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Numerical and experimental analysis of a vibration isolator equipped with a negative stiffness system

E. Palomares *, A.J. Nieto, A.L. Morales, J.M. Chicharro, P. Pintado

Department of Mechanical Engineering, University of Castilla - La Mancha, Avda. Camilo José Cela s/n, 13071, Ciudad Real, Spain

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ABSTRACT

This paper presents a Negative Stiffness System (NSS) based on a set of two double–acting pneumatic linear actuators (PLA). The NSS is added to a system with a single degree of freedom, which consists of a sprung mass and a pneumatic spring. One end of each PLA is jointed to the sprung mass while the other end is jointed to the vibrating frame. In addition, the PLAs are symmetrically arranged so that they remain horizontal while the sprung mass is in static conditions. When the rear chamber is pressurised, the vertical component of the force applied by the PLAs will work against the pneumatic spring reducing the dynamic resonance frequency of the overall system. Experimental tests and simulations showed improvements regarding sprung mass isolation in comparison to the passive system without NSS, decreasing the resonance frequency by up to 58% and improving the vibration attenuation for different experimental excitations.

1. Introduction

Many engineering systems are affected by undesirable vibrations. In the simple case of a mass m supported by a linear spring with stiffness k, isolation does not occur until a frequency of $\sqrt{2k/m}$ is reached. This means that the frequency range of isolation increases while the stiffness of the spring decreases, but this results in a larger static displacement of the mass. To solve this limitation, negative stiffness systems are developed by using nonlinearities such as spring orientation or buckling [1,2].

In the scientific literature one may find a more specific isolator with zero dynamic stiffness, the so–called quasi–zero–stiffness mechanism (QZS) [3,4]. This is commonly used in some special applications, e.g., in areas like space research [5], the isolation of high–precision machinery such as seismographs or gravimeters [6], or vibration isolation platforms [7]. These QZS mechanisms are normally generated by combining a negative stiffness element with a positive stiffness element. The simplest QZS mechanism is loaded with a suitable sized mass, with oblique springs compressed so as to be in the horizontal position. The static load is supported by the vertical spring. This position corresponds to the static equilibrium position and the oblique springs are used as the negative stiffness device in the vertical direction counteracting the positive stiffness of the vertical spring. Some authors [8] studied this trade–off between achieving a low dynamic stiffness for small displacements and acceptable stiffness for large displacements. As a result of this study, an optimum relationship between the ratio of the oblique spring stiffness (assuming these springs to be linear and not preloaded) and the vertical spring stiffness is achieved as well as an optimum angle for the oblique springs.

Carrella et al. [9] compared three cases for the oblique springs: linear springs, pre–stressed springs and pre–stressed non–linear springs which have a softening characteristic with the restoring force expressed by a cubic polynomial. They demon-

* Corresponding author.
E-mail address: Eduardo.Palomares@uclm.es (E. Palomares).

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strated that the latter configuration shows advantages with respect to the linear set of springs and achieves reductions in the transmissibility function of up to 20 dB per decade.

Nonlinear isolators with high–static–low–dynamic–stiffness (HSLDS) have been developed both theoretically and experimentally and their applications have been summarised by Ibrahim [10]. The vibration isolation performance of a nonlinear isolator increases the frequency range of isolation. One way to construct the nonlinear isolator is by using negative stiffness configurations, such as magnetic springs [11,12], Euler buckled beams [13–17], kirigami cellular structures [18], etc. An optimization of the negative stiffness structure can also lead to larger damping ratios [19–21].

Other authors [22] propose a development of a HSLDS system that can act passively but also semi–actively. The semi–active behaviour is achieved by connecting a mechanical spring in parallel with a number of magnetic springs of variable stiffness. The mechanical spring essentially consists of a beam with a stiffness that increases progressively.

Mizuno et al. [23] proposed a vibration isolation system using negative stiffness obtained by actively controlling an actuator. The two–degree–of–freedom isolator consists of the already mentioned actuator connected in series with a coil spring through a middle mass. They used different types of actuators like voice coil motors [23], pneumatic linear actuators [24] or electromagnets [25]. They proved the effectiveness of the system experimentally.

Other authors consider the problem of vibration at low excitation frequencies (0.5–5.0 Hz), focusing on the risk factors for the physical health of drivers and passengers. Their main goal is to design vibration isolation structures for improving vibration isolation effectiveness of the vehicle seat under low excitation frequencies upgrading the ride comfort and safety [26]. The study is based on the work of Carrella et al. [8] which has the negative stiffness structure in parallel with the positive stiffness structure.

Le et al. [26] propose a vibration isolator that incorporates two symmetrical structures of negative stiffness. These are two bars hinged to the sprung mass, perpendicular to the vertical movement of the mass. In turn, these bars are linked to a pre–stressed spring at its other end. The experimental results confirmed better isolation for a wide range of frequencies. They applied those results in order to reduce vibrations of vehicle seats [27]. The passive characteristic in the system gives it limited versatility at low frequencies, and furthermore, this system is not able to offer solutions for different suspended masses. They propose an active system incorporating a pneumatic actuator in the same direction as the sprung mass movement and in parallel with the spring that provides stiffness under static conditions [28]. The experimental results confirmed the improvement in comparison with the systems without negative stiffness.

In this paper, we present a negative stiffness mechanism using a pair of double effect pneumatic actuators initially in a horizontal position, perpendicular to the motion of the mass. The advantage of these pneumatic actuators is the possibility to decrease dynamic stiffness by means of the variation of pressure within these actuators. On the other hand, a pneumatic spring is also included since it is capable of providing sufficient stiffness in static conditions and, in addition, is able to adapt to possible variations of mass. The negative stiffness system (NSS) is then incorporated into a single degree of freedom system (SDOF) where the sprung mass rests on a pneumatic spring. Experimental tests were also made in order to validate the system.

2. Negative stiffness system proposed (NSS)

2.1. Description of the system

The schematic layout of the negative stiffness system can be seen in Fig. 1. This figure shows a pair of pneumatic linear actuators (PLA) placed at each side of the sprung mass m. They are located within a classical one degree of freedom system formed by said sprung mass m and a pneumatic spring. In this mechanism the actuators are set one in front of the other and, in static conditions, they remain in the perpendicular direction to mass relative displacement $z(t) - z_0(t)$. The end of the actuator rod is connected via a ball joint to the mass, while the end of the actuator body is connected in the same way to the vibrating frame, which is excited by an input displacement z_0.

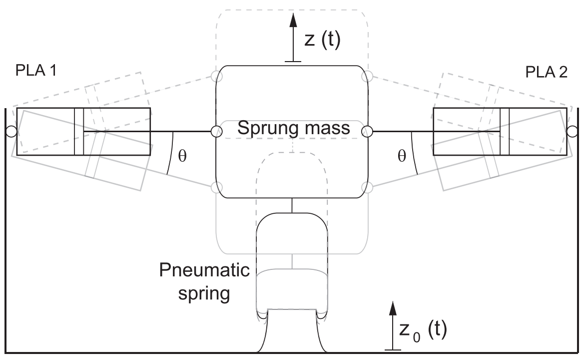

Fig. 1. Schematics of NSS device assembled with the SDOF system.