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Low frequency vibration isolator with adjustable configurative parameter

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ABSTRACT

This paper will develop a low frequency vibration isolator with a mechanism for adjusting the configurative parameter (shorted LFVIM). This isolator offers a large isolation range, ensures a load bearing capacity and maintains the isolation effectiveness when the isolated weight is replaced. This way, adjustment of its configurative parameters is indispensable in accordance with self weight of newly placed isolated object. This will be realized by introducing a mechanism that enables easy and quick adjustment of the configurative parameters of the LFVIM. Then, the effects of the parameters including the dynamic stiffness at the desirable static equilibrium position, the self-weight of the isolated object and the slope of the dynamic stiffness curve on the steady state response as well as the isolation effectiveness of LFVIM are clearly analysed. A seventh order approximate polynomial function is used to explore the effects of these parameters. Then, the normal form method (NF) is employed to predict the frequency response curve of the LFVIM at the steady state. The simulation results show that the LFVIM can offer significant isolation advantages. Besides, these results furnish a useful insight for the design and analysis of the LFVIM.

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1. Introduction

Unwanted vibration is one of main causes which induce many damages of machinery and equipment such as fatigue or failure. Particularly, vibrations that occur in low frequency regions (<5 Hz) are almost harmful and dangerous to human health and activity [1]. Thus, isolators are widely used to lengthen the service life of the machinery, equipment as well as to provide more comfortable and safe condition for human in vehicle. Commonly, a linear vibration isolation system (traditional system) consisting of a linear stiffness spring in parallel to a damper given by S.G. Kelly [2] can only provide a good isolation effectiveness when the excitation frequency is greater than $\sqrt{2}$ times the natural frequency of the system. One strategy for extending attenuation ability to low frequency band is to reduce the stiffness of the system. Many scholars and engineers have been attracted by this idea. Such as by using geometrical nonlinearities, L.N. Virgin et al. [3–4] proposed a low frequency vibration isolator. In which, a thin strip is bent such that the two ends are brought together and clamped to form a teardrop shape. It acts as nonlinear spring for supporting the load and mitigating the transmissibility of a dynamic vertical excitation. By increasing the loop length, the loop becomes less stiff vertically. E. J. Chin et al. [5] showed that by implementing anti-spring technique the Euler spring vibration isolator was significantly improved. It means that the resonance frequency of Euler spring can be reduced. R.H. Paut et al. [6] have been studied application of pairs of pre-columns bonded with a viscoelastic filler as vibration isolator. However, a reduction in stiffness results in a low load bearing capacity and an excessive static deformation of the isolation system. This may be obstacle for improving the isolation effectiveness of the system.

In order to overcome the dichotomy between load bearing capacity and reduction in stiffness, in recent years, quasi-zero stiffness (QZS) vibration isolation systems have been studied and developed [7–11]. These systems are also referred to as high-static low-dynamic stiffness (HSLD) vibration isolation systems. Due to the negative and positive elements connected in parallel, the HSLD obtains the low dynamic stiffness around the static equilibrium position but still remains the load bearing capacity. Hence, this isolation way is available to achieve a low isolation frequency band. Currently, the QZS vibration isolation systems have been studied in great depth and widely applied in many fields such as the suspensions of the ground vehicle and seat, nanotech and the space shuttle. A spring with negative stiffness is designed by C.M. Le et al. [12] to improve vehicle driver vibration isolation seat in low excitation frequency region. This design presents an approach, based on the consistent theory of thin shells, for designing compact springs in terms of their compatibility with the room available for packaging the vehicle suspensions and simultaneous extension of the height control region where fundamental frequencies are kept minimal. Q. Li [13] proposed a negative stiffness magnetic suspension vibration isolator (NSMSVI) in which the magnetic spring acts in repulsion with positive stiffness, whereas the rubber membrane acts with negative stiffness. The NSMSVI can ob-

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Nomenclature

a	Length of bar in millimeter
b	Distance from wall to the mass in millimeter
C	Damping coefficient in N/m/s
d	Range of the displacement from the static equilibrium position over which the dynamic is always smaller unity in millimeter
D_y	Distance from the arbitrary equilibrium position to the desirable one in millimeter
F	Vertical restoring force of system in N
F_{ap}	Approximate restoring force of system in N
h_{sd}	Static deformation of the vertical spring in millimeter
K	Dynamic stiffness of system in N/m^2
K_v	Stiffness of vertical spring in N/m^2
K_h	Stiffness of horizontal spring in N/m^2
K_e	Dynamic stiffness of system at static equilibrium position in N/m^2
L_o	Original Length of horizontal spring in millimeter
M	Mass in kg
u	Relative displacement response of the isolated object in millimeter
U	Relative displacement amplitude of isolated object in millimeter
U_i	Relative displacement amplitude of the ith harmonic term (i=1,2,3…7) in millimeter
w	State vector
W	Amplitude of the trial solution for the vector state w
z	Absolute displacement response of the isolated object in millimeter
Z	Absolute displacement amplitude of isolated object in millimeter
Z_i	Absolute displacement amplitude of the ith harmonic term (i=1,2,3…7) in millimeter
Z_e	Excitation amplitude from base in millimeter
Greek letters	
α	Ratio of spring coefficient
β	Angle of bar to horizon in degree
γ_1, γ_2	Configurative parameters of the proposed system (γ_1=a/L$_o$; γ_2=b/L$_o$)
ξ	Dimensionless viscous damping coefficient
φ	Phase difference between excitation and response of system in radiant
ω	Excitation frequency in rad/s
ω_n	Natural frequency of the system without NSS in rad/s
Δ	The average value of the steady state response
Superscripts	
ˆ	Dimensionless quantity
•	Time derivative

tain low natural frequency as the positive stiffness partly counteracted by negative stiffness. X. Sun et al. [14] introduced a time-delayed active control strategy to improve the performance of a quasi-zero-stiffness vibration isolator. J. Zhou et al. [15] newly designed a quasi-zero stiffness isolator with cam-roller-spring mechanism in which the cam-roller-spring mechanism worked as a negative stiffness element is connected in parallel with a positive stiffness element to reduce the system's stiffness in the vertical direction. The effects of load imperfection on the isolation performance of the QZS isolator are studied by [16]. In addition, Q. Cao et al. [17] proposed a smooth or discontinuous (SD) nonlinear oscillator. The SD isolator consists of an isolated load linked a pair of inclined linear elastic springs which are capable of restricting both tension and compression, each spring is pinned to a rigid support. The limit case responses of the smooth and discontinuous (SD) oscillators ware clearly studied by Q. Cao et al. [18]. R. Tian et al. [19] successfully analyzed the codimension-two bifurcation phenomenon of the SD oscillator. Especially, the complicated nonlinear dynamic of the SD oscillator with quasi-zero stiffness was also presented and analyzed by Z. Hao et al. [20]. The SD oscillator with quasi-zero stiffness in which the lumped mass is connected with a vertical spring of positive stiffness and a pair of horizontally compressed springs providing a negative stiffness to obtain the quasi-zero stiffness, is widely used in vibration isolation.

Drawn by the QZS isolation approach, we developed and experimental investigated a vibration isolation system using negative stiffness structure (VIS with NSS) [21,22]. The simulation and experimental results indicate that isolation effectiveness of the VIS with NSS is better than that of the traditional isolation system. Then, some control algorithms have been designed for enhancing the isolation responses of the system as shown in [23,24]. This structure is simple and easy for application in practice. But, previous works only considered the influence of the dynamic stiffness on the isolation performance of the system when the isolated object is assumed to be unreplaced. As is known, the dynamic stiffness of the VIS with NSS is a nonlinear curve and depends on the static deformation of the load bearing spring. This deformation is changed due to the change of the weight of the isolated object. This phenomenon may affect the isolation capacity of the system.

Therefore, the present paper will broaden the previous works by realizing the dynamic analysis of the system in [21]. Normal form method [25] and seventh order Taylor series expansion around the static equilibrium position are employed for predicting the vibration transmissibility when the dynamic stiffness at the static equilibrium position, the self-weight of the isolated object and the slope of the dynamic stiffness curve are changed. Then, this paper will develop a low frequency vibration isolator with a mechanism for adjusting the configurative parameter so that the desirable static equilibrium position of the system is always remained when the isolation weight is replaced. This paper is organized as follows. The LFVIM is described in section 2. The dynamic stiffness of the LFVIM at the desirable static equilibrium position is presented in Section 3. The dynamic of LFVIM is analyzed in Section 4. In Section 5, Numerical simulation is presented. Finally, some conclusions are drawn in Section 6.

2. Description of LFVIM

The physical model of the developed low frequency vibration isolator is shown in Fig. 1(a). As shown, the isolation object is supported by the vertical spring. Two horizontal springs are used to reduce the stiffness of the LFVIM in the vertical direction. The adjustable mechanism includes the screw jack 1 and 2, the belt driver, the bevel-gear train and two linear motors. One end of vertical spring and damper are fixed on the plate which is slid in the vertical direction through the linear bush. The screw jack 1 is employed to adjust the position of the isolation object so that the LFVIM is always at the desirable static equilibrium position as shown in Fig. 1(b). In order to obtain the low dynamic stiffness at the equilibrium position, the length of the bar is adjusted by using the linear motors. The distance between two horizontal springs is also changed by rotating the screw jack 2. Half of the length of the screw jack 2 is threaded in the opposite direction to that of the other half.

3. The dynamic stiffness of the LFVIM at the desirable static equilibrium position

At the wanted static equilibrium position (denoted by dashed line) as shown in Fig. 1(b), the isolation object is held in equilibrium by the compression force of the vertical spring F_v and the gravity force which is opposite of the force F_v. Therefore, in this case, the load supported capacity of this system only depends on the stiffness of the vertical spring and its static deformation.

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