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US 20160294272 A1 **US 20080136562 A1**

(58) Field of Search:
INT CL **H02K, H02N**
Other: **WPI, EPODOC**

(54) Title of the Invention: **Vibrational energy harvester**
Abstract Title: **Vibrational energy harvester**

(57) A vibrational energy harvester 100 comprises a conductive coil 140 fixed to a first flexure assembly 130, and a magnet fixed 170 to a second flexure assembly 150. The first flexure assembly has a first resonant frequency in which the coil oscillates relative to the magnet back and forth along a first direction, and the second flexure assembly has a second resonant frequency different from the first resonant frequency in which the magnet oscillates relative to the coil back and forth along the first direction so that relative movement of the coil and the magnet induces an electrical current in the coil. A displacement limiter limits the displacement of the coil relative to the magnet in the first direction. The displacement limiter may comprise a cushioning member and a stop surface, wherein the cushioning member contacts the stop surface when the relative displacement of the coil and magnet reaches a predetermined maximum. One of the cushioning member and stop surface may be provided on the first flexure assembly and the other on the second flexure assembly. The cushioning member may absorb and dissipate kinetic energy from the flexure assemblies. The cushioning member may comprise an O-ring.

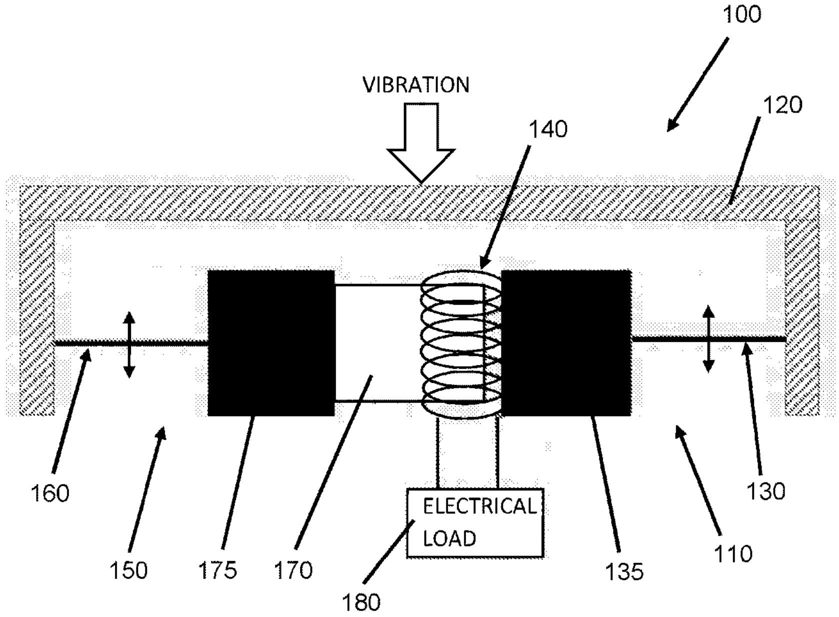


Figure 2

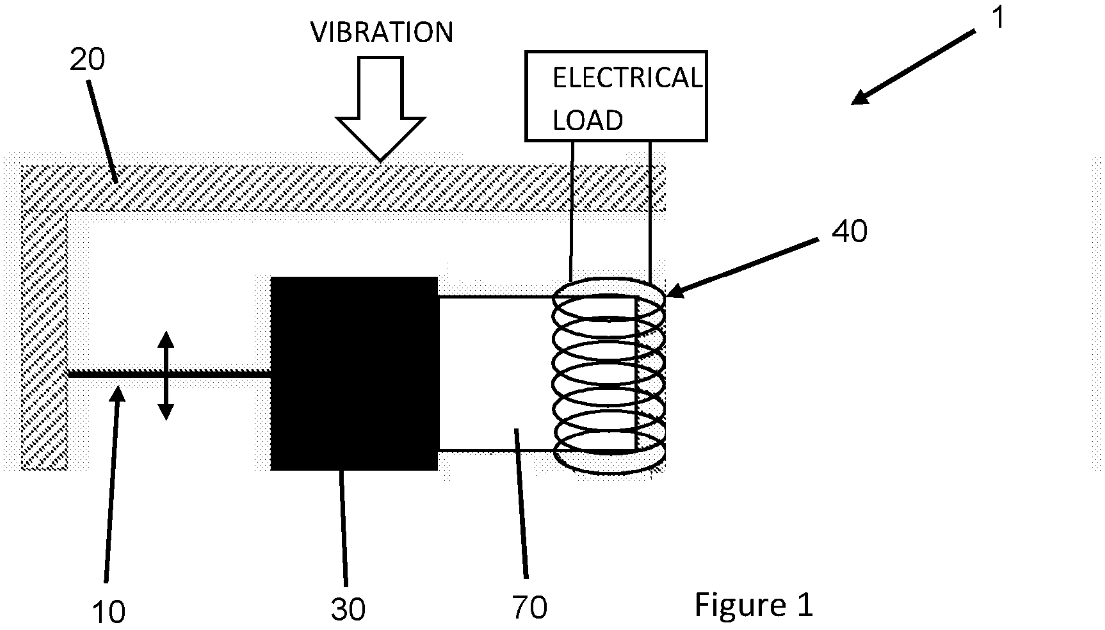


Figure 1

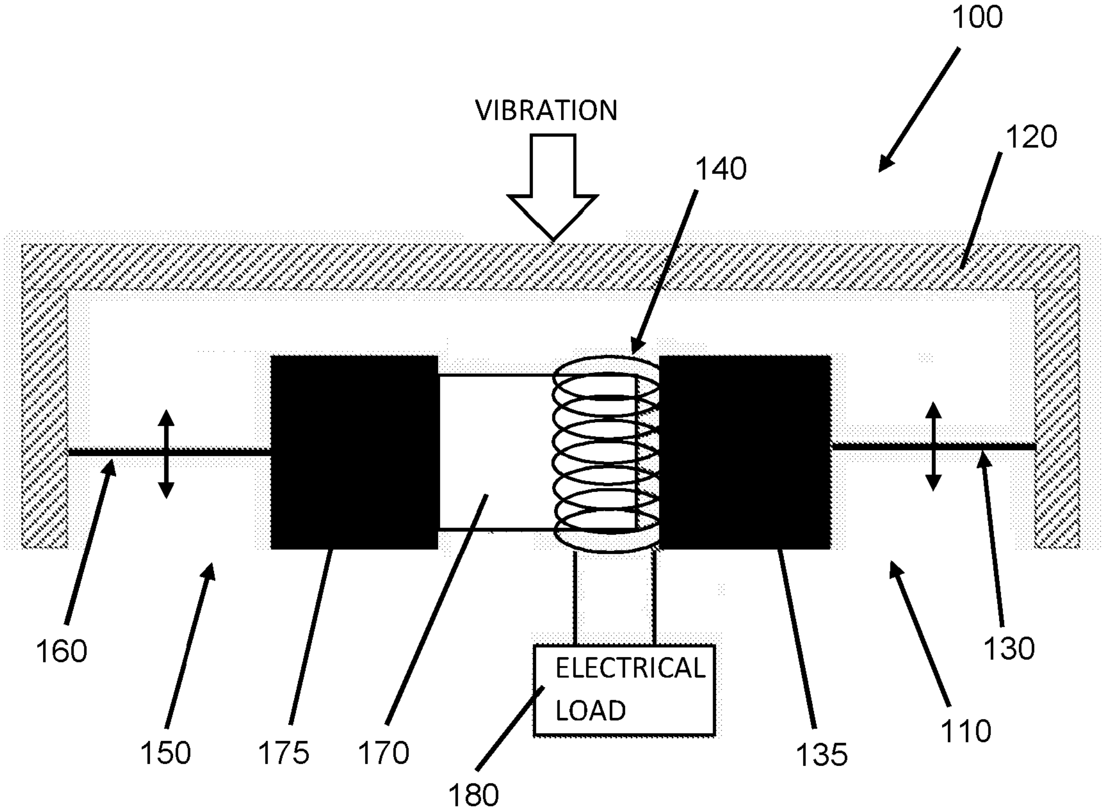


Figure 2

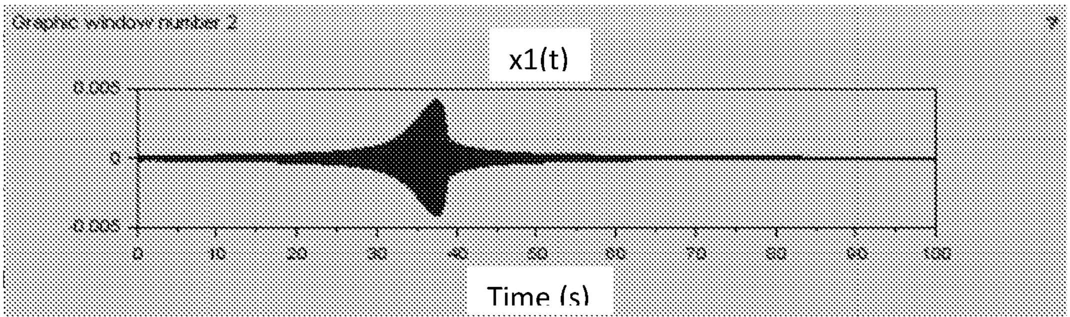


Figure 3a

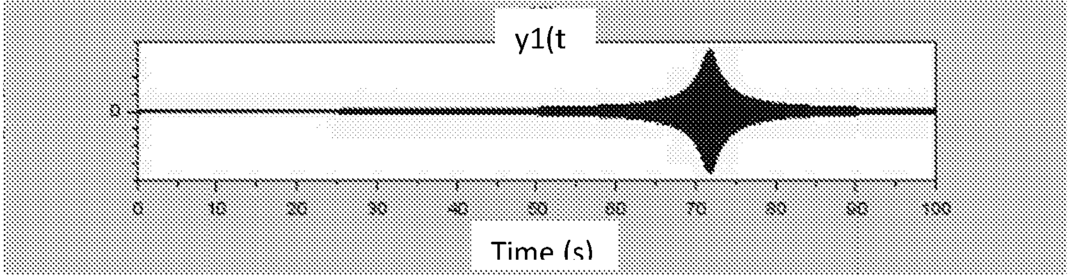


Figure 3b

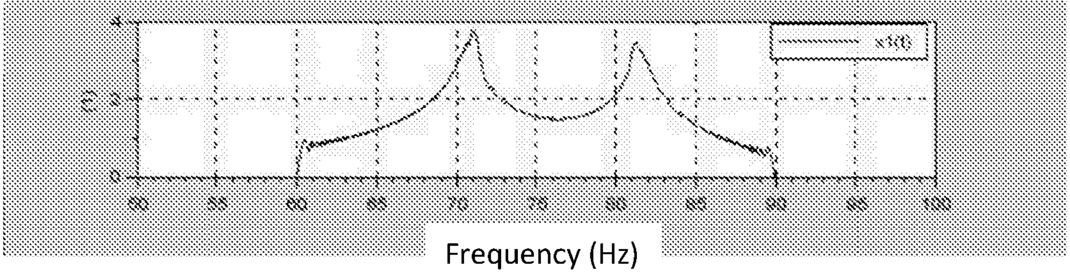


Figure 3c

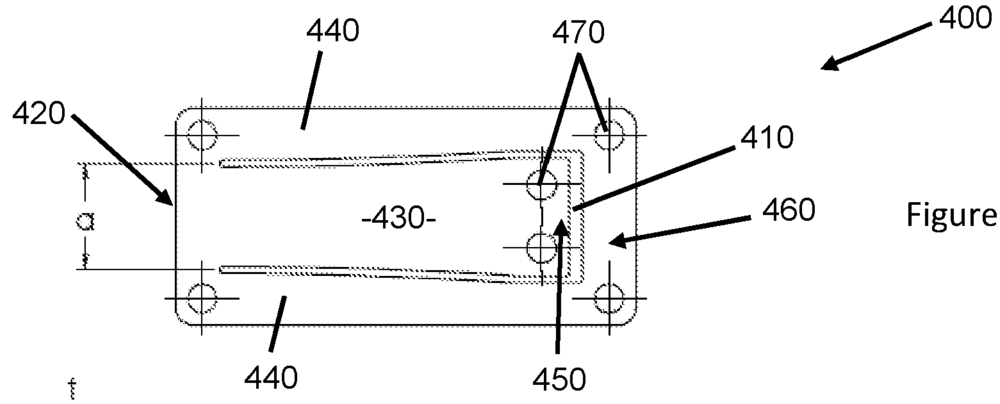


Figure 4a

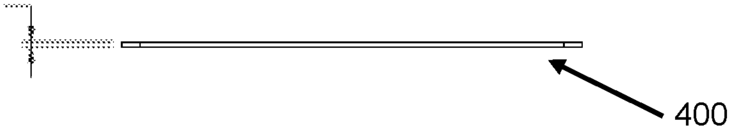


Figure 4b

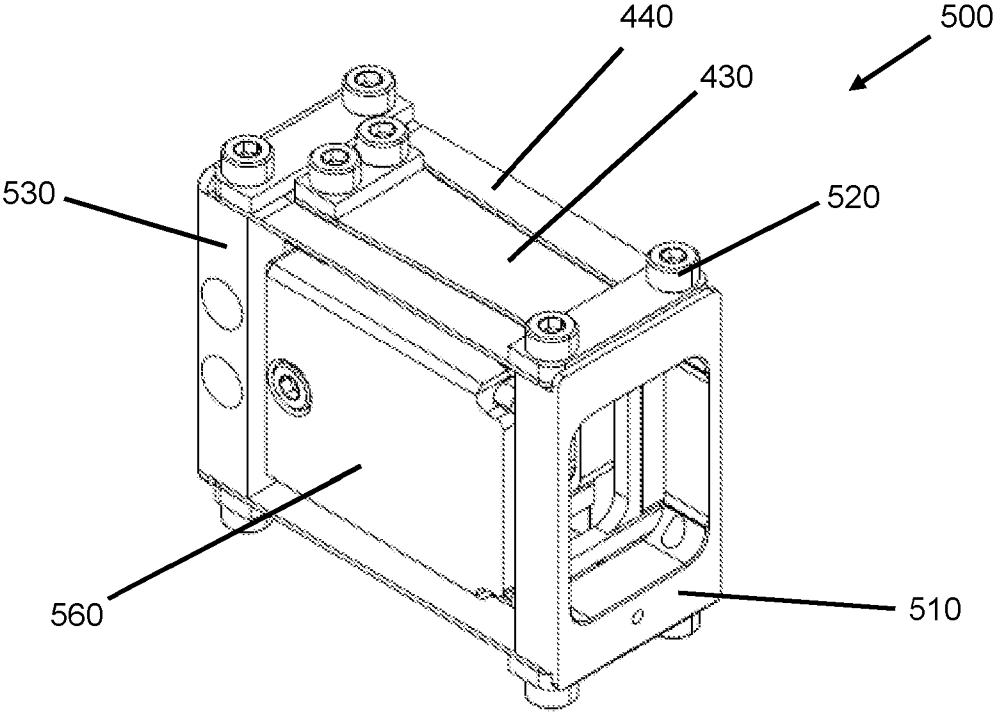


Figure 5a

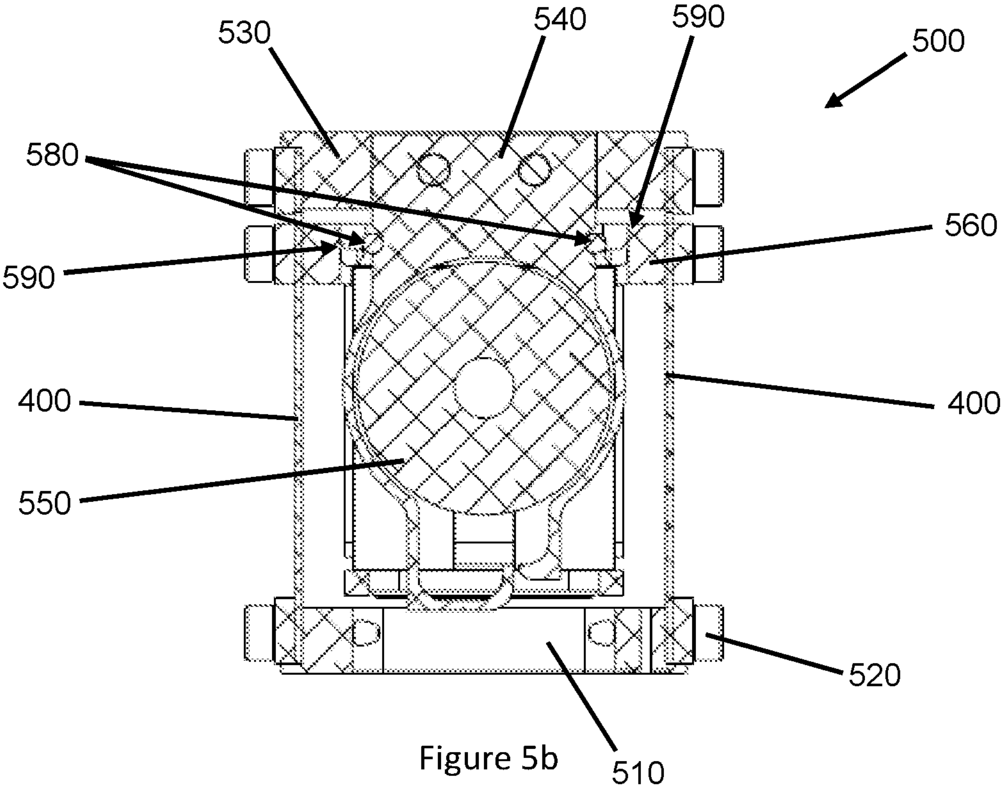


Figure 5b

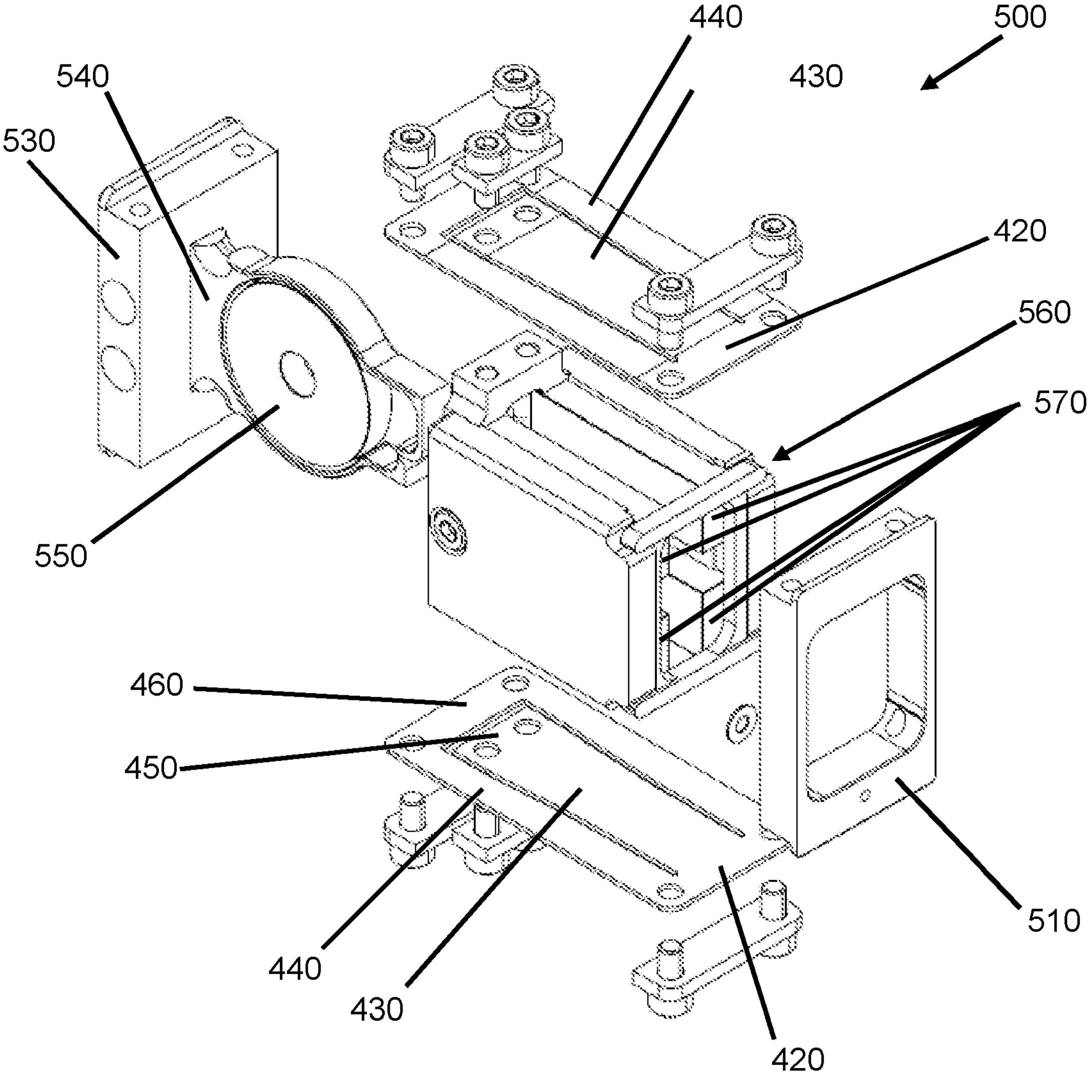
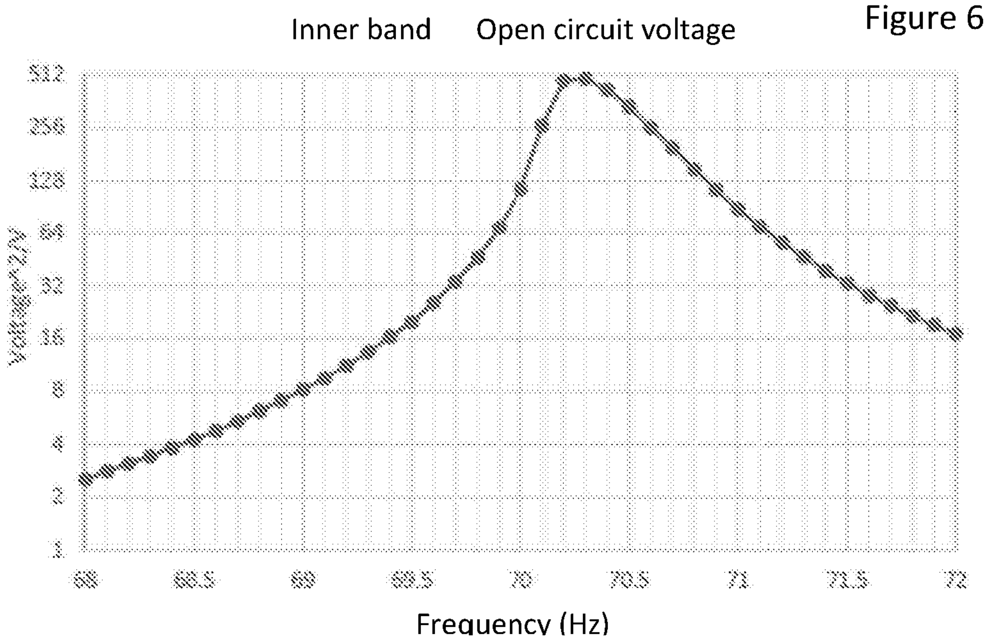
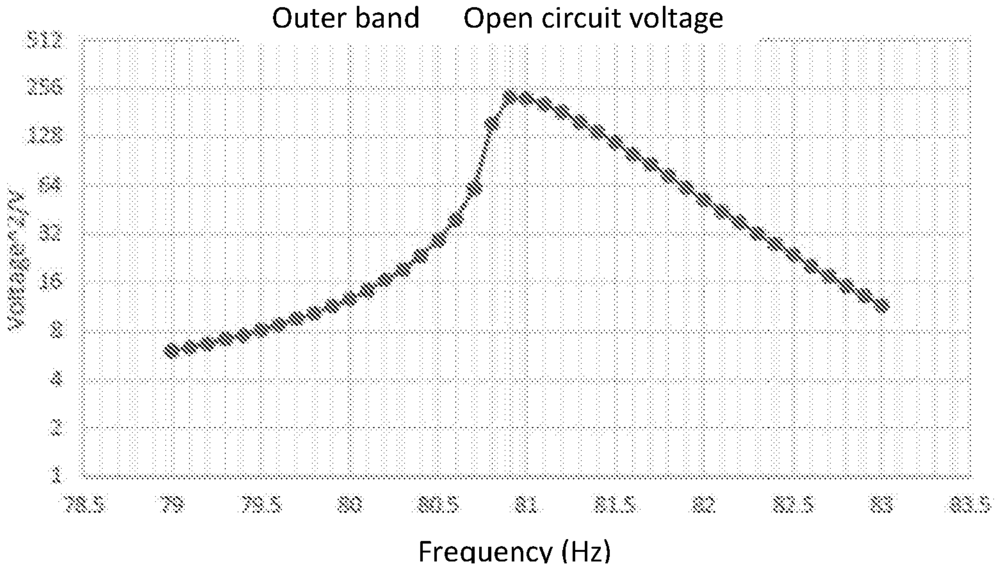


Figure 5c

Figure 6a





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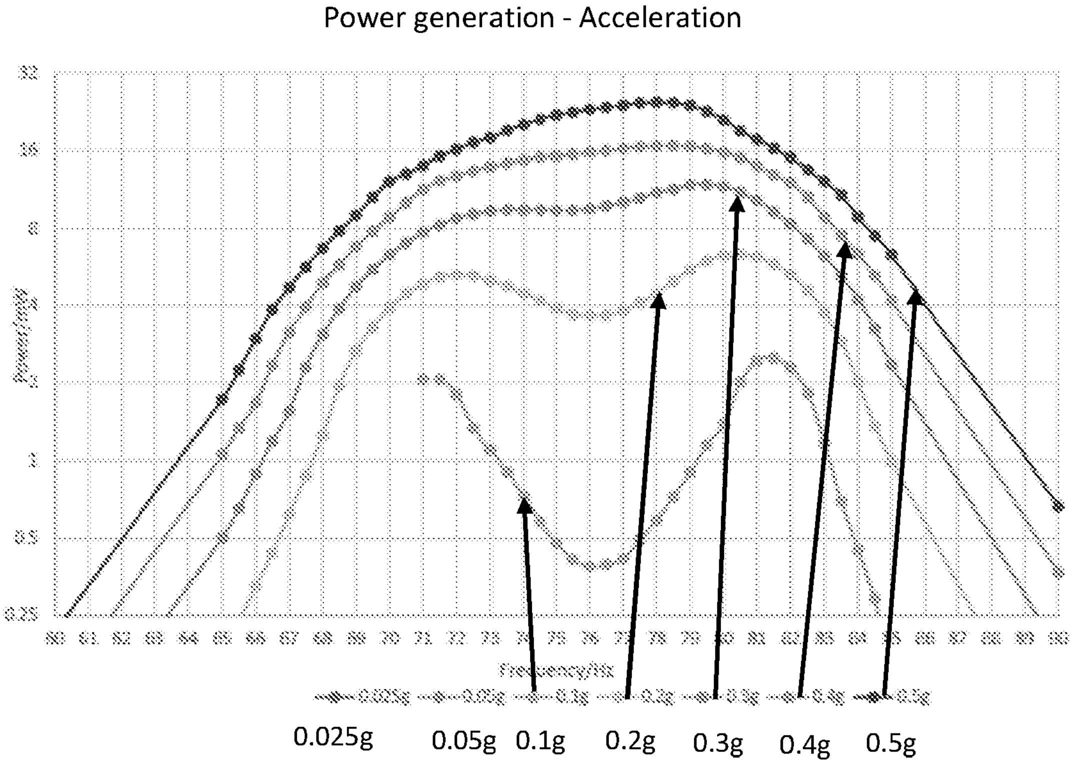

Figure 6c

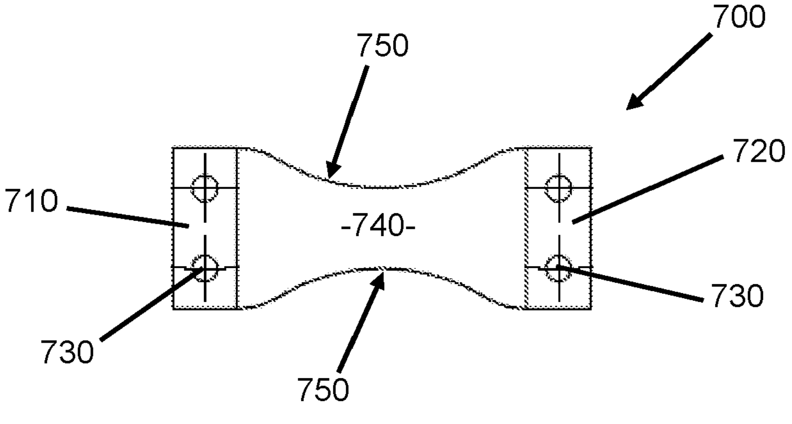

Figure 7a

Figure 7b

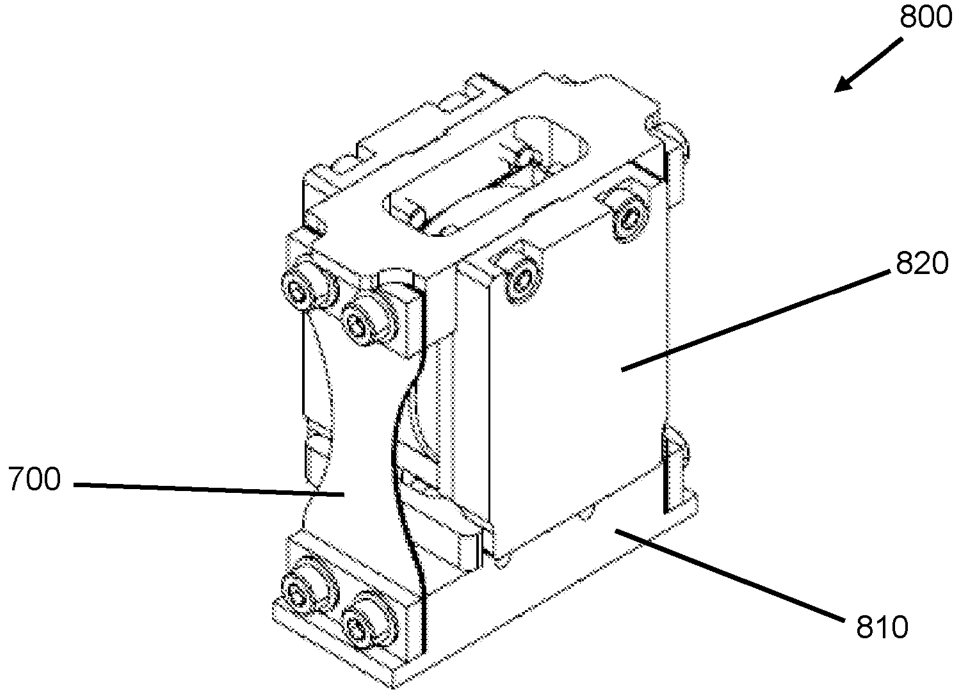

Figure 8a

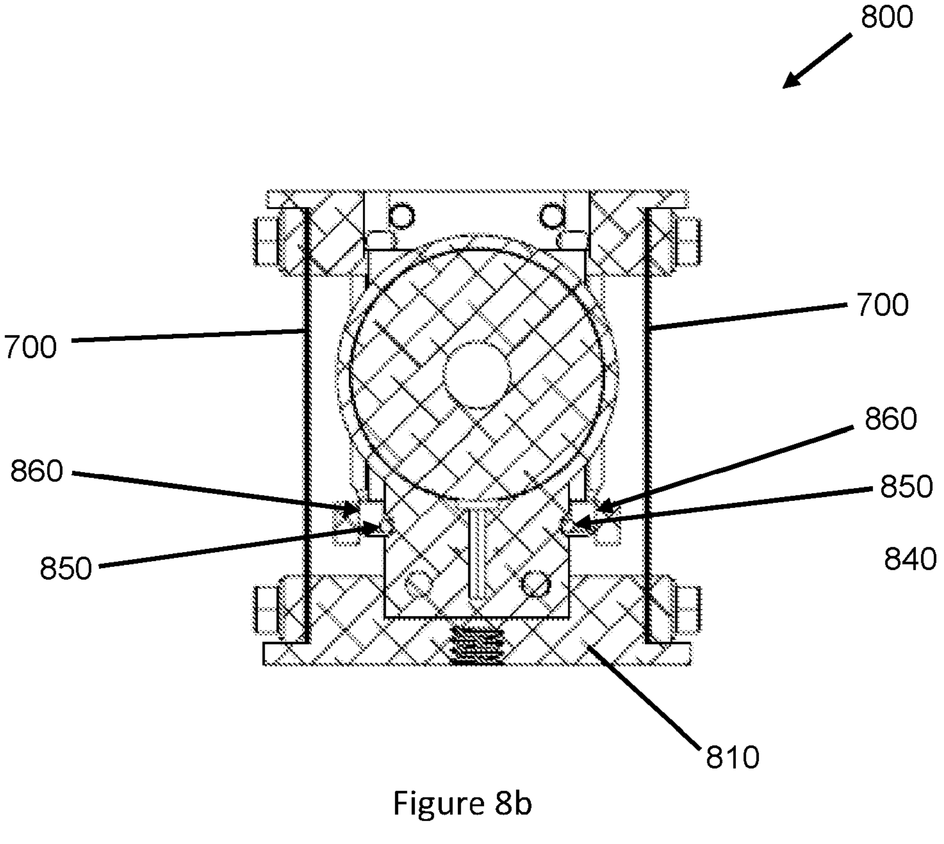

Figure 8b

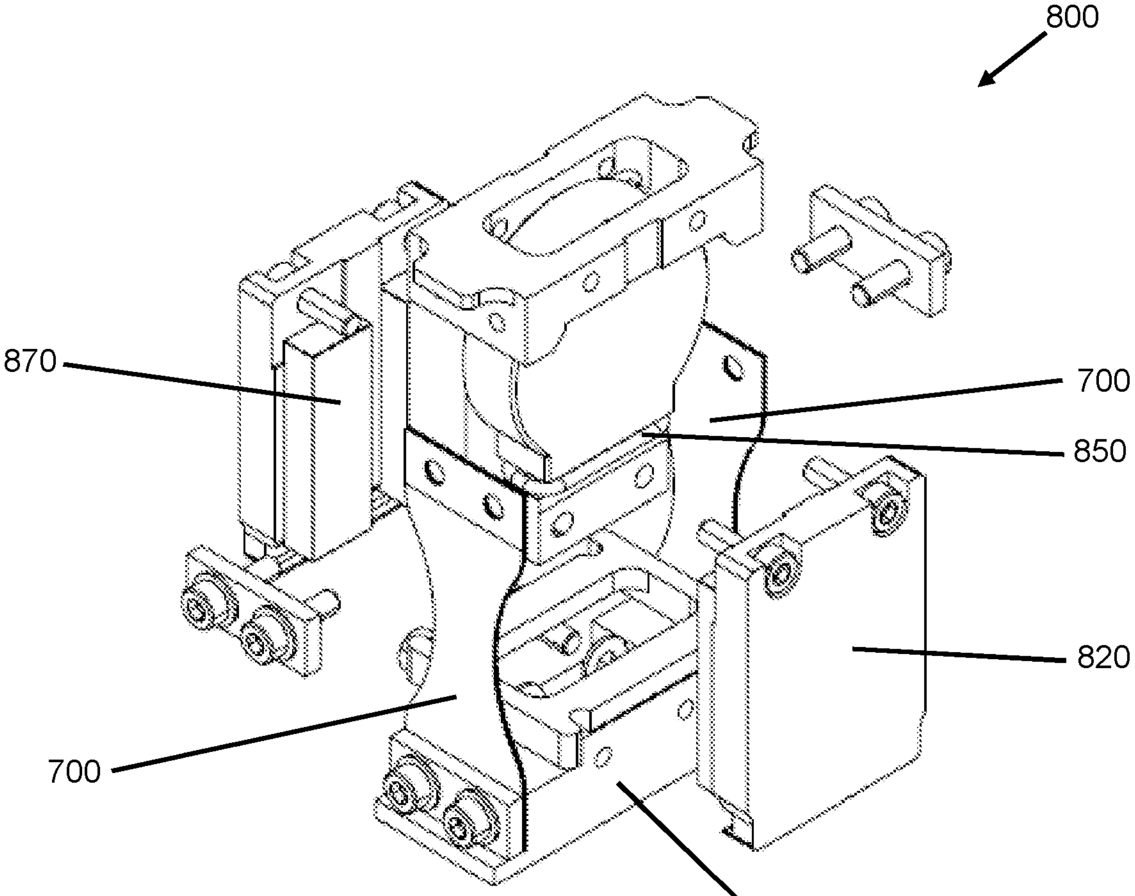

Figure 8c

Vibrational Energy Harvester

The invention relates to a vibrational energy harvester. Preferably the invention relates to a vibrational energy harvester for harvesting or collecting energy from a source of vibration, such as ambient or environmental vibration.

In a conventional energy harvester, a direct resonator, or in some cases a plurality of direct resonators, is responsive to an input vibration, such as vibration of a structure to which the energy harvester is attached. (In the art, direct resonators may also be termed linear, or ordinary, resonators.) The vibration excites the resonator(s) and the resonator(s) are electrically damped, for example by means of a permanent magnet carried by a resonator so that it oscillates in the proximity of a conducting coil, to extract an electrical power output. Such energy harvesters can be used to charge a battery or to operate an electronic device such as a sensor and/or a wireless transmitter in a self-contained device, in a known manner.

Such conventional energy harvesters suffer from several problems which limit their efficacy for converting vibration energy into electrical energy. One particular problem which limits the performance of the resonator(s) is that the resonator in a conventional energy harvester has a specific resonant frequency and can only be effectively excited by vibration frequencies close to that resonant frequency. Away from the resonant frequency the power output drops significantly. Natural or ambient vibrations available for driving an energy harvester tend to contain a variety or spectrum of vibration frequencies and a direct resonator may only be excited by a narrow band of the available vibration frequencies close to the resonant frequency of the resonator.

One approach which has been used to address this is to incorporate into an energy harvester a plurality of direct resonators of different resonant frequencies, but this adds to the complexity of the energy harvester. The prior art harvester disclosed in US Patent No. US 6,858,970, for example, comprises multiple cantilever beam resonators, each resonator comprising a piezoelectric layer. As the cantilever beam vibrates, mechanical strain is generated with the piezoelectric material, which is converted to an electrical potential difference which may be harvested. Providing multiple resonators with different resonant frequencies allows electrical energy to be generated at a range of frequencies.

It would be desirable to provide a reliable and compact vibrational energy harvester that has a significant power output over a wide band of vibration frequencies.

Statement of Invention

The invention provides a vibrational energy harvester as defined in the appended independent claims, to which reference should now be made. Preferred or advantageous features of the invention are set out in dependent sub-claims.

In a first aspect the invention may provide a vibrational energy harvester comprising:

a first flexure assembly and a conductive coil, the coil being arranged in a first plane and fixed to the first flexure assembly;
a second flexure assembly and a magnet, the magnet being fixed to the second flexure assembly;
in which the first flexure assembly has a first resonant frequency in which the coil moves relative to the magnet in a first direction; and in which the second flexure assembly has a second resonant frequency different from the first resonant frequency in which the magnet moves relative to the coil in the first direction,
in which the flexure assemblies are configured so that the coil oscillates in the first plane, and the magnet is positioned adjacent the coil and oscillates in a plane parallel to the first plane, so that relative movement of the coil and the magnet induces an electrical current in the coil.

In use, ambient vibrational energy experienced by the harvester may excite a vibrational mode of one or both of the first and second flexure assemblies, so that the flexure assemblies flex and the coil and/or magnet oscillates back and forth along the first direction. In a vibrational mode in which the coil moves relative to the magnet in the first direction, the first flexure has the first resonant frequency and the second flexure assembly has the second resonant frequency. Oscillation of the coil and/or magnet relative to one another advantageously induces an electrical current in the coil, which may be harvested as electrical energy.

The first flexure assembly may function as a direct resonator. The second flexure assembly may function as a direct resonator.

As the coil is fixed to the first flexure assembly and the magnet is fixed to the second flexure assembly, the magnet and the coil are advantageously both free to oscillate in response to input vibration to the energy harvester. This advantageously increases the range of vibrational frequencies over which the harvester creates relative movement between the magnet and the coil, and over which the relative motion between the two flexure assemblies generates electrical energy.

Input vibration to the energy harvester is preferably provided by a machine or engine which vibrates during use. For example, the harvester may be mounted on a machine or engine, so that vibration of the machine or engine excites vibrational modes of the harvester. The present harvester is advantageously particularly suitable for harvesting energy from a source of input vibrations that operates with a variable-speed, such as a variable speed motor, machine or engine. As the frequency of the input vibrations will vary depending on the running speed of the motor, for example, the wide frequency band of the present harvester allows usable power to be generated over a larger range of motor speeds than would be possible with a single direct resonator.

The first flexure assembly is configured to oscillate at a first resonant frequency, and the second flexure assembly is configured to oscillate at a second resonant frequency different from the first resonant frequency. Thus the magnet and the coil may advantageously oscillate to different extents at different input frequencies of vibration. This may advantageously allow the harvester to generate electrical energy over a wider range of vibrational frequencies. The different resonant frequencies of the first and second flexure assemblies therefore give rise to a broader band of response for the harvester.

When input vibrational energy has a frequency near the first resonant frequency but not near the second resonant frequency, the first flexure assembly may be excited into oscillation while the second flexure assembly is not significantly excited. In this mode, the coil fixed to the first flexure assembly oscillates in proximity to the magnet, inducing an electrical current in the coil. Alternatively, when input vibrational energy has a frequency near the second resonant frequency but not near the first resonant frequency, the second flexure assembly may be excited into oscillation while the first flexure assembly is not significantly excited. In this mode, the magnet fixed to the second flexure assembly oscillates in proximity to the conductive coil, inducing an electrical current in the coil. At frequencies between the first resonant frequency and the second resonant frequency, vibrational modes of both the first and the second flexure assemblies may be excited, so that the coil and the magnet oscillate relative to one another, generating electrical current in the coil.

Therefore at least one of the flexure assemblies of the harvester may be excited into a vibrational mode over a wider range of frequencies than is possible with a single direct resonator. This means that the harvester can output electrical energy over a wider frequency bandwidth. This is particularly beneficial when the frequency of ambient vibrations is variable, for example if the harvester is mounted on a machine or engine which may operate at different frequencies.

The vibrational energy harvester may comprise a frame, which may be fixed relative to a source of input vibration. The first and second flexure assemblies may be fixed to the frame or configured to be fixed relative to a frame or housing, so that in use the coil and the magnet oscillate in the first direction relative to the frame. In a preferred embodiment, a frame portion of the first flexure assembly is fixed relative to a frame portion of the second flexure assembly.

In the present harvester the coil and the magnet are fixed to first and second flexure assemblies, respectively. In a preferred embodiment, the first and second flexure assemblies comprise first and second cantilever flexures, respectively. The present inventors have found that by mounting the coil and magnet on flexures rather than coil springs, the separate assemblies can be aligned with greater consistency, and the parallel alignment can be better maintained during oscillation, as flexures are less prone to out-of-plane movement than coil springs. Coil springs are typically susceptible to out-of-plane vibrational modes that could lead to undesired collisions between the first and second flexure assemblies.

In the present harvester, the coil is arranged in the first plane, so that the circumference of the coil lies in the first plane. The central axis of the coil, around which the coil is wound, extends orthogonally through the first plane. In this arrangement, the “first direction” along which the coil oscillates is contained in the first plane, and lies parallel to a diameter of the coil. This differs from prior art arrangements in which the coil is configured to oscillate along its own axis.

The harvester has a planar construction, in which the coil is arranged in the first plane, and the magnet is positioned adjacent to the coil and arranged to oscillate in a second plane parallel to the first plane. The planes of oscillation are parallel and adjacent to one another.

In use, the coil oscillates back and forth along the first direction in the first plane, and the magnet oscillates back and forth along the first direction in a plane adjacent to the coil. The present parallel-planes-of-oscillation harvester construction has advantageous constructional simplicity, and allows more consistent and reliable alignment between the first and second flexure assemblies.

The first flexure assembly is preferably configured to constrain movement of the coil to the first plane, and the second flexure assembly is preferably configured to constrain movement of the magnet to the plane parallel to the first plane. The plane in which the magnet oscillates may be termed the second plane, while in embodiments comprising magnets on either side of the coil, the magnets may be considered to oscillate in second and third parallel planes positioned on opposite sides of the first plane. The flexure assemblies may thus advantageously maintain the magnet(s) and coil in proximity to allow the generation of electrical current in the coil. By constraining movement to parallel planes, the flexure assemblies may also avoid out-of-plane vibrational modes that could lead to undesired collisions between the first and second flexure assemblies.

The first flexure assembly may comprise one or more first flexures, which are preferably cantilever flexures. The first flexures preferably have a fixed end that is coupled or couplable to a frame or frame portion, and the coil is preferably fixed to the first flexures at a position spaced from the fixed end, for example at a free end of the first flexure opposite the fixed end.

The second flexure assembly may comprise one or more second flexures, which are preferably cantilever flexures. The second flexures preferably have a fixed end that is coupled or couplable to a frame or frame portion, and the magnet is preferably fixed to the second flexures at a position spaced from the fixed end, for example at a free end of the second flexure opposite the fixed end.

The first flexure assembly preferably comprises a pair of first flexures arranged parallel to one another, and connected to one another by the coil, or by a crossbar.

The second flexure assembly preferably comprises a pair of second flexures arranged parallel to one another, and connected to one another by the magnet, or by a crossbar.

The use of pairs of parallel joined flexures in each flexure assembly advantageously increases out-of-plane stiffness and increases the resonant frequency of torsional vibrational modes. This reduces the susceptibility of each flexure assembly to undesirable out-of-plane bending or torsional modes, which may cause the first and second flexure assemblies to collide. This flexure arrangement also desirably stabilises the oscillation of the flexure assemblies in the desired “vertical” bending modes within the first plane (for the coil on the first flexure assembly) and parallel planes (for the magnet(s) on the second flexure assembly).

The flexures of the first and/or second flexure assemblies may be, for example: single-end clamped flexures which are fixed to a frame or frame portion at one end; or clamped – clamped flexures which are fixed to a frame at both ends. The first and second flexure assemblies may comprise the same or different types of flexures.

Preferably the first and second flexure assemblies comprise first and second cantilever flexures, respectively.

The first and second flexures may have different lengths. As flexure length is inversely related to spring constant, the length of the flexures affects their spring constant, and therefore their resonant frequency. The first flexure may preferably be longer than the second flexure. This may advantageously allow the harvester to be made more compact.

As the flexures may be folded, the “length” of a flexure typically relates to the distance from the fixed end of the flexure to its centre of mass, measured along the flexure from its fixed end.

The first and second flexure assemblies are arranged so that, in a resting configuration, the first and second flexures are adjacent to one another in a plane orthogonal to the first direction. The resting configuration of a cantilever flexure refers to the flexure in an unstrained state, for example when the entire energy harvester is at rest and all parts of the harvester are stationary relative to one another.

In the present harvester, the coil is preferably configured so that the axis of the coil is orthogonal to the first plane, and the magnet is configured so that its magnetic N-S axis is parallel with the axis of the coil. In this arrangement, the magnetic flux of the magnet is cut by the coil oscillating adjacent to the magnet.

In a preferred embodiment, the harvester comprises a pair of magnets fixed to the second flexure assembly. The second flexure assembly is preferably configured so that the pair of magnets are arranged one on either side of the coil with their poles aligned N-S-N-S, and so that each magnet oscillates in a plane parallel to the first plane. This may advantageously provide a well contained magnetic field across the coil. In this configuration, the first plane containing the coil is sandwiched by two parallel planes of oscillation each of which contains one magnet. In this arrangement, magnetic flux is directed across the coil from one magnet to another in a direction orthogonal to the first plane. When the coil oscillates in the first direction within the first plane, the coil cuts the lines of magnetic flux so that the relative movement of the coil and the magnet induces an electrical current in the coil.

Preferably a second pair of magnets is fixed to the second flexure assembly and arranged one on either side of the coil with their poles aligned S-N-S-N. Mounting two pairs of magnets on the second flexure assembly, with each pair aligned across the first plane, doubles the magnetic flux that is cut by the oscillating coil, and therefore increases the generated electrical current relative to a single magnet or pair of magnets. Aligning one pair of magnets N-S-N-S across the first plane, and the other pair S-N-S-N across the first plane may advantageously mean that the flux generated forms a closed loop which is cut by the coil during relative movement.

The magnets are preferably bar magnets.

In a preferred embodiment, the magnets are shaped to correspond to the shape of the coil. In a particularly preferred embodiment, for example, the second flexure assembly comprises four semi-circular, or semi-elliptical bar magnets, two magnets being arranged on each side of the coil.

The coil may be a circular coil. Alternatively, the coil may be a square or trapezoidal coil.

In a particularly preferred embodiment, the coil is oblong, or elliptical. The coil is preferably elongated in one dimension relative to a circular coil, so that the coil is oval or elliptical in shape. The coil may be defined by its major axis (the distance across its longest dimension) and its minor axis (the distance across its shortest dimension, which is perpendicular to the major axis). The oblong coil is preferably fixed to the first flexure assembly so that its minor axis is aligned parallel with the first direction, in which the coil oscillates. The inventors have realised that, rather than using a conventional circular coil, lengthening the dimension of the coil that is perpendicular to the first direction of oscillation, means that a greater proportion of the coil generates electrical current during oscillation, so more current is harvested.

The absolute dimensions of the coil may vary depending on the size of harvester required. However, in order to increase the generated current relative to a circular coil, preferably the oblong coil has an aspect ratio of at least 1.25:1, or 1.5:1, or 2:1, or 2.5:1, or 5:1.

In a preferred embodiment, the harvester comprises an oblong coil positioned in the first plane between two pairs of semi-oblong magnets, the perimeters of which have been shaped to correspond to the oblong shape of the coil. This arrangement advantageously allows the coil to cut the maximum amount of magnetic flux as the coil and magnets move relative to one another. If the coil were rectangular with its major axis in the magnetic field, then as the coil moves up and down the short sides (the minor axis) of the coil would generate no power because they are parallel to the direction of travel (Fleming’s right hand rule). By curving the outside surfaces of the magnets to match the oblong (quasi-elliptical) shape of the coil, the flux cut (the resolved component of wire direction, wire movement and magnetic flux) is maximised, so that more electrical current is generated.

There may be a design trade-off between the strength of the magnetic flux, the amount of power generated and the bandwidth of the harvester. For example, more magnetic flux may mean lower generated power but a wider frequency bandwidth.

The first resonant frequency and the second resonant frequency, and the frequency difference between them (which may be termed a frequency bandgap), may be controlled by controlling the spring constants and the masses of the first and second flexure assemblies relative to one another. For example, where the harvester is intended to harvest vibrational energy from a particular source of vibrations, the first and second resonant frequencies may be chosen to correspond to characteristic vibrational frequencies of the source. The frequency bandgap may advantageously be controlled to maximise the usable power bandwidth of the harvester, by maximising the range of frequencies at which the generated electrical energy is above a predetermined threshold.

The first resonant frequency and the second resonant frequency are preferably not harmonics of one another.

When the harvester is connected to an electrical system configured to consume the power generated by the harvester, in order to be providing “usable power” the harvester must be generating at least the minimum electrical power to operate the electrical system. For example, usable power may be defined as power above a threshold power level necessary to operate the electrical system. This must be greater, for example, than the leakage power of the electrical system.

The vibrational energy harvester is preferably configured to generate a power greater than the “usable power” threshold across the widest possible frequency range. The harvester may generate usable power across a frequency range wider than the frequency difference between the first and second resonant frequencies.

ISO10816 and ISO20816 are well known standards that specify different levels of vibration for machines of various sizes. In these ISO standard the amount of vibration is broadly classified into four bands (Good, Satisfactory, Unsatisfactory and Unacceptable), which occupy vibration levels of between 0.28mm/s and 45 mm/s. The harvester of the present invention may for example be configured so that for input vibrations in the “Good” band, the harvester generates sufficient power to power, for example, a sensor.

The first and second resonant frequencies are preferably selected so that for input vibrations of a predetermined magnitude (such as input vibrations in the ISO10816/ISO20816 “Good” band), a predetermined power is generated at input frequencies between the first resonant frequency and the second resonant frequency. The first and second resonant frequencies may be selected so that for input vibrations of a predetermined magnitude, in a graph of usable power vs input vibration frequency, the power between the first and second resonant frequencies is maintained above a predetermined desired power level.

While the flexure assemblies may have their own resonant frequencies at which the amplitude of oscillation, and therefore the collected power, is at its maximum, the oscillation response of the flexure assemblies naturally extends over a range of frequencies either side of the resonant frequency. The 3dB bandwidth is the range of frequency over which the collected power is above half of its maximum amplitude, and may otherwise be termed the full width half maximum (FWHM) of the flexure assembly’s oscillation response.

The output power of the harvester may approximate to and be graphically depicted on a graph of output power vs frequency as two adjacent bell curves, one of which is centred on the first resonant frequency and the other of which is centred on the first resonant frequency. Preferably the harvester is configured so that the two curves overlap one another in the frequency range between the first and second resonant frequencies. Preferably the first and second resonant frequencies are selected to maximise the width of the frequency range over which the total output power of the harvester is above a predetermined threshold power.

In preferred embodiments, the frequency difference between the first resonant frequency and the second resonant frequency may be greater than the 3 dB bandwidth of the first flexure assembly, and up to 30% of the first resonant frequency. The frequency difference is preferably at least equal to the 3 dB bandwidth so that the resonances of the two flexure assemblies don't overlap with one another, which may result in some cancellation during some of the phase shift windows where the coil and magnet are synchronised with one another, so there is no relative motion.

In certain preferred embodiments the frequency difference between the first resonant frequency and the second resonant frequency may be between 10 and 100 Hz, or between 20 and 80 Hz, or between 30 Hz and 50 Hz, depending on the vibrational frequency range of the input vibration source.

The coil may form all or part of a first mass fixed to the first flexure assembly, and the magnet may form all or part of a second mass fixed to the second flexure assembly. The resonator characteristics of the flexure assemblies may thus be calculated based on the first mass and second mass, respectively.

The resonant frequency ω_{res} of a mass on a flexure may be calculated as:

$$\omega_{res} = \sqrt{(k/m)}$$

where ω_{res} is the resonant frequency of the flexure assembly, k is the spring constant of the flexure assembly, and m is the mass fixed to the flexure assembly. The spring constant k is inversely proportional to the length of the flexure cubed, so longer flexures lead to lower spring constants, and thus lower resonant frequencies.

The first mass and/or the second mass may optionally comprise a counterweight in addition to the mass of the coil and magnet respectively. The addition of a counterweight to one or both of the flexure assemblies may advantageously allow the resonant frequency of the flexure assembly to be tuned. A counterweight may also advantageously increase the power output of the harvester assembly, as the power produced is proportional to the mass of the flexure assembly. Increased mass of a flexure assembly may be compensated for by stiffer flexures, which may be advantageous at low frequencies as stiffer springs are affected less by gravity.

An electrical circuit is preferably connected to the coil to harvest the electrical power generated by the harvester. This electrical circuit provides an electrical load, the magnitude of which is preferably variable.

The first flexure assembly and the second flexure assembly may be electromagnetically coupled to one another during oscillation, as movement of the first flexure assembly may affect the movement of the second flexure assembly, and vice versa, due to electromagnetic interaction between the magnet and the coil. This coupling may consist of damping of the movement of each flexure assembly, which may be termed electrical damping.

Electrical damping is caused by electromagnetic interaction between the magnet and the coil when the flexure assemblies move relative to one another. Interactions in the electrical domain of the magnet and coil circuitry manifest as an apparent damping force in the mechanical domain (Lorentz force) that opposes the relative motion between the magnet and coil. Part of this damping force may be attributed to Eddy currents, but the majority of this damping force may be attributed to the current flowing through the coil connected to a circuit.

Electrical damping depends on a load impedance (such as electrical resistance) in the circuit connected to the coil. If the circuit to which the coil is connected is arranged to be an open circuit (in which electrical resistance = infinity), then there will be no electrical damping applied to resist the relative motion of magnet and coil. There will be an open circuit voltage, but no current flow. As the electrical load resistance is connected, and its value decreased from infinity, current flow increases and electrical damping increases. Electrical damping would be maximum if load resistance were zero, i.e. the electromagnetic generator is short circuited.

As the magnet and coil components of the present harvester are located on separate movable flexure assemblies, this electrical damping may mutually damp the movement of both the first and second flexure assemblies. Thus the motion of the first and second flexure assemblies may affect one another, and the flexure assemblies may be “coupled” together by electrical damping. This coupling between separate oscillating flexure assemblies would not be experienced in alternative harvester designs, for example, if piezoelectric transducers were used instead of the magnet and coil transducer of the present invention.

Positioning the magnet and the coil on separate movable flexure assemblies with different resonant frequencies may advantageously provide a vibrational energy harvester that harvests more power than two individual direct resonators. In particular, the electrical coupling between the motion of the two flexure assemblies, and the phase difference between the oscillating coil and magnet, may advantageously broaden the bandwidth over which usable power is collected by the harvester.

The vibrational energy harvester may further comprise electric circuitry connected to the coil. The electric circuitry may comprise, or be connected to, an energy storage device.

The coil may be termed a conductive coil, or an electrically-conductive coil.

The harvester may comprise a flexible cable connected to the coil. Unlike conventional harvesters known in the art, by mounting the coil on a flexure, the coil in the present harvester moves during operation. In order to harvest the electric current induced in the coil, a flexible cable is coupled to the coil and configured to flex with the coil when the coil is oscillating.

The harvester preferably comprises means for limiting or restricting the displacement of the coil relative to the magnet in the first direction, and vice versa. This may be termed a displacement limiter, as described further below.

Dual-Oscillator Harvester with Displacement Limiter

In a further aspect of the invention there is provided a vibrational energy harvester, comprising:
a first flexure assembly and a conductive coil, the coil being fixed to the first flexure assembly;
a second flexure assembly and a magnet, the magnet being fixed to the second flexure assembly;
in which the first flexure assembly has a first resonant frequency in which the coil moves relative to the magnet in a first direction; and the second flexure assembly has a second resonant frequency different from the first resonant frequency in which the magnet moves relative to the coil in the first direction, so that relative movement of the coil and the magnet induces an electrical current in the coil; and
in which the harvester comprises a displacement limiter for limiting the displacement of the coil relative to the magnet in the first direction.

The displacement limiter may advantageously restrict how far the coil and the magnet(s) can move relative to one another when oscillating. This advantageously contains the first and second flexure assemblies to the range of relative displacements at which the largest quantity of usable power is generated. For example, the displacement limiter advantageously prevents the flexure assemblies from reaching relative positions (at large amplitudes of oscillation, for example) where the coil is no longer cutting magnetic flux, and therefore no power is generated. The displacement limiter may also enable the harvester to be provided in a more compact device than would otherwise be possible.

The displacement limiter is preferably configured to restrict the relative displacement of the coil and the magnet in the first direction to ± 5 mm or less, or ± 2.5 mm or less, or ± 1.5 mm or less, or ± 1 mm or less.

The displacement limiter may be configured to restrict the relative displacement of the coil and the magnet to a different extent depending on the operating frequency range of the harvester.

Limiting the relative displacement of the first and second flexure assemblies may also advantageously limit the maximum strain experienced by the flexure assemblies during oscillation. Limiting the strain on the flexures during oscillation may advantageously increase the lifetime of the harvester.

The displacement limiter may comprise one or more cushioning members and one or more stop surfaces, configured so that a cushioning member contacts a stop surface when the relative displacement of the magnet and the coil reaches a predetermined maximum displacement. Contact between the cushioning member and the stop surface preferably imparts a damping force to damp the oscillation of one or both flexure assemblies. configured to cushion impacts between the oscillator assemblies during oscillation.

The cushioning member advantageously cushions impacts between the flexure assemblies to prolong the lifetime of the harvester.

As the flexure assemblies are configured to oscillate back and forth along the first direction, two stop surfaces are preferably provided, to limit relative displacement in both directions. Thus whichever way the cushioning member is travelling, when it reaches the predetermined relative displacement, it will contact one of the stop surfaces, or vice versa.

Preferably, one of the cushioning member and the stop surfaces is provided on the first flexure assembly, and the other of the cushioning member and the stop surfaces is provided on the second flexure assembly. The displacement limiter may therefore function when the cushioning member on one flexure assembly contacts the stop surface on the other flexure assembly, so as to prevent further relative displacement.

The displacement limiter may advantageously allow the harvester to generate even more power at high frequencies because contact between the cushioning member and the stop surface excites resonances in both flexure assemblies when the limiting occurs.

The cushioning member may be configured to absorb and dissipate kinetic energy from the flexure assemblies when the flexure assemblies reach the predetermined maximum displacement, for example when the stop surface comes into contact with the cushioning member.

Alternatively, the cushioning member may be configured to absorb and re-release kinetic energy when the flexure assemblies reach the predetermined maximum displacement. This may advantageously help to conserve kinetic energy of the harvester, while preventing the flexure assemblies from oscillating at too high an amplitude, and potentially breaking from too much strain. For example, the cushioning member may comprise a spring configured to be compressed by the stop surface when the flexure assemblies reach the predetermined maximum displacement. The cushioning member may alternatively comprise a resilient material, for example silicone rubber, which stores and releases energy.

The cushioning member may comprise a resilient material. In a preferred embodiment, the cushioning member is an O-ring. O-rings may advantageously be available in a variety of sizes and thicknesses, so that the cushioning member is straightforward to replace during the lifetime of the harvester.

The cushioning member may be, for example, an O-ring arranged around the coil mount for cushioning contact on either side of the coil mount.

In preferred embodiments of the harvester, the second flexure assembly comprises magnets on either side of the coil. In these embodiments, the second flexure assembly preferably comprises a connector which connects the portions of the assembly on opposite sides of the coil so that both portions of the second flexure assembly always oscillate in phase. In order to connect the portions on opposite sides of the coil, the connector must extend through the first plane in which the coil oscillates. In this embodiment, either the cushioning member or the stop surfaces may be provided on the connector at a position in the plane of the coil, with the other of the cushioning member or the stop surfaces provided on the first flexure assembly, also in the plane of the coil. When the relative displacements of the connector and the first flexure assembly in the first plane reaches the predetermined maximum displacement, the cushioning member and a stop surface will come into contact with one another to prevent further relative displacement.

The second flexure assembly may be configured so that at least a portion of the second flexure assembly oscillates in the same plane as the first flexure assembly.

In some preferred embodiments, the cushioning member may be configured to absorb and dissipate kinetic energy from the flexure assemblies when the flexure assemblies contact one another. For example, the cushioning member may be a damping member configured to damp the oscillations of both flexure assemblies when they come into contact.

In other preferred embodiments, the cushioning member may be configured to absorb kinetic energy from the flexure assemblies when the flexure assemblies contact one another, and to re-release the absorbed kinetic energy to both flexure assemblies following impact. This may advantageously conserve the kinetic energy in the harvester to maximise power output.

For example, the cushioning member may comprise a spring configured to be compressed by the stop surface when the flexure assemblies contact one another. Alternatively the cushioning member may comprise a resilient material, for example silicone rubber.

In a particularly preferred embodiment, the cushioning member comprises a rubber O-ring mounted on one of the flexure assemblies. The O-ring may be compressed by the stop surface when the two flexure assemblies collide, and then expand and release the stored kinetic energy when the flexure assemblies bounce apart.

In a further embodiment, the cushioning member may comprise a first magnet, and a second magnet is mounted on the stop surface and configured to repel the first magnet when the flexure assemblies approach one another. The opposing magnets may store kinetic energy as the flexure assemblies get close, and then release it to accelerate the flexure assemblies apart again, without the two assemblies necessarily contacting one another.

In a preferred embodiment, one of the two flexure assemblies may comprise two stop surfaces, spaced from one another along the first direction, which defines the axis of oscillation of the flexure assemblies. The other of the flexure assemblies may comprise one or more cushioning members, positioned between the two stop surfaces.

Two cushioning members may be provided, one arranged to face either of the two stop surfaces. Alternatively, one cushioning member may be arranged to contact both stop surfaces in the event that the relative displacement becomes too large in either direction.

As the flexure assemblies are configured to oscillate back and forth along the first direction, the maximum relative displacement may be reached in two different directions. It is therefore desirable to have a displacement limiter which is configured to limit the relative displacement of the flexure assemblies both ways along the axis of oscillation.

The first and second flexure assemblies are both configured to oscillate in the first direction, preferably in adjacent planes. The first and second flexure assemblies therefore may have parallel axes of oscillation, which are positioned in adjacent planes and both directed along the first direction.

The flexure assemblies are preferably configured so that the coil oscillates in the first plane, and the magnet is positioned adjacent the coil and oscillates in a plane parallel to the first plane.

Preferably at least a portion of the first flexure assembly and a portion of the second flexure assembly oscillate in the first plane of oscillation. One of the flexure assemblies may comprise two stop surfaces positioned in the first plane of oscillation, and the other flexure assembly may comprise the cushioning member positioned in the first plane of oscillation between the two stop surfaces. Alternatively one of the flexure assemblies may comprise two cushioning members positioned in the first plane of oscillation, and the other flexure assembly may comprise two stop surfaces positioned in the first plane of oscillation between the two cushioning members. Such arrangements may advantageously cushion collisions regardless of the direction in which the flexure assemblies are moving.

Preferably the second flexure assembly comprises two stop surfaces spaced from one another along the first direction (the axis of oscillation of the first flexure assembly). Preferably a first cushioning member, or a first portion of the cushioning member, is mounted at a first position on a flexure assembly, and a second cushioning member, or a second portion of the cushioning member, is mounted on the same flexure assembly at a second position spaced from the first position along the axis of oscillation of the first flexure assembly. The first cushioning member, or the first portion of the cushioning member, is preferably configured to come into contact with one of the two stop surfaces if the relative displacement of the two flexure assemblies reaches a predetermined maximum relative displacement. The second cushioning member, or the second portion of the cushioning member, is preferably configured to come into contact with the other stop surface if the flexure assemblies reach the predetermined maximum relative displacement travelling in the other direction.

The displacement limiter preferably comprises a first cushioning member and a first stop surface that are configured to contact one another when the first flexure assembly reaches the predetermined maximum displacement moving one way along the axis of oscillation, and a second cushioning member and a second stop surface which are configured to contact one another when the first flexure assembly reaches the predetermined maximum displacement moving the other way along the axis of oscillation.

Preferably one of the first cushioning member and the first stop surface is provided on the first flexure assembly and the other of the first cushioning member and a first stop surface is provided on the second flexure assembly. Likewise, preferably one of the second cushioning member and second stop surface is provided on the first flexure assembly and the other of the second cushioning member and second stop surface is provided on the second flexure assembly.

One of the first or second flexure assemblies may comprise the first cushioning member and the second cushioning member spaced from the first cushioning member in the first direction. The other of the first or second flexure assemblies preferably comprises the first stop surface, and the second stop surface spaced from the first stop surface in the first direction.

Alternatively, one of the first or second flexure assemblies may comprise the first cushioning means and the second stop surface spaced from the first cushioning means in the first direction, while the other of the first or second flexure assemblies comprises the second cushioning means and the first stop surface spaced from the second cushioning means in the first direction.

In addition to a displacement limiter that limits relative displacement of the magnet and the coil, the harvester may additionally comprise a stop surface to restrict the amplitude of oscillation of the first and/or second flexure assemblies to a predetermined maximum amplitude. Limiting the relative displacement of the first and second flexure assemblies may advantageously limit the maximum strain experienced by the flexure assemblies during oscillation, to increase the lifetime of the harvester. The stop surface may advantageously be configured for shock loading.

Non-linear Flexure

In a further aspect of the present invention there is provided a vibrational energy harvester comprising a flexure, the flexure having a first end and a second end opposite the first end, in which between the first end and the second end the flexure comprises a non-linear portion having a non-linearly-varying width.

The non-linearly varying width of the flexure advantageously reduces the stress experienced by the flexure during bending, compared to a parallel-sided flexure of equivalent length. This may advantageously allow the stress on the flexure to remain below the fatigue limit of the flexure material during repeated bending, and may prolong the life of the flexure.

This is very different from prior art designs for piezoelectric energy harvesters, which use tapered flexures to try to maximise stress in order to maximise power generated by piezoelectric elements on the flexures.

When a flexure is bent during oscillation in a vibrational energy harvester, bending strain is greatest in the central portion (or “belt”) of the flexure, where it bends most severely.

During bending of a flexure, the stresses and strains experienced by the flexure may be concentrated at vertices in the bent region. By providing a non-linear portion with a non-linearly-varying width, no vertices are present in the region of the flexure being bent, so stresses and strains are more evenly distributed across the flexure.

The non-linear portion is preferably a narrowed portion, having a continuously-varying width less than the width of the first end and the second end.

The inventors have found that when an elongate flexure is bent, bending strain is typically concentrated towards the central axis of the flexure, with less strain experienced towards the lengthways edges of the flexure. By narrowing a portion of the flexure, the overall weight of the flexure may be reduced, while the parts of the flexure that are experiencing the majority of the bending strain are retained. Thus the bending strength and reliability of the flexure may advantageously be maintained, while its overall weight is reduced relative to a conventional straight-edged flexure.

The non-linear portion is preferably narrowest at the position at which greatest bending strain is experienced during bending of the flexure.

In a preferred embodiment, the non-linear portion is narrowest half way between the first end and the second end of the flexure.

The flexure may comprise opposing first and second edges connecting the ends of the flexure, and the first and second edges may comprise concave sections forming the non-linear portion of the flexure. Preferably the concave sections are symmetrical about a central lengthways axis of the flexure.

In an exemplary embodiment the concave sections are parabolic in shape.

The first and second edges may comprise straight sections on either side of the concave sections.

The concave sections may comprise at least 30%, or 40%, or 50%, and less than or equal to 60% or 70% or 80% of the length of the flexure.

The flexure may be a cantilever flexure.

The flexure may be, for example: a single-end clamped flexure which is configured to be fixed to a frame at one end; or a clamped – clamped flexure which is configured to be fixed to a frame at both ends.

In a preferred embodiment, the harvester may comprise a flexure element, in which a first flexure and a second flexure are formed from the same sheet of material, in which the first and second flexures are connected to one another at a first end of the flexure element, so that the first end of the flexure element forms a fixed end of both the first and second flexures, and in which the first and second flexures each comprise free second ends configured to flex independently of one another. In this embodiment, the first and/or second flexures may comprise a non-linear portion having a non-linearly-varying width between the first end and the second end the flexure.

Vibrational Energy Harvester with Amplitude Restrictor

In a further aspect of the invention there is provided a vibrational energy harvester, comprising:
a frame;
an oscillator assembly, coupled to the frame and configured to oscillate relative to the frame; and
an amplitude restrictor, comprising a first part fixed relative to the frame and a second part fixed relative to the oscillator assembly,
in which the amplitude restrictor is configured to restrict the oscillation of the oscillator assembly to a predetermined maximum amplitude.

The first and second parts of the amplitude restrictor may comprise a cushioning member and a stop surface, one of the cushioning member and the stop surface being fixed relative to the frame, and the other being positioned on the oscillator assembly. The cushioning member and the stop surface may be configured to contact one another when the oscillator assembly oscillates at the predetermined maximum amplitude.

Contact between the cushioning member and the stop surface preferably imparts a damping force to damp the oscillation of the oscillator assembly.

The amplitude restrictor is preferably configured for shock loading.

The cushioning member and stop surface may advantageously come into contact to limit the motion of the oscillator assembly when the oscillator assembly reaches or exceeds a predetermined maximum amplitude. The relative positioning of the stop surface and the cushioning member may thus set a maximum displacement limit beyond which the oscillator assembly cannot oscillate. This may advantageously limit the maximum strain experienced by the oscillator assembly during oscillation. Limiting the strain on the oscillator assembly, for example on flexures of the oscillator assembly, during oscillation may advantageously increase the lifetime of the harvester.

The cushioning member may be configured to absorb and dissipate kinetic energy from the oscillator when the oscillator assembly reaches the predetermined maximum amplitude, for example when the stop surface comes into contact with the cushioning member.

Alternatively, the cushioning member may be configured to absorb and re-release kinetic energy from the oscillator when the oscillator reaches the predetermined maximum amplitude. This may advantageously help to conserve kinetic energy of the harvester, while preventing the oscillator from oscillating at too high an amplitude, and potentially breaking from too much strain. For example, the cushioning member may comprise a spring configured to be compressed by the stop surface when the oscillations of the oscillator exceed a threshold amplitude. The cushioning member may alternatively comprise a resilient material, for example silicone rubber, which stores and releases energy.

In a preferred embodiment, the cushioning member comprises an O-ring mounted on the frame or the oscillator assembly.

Alternatively the cushioning member and the stop surface may comprise a pair of opposing magnets, so that the opposing magnets interact with one another to store kinetic energy as the opposing magnets come close to one another, and repel one another to re-release kinetic energy back to the oscillator assembly. Thus the movement of the oscillator assembly may be limited and cushioned without the cushioning member coming into contact with a stop surface.

Either the stop surface or the cushioning member may be fixed relative to the frame, and the other of the stop surface or the damping member may be mounted on the oscillator assembly. Thus, when the oscillator assembly is oscillating relative to the frame and the amplitude of oscillations becomes too great, the stop surface and the cushioning member come into contact and restrict the movement of the oscillator assembly.

The cushioning member may comprise a shock-absorbent material, so that contact between the cushioning member and the stop surface imparts a damping force to damp the oscillation of the oscillator assembly.

In a preferred embodiment, the harvester may comprise two stop surfaces positioned on the axis of oscillation of the oscillator assembly, and one or two cushioning members may be positioned between the two stop surfaces on the axis of oscillation. Oscillation in either direction may thus be restricted to a maximum amplitude by contact between a cushioning member and one of the two stop surfaces.

Alternatively the harvester may comprise two cushioning members positioned on the axis of oscillation of the oscillator assembly, and two stop surfaces positioned between the two cushioning members on the axis of oscillation. Oscillation in either direction may thus be restricted to a maximum amplitude by contact between a cushioning member and one of the two stop surfaces.

The amplitude restrictor may comprise a pair of magnets with opposing magnetic fields, the first part of the amplitude restrictor comprising one of the pair of magnets fixed relative to the frame, and the second part of the amplitude restrictor comprising a magnet fixed to the oscillator assembly. The magnets may be configured to repel one another when the oscillator assembly oscillates at the predetermined maximum amplitude, so that the oscillator assembly is slowed as it approaches maximum amplitude, and then accelerated away as the magnets repel one another.

The features of the other aspects of the invention described above may advantageously be combined with one another in a vibrational energy harvester. Features described in relation to one aspect of the invention are equally applicable to another aspect of the invention unless otherwise indicated.

Brief Description of the Drawings

Specific embodiments of the invention will now be described with reference to the figures, in which:

Figure 1 is a schematic illustration of a single-resonator vibrational energy harvester known in the prior art;

Figure 2 is a schematic illustration of a vibrational energy harvester comprising two resonators, according to an aspect of the present invention;

Figures 3a – 3c show the simulated frequency response of a vibrational energy harvester as shown in Figure 2;

Figures 4a and 4b are plan and side views of a double flexure element according to a preferred embodiment of the present invention;

Figure 5a shows a perspective view of an assembled dual-resonator vibrational energy harvester according to a preferred embodiment of an aspect of the present invention;

Figure 5b is a cross-sectional view of the dual-resonator vibrational energy harvester of Figure 5a;

Figure 5c is an exploded view of the dual-resonator vibrational energy harvester of Figures 5a and 5b;

Figures 6a and 6b are graphs of the open circuit voltage vs frequency of vibration for the two resonators of the vibrational energy harvester of Figures 5a-5c;

Figure 6c is a graph of power generation vs frequency of vibration for the dual resonator vibrational energy harvester of Figures 5a-5c;

Figures 7a and 7b are plan and side views of a flexure element according to a preferred embodiment of an aspect of the present invention;

Figure 8a shows a perspective view of an assembled single-resonator vibrational energy harvester according to a preferred embodiment of an aspect of the present invention;

Figure 8b is a cross-sectional view of the vibrational energy harvester of Figure 8a;

Figure 8c is a partially exploded view of the vibrational energy harvester of Figures 8a and 8b.

In the known vibrational energy harvester 1 shown in Figure 1, a cantilever flexure 10 is fixed at one end to a vibrating support structure 20. A proof mass 30 is suspended from the other, free end of the cantilever flexure. A magnet 70 is fixed relative to the proof mass, and an electrically-conductive coil 40 is fixed relative to the support structure or frame of the harvester. Vibration of the support structure 20 excites a vibrational mode of the flexure so that the proof mass oscillates relative to the support structure. This creates relative movement of the magnet 70 and conducting coil 40, inducing an electrical current in the coil which may be extracted as an electrical power output.

This prior art harvester 1 may be termed a direct resonator.

The resonator has a resonant frequency which depends on the spring constant of the flexure and the mass of the proof mass as described above in the summary of invention.

The flexure is excited into a vibrational mode only when ambient vibrations are at or near the resonant frequency of the resonator. The amplitude of oscillation, and therefore the output power of the harvester, is at a maximum when ambient vibrations are at the resonant frequency of the resonator. As the frequency of the ambient vibrations gets further away from the resonant frequency of the resonator however, the oscillation response of the resonator falls away relatively quickly, with the amplitude of oscillation decreasing quickly at frequencies further from the resonant frequency. This means that the output power from the coil is only high enough to be useful when the resonator is vibrating at or close to its resonant frequency.

As shown in Figure 2, a vibrational energy harvester 100 according to a preferred embodiment of the present disclosure comprises a first flexure assembly 110 fixed to a frame 120. In the illustrated embodiment the first flexure assembly comprises a first flexure 130, one end of which is fixed to the frame 120, and the other end of which is free to flex relative to the frame, and a coil 140 of electrically-conductive wire mounted with a first proof mass 135 on the free end of the first flexure 130. Movement of the free end of the first flexure 130 causes the coil 140 to move relative to the frame 120.

A second flexure assembly 150 is also fixed to the frame 120, and comprises a second flexure 160 on which a magnet 170 and a second proof mass 175 are mounted. The first and second flexure assemblies are arranged on the frame so that, when the harvester 100 is stationary and the flexures are in their unstrained rest positions as shown in Figure 1, the coil 140 and the magnet 170 are adjacent to one another. The coil 140 and the magnet 170 are close enough so that the coil experiences the magnetic field of the magnet.

An electrical circuit 180 is connected to the coil 140 by flexible wires so that the electrical load of the circuit is applied across the coil.

In use, vibrational energy is applied to the harvester 100, which causes the frame 120 to vibrate. These vibrations in turn cause the first and second flexures to flex, so that the flexure assemblies 110, 150 oscillate relative to the frame along a first direction (indicated by double ended arrows in Figure 1).

The oscillation behaviour of the flexure assemblies depends on their resonant frequencies ω_{res}, which in turn depends on the spring constants k of the first and second flexures, and the masses of the flexure assemblies.

The first and second flexure assemblies 110, 150 have different resonant frequencies, so for a given frequency of input vibrations ω_{input}, the oscillation response of the first flexure assembly 110 differs from the oscillation response of the second flexure assembly 150. Vibration of the harvester 100 at ω_{input} therefore causes the coil 140 to move relative to the magnet 170. Relative movement of the coil 140 and the magnet 170 causes the coil to experience a changing magnetic field, which induces an electrical current in the coil. The induced current flows to the electrical circuit 180 and is harvested as electrical energy.

In use, the vibrational energy harvester 100 may be mounted on a source of ambient vibrations, for example a motor, which vibrates during regular operation.

In prior art vibrational energy harvesters with a single direct resonator, the frequency response of the flexure typically means that usable electrical power is only generated in a narrow range of frequencies close to the resonant frequency of the flexure. If the input vibrations, which may be termed ambient vibrations, have a frequency not near the resonant frequency of the resonator, then no power is generated.

The dual-resonator harvester of the present invention, however, allows usable electrical power to be generated over a much broader frequency range. Both the first flexure assembly 110 and the second flexure assembly 150 have their own resonant frequencies at which their respective oscillations are at their greatest amplitude, and the resonant frequencies may be selected so that usable power is generated across a broad range of frequencies containing the two resonant frequencies. By covering a broader frequency bandwidth, at least one of the flexure assemblies of the harvester may be excited into a vibrational mode over a wider range of input frequencies than is possible with a single direct resonator. This means that the harvester can output electrical energy over a wider frequency bandwidth. This is particularly beneficial when the frequency of ambient vibrations is variable, for example if the source of ambient vibrations is a motor that operates across a range of frequencies.

The first and second flexure assemblies 110, 150 do not touch one another, so the first flexure is free to oscillate independently of the second and vice versa. In use, however, the oscillations of the first and second flexure assemblies may be coupled together by electrical damping caused by electromagnetic interactions between the magnet and the coil with its electrical load.

Simulated response curves of a dual-band harvester are shown in Figures 3a, 3b and 3c.

For these simulations, the first flexure assembly (response shown in Figure 3a) was modelled with a resonant frequency of 70.62 Hz, while the second flexure assembly (response shown in Figure 3b) was modelled with a resonant frequency of 81.53 Hz.

In the example diagrams of Figures 3a to 3c, a swept frequency, from low frequency (60Hz) to high frequency (90Hz), was applied to a Scilab simulation of the vibrational energy harvester 100 over a period of 100 seconds. The resulting graphs are as follows:

Figure 3a: Time domain response of the first flexure assembly 110 plotted on a linear scale. As expected, the first flexure assembly only responds by oscillating near its natural resonant frequency of 71 Hz.

Figure 3b: Time domain response of the second flexure assembly 150 plotted on a linear scale. As expected, the second flexure assembly only responds near its natural resonance frequency of 82 Hz, which in this case is higher than that of the first flexure assembly 110 as determined by its spring stiffness and mass.

Figure 3c: Figure 3c shows the combined frequency response of the dual-resonator harvester plotted on a logarithm scale. This shows that useful power is generated across a wide range, for this case from 60Hz through to 90Hz. The 3dB bandwidth is narrower than the usable power bandwidth, but for many applications the usable power is a more meaningful measure. For example if sufficient (useful) power is generated across the range of a variable frequency drive (VFD) motor then a harvester mounted on that motor can extract energy to power sensors across the normal operating range of the motor. A conventional single cantilever harvester, however, would have a much narrower useful power bandwidth and hence would not be applicable in a VFD environment.

Figures 4a and 4b illustrate a preferred embodiment of a flexure element 400 according to the second aspect of the invention.

The flexure element 400 is a single sheet of metal, such as spring steel, and the outer perimeter of the flexure element is substantially rectangular with four rounded corners. A slot 410 is formed through the sheet of spring steel so that the slot approximately defines three sides of a rectangle, with an open end adjacent a first end 420 of the flexure element. The section of the flexure element inside the slot 410 forms an internal flexure 430, while the section of the flexure element between the slot and the perimeter of the flexure element forms a U-shaped external flexure 440. The internal flexure and the external flexure are connected to one another at the first end 420 of the flexure element where the slot 410 ends. As the slot 410 separates the two flexure elements at their other ends, however, the internal flexure has a second end 450 that is free to flex independently of a second end 460 of the external flexure.

Six holes 470 are formed in the flexure element, to allow the flexure element to be attached to other components of a vibrational energy harvester. Two of the holes 470 are positioned at the corners of the first end 420 of the flexure element, to allow the first end 420 to be attached to the frame of a vibrational energy harvester. Two of the holes 470 are positioned at the corners of the second end 460 of the external flexure, to allow a component to be mounted on the second end 460 of the external flexure. Two of the holes 470 are positioned at the corners of the second end 450 of the internal flexure, to allow a component to be mounted on the second end 450 of the internal flexure.

The slot 410 is shaped so that the internal flexure 430 is slightly narrower adjacent the first end 420 than at its second end 450. Between the ends of the slot and the second end 450 of the internal flexure, the width of the internal flexure varies non-linearly. As the outer edges of the flexure element 400 are straight, this means that the legs of the U-shaped external flexure also have a non-linearly varying width between the first end 420 and the second end 460 of the external flexure.

Figures 5a to 5c illustrate a preferred embodiment of a dual-resonator vibrational energy harvester 500 embodying the harvester shown schematically in Figure 2. The harvester 500 comprises two of the flexure elements 400 of Figures 4a and 4b.

The first ends 420 of two identical dual flexure elements 400 are attached to opposite sides of a frame 510 by screws 520, so that the width of the frame 510 separates the two flexure elements 400, and the flexure elements extend from the frame so that they are parallel to one another.

A crossbar 530 is positioned between the two flexure elements 400, and screwed to the second ends 460 of both of the U-shaped external flexures 440 to rigidly connect the two flexures. A coil mount 540 is connected to the crossbar 530 between the two parallel flexures, and extends out of the crossbar in the direction of the frame.

A round coil 550 of electrically-conductive wire is mounted on the coil mount, so that the coil is positioned mid-way between the parallel flexure elements 400 and roughly mid-way between the crossbar 530 and the frame 510. The circumference of the coil 550 is arranged in plane with the two flexure elements, the frame and the crossbar.

A magnet assembly 560 is screwed to the second ends 450 of the internal flexures 430, so that the magnet assembly is suspended between the parallel internal flexures. The magnet assembly has two opposing halves, with two magnets 570 mounted on each half. The magnets 570 are arranged in pairs with the magnetic poles of one half opposing the magnetic poles of the magnets on the other half of the assembly. A space between the sides of the magnet assembly is configured to receive the coil 550, and the magnets are oriented so that their magnetic poles are orthogonal to the plane of the coil.

The harvester comprises a displacement limiter which restricts the relative displacement of the first and second flexure assemblies during oscillation. The displacement limiter is made up of a cushioning member, which is a silicone O-ring 580 arranged around the coil mount 540, and two stop surfaces 590 on the magnet assembly. The two stop surfaces 590 are located above and below the O-ring on the coil-mount, so that when the harvester 500 is assembled the O-ring is aligned with both stop surfaces 590 on the magnet assembly.

When assembled, as shown in Figure 5a, the coil 550 is suspended between the second ends 460 of the external flexures 440, and the magnet assembly is suspended between the second ends of the internal flexures 430, with the coil 550 positioned in the space between the two sides of the magnet assembly. In this arrangement, the coil 550 experiences the magnetic flux of the magnet assembly, so that relative movement between the coil and the magnets induces an electrical current in the coil.

The parallel orientation of the two flexure elements 400 allows the coil and the magnet assembly each to oscillate relative to the frame along a first direction, or a first axis.

As the internal flexures have a different length from the external flexures, and as different masses are fixed to the internal and external flexures, the resonant frequency of the internal flexures and magnet assembly is different from the resonant frequency of the external flexures and coil.

In use, when the frame vibrates as a result of ambient vibrations, the flexure assemblies (the internal flexures plus magnet assembly, and the external flexures plus coil, respectively) are excited and oscillate relative to the frame and each other. As the flexure assemblies have different resonant frequencies, however, the oscillation response of the two flexure assemblies differs according to the frequency of the input vibration and its proximity to the resonant frequencies of the flexure assemblies.

If the relative displacement of the flexure assemblies reaches a predetermined maximum, the O-ring 580 comes into contact with one of the stop surfaces 590 to limit the travel of the flexure assembly. For example, if the displacement limiter is configured to limit relative displacement of the two flexure assemblies to +/- 2.5 mm, then when one of the two flexure assemblies moves 2.5 mm from its rest position relative to the other flexure assembly, the O-ring will collide with the stop surface. During a collision, the silicone O-ring cushions the impact by absorbing some kinetic energy from the oscillating assemblies, and re-releases the energy to both flexure assemblies as the flexure assemblies bounce apart. This may advantageously retain kinetic energy in the harvester to maximise the power generated, while preventing damaging collisions between the two oscillating flexure assemblies, and limiting the strain experienced by the flexures due to high-amplitude oscillations. By restricting the relative displacement of the coil and the magnet, this also contains the flexure assemblies to the range of travel over which the greatest quantity of power is generated.

An electrical circuit (not shown) is connected to the coil 550 by flexible cables, so that the electrical load of the circuit is applied across the coil. Electrical power generated in the coil can then be used by the electrical circuit.

Figure 6a shows the open circuit voltage produced across the coil over a range of input vibration frequencies, when only the internal flexures 430 and the magnet assembly 560 is oscillating. Figure 6b shows the open circuit voltage produced across the coil over a range of input vibration frequencies, when only the external flexures 440 and the coil 550 is oscillating. The different amplitude responses at different frequencies illustrate the different resonant frequencies of the two flexure assemblies.

Figure 6c shows the power produced by the vibrational energy harvester of Figures 5a to 5c over a range of input vibration frequencies, when both the internal and external flexure assemblies are free to oscillate relative to the frame. The acceleration of the input vibrations was varied from 0.1 g (g = gravitational acceleration = 9.8 m/s^2) to 0.5 g in 0.1 g increments, and the output power of the harvester 500 was measured.

At low accelerations of 0.1 g, the power output of the harvester 500 shows two distinct peaks at the resonant frequencies of the internal and external flexure assemblies, respectively. As the acceleration increases, however, the power generated at frequencies between the two resonant frequencies increases significantly, and a greater magnitude of power is generated over a wider frequency bandwidth.

The inventors have found that the electrical coupling effect achieved by allowing both the coil and the magnet assembly to oscillate relative to the frame and relative to one another, advantageously leads to this increased power generation, which is significantly greater than that of two individual direct resonators with equivalent resonant frequencies.

Figures 7a and 7b illustrate an embodiment of a flexure 700 suitable for use in vibrational energy harvesters.

The flexure 700 has a first end section 710 and a second end section 720 opposite the first end. Each end section contains two holes 730 suitable for receiving screws to attach the ends of the flexure to other components of a vibrational energy harvester, as shown in Figures 8a to 8c.

A central portion 740 of the flexure connects the first end section 710 and the second end section 720. The central portion 740 is narrower than the two end portions, and is defined by two concave edges 750 of the flexure 700. Between the first end section 710 and the second end section 720, the width of the flexure 700 varies non-linearly along the central portion 740. The concave edges 750 are symmetrical, so the narrowest point of the flexure is positioned half way along its length.

The non-linearly varying width of the flexure advantageously reduces the stress experienced by the flexure during bending, compared to a parallel-sided flexure of equivalent length, so that the flexure remains below the fatigue limit during bending.

The flexure 700 is formed from a sheet of metal such as spring steel, so that the flexure may flex resiliently on application of a force.

As shown in Figures 8a to 8c, the flexure 700 may be integrated into a vibrational energy harvester 800.

The vibrational energy harvester 800 is a single-resonator harvester similar to the harvester shown schematically in Figure 1.

In the single-resonator harvester 800, the first end sections of two parallel flexures 700 are attached to opposite sides of a frame 810. A magnet assembly 820 is mounted between the second end sections of the flexures 700, so that the magnet assembly may oscillate relative to the frame by bending the flexures. A coil 830 is mounted on a coil mount 840 which is rigidly connected to the frame 810, so that the coil is positioned between magnets 870 on the two halves of the magnet assembly in use.

A silicone O-ring 850 is located around the coil mount so that, when the harvester is constructed, the O-ring is aligned with two stop surfaces 860 on the magnet assembly. As the coil mount is rigidly connected to the frame 810, the O-ring always remains stationary relative to the frame.

In use, vibration of the frame 810 causes the flexures 700 to flex so that the magnet assembly and the flexures oscillate back and forth along a first direction orthogonal to the parallel flexures. This creates relative movement between the magnets in the magnet assembly and the coil, so that an electrical current is induced in the coil and harvested by an electrical circuit (not shown).

When the amplitude of oscillation of the magnet assembly reaches a predetermined amplitude relative to the frame, one of the stop surfaces 860 comes into contact with the O-ring, which limits the movement of the magnet assembly and damps the oscillation. This prevents the oscillations from getting too large and putting the flexures 700 under too much strain during operation.

Preferred Aspects

Preferred aspects of the present invention are defined in the following numbered clauses:

1. A vibrational energy harvester having a flexure, the flexure having a first end and a second end opposite the first end, in which between the first end and the second end the flexure comprises a non-linear portion having a non-linearly-varying width.

2. A vibrational energy harvester according to clause 1, in which the non-linear portion is narrowest at the position at which greatest flexural strain is experienced during bending of the flexure.

3. A vibrational energy harvester according to clause 1 or 2, in which the non-linear portion is a narrowed portion having a continuously-varying width less than the width of the first end and the second end.

4. A vibrational energy harvester according to clause 3, in which the non-linear portion is narrowest half way between the first end and the second end of the flexure.

5. A vibrational energy harvester according to clause 3 or 4, in which the flexure comprises opposing first and second edges connecting the ends of the flexure, and in which the first and second edges comprise concave sections forming the non-linear portion of the flexure.

6. A vibrational energy harvester according to clause 5, in which the concave sections are symmetrical about a central lengthways axis of the flexure.

7. A vibrational energy harvester according to clause 5 or 6, in which the concave sections are parabolic in shape.

8. A vibrational energy harvester according to clause 5, 6 or 7, in which the first and second edges comprise straight sections on either side of the concave sections.

9. A vibrational energy harvester according to any of clauses 5 to 8, in which the concave sections comprise at least 30%, or 40%, or 50%, and less than or equal to 60% or 70% or 80% of the length of the flexure.

10. A vibrational energy harvester according to any preceding clause, in which the flexure is a cantilever flexure.

11. A vibrational energy harvester, comprising:
a frame;
an oscillator assembly, coupled to the frame and configured to oscillate relative to the frame; and
an amplitude restrictor, comprising a first part fixed relative to the frame and a second part fixed relative to the oscillator assembly,
in which the amplitude restrictor is configured to restrict the oscillation of the oscillator assembly to a predetermined maximum amplitude.

12. A vibrational energy harvester according to clause 11, in which the first and second parts of the amplitude restrictor comprise a cushioning member and a stop surface, one of the cushioning member and the stop surface being fixed relative to the frame, and the other being positioned on the oscillator assembly, in which the cushioning member and the stop surface are configured to contact one another when the oscillator assembly oscillates at the predetermined maximum amplitude.

13. A vibrational energy harvester according to clause 12, in which the cushioning member is configured to absorb and dissipate kinetic energy from the oscillator assembly when the oscillator assembly reaches the predetermined maximum amplitude.

14. A vibrational energy harvester according to clause 12, in which the cushioning member is configured to absorb and re-release kinetic energy from the oscillator assembly when the oscillator assembly reaches the predetermined maximum amplitude.

15. A vibrational energy harvester according to clause 14, in which the cushioning member comprises a spring configured to be compressed by the stop surface when the oscillations of the oscillator assembly reach the maximum amplitude.

16. A vibrational energy harvester according to clause 14, in which the cushioning member comprises a resilient material, for example silicone rubber.

17. A vibrational energy harvester according to clause 16, in which the cushioning member comprises an O-ring mounted on the frame or the oscillator assembly.

18. A vibrational energy harvester according to any of clauses 12 to 17, in which the harvester comprises two stop surfaces positioned on the axis of oscillation of the oscillator assembly, and in which the cushioning member is positioned on the axis of oscillation between the two stop surfaces.

19. A vibrational energy harvester according to any of clauses 12 to 17, in which the harvester comprises two cushioning members positioned on the axis of oscillation of the oscillator assembly, and two stop surfaces positioned on the axis of oscillation between the two cushioning members.

20. A vibrational energy harvester according to clause 11, in which the amplitude restrictor comprises a pair of magnets with opposing magnetic fields, the first part of the amplitude restrictor comprising one of the pair of magnets fixed relative to the frame, and the second part of the amplitude restrictor comprising a magnet fixed to the oscillator assembly, in which the magnets are configured to repel one another when the oscillator assembly oscillates at the predetermined maximum amplitude.

21. A vibrational energy harvester, comprising:
a frame;
a first oscillator assembly, coupled to the frame and configured to oscillate relative to the frame;
a second oscillator assembly, coupled to the frame and configured to oscillate relative to the frame and the first oscillator assembly; and
a cushioning means;
in which the cushioning means is positioned between the first oscillator assembly and the second oscillator assembly and configured to cushion impacts between the oscillator assemblies during oscillation.

22. A vibrational energy harvester according to clause 21, in which the cushioning means comprises a cushioning member and a stop surface, one of the cushioning member and the stop surface being positioned on the first oscillator assembly, and the other being positioned on the second oscillator assembly, in which the cushioning member and the stop surface are configured to contact one another to cushion impacts between the oscillator assemblies.

23. A vibrational energy harvester according to clause 22, in which the cushioning member is configured to absorb and dissipate kinetic energy from the oscillator assemblies when the oscillator assemblies contact one another.

24. A vibrational energy harvester according to clause 22, in which the cushioning member is configured to absorb kinetic energy from the oscillator assemblies when the oscillator assemblies contact one another, and to re-release the absorbed kinetic energy to both oscillator assemblies following impact.

25. A vibrational energy harvester according to clause 24, in which the cushioning member comprises a spring configured to be compressed by the stop surface when the oscillator assemblies contact one another.

26. A vibrational energy harvester according to clause 24, in which the cushioning member comprises a resilient material, for example silicone rubber.

27. A vibrational energy harvester according to clause 26, in which the cushioning member comprises an O-ring mounted on one of the oscillator assemblies.

28. A vibrational energy harvester according to clause 24, in which the cushioning member comprises a first magnet, and in which a second magnet is mounted on the stop surface and configured to repel the first magnet when the oscillator assemblies approach one another.

29. A vibrational energy harvester according to any of clauses 22 to 28, in which a first cushioning member, or a first portion of the cushioning member, is mounted at a first position on the first oscillator assembly, and a second cushioning member, or a second portion of the cushioning member, is mounted on the first oscillator assembly at a second position spaced from the first position along the axis of oscillation of the first oscillator assembly.

30. A vibrational energy harvester according to any of clauses 22 to 29, in which the first oscillator assembly and the second oscillator assembly oscillate along the same axis of oscillation, one of the oscillator assemblies comprising two stop surfaces positioned on the axis of oscillation, and the other oscillator assembly comprising the cushioning member positioned on the axis of oscillation between the two stop surfaces.

31. A vibrational energy harvester according to any of clauses 22 to 29, in which the first oscillator assembly and the second oscillator assembly oscillate along the same axis of oscillation, one of the oscillator assemblies comprising two cushioning members positioned on the axis of oscillation, and the other oscillator assembly comprising two stop surfaces positioned on the axis of oscillation between the two cushioning members.

Claims

1. A vibrational energy harvester, comprising:
   a first flexure assembly and a conductive coil, the coil being fixed to the first flexure assembly;
   a second flexure assembly and a magnet, the magnet being fixed to the second flexure assembly;
   in which the first flexure assembly has a first resonant frequency in which the coil oscillates relative to the magnet back and forth along a first direction; and the second flexure assembly has a second resonant frequency different from the first resonant frequency in which the magnet oscillates relative to the coil back and forth along the first direction, so that relative movement of the coil and the magnet induces an electrical current in the coil; and
   in which the harvester comprises a displacement limiter for limiting the displacement of the coil relative to the magnet in the first direction.

2. A vibrational energy harvester according to claim 1, in which the displacement limiter comprises a cushioning member and a stop surface, configured so that the cushioning member contacts the stop surface when the relative displacement of the magnet and the coil reaches a predetermined maximum displacement.

3. A vibrational energy harvester according to claim 2, in which one of the cushioning member and the stop surfaces is provided on the first flexure assembly, and the other of the cushioning member and the stop surfaces is provided on the second flexure assembly.

4. A vibrational energy harvester according to claim 2 or 3, in which two stop surfaces are provided, to limit relative displacement in both directions of oscillation back and forth along the first direction.

5. A vibrational energy harvester according to claim 2, 3 or 4, in which the cushioning member is configured to absorb and dissipate kinetic energy from the flexure assemblies when the cushioning means and the stop surface contact one another.

6. A vibrational energy harvester according to claim 2, 3 or 4, in which the cushioning member is configured to absorb kinetic energy from the oscillator assemblies when

the cushioning means and the stop surface contact one another, and to re-release the absorbed kinetic energy to both flexure assemblies following impact.

7. A vibrational energy harvester according to claim 6, in which the cushioning member comprises a spring configured to be compressed by the stop surface when the oscillator assemblies contact one another.

8. A vibrational energy harvester according to claim 6, in which the cushioning member comprises a resilient material, for example silicone rubber.

9. A vibrational energy harvester according to claim 8, in which the cushioning member comprises an O-ring.

10. A vibrational energy harvester according to any of claims 2 to 9, in which the displacement limiter comprises a first cushioning member and a first stop surface configured to contact one another when the first flexure assembly reaches the predetermined maximum displacement moving one way along the axis of oscillation, and a second cushioning member and a second stop surface configured to contact one another when the first flexure assembly reaches the predetermined maximum displacement moving the other way along the axis of oscillation.

11. A vibrational energy harvester according to claim 10, in which one of the first cushioning member and the first stop surface is provided on the first flexure assembly and the other of the first cushioning member and a first stop surface is provided on the second flexure assembly.

12. A vibrational energy harvester according to claim 10 or 11, in which one of the second cushioning member and second stop surface is provided on the first flexure assembly and the other of the second cushioning member and second stop surface is provided on the second flexure assembly.

13. A vibrational energy harvester according to claim 10, 11 or 12, in which one of the first or second flexure assemblies comprises the first cushioning member, and the second cushioning member spaced from the first cushioning member in the first direction.

14. A vibrational energy harvester according to claim 13, in which the other of the first or second flexure assemblies comprises the first stop surface, and the second stop surface spaced from the first stop surface in the first direction.

15. A vibrational energy harvester according to claim 10, 11 or 12, in which one of the first or second flexure assemblies comprises the first cushioning means and the second stop surface spaced from the first cushioning means in the first direction, while the other of the first or second flexure assemblies comprises the second cushioning means and the first stop surface spaced from the second cushioning means in the first direction.

16. A vibrational energy harvester according to any preceding claim, in which the displacement limiter is configured to restrict the relative displacement of the coil and the magnet in the first direction to ± 5 mm or less, or ± 2.5 mm or less, or ± 1.5 mm or less, or ± 1 mm or less.

17. A vibrational energy harvester according to any preceding claim, in which the harvester comprises a frame, and additionally comprises an amplitude restrictor configured to restrict the amplitude of oscillation of the first and/or second flexure assemblies to a predetermined maximum amplitude relative to the frame.

18. A vibrational energy harvester according to claim 17, in which the amplitude restrictor comprises a cushioning member and a stop surface, one of the cushioning member and the stop surface being fixed relative to the frame, and the other being positioned on a flexure assembly, in which the cushioning member and the stop surface are configured to contact one another when the flexure assembly reaches the predetermined maximum amplitude.

19. A vibrational energy harvester according to claim 17, in which the cushioning member is configured to absorb and dissipate kinetic energy from the oscillator assembly when the flexure assembly reaches the predetermined maximum amplitude.

20. A vibrational energy harvester according to claim 17, in which the cushioning member is configured to absorb and re-release kinetic energy from the oscillator assembly when the flexure assembly reaches the predetermined maximum

amplitude.

21. A vibrational energy harvester according to clause 20, in which the cushioning member comprises a spring configured to be compressed by the stop surface when the oscillations of the flexure assembly reach the maximum amplitude.

22. A vibrational energy harvester according to clause 20, in which the cushioning member comprises a resilient material, for example silicone rubber, preferably in which the cushioning member comprises an O-ring mounted on the frame or the oscillator assembly.

23. A vibrational energy harvester according to any of clauses 17 to 22, in which the harvester comprises two stop surfaces positioned on the axis of oscillation of the flexure assembly, and in which the cushioning member is positioned on the axis of oscillation between the two stop surfaces.

24. A vibrational energy harvester according to any of clauses 17 to 22, in which the harvester comprises two cushioning members positioned on the axis of oscillation of the flexure assembly, and two stop surfaces positioned on the axis of oscillation between the two cushioning members.

25. A vibrational energy harvester according to clause 17, in which the amplitude restrictor comprises a pair of magnets with opposing magnetic fields, one of the pair of magnets being fixed relative to the frame, and the other magnet being fixed to the flexure assembly, in which the magnets are configured to repel one another when the flexure assembly oscillates at the predetermined maximum amplitude.

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Claims searched:	1-25	**Date of search:**	23 February 2021

Patents Act 1977: Search Report under Section 17

Documents considered to be relevant:

Category	Relevant to claims	Identity of document and passage or figure of particular relevance
X	1, 2, 4, 5-8 and 16-22	US 2008/0136562 A1 KULAH: See the abstract, figures 4a-c and paragraphs [0055-57]
A	-	US 2016/0294272 A1 MAK

Categories:

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
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Field of Search:

Search of GB, EP, WO & US patent documents classified in the following areas of the UKC^X :

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H02K; H02N

The following online and other databases have been used in the preparation of this search report

WPI, EPODOC

International Classification:

Subclass	Subgroup	Valid From
H02K	0035/00	01/01/2006
H02K	0001/34	01/01/2006