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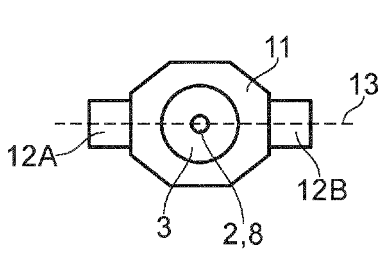

FIG. 4A

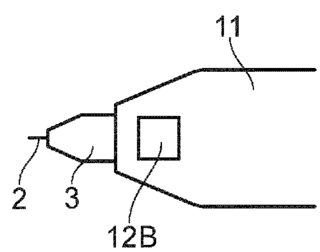

FIG. 4B

(57) **Abstract:** A multi-axis milling machine comprising a spindle (3), the spindle arranged to receive a tool holder (3) and in use to cause rotation of the tool holder within the spindle, wherein a portion of the spindle housing (11) surrounding the tool holder (3) is provided with one or more damping units (12A, 12B).

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DAMPER FOR A MILLING MACHINE

Background

The present invention is concerned with a machining apparatus particularly, but not exclusively, for machining super-alloyed components.

Super-alloy materials are metal alloys which are extremely hard and durable and which can withstand high temperatures and stresses. They are used in many applications including the aerospace industry where such materials are often required for high performance components. These components also often have small geometrical tolerances and so careful machining is required using specialist hardened cutting tools and materials.

Complex geometries can be created using multi-axis machining centres such as a 5-axis machining centre of the type manufactured by GROB-WERKE GmbH & Co. KG.

Multi-axis machining centres are typically computer numerically controlled (CNC) machining centres which allow highly accurate control of a cutting head at the high speeds needed to economically and accurately form components from super alloy materials. Cutting speed can typically be in the region of 12,000 revolutions per minute of the cutting tool.

The inventors have established that the performance of such machinery can be improved in terms of accuracy, reliability and cutting performance using an unconventional modification. Furthermore, the arrangement described herein advantageously dramatically reduces incidents of machine stop alarms. This can dramatically increase productivity and reduce machine down-time.

Summary of the Invention

Aspects of the invention are set out in the accompanying claims.

Viewed from a first aspect there is provided a multi-axis milling machine comprising a spindle, the spindle arranged to receive a tool holder and in use to cause rotation of the tool holder within the spindle, wherein a portion of the spindle housing surrounding the tool holder is provided with one or more damping units.

Thus, providing a multi-tuned mass damper in such a configuration advantageously allows the vibrations that otherwise propagate through the machine and tool holder to be controlled. Each machine tool (for example a CNC multi-axis milling machine) has its own harmonic or natural frequencies which are a result of the design and build of the machine and the operating (cutting) conditions. During a cutting operation the natural harmonic frequencies can cause the machine to vibrate above predetermined alarm thresholds.

Each machine exhibits its own harmonic 'weakness' in a certain direction or axis i.e. a specific direction. Because of the build of each machine the exact axis of harmonic 'weakness' varies. The term 'weakness' is intended to refer to an axis and position at which the natural harmonics of the machine are most prevalent i.e. where a machine begins to vibrate excessively.

Conventionally unwanted vibrations are addressed by providing the cutting tool holder and machine with sufficient mass and rigidity to prevent unwanted vibrations and harmonics developing.

Typically, when cutting super-alloys at high cutting speeds with ceramic tools, vibrations can develop in the cutting head and/or machine. These vibrations can exceed safe limits for the machine and or cutting tip. Machine manufacturers therefore incorporate vibration alarms that are activated at predetermined levels to stop a machine. This thereby prevents machine and/or tool damage. The machine can then be reset and restarted.

This safety system in conventional machinery advantageously prevents machine or tool damage but dramatically reduces productivity as the number of alarm activations can increase.

The present invention dramatically reduces unwanted alarm activations by identifying the ‘weak’ harmonic points on the machine and controlling and attenuating the vibration at or close to the point where the vibrations are most severe.

The cutting speeds used with ceramics cutting tips can be extremely high. For example the cutting speeds may exceed 12,000 revolutions per minute. Natural harmonics of the machine can thereby very quickly reach an alarm threshold causing a machine to stop operating. In overnight machining, where operatives set a machine to operating overnight, an alarm early in the machining time can result in very low productivity due to excessive machine stoppages.

In an arrangement described herein, the one or more damping units may be coupled to the spindle housing at a position corresponding to the location of a resonant frequency of the housing at a predetermined spindle operating speed.

For a given cutting speed (which will be known for the part being machined and the materials involved) a position corresponding to the point at which the machine will vibrate resonantly at or close to that frequency can be determined. A damping arrangement as described herein can then be located at or close to that position. Furthermore, the damper can be arranged in line or parallel with the direction of the predicted resonant vibration. The resonance that would ordinarily occur for the cutting speed can then be attenuated.

The or each damping unit may comprise an outer housing enclosing a damping mass, the damping mass being movable within and with respect to the housing. Housing a damping mass within the housing and allowing the mass to move allows for the damping to take place. The latency of movement of the mass with respect to the housing causes the damping to take place.

The housing and damping mass may be in the form of a concentric cylinder and a surrounding (also cylindrical) housing. The concentric axis of the cylinder and housing may be arranged to be parallel with the axis along which the spindle housing vibrates at resonance. Thus, the damper is arranged to attenuate the resonant vibration of the machine in the most effective way.

Other cross-sections of damper mass and housing may also be used. For example the housing and mass could be elliptical allowing for greater mass with a smaller vertical height. Equally a square or rectangular arrangement may be used. However, a cylindrical arrangement allows for convenient manufacture and also allow for a more accurate way to align the mass with respect to the housing such that the damper itself is dynamically stable.

The damping mass may be spaced from the surrounding housing by one or more elastic/elastomeric coupling(s). Importantly the couplings must allow relative movement of the damping mass with respect to the housing to allow the damping to take place.

The elastic coupling may be in the form of a plurality of elastomeric or rubber elements. These may be arranged uniformly around the damping mass so as to be in contact with the damping mass on one side and in contact with the housing on the other. Conveniently the elements may be in the form of O-ring seals positioned proximate to the perimeter of the damping mass. This allows for convenient installation and alignment of the member. For example, the member may be arranged in a groove with the housing and/or the damping mass.

Depending on the vibrational performance of the machine a single damping unit may be all that is needed to attenuate the resonant frequency of the machine at the cutting frequency/cutting speed and load. Alternatively, one or more pairs of damping units may be arranged around the machine, for example on the spindle housing of a milling machine. This advantageously allows the damping mass to be distributed to allow for convenient coupling of the dampers to the body of the machine and to avoid any issues with the damper body reducing the envelope of movement of the cutting head with respect to the workpiece.

The dampers may be conveniently positioned circumferentially and optionally equally spaced around the spindle housing of the machine.

The elastomeric or rubber elements may be continuous, for example O-ring seals. Portions of the inner surface of the end faces of the damper or portions of the end faces of the damping mass may optionally be provided with recesses in alignment with the normal line of contact of the elastomeric elements and the end faces. The term end faces is intended to refer to the ends of the damper in line with the axis of vibration that the damper it working to attenuate. In a cylindrical arrangement this would be on the end faces of the cylinder.

By providing recesses which are arranged to be in line with or overlap the normal line of contact of the O-ring seal the O-ring is effectively de-coupled from compression contact with the housing and damper mass at that particular point or over the length of overlap of the recess. Relative movement of the damping mass with respect to the housing will not then compress the elastomer at these points. This reduces the overall contact area the elastomer has with the damper mass and housing. For a given input force the contact area is less and the pressure on the remaining elastomer material in contact between the two surfaces is increased.

Advantageously the increase in pressure causes the damper to perform (i.e. to initiate damping) at a lower input force than if full contact of the elastomer is provided. The increased pressure initiates damping much more quickly resulting in damping at lower input forces, for example associated with low levels of cutting force for finishing machining.

Advantageously the recesses and O-ring arrangement are located on either end of the damping unit i.e. on an end of the damping unit facing the machine and on an opposing end of the damping unit. Vibrations along the axis of the damper towards the machine can then be effectively damped.

Viewed from another aspect there is provided a damping unit for a multi-axis machining centre arranged in use to be coupled to a spindle of the machining centre and comprising an outer housing enclosing a damping mass, the damping mass being movable within and with respect to the housing.

Viewed from yet another aspect there is provided a multi-axis milling machine comprising a spindle, the spindle arranged to receive a tool holder and in use to cause rotation of the tool holder within the spindle, wherein the spindle comprises a housing incorporating at least 1 pair of opposing damping units.

Viewed from a still further aspect of an invention described herein there is provided a machining centre comprising a spindle, the spindle arranged to receive a tool holder and in use to cause rotation of the tool holder within the spindle, wherein a portion of the spindle housing surrounding the tool holder is provided with at least one damping unit, the damping unit comprising a damping mass positioned within a housing and spaced therefrom by a plurality of elastomeric portions, each elastomeric portion being in contact with the damping mass on a first side and with the housing on an opposing side.

In such an arrangement portions of the housing and or damping mass may be provided with one or more recesses arranged along a line of contact between the elastomeric portion and the damping mass/housing so as to prevent contact between the elastomeric portion and the damping mass/housing across the one or more recesses.

The recesses may be formed on end faces of the damping mass and or opposing inner surfaces of the damping housing. Thus, the increased pressure effect of providing the recesses is only active in the direction of the damper in alignment with or parallel to the direction along which damping is primarily desired.

Viewed from another aspect there is provided a method of damping a milling machine comprising the steps of: (A) identifying one or more positions and directions of resonant vibration at a frequency approximately equal to a cutting frequency for a machining operation; and (B) positioning a damper as described herein to the milling machine at the identified position and direction.

Drawings

Aspects of the invention will now be described, by way of example only, with reference to the accompanying figures in which:

Figures 1 shows a schematic of the main components of a computer numerically controlled (CNC) milling station or machine;

Figure 2 shows a spindle of such a milling machine;

Figure 3 is a schematic graph showing natural frequencies for an example spindle housing;

Figures 4A and 4B show an example spindle comprising a pair of damping units;

Figures 5A and 5B show an example spindle with and without a pair of damping units;

Figure 6 shows a cross-section through one damping unit;

Figures 7A and 7B show a cross-section through a damping unit and the positioning of elastomeric seals (such as O-ring seals or similar flexible/elastomeric seals);

Figures 8A, 8B and 8C show a further improved arrangement of a damper described herein; and

Figures 9A, 9B and 9C show alternatively configurations of damping units with respect to a spindle housing.

While the invention is susceptible to various modifications and alternative forms, specific embodiments are shown by way of example in the drawings and are herein described in detail. It should be understood however that drawings and detailed description attached hereto are not intended to limit the invention to the particular form disclosed but rather the invention is to cover all modifications, equivalents and alternatives falling within the spirit and scope of the claimed invention

It will be recognised that the features of the aspects of the invention(s) described herein can conveniently and interchangeably be used in any suitable combination. It will also be recognised that the invention covers not only individual embodiments but also combinations of the embodiments that have been discussed herein.

Detailed Description

Figure 1 is a schematic of a typical multi-axis machining centre 1. The example shown is a milling machine comprising a cutting tool 2 coupled to a spindle 3. The spindle 3 incorporates a tool holder which holds the cutting tool at one end and provides a connection for coupling the tool holder and cutting tool to the spindle 3.

The spindle is coupled to a spindle drive arrangement 4 which is arranged to rotate the spindle (and thus the tool) at high speed. The drive arrangement 4 is also arranged so that it can move the spindle and tool in an axial direction along the x axis shown to the right of figure 1.

At the opposing end of the machining centre (to the drive arrangement) is the chamber that houses the component to be machined (the workpiece) 5 which is secured to a movable table 6. The table 6 is movable in multiple places (vertically and rotationally). In combination with the axial movement of the cutting spindle the cutting tool can be moved relative to the workpiece in each of the axes shown to the right of figure 1.

The machining centre comprises a large number of sensors which accurately detect the position of the table 6 and cutting tool so that in operation accurate machining can be performed.

The table 6, spindle 3 and drive arrangement 4 each have a large mass to prevent excessive vibrations during a machining operation. Increasing the mass and rigidity of each of the components of the machining centre reduces vibrations during machining, allowing for accurate machining. A vibration alarm 7 is however provided which deactivates the cutting (milling) operation if vibration levels exceed a predetermined threshold. In this scenario a vibration or ‘excessive movement’ alarm is used which detects machine vibration and compares it to a predetermined threshold or level. Vibrations may increase beyond a predetermined level for a number of reasons such as increased tool wear, an increased hardness in a portion of the workpiece or unfavourable cutting characteristics/conditions.

Figure 2 shows a simple cross-section through the spindle 3, cutting tool 2 and workpiece 5. The cutting tool 2 is connected to the spindle 3 by means of a tapered tool holder 8 which allows cutting tools to be conveniently changed within the spindle 3. During operation of the machine the spindle 3, tool holder 8 and tool 2 are caused to rotate at speed of in excess of 12,000 revolutions per minute as shown by the arrow 9.

To allow for intricate shapes and geometries to be machined the end of the spindle 3 closest to the cutting tool 2 (and thus workpiece 5) may on some machines be tapered such that the tool can cut at angles with respect to the workpiece without the spindle or tool holder connecting with or colliding with the workpiece. In other machines no taper may be provided on the spindle. It will be appreciated that the invention described herein applies equally to a machine with and without a tapered head.

Each machine tool has a number of natural frequencies. The person skilled in the art will understand that in the context of this invention a natural frequency is a frequency at which parts of the milling machine will vibrate in the absence of any force either exciting a vibration or acting to damp a vibration i.e. it is the frequency at which the part naturally vibrates.

When a machine is caused to vibrate at or close to the natural frequency resonance may occur which can cause vibration alarms to be triggered.

The way a machine vibrates during use is a function of two stiffness considerations, namely:

(i) Static stiffness; and
(ii) Dynamic stiffness.

During the operation of a machine a damping arrangement described herein can advantageously influence the dynamic stiffness. Specifically, adding damping increases the dynamic stiffness of the machine (and/or subcomponents of the machine) when the damping is applied in a specific way as described herein.

All machine tools have a number of natural frequencies in their basic construction. Some of these natural frequencies can be disruptive and lead to excessive vibrations during a machining operation. The natural frequencies originate from each part or element in the machine tool. The different vibrations interact (both constructively and destructively) resulting in an overall vibrational performance of a machine being complex and specific to the given machine and, importantly, operating or cutting speeds and conditions. Different cutting speeds for example result in different harmonics of vibration being exhibited by the machine.

In one example, there is a need to improve the dynamic stability of a machine at a given cutting speed in revolutions per minute (rpm). For example, for a ceramic cutting tip this may be a cutting speed of 12,000 rpm and a frequency of 200Hz.

By performing vibrational analysis of each part of the machine using accelerometers and an impact hammer it is possible to establish how each part of a machine vibrates and specifically the natural frequencies of each part.

Example equipment that can be used to measure the vibrational characteristics of the machine is as follows:

Accelerometers – Kistler model 8776A50
Impulse Hammer – Kistler model 9726A5000

The outputs from the accelerometer and impulse hammer may be communicated to a conventional computer through a input/output interface of the type manufactured by National Instruments. Analysis of the data can then be formed using commercially available software such as CUTPRO. The vibrational characteristics i.e. the force/vibration characteristics of each part of the machine can be mapped my moving the accelerometer and/or using multiple accelerometers to determine where vibrations occur and their associated natural frequencies. The damper can then be positioned according to this determination in combination with the expected operating speed of the machine.

Specifically, a model can be built and the location and direction of the part of the component with the natural frequency which is most closely matched to the cutting frequency can be identified. This is termed herein as the 'weakest' part of the machine i.e. the part of the machine that needs to be damped to prevent resonance at the given cutting speed and frequency (12,000 rpm and 200Hz in one example).

The vibration analysis can be conducted using conventional accelerometers and impact hammer(s). The accelerometers are distributed around the machine and a predetermined impact is applied to the machine to cause the machine to vibrate. The accelerometer outputs can then be recorded and the location of the machine sub-component with a natural frequency closest to the cutting frequency (that is the rpm speed divided by 60) can be identified. Similarly the accelerometers can be used to determine the direction in which this part of the machine vibrates.

Thus, using accelerometers and an impact hammer it is possible to locate the part or parts of the machine which will vibrate close to the cutting frequency. This part is therefore the part that is likely to resonate during the cutting process because the cutting frequency will cause resonance which in turn will cause that part of the machine to vibrate at increasing amplitude. This, in turn, can cause the vibration alarm of the machine to be triggered.

In one example embodiment a machine has a weakest dynamic stability at 200Hz in an X direction of the machine. The vibration analysis described above established that the natural frequency of 200Hz can be found at the machine's spindle housing in an X direction.

Once the location and direction of the ‘weakest’ natural frequency is found, it then defines the position and direction at which a multi-tuned mass damper (MTMD) should be located. In one example this is the top of the spindle housing.

Figure 3 shows a graph illustrating frequency versus displacement and the associated natural frequencies for an example spindle housing

As shown in figure 3 the displacement in the y axis is caused by the vibration of the machine. Three separate frequencies are illustrated by figure 3.

- The first is frequency M at approximately 100Hz which reflects the resonance of the machine tool foundations i.e. the base structure of the machine.
- The second frequency A is at 200Hz and represents the resonance of the spindle housing.
- The third frequency B is at 1200Hz and represents the spindle shaft frequency.

Figure 4A shows an end view of the spindle 3 holding the tool holder 8 and cutting tool 2 along its central axis. The surrounding housing 11 which supports the spindle is also shown. In Figure 4B a side view of the housing 11 is shown illustrating the optionally tapering of the housing.

Returning to Figure 4A the modified housing arrangement according to an invention described herein is shown. Specifically a pair of opposing damping units 12A and 12B are shown. However, a pair of opposing dampers may not be required; a single damper may provide sufficient damping to achieve the desired results for a given machine.

The damping units 12A and 12B are mounted onto the sides of the spindle housing 11 and, in the example shown, are arranged to align with an axis 13 (that passes through the centre of the spindle and cutting tool). The precise axis may vary depending on the machine and the results of the vibration analysis and may not always pass through the centre of the spindle shaft. The damping units 12A and 12B are firmly connected to the spindle housing for example using a nut and bolt arrangement. They may advantageously be integrally formed into the spindle housing to form a single housing component incorporating the pair of opposing damping units or zones. They may also be retrofitted to the outer surface of existing housings.

The damping units 12A and 12B act to interact with the vibration of the spindle housing which passes through the spindle shaft. In effect the damping units provide a multi-tuned mass damper for a machining centre which allows for the control of machining vibrations.

The damping units 12A, 12B will now be described in further detail with reference to figures 5A, 5B, 6, 7A and 7B.

Figures 5A and 5B show an example spindle housing with and without the pair of damping units 12A and 12B respectively. As shown in figure 5B the pair of damping units are coupled directly to the sides of the spindle housing 11 by means of a bolt arrangement. Such an arrangement can be conveniently retro-fitted to the spindle housing.

Figure 6 shows a cross-section through the damping unit 12B shown in figure 5B.

The outer housing of the damping unit 12B is formed of 3 parts defining a central cavity or space to receive a damping mass. The first part 14 is arranged for abutment with the spindle housing 11 as shown in figure 5B. The second part of the housing is opposing face 15. A cylindrical body 16 extends between the two faces and can be coupled to each one by means of a flange extending around the perimeter of the cylindrical portion 16. The flange can be conveniently bolted on both sides of the central body 16 to define the cavity within the housing and to rigidly connect the housing to the spindle housing. The same arrangement is optionally replicated on the opposing side of the spindle housing for damping unit 12A.

The outer housing of the damping unit defines an inner cavity or chamber 17 which is described with reference to figure 7A and 7B.

The chamber contains a damping mass 18 which is contained within and enclosed by the outer casing. The damping mass is also spaced from the inner surface of the outer case via a plurality of elastic/elastomeric members 19. The elastic members shown are in the form of elastomeric O-ring seals that extend around perimeter of the damping mass. Other types of elastic spacers may also be used and may not be in the form of a continuous profile but may be a number of discrete elastomeric portions providing the same spacing effect as an O-ring seal.

As shown in figures 7A and 7B a first pair of O-ring seals are arranged on the end faces of the cylindrical damping mass and a second pair are arranged on the outer peripheral surfaces of the damping mass. As shown in figure 7B two clearances Δx and Δy are thereby provided to space the damping mass from the outer casing.

During operation of the machining apparatus the vibration of the spindle housing causes vibration of the outer casing of the damping unit. This in turn causes vibration of the damping mass through the elastomeric O-ring connections and this creates a damping effect because of the latency of movement of the damping mass with respect to the damping unit as the elastomeric couplings compress and expand out of synchronisation with the excitation frequency.

The precise level of damping can be optimised by adjusting the clearances Δx and Δy and the size and material (hardness or elasticity) of the O ring material (and also the specific mass of the damping mass itself).

In one example the damping units are configured to provide a desired dynamic stability and damping rate for a given machine design and for the anticipated machining speeds. This defines the frequency of damping needed.

For example, for a milling machine as manufactured by GROB (as discussed above) the parameters determined include:

Damping mass:	Densimet 19,0 kg/dm^3, damping mass = 4,7 kg each (2 dampers).
Material Elasticity:	O-ring 70x3 Nitril
	O-ring 80x3 Nitril

A damping unit with these characteristics is then located on the machine at a position corresponding to the position of weakest dynamic stability (as described above). Importantly, this acts to increase the dynamic stability of the spindle housing (this having been identified by the frequency analysis). The effect of improving the dynamic stability is that machine vibration is dramatically reduced at the desired cutting speed for super alloy materials. This, in turn, prevents vibration levels exceeding those of the manufacturer defined limits and thereby prevents an automatic shut-down of the machine (a vibration alarm trigger). Furthermore, the accuracy of machining can also be increased through a reduction in vibration and thus displacement of the cutting tool with respect to the workpiece.

Advantageously the performance of the damping units described herein can be further optimised beyond the advantages described above. Figures 8A, 8B and 8C illustrate such a further improvement.

Specifically, the inventors have established that creating a discontinuous or interrupted elastomeric coupling between the damping mass and the damper housing provides a surprising improvement in dynamic performance of the damper.

In the examples described above a continuous elastomeric O-ring seal is used in the damping arrangement. Thus, there is a continuous contact line between:

(i) the O-ring and the damper mass (in the cavity of the damper); and
(ii) the O-ring and the housing of the damper (around the periphery of the damper mass)

In effect the O-ring is sandwiched between the damper mass and the damper housing. Compression and expansion of the elastomer allows for the damping described herein.

Figure 8A illustrates the damping performance of the modified damper with damping percentage on the y axis and input cutting force on the x axis.

Line A in figure 8A shows damping performance of the first embodiment of a damper as described above i.e. a damper with a continuous elastomeric coupling between damping mass and damper housing. As shown damping percentage increases slowly with input force and peaks at around 1200 Newtons before tailing off as input force increases and damping effect reduces.

Line B illustrates a modified damper performance incorporating a discontinuous coupling of elastomer to damping mass and housing. As illustrated by line B the percentage of damping increases dramatically faster than the conventional arrangement. As shown the damping percentage increases at very low input forces providing damping at low machining loads.

Also, the damping percentage remains higher than the conventional arrangement for all input loads.

In operational terms this means that a modified damper can provide effective damping at both low machining loads associated with finishing machining and also high machining loads associated with roughing machining. This broad range of damper performance as provided by the damper arrangement described herein is highly advantageous in machine tools.

Figures 8B and 8C show one arrangement to achieve these performance improvements. In the example shown a series of recesses are formed in the damper housing and/or damper mass in line with the contact line of the O-ring seal. A recess 20 is formed so as to de-couple or disconnect the O-ring 19 from compression contact with the damper mass 18 and end plate 14. In effect, the recess 20 disconnects the O-ring seal from connecting between the mass and the housing of the damper. This effectively reduces the contact area of the O-ring seal between the damper mass and housing at either end of the damper. Reducing the contact area increases the pressure on the remaining portions of the O-ring which are in contact with the damper mass on one side and the housing on the other.

The inventors have established that reducing the contact area of the elastomer seals using the recess leads to the performance change shown in figure 8A i.e. further improved dynamic damping performance.

The specific performance of the damper can be selected depending on the characteristics of the machine and the desired damping performance. For example the contact area may be modified in combination with the elastomer hardness to achieve the desired performance. In one example, a shore hardness of between 70 and 90 shore A may be used.

Figure 8C shows examples of how the recesses may be incorporated into the end housings of the damper.

At the left hand side of figure 8C the damper connection to the machine is shown and incorporates a plurality (4 in this example) of slots. The dotted line indicates the normal contact line of the O-ring. As shown in figure 8B, the way the recesses overlap this contact line creates a space which results in the de-coupling of the compression contact between the O-ring and the damping mass.

At the right hand side of figure 8C the damper connection to the end housing of the damper is shown. Here, for simplicity of manufacture, 4 semicircles have been machined into the end portion of the housing. Again, the dotted line shows the contact line of the O-ring which overlaps with the 4 semicircles.

It will be recognised that the de-coupling of the contact line of the elastomer can be achieved in a variety of ways and is not limited to the two examples shown. For example, the O-ring itself could be modified to have a reduced thickness or to include recesses or cut-away portions creating the same effect. Similarly instead of a single continuous O-ring or elastomeric member, a plurality of discrete elastomeric portions may be provided.

Figures 9A-9C illustrate alternative arrangements of damping units with respect to the spindle. The dampers may be single dampers or dampers arranged in pairs or groups. Importantly in each case the damping effect is arranged to improve the dynamic stability of the machine at the point (and in a direction) where a resonance will occur during the anticipated cutting parameters.

As shown in figures 9A to 9C the dampers may be arranged at a variety of positions around the spindle housing.

It will be recognised that the damping arrangement and approach described herein may be situated at any position of a machine according to the natural frequency measurements and the desired cutting speed/frequency. For particular 5 axis milling machines it has been determined that the optimal position may, as described above, be on the spindle housing itself.

CLAIMS

1. A multi-axis milling machine comprising a spindle, the spindle arranged to receive a tool holder and in use to cause rotation of the tool holder within the spindle, wherein a portion of the spindle housing surrounding the tool holder is provided with one or more damping units.

2. A machine as claimed in claim 1, therein the one or more damping units is coupled to the spindle housing at a position corresponding to the location of a resonant frequency of the housing at a predetermined spindle operating speed.

3. A machine as claimed in any preceding claim wherein the or each damping unit comprises an outer housing enclosing a damping mass, the damping mass being movable within and with respect to the housing.

4. A machine as claimed in claim 3, wherein the housing and damping mass are in the form of a concentric cylinder and surrounding housing and wherein the concentric axis of the cylinder and housing is arranged to be parallel with the axis along which the spindle housing vibrates at resonance.

5. A machine as claimed in claim 5, wherein the damping mass is spaced from the surrounding housing by one or more elastic coupling(s).

6. A machine as claimed in claim 5, wherein the elastic coupling is in the form of a plurality of elastomeric or rubber elements.

7. A machine as claimed in claim 6 wherein the elastomeric or rubber elements are in the form of O-ring seals positioned proximate to the perimeter of the damping mass.

8. A machine as claimed in any preceding claim comprising one or more pairs of damping units circumferentially spaced around the spindle.

9. A machine as claimed in any preceding claim, wherein the elastomeric or rubber elements are continuous and portions of the inner surface of the end faces of the damper and/or portions of the end faces of the damping mass are provided with recesses in alignment with the line of contact of the elastomeric elements and the end faces.

10. A damping unit for a multi-axis machining centre arranged in use to be coupled to a spindle of the machining centre and comprising an outer housing enclosing a damping mass, the damping mass being movable within and with respect to the housing.

11. A multi-axis milling machine comprising a spindle, the spindle arranged to receive a tool holder and in use to cause rotation of the tool holder within the spindle, wherein the spindle comprises a housing incorporating at least 1 pair of opposing damping units.

12. A machining centre comprising a spindle, the spindle arranged to receive a tool holder and in use to cause rotation of the tool holder within the spindle, wherein a portion of the spindle housing surrounding the tool holder is provided with at least one damping unit, the damping unit comprising a damping mass positioned within a housing and spaced therefrom by a plurality of elastomeric portions, each elastomeric portion being in contact with the damping mass on a first side and with the housing on an opposing side.

13. A machining centre as claimed in claim 12, wherein portions of the housing and or damping mass are provided with one or more recesses arranged along a line of contact between the elastomeric portion and the damping mass/housing so as to prevent contact between the elastomeric portion and the damping mass/housing across the one or more recesses.

14. A machining centre as claimed in claim 13, wherein the recesses are formed on end faces of the damping mass and or opposing inner surface of the damping housing.

15. A method of damping a milling machine comprising the steps of:

(A) identifying one or more positions and directions of resonant vibration at a frequency approximately equal to a cutting frequency for a machining operation; and

(B) positioning a damper as claimed in claim 11 to the milling machine at the identified position and direction.

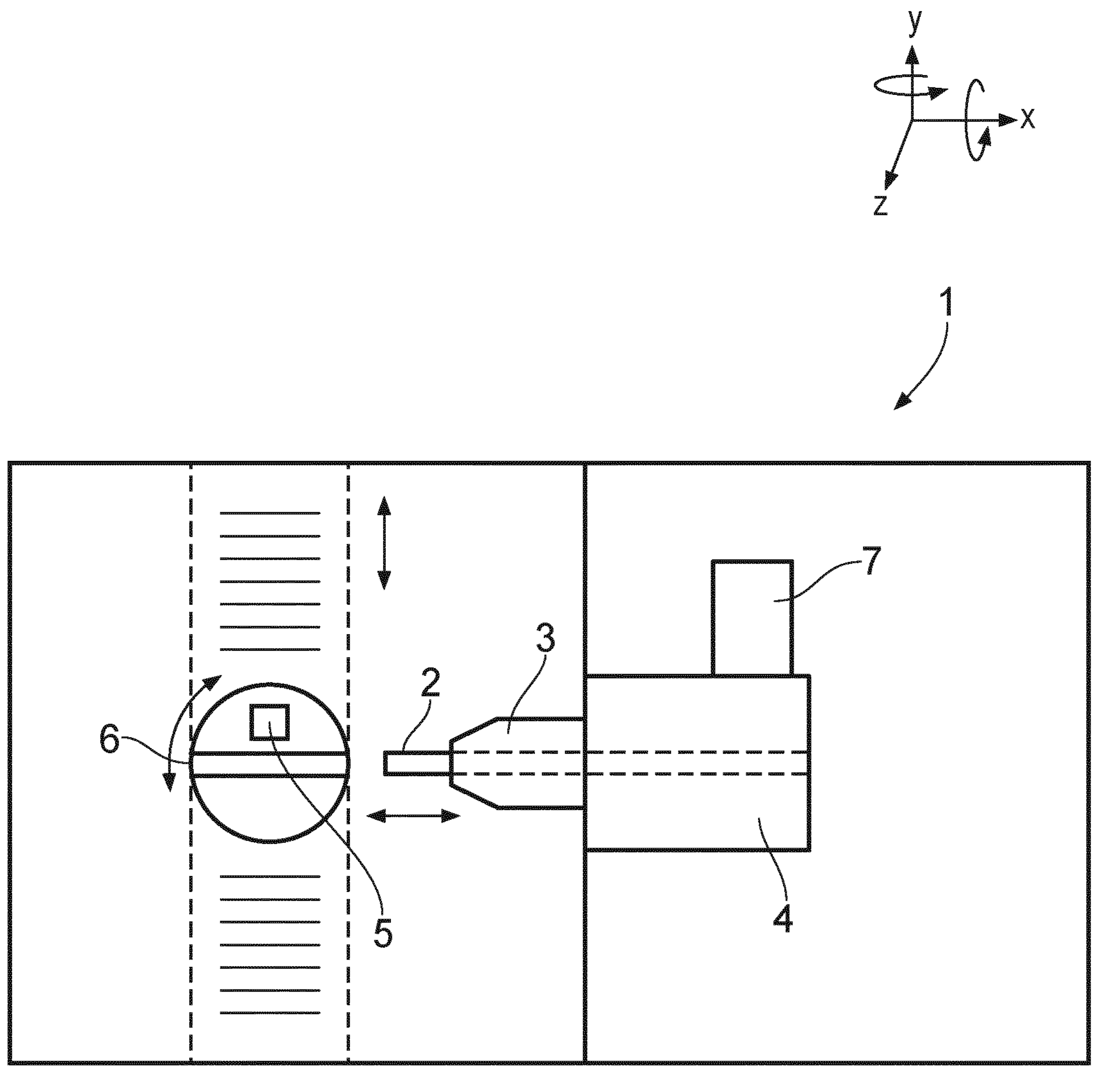

FIG. 1

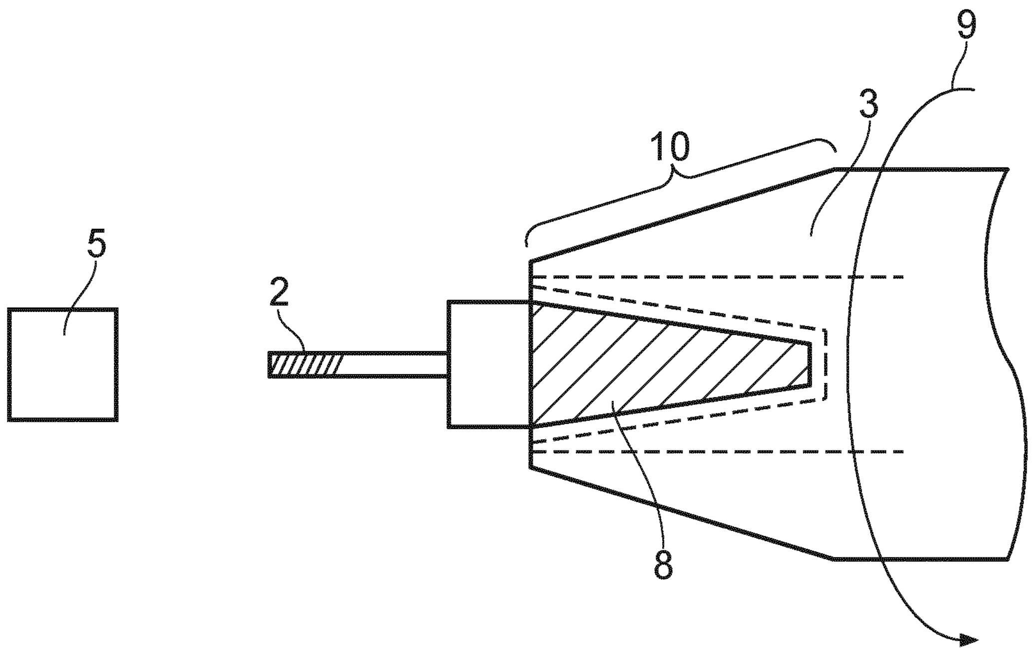

FIG. 2

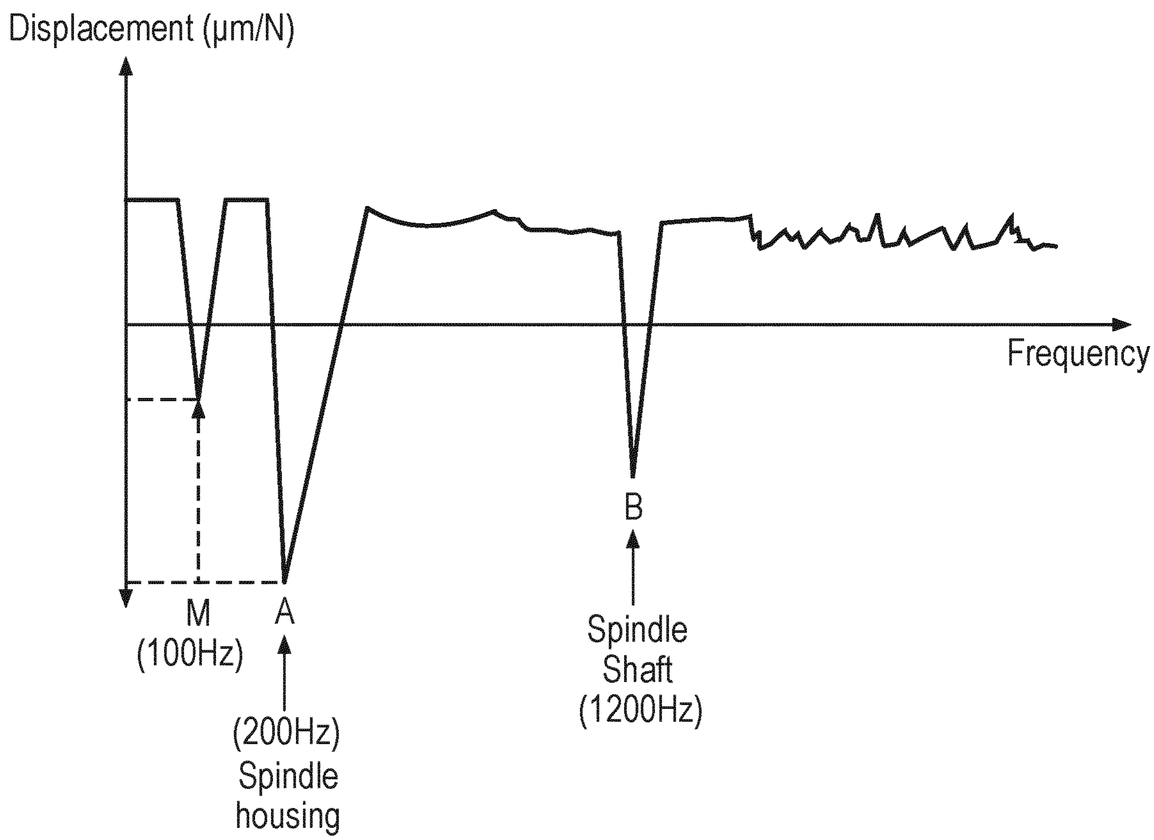
Displacement (μm/N)
Frequency
M
(100Hz)
A
(200Hz)
Spindle
housing
B
Spindle
Shaft
(1200Hz)

FIG. 3

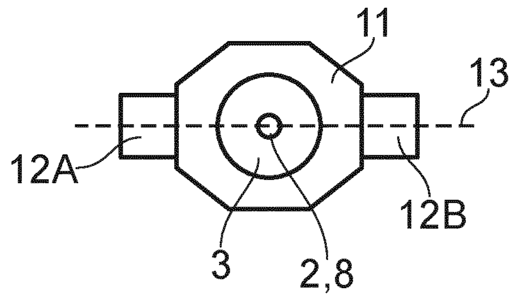

FIG. 4A

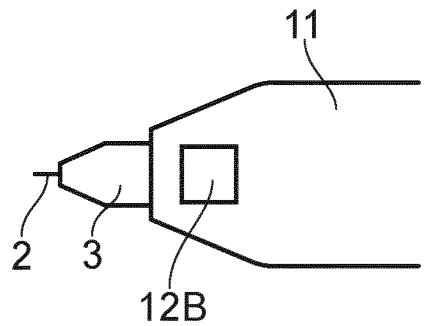

FIG. 4B

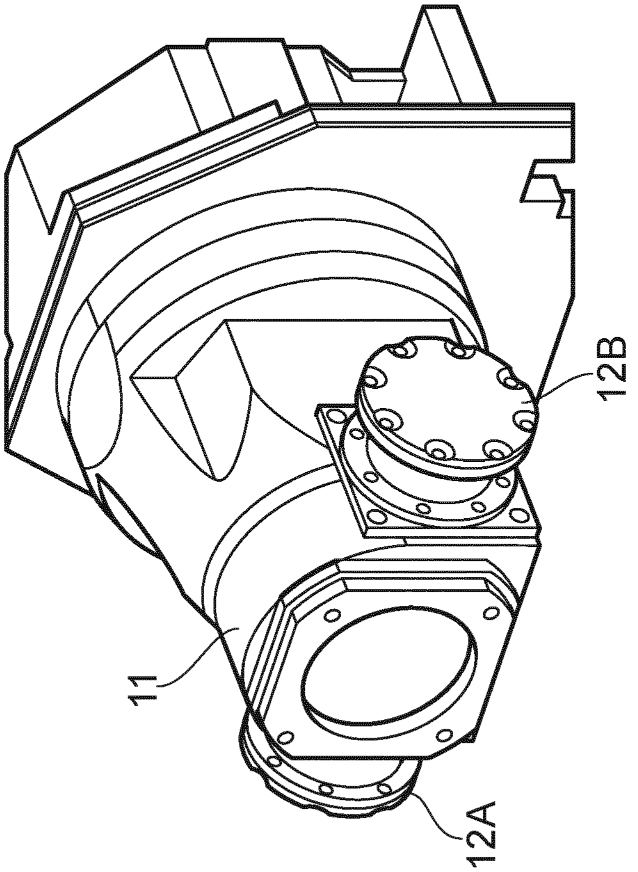

FIG. 5B

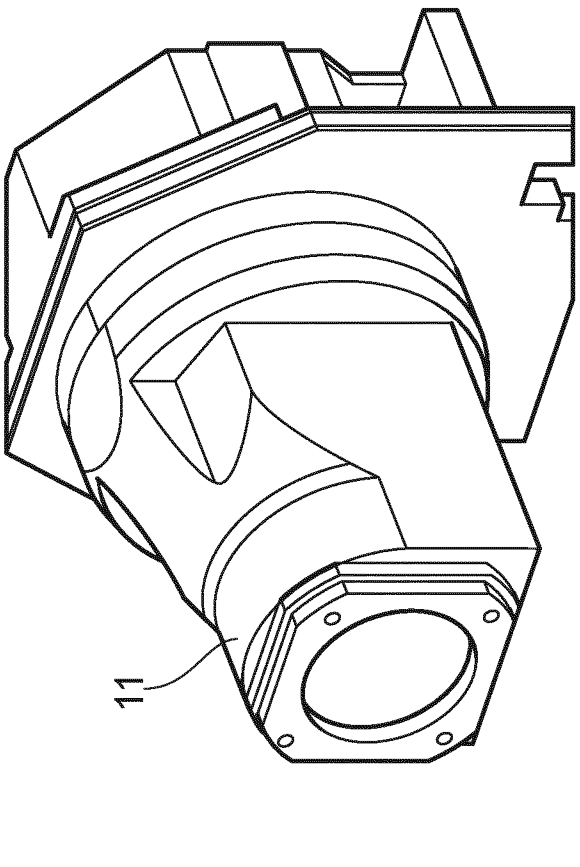

FIG. 5A

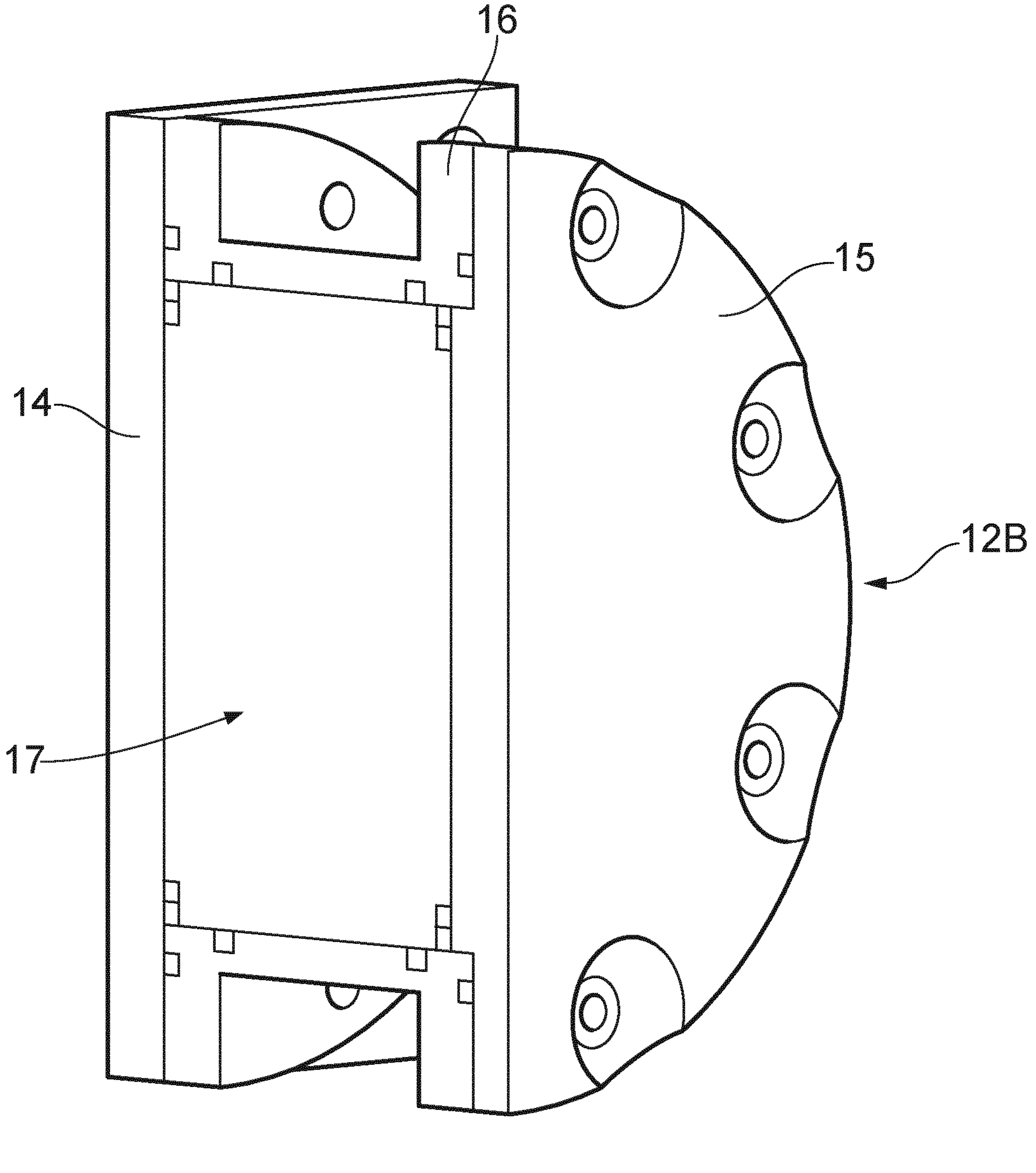

FIG. 6

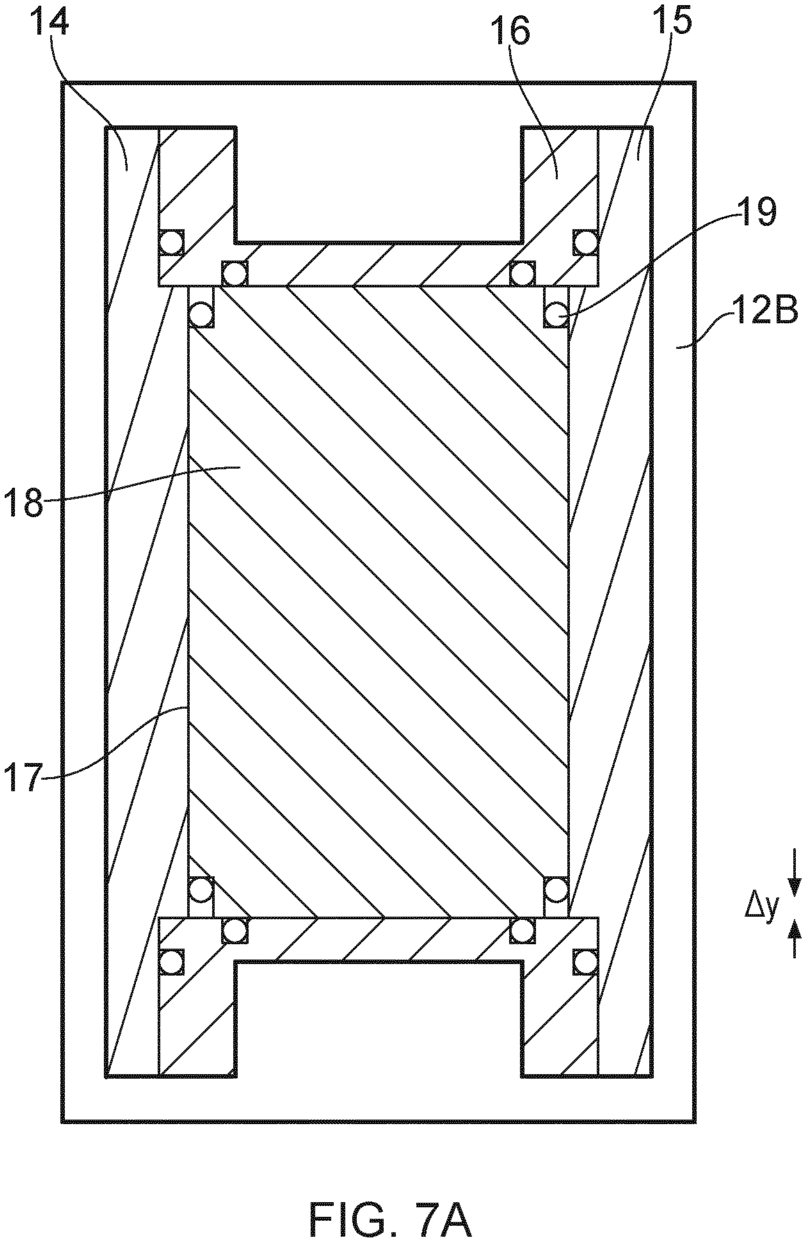

FIG. 7A

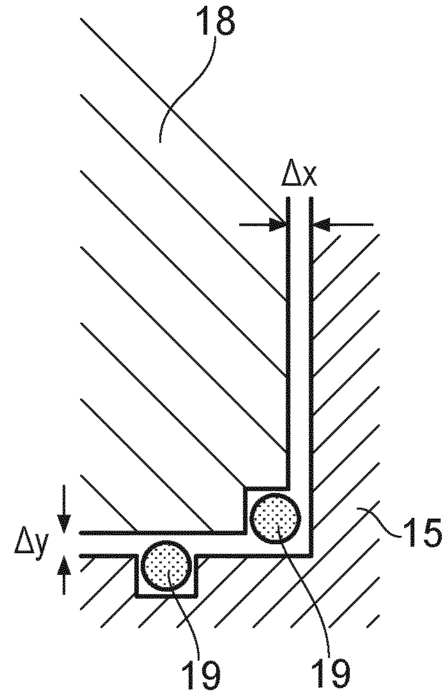

FIG. 7B

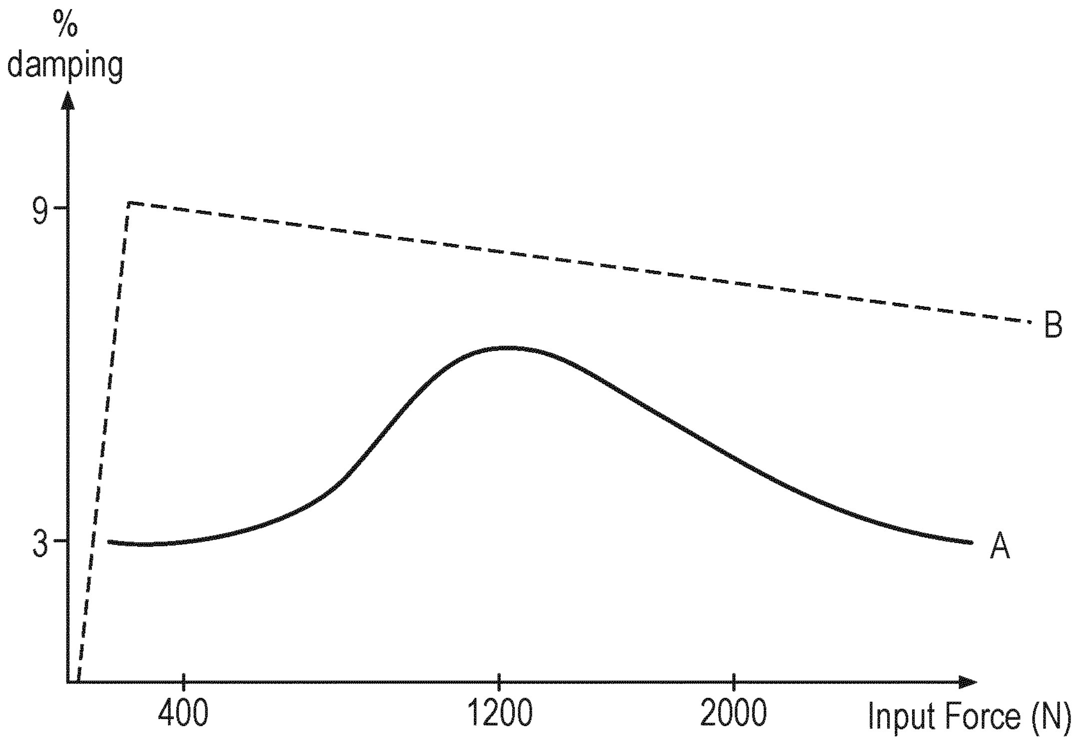

FIG. 8A

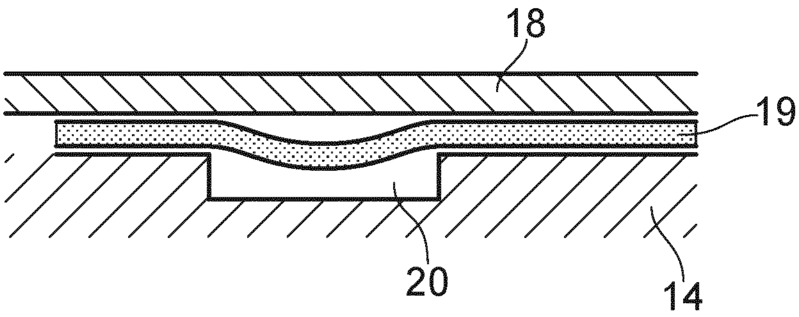

FIG. 8B

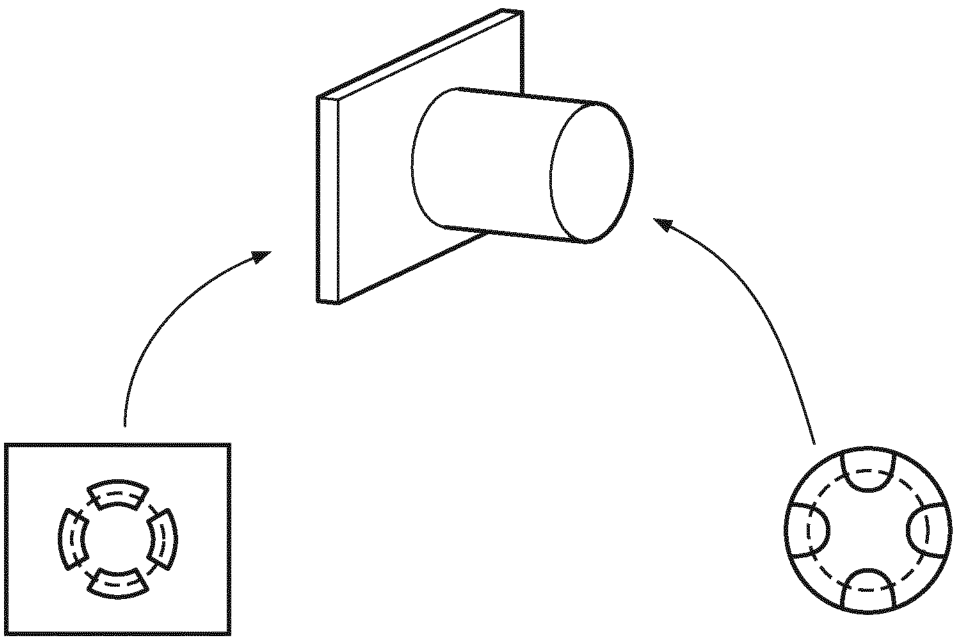

FIG. 8C

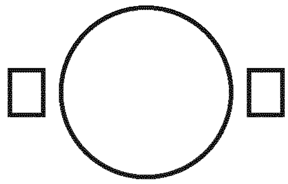

FIG. 9A

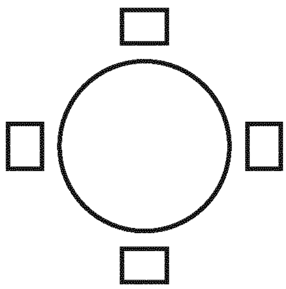

FIG. 9B

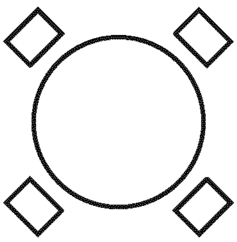

FIG. 9C

INTERNATIONAL SEARCH REPORT

International application No
PCT/EP2020/069425

A. CLASSIFICATION OF SUBJECT MATTER
INV. B23Q11/00 B23Q1/70
ADD.

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
B23Q

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	DE 10 2016 210233 A1 (HOMAG GMBH [DE]) 14 December 2017 (2017-12-14) paragraphs [0049], [0051]; figures 4,6	1
X	DE 10 2007 025934 A1 (SIEMENS AG [DE]) 11 December 2008 (2008-12-11) paragraph [0031]; figures 1-4	1-4,8
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X	US 2016/207154 A1 (NEBUKA TEPPEI [JP]) 21 July 2016 (2016-07-21) paragraph [0059]; figures 1,2,4,5	1,15
	-/--	

[X] Further documents are listed in the continuation of Box C.

[X] See patent family annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
8 October 2020	22/10/2020

Name and mailing address of the ISA/	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Fax: (+31-70) 340-3016	Lasa Goñi, Andoni

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C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

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X	EP 3 210 718 A2 (DECKEL MAHO PFRONTEN GMBH [DE]) 30 August 2017 (2017-08-30) figures 3,4	1
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A	figures 1-2	1,4-7

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Information on patent family members

International application No
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