TITLE: VIBRATION CONTROL OF FREE PISTON MACHINE THROUGH FREQUENCY ADJUSTMENT

BACKGROUND OF THE INVENTION

1. Field Of The Invention

This invention relates generally to minimizing mechanical vibrations of a mechanical apparatus that includes one or more masses driven in reciprocation by a linear, freely reciprocating, prime mover and using the electronic controller that controls that prime mover.

2. Description Of The Related Art

Linear, freely reciprocating machines are often used because they provide improved durability, reduced wear, controllability and efficiency. Freely reciprocating machines include linear compressors, free piston Stirling engines, Stirling cooler, cryocoolers and heat pumps, linear motors and linear alternators. Linear, freely reciprocating machines reciprocate with a controllable stroke and are unconfined by conventional crankshafts and connecting rods. However, linear, freely reciprocating machines cause substantial vibration because they have one or more masses that are linearly reciprocating within a common housing and/or are attached to a common support frame.

Typically, a main machine or system consists of multiple, freely reciprocating machines connected together. One reciprocating machine is a linear, freely reciprocating, prime mover such as an electric, linear motor or free piston Stirling engine which can also be termed a Stirling linear motor. The second reciprocating machine is a linear, freely reciprocating load driven through a mechanical link by the prime mover and may be, for example, a free piston compressor, a Stirling heat pump or cooler or an electric alternator. The composite reciprocating masses of both the prime mover and the load it drives contribute to the vibration. This vibration is ordinarily undesirable and a variety of systems have been developed to minimize the amplitude of such vibrations.

Typically, free piston and other linear, freely reciprocating machines are constructed with one or more springs applying a spring force to the reciprocating masses. Both the prime mover and the machine it drives may include springs. The springs may include one or a combination of mechanical springs as well as gas springs and magnetic springs. The gas and magnetic springs may be devices designed to provide a spring force or, more commonly, they are the result of gas acting upon a component of the machine and/or magnetic forces from electromagnetic devices or permanent magnet system used in the machines, such as electric linear motors and alternators. Together, the masses and springs of the linear, freely reciprocating prime mover and the driven, linear, freely reciprocating machine form a main machine that is a resonant system

Commonly, the main machines are designed to be operated at or near a resonant frequency because that maximizes their efficiency. The natural frequency of such a system is described in accordance with the equation:

$$\text{Eq. 1} \quad f = \frac{1}{2\pi}\sqrt{\frac{K}{m}}$$

where f = the resonant frequency in cycles per second or Hertz, K is the composite spring constant in Newton/meter and m is the composite mass in Kg. The word "composite" is used to designate the sum of the respective masses and springs of the main machine and the terms "mass" and "spring" are used to include the composite mass or spring when their effects sum together.

The vibration problem can be further complicated if a mechanical apparatus consists of a main machine or system, comprising a prime mover driving a driven machine, that is also coupled to other equipment that includes one or more secondary vibrating systems. Such secondary systems may be coupled to the main machine by mounting the secondary vibrating system so it is mechanically connected to the main system, for example because both systems are mounted to the same support frame. Secondary vibrating systems can be devices that are designed with masses and springs to oscillate during their operation or they can be devices that are not intended to oscillate as part of their normal function but nonetheless have a mass connected to a structure acting as a spring. A secondary vibrating system that is coupled to the main system is a parasitic

resonant system if it is not intended to vibrate during its normal operation. If the resonant frequency of the parasitic resonant system is sufficiently near the driving frequency of the main machine, the parasitic resonant system may vibrate at an excessive amplitude. If the parasitic resonant system vibrates at the drive frequency and at less than 90° out of phase with the main system, it can increase the total vibration of the mechanical apparatus.

The prior art has developed a variety of devices for reducing the vibration of a main machine. These are known by a variety of names including "vibration absorbers", although they are more accurately called vibration balancers because they do not "absorb" vibration. A vibration balancer is a secondary vibrating system that is mechanically coupled to the main system usually by direct connection to it. Although the purpose of vibration balancers is to diminish the vibration resulting from the reciprocations of the main machine, they are desirably viewed as a form of a secondary vibrating system because they are not a part of the main machine or system. One common vibration balancing system seeks to drive a reciprocating, counterbalancing mass in a manner that applies forces to the vibrating main machine that are equal but opposite to the forces generated by the vibrating masses of the main machine. The driven mass of the vibration balancer can be driven by its own prime mover or, alternatively, it can be driven by the vibrations of the vibrating main machine and tuned to be resonant at the same driving frequency but designed to reciprocate 180° out of phase with the vibrations of the vibrating main machine. An example of a system of the former nature is shown in U.S. Patent 5,620,068.

Another system for reducing vibration is illustrated in U.S. patent 6,040,672. A waveform induced in an electric motor drive signal is sensed, translated into a control waveform and summed with the motor drive current to reduce the vibration.

Although these systems perform satisfactorily under relatively stable operating conditions, under extreme variations in operating conditions they can encounter difficulties. For example, a Stirling cycle cooler may undergo extreme ambient temperature variations. It may operate anywhere in the range of -40° C to +60° C. If a Stirling cooler has a vibration balancer attached to it, these variations in temperature change the stiffness of its springs, thereby varying their spring constant, and therefore

cause the natural frequency of the vibration balancer to change. The effective spring stiffness of the spring forces in the cooler may also vary somewhat, although these variations usually have less effect because Stirling coolers typically have a relatively low Q while vibration balancers typically have a high Q, (i.e. sharp resonant peak). Therefore, a relatively small variation in the natural frequency of the vibration balancer results in a large variation in its effective amplitude of oscillation if the driving frequency remains the same. Consequently the ability of the vibration balancer to cancel the vibrations of the vibrating main machine is substantially diminished. Similarly, changes of temperature can also result in changes in electrical parameters which, in turn, can change the effective spring constant of magnetic spring effects. Temperature can also change the dynamic behavior of a Stirling engine resulting in a shift of its operating frequency. Non-linear behavior of mechanical springs or structural components in response to varying strokes may shift the natural frequency as well.

As a result, a mechanical apparatus with a vibration balancer can be well balanced and exhibit an acceptable amplitude of vibration under some operating conditions, but if the operating conditions depart sufficiently from the preset operating conditions, the vibration balancer will become less effective because the change in operating conditions changes the resonant or natural frequency of the vibration balancer or changes it phase relation to the main system or both. If the vibration balancer becomes less effective, the amplitude of the vibrations increases.

Similarly, the resonant frequency of secondary parasitic vibrating systems coupled to the main system can also change as a result of changes in operating conditions. As a result, a secondary system that does not aggravate the vibration of the mechanical apparatus under some operating conditions can become a problem when the operating conditions change sufficiently. A component of a mechanical apparatus that was not a vibration problem can become a problem when operating conditions change sufficiently. A parasitic vibration system can also be discovered after a machine is constructed.

Although it is possible to construct a vibration balancer that would be able to vary its spring constant or otherwise vary its natural frequency of oscillation, such a vibration balancer would be even more expensive than conventional vibration balancers. Vibration

balancers are not only a considerable cost, they also take up space and add weight to a product.

It is a feature and object of the invention to supplement a vibration balancer by electrically compensating for variations in the ability of the vibration balancer to cancel vibrations as a result of variations in operating conditions.

Another object and feature of the invention is to provide a control system for a linear, freely reciprocating, main machine that can compensate for secondary, parasitic vibration systems.

Another object and feature of the invention is to reduce vibration electronically by altering the controlled operating characteristics of a linear, freely reciprocating prime mover to compensate for any of the variety of causes of changes in the resonant frequency associated with a linear, freely reciprocating main machine and any vibration balancer connected to it where the uncompensated changes resulting from changes in operating conditions would otherwise result in an increase in vibration.

Yet another object of the invention is to compensate for changes in the resonant frequency of a linear, freely reciprocating main machine independently of, and in response to, changes in the operating conditions of the machine.

BRIEF SUMMARY OF THE INVENTION

The invention is a method and apparatus for minimizing the amplitude of mechanical vibrations of a mechanical apparatus that includes a linear motor coupled to and driving a reciprocating mass of a driven machine in reciprocation at a driving frequency. The coupled motor and driven machine have one or more springs applying a force upon the composite reciprocating mass to form a resonant main system having a main system resonant frequency of reciprocation. A driving frequency range over which the driven machine operates at an acceptable efficiency of operation is determined and stored. A parameter of the operation of the mechanical apparatus is sensed and the linear motor is driven in response to the sensed parameter at a driving frequency that is offset from the main system resonant frequency of reciprocation, is within the driving frequency

range of acceptable efficiency of operation and reduces or minimizes the amplitude of mechanical vibration of the mechanical apparatus under existing operating conditions.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

Fig. 1 is a block diagram illustrating an example of an embodiment of the invention.

Fig. 2 is a block diagram illustrating a second example of an embodiment of the invention.

Fig. 3 is a graphical plot in the frequency domain of resonant peaks and illustrating the operation of the invention.

Fig. 4 is a block diagram showing an example of a control circuit embodying the invention and using a temperature sensor.

Fig. 5 is a block diagram showing an example of a control circuit embodying the invention and using a vibration amplitude sensor.

Fig. 6 is a block diagram showing an example of a control circuit embodying the invention and using a variable frequency generator.

Fig. 7 is a block diagram showing an example of a control circuit embodying the invention and having a temperature sensor.

In describing the preferred embodiment of the invention which is illustrated in the drawings, specific terminology will be resorted to for the sake of clarity. However, it is not intended that the invention be limited to the specific term so selected and it is to be understood that each specific term includes all technical equivalents which operate in a similar manner to accomplish a similar purpose.

DETAILED DESCRIPTION OF THE INVENTION

The invention makes use of the observation that, for a mechanical apparatus that includes a vibrating or reciprocating main machine coupled to a secondary vibrating system, which can include a vibration balancer, there are three frequencies that are important. There are the resonant (or natural) frequency of the vibrating main machine, the resonant (or natural) frequency of the secondary vibrating system and the operating

frequency of the main machine. The operating frequency of the main machine is also the operating frequency of the vibration balancer and any other secondary vibrating system coupled to the main machine.

If a graph is made of frequency vs. amplitude of vibration for any resonant system, the plotted amplitudes form a resonant peak centered at a resonant frequency. These peaks can rise and fall in a range extending from a broad, gradual manner, to a sharp, steep manner. The sharper the peak, the higher the quality factor "Q" of the resonant system, as known to those skilled in the art.

The invention also makes use of the observation that those main machines, that have a linear, freely reciprocating, prime mover driving a linear, freely reciprocating driven machine and that operate efficiently at or near their composite resonant frequency, nonetheless ordinarily have a band of driving frequencies over which they can operate at an acceptable efficiency. They are not confined to operating exactly at their resonant frequency. In part this is because a typical main machine, such as a linear motor driving a Stirling cooler, ordinarily exhibits a low Q resonance peak. Although this is helpful, the driving frequency range over which the driven machine operates at an acceptable efficiency of operation is determined not only by the Q of the mechanically resonant, reciprocating components of the main machine, but is also dependent upon other design and operating characteristics of the main machine. However, a designer of any particular machine is able to determine an acceptable range of driving frequency by applying ordinary engineering principles to the design of a particular main machine and its application.

Fig. 1 diagrammatically illustrates a mechanical apparatus 10 that has a main machine 12, consisting of an electromagnetic linear motor 14 driving a Stirling cooler 16 in reciprocation, a motor control circuit 18 that has a data storage 20 and controls the operation of the linear motor 14. The main machine 10 may also include secondary vibrating systems 24, such as a vibration balancer 26 and parasitic resonant systems 28. All of the illustrated components are mechanically coupled together to form the mechanical apparatus 10. For example, they may be physically connected together in the

same housing or on the same frame or they may be linked together by intermediate physical structures that can transmit vibrations.

The control system also has a temperature sensor 22 that senses the temperature of the vibrating systems 24, which may be sensed at the casing, and inputs the temperature data to the motor control 18. The temperature sensor may sense the temperature of the spring of the vibration balancer because that is the principal component for which temperature changes most directly affect the resonant frequency of the vibration balancer. Alternatively, the temperature of the ambient environment or a component in thermal connection to the spring can be sensed to approximate the spring temperature.

The motor control 18 can be of a conventional type which typically is a microprocessor based computing system or microcontroller or a digital signal processor and may also include additional sensors. Although the preferred control circuit is a microprocessor controller, many alternative means exist to provide the control circuit functions. As known to those skilled in the art, there are a variety of commercially available, non-microprocessor based controllers that also can provide the controller functions and therefore are equivalent and can be substituted for the microprocessor controller. The sensing functions can be performed by separate circuitry or can be provided on-board a controller. Suitable controllers can include equivalent digital and analog circuits available in the commercial marketplace. Examples of controllers that can be used for the control circuit of the invention include microprocessors, microcontrollers, programmable gate arrays, digital signal processors, field programmable analog arrays and logic gate arrays. Such circuits can be elementary digital logic circuits and can be constructed of discrete components such as diodes and transistors. Therefore the term "controller" is used to generically refer to any of the combinations of digital logic and analog signal processing circuits that are available or known and can be constructed, programmed or otherwise configured for performing the logic functions of the control circuit as described above.

As shown widely in the prior art, the linear motor has a reciprocating set of magnets that are located to reciprocate within a stationary armature winding. The magnets are driven in reciprocation by an alternating magnetic field generated by an

alternating current applied to the armature winding. A support for the magnets is connected to the piston of the Stirling cooler 16 and drives it in reciprocation. This reciprocation causes the Stirling cooler to pump heat energy from one area of the cooler to another where it is rejected. As also known in the art, such Stirling devices are also more generally known as heat pumps because of their above described heat pumping capability. Stirling heat pumps can be used either to heat objects from the rejected heat or to cool objects by accepting heat at their cooled area and rejecting it into the ambient environment. Therefore, the latter are often referred to as coolers and this includes coolers that cool to cryogenic temperatures. The details of the linear motor and the driven machine, such as a Stirling heat pump, compressor or fluid pump, are not illustrated because they are not the invention and are illustrated in numerous examples in the prior art. The principles of this circuit can also be applied for other linear, freely reciprocating prime movers and driven loads such as linear compressors and free-piston Stirling engines which use vibration balancers.

Fig. 2 illustrates an example of an alternative embodiment of the invention. The mechanical apparatus 30 has a main machine 32 that comprises a linear motor 34 driving a Stirling cooler 36 in reciprocation, and a motor control 38 for controlling the linear motor 34, including control of its driving frequency. The motor control has a vibration amplitude sensor 40, such as an accelerometer, to sense and input to the motor control 38 a signal representing the amplitude of the vibrations of the mechanical apparatus 30. All of these devices are physically coupled together as described in connection with Fig. 1. The embodiment of Fig. 2 also may be coupled to one or more secondary vibrating systems, such as a vibration balancer 46 and parasitic vibrating systems 48.

Fig. 3 illustrates the principles upon which the invention operates. It refers to frequency values and curves that are representative and typical but the invention is not limited to those values and curves. For example, it is common for a reciprocating main machine to be designed to be resonant at and to operate at 60 Hz. However, many other frequencies of operation are practical, such as 50 Hz, 120 Hz or 400 Hz. Main machines of the type described are typically designed to be resonant at a natural frequency of vibration or resonant frequency f_0 which corresponds to f in the above equation 1.

Resonant peak M illustrates a typical resonant peak for the mechanical vibrations of a main machine that has a resonant frequency of 60Hz. Its resonant peak is relatively broad about its resonant frequency, thus exhibiting a relatively low Q characteristic. Resonant peaks S1 and S2 illustrate a typical resonant peak for a secondary vibrating system. They are relatively sharp and steep, thus exhibiting a relatively high Q characteristic.

Although half power points (70.7 % of amplitude) are one well known measure of the width of a resonant peak, this measure is applicable only to the mechanically resonant aspects of the system. Other operating characteristics of the main machine, such as its cooling efficiency or coefficient of performance, determine the operating efficiency of the driven machine. Therefore, the driving frequency range in which the driven machine operates with acceptable efficiency may, and usually is, different from the width of the resonant peak of the mechanically oscillating system. However, this acceptable driving frequency range can be, and ordinarily is, determined by the designer. In Fig. 3, an example of the acceptable driving frequency range R is illustrated as between 58 Hz and 62 Hz although it will be different for different main machines.

The resonant peaks S1 and S2 can be used, in explaining the operation of the invention, to represent either a secondary vibrating system that is a vibration balancer or a secondary vibrating system that is a parasitic vibration system. Each is addressed in sequence.

If the secondary vibrating system is a vibration balancer and the main machine is operating under its nominal or design conditions, it will be operating with the peak M representing the main machine and the peak S1 representing the vibration balancer. Under that condition, the driven machine can be driven at the nominal resonant frequency of the main machine, 60 Hz in the example, because that frequency coincides with the resonant frequency of the vibration balancer. However, if the operating conditions, such as temperature, change sufficiently that the physical parameters of the main machine or the secondary vibration system cause a change in the resonant frequency of one or both, that change will appear on the graph of Fig. 3 as one or both peaks becoming displaced horizontally with respect to each other. For example, the peak S1 may move to the position of peak S2, although it could move in either direction and different distances.

The displacement of the peak S1 to the position of peak S2 causes the vibration balancer to become far less effective if the main machine continues to be driven at the resonant frequency of the main machine. However, if the driving frequency of the main machine were changed to be closer to the center of peak S2, the vibration balancer would become more effective at that frequency, 62 Hz in the example, so long as the changed operating conditions remained. If the vibration balancer peak were to shift to a center at 61 Hz or 62 Hz, the driving frequency would be moved to 61 Hz or 62 Hz respectively. Thus, in the invention the motor control system 18 or 38 drives the linear motor at a driving frequency that is offset from the resonant frequency of the main machine or system, that is nearer to or at the displaced resonant frequency of the vibration balancer but is within the driving frequency range of acceptable efficiency of operation of the main system.

Therefore, one aspect of the invention is that, in response to changes in operating conditions that cause divergence of the center frequency of the resonant peaks of the main machine and a vibration balancer, the driving frequency of the linear motor is moved closer to the altered center frequency of the vibration balancer. Although the driving frequency can be changed to bring it closer to the resonant frequency of the shifted peak S2, it can not be moved beyond the limits of the acceptable driving frequency range R because doing so would cause an unacceptable deterioration of the operation of the main machine.

If the secondary vibration system is a parasitic vibration system, hopefully its resonant peak is and remains sufficiently far from the center frequency f_0 of the main machine that it never becomes a factor in the vibration. However, other equipment mounted to the mechanical apparatus 10 or 30 can introduce one or more parasitic vibration systems that either has a resonant peak unexpectedly near the center frequency f_0 or that moves near it as a result of changes in operating conditions. The peaks S1 and S2 can represent such resonant peaks of parasitic, secondary vibration systems. If peak S1 is the peak of the parasitic vibration system, Fig. 3 illustrates that the vibration of the secondary, parasitic system can be substantially reduced by changing the driving frequency to either side of the center frequency of peak S1 but retaining it within the

range R. Consequently, the driving frequency would optimally be made 58 Hz or 62 Hz. If the peak S2 is the peak of a parasitic vibration system, the vibration of the parasitic system would be minimized by moving the driving frequency to 58 Hz in the example, which is as far as possible from the center frequency of peak S2, but not beyond the range R.

There are multiple ways to design and construct a control system to vary the driving frequency of the linear, freely reciprocating, prime mover in accordance with the above principles and six examples will be described. All include sensing a parameter of the operation of the mechanical apparatus, such as the amplitude of vibration of the mechanical apparatus or a temperature of a component part of the mechanical apparatus. The sensed parameter can be sensed by a sensor provided for practicing the invention or it can be a sensor that is part of another control system.

Fig. 2 is the first of the six examples. The vibration amplitude sensor 40 senses the amplitude of the vibration of the mechanical apparatus. Generally, it can be placed on any of the component parts of the mechanical apparatus that are mechanically coupled together because the vibrations of any one part are ordinarily transmitted to the other component parts. The motor control 38 drives the linear motor 34 at each of several representative frequencies distributed within the driving frequency range of acceptable efficiency of operation. Each frequency is stored in association with the amplitude of the sensed, resulting vibrations. This is done on start up, periodically after start up, in response to a sensed vibration amplitude above a selected level and/or in response to other conditions or algorithms. The software or logic circuitry of the motor control 38 then selects the least vibration amplitude and drives the linear motor at the frequency associated with the least vibration amplitude. The vibrations are therefore reduced or minimized for the existing operating conditions at the time this procedure is performed. Repetitions of the procedure permit the control system to respond to changed operating conditions.

More specifically, the motor control circuit 38 finds the frequency of least vibration by driving the linear motor at a plurality of frequencies within the acceptable range R by any of several techniques which are referred to herein as dithering or

sweeping the frequency across the frequency range R. The most common form of sweeping the frequency is to progressively vary the frequency from one side of the range R to the other, either continuously or in incremental steps. However, this sweep alternatively can be performed in a non-sequential or in a random manner and can be done at selected spaced intervals.

Fig. 1 illustrates a second example of the invention. The temperature sensor 22 senses the temperature of the vibration balancer 26, or its ambient environment, and inputs the temperature data to the motor control 18. As with all embodiments of the invention, the driving frequency range of acceptable efficiency of operation is determined and stored in response to testing of at least one mechanical apparatus. The testing is ordinarily performed in a laboratory setup but can be based upon engineering design specifications. This experimentally determined operating frequency range of acceptable efficiency of operation is then stored in production replications of the mechanical apparatus.

At least one heat pumping mechanical apparatus is also tested, ordinarily in a laboratory environment, by being operated at a plurality of different operating temperatures and, for each of the operating temperatures, the driving frequency is swept within the acceptable driving frequency range. The driving frequency resulting in the least amplitude of vibration of the mechanical apparatus is stored in association with each operating temperature. As a result, for each operating temperature that is sensed, there is a stored driving frequency that provides the minimum vibration. These operating temperatures and their associated driving frequencies are stored as a lookup table in a memory device 20 connected in the frequency control system of production replications of the tested mechanical apparatus. Alternatively, the lookup table can store a spring constant for the spring of the vibration balancer in association with each measured temperature and an algorithm used to convert the spring constant to an operating frequency. As yet another alternative, instead of a lookup table, an equation can be developed using well know mathematical techniques, such as a polynomial series, for approximating a plot of the lookup table and thereby relating the vibration absorber spring

constant or operating frequency to the sensed temperature with the result computed by the motor control microprocessor.

During operation of the production replications, the corresponding operating temperature of the replications are sensed and the associated driving frequency or spring constant is fetched from the data storage 20 or, alternatively, computed by the equation. The linear motor is then driven at the stored or computed driving frequency associated with the sensed temperature. This process is repeated during the operation of the production machines so that the mechanical apparatus is always driven at the driving frequency that provides the least vibration amplitude for the most recently sensed temperature.

For mechanical apparatus that has both parasitic secondary vibration systems and a vibration balancer, there will be more resonant peaks to represent on a graph similar to Fig. 3. However, the method and procedures of the invention remain the same.

Fig. 4 illustrates a third example of an embodiment of the invention. It shows a prior art mechanical apparatus and control system to which components have been added for implementing the invention.

The illustrated prior art system has a main machine consisting of an electric linear motor 50 mechanically linked to drive internal moving masses 52 of itself and a driven load and mechanically connected to secondary vibrating systems 54. A sensor 56 senses a parameter of main machine operation, such as the top dead center (TDC) or piston position of one of the moving masses 52 and motor current and voltage are also detected. These signals are applied to a signal conditioner 58 and applied as a feedback signal to the summing junction 60 of a feedback control system, which are implemented in the software of a microcontroller or digital signal processor 62. A reference value is also applied to the summing junction 60 to provide an error signal that is applied to a transfer function and develop a control signal in accordance with well known feedback control system principles. For controlling the linear motor 50, the control signal is applied to a variable frequency inverter duty cycle generator circuit 64 that generates a square wave for which both the duty cycle and the frequency of the square wave can be controllably varied. The duty cycle is controlled in the prior art manner by the control signal. The

output of the generator circuit 64 is applied to an inverter output stage 66 that converts the square wave to oppositely directed dc pulses for driving the linear motor 50, the pulses having a pulse width corresponding to the duty cycle of the square wave. A variety of such circuits are known in the prior art and the invention is not limited to any particular drive circuit having these general characteristics.

In order to implement the present invention, a temperature sensor 68 is mounted close to the vibration balancer 54A and applies its output signal to the microcontroller 62 for storing temperature data for use in a lookup table 70 or a corresponding equation for use in determining the operating frequency as described in connection with Fig. 1. If the lookup table or equation provides values corresponding to the vibration balancer spring constant for each sensed temperature, that output is converted by a frequency adjustment algorithm 72 to determine the operating frequency. If the lookup table or equation directly provides frequency, the algorithm 72 may be omitted.

Fig. 5 illustrates a fourth example of an embodiment of the invention. It shows the same prior art mechanical apparatus and control circuit as illustrated in Fig. 4 to which components have been added for implementing the invention. For implementing the invention, a vibration sensor 80 is connected to provide a vibration amplitude input to the microcontroller 82. The microcontroller 82, under control of software module 84, scans, dithers or otherwise varies the operating drive frequency within the limits of the acceptable range of operating frequencies as described above and implements storage of the least amplitude of vibration within that range. The microcontroller 82 then selects and operates the linear motor 86 at the stored operating frequency that is associated with the least amplitude of vibration. It also repeats that process at selected intervals or under selected conditions, such as those described above, so that as operating conditions may change, the electric linear motor 86 is always driven at the frequency of least amplitude of vibration.

Figs. 6 and 7 are diagrams illustrating the fifth and sixth examples of embodiments of the invention. Both include the identical prior art device to which circuitry implementing the invention is added. Therefore, the common, prior art portion of both figures will be described first. Figs. 6 and 7 show a main machine that is a free

piston Stirling engine 100 driving an electric linear alternator 102 for generating electric power from a heat source input. They may be of a design known in the prior art and are typically mounted in the same housing. The reciprocating power piston of the Stirling engine is mechanically linked to the reciprocating component of the alternator, usually a series of permanent magnets supported on a carrier that reciprocates within an armature winding or coil mounted within the common housing. The composite mass 104 of these reciprocating structures is mechanically linked to a vibration balancer 106 and any parasitic vibration systems 108 through the reaction of the cylinders and casing of the Stirling engine and alternator. These reaction forces are transmitted to the cylinder and casing through the usual mechanical springs, working gas within the Stirling engine and the electromagnetic coupling from the magnets to the armature coil. The output of the alternator 102 is connected through a conventional tuning capacitor 109, used for power factor correction, to supply power to a useful load 110.

The frequency of the Stirling engine 100 is controlled using a principle known in the prior art for connecting a Stirling engine driven alternator to a power grid to supply power to the grid. The oscillations of the Stirling engine will synchronize with the AC power oscillations of the power grid if the Stirling engine driven alternator is designed to be resonant near the frequency of the power grid. The applied principle is that the oscillations of a Stirling engine driven alternator, when connected to an external AC power supply, will synchronize with the voltage oscillations of the AC power supply if the AC power supply has a lower internal impedance than the alternator and if there are only small variations in frequency. The frequency variations applicable to the invention and described above are such small variations.

A variable frequency, variable amplitude power source 111 is used as an engine output frequency controller and its AC output terminals are connected to the alternator 102. Such power sources are commercially available and are therefore not described further. The output of the variable frequency generator 118 of the microcontroller 114 is connected to the control input terminal 112 of the variable AC power source 111 which is the input that controls the frequency of the variable AC power source 111. The output of the variable frequency generator 118 controls the AC power source 111. Consequently,

within the small frequency variations used in the present invention, the operating frequency of the Stirling engine driven linear alternator 102 tracks the frequency of the variable AC power source 111 and therefore is controlled by the microcontroller 114.

The implementation of the invention illustrated in Fig. 6 operates similarly to the embodiment illustrated in Fig. 5. The controller 114 is programmed to perform the logic and arithmetic functions for computing the operating frequency of the main machine that minimizes the amplitude of vibration. A vibration sensor 116, such as an accelerometer, has an output connected to an input of the controller 114 for providing a signal representing the amplitude of the sensed vibration of the prior art, main machine illustrated in Fig. 6. However, instead of inputting a nominal operating frequency as a command input to an engine frequency controller, a variable frequency generator 118 is interposed between the nominal operating frequency command input and the input to the variable AC source 111. This permits the controller 114 to offset the commanded operating frequency from the nominal operating frequency in order to minimize vibration. The controller 114, under software control, scans, dithers or otherwise varies the operating drive frequency within the limits of the acceptable range of operating frequencies and implements storage of the least amplitude of vibration within that range. The controller 114 then selects and operates the Stirling engine 100 at the stored operating frequency that is associated with the least amplitude of vibration. It also repeats that process at selected intervals or under selected conditions, such as those described above, so that, as operating conditions may change, the free piston Stirling engine 100 is always driven at the frequency of least amplitude of vibration within the limits of the acceptable range of operating frequencies.

The implementation of the invention illustrated in Fig. 7 operates similarly to the embodiment illustrated in Fig. 2 and applies the same operating principles for controlling the operating frequency of the free piston Stirling engine 100 as described in connection with the embodiment of Fig. 6. The output of a temperature sensor 120 is connected to an input of a controller 122 to provide a signal representing the temperature of the vibration balancer 106. The controller 122 then operates the Stirling engine 100 over the range of frequencies of acceptable operation and stores each temperature in association with an

operating frequency to provide a lookup table 124 by the same process as described above in connection with Figs. 2 and 4.

While certain preferred embodiments of the present invention have been disclosed in detail, it is to be understood that various modifications may be adopted without departing from the spirit of the invention or scope of the following claims.

CLAIMS

1. A method for minimizing the amplitude of mechanical vibrations of a mechanical apparatus including a linear, freely reciprocating, prime mover coupled to and driving a reciprocating mass of a driven machine in reciprocation at a driving frequency, the coupled prime mover and driven machine having a spring applying a force upon the reciprocating mass to form a resonant main system having a main system resonant frequency of reciprocation, the method comprising:

(a) determining and storing a driving frequency range over which the driven machine operates at an acceptable efficiency of operation;

(b) sensing a parameter of the operation of the mechanical apparatus; and

(c) driving the prime mover in response to the sensed parameter at a driving frequency that

(i) is offset from the main system resonant frequency of reciprocation;

(ii) is within the driving frequency range of acceptable efficiency of operation; and

(iii) reduces or minimizes the amplitude of mechanical vibration of the mechanical apparatus under existing operating conditions.

2. A method in accordance with claim 1 wherein the sensing step comprises sensing the amplitude of vibration of the mechanical apparatus and wherein the method further comprises:

(a) sweeping the driving frequency over a frequency range that includes the main system resonant frequency of reciprocation;

(b) storing the sensed amplitude of vibration in association with a plurality of the sweeping drive frequencies; and

wherein the prime mover is driven at a driving frequency that is a stored driving frequency associated with the smallest, sensed, stored amplitude.

3. A method in accordance with claim 2 wherein the method is periodically repeated.

4. A method in accordance with claim 1 for controlling a mechanical apparatus in which the driven machine is a free piston, Stirling, heat pumping apparatus, wherein

(a) the driving frequency range is determined and stored in response to testing of at least one component of said mechanical apparatus;

(b) at least one heat pumping apparatus of the mechanical apparatus is operated during a test at a plurality of operating temperatures, for each of the operating temperatures the driving frequency is varied within the acceptable driving frequency range and the driving frequency resulting in the least amplitude of vibration of the mechanical apparatus is stored in association with each operating temperature;

(c) the operating temperatures and associated driving frequencies are stored as a lookup table in a memory device connected in a frequency control system of replications of the tested mechanical apparatus;

(d) the sensing step comprises sensing the operating temperature of the replications of the tested mechanical apparatus; and

(e) the prime mover is driven at the stored driving frequency associated with the sensed temperature.

5. A computer or logic circuit control system for minimizing the amplitude of mechanical vibrations of a mechanical apparatus including a main machine having a linear, freely reciprocating prime mover driving a driven machine in reciprocation, the main machine including a mass and springs applying a force upon the mass to provide a resonant mechanical oscillator having a resonant frequency near which the main machine is designed to be operatively driven, the control system comprising:

(a) a sensor for sensing a parameter of machine operation;

(b) a data storage for storing a driving frequency range over which the driven machine operates at an acceptable efficiency of operation; and

(c) a microcontroller connected to receive inputs from the sensor and the data storage for controlling the prime mover and programmed for driving the prime mover in response to the sensed parameter at a driving frequency that is offset from the

main machine's resonant frequency of reciprocation, is within the stored driving frequency range of acceptable efficiency of operation and minimizes the amplitude of mechanical vibration of the mechanical apparatus under existing operating conditions.

6. A control system according to claim 5 wherein the sensor senses the amplitude of vibrations of the mechanical apparatus.

7. A control system according to claim 6 wherein the sensor is an accelerometer.

8. A control system according to claim 5 wherein the sensor is a temperature sensor.

9. A control system in accordance with claim 8 wherein the sensor is connected to sense the temperature of a spring of a vibration balancer.

10. A control system in accordance with claim 5 wherein the prime mover is a linear, electric motor.

11. A control system in accordance with claim 5 wherein the prime mover is a free piston Stirling engine.

12. A control system in accordance with claim 5 wherein the driven machine is a free piston Stirling cooler.