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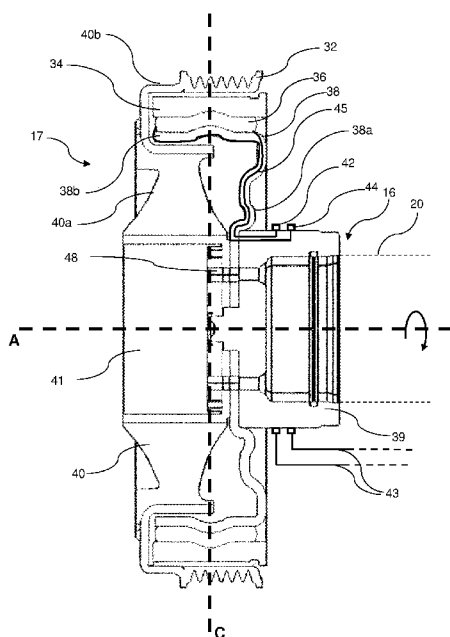


Figure 3

(57) Abstract: **Vibration damper for a vehicle powertrain** A vibra-
tion damping system (16) for use in a vehicle powertrain (12), com-
prising: a first rotatable component (38); a second rotatable component
(40); and, a damper (36) coupled to at least one of, or between, the first
component (38) and the second component (40); the damper compris-
ing at least one piezoelectric element (50) responsive to vibrations of
the first component (38) and/or the second component (40), wherein
the damper (36) comprises a resilient matrix (56), the resilient matrix
(56) being associated with the at least one piezoelectric element (50);
wherein the second rotatable component (40) is an inertia mass (34) or
an output component (40) of the powertrain (12) configured to trans-
mit drive from an engine or gearbox of the vehicle.



Vibration damper for a vehicle powertrain

TECHNICAL FIELD

- 5 The present disclosure relates to a vibration damper to attenuate vibrations induced in certain rotatable engine components, for example in the vehicle powertrain or cranktrain. Aspects of the invention relate to a vibration damper, a powertrain or cranktrain incorporating a vibration damper, to a control system for controlling the vibration damper and to a vehicle comprising such a powertrain or cranktrain.

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BACKGROUND

- In a conventional automotive vehicle, the rotational force of the internal combustion engine (ICE) is transmitted to the wheels via a powertrain of the vehicle. During
15 operation of the vehicle, high combustion load input events in the ICE are known to transmit torsional vibrations along components of the powertrain, such as the crankshaft, which causes the components to twist and bend.

- The torque fluctuations induced by the torsional vibration can be sufficient to cause the
20 components of the powertrain to fatigue over time. It can also result in significant noise vehicle harshness (NVH) issues. Internal combustion engines are typically fitted with passive torsional vibration dampers (TVDs), which are arranged to absorb torsional vibration energy through the elastic deformation of rubber dampers or through viscous losses of a hydraulic fluid. In this way the TVDs convert torsional vibration energy into
25 thermal energy, which is then dissipated to the local environment. Consequently, such passive dampers are inherently inefficient at transmitting power from the ICE as a significant proportion of the torque is lost as heat which is radiated from the TVD. Furthermore, passive dampers are unable to react to the varying output of the ICE such that they are ineffective at certain engine speeds and conditions.

30

More recent attempts to address the problem of torsional vibration have led to the development of TVDs which are arranged to dynamically damp the torsional vibrations produced by the ICE. These dynamic dampers typically include electromagnetic

actuators which are arranged to apply an opposing torque to the crankshaft so as to counteract the effects of the torsional vibration.

5 However, the frequency range of the torque fluctuations means that the attenuation of torsional vibrations is challenging, particularly at very high vibration frequencies where the electromagnetic actuators struggle to match the vibrational frequency of the vibrations. Conversely, at very low vibration frequencies the inherent stiffness and structural rigidity of the actuator results in a significant proportion of the vibrational energy to be transmitted through the powertrain.

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Furthermore, with the introduction of increasingly stringent emission regulations, vehicle manufacturers are being encouraged to produce ever more efficient vehicles whilst maintaining high levels of refinement and increased vehicle service life. Consequently, there is a need to harness the vibrational energy whilst simultaneously
15 damping providing more effective vibrational damping.

The present invention has been devised to mitigate or overcome at least some of the above-mentioned problems.

20 SUMMARY OF THE INVENTION

According to an aspect of the present invention there is a vibration damping system for use in a vehicle powertrain, comprising: a first rotatable component; a second rotatable component; and, a damper coupled to at least one of, or between, the first component
25 and the second component; the damper comprising at least one piezoelectric element responsive to vibrations of the first component and/or the second component, wherein the damper comprises a resilient matrix, the resilient matrix being associated with the at least one piezoelectric element; wherein the second rotatable component is an inertia mass or an output component of the powertrain configured to transmit drive
30 from an engine or gearbox of the vehicle.

The elements are able to absorb a displacement of the damper caused by the vibrations that are transmitted along the first and/or second component. The elements are also able to sense the vibrations and/or harvest electrical energy from the kinetic

energy associated with the vibrations that are transmitted along the first and/or second component. Advantageously, the piezoelectric elements may be configured to dampen vibrations that exhibit a wide range of frequencies and/or amplitudes. Moreover, the piezoelectric elements are configurable to selectively attenuate only vibrations that
5 cause fatigue to the first and/or second components, whilst allowing other, none fatiguing, vibrations to continue un-attenuated.

The piezoelectric element may be configured to adjust the stiffness of the resilient matrix so as to dampen the vibrations of the first and/or second components.
10

The at least one piezoelectric element may be embedded within the resilient matrix. Advantageously, the resilient matrix provides minimum amount of damping even if the piezoelectric elements were to fail during use, due to a system fault. In this way, the damper may provide hybrid structure of the damper provides an inbuilt resilience to
15 failure.

At least one of the piezoelectric elements may be bonded directly to at least one of the first or second rotatable components. By bonding the piezoelectric elements directly to the are able to counteract to first or second rotatable components, the piezoelectric
20 elements are able to more readily attenuate the vibrations therein.

The at least one piezoelectric element may extend about the first rotatable component. In this way, the piezoelectric element is able to conveniently apply torque to the first component in order to attenuate torsional vibrations of the first component.
25

The above described vibration damping system may include a plurality of piezoelectric elements. Advantageously, the plurality of piezoelectric elements may be configured to simultaneously attenuate vibrations with different frequencies and/or amplitudes.

30 The plurality of piezoelectric elements may be arranged to extend circumferentially about the first rotatable component.

The plurality of piezoelectric elements may be arranged in a plurality of rows, in which the rows are circumferentially spaced about the first rotatable component. The rows

may be arranged in more than one deck. Advantageously, the piezoelectric elements when arranged in this decked or stacked configuration provide a greater displacement when activated and thus, the damper is able to exert a greater damping force on the first and/or second component. Equally, the elements are also able to absorb a larger
5 displacement of the damper and harvest more electrical energy from the vibrations that are transmitted along the first and/or second component.

The at least one piezoelectric element may be substantially H-shaped in cross section.

- 10 The above described vibration damping system may further comprise a control system that is operatively coupled to the at least one piezoelectric element and configured to control the operation of the system in at least one of a damping mode or a generating mode.
- 15 The control system, in the damping mode, may be operable to apply a control signal to the at least one piezoelectric element to attenuate vibrations of the first or second rotatable member.

The control system may be configured to apply the control signal to the at least one
20 piezoelectric element in dependence on receiving an input signal indicative of a level of vibration of the first or second rotatable component. The input signal may be provided by the at least one piezoelectric element.

The control system, in the generating mode, may be operable to harvest electrical
25 energy from a voltage signal generated by the at least one piezoelectric element.

The first component may be an output component of the powertrain, the output component configured to drive a peripheral system of the vehicle.

- 30 The peripheral system may be at least one of an alternator, a fluid pump or a compressor for an air conditioning unit of the vehicle.

The second rotatable component may be an inertia mass. The second rotatable component may be an output component of the powertrain, the output component configured to transmit drive from an engine or gearbox of the vehicle.

- 5 According to a further embodiment of the present invention, a vehicle powertrain is provided; the vehicle powertrain comprising the vibration damping as described above.

According to a yet further embodiment of the present invention, a vehicle is provided; the vehicle comprising the vehicle powertrain as described above.

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According to an aspect of the present invention there is provided a vibration damping system for use in a vehicle powertrain, comprising: a crankshaft; an inertia mass; and, a damper coupled to at least one of, or between, the crankshaft and the inertia mass; the damper comprising at least one piezoelectric element responsive to vibrations of
15 the crankshaft and/or the inertia mass.

According to an aspect of the present invention there is provided a damper suitable for coupling a first rotatable component to a second rotatable component of a vehicle powertrain; the damper comprising at least one actuator disposed within a resilient
20 matrix, wherein the actuator is responsive to relative vibrations between the first component and the second component.

The actuator may be configured to modify the stiffness of the resilient matrix in response to the relative vibrations between the first component and the second
25 component. The actuator may be formed from a piezoelectric material.

As used herein, the term "control unit" will be understood to include both a single control unit or controller and a plurality of control units or controllers collectively operating to provide the required control functionality. A set of instructions could be provided which, when executed, cause said controller(s) or control unit(s) to implement
30 the control techniques described herein (including the method(s) described below). The set of instructions may be embedded in one or more electronic processors, or alternatively, the set of instructions could be provided as software to be executed by one or more electronic processor(s). For example, a first controller may be

implemented in software run on one or more electronic processors, and one or more other controllers may also be implemented in software run on or more electronic processors, optionally the same one or more processors as the first controller. It will be appreciated, however, that other arrangements are also useful, and therefore, the present invention is not intended to be limited to any particular arrangement. In any event, the set of instructions described above may be embedded in a computer-readable storage medium (e.g., a non-transitory storage medium) that may comprise any mechanism for storing information in a form readable by a machine or electronic processors/computational device, including, without limitation: a magnetic storage medium (e.g., floppy diskette); optical storage medium (e.g., CD-ROM); magneto-optical storage medium; read only memory (ROM); random access memory (RAM); erasable programmable memory (e.g. EPROM and EEPROM); flash memory; or electrical or other types of medium for storing such information/instructions.

It will be appreciated that the foregoing represents only some of the possibilities with respect to the particular subsystems of a vehicle that may be included, as well as the arrangement of those subsystems with the control unit. Accordingly, it will be further appreciated that embodiments of a vehicle including other or additional subsystems and subsystem arrangements remain within the spirit and scope of the present invention.

Within the scope of this application it is expressly intended that the various aspects, embodiments, examples and alternatives set out in the preceding paragraphs, in the claims and/or in the following description and drawings, and in particular the individual features thereof, may be taken independently or in any combination. That is, all embodiments and/or features of any embodiment can be combined in any way and/or combination, unless such features are incompatible. The applicant reserves the right to change any originally filed claim or file any new claim accordingly, including the right to amend any originally filed claim to depend from and/or incorporate any feature of any other claim although not originally claimed in that manner.

BRIEF DESCRIPTION OF THE DRAWINGS

In order that the invention may be more readily understood, reference will now be made, by way of example only, to the accompanying drawings, in which:

Figure 1 is a schematic drawing of a vehicle provided with a powertrain including an
5 internal combustion engine, a control system and a vibration damper system;

Figure 2 is a schematic diagram of the vehicle powertrain of Figure 1;

Figure 3 is a schematic longitudinal-sectional view of the vibration damping system of
10 Figure 1;

Figure 4 is a schematic longitudinal-sectional view of the vibration damping system of
Figure 3, showing a longitudinally arranged damper according to an embodiment of the
invention;

15 Figure 5 is a schematic perspective view of a piezoelectric element of the vibration
damping system of Figure 4;

Figures 6 and 7 are schematic longitudinal-sectional views of the vibration damping
20 system of Figure 3, each showing a longitudinally arranged damper according to an
alternative embodiment of the invention;

Figures 8, 9 and 10 are schematic longitudinal-sectional views of the vibration
damping system of Figure 3, each showing a radially arranged damper according to an
25 alternative embodiment of the invention;

Figure 11 is a schematic view of a damping control unit which is arranged to control
the vibration damper system according to an embodiment of the invention; and

30 Figure 12 is a schematic longitudinal-sectional view of the vibration damping system of
Figure 1, according to an alternative embodiment of the invention.

DETAILED DESCRIPTION

A specific embodiment of the invention will now be described in which numerous specific features will be discussed in detail in order to provide a thorough understanding of the inventive concept as defined in the claims. However, it will be apparent to the skilled person that the invention may be put in to effect without the specific details and that in some instances, well known methods, techniques and structures have not been described in detail in order not to obscure the invention unnecessarily.

To place the embodiments of the invention in a suitable context, reference will firstly be made to Figure 1, which schematically illustrates a vehicle 10 provided with a powertrain assembly 12 including an internal combustion engine 14 and a vibration-damping system 16. In the following description, the powertrain assembly 12 refers to both a powertrain and a cranktrain of the vehicle, each being rotatable components of the engine 14 that are powered thereby. In use, the engine 14 drives a crankshaft assembly comprising a front crankshaft 20 that is coupled to a peripheral system 17, which is located at a position that is forward of a front end of the engine 14, with respect to the vehicle 10. The peripheral system 17 is a front end accessory device (FEAD) such as a compressor for an air-conditioning system, an alternator for charging a battery or a fluid pump for supplying coolant or oil to the engine 14. The front crankshaft 20 is arranged to transmit the output from the engine 14 to power the operation of the FEAD.

The front crankshaft 20 is further coupled to the vibration-damping system 16, which is arranged between the engine 14 and the peripheral system 17, i.e. upstream, along the front crankshaft 20. Consequently, the vibration-damping system 16 is arranged to modulate the output from the engine 14 as it is transferred from the engine 14 to the peripheral system 17 so as to attenuate the torque oscillations in that output.

A rear crankshaft 21 is arranged to transfer torque from the engine 14 to a rear axle 22, which, in turn, is arranged to drive a pair of rear wheels 24 of the vehicle 10. In this way, the rear crankshaft 21 is configured to transfer power from the engine 14 to downstream components of the powertrain 12.

Figure 1 describes features that are significant for this discussion, although it should be appreciated that many features that are common to vehicle powertrains are not shown here for brevity, for example the transmission and engine control unit (ECU) required for a functioning powertrain assembly, gearbox, dual-mass flywheel and the like are not described in detail. However, the skilled person would understand that these features are implicitly present in a conventional powertrain assembly 12 as described herein.

A control system or 'controller' in the form of a damping control unit 18 is arranged to receive an input signal that is indicative of the level of vibration of the powertrain assembly. The input signal comprises an engine-parameter signal from an engine sensor 26 and a vibration signal from a vibration sensor 28.

The vibration sensor 28 is a piezoelectric element (not shown) which is housed within the vibration-damping system 16 as will be described in more detail below. In alternative embodiments the vibration sensor 28 may be one of a number of sensors including, for example, an accelerometer, a strain gauge, a Hall Effect sensor, a rotary potentiometer, a piezoelectric sensor or any other sensing means for either directly or indirectly measuring the torsional vibration of the crankshaft assembly. Alternatively, the torsional vibrations may be predicted using an output of a computational model of the powertrain assembly.

The engine sensor 26 is connected to the engine 14 and is configured to detect operational parameters of the engine 14, whereas the vibration sensor 28 is coupled to the front crankshaft 20 and is configured to detect the torsional vibrations of the front crankshaft 20, which are caused by torque fluctuations in the engine 14. The engine-parameter signal includes information relating to the engine load, as determined by the cylinder pressure, and the engine speed. The damping control unit 18 is also arranged to provide control signals to the vibration-damping system 16 in dependence on the sensor signals it receives from each sensor. The control signals control the vibration-damping system 16 to actively attenuate or absorb the torsional vibrations that would otherwise be transmitted along the powertrain 12. More specifically, the damping control unit 18 is configured to dynamically control the operation of the vibration-

damping system 16 in dependence on sensor information it receives from each respective sensor.

5 Additionally, a power supply in the form of a vehicle battery (not shown) is provided to supply power for all electrical systems of the vehicle 10. Electrical power is directed to and from the vibration-damping system 16 through a power electronics module (not shown) of the damping control unit 18.

10 Figure 2 shows the powertrain assembly 12 and the vibration-damping system 16 in more detail. As is shown in Figure 2, the front crankshaft 20 is attached, at one end, to the peripheral system 17, and at another end to a flywheel 30 of the engine 14. As would be well known to the skilled person, the crank shaft is coupled to connecting rods of the engine by which means torque is transmitted to the crank shaft. In the illustrated embodiment, the vibration-damping system 16 is arranged between the
15 engine 10 and the peripheral system 17.

The vibration sensor 28 is housed within the vibration-damping system 16 and is configured to measure the acceleration and deceleration of the crankshaft and provide the vibration signal to the damping control unit 18. In a variant embodiment, the
20 vibration sensor 28 may be coupled to the input crankshaft 20a at a point where it meets the vibration-damping system 16. The damping control unit 18 is electrically connected to the vibration sensor 28 and further connected to the engine sensor 26, which, in turn, is in communication with the engine 14.

25 The structure and operation of the damping system 16 will now be described in more detail with reference to Figure 3, which shows a schematic longitudinal-sectional view of the vibration-damping system 16, which is arranged with the peripheral system 17 to form a single structural assembly according to an embodiment of the invention.

30 The vibration-damping system 16 comprises a first rotatable component and a second rotatable component. The first rotatable component is an input component in the form of an input-assembly 38, which is located adjacent to and co-axially aligned with the second rotatable component, which is an output component in the form of an output-assembly 40 of the peripheral system 17.

The input-assembly 38 is nested within the output-assembly 40 such that a radial inward-facing surface of the input-assembly 38 opposes a radial inward-facing surface of the output-assembly 40. Where the radial inward-facing surfaces of the input and output assemblies 38, 40 meet defines a central-radial plane C of the vibration-damping system 16.

The input and output assemblies 38, 40 are arranged such that they share a common longitudinal axis, which is shown as the dotted line A in Figure 3. The front crankshaft 20 is co-axially aligned with and fixedly coupled to an input-hub 39 of the input-assembly 38, which is rigidly coupled to an output-hub 41 of the output assembly 40.

The input-assembly 38 comprises a radial portion 38a and an axially extending annular 'flange' portion 38b. The radial portion 38a extends in a substantially radial direction from the front crankshaft 20 and comprises an inward-facing surface and an outward-facing surface, which are arranged, respectively, to face towards and away from the central-radial plane C of the vibration-damping system 16. The annular portion 38b extends perpendicularly from the inward-facing surface of the hub at a circumferential edge of the radial portion 38a and comprises an inward-facing surface and an outward-facing surface, which are arranged, respectively, to face towards and substantially away from the longitudinal axis of the vibration-damping system 16.

The output-assembly 40 also includes a radial portion 40a, which extends in a substantially radial direction from the output-hub 41. The radial portion 40a has an inward-facing surface and an outward-facing surface, which are arranged to face towards and away from the central-radial plane of the vibration-damping system 16, respectively. The output-assembly 40 further comprises an annular portion 40b which extends perpendicularly from the inward-facing surface at a circumferential edge of the radial portion 40a. The annular portion 40b has an inward-facing surface and an outward-facing surface, which are arranged, respectively, to face towards and away from the longitudinal axis of the vibration-damping system 16.

The input and output assemblies 38, 40 are arranged such that the annular-portion 40b of the output-assembly 40 overlaps the annular-portion 38b of the input-assembly

38. Thus, the annular portions 38b, 40b protrude towards each other, from their respective assemblies, in a direction that is substantially parallel to the longitudinal axis A. Although not shown here, in some embodiments, a bearing may be located between the annular portion 40b of the output-assembly 40 and the inertia ring 34 in order to ensure the movement of the inertia ring 34 remains coaxial.

The vibration-damping system 16 includes a damper 36, which is arranged between the input assembly 38 and the inertia ring 34 to attenuate the torsional vibrations that are transmitted therebetween. More precisely, the damper 36 is arranged between the annular portion 38b of the input-assembly 38 and the inertia ring 34. The inertia ring 34 is an inertia mass made from a heavy material such as, for example, steel. The inertia mass forms an annulus which has a width that is substantially equal to the annular portion 38b of the input-assembly 38.

The damper 36 is arranged between the inertia ring 34 and the input-assembly 38. The damper 36 forms a belt which has a width that is substantially equal to the annular portion 38b of the input-assembly 38 and the inertia ring 34. The inner surface of the inertia ring 34 is bonded to an outer surface of the damper 36. An inner surface of the damper 36 is bonded to the outward-facing surface of the annular portion 38b of the input-assembly 38.

The vibration-damping system 16 also includes a means of connecting the damper 36 to the damping control unit 18 through which the control unit 18 can send and receive signals to and from the damper 36. The connecting means comprise a first and second control ring 42, 44 which are arranged on an external surface of the assembly-hub 39. A first set of cables 43, of the connecting means, form an electrical connection between the damping control unit 18 and an external surface of the control rings 42, 44. A stationary connector assembly in the form of a pair of slip rings provides an electrical connection between the stationary cables 43 and the external surface of the control rings 42, 44. A second set of cables 45 provide a connection between an internal surface of the control rings 42, 44 and the damper 36 of the vibration-damping system 16.

In alternative embodiments, the control unit 18 may be arranged to send and receive signals to and from the damper and engine sensors 28, 26 using a wireless connection, for example Bluetooth™.

- 5 The outward-facing surface of the annular-portion 40b of the output-assembly 40 defines a grooved pulley surface 32 that couples to a belt (not shown), which is driven by the rotation of the output-assembly 40. The belt may be driven by the pulley to power an FEAD of the vehicle 10, such as a compressor for an air-conditioning system, an alternator for charging a battery or a fluid pump for supplying coolant or oil
10 to the engine 14. The skilled person will appreciate that the pulley 32 is not an essential feature of the vibration-damping system 16. Rather, it is a component of the peripheral system 17 in the embodiment illustrated here.

- As described earlier, the input assembly 38 of the vibration damping system 16 and
15 the output assembly 40 of the peripheral system 17 are arranged to abut each other to define the central-radial plane C of the damping system 16. More specifically, a mating-surface of the input-assembly 38 is arranged in contact with a mating surface of the output-assembly 40 in a central region of damping system 16. The mating surfaces are both aligned in parallel with the central-radial plane C of the damping
20 system 16 and perpendicularly to the longitudinal axis A of the vibration-damping system 16. The two opposing mating surfaces are retained by a plurality of bolts (not shown) that are housed within a number of corresponding passages 48 that are formed within the input and output assemblies 38, 40. The passages 48 are arranged in parallel with the longitudinal axis A of the vibration-damping system 16.

25

The damper 36 will now be described in more detail with reference to Figures 4 to 6, which show a longitudinal-sectional view of a portion of the vibration-damping system 16 according to an embodiment of the invention.

- 30 It will be apparent to a person skilled in the art that the vibration-damping system 16 may include an inertia ring 34 as shown in Figure 3, in which the damper 36 is arranged between the inertia ring 34 and the input-assembly 38 where it provides structural support for the inertia ring 34.

The damper 36 is mechanically bonded at a lower surface 55 to the input-assembly 38 and at an upper surface 57 to the inertia ring 34. In this way, the upper and lower surfaces 57, 55 define an inner and outer surface of the damper 36. The damper 36 also has an inner-radial surface 51 and an outer-radial surface 53, which are arranged
5 orthogonally to the inner and outer surfaces 55, 57.

With reference to Figure 4, the damper 36 has a composite structure as it is formed of two main components, which include a plurality of piezoelectric elements 50 that are associated with a resilient matrix 56. More specifically, the piezoelectric elements 50
10 are embedded within the resilient matrix of the damper 36. In embodiments, the elements 50 may be substantially or completely disposed within the external envelope that is defined by the resilient matrix of the damper 36. Alternatively, the elements 50 may be sandwiched between two opposing regions of resilient matrix to form the damper 36. In alternative embodiments, the elements 50 may be only partially
15 disposed so that the resilient matrix doesn't encompass each and every surface of the elements 50.

Attached to the piezoelectric elements 50 are a pair of electrodes 54 (in the form of an anode and a cathode) which are electrically connected to the damping control unit by a
20 series of wires (not shown).

The resilient matrix 56 is made from an elastic material which is configured to passively damp the vibrations of the input-assembly 38. It is envisaged that suitable materials for the matrix may be polyurethane-based polymers, which exhibit suitable
25 resistance to chemicals, oils and high-temperatures, and is generally robust to the harsh environment in which the damper 36 would be operating. The skilled person would understand that other materials could also be suitable. The matrix 56 also acts as an electrical insulating material (or isolator), which is configured to electrically isolate the electrodes 54 from each other and to further isolate the electrodes from the
30 input assembly 38 and the inertia ring 34.

The piezoelectric elements 50 are configured to deform (i.e. expand and contract) in dependence on a voltage being applied across them by the electrodes 54, which applies a compressive and tensile stress, respectively, on the surrounding matrix 56.

The surrounding matrix 56 material will be deformed by the change in shape of the piezo elements 50. The stiffness of the surrounding matrix 56 material in some examples will be increased as the material is compressed and may follow the general rule $f=kx$, where f is force applied, k is stiffness (N/mm rate) and x is displacement.

- 5 In some embodiments the piezo elements 50 could be configured within a matrix 56 material such that the stress could be reduced in the material as the piezo elements 50 contract or expand for example.

10 In this way, the piezoelectric elements 50 are able to harness the inverse piezoelectric effect to convert electrical energy into mechanical energy and thereby alter the elastic properties, or stiffness, of the damper 36. In this way, the damper 36 is operable to absorb a greater or smaller proportion of a damping torque M_d that is imposed on the damper 36 by the motion of the inertia ring 34 relative to the input-assembly 38. Thus, the damper 36 is operable to control the transmission of the damping torque M_d from
15 the inertia ring 34 to the input-assembly 38 in order to actively counteract the torsional vibrations that are exerted on output-assembly 40. Put another way, the damper 36 is configured to control the dampening force that is applied to the input-assembly 38 due to the counter rotation of the inertia ring 34.

- 20 The application of an increasing voltage across the piezoelectric elements 50 leads to the application of a higher damping torque M_d to the input-assembly 38, which thereby reduces the vibration amplitude of the harmonic torques. Thus, the torsional vibrations which result from the harmonic torques of the engine 14 are counteracted by actuation of the piezoelectric elements 50 of the damper 36 in order to damp the transmission of
25 the vibrations from the input-assembly 38 to the output-assembly 40.

Conversely, the expansion and contraction of the elements 50 under the influence of an external force produces a flow of charge across the surface of the elements 50. In this way, the elements 50 are able to harness the piezoelectric effect to detect the
30 compression and expansion of the damper 36.

With reference to Figure 4 and 5, the piezoelectric elements 50 have an H-shaped cross-section and comprise an elongated rectangular shaped spine 58 with elongated rectangular shaped flanges 60 arranged across upper and lower ends of the spine 58.

The spine 58 and flanges 60 define a first and second channel on either side of the spine 58 into which the electrodes 54 are located, respectively.

5 In the illustrated embodiment, the piezoelectric elements 50 form a linear array of one row and six columns. Thus, the array is six elements long, as between the inner and outer radial surfaces 51, 53 of the damper 36, and one element wide or high, as between the upper and lower surfaces 57, 55 of the damper 36. The piezoelectric elements 50 are orientated within the resilient matrix 56 such that the flanges 60 lie in parallel or co-planar with the upper and lower surfaces 57, 55 of the damper 36.

10

In alternative embodiments the piezoelectric elements 50 may comprise numerous different sizes and shapes including, for example, a box section or an annular ring. It will be clear to the skilled person that the piezoelectric elements 50 may be arranged at 90° to the alignment shown in Figure 4 or in any other arrangement that results in a
15 change in the stiffness of the damper 36. It is envisaged that the damper 36 may comprise many more piezoelectric elements 50 than are shown in the embodiments described herein. Moreover, each of the piezoelectric elements 50 may be configured to exhibit a particular shape and/or orientation such that they are able to preferentially counteract torsional vibrations of different frequencies.

20

The piezoelectric elements 50 are ferroelectrically soft piezoceramics with low polarity reversal field strengths which are configured to expand and contract in dependence on a voltage being applied across them by the electrodes 54.

25 Figure 6 shows an alternative embodiment of the damper 36 shown in Figure 4, in which each of the plurality of piezoelectric elements 50 are directly mounted to the input assembly 38 and the inertia ring 34. For each piezoelectric element 50, a first bonding strip 58a is located between an upper surface of the element 50 and the inner surface of the inertia ring 34. A second bonding strip 58b is located between a lower
30 surface of the element 50 and the outer surface of the radial portion 38b of the input-assembly 38. The bonding strips 58a, 58b are formed from an epoxy material. In embodiments, the bonding strips 58a, 58b may be formed from any material with the appropriate adhesive and structural properties.

As described above, the first and second bonding strips 58a, which mechanically couples the flanges 60 of the piezoelectric element 50 directly to the input-assembly 38 and the inertia ring 34. Consequently, there is no elastic material provided between the piezoelectric elements 50 and the input-assembly 38 or inertia ring 34. Hence, the kinetic energy is transmitted directly to and from the elements 50 and the flywheels 38, 40 and cannot be absorbed by the elastic material of the matrix 56, as with the embodiment described in Figure 5. Directly bonding the piezoelectric elements 50 reduces the amount of rubber that is used to form the damper 36. Consequently, packaging of the damper 36 within the rotatable components of the damping system 16 is improved. As a consequence of the piezoelectric elements 50 being directly bonded to the damper 36, the damper 36 is much stiffer than it would be if the elements 50 were merely embedded within the matrix 56 as shown in Figure 4.

Figure 7 shows an alternative embodiment of the damper 36 of Figure 6 in which the plurality of piezoelectric elements 50 form a rectangular array of two rows and six columns. Expressed another way, the rows are arranged in more than one deck, or in a double-deck configuration. According to the embodiment described herein, the array is six elements long, as between the inner and outer radial surfaces 51, 53 of the damper 36, and two elements wide or high, as between the upper and lower surfaces 57, 55 of the damper 36. Advantageously, the piezoelectric elements 50 when arranged in this stacked configuration provide a greater displacement when activated and thus, the damper 36 is able to exert a greater damping force on the input-assembly 38 and inertia ring 34. Equally, the elements 50 are also able to absorb a larger displacement of the damper 36 and harvest more electrical energy from the torsional vibrations that are transmitted along the front crankshaft 20.

As with the embodiments shown in Figures 4 and 6, the piezoelectric elements 50 are orientated within the resilient matrix 56 such that the flanges 60 of the elements 50 lie parallel with the upper and lower surfaces 57, 55 of the damper 36.

30

Likewise, as in the previous embodiment the piezoelectric elements 50 are bonded to the surface the nearest flywheel. That is, a first bonding strip 58a is located between an upper surface of an upper piezoelectric-element 50a and the inner surface of the inertia ring 34. A second bonding strip 58b is located between a lower surface of a

lower piezoelectric-element 50b and the outer surface of the radial portion 38b of the input-assembly 38. However, the two separate rows of piezoelectric elements 50 are also bonded to each other. That is, a third bonding strip 58c is located between an upper surface of the lower element 50b and a lower surface of the upper piezoelectric-
5 element 50a. Thus, the first and second bonding strip 58a, 58b mechanically couple the upper and lower piezoelectric elements 50a, 50b to the input-assembly 38 and inertia ring 34, respectively, whereas the third bonding strip 58c mechanically couples the upper and lower piezoelectric elements 50a, 50b to each other.

10 An alternative embodiment of the vibration-damping system 16 will now be described with reference to Figures 8 to 10, which show a longitudinal-sectional view of a portion of the vibration-damping system 16 in which the damper 36 is located in an axial arrangement relative to the input-assembly 38 and the inertia ring 34.

15 According to the embodiment of the invention described herein, the damper 36 has an inner-radial surface 51 and an outer-radial surface 53 which face towards the input-assembly 38 and the inertia ring 34, respectively. The damper 36 is mechanically bonded at the inner-radial surface 51 to the inertia ring 34 and at the outer-radial surface 53 to the input-assembly 38. The upper surface 57 and the lower surface 55
20 are arranged orthogonally to the inner and outer radial-surfaces 51, 53, which define an inner and outer circumferential surface of the damper 36, respectively.

The damper 36 comprises a plurality of piezoelectric elements 50 which form a rectangular array of two rows and one column. Thus, the array is one element long, as
25 between the inner and outer radial surfaces 51, 53 of the damper 36, and two elements wide or high, as between the upper and lower surfaces 57, 55 of the damper 36. The piezoelectric elements 50 are orientated within the resilient matrix 56 such that the flanges 60 lie in parallel with the inner and outer radial surfaces 51, 53 of the damper 36, i.e. in an orthogonal orientation to the piezoelectric elements 50 in the
30 embodiments described in Figures 4, 6, and 7.

Figure 9 shows an alternative embodiment of the damper 36 of Figure 8 in which each of the plurality of piezoelectric elements 50 are directly mounted to the input-assembly 38 and the inertia ring 34. For each piezoelectric element 50, a first bonding strip 58a

is located between an inner-radial surface of the element 50 and the inner surface of the inertia ring 34. A second bonding strip 58b is located between an outer-radial surface of the element 50 and an inner-radial surface of the annular portion 38b of the input-assembly 38.

5

Figure 10 shows an alternative embodiment of the damper 36 of Figure 9 in which the plurality of piezoelectric elements 50 form a rectangular array of two rows and two columns. According to the embodiment described herein, the array is two elements long, as between the upper and lower surfaces 57, 55 of the damper 36, and two elements wide or high, as between the inner and outer radial surfaces 51, 53 of the damper 36.

The operation of the damping control unit 18 will now be described in more detail with reference to Figure 11, which shows a schematic view of the damping control unit 18 in combination with the vibration-damping system 16. For clarity, reference will also be made to the vibration-damping system 16 as shown in Figures 3 and 4.

In use, the input-assembly 38 is exposed to mechanical excitation caused by the harmonic torque oscillation, associated with the combustion events of the engine 14. The cylinder pressures that occur at the peak of the combustion cycle apply large forces to accelerate a crank which is coupled to the crankshaft assembly. The inertia of the crank and the crankshaft assembly react to these forces, which causes torsional vibrations (or angular vibrations) to be transmitted along the crankshaft assembly towards the vibration-damping system 16. The magnitude and frequency of these torsional vibrations typically depends on the length and stiffness of the crank, the engine speed and the engine load.

The damping control unit 18 is operable to monitor these vibrations and control the operation of the piezoelectric elements 50 in order to dampen the vibrations thereby inhibiting their transmission along the powertrain. More specifically, the piezoelectric elements 50 are operated in a damping mode in which they are able to increase or decrease the stiffness of the damper 36 thereby causing it to dampen more or less of the torsional vibrations as they are transmitted along the crankshaft 20 to the peripheral system 17.

The damping control unit 18, as shown in Figure 11, includes a processor 62 in communication with the engine sensor 26 and the vibration sensor 28 via an amplifier 66 of the control unit 18. The control unit 18 is connected to the damper 36 of the vibration-damping system 16 via a driver of the control unit 18. The driver is shown in this embodiment as a capacitive charge pump 68. A power supply 72 in the form of a vehicle battery is connected to the damping control unit 18. The power supply 72 is arranged to supply electrical power to and from the vibration-damping system 16 through a voltage inverter 70 of the damping control unit 18.

10

The amplifier 66 is arranged to amplify the sensor signals so that they can be easily interpreted by the processor 62. The processor 62 analyses the vibration signal using a frequency analysis algorithm to determine the amplitude and frequency of the waveform associated with the detected torsional vibration. The vibration signal values are compared with a database of engine-parameter signal values and vibration signal values which are stored electronically within a memory module 64 of the damping control unit 18. The engine-parameter signal values relate to the load and speed parameters of the engine, whereas the vibration signal values include information relating to the magnitude and frequency of the detected torsional vibrations. The engine and vibration signal values form a vibration map, which is used by the processor 62 to determine a frequency and amplitude relating to an inverse waveform that will provide the most effective dampening of the detected torsional vibrations for a given engine condition. The determined frequency and amplitude values are then sent to the charge pump 68, which generates a control signal which is then provided to the vibration-damping system 16.

25

The control signal is supplied to the damper 36 to energize the piezoelectric elements 50 in such a way that they actuate to resist the deformation caused by the torsional vibrations. The piezoelectric elements 50 may be energized so as to partially or completely cancel out the waveform associated with the torsional vibrations in order so that it will either reduce or eliminate the resonant vibrations, respectively.

30

The database is provided in the form of a protocol which is both readable and addressable by the processor 62 such that it can be updated with new signal data

values over time. The processor 62 is configured to utilise the protocol in order to predict an inverse waveform that is required to dampen the torsional vibration of the crank shaft in dependence on the detected engine speed and load.

5 It has been described that one of the benefits of vibration-damping system 16 is that it can be operated to actively dampen the torsional vibrations that are transmitted through the powertrain. However, the damping system 16 may also be operated in a sensing mode whereby the forces exerted on the piezoelectric elements 50 are converted into vibration signals which are sent to the damping control unit 18. In use,
10 the vibrations that are transmitted along the crankshaft to the input assembly 38 result in the generation of charges on the surface of piezoelectric elements 50 in the damper 36. The electric charges are transmitted to the damping control unit 18 in the form of vibration signals. In this way, the piezoelectric elements 50 are able to harness the direct piezoelectric effect to convert mechanical energy into electrical energy. Hence,
15 the piezoelectric elements 50 operate in the sensing mode to detect the torsional vibrations and send corresponding vibration signals to the damping control unit 18.

The damping control unit 18 determines the control signal from the control map of the protocol in dependence on the received engine and vibration signals. The control
20 signal is then applied to the piezoelectric elements 50 inside the damper 36. As the control signal is applied across the piezoelectric elements 50, the dimension of the elements 50 change thereby adjusting the damping coefficient of the damper 36. In this way the damper 36 is stiffened in phase with the frequency and amplitude of the vibration to counter the unwanted torque oscillation. Put another way, the damper 36 is
25 effectively configured to produce an anti-torque that is equal in frequency and amplitude to the torque associated with the torsional vibration but is opposite in direction so as to cancel out the vibration. By addressing the piezoelectric elements 50 in this way, the damping control unit 18 is able to configure the physical parameters of the damper 36 in order to adapt to the changing frequency of the torsional vibrations.

30

Thus, when operating in the damping mode, the damping control unit 18 is configured to send a damping control signal to control the elements 50 in dependence on the sensor signals it receives from the damper sensor 28 and the engine sensor 26.

The piezoelectric elements 50 are also configurable to operate in a generator mode in which the elements 50 are allowed to be driven by the torsional vibrations so that they deform, thereby converting the kinetic energy of the vibrations to electrical energy which is then transmitted to the damping control unit 18. In the generating mode, the piezoelectric elements 50 are allowed to oscillate naturally with the vibrations of the crankshaft. The resulting displacements cause electrical charge to be generated according to the piezoelectric effect, as would be readily understood by a person skilled in the art. The electrical charge, in the form of a voltage signal, is then transferred to the battery 72 via the voltage inverter 70 of the control unit 18. The damping control unit 18 sends a generating control signal to control the elements 50 in dependence on the sensor signals it receives from the damper sensor 28 and the engine sensor 26.

The damping control unit 18 is configured to determine whether to operate the damper 36 in either the sensing, damping or generating mode, or in any combination thereof, in dependence on the sensor signals it receives from the damper sensor 28 and the engine sensor 26. In other words, the control unit 18 receives vibration signals from the piezoelectric elements 50 and engine-parameter signals from the engine sensor 26, and determines an appropriate control signal in accordance with the predetermined protocol.

During normal operation of the engine 14, the torsional vibrations cause the crankshaft assembly to twist which can cause undesirable fatigue loading however, not all torsional vibrations are detrimental to the crankshaft assembly. Fatigue loading only occurs at specific vibration frequencies. Conventional damping systems are unable to differentiate between the different vibration frequencies such that some vibrations are damped unnecessarily whilst fatigue load vibrations go undamped. Advantageously, the vibration-damping system 16 according to the embodiment of the invention as described herein is able to operate the piezoelectric elements 50 in a sensing mode and a damping mode so as to accurately counteract only the vibrations which may result in the undesirable fatigue loading of the powertrain components. Moreover, the piezoelectric elements 50 may be operable according to the generating mode in order to extract energy from the damper 36. This is in contrast to a conventional damper, with a

fixed coefficient of elasticity which absorbs energy regardless of the operating parameters of the engine.

Advantageously, the piezoelectric elements 50 are configurable to rapidly detect
5 changes in the frequency of the vibrations and to detect vibrations across a very broad range of frequencies. The piezoelectric elements 50 are also configurable to deform at a wide range of frequencies and to rapidly change the frequency at which they actuate. The level or damping, in both amplitude and frequency, is proportional to the sensed vibration.

10

Thus, the piezoelectric elements 50 enable the damping control unit 18 to detect rapid changes in the frequency of the vibrations of a wide frequency range. The elements 50 are also able to rapidly change the frequency at which they actuate, over a wide frequency range. Consequently, the piezoelectric elements 50 are able to match any
15 rapid changes in frequency of the torsional vibrations. In other words, the damper 36 is able to make near instantaneous changes to the damping strategy in order to counteract the torsional vibrations. By reducing the effect of the torsional vibration as it varies with engine speed and load, the vibration-control system 16 allows for lighter and less stiff powertrain components to be used on the vehicle. This allows for a
20 significant reduction in both the stationary and movable mass of the vehicle.

Advantageously, the vibration control system 16 is able to damp specific torsional vibration frequencies completely or partially. The damper 36 is also able to simultaneously convert the kinetic energy associated with the torsional vibrations into
25 electrical energy.

The damper 36 has been described, above, as being configured to control the dampening force that is applied to the input-assembly 38 due to the counter rotation of the inertia ring 34. With reference to Figure 12, an alternative embodiment of the
30 invention will now be described in which a damper 136 provides a mechanical coupling between an input-assembly 138 and an output-assembly 140 of the vibration-damping system 16.

The input-assembly 138 is directly coupled to the front crankshaft 20 of the powertrain 12, the output-assembly 140 is coupled to the peripheral system (not shown) and the damper 136 forms a direct coupling between the input and output assemblies 138, 140. More precisely, the damper 136 forms an annular belt (or ring) which is arranged
5 between the input and output flywheels 138, 140 and extends substantially across an annular portion 138b, 140b of the input and output assemblies 138, 140. The damper 136 is mechanically bonded at an inner surface 155 to the annular portion 138b of the input-assembly 138 and at an outer surface 157 to the annular portion 138b of the output assembly 134. The damper 136 also has an inner-radial surface and an outer-
10 radial surface, which are arranged orthogonally to the inner and outer surfaces 155, 157. The damper 136 has a composite structure as it is formed of two main components, which include a plurality of piezoelectric elements (not shown) that are associated with a resilient matrix 156. The piezoelectric elements are embedded within the resilient matrix 156 of the damper 136 in a similar manner to that which was
15 described in the earlier embodiments of the invention. In this configuration, the damper 136 is operable to directly dampen the torsional vibrations that are transmitted from the input-assembly 138 to the output-assembly 140.

In embodiments, the vibration damping system 16 may be coupled to the rear
20 crankshaft 21 of the vehicle 10, which is arranged to transmit power from the engine 14 to other components of the vehicle powertrain, such as the rear axle, for example. In this way, the vibration damping system 16 may be arranged to dampen vibrations that are transmitted from the engine 14 to components of the vehicle that are used to drive the vehicle 10.

25 In embodiments the vibration-control unit 18 is provided as electronic data stored on a non-volatile memory component of a computer or logic system embedded within an external control unit. In embodiments the engine-parameter signals are received from an engine control unit of the engine rather than a separate engine sensor 28.

30 In embodiments, the sensed signals are stored within a non-volatile storage device from which they can be downloaded to provide a vehicle user profile for analysis or to be used in conjunction with information sourced from the vehicle ECU to predict the damping requirements for a number of different vehicle conditions.

In embodiments, the control signals may be generated without the vibration sensor inputs. In this case, the control signals would be determined solely from a pre-determined protocol comprising engine-parameter signal values that have been pre-
5 recorded, for example, using a development engine.

Many modifications may be made to the above examples without departing from the scope of the present invention as defined in the accompanying claims.

10

CLAIMS

1. A vibration damping system for use in a vehicle powertrain, comprising:
a first rotatable component;
5 a second rotatable component; and,
a damper coupled to at least one of, or between, the first component and the second component; the damper comprising at least one piezoelectric element responsive to vibrations of the first component and/or the second component, wherein the damper comprises a resilient matrix, the resilient matrix being associated with the
10 at least one piezoelectric element;
wherein the second rotatable component is an inertia mass or an output component of the powertrain configured to transmit drive from an engine or gearbox of the vehicle.
- 15 2. The system of Claim 1, wherein the at least one piezoelectric element is embedded within the resilient matrix.
3. The system of Claim 1 or Claim 2, wherein at least one of the piezoelectric elements is bonded directly to at least one of the first or second rotatable components.
20
4. The system of any one of the preceding claims, wherein the piezoelectric element extends about the first rotatable component.
5. The system of Claim 4, including a plurality of piezoelectric elements.
25
6. The system of Claim 5, wherein the plurality of piezoelectric elements are arranged to extend circumferentially about the first rotatable component.
7. The system of Claim 6, wherein the plurality of piezoelectric elements are
30 arranged in a plurality of rows, wherein the rows are circumferentially spaced about the first rotatable component.
8. The system of Claim 6, wherein the rows are arranged in more than one deck.

9. The system of any one of the preceding claims, wherein the at least one piezoelectric element is substantially H-shaped in cross section.
10. The system of Claim 6, further comprising a control system operatively coupled to the at least one piezoelectric element and configured to control the operation of the system in at least one of a damping mode or a generating mode.
11. The system of Claim 10, wherein the control system, in the damping mode, is operable to apply a control signal to the at least one piezoelectric element to attenuate vibrations of the first or second rotatable member.
12. The system of Claim 11, wherein the control system is configured to apply the control signal to the at least one piezoelectric element in dependence on receiving an input signal indicative of a level of vibration of the first or second rotatable component.
13. The system of Claim 12, wherein the input signal is provided by the at least one piezoelectric element.
14. The system of Claim 6, wherein the control system, in the generating mode, is operable to harvest electrical energy from a voltage signal generated by the at least one piezoelectric element.
15. The system of any one of preceding claims, wherein the first component is an output component of the powertrain, the output component configured to drive a peripheral system of the vehicle.
16. The system of Claim 15, wherein the peripheral system is at least one of an alternator, a fluid pump or a compressor for an air conditioning unit of the vehicle.
17. A vehicle powertrain comprising the vibration damping system of any one of claims 1 to 16.
18. A vehicle comprising the vehicle powertrain of Claim 17.

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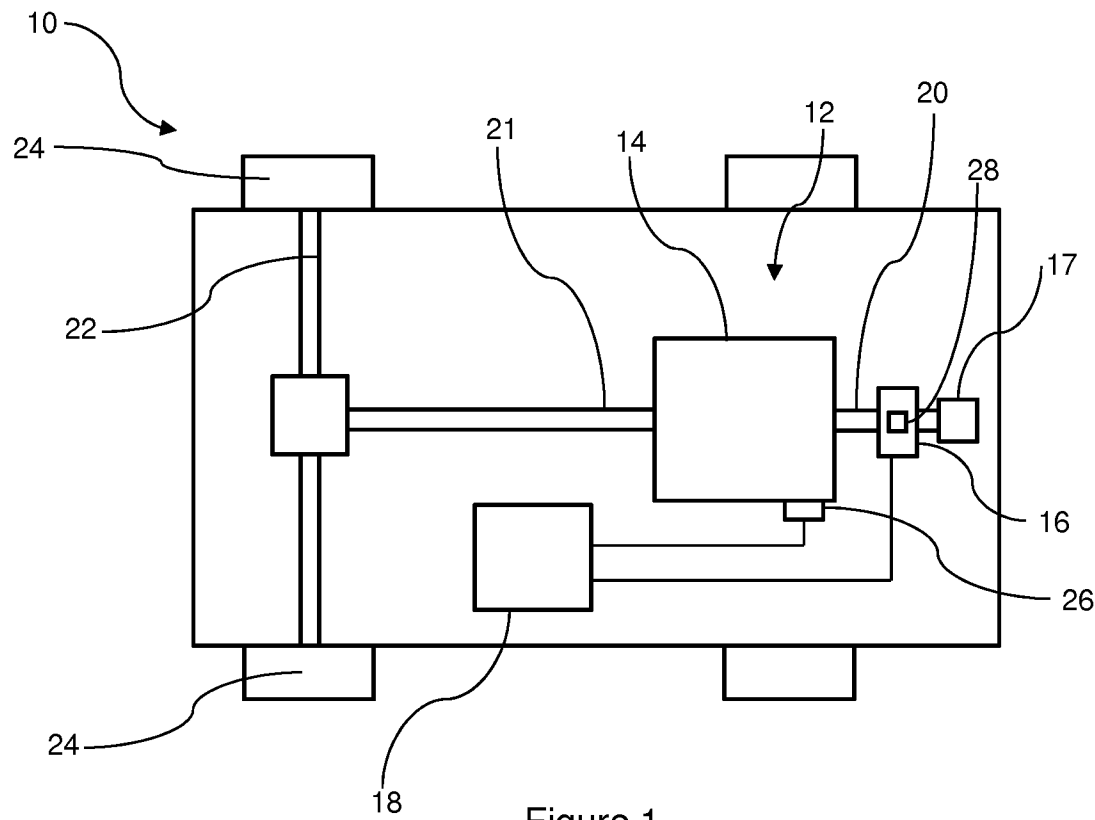


Figure 1

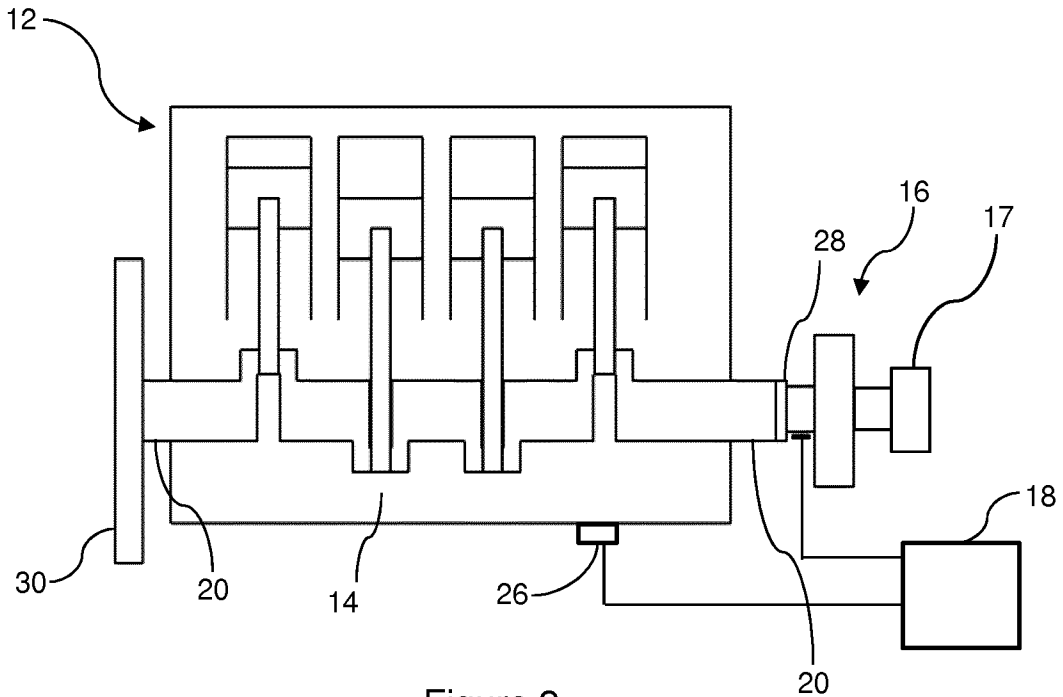


Figure 2

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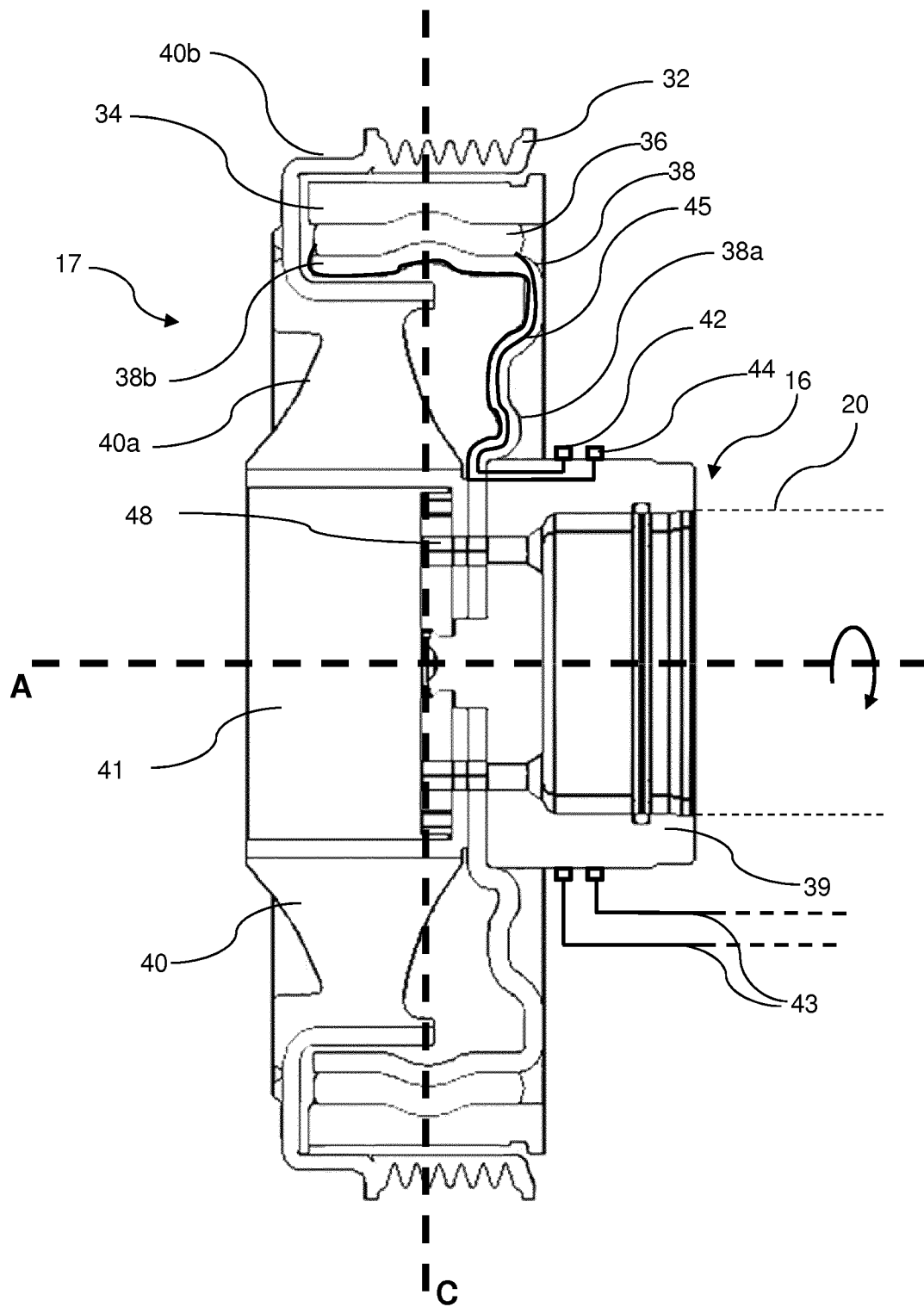


Figure 3

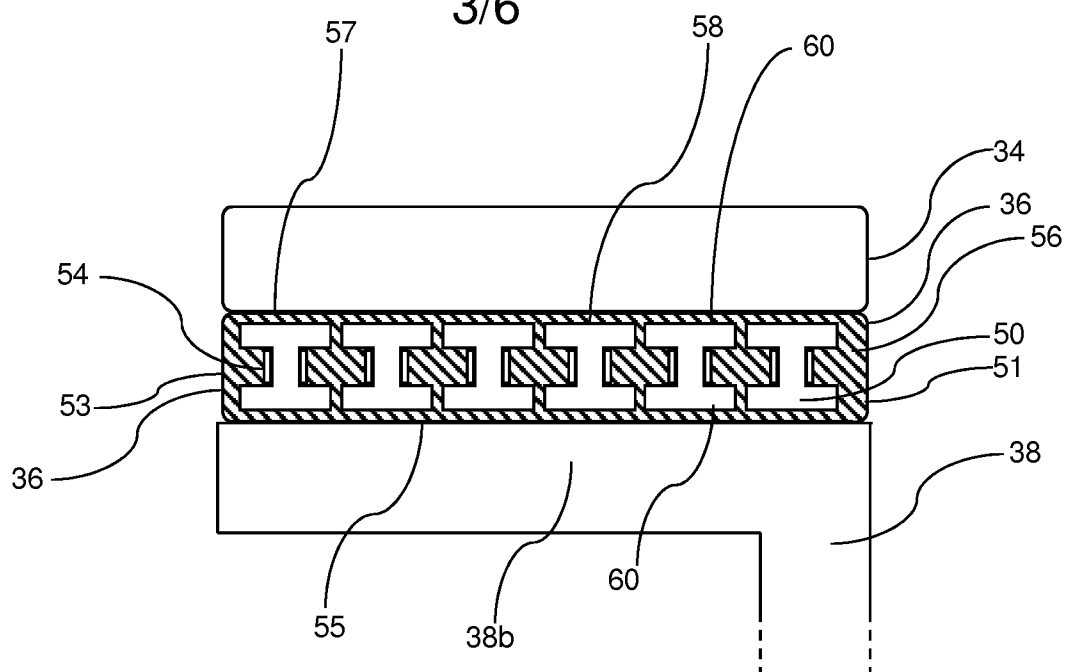


Figure 4

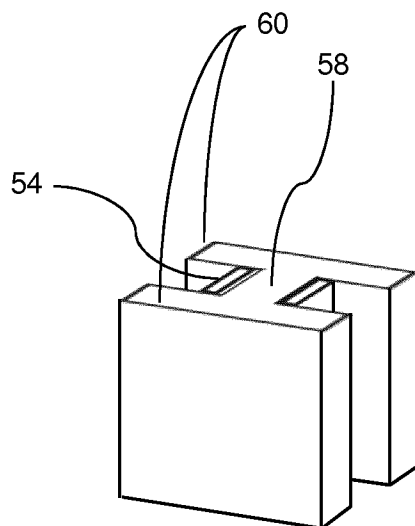


Figure 5

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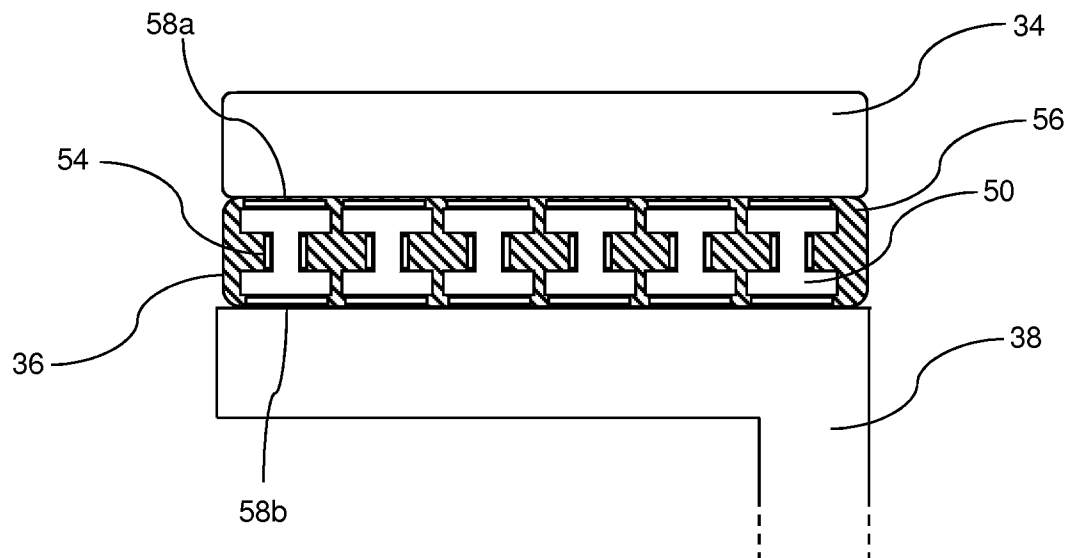


Figure 6

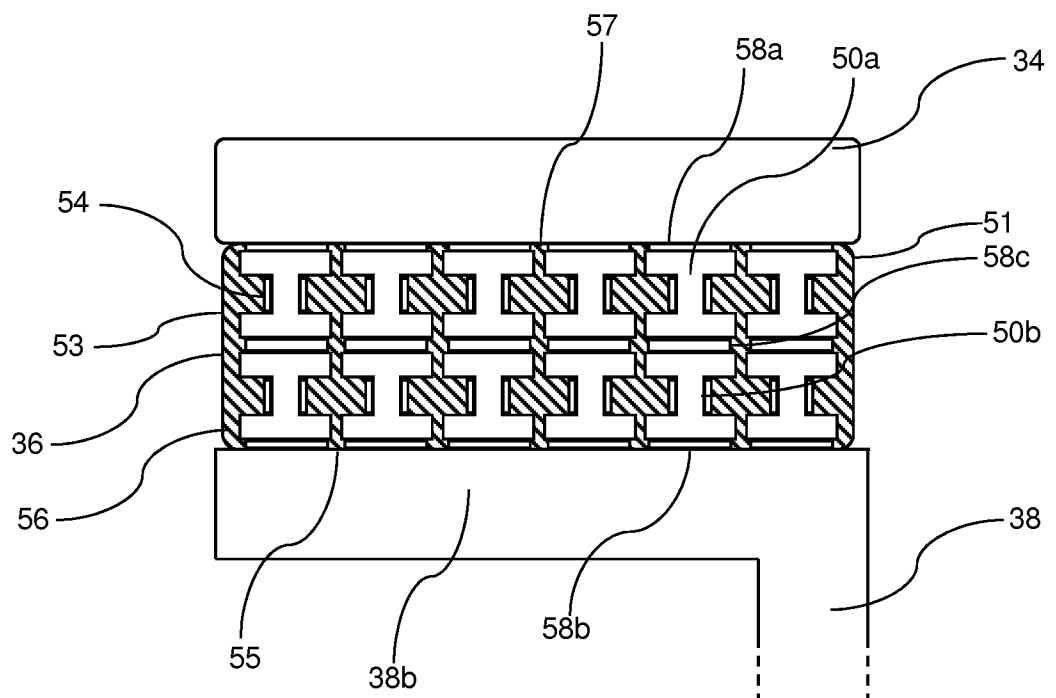


Figure 7

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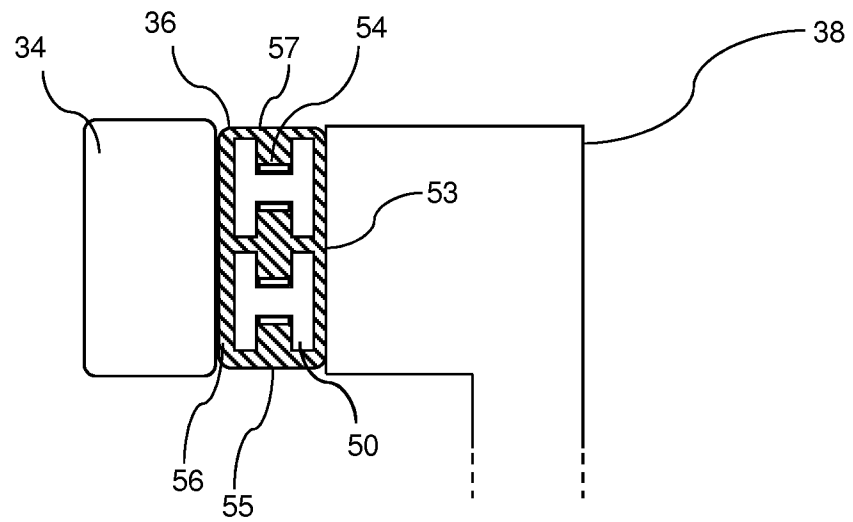


Figure 8

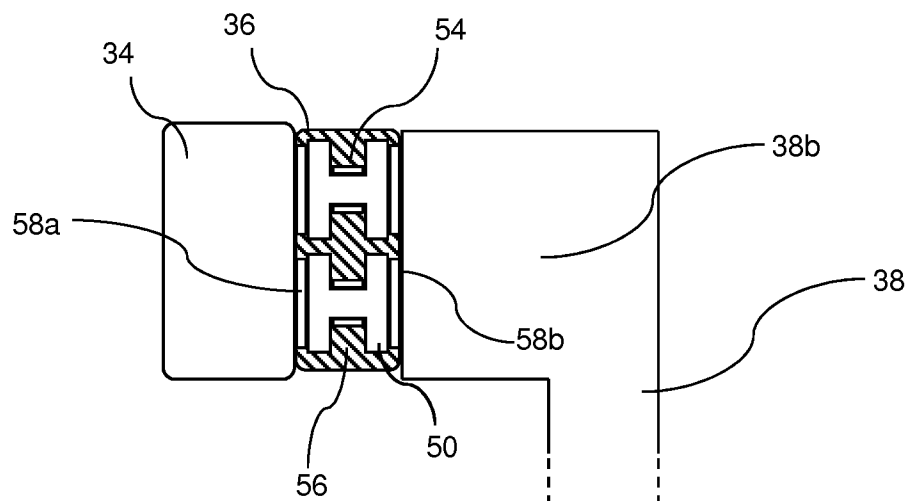


Figure 9

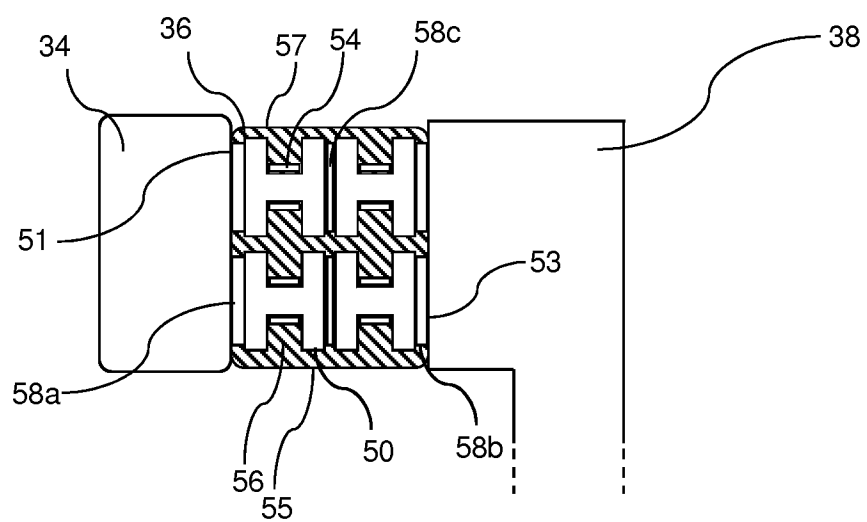


Figure 10

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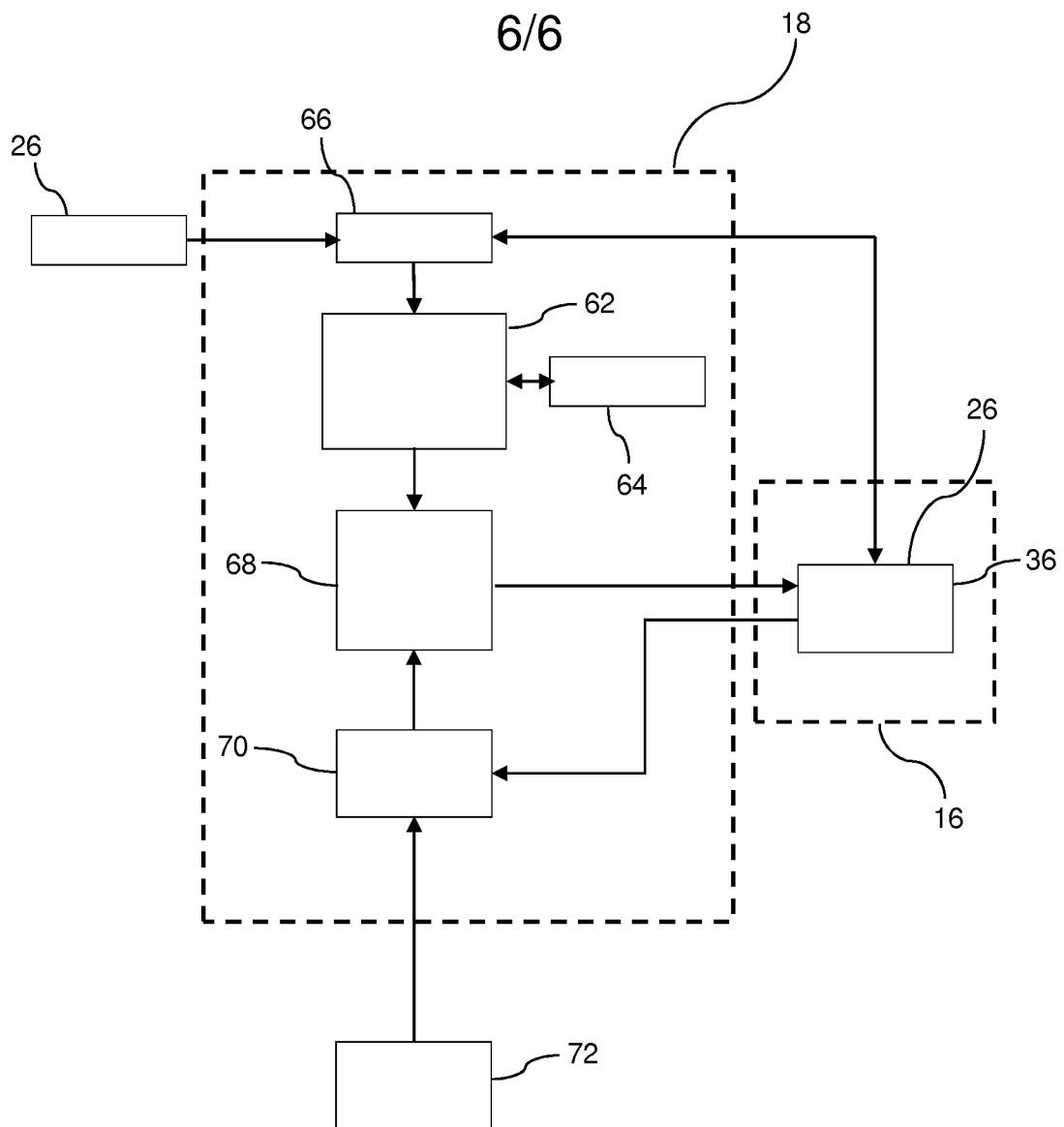


Figure 11

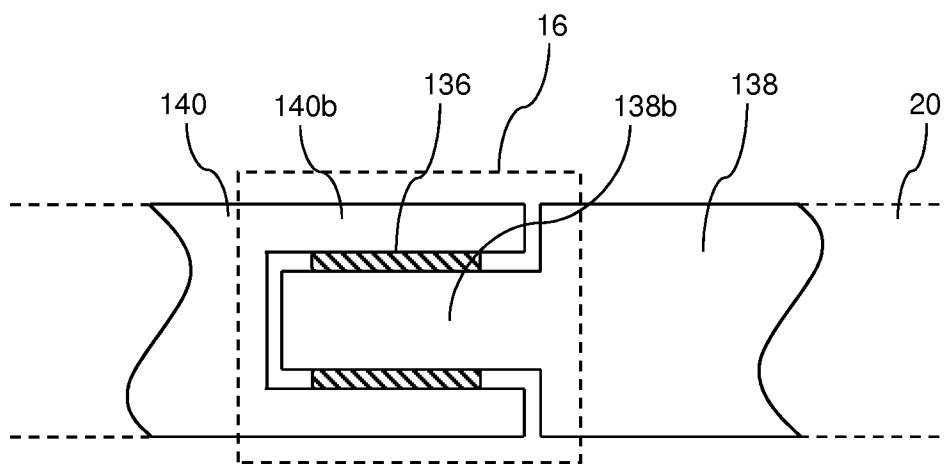


Figure 12

INTERNATIONAL SEARCH REPORT

International application No

PCT/EP2017/077974

A. CLASSIFICATION OF SUBJECT MATTER

INV. F16F15/00 F16F15/126 F16F15/14
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)

F16F

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, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y A	US 2011/159968 A1 (EXNER JOCHEN [DE]) 30 June 2011 (2011-06-30) paragraph [0001] - paragraph [0047] paragraph [0054] - paragraph [0055] figures 3,5	1,2,9, 17,18 13 3-8, 10-12, 14-16
X Y A	----- EP 1 146 248 A1 (FERRARI SPA [IT]) 17 October 2001 (2001-10-17) column 1 - column 4 figures 1,2	1,2,4-8, 10-12 13 3,9, 14-18
X A	----- WO 2015/103820 A1 (UNIV TIANJIN [CN]; LI JIAJUN [CN]) 16 July 2015 (2015-07-16) abstract figure 1 ----- -/-	1,3-6, 10,14-16 2,7-9, 11-13, 17,18



Further documents are listed in the continuation of Box C.



See patent family annex.

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"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

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"&" document member of the same patent family

Date of the actual completion of the international search

8 February 2018

Date of mailing of the international search report

16/02/2018

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Authorized officer

Maroño Martínez, J

INTERNATIONAL SEARCH REPORT

International application No

PCT/EP2017/077974

C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 6 012 333 A (URUSHIYAMA YUTA [JP] ET AL) 11 January 2000 (2000-01-11) column 2 - column 4 figures 1-8 -----	1-18
A	GB 2 454 859 A (IWIS MOTORSYSTEME GMBH & CO KG [DE]) 27 May 2009 (2009-05-27) abstract figures 1a-1d,2,4 -----	1-18
A	US 2004/069092 A1 (SCHANKIN DAVID PAUL [US] ET AL) 15 April 2004 (2004-04-15) abstract figures 1-4 -----	10-12
A	GB 814 123 A (POST OFFICE) 27 May 1959 (1959-05-27) figures 1-1b -----	9

INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No

PCT/EP2017/077974

Patent document cited in search report	Publication date	Patent family member(s)	Publication date
US 2011159968	A1	30-06-2011	CN 102016338 A 13-04-2011
			DE 112008003984 A5 26-05-2011
			EP 2153079 A1 17-02-2010
			EP 2623813 A1 07-08-2013
			ES 2464091 T3 30-05-2014
			US 2011159968 A1 30-06-2011
			WO 2009149676 A1 17-12-2009
EP 1146248	A1	17-10-2001	AT 364804 T 15-07-2007
			DE 60128851 T2 07-02-2008
			EP 1146248 A1 17-10-2001
			IT B020000219 A1 15-10-2001
WO 2015103820	A1	16-07-2015	CN 103715942 A 09-04-2014
			WO 2015103820 A1 16-07-2015
US 6012333	A	11-01-2000	JP H10309951 A 24-11-1998
			US 6012333 A 11-01-2000
GB 2454859	A	27-05-2009	CN 101568745 A 28-10-2009
			DE 102006041417 B3 03-04-2008
			GB 2454859 A 27-05-2009
			JP 5484055 B2 07-05-2014
			JP 2010502878 A 28-01-2010
			US 2009293823 A1 03-12-2009
			WO 2008028541 A1 13-03-2008
US 2004069092	A1	15-04-2004	EP 1418365 A2 12-05-2004
			US 2004069092 A1 15-04-2004
			US 2005051402 A1 10-03-2005
GB 814123	A	27-05-1959	NONE