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(54) Title: DEVICE AND METHOD FOR CONTROLLING VIBRATION WITH PIECEWISE-LINEAR NONLINEARITY

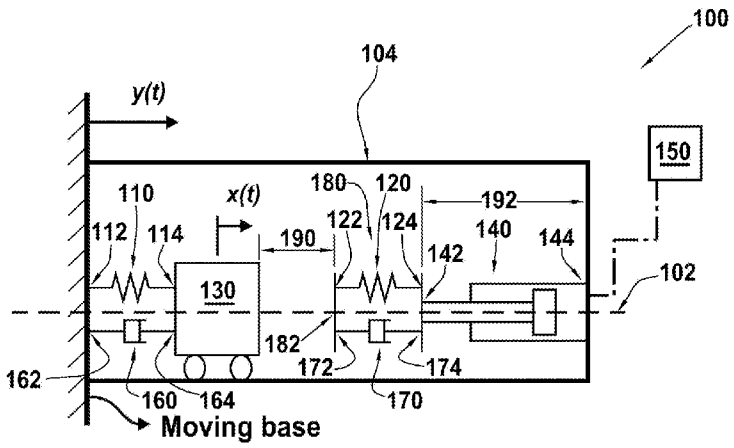


FIG. 1

(57) Abstract: Various implementations include a device for controlling vibration with piecewise-linear nonlinearity. The device includes a stiffness element, a mass, a stopper, and an actuator. The stiffness element is expandable and compressible along an axis. The mass is coupled to the stiffness element. The mass has a resting mass position along the axis. The actuator is coupled to the stopper. The actuator is configured to move the stopper along the axis to vary a gap size. The gap size is measured as a distance between the resting mass position and a resting stopper position.

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DEVICE AND METHOD FOR CONTROLLING VIBRATION WITH PIECEWISE-LINEAR NONLINEARITY

CROSS REFERENCE TO RELATED APPLICATIONS

[0001] This application claims the benefit of U.S. Provisional Patent Application No. 62/881,113, filed July 31, 2019, the content of which is incorporated herein by reference in its entirety.

BACKGROUND

[0002] Vibration energy harvesting is considered one of the most promising green energy sources that can self-power small-scale devices and fulfill the need of large-scale electricity generation. On one hand, as electronic devices are becoming more compact and portable, a sustainable and scalable energy source with high energy density is in high demand. Ambient vibration is an ideal energy source for these systems since it is ubiquitous and easily accessible. On the other hand, large-scale vibrations such as ocean waves are now considered to be a potential energy resource to power entire cities since they contain a remarkable amount of recoverable energy. Thus, vibration energy harvesting is an attractive solution in fulfilling a variety of energy needs and a great amount of research has been conducted to optimize energy extraction from vibration sources.

[0003] Vibration energy harvesting can be realized using piezoelectric, electromagnetic, or electrostatic approaches. Early designs of these power generators were based on linear models that provide the maximum power generation efficiency at resonance with their efficiency decreasing dramatically even when the excitation frequency shifts slightly. Thus, traditional linear energy harvesters are usually limited to very narrow frequency ranges. In order to broaden the frequency bandwidth for effective energy harvesting, different techniques such as nonlinear energy harvesting, array-harvester systems, and frequency-tunable systems have been developed so the harvester can accommodate a broader frequency range. Each technique has its own advantages and disadvantages. For instance, the array-harvester design can harvest the vibration energy over the resonant frequencies of each linear system; however, the system setup and the corresponding electronic configuration are complex, which makes the utilization very challenging. Nonlinear energy harvesters, which are commonly used in piezoelectric and electromagnetic generators, can broaden the effective frequency bandwidth by exploiting geometric and material nonlinearities. However, these nonlinear techniques are not as efficient

as linear energy harvesters at resonance.

[0004] Recently, new designs of energy harvesters with piecewise-linear (PWL) nonlinearities have been proposed. These nonlinear devices incorporate mechanical stoppers into conventional linear energy harvesters. It is found that these PWL harvesters can effectively broaden the effective frequency range for an up-sweep excitation condition. However, these devices do not guarantee the best vibration performance at an arbitrary excitation frequency. Also, they do not achieve the efficiency of linear harvesters at resonance.

[0005] Thus, a need exists for a vibration energy harvester that can perform efficiently at varying frequencies.

SUMMARY

[0006] Various implementations include a device for controlling vibration with piecewise-linear nonlinearity. The device includes a stiffness element, a mass, a stopper, and an actuator. The stiffness element is expandable and compressible along an axis. The mass is coupled to the stiffness element. The mass has a resting mass position along the axis. The actuator is coupled to the stopper. The actuator is configured to move the stopper along the axis to vary a gap size. The gap size is measured as a distance between the resting mass position and a resting stopper position.

[0007] In some implementations, the stiffness element is a spring. In some implementations, the spring is a linear spring. In some implementations, the stiffness element is a cantilevered beam. In some implementations, the stiffness element is a coil of wire.

[0008] In some implementations, the device includes a damping element coupled to the mass. The damping element is expandable and compressible along the axis. In some implementations, the damping element is a linear viscous damper.

[0009] In some implementations, the stopper comprises a rigid material. In some implementations, the stopper comprises a stopper stiffness element. In some implementations, the stopper stiffness element is a stopper spring. In some implementations, the stopper spring is a linear spring. In some implementations, the stopper stiffness element is a cantilevered beam. In some implementations, the stopper stiffness element is a coil of wire.

[0010] In some implementations, the stopper includes a stopper damping element. In some implementations, the stopper damping element is a linear viscous damper.

[0011] In some implementations, the device includes a controller configured to determine an optimized gap size based on an identified dominant frequency and amplitude of an

excitation signal and to cause the actuator to move the stopper along the axis toward the optimized gap size. In some implementations, the optimized gap size is determined using bilinear amplitude approximation (BAA). In some implementations, the optimized gap size is determined using non-dimensional calculations.

[0012] In some implementations, the gap size is a negative distance such that the stopper prestresses the stiffness element.

[0013] Various other implementations include a method of controlling vibration with piecewise-linear nonlinearity. The method includes (a) introducing a device like the device described above for controlling vibration with piecewise-linear nonlinearity to an excitation signal, (b) measuring or computing the excitation signal, (c) identifying a dominant frequency and amplitude of the excitation signal, (d) determining an optimized gap size based on the identified dominant frequency and amplitude of the excitation signal, and (e) moving the stopper along the axis toward the optimized gap size.

[0014] In some implementations, the method further includes repeating steps (b)-(e).

[0015] In some implementations, the initial gap size is zero.

[0016] In some implementations, the stiffness element is a spring. In some implementations, the spring is a linear spring. In some implementations, the stiffness element is a cantilevered beam. In some implementations, the stiffness element is a coil of wire.

[0017] In some implementations, the device further includes a damping element coupled to the mass. The damping element being expandable and compressible along the axis. In some implementations, the damping element is a linear viscous damper.

[0018] In some implementations, the stopper includes a rigid material. In some implementations, the stopper includes a stopper stiffness element. In some implementations, the stopper stiffness element is a stopper spring. In some implementations, the stopper spring is a linear spring. In some implementations, the stopper stiffness element is a cantilevered beam. In some implementations, the stopper stiffness element is a coil of wire.

[0019] In some implementations, the stopper includes a stopper damping element. In some implementations, the stopper damping element is a linear viscous damper.

[0020] In some implementations, the device further includes a controller configured to determine the optimized gap size. In some implementations, the optimized gap size is determined using bilinear amplitude approximation (BAA). In some implementations, the optimized gap size is determined using non-dimensional calculations.

[0021] In some implementations, the gap size is a negative distance such that the stopper

prestresses the stiffness element.

[0022] Various other implementations include a device for controlling vibration with piecewise-linear nonlinearity. The device includes one or more stiffness elements, one or more masses, one or more stoppers, and one or more actuators. Each of the one or more stiffness elements is expandable and compressible along an axis. Each of the one or more masses is coupled to one or more stiffness elements. Each of the one or more masses has a resting mass position along its axis. Each of the one or more actuators is coupled to one of the one or more stoppers. Each of the one or more actuators is configured to move the one of the one or more stoppers along the axis of one of the one or more stiffness elements to vary a gap size. The gap size is measured as a distance between the resting mass position and a resting stopper position of the one of the one or more stoppers.

[0023] In some implementations, at least one of the stiffness elements is a spring. In some implementations, at least one of the springs is a linear spring. In some implementations, at least one of the stiffness elements is a cantilevered beam. In some implementations, at least one of the stiffness elements is a coil of wire.

[0024] In some implementations, the device includes one or more damping elements. Each of the one or more damping elements is coupled to one of the one or more masses. The one or more damping elements are expandable and compressible along the axis. In some implementations, at least one of the damping elements is a linear viscous damper.

[0025] In some implementations, at least one of the stoppers comprises a rigid material. In some implementations, at least one of the stoppers comprises a stopper stiffness element. In some implementations, at least one of the stopper stiffness elements is a stopper spring. In some implementations, at least one of the stopper springs is a linear spring. In some implementations, at least one of the stopper stiffness elements is a cantilevered beam. In some implementations, at least one of the stopper stiffness elements is a coil of wire.

[0026] In some implementations, at least one of the stoppers includes a stopper damping element. In some implementations, at least one of the stopper damping elements is a linear viscous damper.

[0027] In some implementations, the device includes a controller configured to determine an optimized gap size between each of the masses and corresponding stopper based on one or more identified frequencies and amplitudes of one or more excitation signals and to cause the one or more actuators to move the corresponding stoppers along the axes toward the optimized gap sizes. In some implementations, the optimized gap sizes are determined using

bilinear amplitude approximation (BAA). In some implementations, the optimized gap sizes are determined using non-dimensional calculations.

[0028] In some implementations, at least one of the gap sizes is a negative distance such that one of the stoppers prestresses the corresponding stiffness element.

[0029] Various other implementations include a method of controlling vibration with piecewise-linear nonlinearity. The method includes (a) introducing a device like the device described above for controlling vibration with piecewise-linear nonlinearity to one or more excitation signals, (b) measuring or computing each of the excitation signals, (c) identifying one or more frequencies and amplitudes of each of the excitation signals, (d) determining an optimized gap size for each of the masses and corresponding stopper based on the identified one or more frequencies and amplitudes of each of the excitation signals, and (e) moving each of the stoppers along the corresponding axis toward the corresponding optimized gap size.

[0030] In some implementations, the method includes repeating steps (b)-(e).

[0031] In some implementations, at least one of the initial gap sizes is zero.

[0032] In some implementations, at least one of the stiffness elements is a spring. In some implementations, at least one of the springs is a linear spring. In some implementations, at least one of the stiffness elements is a cantilevered beam. In some implementations, at least one of the stiffness elements is a coil of wire.

[0033] In some implementations, the device includes one or more damping elements. Each of the one or more damping elements is coupled to one of the one or more masses. The one or more damping elements are expandable and compressible along the axis. In some implementations, at least one of the damping elements is a linear viscous damper.

[0034] In some implementations, at least one of the stoppers comprises a rigid material. In some implementations, at least one of the stoppers comprises a stopper stiffness element. In some implementations, at least one of the stopper stiffness elements is a stopper spring. In some implementations, at least one of the stopper springs is a linear spring. In some implementations, at least one of the stopper stiffness elements is a cantilevered beam. In some implementations, at least one of the stopper stiffness elements is a coil of wire.

[0035] In some implementations, at least one of the stoppers includes a stopper damping element. In some implementations, at least one of the stopper damping elements is a linear viscous damper.

[0036] In some implementations, the device includes a controller configured to determine an optimized gap size between each of the masses and corresponding stopper. In

some implementations, the optimized gap sizes are determined using bilinear amplitude approximation (BAA). In some implementations, the optimized gap sizes are determined using non-dimensional calculations.

[0037] In some implementations, at least one of the gap sizes is a negative distance such that one of the stoppers prestresses the corresponding stiffness element.

BRIEF DESCRIPTION OF DRAWINGS

[0038] Example features and implementations are disclosed in the accompanying drawings. However, the present disclosure is not limited to the precise arrangements and instrumentalities shown.

[0039] FIG. 1 is a side schematic view of a device for controlling vibration with piecewise-linear nonlinearity, according to one implementation.

[0040] FIG. 2 is a side schematic view of a device for controlling vibration with piecewise-linear nonlinearity, according to another implementation.

[0041] FIG. 3 is a graph of a nonlinear vibration cycle of the device of FIG. 1.

[0042] FIGS. 4(a)-4(d) are graphs of the performance and stability of the device of FIG. 1.

[0043] FIG. 5(a) is a graph of the δ -sweep process of the device of FIG. 1. FIG. 5(b) is a graph of the maximum response envelope and peak gap size over the bounded frequency range for the device of FIG. 1.

[0044] FIG. 6 is a flow chart and graph of the control process for optimizing the vibration performance of the device of FIG. 1.

[0045] FIGS. 7(a)-(d) are graphs of numerical simulation with unperturbed stationary excitation, according to another implementation.

[0046] FIGS. 8(a)-(d) are graphs of numerical simulation with perturbed stationary excitation, according to another implementation.

[0047] FIGS. 9(a)-(d) are graphs of numerical simulation with drifting excitation, according to another implementation.

[0048] FIGS. 10(a)-(d) are graphs of numerical simulation with abruptly changing excitation, according to another implementation.

DETAILED DESCRIPTION

[0049] The devices, systems, and methods disclosed herein provide a new way to efficiently control vibration by manipulating gaps of PWL nonlinear systems. The devices, systems, and methods can be exploited to design next-generation energy harvesters, vibration isolators, and vibration absorbers. Traditional linear vibration enhancement/reduction devices have the best performance when operating at resonant frequencies, while being limited to very narrow frequency ranges about these resonances. On the other hand, nonlinear devices can broaden the effective frequency bandwidth by exploiting different nonlinearities such as geometric or material nonlinearities. However, these nonlinear techniques are not as efficient as linear devices at resonance. The disclosed devices, systems, and methods provide for a novel solution for vibration enhancement and reduction by incorporating PWL nonlinearities into system design. The new designs incorporate mechanical stoppers or additional sets of springs and dampers into traditional linear systems. The resonant frequencies and amplitude of these systems can then be manipulated by adjusting the gap size between mechanical elements. The benefit of utilizing PWL nonlinearity in these systems is that a wide frequency range can be covered by shifting the resonant frequency through actively controlling the gap size in these systems while maintaining the high performance of a linear system at resonance.

[0050] Various implementations include a new vibration harvester composed of a PWL oscillator and a controllable gap. The resonant frequency of the harvester device can be tuned to match the dominant frequency of the excitation signal to provide the instantaneous optimized vibration performance. The frequency tunability is enabled by adjusting the gap size in the PWL oscillator.

[0051] Furthermore, a control method combining the response prediction, signal measurement, and gap adjustment mechanism is proposed to optimize the system's performance. In this control method, a response approximation technique referred to as the bilinear amplitude approximation (BAA) method is first used to compute the gap size that can tune the system to resonance over an effective frequency range. BAA is used to efficiently capture the nonlinear response of PWL nonlinear systems using linear techniques. Frequency and amplitude estimators can be employed to analyze the excitation signal over a specified time window. The gap in the PWL oscillator is then adjusted to the appropriate size that can optimize the vibration performance based on the measured excitation signal and the pre-computed optimized gap size. The proposed energy harvesting strategy has a better performance than current PWL harvesters since it provides a better vibration performance for both stationary excitation and drifting

excitation conditions.

[0052] Various implementations include a device for controlling vibration with piecewise-linear nonlinearity. The device includes a stiffness element, a mass, a stopper, and an actuator. The stiffness element is expandable and compressible along an axis. The mass is coupled to the stiffness element. The mass has a resting mass position along the axis. The actuator is coupled to the stopper. The actuator is configured to move the stopper along the axis to vary a gap size. The gap size is measured as a distance between the resting mass position and a resting stopper position.

[0053] Various other implementations include a method of controlling vibration with piecewise-linear nonlinearity. The method includes (a) introducing a device like the device described above for controlling vibration with piecewise-linear nonlinearity to an excitation signal, (b) measuring or computing the excitation signal, (c) identifying a dominant frequency and amplitude of the excitation signal, (d) determining an optimized gap size based on the identified dominant frequency and amplitude of the excitation signal, and (e) moving the stopper along the axis toward the optimized gap size.

[0054] Various other implementations include a device for controlling vibration with piecewise-linear nonlinearity. The device includes one or more stiffness elements, one or more masses, one or more stoppers, and one or more actuators. Each of the one or more stiffness elements is expandable and compressible along an axis. Each of the one or more masses is coupled to one or more stiffness elements. Each of the one or more masses has a resting mass position along its axis. Each of the one or more actuators is coupled to one of the one or more stoppers. Each of the one or more actuators is configured to move the one of the one or more stoppers along the axis of one of the one or more stiffness elements to vary a gap size. The gap size is measured as a distance between the resting mass position and a resting stopper position of the one of the one or more stoppers.

[0055] Various other implementations include a method of controlling vibration with piecewise-linear nonlinearity. The method includes (a) introducing a device like the device described above for controlling vibration with piecewise-linear nonlinearity to one or more excitation signals, (b) measuring or computing each of the excitation signals, (c) identifying one or more frequencies and amplitudes of each of the excitation signals, (d) determining an optimized gap size for each of the masses and corresponding stopper based on the identified one or more frequencies and amplitudes of each of the excitation signals, and (e) moving each of the stoppers along the corresponding axis toward the corresponding optimized gap size.

[0056] FIG. 1 shows a device 100 for controlling vibration with piecewise-linear nonlinearity. The device 100 includes a housing 104, a mass 130, a stiffness element 110, a damping element 160, a stopper 180, an actuator 140, and a controller 150.

[0057] The stiffness element 110 shown in FIG. 1 is a linear spring having a first end 112 and a second end 114 opposite and spaced apart from the first end 112 of the stiffness element 110. The first end 112 of the stiffness element 110 is coupled to the housing 104 and the second end 114 of the stiffness element 110 is coupled to the mass 130.

[0058] The damping element 160 shown in FIG. 1 is a linear viscous damper having a first end 162 and a second end 164 opposite and spaced apart from the first end 162 of the damping element 160. The first end 162 of the damping element 160 is coupled to the housing 104 and the second end 164 of the damping element 160 is coupled to the mass 130.

[0059] The stiffness element 110 and the damping element 160 are aligned such that each are expandable and compressible along, or parallel to, the same axis 102. Thus, the mass 130 coupled to the second end 114 of the stiffness element 110 and the second end 164 of the damping element 160 is movable along, or parallel to, the axis 102 by expanding and/or compressing the stiffness element 110 and the damping element 160.

[0060] The stiffness element 110, damping element 160, and mass 130 are selected for the device 100 based on the expected excitation frequencies and amplitudes of the system into which the device 100 is designed to be introduced.

[0061] The stopper 180 shown in FIG. 1 includes a stopper stiffness element 120 and a stopper damping element 170. The stopper stiffness element 120 is a linear spring having a first end 124 and a second end 122 opposite and spaced apart from the first end 124 of the stopper stiffness element 120. The first end 124 of the stopper stiffness element 120 is coupled to the actuator 140 and the second end 122 of the stopper stiffness element 120 is coupled to a stopper plate 182.

[0062] The stopper damping element 170 shown in FIG. 1 is a linear viscous damper having a first end 174 and a second end 172 opposite and spaced apart from the first end 174 of the stopper damping element 170. The first end 174 of the stopper damping element 170 is coupled to the actuator 140 and the second end 172 of the stopper damping element 170 is coupled to the stopper plate 182.

[0063] The stopper stiffness element 120 and the stopper damping element 170 are aligned such that each are expandable and compressible along, or parallel to, the axis 102. Thus, as the mass 130 moves along the axis 102, the mass 130 moves away from and toward the

stiffness plate 182 and can contact the stiffness plate 182 to compress each of the stopper stiffness element 120 and the stopper damping element 170.

[0064] The mass 130 has a resting mass position along the axis 102 at which the stiffness element 110 and damping element 160 are neither expanded nor compressed. The stopper plate 182 has a resting stopper position along the axis 102 at which the stopper stiffness element 120 and stopper damping element 170 are neither expanded nor compressed. The distance between the mass 130 at the resting mass position and the stopper 180 at the resting stopper position is the gap size 190. The gap size 190 can either be a positive number in which the mass 130 and stopper 180 are spaced apart, a negative number in which the mass 130 and stopper 180 abut each other and are prestressing the stiffening element 110 and damping element 160, or zero in which the mass 130 and stopper 180 abut each other but are not compressing the stiffening element 110 or damping element 160.

[0065] The actuator 140 shown in FIG. 1 is a linear actuator. The actuator 140 has a first end 142 and a second end 144 opposite and spaced apart from the first end 142 of the actuator 140. The stopper 180 is coupled to the first end 142 of the actuator 140 and the second end 144 of the actuator 140 is coupled to the housing 104. The actuator 140 is actuatable along, or parallel to, the axis 102 to move the stopper 180 closer or further from the mass 130 to vary the gap size 190.

[0066] The controller 150 is configured to determine an optimized gap size and actuate the actuator 140 to move the stopper 180 relative to the mass 130, thus varying the gap size 190 closer to the optimized gap size. When the device 100 is introduced to an excitation signal of a system, the starting gap size is zero to ensure that the mass 130 and the stopper 180 are in contact with each other. The controller 150 measures the excitation signal, if possible, using an accelerometer. In some systems, the excitation signal cannot be measured directly and must instead be computed based on the response of the system. The controller 150 uses the measured or computed excitation signal to identify a dominant frequency and amplitude of the excitation signal. The controller 150 then uses bilinear amplitude approximation (BAA) to determine the optimized gap size based on the identified dominant frequency and amplitude of the excitation signal, as discussed below. The controller 150 then sends a signal to actuate the actuator 140 to move the stopper 180 along the axis 102 in the direction of the optimized gap size. The controller 150 continuously iterates this process to vary the gap size 190 in the direction of the optimized gap size. By adjusting the gap size 190 to the optimized gap size, the vibration performance of the device 100 can be optimized.

[0067] FIG. 2 shows a device 200 for controlling vibration with piecewise-linear nonlinearity, according to another implementation. The device 200 can be introduced into a system having two or more excitation signals and can optimize the energy harvesting of the two or more excitation signals. The device 200 includes a first system including a first mass 230, a first stiffness element 210, a first damping element 260, a first stopper 280, and a first actuator 240 and a second system including a second mass 230', a second stiffness element 210', a second damping element 260', a second stopper 280', and a second actuator 240'. The first stopper 280 includes a first stopper stiffness element 220 and a first damping element 270, and the second stopper 280' includes a second stopper stiffness element 220' and a second damping element 270'. The first mass 230, first stiffness element 210, first damping element 260, first stopper 280, and first actuator 240 each move along, or parallel to, a first axis 202, and the second mass 230', second stiffness element 210', second damping element 260', second stopper 280', and second actuator 240' each move along, or parallel to, a second axis 202'. The first and second axes are transverse to each other. The first and second systems are disposed within the housing 204 and both systems are controlled by the controller 250.

[0068] Using the same process described above, the controller 250 determines the optimized gap size of each of the systems and actuates the first and second actuators 240, 240' to vary the first and second gap sizes 290, 290' toward the corresponding optimized gap sizes. Each of the optimized gap sizes can be determined based on the frequency and amplitude of two separate excitation signals or can be determined based on multiple frequencies and amplitudes within the same excitation signal.

[0069] Although the device shown in FIG. 2 has two systems along two different axes, in other implementations, the device includes three or more systems and the components of each system move along different axes. In some implementations, all of the axes are transverse to each other. In some implementations, one or more axes are parallel to each other.

[0070] Although the stiffness elements and the stopper stiffness elements shown in FIGS. 1 and 2 are all linear springs, in other implementations, the stiffness element and/or the stopper stiffness element are a non-linear spring, any other type of spring, a cantilevered beam, a coil of wire, an elastic band, or any device capable of resiliently deflecting under a load. The stoppers shown in FIGS. 1 and 2 all include a stopper stiffness element and a stopper damping element, but in other implementations, the stopper includes a rigid material and does not include a stopper stiffness element and a stopper damping element.

[0071] Although the damping elements and stopper damping elements shown in FIGS. 1

and 2 are all linear viscous dampers, in other implementations, the damping element is any device capable of dissipating the mechanical energy of the mass into another form of energy (e.g., heat energy). In some implementations, the device does not include a damping element or damping is provided by the inherent damping of the stiffening element or other features of the device.

[0072] In some implementations, a system within a device can include more than one stiffness element, damping element, stopper stiffness element, and/or stopper damping element.

[0073] Although the devices shown in FIGS. 1 and 2 are energy harvesting devices, in other implementations, the devices are vibration absorbers and the optimized gap size is a resonant frequency for an excitation signal.

[0074] The equations of motion for the system of the device 100 shown in FIG. 1 can be expressed as

$$\begin{aligned} m\ddot{\overline{x}}_c(t) + (c+c^*)\dot{\overline{x}}_c(t) + (k+k^*)\overline{x}_c(t) &= -m\ddot{y}(t) + k^*g \quad \text{when } \overline{x} \geq g, \\ m\ddot{\overline{x}}_o(t) + c\dot{\overline{x}}_o(t) + k\overline{x}_o(t) &= -m\ddot{y}(t) \quad \text{when } \overline{x} < g, \end{aligned} \tag{2.1}$$

[0075] where m is the mass 130, k is the linear spring coefficient for the stiffness element 110, c is the damper coefficient for the damping element 160, k^* is the linear spring coefficient for the stopper stiffness element 120, c^* is the damper coefficient for the stopper damping element 170, $y(t)$ is the displacement of the device 100 by the excitation signal, $x(t)$ is the displacement of mass m 130 along the axis 102, and g is the gap size 190. $\overline{x} = x - y$ represents the relative displacement between the mass m 130 and the moving device 100. Note that the subscript c represents the displacement of the mass 130 when the gap 190 is closed, i.e., $\overline{x} \geq g$, and the subscript o represents the displacement of the mass 130 when the gap 190 is open, i.e., $\overline{x} < g$. Eqn. (2.1) can then be rewritten as

$$\begin{aligned} \ddot{\overline{x}}_c(t) + 2(\zeta\omega + \zeta^*\omega^*)\dot{\overline{x}}_c(t) + (\omega^2 + \omega^{*2})\overline{x}_c(t) &= -\ddot{y}(t) + \omega^{*2}g \quad \text{when } \overline{x} \geq g, \\ \ddot{\overline{x}}_o(t) + 2\zeta\omega\dot{\overline{x}}_o(t) + \omega^2\overline{x}_o(t) &= -\ddot{y}(t) \quad \text{when } \overline{x} < g, \end{aligned} \tag{2.2}$$

[0076] where $\omega^2 = \frac{k}{m}$, $\omega^{*2} = \frac{k^*}{m}$, $\zeta = \frac{c}{2m\omega}$, and $\zeta^* = \frac{c^*}{2m\omega^*}$. Herein, it is assumed that the system is driven by a harmonic excitation signal $y(t) = y_0 \sin(\alpha t)$, where

y_0 is the excitation amplitude and α is the excitation frequency.

[0077] Next, dimensionless variables are introduced to simplify the subsequent analysis. First, the time-related dimensionless variables are introduced:

$$\tau = \omega t, \quad \rho = \frac{\alpha}{\omega}, \quad \rho^* = \frac{\omega^*}{\omega}, \tag{2.3}$$

[0078] Eqn. (2.2) can then be expressed as

$$\begin{aligned} \overline{x}_o''(\tau) + 2(\zeta + \zeta^*\rho^*)\overline{x}_o'(\tau) + (1+\rho^{*2})\overline{x}_o(\tau) &= \rho^2 y_0 \sin(\rho\tau) + \rho^{*2} g \quad \text{when } \overline{x} \geq g, \\ \overline{x}_o''(\tau) + 2\zeta\overline{x}_o'(\tau) + \overline{x}_o(\tau) &= \rho^2 y_0 \sin(\rho\tau) \quad \text{when } \overline{x} < g, \end{aligned} \tag{2.4}$$

[0079] where the prime symbol (') indicates differentiation with respect to the dimensionless time variable τ. Finally, the following spatial dimensionless variables are used:

$$u = \frac{\overline{x}}{y_0}, \quad \delta = \frac{g}{y_0}, \tag{2.5}$$

[0080] The dimensionless equations of motion can be written as

$$\begin{aligned} u_o''(\tau) + 2(\zeta + \zeta^*\rho^*)u_o'(\tau) + (1+\rho^{*2})u_o(\tau) &= \rho^2 \sin(\rho\tau) + \rho^{*2}\delta \quad \text{when } u \geq \delta, \\ u_o''(\tau) + 2\zeta u_o'(\tau) + u_o(\tau) &= \rho^2 \sin(\rho\tau) \quad \text{when } u < \delta. \end{aligned} \tag{2.6}$$

[0081] Note that the nondimensionalization allows pre-computation of all the responses for all base displacement levels since the response has been scaled by y_0. This enables offline pre-computation of all information needed in the control method described herein. Moreover, scaling the frequency by ω enables efficient design of the parameters of the harvester system without reconducting the analysis.

[0082] In order to efficiently analyze the steady-state dynamics of the device 100, the BAA method is applied to find the solution to Eqn. (2.6). First, the coordinates of the device 100 in its closed and open state are analytically expressed as combinations of the linear transient

response and the linear steady-state response:

$$\begin{aligned} u_c(\tau) &= e^{-\overline{\zeta}\overline{\rho}\tau} a_c \sin\left(\sqrt{1-\overline{\zeta}^2}\overline{\rho}\tau + \phi_c\right) + \frac{(\rho/\overline{\rho})^2 \sin(\rho\tau - \theta_c + \psi)}{\sqrt{[1-(\rho/\overline{\rho})^2]^2 + (2\overline{\zeta}\rho/\overline{\rho})^2}} + \delta\left(\frac{\rho^{*2}}{1+\rho^{*2}}\right), \\ u_o(\tau) &= e^{-\zeta\tau} a_o \sin\left(\sqrt{1-\zeta^2}\tau + \phi_o\right) + \frac{\rho^2 \sin(\rho\tau - \theta_o + \psi)}{\sqrt{(1-\rho^2)^2 + (2\zeta\rho)^2}}, \end{aligned} \tag{3.1}$$

[0083] where $\overline{\rho} = \sqrt{1+\rho^{*2}}$, $\overline{\zeta} = \frac{\zeta + \zeta^*\rho^*}{\sqrt{1+\rho^{*2}}}$, $\theta_c = \tan^{-1}\left(\frac{2\overline{\zeta}\rho\overline{\rho}}{\overline{\rho}^2 - \rho^2}\right)$, and $\theta_o = \tan^{-1}\left(\frac{2\zeta\rho}{1-\rho^2}\right)$; $[a_c, a_o]$ and $[\phi_c, \phi_o]$ are scalar coefficients and phase angles of the linear transient responses, respectively. The angle ψ reflects the phase difference between the excitation and the linear steady-state responses.

[0084] The key idea of the BAA method is that an entire vibration cycle of the PWL oscillator can be obtained by coupling the responses in the closed and open states. The motion of one vibration cycle is schematically shown in FIG. 3. The BAA method assumes that a single vibration cycle has only one time interval when the response is in the closed state and only one time interval when the response is in the open state. These time intervals are denoted using T_c and T_o, respectively. Note that $T = T_c + T_o$ is the period of the harmonic excitation. Next, a nonlinear optimization solver is used to solve for the unknowns a_c, a_o, ϕ_c, ϕ_o, and ψ in Eqn. (3.1) by minimizing the residual of a set of compatibility conditions listed as follows:

$$\begin{aligned} u_c(0) &= \delta, \\ u_o(T_c) &= \delta, \\ u_o(T_c) &= \delta, \\ u_o(T_c + T_o) &= \delta, \\ u_c'(T_c) &= u_o'(T_c), \\ u_c'(0) &= u_o'(T_c + T_o). \end{aligned} \tag{3.2}$$

[0085] The first four equations in Eqn. (3.2) represent the displacement compatibility conditions at the transition moment when the dimensionless displacement u equals the dimensionless gap size δ when the system switches from one state to the other. The last two equations in Eqn. (3.2) represent the velocity compatibility conditions whereby the velocity of

the mass 130 must be continuous at the moment of transition. Note that T_c in Eqn. (3.2) is also an unknown since the time fraction that the system stays in two linear states cannot be predetermined. The function "lsqnonlin" in MATLAB can be used to solve for all the unknown parameters. One nonlinear vibration cycle can then be constructed once these unknowns are solved. The detailed description of the BAA method can be found in Tien MH, D'Souza K., "A generalized bilinear amplitude and frequency approximation for piecewise-linear nonlinear systems with gaps or prestress," Nonlinear Dynamics 88, 2403–2416 (2017), which is incorporated in its entirety by reference.

[0086] The nonlinear forced responses of the PWL oscillator device for different gap sizes δ are computed using the BAA method and plotted in FIG. 4(a). The system parameters used in this work are $\zeta = 0.015, \zeta^* = 0.015, \text{and } \rho^* = 2$. The linear response of the system when the gap 190 is always closed and the linear response of the device 100 when the gap 190 is always open are also plotted in FIG. 4(a). FIG. 4(a) shows that the resonant frequency of the system can be shifted by changing the gap size δ. The resonant frequency associated with $\delta = 0$ is referred to as the bilinear frequency (BF). This resonant frequency can be lowered by increasing δ and can be raised by decreasing δ. This implies that the resonant frequency of the harvester can be tuned to match an arbitrary excitation frequency over the frequency range bounded by the natural frequencies of the linear systems by adjusting the gap to the appropriate size. Furthermore, nonlinear characteristics such as multi-stability, instability, and jump phenomenon can be observed when $\delta \neq 0$.

[0087] Next, time integration can be used to investigate the basins of attraction for the system since multiple stable periodic responses can be observed. The Runge-Kutta method and the event function in MATLAB can be used to conduct time integration. The results of $\rho = 1.06$, 1.17, and 1.25 when $\delta = 10$ are plotted in FIGS. 4(b)-4(d). FIGS. 4(b)-4(d) show that the system, when $\delta > 0$, can settle into either the nonlinear response where the intermittent contact happens or the linear response where there is no contact. Similarly, the system can settle into either the nonlinear response or the linear response where the gap 190 is always closed when $\delta < 0$. Furthermore, it is observed that the area of the basins of attraction of the nonlinear response shrinks significantly when ρ approaches the resonant frequency. Although the nonlinearity can provide a larger vibration motion when the excitation frequency is extremely close to the resonant frequency, it is also more likely that the system will jump from the nonlinear response to the linear response when it is slightly perturbed.

[0088] In order to determine the gap size that can tune the system to resonance, the BAA method is employed to conduct two consecutive sweep processes. In the first sweep, the BAA method is used to sweep through the frequency range bounded by the natural frequencies of the linear systems (i.e., open and sliding systems) by setting $\delta = 0$. In this ρ-sweep process, the nonlinear solver is provided with random initial values at the starting frequency point. The solution that results in the minimum residual and that obeys the physical constraints is chosen for the frequency sweep. Next, the peak response and the corresponding gap size at each frequency ρ is found by sweeping through the gap size δ. The δ-sweep process starts from $\delta = 0$ using the ρ-sweep result as the initial value and ends at a δ value where the solution cannot be found using BAA. The results of the δ-sweep process for five different ρ values are shown in FIG. 5(a). The peak responses are indicated using (O)'s in FIG. 5(a). Note that when ρ is greater than the BF, δ is decreased to locate the peak response; while when ρ is less than the BF, δ is increased to find the peak response. The gap size associated with the peak response is referred to as the resonant gap size δ_r. These resonant gap sizes are indicated using downwardly facing arrows in FIG. 5(a). δ_r represents the gap size that can theoretically provide the maximum steady-state vibration amplitude at a specified excitation frequency ρ if the excitation signal is perfectly harmonic. FIG. 5(a) also shows that the vibration displacement amplitude is roughly proportional to δ. By successively applying the δ-sweep process to each excitation frequency ρ, the maximum response and the corresponding δ_r over the bounded frequency range can be efficiently determined using the BAA method. The maximum response envelope and the corresponding δ_r values over the frequency range are plotted in FIG. 5(b).

[0089] With δ_r values computed, the vibration amplitude of the PWL oscillator can be amplified by adjusting the gap size to approach δ_r at any excitation frequency within the bounded frequency range. However, energy harvesters are rarely driven by perfectly a harmonic excitation with a stable frequency and amplitude. Thus, a signal measurement and analysis process and a gap size adjustment strategy are introduced in this work. This control process assumes that the excitation signal is measurable through sensing devices and the gap size can be adjusted instantaneously based on the measured signal. Moreover, the excitation is assumed to contain a single frequency component that dominates the base motion at any given time. However, the excitation is accompanied by perturbation, and the dominant frequency and the corresponding amplitude can drift through time. The control process is described as follows:

(i) The measured excitation signal is collected over a specified time duration T_s.

(ii) The collected signal is analyzed using frequency and amplitude estimators to extract the frequency ω_{est} and amplitude $y_{0,est}$ that dominate the base motion.

(iii) The δ_r associated with $\rho_{est} = \omega_{est}/\omega$ that has been pre-computed using BAA is identified. Note that ρ_{est} generally does not hit a frequency point that has been pre-computed. δ_r at arbitrary ρ_{est} can be obtained by interpolating adjacent δ_r values.

(iv) The optimized gap size δ_{opt} is determined by using only a fraction of δ_r to avoid jumping from the nonlinear response to the linear response: $\delta_{opt} = a\delta_r$, where $0 < a < 1$.

(v) The gap size is adjusted to $g_{opt} = y_{0,est}\delta_{opt}$.

(vi) Steps (i)-(v) are repeated for next time window.

[0090] The dominant frequency and the associated amplitude can be estimated using methods developed by Zhivomirov et al. and Lyons, respectively. These methods build on discrete Fourier transform techniques and estimate the frequency and amplitude by analyzing the windowed signal in the frequency domain. It should also be noted that the system is more likely to jump to the linear response if a is close to 1. The overall control process is summarized in FIG. 6.

EXAMPLES

[0091] In this section, the PWL harvester integrated with the proposed control method is numerically investigated using time integration. Different excitation conditions are applied in case studies to validate the vibration performance of the system.

Example 1

[0092] The system subjected to a perfectly harmonic excitation with fixed excitation frequency and amplitude is studied first. In order to present the vibration amplitude variation, equations of motion with physical spatial variables, i.e., Eqn. (2.4), are used for time integration. The actual excitation amplitude and frequency used in this case study are $y_0 = 1$ and $\rho = 1.15$. Moreover, 94% of the resonant gap size is used as the optimized size to avoid the jump

phenomenon, i.e., $a = 0.94$. Two gap size control strategies are compared in the study: (1) fixing the gap at the optimized size and (2) starting from $g = 0$ and gradually approaching the optimized gap size with the proposed control method. The mass is assumed to have zero initial displacement and zero initial velocity, i.e., $(\bar{x}_0, \bar{x}_0') = (0, 0)$, in the simulation. The results are plotted in FIGS. 7(a)-7(d). FIGS. 7(a)-7(d) show that the signal estimators provide very accurate estimations for both the excitation frequency and amplitude when the base motion is perfectly harmonic. Furthermore, the vibration response with the gradual adjustment strategy has a root mean square (RMS) amplitude of 19.94. However, the RMS amplitude of the vibration response without this adjustment strategy is only 2.92 even when the gap size is optimized. When the gap size is fixed at the optimized size, the PWL oscillator converges to the linear response with a lower vibration amplitude since $(\bar{x}_0, \bar{x}_0') = (0, 0)$ in this case is in the basin of the attractor of the linear response as discussed above. Thus, current PWL harvesters do not always provide an improved vibration performance in an arbitrary excitation frequency given a random initial condition since they do not have the gap adjustment mechanism. This new vibration energy harvester is able to accumulate vibration energy when the gap size is initially zero and gradually adjust the gap 190 to the optimized size. This gradual adjustment process ensures that the system preserves the intermittent contact behavior during the transient response. Controlling the gap size is required even when the system is subjected to stationary excitation.

Example 2

[0093] The system subjected to a perturbed harmonic excitation with a fixed excitation frequency is studied in this case. In this study, the excitation is assumed to be perturbed by a normally distributed noise: $y(\tau) = y_0 \sin(\rho\tau) + p(\tau)$, where $y_0 = 1$, $\rho = 1.15$ and $p(\tau)$ is the random perturbation with the variance being 35% of y_0. The results are plotted in FIGS. 8(a)-8(d). FIGS. 8(a)-8(d) show that the estimated frequency and amplitude both fluctuate around the ρ and y_0 values used in the excitation input, respectively. In particular, the estimated amplitude has a larger variation due to the excitation perturbation. Furthermore, the system responses of using two optimized gap sizes are compared in FIG. 8(b). The optimal gap size fluctuates due to changing estimated amplitude value as can be seen in FIGS. 8(c) and 8(d). When the system is tracking 88% of the resonant gap size, the PWL oscillator maintains intermittent contact over the entire simulation time range. In contrast, the system loses the intermittent contact when it is tracking 92% of the resonant gap size. This implies that the

absolute optimized gap size needs to be reduced to prevent the system from jumping to the linear response when subject to large variation in the excitation. The jump phenomenon significantly reduces the vibration performance. Thus, there is a trade-off between maintaining a high vibration amplitude and avoiding the undesired jump phenomenon. Note that the nonlinear vibration amplitude is roughly proportional to the gap size, as discussed above.

Example 3

[0094] The case when the system is subjected to a gradually drifting excitation is studied next. In order to test how the proposed control method responds to changing excitation, both excitation frequency and amplitude are set up to drift through time. In this case study, the frequency gradually changes from $\rho = 1.05$ to $\rho = 2.2$ and the amplitude changes from $y_o = 2.0$ to $y_o = 0.5$ over the simulation time range. Note that a 35% perturbation of the amplitude is applied to the excitation signal and 87% of the resonant gap size is used for gap optimization. The results are shown in FIGS. 9(a)-9(d). FIGS. 9(a)-9(d) show that both the estimated frequency and amplitude fluctuate around the actual values due to the perturbation and signal drifting; however, the estimators track the trend of changing excitation very well. The response of using the adjustable gap with the proposed control method is compared with the response of using $g = 0$ in FIG. 9(b). It is shown that the PWL oscillator can maintain intermittent contact after the gradual adjustment process when the gap 190 is tracking the optimized size that changes with the excitation. In contrast, the system only experiences resonance when the excitation frequency is close to the BF when the gap size is fixed at $g = 0$. The RMS vibration amplitudes are 32.94 when using the adjustable gap and 6.96 when $g = 0$. In summary, the proposed control strategy can provide the optimized vibration performance when the PWL harvester is driven by a changing excitation where the frequency drifts within the bounded frequency range.

Example 4

[0095] It has been shown that the proposed system can handle excitation with gradual changes in its frequency and amplitude. The case where the system is subjected to abruptly changing excitation is discussed next. In this case study, the system is subjected to piecewise stationary excitation with 35% perturbations in amplitude. The excitation frequency and amplitude within the three stationary regions are $(\rho, y_0) = (1.05, 1.3), (1.20, 2.0)$, and $(1.70, 1.1)$, respectively. The excitation condition

changes abruptly when it switches from one stationary region to the next. The result of applying the controllable gap is plotted in FIGS. 10(a)-10(d). Note that 87% of the resonant gap size is used for the optimized gap size. FIG. 10(b) shows that the system loses the intermittent contact and tends to converges to the linear response shortly after the sudden changes in the excitation. In order to maximize the response, the gap size is adjusted to $g = 0$ to trigger the intermittent contact behavior. Once the intermittent contact event is detected again, the gradual adjustment process is activated to bring the system back to the nonlinear response to enhance the vibration performance. Note that this control strategy can also be applied to the circumstance where the system loses the intermittent contact due to perturbations in the excitation, as discussed above in Example 2.

[0096] Disclosed are a device and method for an energy harvesting system with a PWL nonlinear oscillator and an adjustable gap. In this system, the resonant frequency of the device can be tuned to match the excitation frequency using a real-time control mechanism by adjusting the gap to the appropriate size. The control method integrates the fast prediction of the optimized gap size, signal estimators, and a gap adjustment mechanism to optimize the vibration performance over a broad frequency range while also being able to achieve the best performance at resonance. The simulation results show that the control method can enhance the vibration amplitude of the PWL harvester for both stationary and changing excitation conditions. The system can achieve a better vibration performance than traditional PWL harvesters with the gap size fixed at a constant. Furthermore, the energy harvesting strategy can be applied to piezoelectric or electromagnetic devices.

[0097] A number of example implementations are provided herein. However, it is understood that various modifications can be made without departing from the spirit and scope of the disclosure herein. As used in the specification, and in the appended claims, the singular forms "a," "an," "the" include plural referents unless the context clearly dictates otherwise. The term "comprising" and variations thereof as used herein is used synonymously with the term "including" and variations thereof and are open, non-limiting terms. Although the terms "comprising" and "including" have been used herein to describe various implementations, the terms "consisting essentially of" and "consisting of" can be used in place of "comprising" and "including" to provide for more specific implementations and are also disclosed.

[0098] Disclosed are materials, systems, devices, methods, compositions, and components that can be used for, can be used in conjunction with, can be used in preparation for, or are products of the disclosed methods, systems, and devices. These and other components are

disclosed herein, and it is understood that when combinations, subsets, interactions, groups, etc. of these components are disclosed that while specific reference of each various individual and collective combinations and permutations of these components may not be explicitly disclosed, each is specifically contemplated and described herein. For example, if a device is disclosed and discussed each and every combination and permutation of the device, and the modifications that are possible are specifically contemplated unless specifically indicated to the contrary. Likewise, any subset or combination of these is also specifically contemplated and disclosed. This concept applies to all aspects of this disclosure including, but not limited to, steps in methods using the disclosed systems or devices. Thus, if there are a variety of additional steps that can be performed, it is understood that each of these additional steps can be performed with any specific method steps or combination of method steps of the disclosed methods, and that each such combination or subset of combinations is specifically contemplated and should be considered disclosed.

WHAT IS CLAIMED IS:

1. A device for controlling vibration with piecewise-linear nonlinearity, the device comprising:

a stiffness element that is expandable and compressible along an axis;

a mass coupled to the stiffness element, wherein the mass has a resting mass position along the axis;

a stopper; and

an actuator coupled to the stopper, the actuator being configured to move the stopper along the axis to vary a gap size, wherein the gap size is measured as a distance between the resting mass position and a resting stopper position.

2. The device of claim 1, wherein the stiffness element is a spring.

3. The device of claim 2, wherein the spring is a linear spring.

4. The device of claim 1, wherein the stiffness element is a cantilevered beam.

5. The device of claim 1, wherein the stiffness element is a coil of wire.

6. The device of claim 1, further comprising a damping element coupled to the mass, the damping element being expandable and compressible along the axis.

7. The device of claim 1, wherein the damping element is a linear viscous damper.

8. The device of claim 1, wherein the stopper comprises a rigid material.

9. The device of claim 1, wherein the stopper comprises a stopper stiffness element.

10. The device of claim 1, wherein the stopper stiffness element is a stopper spring.

11. The device of claim 10, wherein the stopper spring is a linear spring.

12. The device of claim 1, wherein the stopper stiffness element is a cantilevered beam.

13. The device of claim 1, wherein the stopper stiffness element is a coil of wire.

14. The device of claim 1, wherein the stopper comprises a stopper damping element.

15. The device of claim 1, wherein the stopper damping element is a linear viscous damper.

16. The device of claim 1, further comprising a controller configured to determine an optimized gap size based on an identified dominant frequency and amplitude of an excitation signal and cause the actuator to move the stopper along the axis toward the optimized gap size.

17. The device of claim 16, wherein the optimized gap size is determined using bilinear amplitude approximation (BAA).

18. The device of claim 16, wherein the optimized gap size is determined using non-dimensional calculations.

19. The device of claim 1, wherein the gap size is a negative distance such that the stopper prestresses the stiffness element.

20. A method of controlling vibration with piecewise-linear nonlinearity, the method comprising:

(a) introducing a device for controlling vibration with piecewise-linear nonlinearity to an excitation signal, the device comprising:

a stiffness element that is expandable and compressible along an axis,

a mass coupled to the stiffness element, wherein the mass has a resting position along the axis,

a stopper, and

an actuator coupled to the stopper, the actuator being configured to move the stopper along the axis to vary a gap size, wherein the gap size is measured as a distance between the resting mass position and a resting stopper position;

(b) measuring or computing the excitation signal;

(c) identifying a dominant frequency and amplitude of the excitation signal;

(d) determining an optimized gap size based on the identified dominant frequency and amplitude of the excitation signal; and

(e) moving the stopper along the axis toward the optimized gap size.

21. The method of claim 20, further comprising repeating steps (b)-(e).

22. The method of claim 20, wherein the initial gap size is zero.

23. The method of claim 20, wherein the stiffness element is a spring.

24. The method of claim 23, wherein the spring is a linear spring.

25. The method of claim 20, wherein the stiffness element is a cantilevered beam.

26. The method of claim 20, wherein the stiffness element is a coil of wire.

27. The method of claim 20, wherein the device further comprises a damping element coupled to the mass, the damping element being expandable and compressible along the axis.

28. The method of claim 20, wherein the damping element is a linear viscous damper.

29. The method of claim 20, wherein the stopper comprises a rigid material.

30. The method of claim 20, wherein the stopper comprises a stopper stiffness element.

31. The method of claim 20, wherein the stopper stiffness element is a stopper spring.

32. The method of claim 31, wherein the stopper spring is a linear spring.

33. The method of claim 20, wherein the stopper stiffness element is a cantilevered beam.

34. The method of claim 20, wherein the stopper stiffness element is a coil of wire.

35. The method of claim 20, wherein the stopper comprises a stopper damping element.

36. The method of claim 20, wherein the stopper damping element is a linear viscous damper.

37. The method of claim 20, wherein the device further comprises a controller configured to determine the optimized gap size.

38. The method of claim 20, wherein the optimized gap size is determined using bilinear amplitude approximation (BAA).

39. The method of claim 20, wherein the optimized gap size is determined using non-dimensional calculations.

40. The method of claim 20, wherein the gap size is a negative distance such that the stopper prestresses the stiffness element.

41. A device for controlling vibration with piecewise-linear nonlinearity, the device comprising:

one or more stiffness elements, each of the one or more stiffness elements is expandable and compressible along an axis;

one or more masses, each of the one or more masses being coupled to one or more stiffness elements, wherein each of the one or more masses has a resting mass position along its axis;

one or more stoppers; and

one or more actuators, each of the one or more actuators being coupled to one of the one or more stoppers, each of the one or more actuators being configured to move the one of the one or more stoppers along the axis of one of the one or more stiffness elements to vary a gap size, wherein the gap size is measured as a distance between the resting mass position and a resting stopper position of the one of the one or more stoppers.

42. The device of claim 41, wherein at least one of the stiffness elements is a spring.

43. The device of claim 42, wherein at least one of the springs is a linear spring.

44. The device of claim 41, wherein at least one of the stiffness elements is a cantilevered beam.

45. The device of claim 41, wherein at least one of the stiffness elements is a coil of wire.

46. The device of claim 41, further comprising one or more damping elements, wherein each of the one or more damping elements is coupled to one of the one or more masses, the one or more damping elements being expandable and compressible along the axis.

47. The device of claim 41, wherein at least one of the damping elements is a linear viscous damper.

48. The device of claim 41, wherein at least one of the stoppers comprises a rigid material.

49. The device of claim 41, wherein at least one of the stoppers comprises a stopper stiffness element.

50. The device of claim 41, wherein at least one of the stopper stiffness elements is a stopper spring.

51. The device of claim 50, wherein at least one of the stopper springs is a linear spring.

52. The device of claim 41, wherein at least one of the stopper stiffness elements is a cantilevered beam.

53. The device of claim 41, wherein at least one of the stopper stiffness elements is a coil of wire.

54. The device of claim 41, wherein at least one of the stoppers comprises a stopper damping element.

55. The device of claim 41, wherein at least one of the stopper damping elements is a linear viscous damper.

56. The device of claim 41, further comprising a controller configured to determine an optimized gap size between each of the masses and corresponding stopper based on one or more identified frequencies and amplitudes of one or more excitation signals and cause the one or more actuators to move the corresponding stoppers along the axes toward the optimized gap sizes.

57. The device of claim 56, wherein the optimized gap sizes are determined using bilinear amplitude approximation (BAA).

58. The device of claim 56, wherein the optimized gap sizes are determined using non-dimensional calculations.

59. The device of claim 41, wherein at least one of the gap sizes is a negative distance such that one of the stoppers prestresses the corresponding stiffness element.

60. A method of controlling vibration with piecewise-linear nonlinearity, the method comprising:

(a) introducing a device for controlling vibration with piecewise-linear nonlinearity to one or more excitation signals, the device comprising:

one or more stiffness elements, each of the one or more stiffness elements is expandable and compressible along an axis,

one or more masses, each of the one or more masses being coupled to one or more stiffness elements, wherein each of the one or more masses has a resting mass position along its axis,

one or more stoppers, and

one or more actuators, each of the one or more actuators being coupled to one of the one or more stoppers, each of the one or more actuators being configured to move the one of the one or more stoppers along the axis of one of the one or more stiffness elements to vary a gap size, wherein the gap size is measured as a distance between the

resting mass position and a resting stopper position of the one of the one or more stoppers;

(b) measuring or computing each of the excitation signals;

(c) identifying one or more frequencies and amplitudes of each of the excitation signals;

(d) determining an optimized gap size for each of the masses and corresponding stopper based on the identified one or more frequencies and amplitudes of each of the excitation signals; and

(e) moving each of the stoppers along the corresponding axis toward the corresponding optimized gap size.

61. The method of claim 60, further comprising repeating steps (b)-(e).

62. The method of claim 60, wherein at least one of the initial gap sizes is zero.

63. The method of claim 60, wherein at least one of the stiffness elements is a spring.

64. The method of claim 63, wherein at least one of the springs is a linear spring.

65. The method of claim 60, wherein at least one of the stiffness elements is a cantilevered beam.

66. The method of claim 60, wherein at least one of the stiffness elements is a coil of wire.

67. The method of claim 60, further comprising one or more damping elements, wherein each of the one or more damping elements is coupled to one of the one or more masses, the one or more damping elements being expandable and compressible along the axis.

68. The method of claim 60, wherein at least one of the damping elements is a linear viscous damper.

69. The method of claim 60, wherein at least one of the stoppers comprises a rigid material.

70. The method of claim 60, wherein at least one of the stoppers comprises a stopper stiffness element.

71. The method of claim 60, wherein at least one of the stopper stiffness elements is a stopper spring.

72. The method of claim 71, wherein at least one of the stopper springs is a linear spring.

73. The method of claim 60, wherein at least one of the stopper stiffness elements is a cantilevered beam.

74. The method of claim 60, wherein at least one of the stopper stiffness elements is a coil of wire.

75. The method of claim 60, wherein at least one of the stoppers comprises a stopper damping element.

76. The method of claim 60, wherein at least one of the stopper damping elements is a linear viscous damper.

77. The method of claim 60, further comprising a controller configured to determine an optimized gap size between each of the masses and corresponding stopper.

78. The method of claim 77, wherein the optimized gap sizes are determined using bilinear amplitude approximation (BAA).

79. The method of claim 77, wherein the optimized gap sizes are determined using non-dimensional calculations.

80. The method of claim 60, wherein at least one of the gap sizes is a negative distance such that one of the stoppers prestresses the corresponding stiffness element.

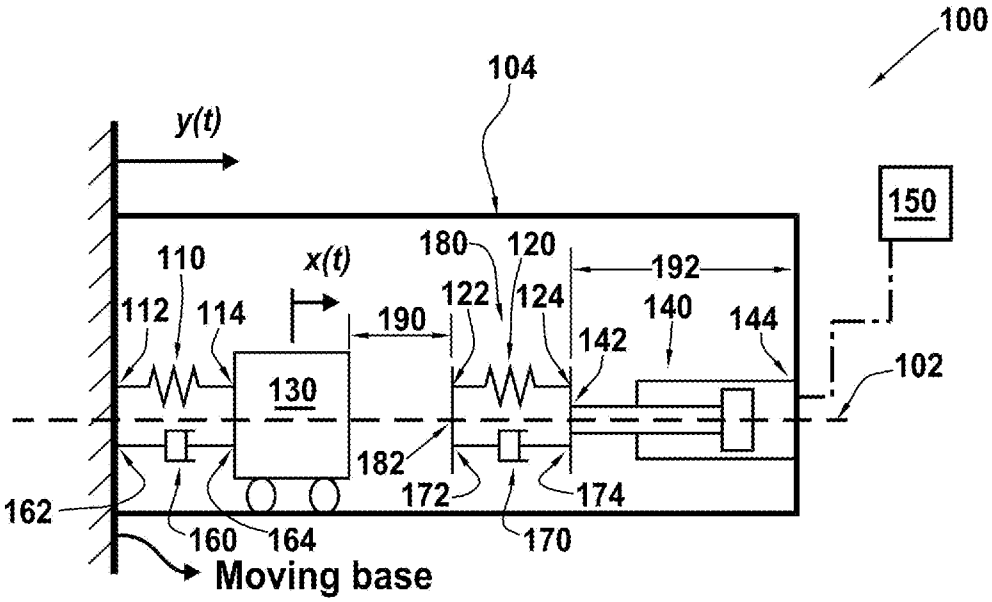

100
104
y(t)
150
110
x(t)
180
120
192
112
114
122
124
140
144
190
142
130
102
182
172
174
162
160
164
170
Moving base

FIG. 1

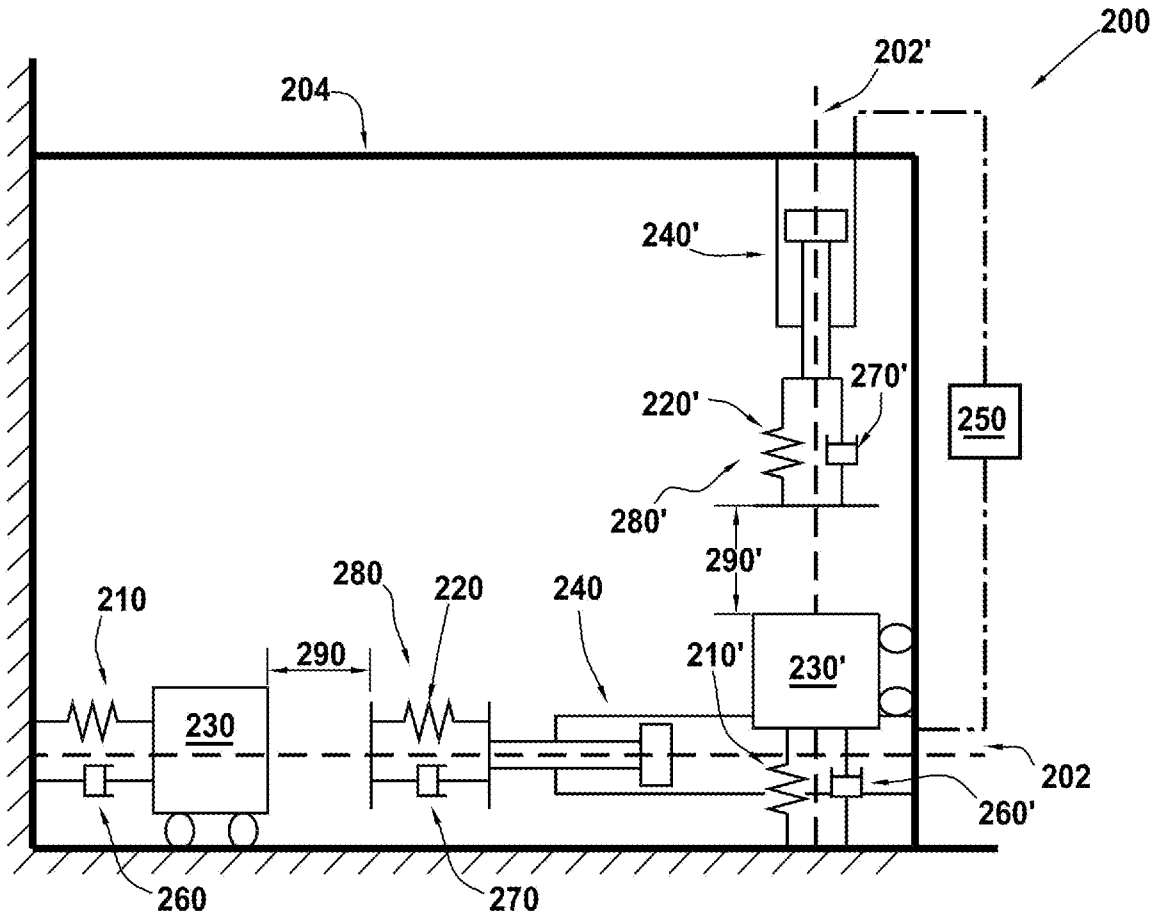

200
202'
204
240'
270'
220'
250
280'
290'
280
220
240
210
210'
230'
290
230
202
260'
260
270

FIG. 2

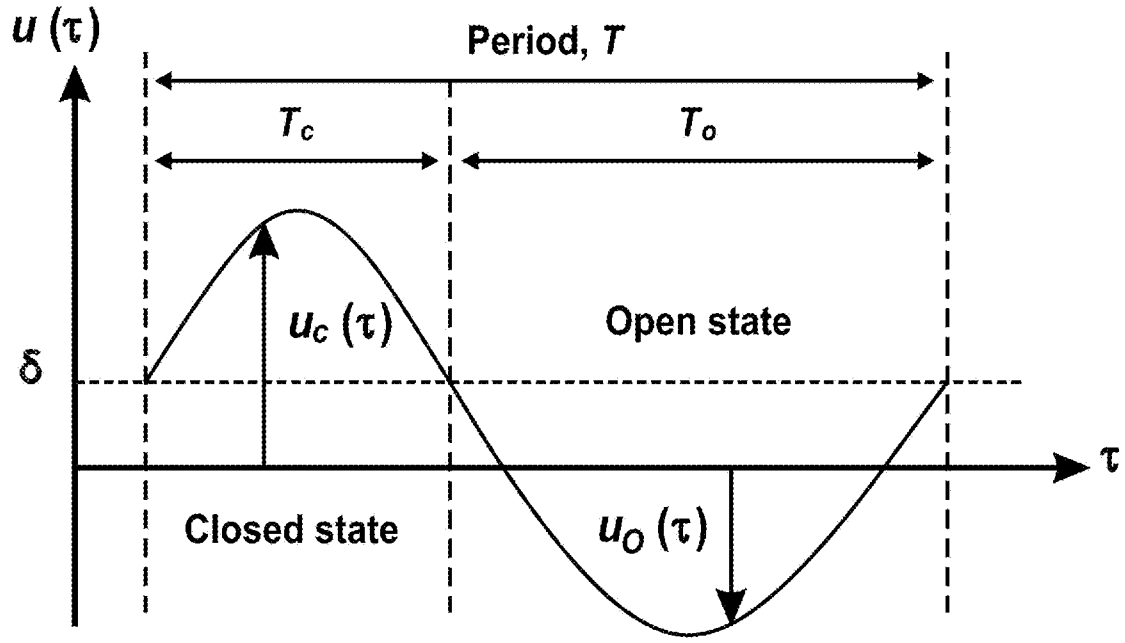

u (τ)
Period, T
Tc
To
uc (τ)
Open state
δ
τ
Closed state
uO (τ)

FIG. 3

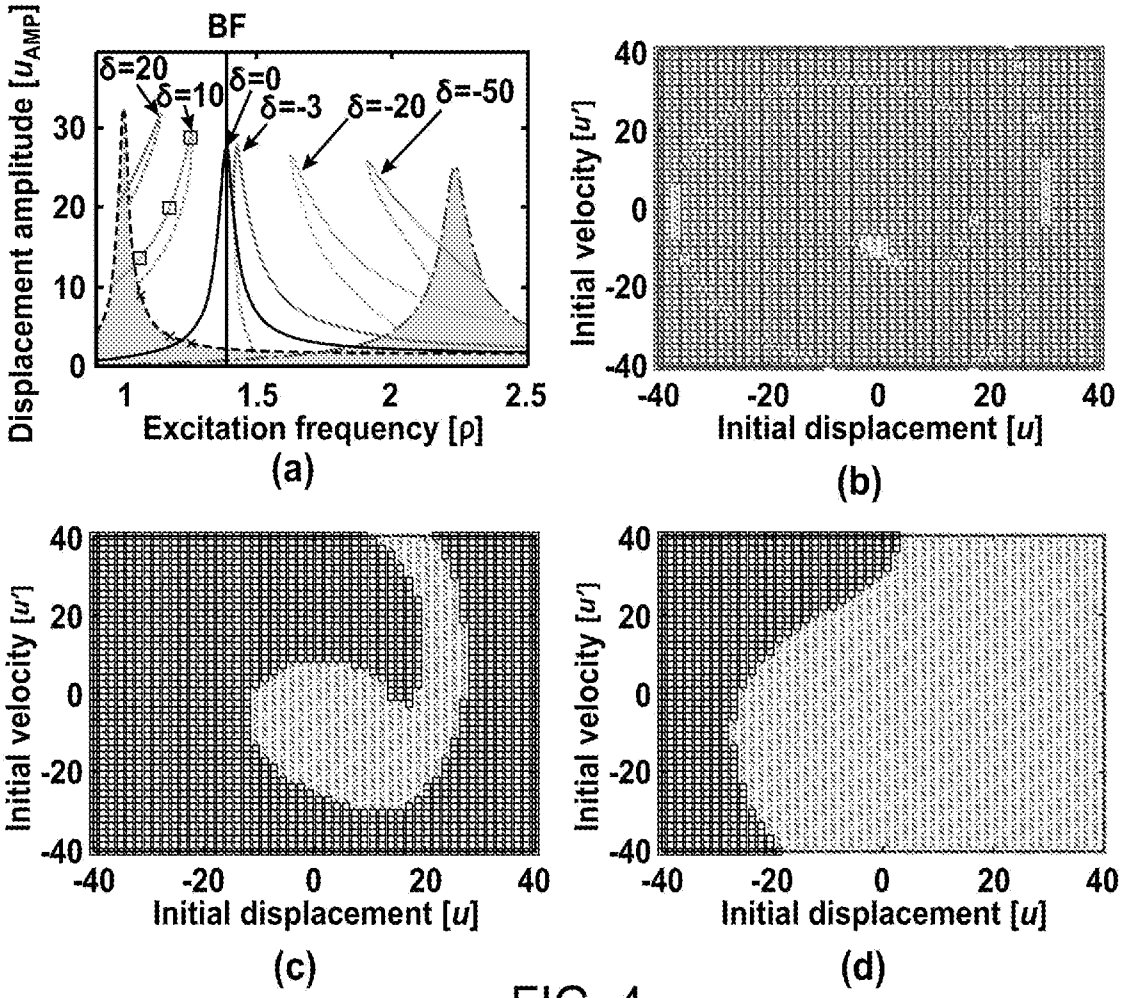

BF
δ=20
δ=10
δ=0
δ=-3
δ=-20
δ=-50
Displacement amplitude [uAMP]
Excitation frequency [ρ]
(a)
Initial velocity [u']
Initial displacement [u]
(b)
(c)
(d)

FIG. 4

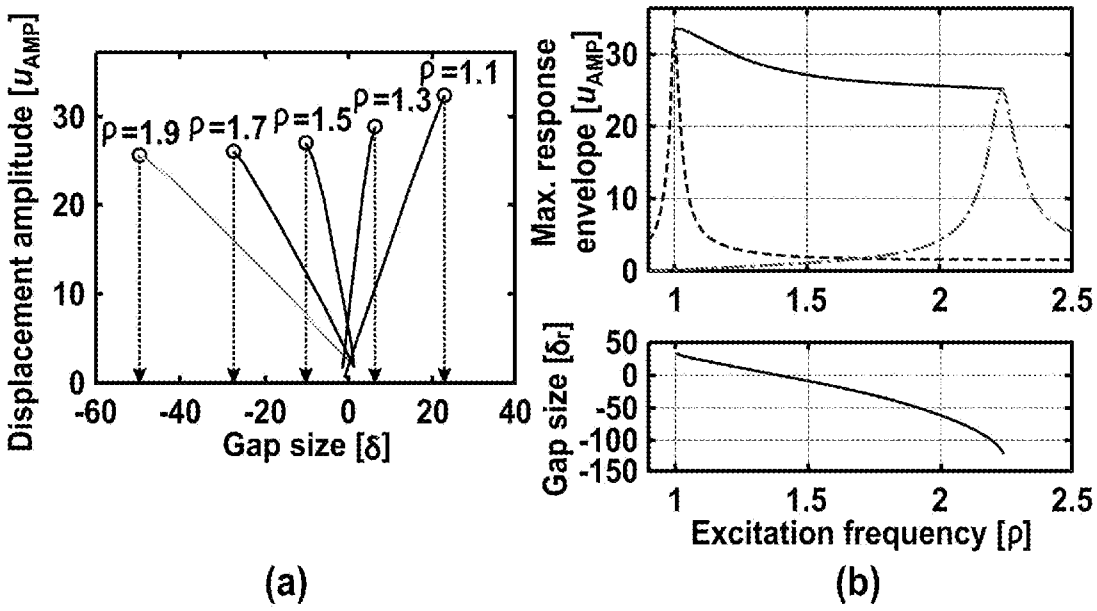

Displacement amplitude [u_{AMP}]
ρ=1.9
ρ=1.7
ρ=1.5
ρ=1.3
ρ=1.1
Gap size [δ]
Max. response envelope [u_{AMP}]
Gap size [δ_r]
Excitation frequency [ρ]
(a)
(b)

FIG. 5

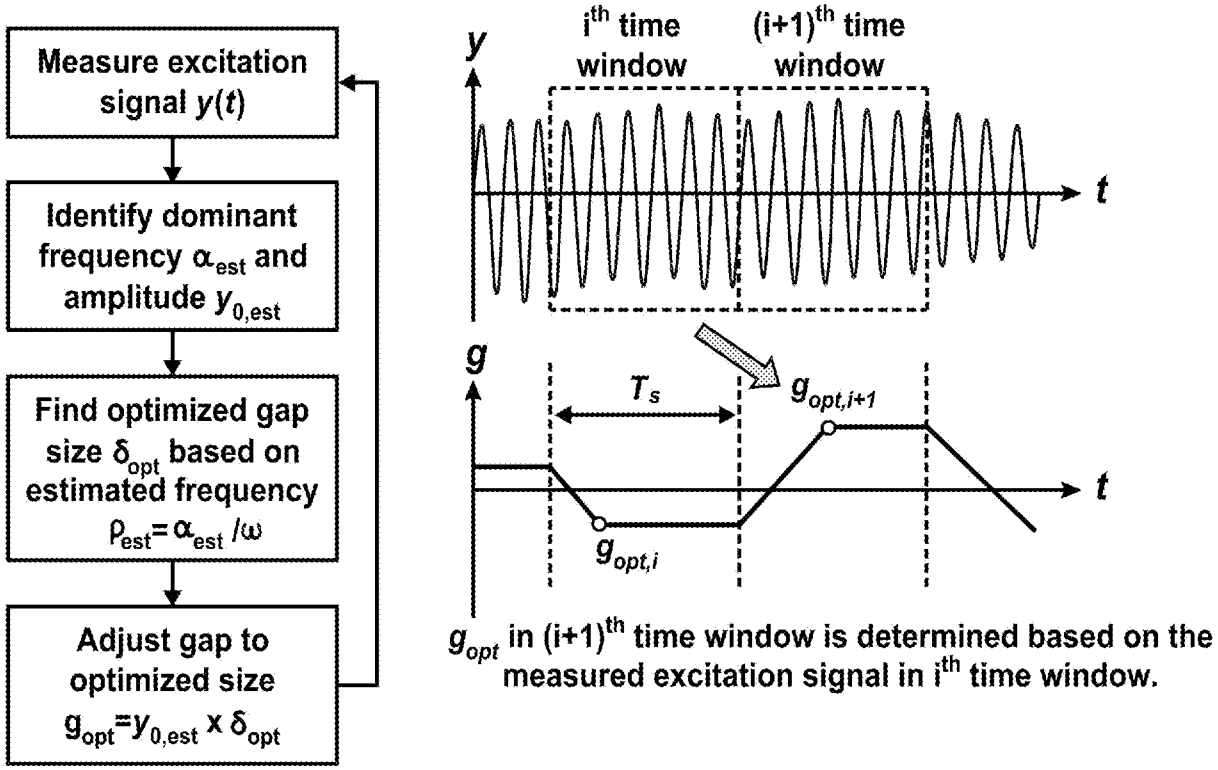

Measure excitation signal $y(t)$
Identify dominant frequency α_{est} and amplitude $y_{0,est}$
Find optimized gap size δ_{opt} based on estimated frequency $\rho_{est}=\alpha_{est}/\omega$
Adjust gap to optimized size $g_{opt}=y_{0,est} \times \delta_{opt}$
i^{th} time window
$(i+1)^{th}$ time window
T_s
$g_{opt,i+1}$
$g_{opt,i}$
g_{opt} in $(i+1)^{th}$ time window is determined based on the measured excitation signal in i^{th} time window.

FIG. 6

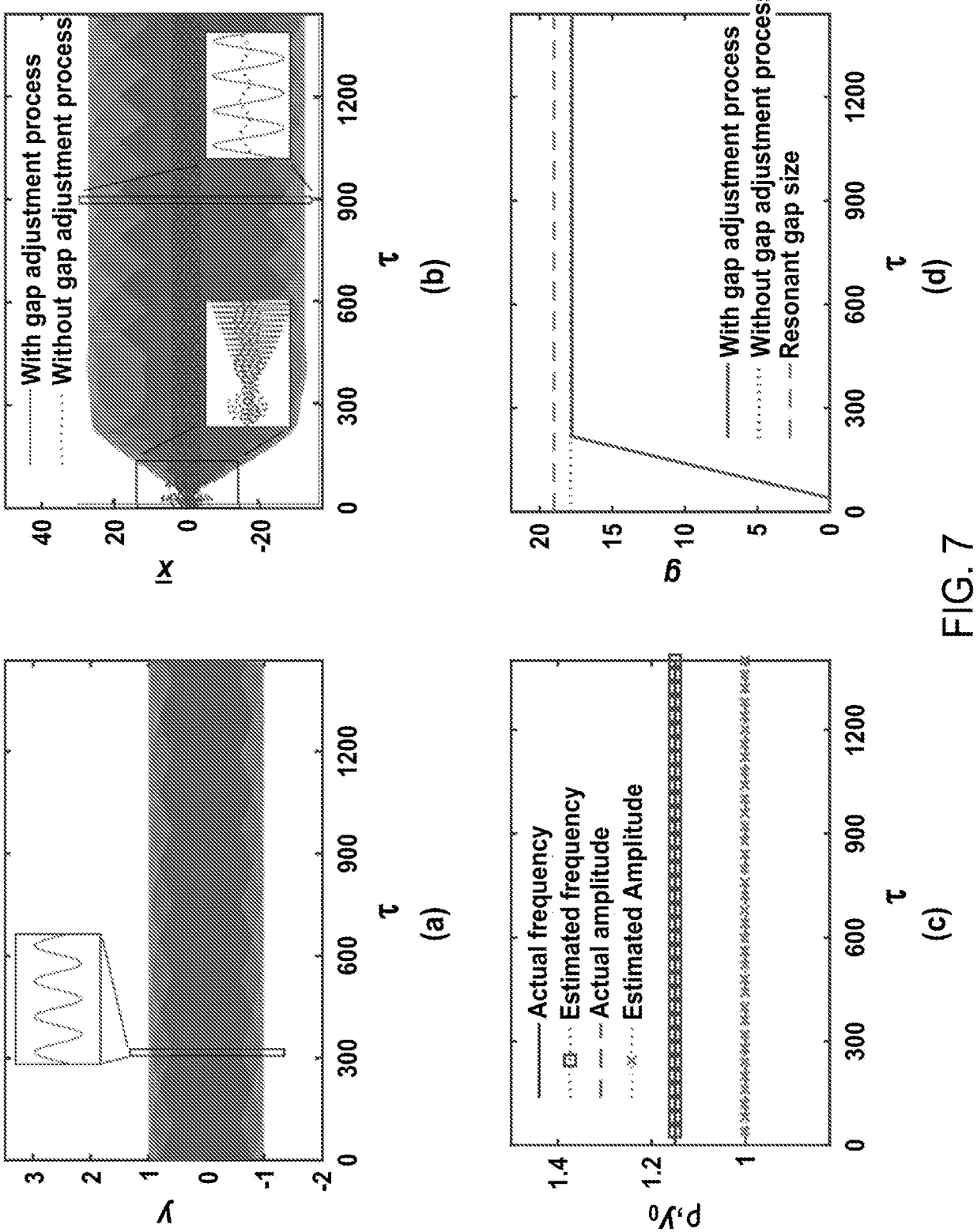

With gap adjustment process
Without gap adjustment process
x
τ
(b)
y
τ
(a)
With gap adjustment process
Without gap adjustment process
Resonant gap size
g
τ
(d)
Actual frequency
Estimated frequency
Actual amplitude
Estimated Amplitude
p, y0
τ
(c)

FIG. 7

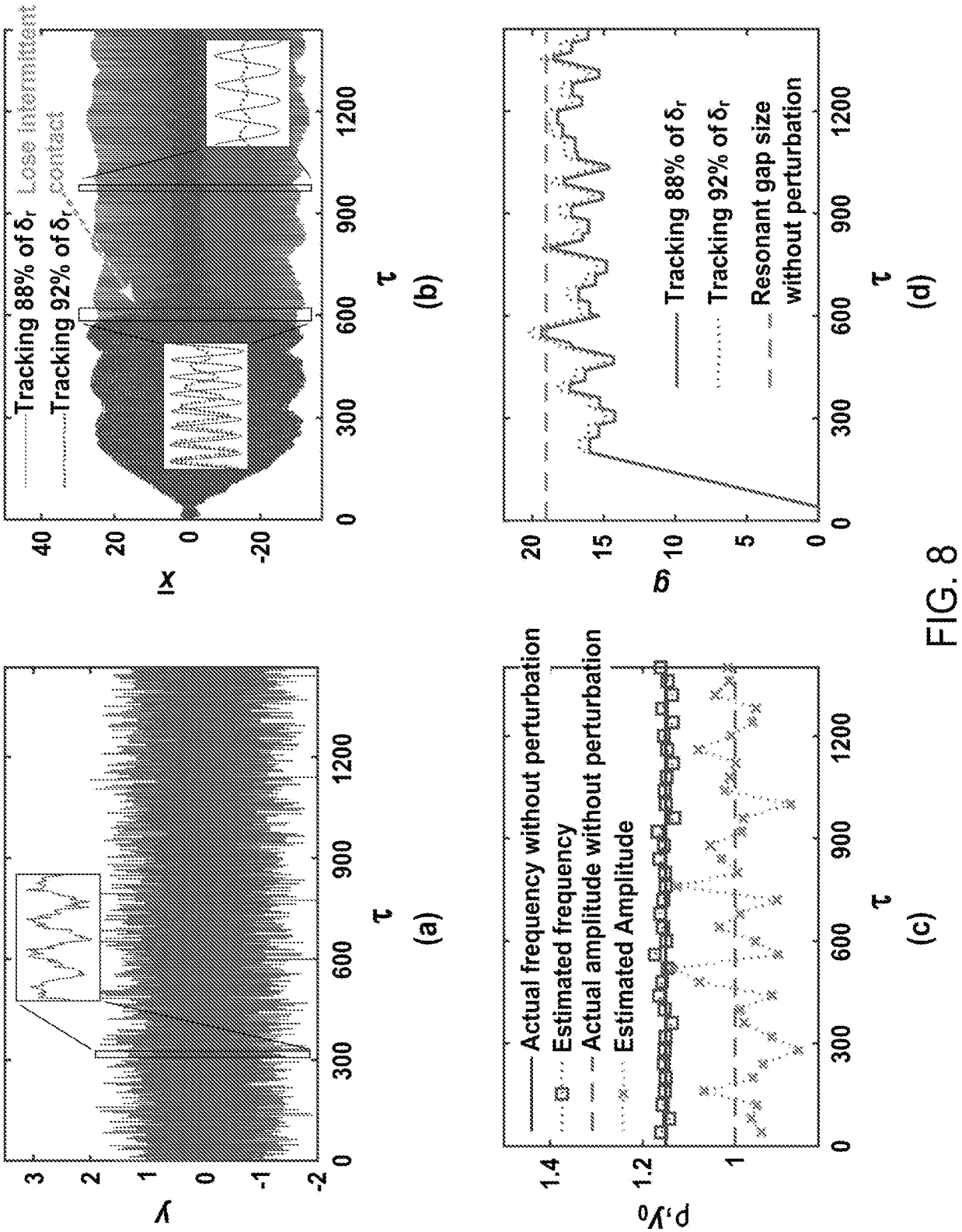

Tracking 88% of δr Lose intermittent contact
Tracking 92% of δr
x
τ
(b)
y
τ
(a)
δ
Tracking 88% of δr
Tracking 92% of δr
Resonant gap size without perturbation
τ
(d)
p1,y0
Actual frequency without perturbation
Estimated frequency
Actual amplitude without perturbation
Estimated Amplitude
τ
(c)

FIG. 8

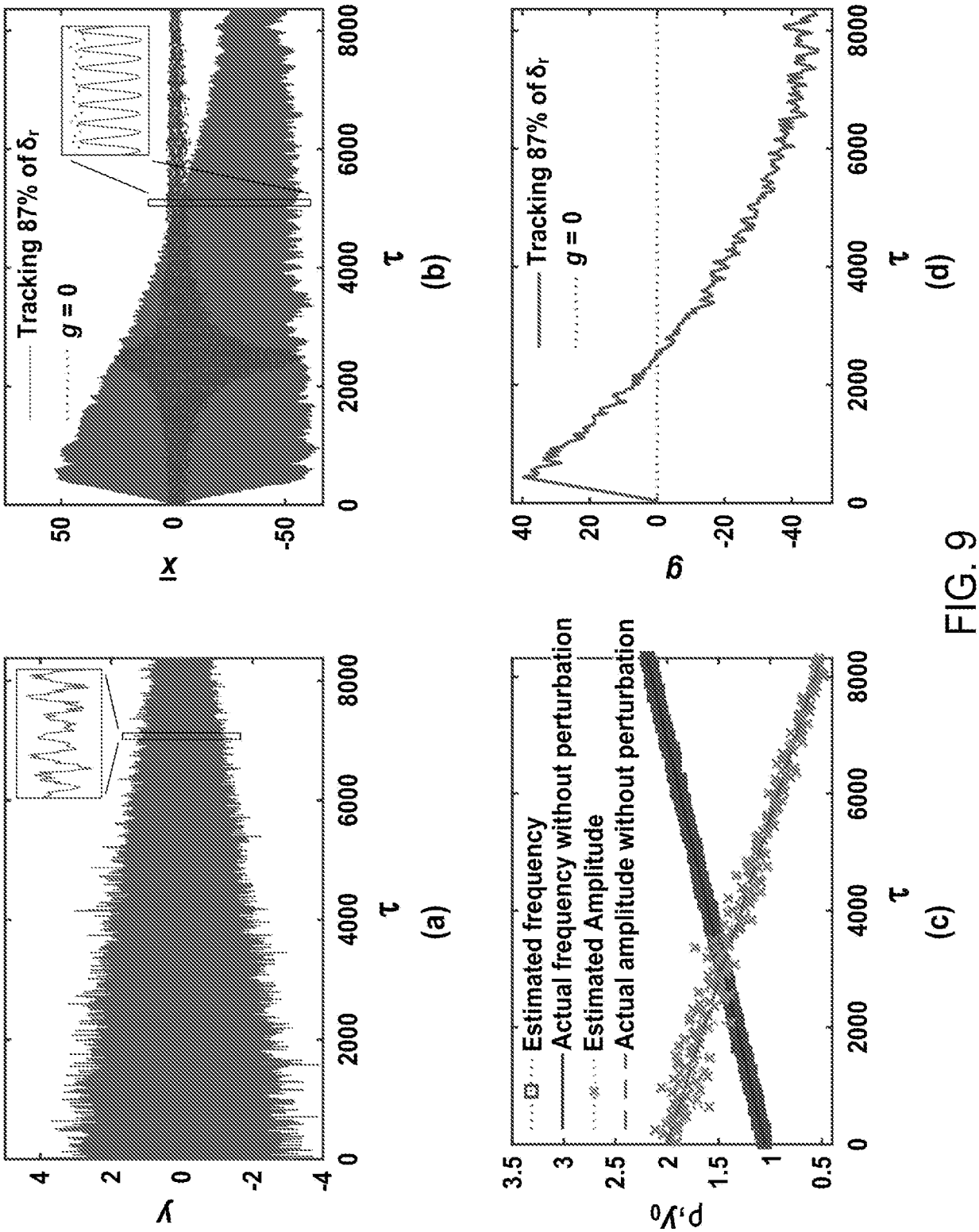

Tracking 87% of δ_r
$g = 0$
$\bar{x}$
τ
(b)
g
(d)
Estimated frequency
Actual frequency without perturbation
Estimated Amplitude
Actual amplitude without perturbation
p, y_0
(c)
y
(a)

FIG. 9

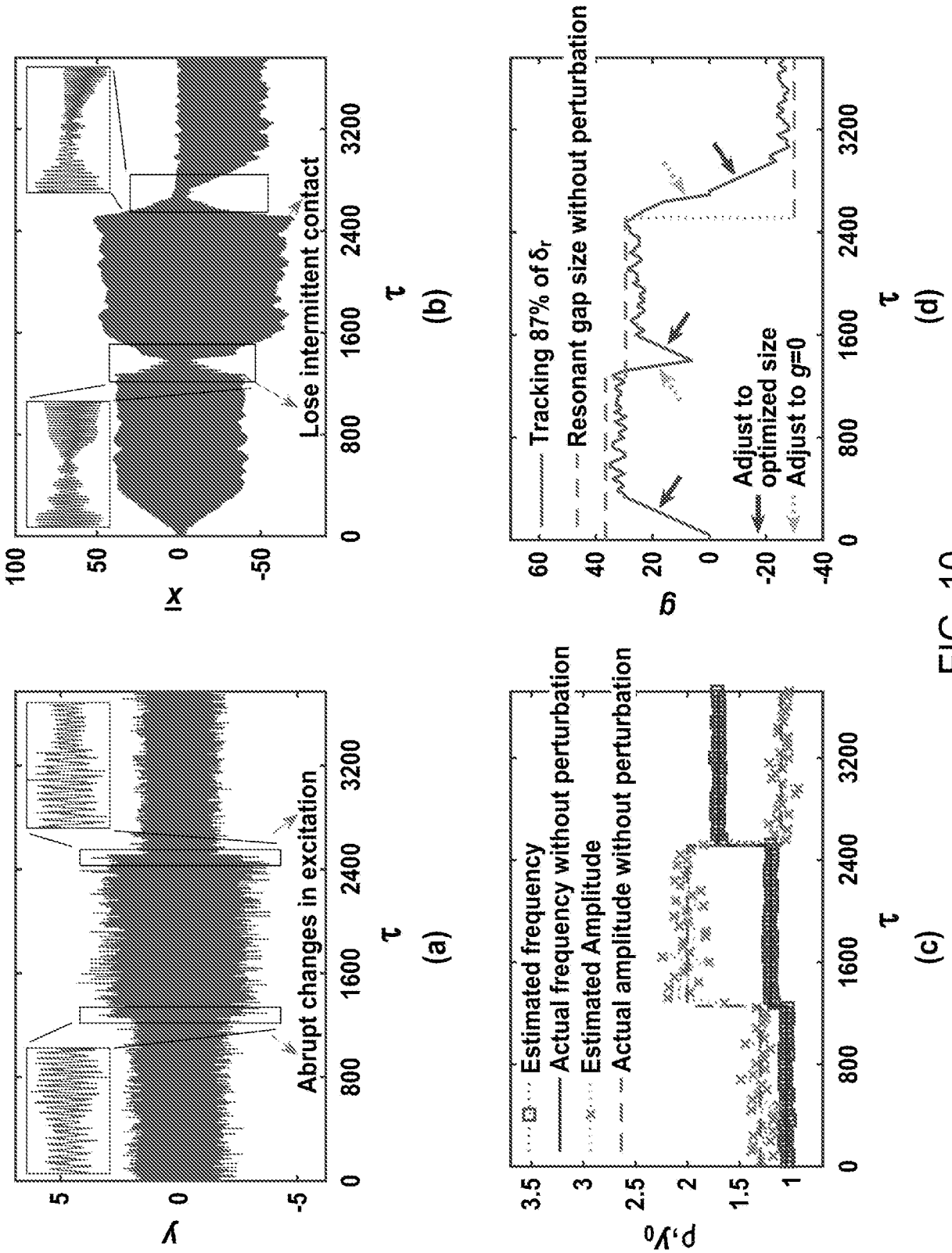

Abrupt changes in excitation
y
τ
(a)
Lose intermittent contact
x
τ
(b)
Estimated frequency
Actual frequency without perturbation
Estimated Amplitude
Actual amplitude without perturbation
p, y0
τ
(c)
Tracking 87% of δr
Resonant gap size without perturbation
Adjust to optimized size
Adjust to g=0
g
τ
(d)

FIG. 10

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US2020/044472

A. CLASSIFICATION OF SUBJECT MATTER

IPC(8) - F16F 7/10; E04B 1/98; E04H 9/02; F16F 15/00; F16F 15/02; G05B 19/404 (2020.01)
CPC - F16F 7/1005; E04H 9/02; E04H 9/0215; E04H 9/0235; E04H 9/027; G05B 19/404; G05B 2219/39199 (2020.08)

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
see Search History document

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

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X --- Y	US 2015/0361658 A1 (MITSUBISHI HEAVY INDUSTRIES MECHATRONICS SYSTEMS, LTD. et al) 17 December 2015 (17.12.2015) entire document	1-15, 41-55 --- 16-40, 56-80
Y	TIEN et al. A generalized bilinear amplitude and frequency approximation for piecewise-linear nonlinear systems with gaps or prestress. Nonlinear Dynamics, 88(4), pages 2403-2416. Published on 08 February 2017 (08.02.2017) entire document	16-40, 56-80
A	US 2016/0209829 A1 (FANUC CORPORATION) 21 July 2016 (21.07.2016) entire document	1-80
A	JP 3786489 B2 (FUJITA CORPORATION) 14 June 2006 (14.06.2006) machine translation	1-80
A	SUN et al. Dynamics of Nonlinear Primary Oscillator with Nonlinear Energy Sink under Harmonic Excitation: Effects of Nonlinear Stiffness. Mathematical Problems in Engineering Volume 2018, Article ID 5693618, 02 September 2018 (02.09.2018) entire document	1-80

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

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

Date of the actual completion of the international search	Date of mailing of the international search report
01 October 2020	16 OCT 2020
Name and mailing address of the ISA/US Mail Stop PCT, Attn: ISA/US, Commissioner for Patents P.O. Box 1450, Alexandria, VA 22313-1450 Facsimile No. 571-273-8300	Authorized officer Blaine R. Copenheaver Telephone No. PCT Helpdesk: 571-272-4300