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# Dynamic counterbalancing the single-piston linear compressor of a Stirling cryogenic cooler

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ABSTRACT

Low vibration Stirling cryocoolers often rely on dual-piston linear compressors, the known disadvantages of which, as compared to their single-piston rivals, are: low reliability, increased power consumption, price, bulk, sensitivity to external vibration and *g*-forces. However, because of the inherently low level of vibration export, as required in numerous vibration sensitive electronic and electro-optic applications, the dual-piston approach has become prevalent in today's industrial practice.

The authors report on the novel approach to the passive control of a fundamental component of a vibration export from a single-piston compressor down to the levels typical for the actively controlled dual-piston rival. The technique relies on the newly proposed principle of dynamic counterbalancing, where an auxiliary movable mass is flexibly attached to a movable piston assembly and to the stationary compressor casing using auxiliary mechanical springs. The proper design of such a "spring–mass–spring" counterbalancer yields zero vibration export at minimum electrical power and current consumed by the motor.

Based on the theoretical analysis, the design of the single-piston compressor of 1 W@77 K Ricor model K529N Stirling cryocooler was enhanced by adding such a counterbalancer. The obtained experimental results are in full agreement with the theoretical prediction. From experiment, the vibration export at driving frequency was reduced 57-fold at practically the same electrical current and power consumed by the compressor actuator as compared with the basic cooler.



## 1. Introduction

Linear Stirling coolers typically comprise a linear compressor providing for the required pressure pulses and the volumetric reciprocal change of a working agent (Helium, typically) in the expansion space of the cold finger.

The above linear compressors are typically driven by linear motors of the "moving coil" or "moving magnet" types. In the "moving coil" linear actuator an alternating current is passed through a coil attached to a piston assembly and located between the poles of a stationary permanent magnet. In a "moving magnet" linear actuator the coil is fixed and the permanent magnet is attached to the piston assembly.

The single-piston linear compressors are known to be major sources of harmful vibration export produced by the cryogenic coolers. This is caused, primarily, due to an unbalanced motion of the moving piston assembly, where the resulting vibration export is, in essence, the force of inertia developed by the moving parts with the magnitude being the product of the total moving mass, the magnitude of reciprocation and the driving frequency squared. The above unbalance leads to an essential vibration export comprising a well-pronounced fundamental frequency component, along with comparatively small higher-order harmonics. Since the prior-art designs do not allow for canceling such vibration at the source level, it is inevitably exported into the supporting structure to which the cooler is normally rigidly attached.

The simplest known approach to minimize this force relies on reducing the moving assembly mass, piston stroke and driving frequency. However, with such manner, the force export may be reduced to a particular limit only.

Another approach is based on using internal vibration isolation when the cylinder assembly is flexurally mounted inside the compressor casing and the gas transfer from the above movable cylinder assembly to the stationary casing relies on the flexible transfer line [1]. By choosing the natural frequency of such a vibration isolator to be well below the driving frequency, the vibration export may be essentially reduced. The drawback of such an approach is that under severe environmental conditions, such as shock, wideband random vibration, sine vibration with variable frequency and high *g*-forces, the above cylinder assembly is capable of developing excessive dynamic deflection damaging the internal transfer line along with the compressor interior. Tight tolerances, imposed on the free rattle space, call for the use of a stiff and heavily damped internal isolator, thus diminishing its performance. Additional drawback is the limited lifespan of the internal flexible transfer line.

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A further approach is based on the principle of external vibration isolation when the entire compressor is supported from the enclosure of a cooled device using a compliant vibration-isolating and thermo-conductive interface. Examples of such an arrangement are disclosed, for instance, in US patents 5129,232 and 5864,273 [2,3]. The known disadvantages of this approach are: complication of the mechanical design, increase in mass and bulk, sensitivity to external vibration, shock and *g*-forces, additional heat load on the cooler, complicated heat sinking and, therefore, degraded cooling performance.

In US patent 5895,033 [4] it is shown how a tuned dynamic absorber may be applied to the vibration protection of the sensitive equipment mounted in close proximity to the liner compressor. In this approach, the natural frequency of an auxiliary undamped spring-mass system, which is externally attached to the compressor housing, has to be precisely equal to the driving frequency. Since the compressor housing is normally attached rigidly to the system's enclosure, the known disadvantage of such an approach is that the relatively bulky dynamic absorber (20% of the entire system, typically) is required in order to achieve the desired vibration control. Further known disadvantages inherent to this approach are high sensitivity to even a small variation in the driving frequency and to environmental vibration and shock.

A combination of a stiff and heavily damped vibration isolation mounting and a tuned dynamic absorber is disclosed in [5] and US patent 4860,543 [6]. Using an optimally stiffened and damped vibration isolator allows the use of an essentially smaller dynamic absorber (20% of the compressor mass, typically). However, the above-mentioned sensitivity to the driving frequency, complication of mechanical design, heat sinking and degraded performance of such an arrangement, prevents this approach from being used in many practical applications.

Recent US patents disclose active vibration cancellation systems containing a vibration sensor, sophisticated digital controller and an actuator, as may be seen, for example, in US patents 5582,013 and 6169,404 [7,8]. Such active systems yield very accurate suppression of vibration disturbance produced by the compressor. The disadvantages of this approach are, however, higher costs and bulk, lower reliability, additional power consumption in use and, consequently, additional heat emission.

The vibration export from the linear compressor may also be eradicated using a dual-piston design where the fine balancing of the fundamental and higher-order harmonics is usually attained by an active control system relying on appropriate sensors and a sophisticated digital controller, as disclosed, for example, in US patent 5535,593 [9]. This approach to vibration control in the linear compressors of Stirling cryogenic coolers has become prevalent in today's industrial practice. The known disadvantages are, however: increase in price and bulk, decrease of reliability, cryocooler efficiency and excessive sensitivity to the *g*-forces [10].

It is, therefore, our primary objective to practically eliminate vibration export from the linear single-piston compressor of a Stirling cryogenic cooler without compromising the power consumed by the compressor motor while incurring minimally increased construction costs, allowing said cryogenic cooler to be sold at practically the same price as is prevalent today for prior-art coolers.

The above objective may be achieved by using a passive auxiliary "spring-mass-spring" counterbalancer, as detailed below.

## 2. Dynamic model of a single-piston linear compressor with dynamic counterbalancer

In Fig. 1, the piston assembly of total mass $m_0$ is driven by the force $F(t) = \alpha i$, where $\alpha$ is the motor force/current constant and $i$ is the motor current resulting from application of the driving voltage $u$. It is important to note that the above driving force is applied to not only the piston assembly, but also to the compressor housing, as shown in Fig. 1. The exact definition of gasodynamic forces produced by the piston motion is quite complex and determined by the entire cooler thermodynamics; a detailed knowledge of these forces, however, is not required for the purpose of the present study. In a simplistic approach [11], the entire gasodynamic interaction of the piston and the cylinder head may be thought of as a linear parallel spring-dashpot combination $k_0$ and $c_0$. The counterbalancer of total mass $m_1$ is driven by the piston through the first auxiliary spring having spring rate $k_1$ and flexibly attached to the stationary housing using the second auxiliary spring having spring rate $k_2$. The piston and counterbalancer motions are $x_0(t)$ and $x_1(t)$, respectively.

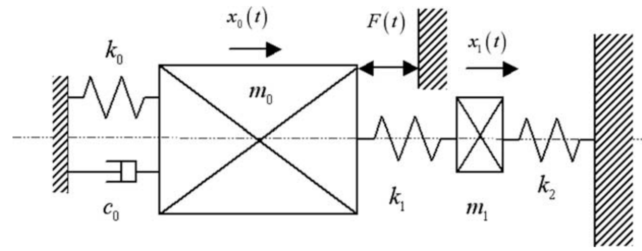


**Fig. 1.** Dynamic model of a single-piston linear compressor with dynamic counterbalancer.

The equations of motion [12,13] take into account the interaction of mechanical and electrical subsystems; these are

$$\left\{\begin{array}{l} m_0\frac{d^2x_0}{dt^2} + k_0x_0 + c\frac{dx_0}{dt} + k_1(x_0 - x_1) = F(t), \\ m_1\frac{d^2x_1}{dt^2} + k_1(x_1 - x_0) + k_2x_1 = 0, \\ L\frac{di}{dt} + Ri + \alpha\frac{dx_0}{dt} = u. \\ F(t) = \alpha i \end{array}\right\} \tag{1}$$

The vibration export may be easily calculated as a sum of all forces applied to the housing, that is

$$\begin{aligned} Q(t) &= -k_0x_0 - c\frac{dx_0}{dt} + F(t) - k_2x_1 \\ &= -k_0x_0 - c\frac{dx_0}{dt} + \alpha i - k_2x_1. \end{aligned} \tag{2}$$

From Eqs. (1) and (2),

$$Q(t) = m_0\frac{d^2x_0}{dt^2} + m_1\frac{d^2x_1}{dt^2} \tag{3}$$

which is, in essence, the sum of the inertial forces developed by the moving masses.

From Eq. (3) it is seen that vibration export from such a compressor may be cancelled if the piston and counterbalancer reciprocate oppositely and the magnitudes of their reciprocation are related as their mass ratio, that is

$$m_0\frac{d^2x_0}{dt^2} = -m_1\frac{d^2x_1}{dt^2}. \tag{4}$$

It is evident, that developing such a dynamic counterbalancing requires optimal tuning of its parameters, namely: the mass and rates of the two auxiliary springs. Now we turn our attention to finding such optimal parameters.

To start with, we further ignore the presence of the small higher-order harmonics in the time histories representing displacements, current and voltage. A transition into the frequency domain yields: $x_0(t) \Longleftrightarrow X_0(j\omega), x_1(t) \Longleftrightarrow X_1(j\omega); i(t) \Longleftrightarrow I(j\omega)$