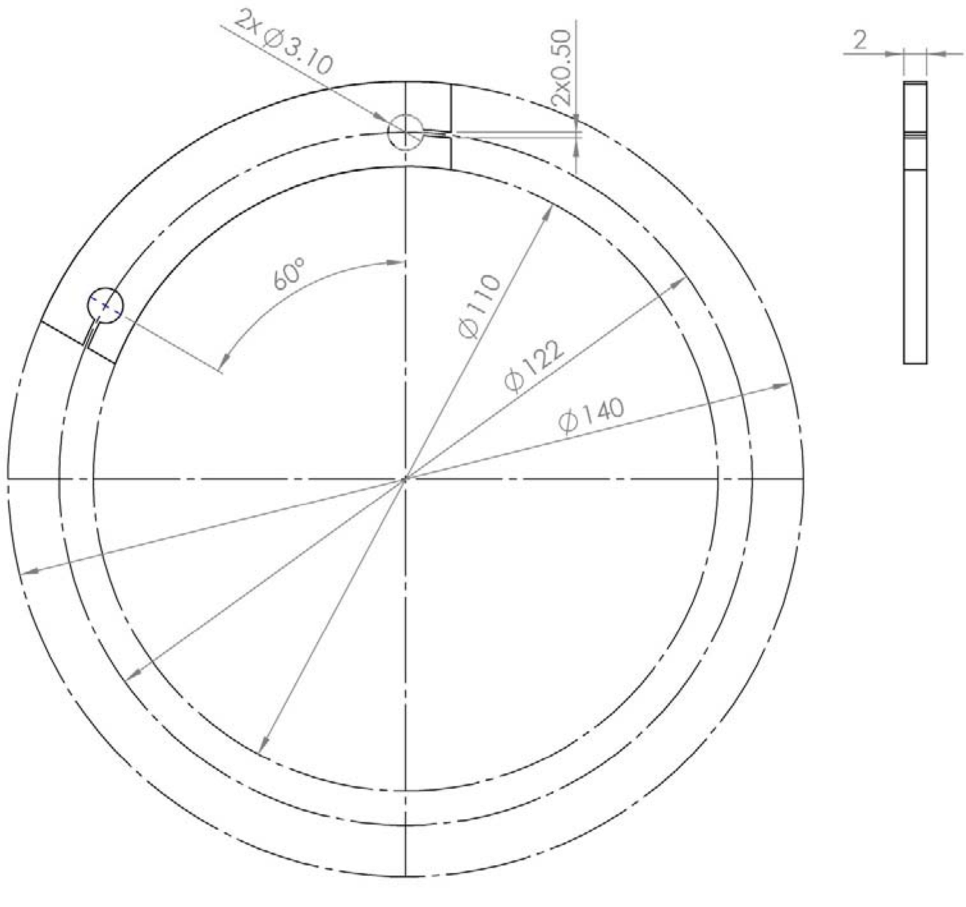


Figure A. 17. Cover

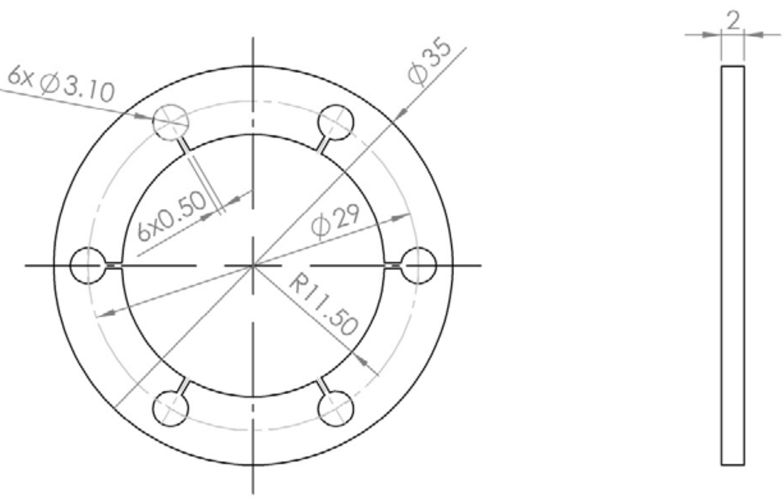
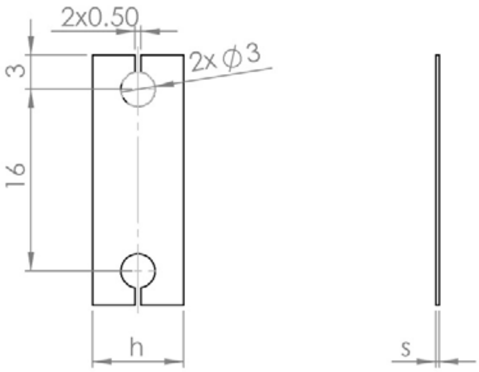


Figure A. 18. Plate



Name	s	h	Material
Spring	0.3	6	Spring Steel SK4
Spring-01	0.6	6	Spring Steel SK5
Spring-02	0.6	8	Spring Steel SK5
Spring-03	0.6	10	Spring Steel SK5

Figure A. 19. Spring plate

Horizontal spring system

Position	Name	Quantity	Figure
	Horizontal spring system		A. 20
1	External collar (from vertical spring system)	1	A.15
2	Bearing “NSK 983A-MC1-U1-AF2-Q”	6	Purchased
3	Belleville spring “Iwata Denko H=12.5 or L=12.5”	18	Purchased
4	Box	3	A. 21
5	Axis	3	A. 22a
6	Plate	3	A. 22b
7	Stopper	3	A. 23a
8	Axis	3	A. 23b
9	Bush	3	A. 24a
10	Cover	6	A. 24b

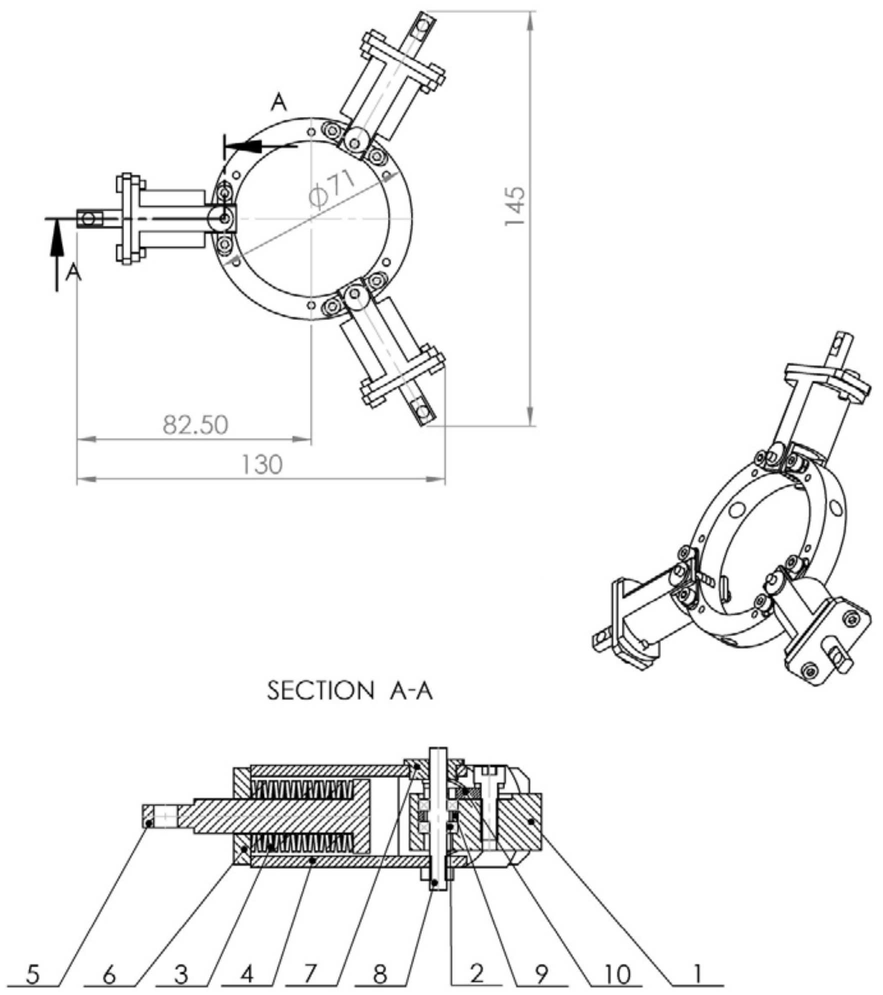


Figure A. 20. Horizontal spring system

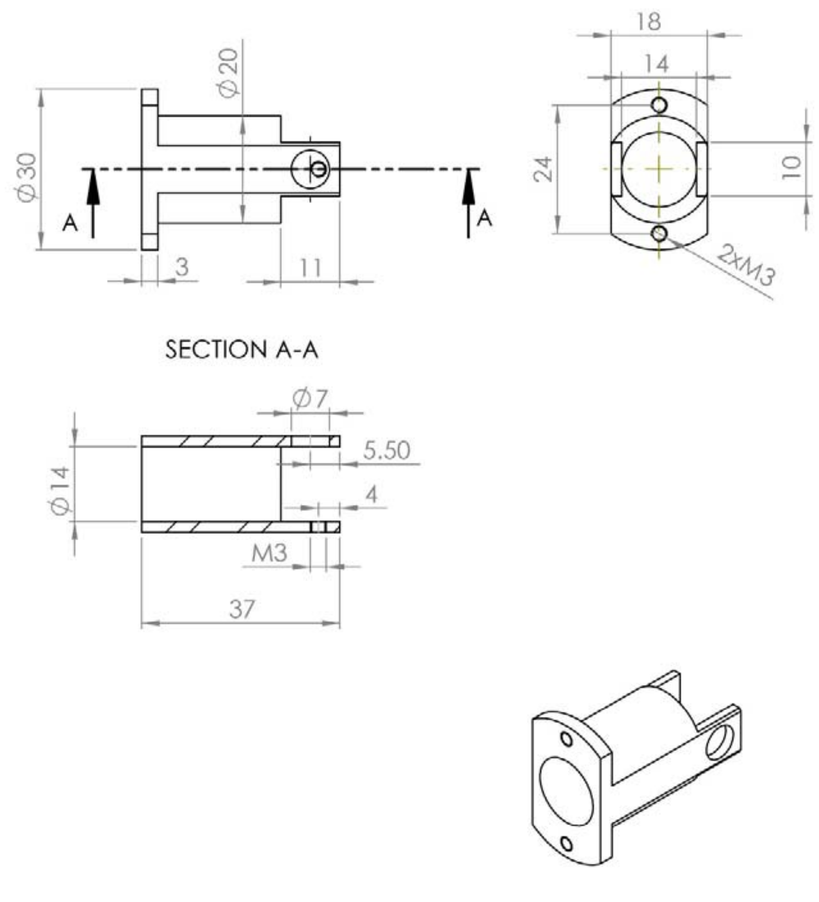
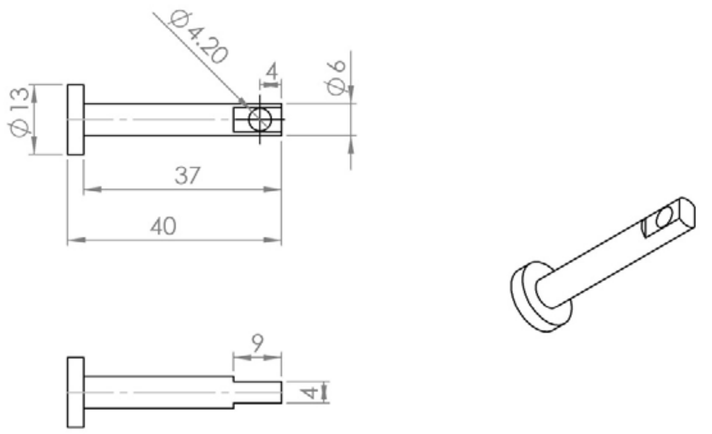
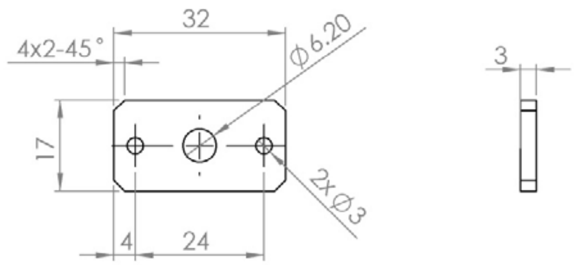


Figure A. 21. Box

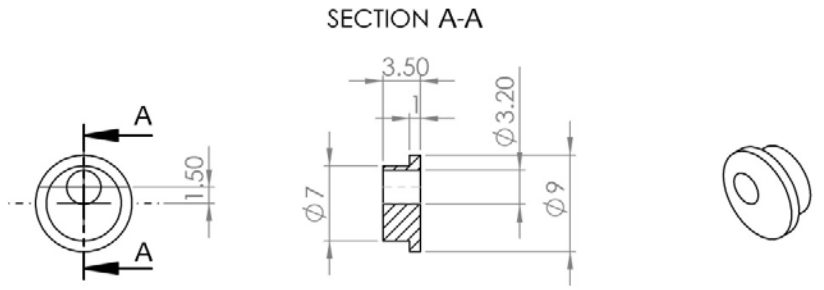


(a) Axis

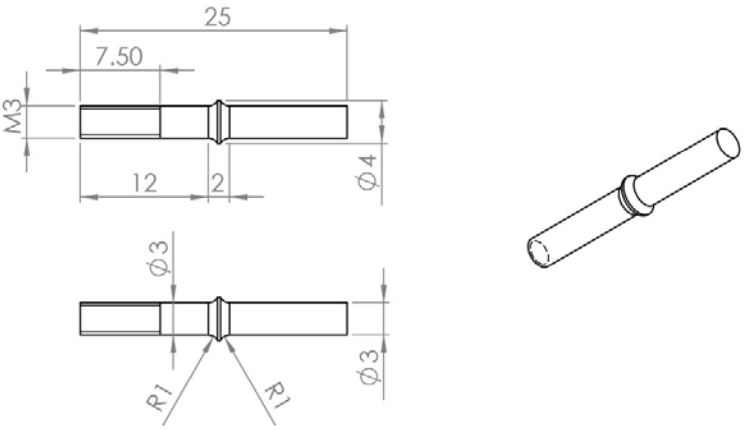


(b) Plate

Figure A. 22. Parts of horizontal spring system

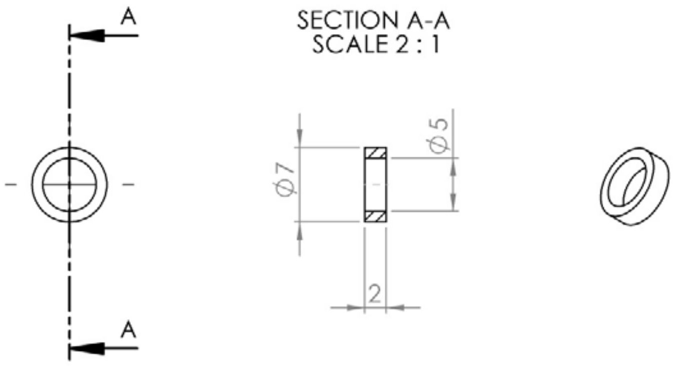


(a) Stopper

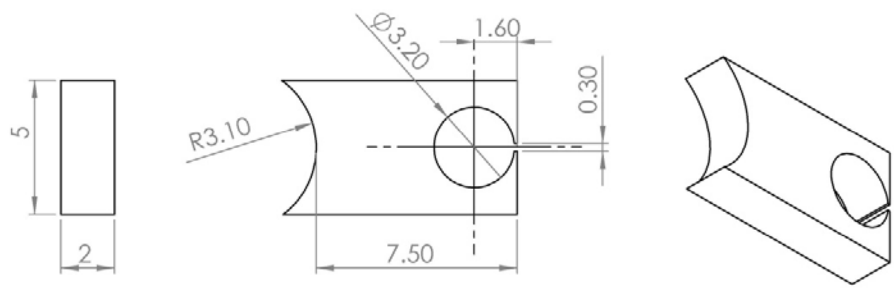


(b) Axis

Figure A. 23. Parts of horizontal spring system



(a) Bush



(b) Cover

Figure A. 24. Parts of horizontal spring system

Vertical bearing slider

Position	Name	Quantity	Figure
	Vertical bearing slider		A. 25
1	Bearing slider “THK RSR3NC1+20LM”	2	Purchased
2	Bracket	1	A. 26
3	Bracket	1	A. 27

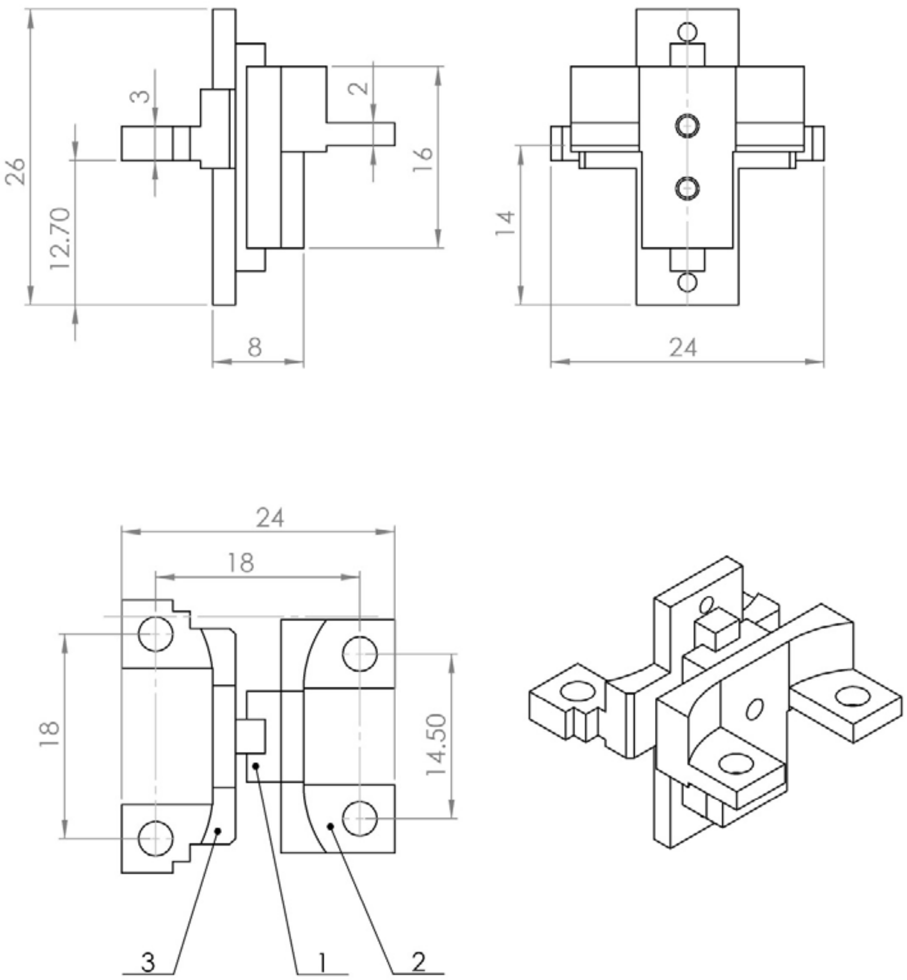


Figure A. 25. Vertical bearing slider

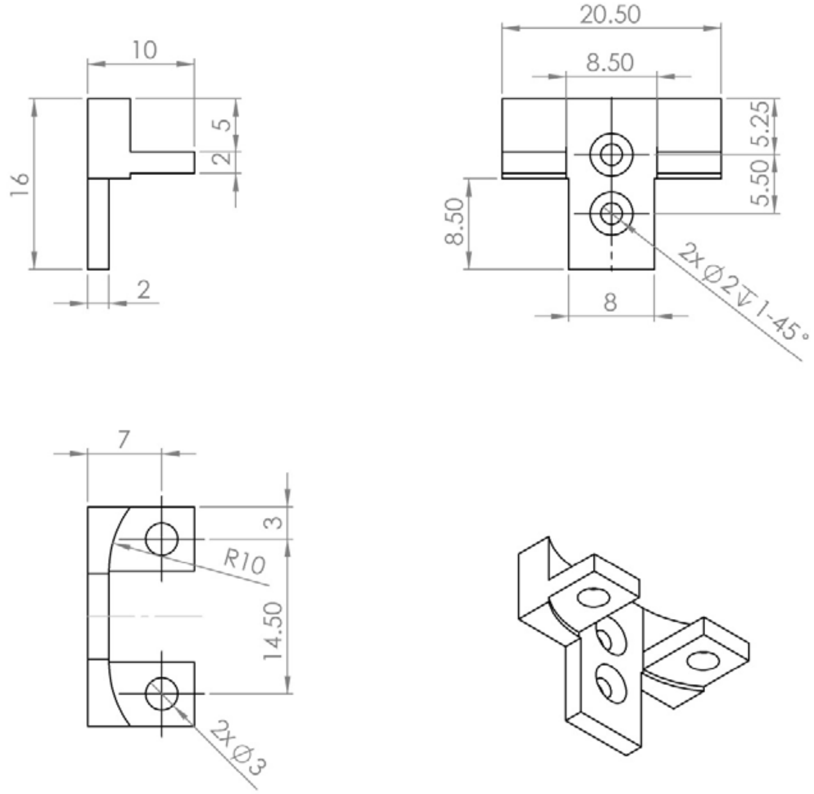


Figure A. 26. Bracket

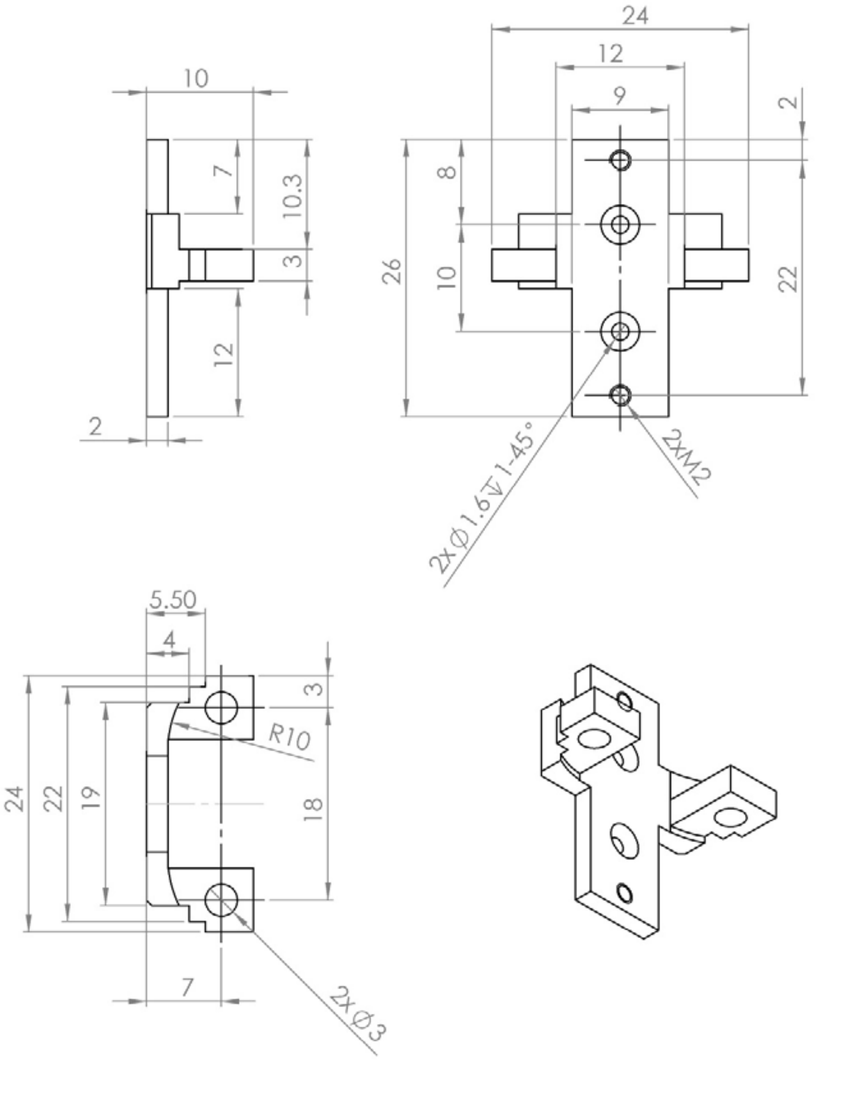
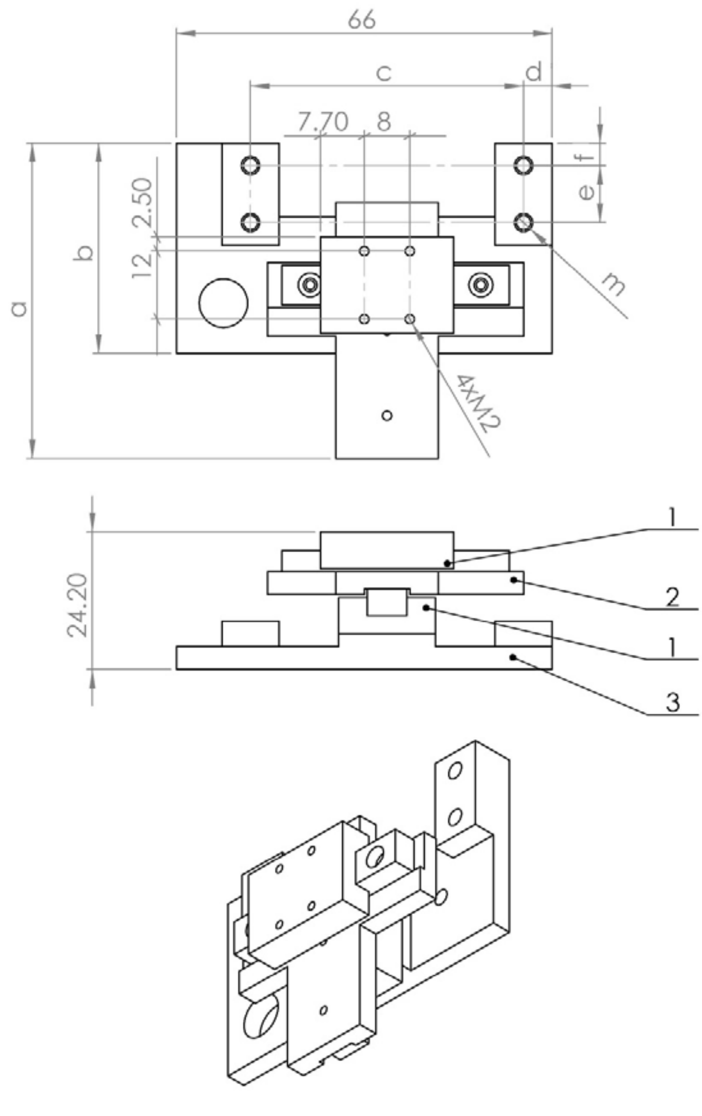


Figure A. 27. Bracket

Horizontal bearing slider

Position	Name	Quantity	Figure
	Horizontal bearing slider		A. 28
1	Bearing slider “THK RSR-7M-UU-G1+40LM”	2	Purchased
2	Bracket	1	A. 29
3	Base	1	A. 30
	Horizontal bearing slider-01		A. 28
1	Bearing slider “THK RSR-7M-UU-G1+40LM”	2	Purchased
2	Bracket	1	A. 29
3	Base-01	1	A. 30



Name	a	b	c	d	e	f	m
Horizontal bearing slider	56	37	48	5	4	10	4xM3
Horizontal bearing slider-01	43	24	-	-	-	-	-

Figure A. 28. Horizontal bearing slider

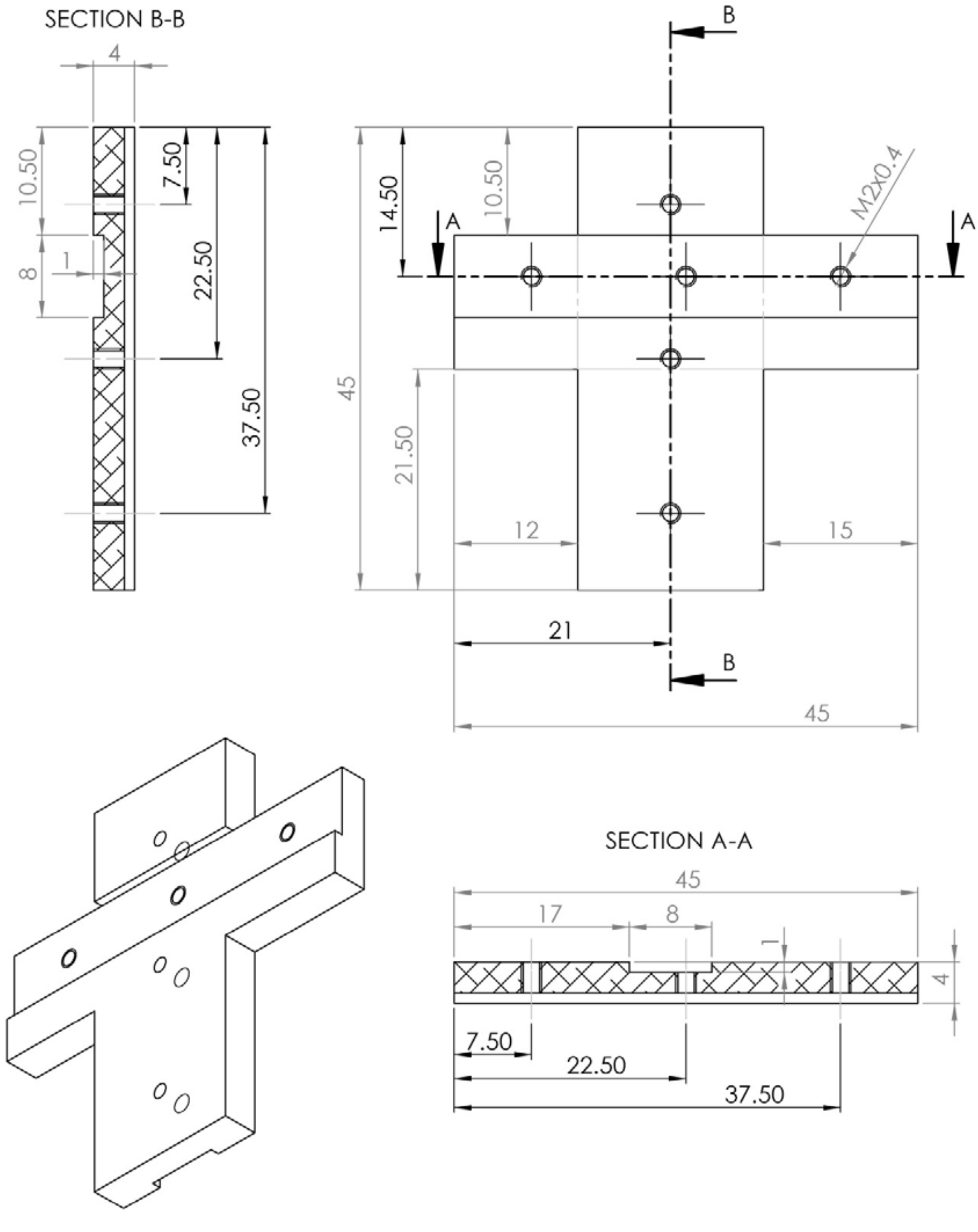
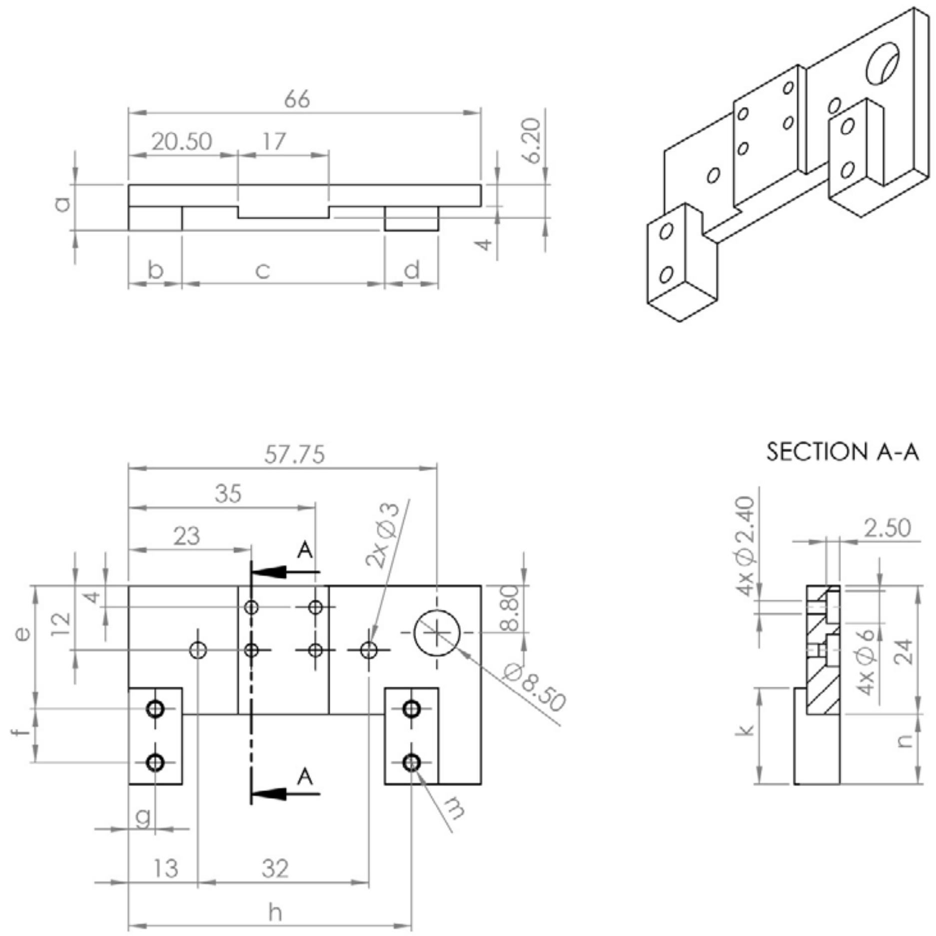


Figure A. 29. Bracket



Name	a	b	c	d	e	f	g	h	k	n	m
Base	8.5	5	38	10	23	10	5	53	18	13	4xM3
Base-01	6.2	0	0	0	-	-	-	-	0	0	-

Figure A. 30. Base

Appendix B. Controlled 2-dimensional vibration

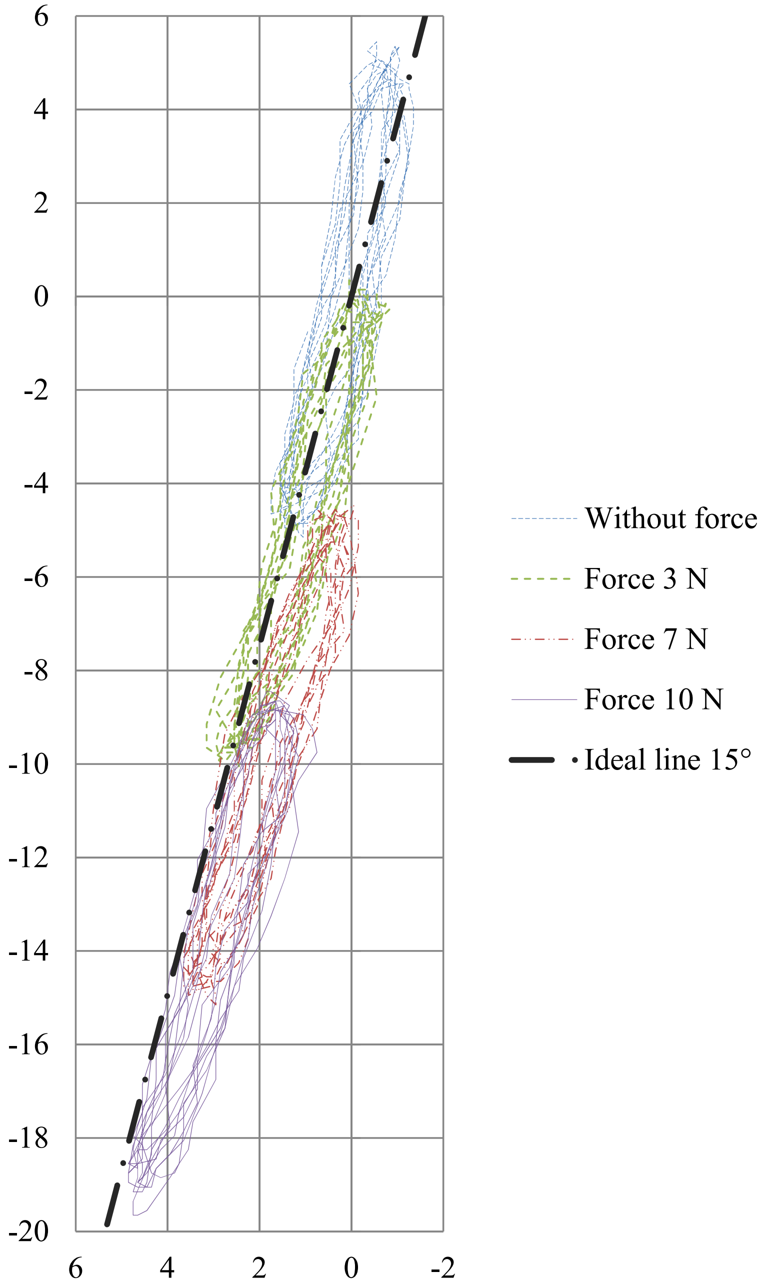
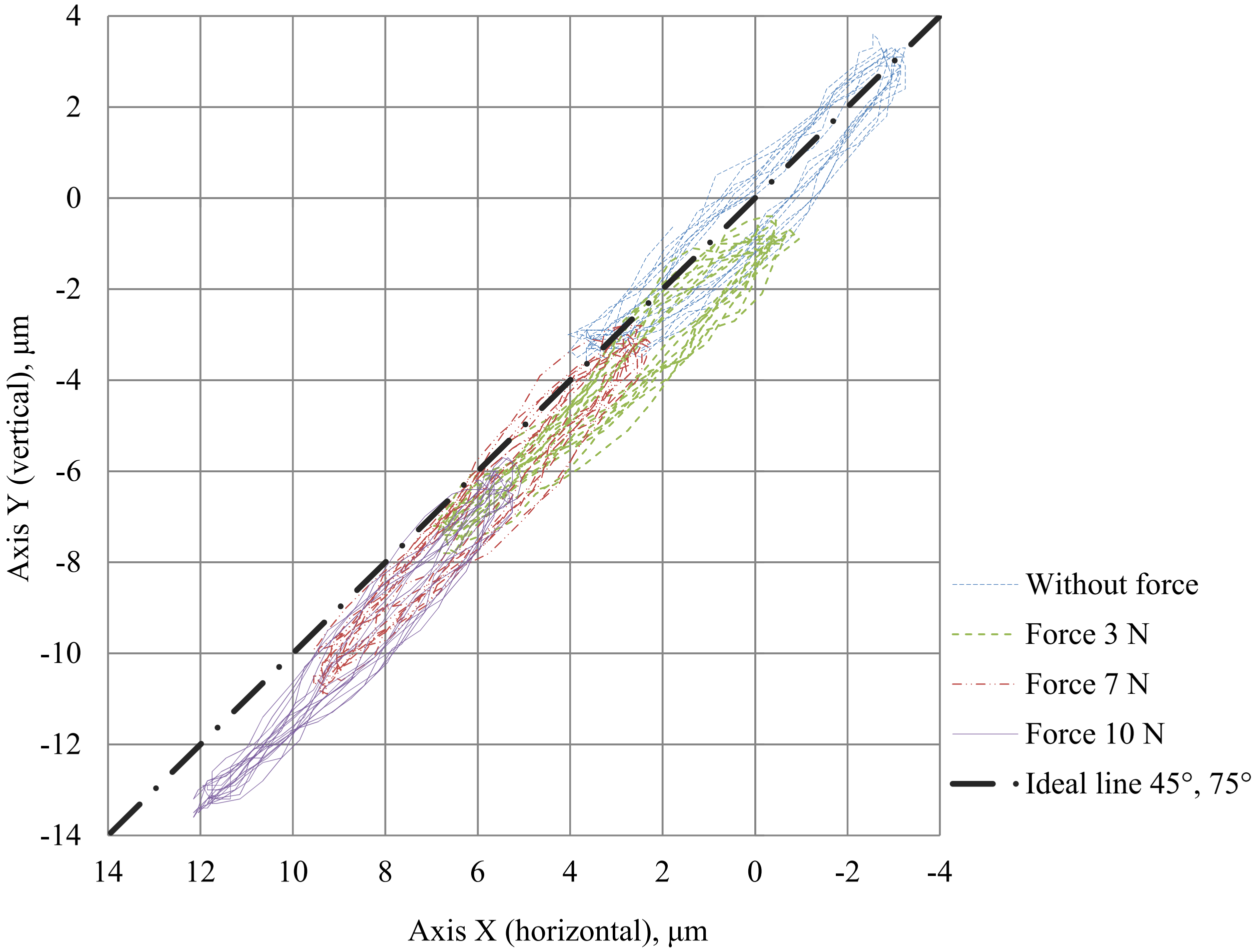
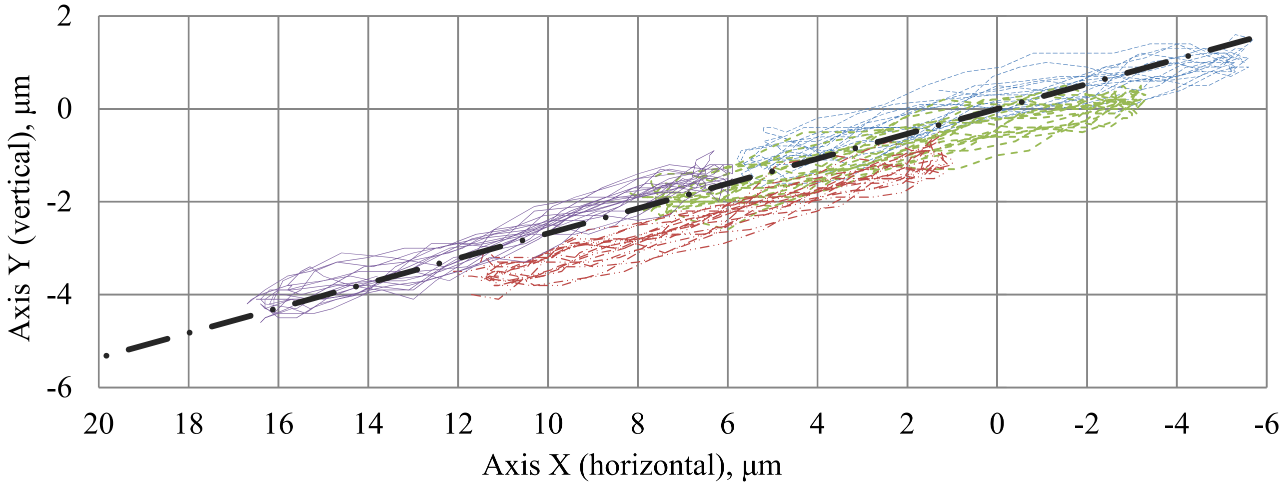


Figure B.1. Controlled vibration with frequency of 149.5 Hz with 15°



(b) with angle 45°



(a) with angle 75°

Figure B.2. Controlled vibration with frequency of 149.5 Hz

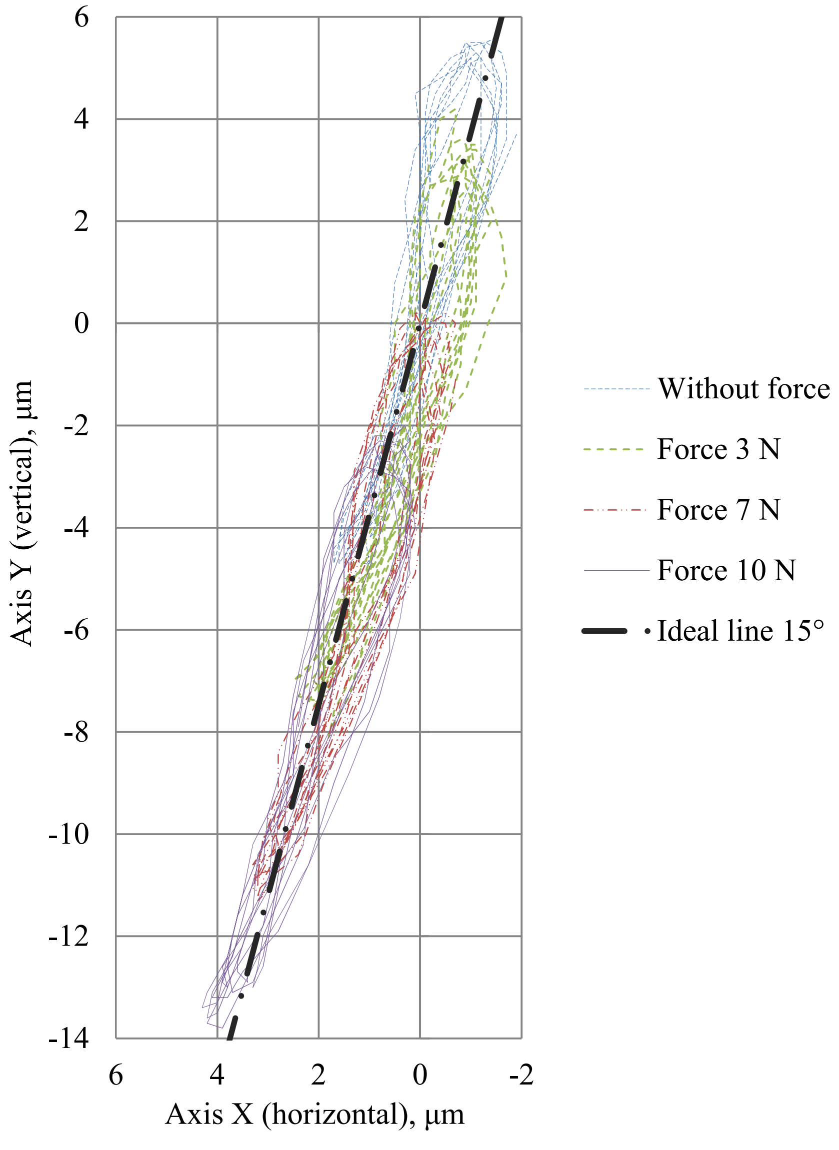
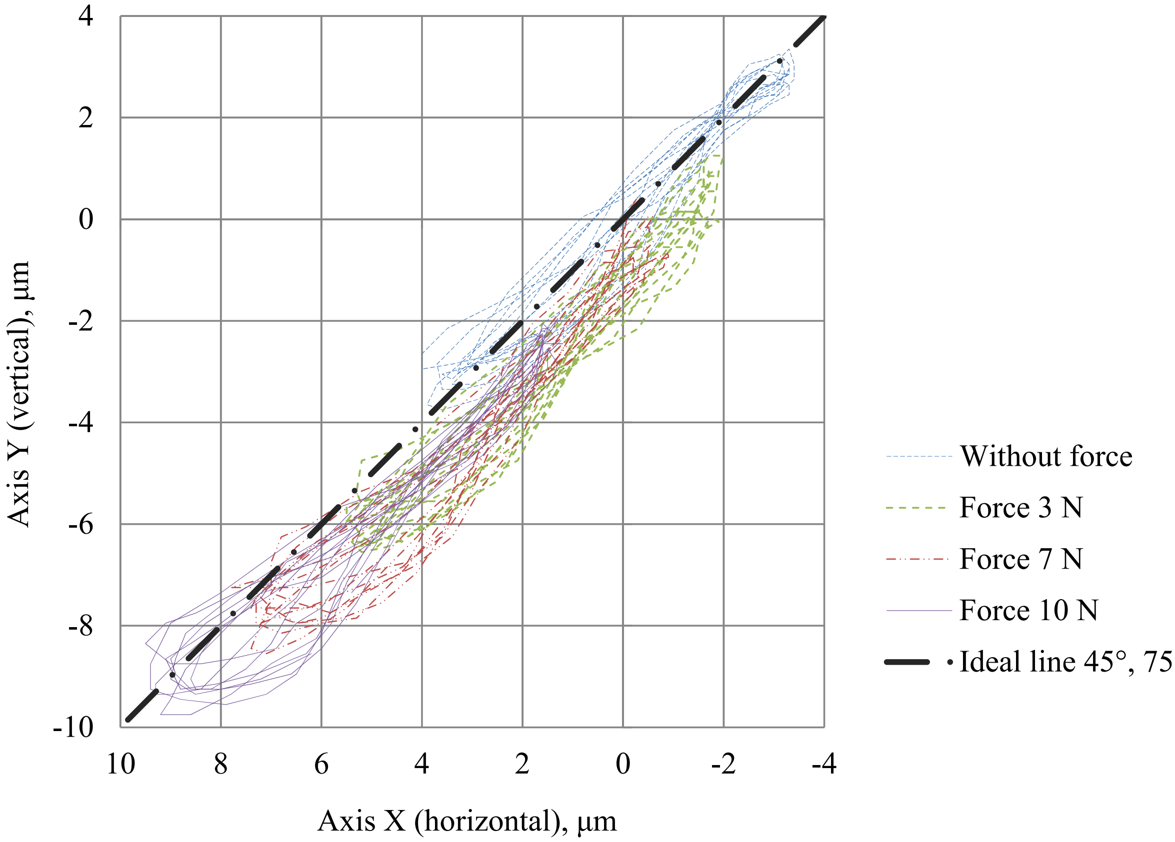
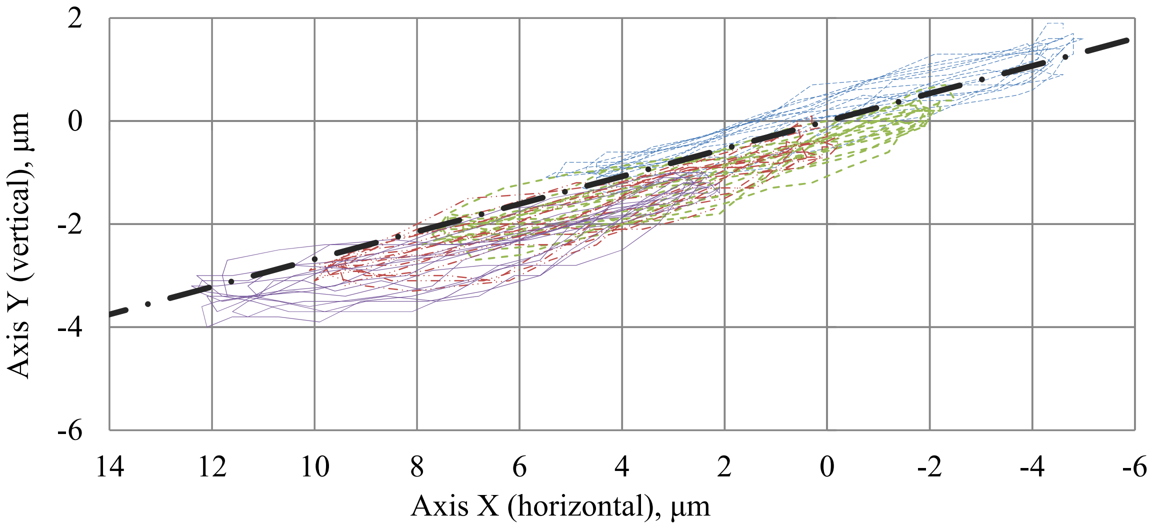


Figure B.3. Controlled vibration with frequency of 198.0 Hz with 15°



(b) with angle 45°



(b) with angle 75°

Figure B.4. Controlled vibration with frequency of 198.0 Hz

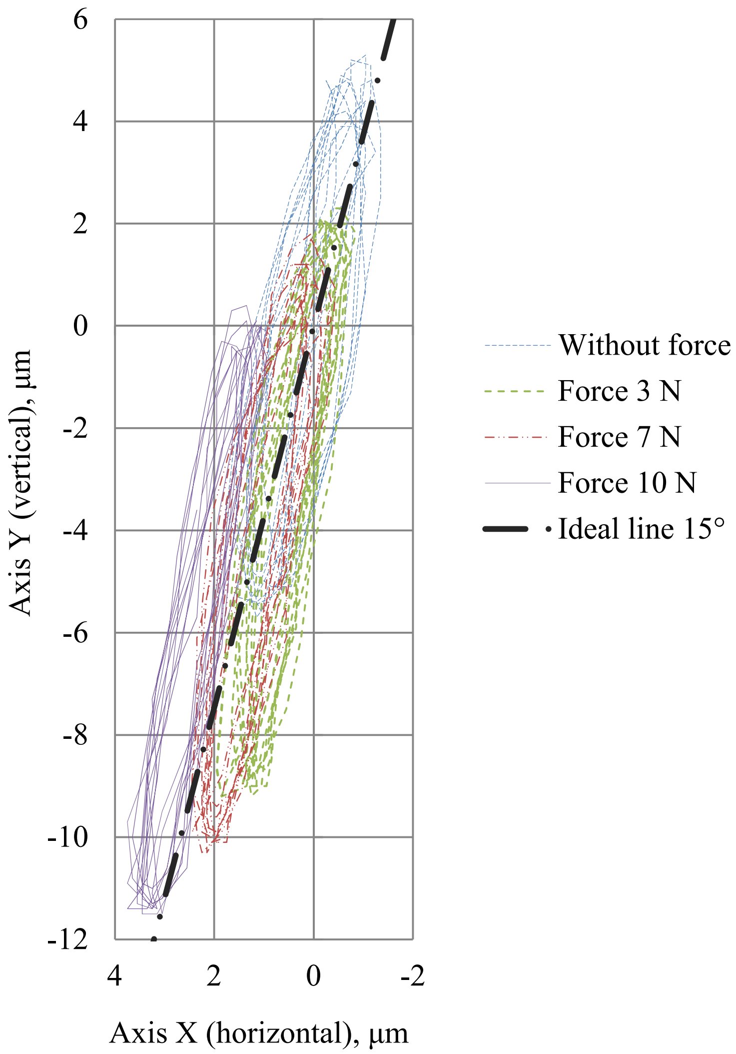
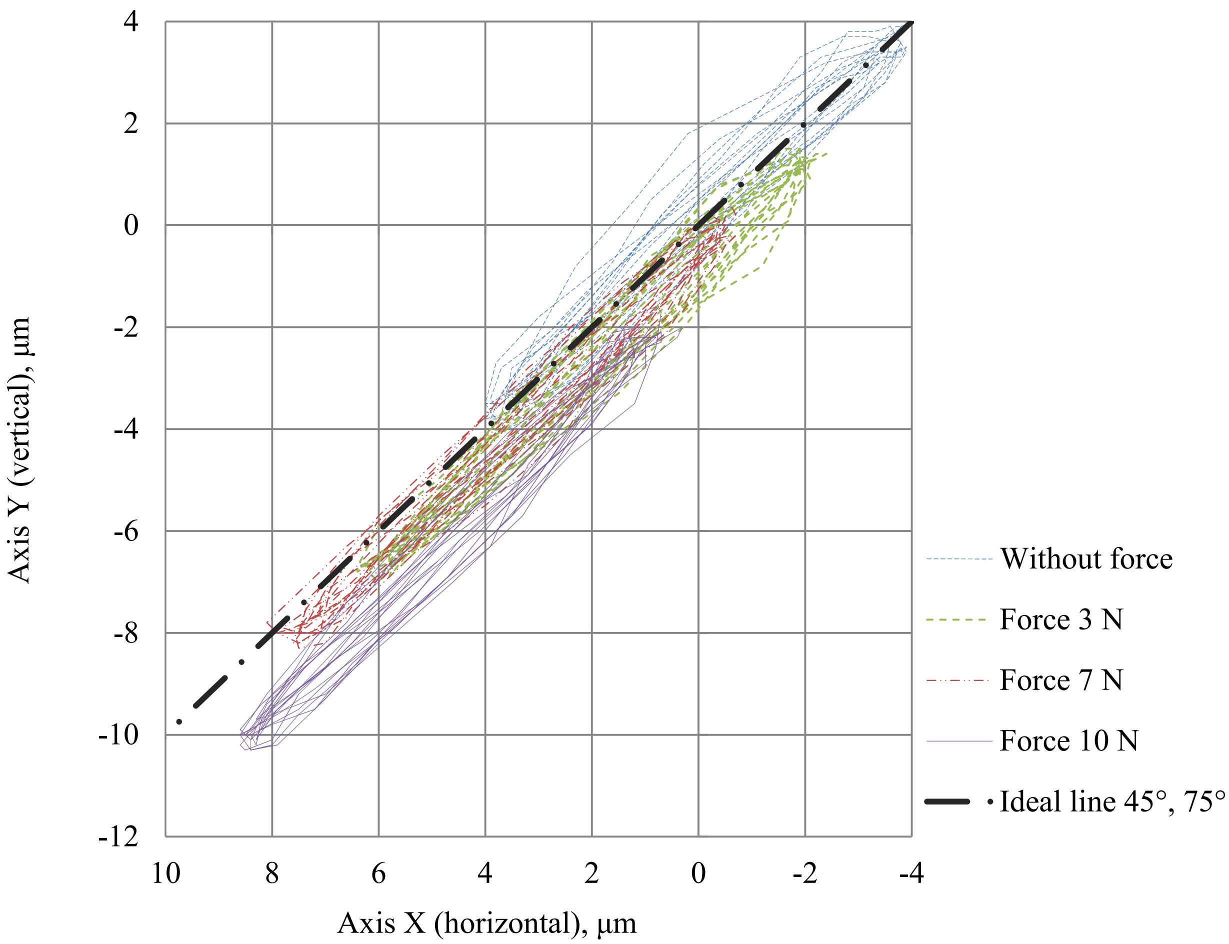
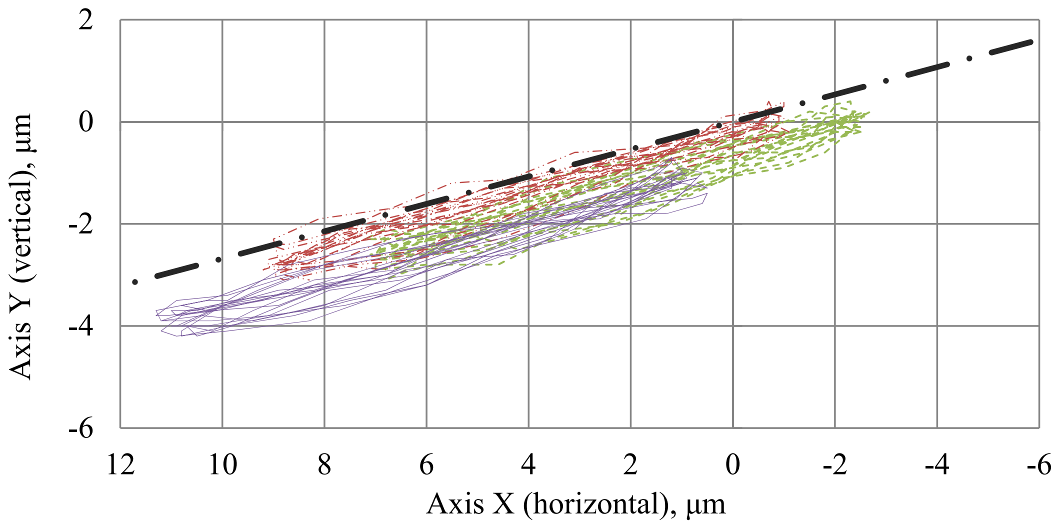


Figure B.5. Controlled vibration with frequency of 252.3 Hz with 15°



(a) with angle 45°



(b) with angle 75°

Figure B.6. Controlled vibration with frequency of 198.0 Hz