ADJUSTABLE SUSPENSION ELEMENTS AND SEMI-ACTIVE SUSPENSION DESIGN

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Abstract: Semiactive – (SA) - suspensions are those which otherwise passively generated damping or spring forces modulated according to a parameter tuning policy with only a small amount of control effort. SA suspensions, as their name implies, fill the gap between purely passive and fully active suspensions and offer the reliability of passive systems, yet maintain the versatility and adaptability of fully active devices. Because of their low energy requirement and cost, considerable interest has developed during recent years toward practical implementation of these systems. This paper presents the basic theoretical concepts for SA suspensions’ design and implementation, followed by an overview of recent developments and control techniques. Some related practical developments ranging from vehicle suspensions to civil and aerospace structures are also reviewed.

1. Introduction

In most of today’s mechatronic systems a number of possible devices, such as reaction or momentum wheels, rotating devices, and electric motors are essential to the systems’ operations. These devices, however, can also be sources of detrimental vibrations that may significantly influence the mission performance effectiveness, and accuracy of operation. Several techniques are utilized to either limit or alter the vibration response of such systems.

Vibration isolation suspensions and vibration absorbers are quoted in the literature as the two most commonly used techniques for such utilization. In vibration isolation either the source of vibration is isolated from the system of concern also called “force transmissibility, see Figure 1a, or the device is protected from vibration

![Diagram](image-url)

**Fig. 1.** Schematic of (a) force transmissibility for foundation isolation, (b) displacement transmissibility for protecting device from vibration of the base, and (c) application of vibration absorber for suppressing primary system vibration.
of its point of attachment - also called displacement transmissibility, see Figure 1b. Unlike the isolator, a vibration absorber consists of a secondary system-usually mass–(spring)–damper trio added to the primary device to protect it from vibrating see Figure 1c. By properly selecting absorber mass, stiffness and damping, the vibration of the primary system can be minimized [1].

Passive, active, and semiactive are referred to in the literature as the three most common classifications of suspension systems (either as isolators or absorbers), see Figure 2, [2]. A suspension system is said to be active, passive, or semiactive depending on the amount of external power required for the suspension to perform its function.

A passive suspension consists of a resilient member (stiffness) and an energy dissipater (damper) to either absorb vibratory energy or load the transmission path of the disturbing vibration, Figure 2a, [3]. It performs best within the frequency region of its highest sensitivity.

For wideband excitation frequency, its performance can be improved considerably by optimizing the suspension parameters, [4], [5]. However, this improvement is achieved at the cost of lowering narrowband suppression characteristics.

The passive suspension has significant limitations in structural applications where broadband disturbances of highly uncertain nature are encountered. To compensate for these limitations, active suspension systems are utilized.

With an additional active force introduced as a part of suspension, \( u(t) \) in Figure 2b, the suspension is then controlled using different algorithms to make it more responsive to source of disturbances [6], [7]. A combination of active/passive treatment is intended to reduce the amount of external power necessary to achieve the desired performance characteristics.

![Fig.2. A typical primary structure equipped with three versions of suspension systems: (a) passive, (b) active, and (c) semi-active configuration.](image)

**2. Adjustable Suspension Elements**

Adjustable suspension elements typically are comprised of a variable rate damper and stiffness. Significant efforts have been devoted to the development and implementation of such devices for a variety of applications. Examples of such devices include electro-rheological (ER), magnetorheological (MR) fluid dampers, variable orifice dampers, controllable friction braces, controllable friction isolators, and variable stiffness and inertia devices. The conceptual devices for such adjustable properties are briefly reviewed in this section, [8].

A common and very effective way to reduce transient and steady-state vibration is to change the amount of damping in the SA suspension. Considerable design work of semi-active damping was done in the 1960s through 1980s, [8] for vibration control of civil structures such as buildings and bridges, [9] and for reducing machine tool oscillations.

Since then, SA dampers have been utilized in diverse applications ranging from trains [39] and other off-road vehicles [40] to military tanks, [10]. During recent years considerable interest in improving and refining the SA concept has arisen in industry.
Recent advances in smart materials have led to the development of new SA dampers, which are widely used in different applications. In view of these SA dampers, electro-rheological (ER) and magneto-rheological (MR) fluids probably serve as the best potential hardware alternatives for the more conventional variable-orifice hydraulic dampers, [11].

From a practical standpoint, the MR concept appears more promising for suspension because it can operate, for instance, on a vehicle’s battery voltage, whereas the ER damper is based on high-voltage electric fields. Due to their importance in today’s SA damper technology, we briefly review their operation and fundamental principles.

3. Semi-Active suspensions design

In the design of a suspension system, the system is often required to operate over a wideband load and frequency range which is impossible to meet with a single choice of suspension stiffness and damping. If the desired response characteristics cannot be obtained, active suspension may provide an attractive alternative vibration control for such broadband disturbances.

However, active suspensions suffer from control-induced instability in addition to the large control effort requirement. This is a serious concern that prevents common usage in most industrial applications. On the other hand, passive suspensions are often hampered by a phenomenon known as “de-tuning.”

De-tuning implies that the passive system is no longer effective in suppressing the vibration as it was designed to do. This occurs because of one of the following reasons:

- the suspension structure may deteriorate and its structural parameters can be far from the original nominal design,
- the structural parameters of the primary device itself may alter, or
- the excitation frequency and/or nature of disturbance may change over time.

Semi-Active also known as adaptive-passive suspension addresses these limitations by effectively integrating a tuning control scheme with tunable passive devices. For this, active force generators are replaced by modulated variable compartments such as a variable rate damper and stiffness, see Figure 2c. [12].

These variable components are referred to as “tunable parameters” of the suspension system, which are re-tailored via a tuning control and thus result in semi-actively inducing optimal operation. Much attention is being paid to these suspensions for their low energy requirement and cost.

Recent advances in smart materials and adjustable dampers and absorbers have significantly contributed to the applicability of these systems. SA suspensions can achieve most of the performance characteristics of fully active systems, thus allowing for a wide class of applications.

![Fig. 3. Application of a Semi-Active absorber to SDOF primary system with adjustable stiffness $k_a$ and damping $c_a$.](image)
The idea of SA suspension is very simple: to replace active force generators with continually adjustable elements which can vary and/or shift the rate of energy dissipation in response to an instantaneous condition of motion. This section presents basic understanding and fundamental principles and design issues for SA suspension systems, which are discussed in the form of a vibration absorber and vibration isolator.

With a history of almost a century, vibration absorbers have proven to be useful vibration suppression devices, widely used in hundreds of diverse applications. It is elastically attached to the vibrating body to alleviate detrimental oscillations from its point of attachment see Figure 2. The underlying proposition for an SA absorber is to properly adjust the absorber parameters so that it absorbs the vibratory energy within the frequency interval of interest. To explain the underlying concept, a single-degree-of-freedom (SDOF) primary system with a SDOF absorber attachment is considered Figure 3. The governing dynamics are expressed as:

\[ m_a \ddot{x}_a(t) + c_a \dot{x}_a(t) + k_a x_a(t) = c_a \dot{x}_p(t) + k_a x_p(t), \]  
\[ m_p \ddot{x}_p(t) + (c_p + c_a) \dot{x}_p(t) + (k_p + k_a) x_p(t) - c_a \dot{x}_a(t) - k_a x_a(t) = f(t), \]

where \( x_p(t) \) and \( x_a(t) \) are the respective primary and absorber displacements, \( f(t) \) is the external force, and the rest of the parameters including adjustable absorber stiffness \( k_a \) and damping \( c_a \) are defined per Figure 3. The transfer function between the excitation force and primary system displacement in Laplace domain is then written as:

\[ T_F(s) = \frac{X_p(s)}{F(s)} = \frac{m_a s^2 + c_a s + k_a}{H(s)}, \]

where:

\[ H(s) = m_p s^2 + (c_p + c_a) s + k_p + k_a (m_a s^2 + c_a s + k_a) - (c_a s + k_a)^2, \]

and \( X_p(s) \), \( X_a(s) \) and \( F(s) \) are the Laplace transformations of \( x_p(t) \), \( x_a(t) \) and \( f(t) \) respectively, [7], [9], [12].

The steady-state displacement of the system due to harmonic excitation is then:

\[ \left| \frac{X_p(j\omega)}{F(j\omega)} \right| = \left| \frac{k_a - m_a \omega^2 + j c_a \omega}{H(j\omega)} \right|. \]

where \( \omega \) is the disturbance frequency and \( j = \sqrt{-1} \). Utilizing adjustable properties of the SA unit (i.e., variable rate damper and spring), an appropriate parameter tuning scheme is selected to minimize the primary system’s vibration subject to external disturbance \( f(t) \).

When excitation is tonal, the absorber is generally tuned at the disturbance frequency, [3], [9]. For complete attenuation, the steady state \( X_p(j\omega) \) must equal zero. Consequently, from Equation (5), the ideal stiffness and damping of SA absorber are adjusted as:

\[ k_a = m_a \omega^2, \quad c_a = 0. \]
Note that this tuned condition is only a function of absorber elements \((m_a, k_a \text{ and } c_a)\). That is, the absorber tuning does not need information from the primary system and hence its design is stand-alone. For tonal applications, theoretically zero damping in an absorber subsection results in improved performance. In practice, however, damping is incorporated to maintain a reasonable tradeoff between the absorber mass and its displacement.

Hence, the design effort for this class of applications is focused on having precise tuning of an absorber to the disturbance frequency and controlling damping to an appropriate level. Referring to Snowdon, [12] it can be proven that the absorber, in the presence of damping, can be most favorably tuned and damped if adjustable stiffness and damping are selected as:

\[
k_{opt} = \frac{m_a m_p^2 \omega^2}{(m_a + m_p)^2}, \quad c_{opt} = \sqrt{\frac{3k_{opt}}{2(m_a + m_p)}}.
\]

### 4. Broadband Excitation

In broadband vibration control, the absorber subsection is generally designed to add damping to and change the resonant characteristics of the primary structure to maximally dissipate vibrational energy over a range of frequencies. The objective of SA suspension design is, therefore, to adjust the absorber parameters to minimize the peak magnitude of the frequency transfer function \(\{FTF(\omega) = \text{FTF}_s(\omega)\} \) over the absorber variable suspension parameter vector \(p\) as:

\[
\min_p \left\{ \max_{\omega_{min} < \omega < \omega_{max}} \{\text{FTF}(\omega)\} \right\}
\]

Alternatively, one may select the mean square displacement response (MSDR) of the primary system for vibration suppression performance. That is, the absorber variable parameters' vector \(p\) is selected such that the MSDR:

\[
E\{\ddot{x}_p\} = \int_0^\infty \{\text{FTF}(\omega)\}^2 S(\omega) d\omega
\]

is minimized over a desired wideband frequency range. \(S(\omega)\) is the power spectral density of the excitation force \(f(t)\), and FTF was defined earlier.

This optimization is subjected to some constraints in space, where only positive elements are acceptable. Once the optimal absorber suspension properties, \(c_a\) and \(k_a\) are determined they can be implemented using adjustment mechanisms on the spring and the damper elements.

This is viewed as a semi-active adjustment procedure as it introduces no added energy to the dynamic structure. The conceptual devices for such adjustable suspension elements will be discussed later in (3).

To better recognize the effectiveness of the SA absorber over the passive and optimum passive absorber settings, a simple example case is presented. For the simple system shown in Figure 3, the following nominal structural parameters (marked by over score) are taken:

\[
\begin{align*}
m_p &= 5.75 \text{ kg}, \quad k_p = 252 \times 10^6 \text{ N/m}, \quad c_p = 198 \text{ kg/s}; \\
\bar{m}_a &= 0.225 \text{ kg}, \quad \bar{k}_a = 9.81 \times 10^6 \text{ N/m}, \quad \bar{c}_a = 356 \text{ kg/s}
\end{align*}
\]

These are from an actual test setting which is optimal by design. That is, the peak of FTF is minimized, see thinner line in Figure 4.

When the primary stiffness and damping increase 5% (for instance, during the operation) the FTF of the primary system deteriorates considerably (dashed line in Figure 4), and the absorber is no longer an optimum one for the present primary.
When the absorber is optimized based on optimization problem (8), the re-tuned setting is reached as:

\[ k_a = 10.3 \times 10^6 \, \text{N/m}, \quad c_a = 364 \, \text{kg/s}. \]  

which yields a much better frequency response, see darker line in Figure 4.

The SA absorber effectiveness is better demonstrated at different frequencies by a frequency sweep test. For this, the excitation amplitude is kept fixed at unity and its frequency changes every 0.15 seconds from 1860 to 1970 Hz.

References