Vibration control of a medium-sized vehicle by a novel active engine mount

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Abstract

The paper presents the development and the performance testing of an engine mount featuring a stroke-amplified piezo stack actuator that is decoupled from the quasi-static loads. The actuator introduces its counteracting force through a viscous damper, acting as a high-pass filter. An elastic coupling element arranged in parallel with the actuator carries the static and quasi-static loads. This arrangement reduces the load requirements for the actuator.

Because the engine induces multi-order narrowband distortions, the controller uses a narrowband feedforward control strategy (Filtered-X LMS) to compensate forced and resonant distortions, minimizing the acceleration of the base structure. For the performance tests the active engine mount is installed in a test vehicle mounted on a full-vehicle test bench [1].

1. Introduction

In many cases, the potential for improvement of passive vibration reduction has been all but exhausted. Active vibration control systems provide further opportunities to suppress unwanted vibrations [2].

Existing active mounting systems often utilize a serial arrangement of an actuator and a passive elastic coupling element [3]. This arrangement carries the disadvantage that the actuator is fully exposed to the static loads, which may exceed the dynamic loads by an order of magnitude. At the same time the actuator must be capable of a large enough stroke, in order to exert enough force on the vibrating structure through the elastic coupling element. Usually these requirements will result in a relatively large actuator.

Previously published work describes the model-based development of a novel engine mount topology and experimental testing on a test bench [4, 5].

The mount features a stroke-amplified piezo stack actuator which introduces its counteracting force through a coupling element consisting of a serial arrangement of a spring and a damper.

The purpose of this is to minimize the transmission of static and quasi-static loads onto the actuator, while ensuring a stiff connection to the vibrating structure at higher frequencies. The bulk of the static and quasi-static loads are carried by a second elastic coupling element arranged in parallel with the actuator.

This reduces the load requirements for the actuator and better suits the use of a stroke-amplified piezo actuator.

In this paper the advancement of this mount for the installation in a vehicle, including the test results is presented.

2. Measurement of the Series Configuration

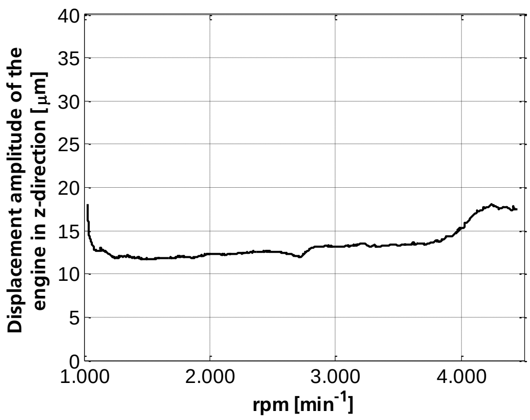

Fig. 1: *Dynamic, 2nd-order displacement of the engine at the left mount*

The most dominant disturbing forces working on the car body result from the combustion process in the engine.

As the test vehicle is powered by a 4-cylinder engine, the second engine order dominates the vibration along the working direction of the active mount.

For the dimensioning of the active mount it is necessary to know the present motor displacement amplitude at the engine mount position. The engine displacement amplitude is derived from acceleration measurements at the left engine mount, as this mount is to be replaced by the developed active one. Fig. 1 presents the resulting second order displacements during an engine run-up, showing an almost constant displacement of around 15µm.

The total engine displacement might be much greater due to static deflection or excitation of different engine orders.

With the measured motor displacement and the dynamic stiffness of the engine mount, the disturbance forces are estimated along the working direction.

3. Design of the Engine Mount

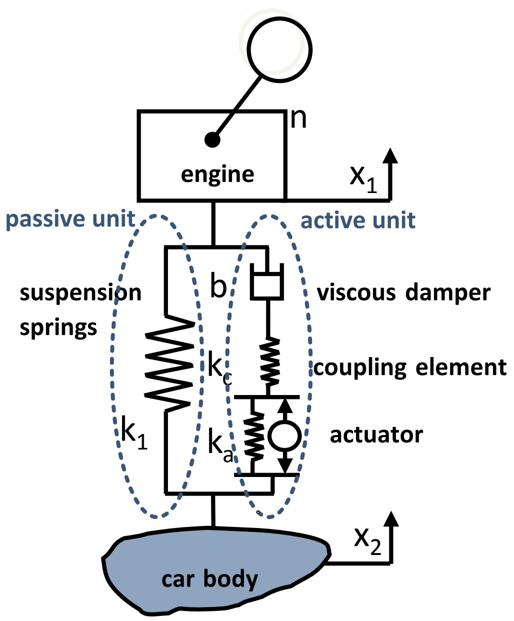

Fig. 2: *Topology of the active mount*

As shown in Fig. 2 the engine mount is functionally divided into an active and a passive assembly. The active unit, which exerts the counteracting force along the z-axis, consists of a piezo stack actuator in series with a coupling element and a viscous damper. The coupling element ensures the correct total stiffness of the active unit and protects the actuator against overload. This arrangement can be described as a Maxwell unit. According to this, the whole system consists of a Maxwell unit in parallel with the suspension spring. By use of this topology two beneficial properties arise. At first, the actuator does not have to carry static and low frequency loads. Moreover this topology leads to a high isolation with a decay of -40dB/decade [6]. At high frequencies the damper is stiff and the system behaves like a pure spring consisting of k_1 and k_2.

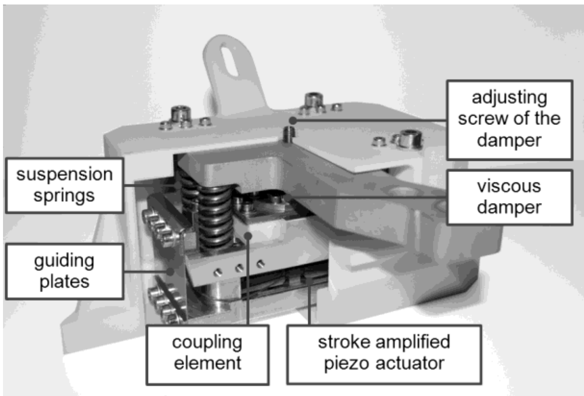

Fig. 3: *Active mount in cross-sectional view*

The passive assembly, which carries the horizontal and static vertical engine loads, consists of three nested sets of springs; one set for each spatial direction. The vertical, coil type springs are selected to provide a stiffness similar to that of the original mount. The two sets of parallel plate springs that enable a defined horizontal mobility while being as stiff as possible in the vertical.

The entire spring assembly is suspended from the lid of the mount housing to ensure that the plate springs are always in tension. The engine bracket is placed on top of the coil type springs. Rubber stops are placed at the cantilever to protect the active unit from overshoots. A cross-sectional view of the mount without the rubber stops is shown in Fig. 3.

3.1 The Active Unit

As the rotational speed of the engine may reach 6000rpm, a resonance-free operation of the active mount in the second order frequency range up to 200Hz has to be ensured. For the technical design, a stroke amplified piezo actuator with a first resonance frequency at about 220Hz is chosen. A 200V multilayer actuator is implemented.

Using the known engine displacement amplitude and the dynamic stiffness of the mount and including a safety factor, the force required to compensate the engine induced vibrations are roughly estimated to be 25N.

The arrangement of the actuator, the coupling element and the viscous damper are designed to ensure that the necessary force is provided over a wide frequency range. In addition to this, the coupling element must be designed to protect the actuator against overload, and to ensure the correct total stiffness.

To ensure that these requirements are met, the active unit is characterized in a stiff frame including a force sensor as shown in Fig. 4.

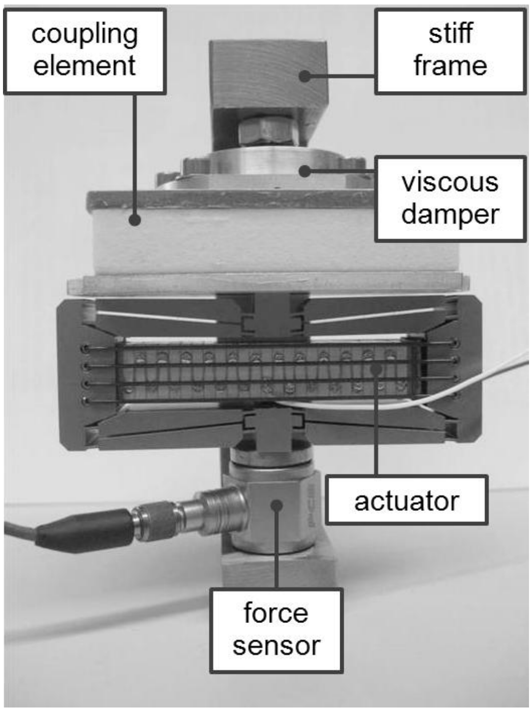

Fig. 4: *Measurement set-up of the active assembly*

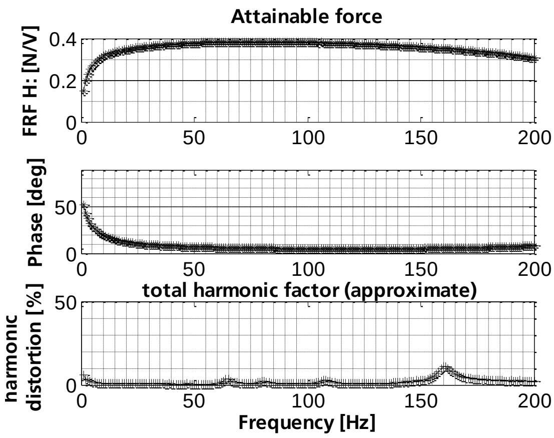

Fig. 5: *Measurement results of the active assembly*

As the measurement results in Fig. 5 show, no force is transmitted through the active unit for the static case because of the viscous damper. With rising frequency the force transmission increases and remains nearly constant from approximately 30Hz up to 150Hz. Then it starts to decrease due to a root in the mechanical system at higher frequencies. The measurements show that the required forces of around 25N are reached at a supply voltage of 50 - 70V.

Another decisive criterion to evaluate the suitability of the active unit is the linearity. Here the linearity is assessed by the harmonic distortion factor that stays level at 3% with the exception of a maximum at 170Hz with 12%. Although the measurements shown here were captured with 10V supply voltage, measurements have shown that the linearity does not deteriorate significantly by the use of voltages up to 50V.

4. Control Algorithm for Vibration Cancellation

Concerning the reduction of periodic engine induced vibrations, feedforward control concepts are commonly used for those systems, implemented e.g. by a narrowband FxLMS algorithm [3, 7].

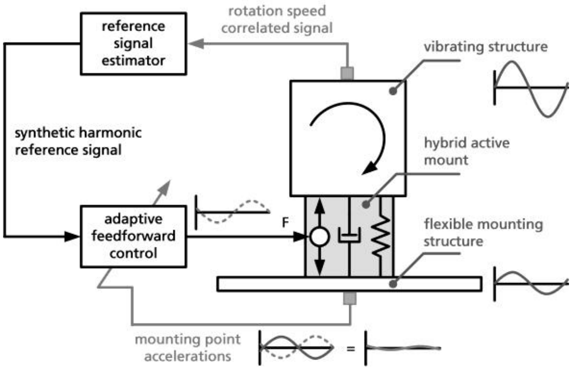

Fig. 6: *Adaptive narrowband feedforward control scenario*

As shown in Fig. 6 a vibrating structure is mounted on a flexible resonant structure by means of a hybrid active mount. By measuring the mounting point accelerations as well as the rotation speed it is possible to induce harmonic forces in such a way that the measured acceleration is minimized. Fig. 7 shows the applied narrowband FxLMS algorithm [7] which is used to minimize the error signal e_k which is composed of the disturance signal d_k and the output of the control path $S(z)$.

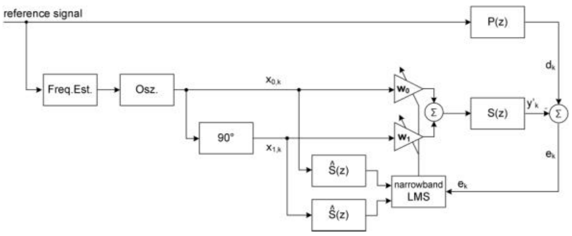

Fig. 7: *Narrowband FxLMS algorithm*

The adaptive filter weights w_i are readjusted in every single sampling step to ensure an optimal performance as rotation speed or ambient conditions change.

However, the attainable vibration reduction mostly depends on precise frequency estimation [8] and as former investigations have shown, the rotational speed information provided by the motor control unit via CAN bus does not represent optimal frequency estimation for the control system. For that reason an optical encoder and a 16MHz sampling frequency measurement unit with RS232 bus connection is used to obtain suitable frequency estimation.

The control algorithm is implemented on a rapid control prototyping system but has also successfully been implemented on embedded control platforms.

5. In-vehicle Measurement

5.1 Measurement Set-up

Performance measurements of the engine mount were taken in the Adaptive Structures Test Facility (ASF) at the Fraunhofer Institute LBF. The vehicle is mounted on a stationary rig; the driven wheels attached to a speed controlled asynchronous motor as shown in Fig. 8.

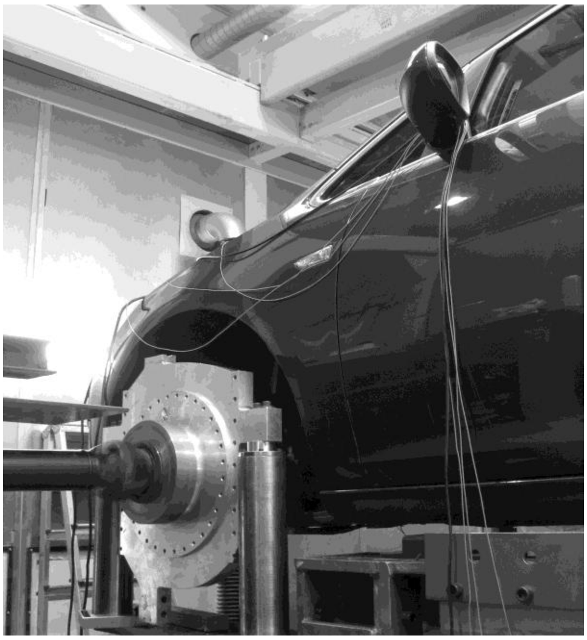

Fig. 8: *Measurement set-up of the in-vehicle tests*

The measurement schedule consists of acceleration and sound pressure measurements during run-ups in second gear and an accelerator pedal position of 30% of maximum position. In this paper only the results of the acceleration measurements are presented. The run-ups were performed within 60 seconds from 1500 to 4500 rpm. During these measurements the adaptive controller reduces the mounting point accelerations by means of the previously introduced control concept. The engine speed is set using the asynchronous motors, thereby determining the engine load during the run-up. For comparison, measurements were taken for the serial mount (Fig. 9), for the uncontrolled active mount and the controlled active mount (Fig. 10) as well.

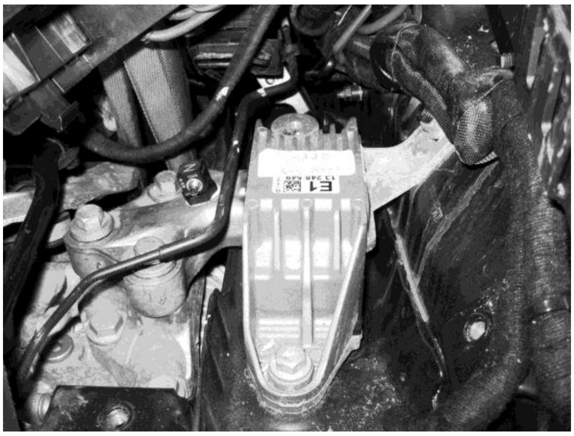

Fig. 9: *Serial mount*

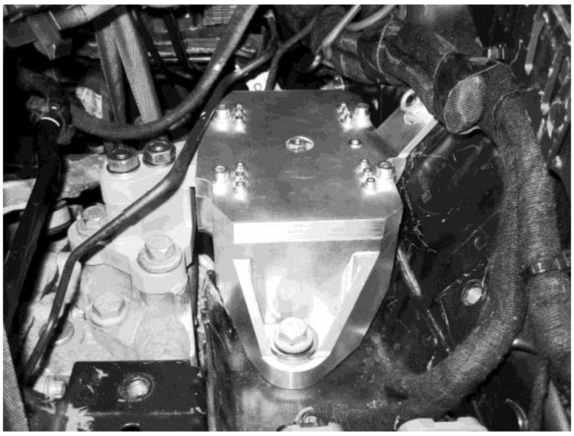

Fig. 10: *Attached active mount*

Accelerations were measured at the chassis side of the engine mount; the sound pressure was measured in the passenger cabin at driver's ear position. The acceleration and sound pressure signals were sampled at 2048 Hz and collected by a LMS data acquisition system.

5.2 Measurement Results

Because of the fact that the second engine order represents the dominating harmonic distortion, the second engine order disturbance at the mounting point as a function of the current rotation speed was chosen to evaluate the controller performance as well as the performance of the serial engine mount.

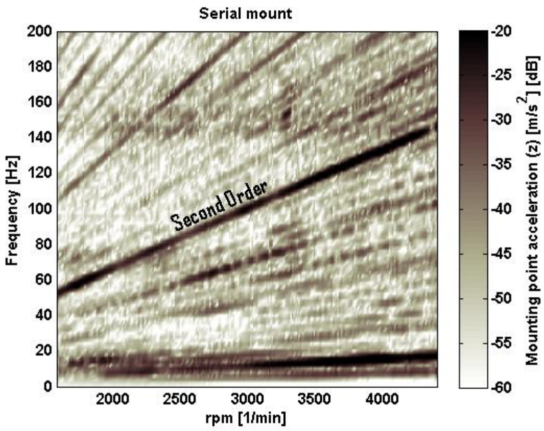

Fig. 11: *Serial mount - mounting point acceleration*

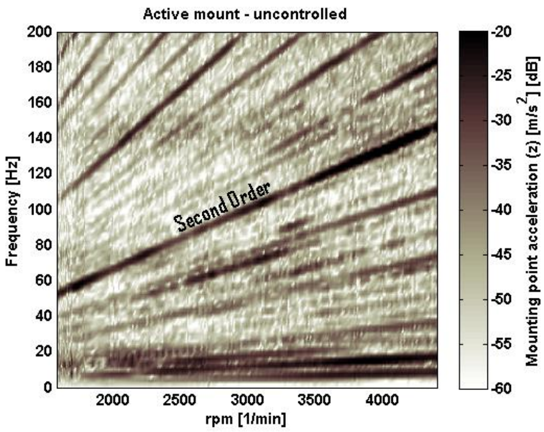

Fig. 12: *Active mount (uncontrolled) – mounting point acceleration*

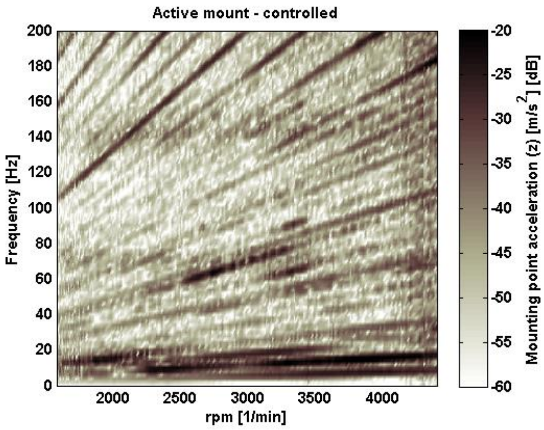

Fig. 13: *Active mount (controlled) – mounting point acceleration*

Fig. 11 to Fig. 13 show the Campbell diagrams of the acceleration at the mounting point during a complete run-up for the serial mount, the uncontrolled active engine mount and the controlled active engine mount. Because the active mount is designed to have a comparable dynamic stiffness to the serial mount, there is no significant deviation between the two run-ups shown in Fig. 11 and Fig. 12.

The Campbell diagram in Fig. 13 shows the measurement results for the controlled active engine mount. Compared to the serial mount and the uncontrolled active mount, it's apparent that the second engine order is significantly reduced.

In Fig. 14 the second order cuts of the run-ups are shown. A significant reduction of the acceleration amplitude up to 20dB is observed considering the serial mount and controlled active engine mount, the improvement is highlighted gray in the Figure. By comparison of the serial engine mount with the uncontrolled active engine mount the difference of the amplitude is mostly in between -5 and 5dB. This indicates that the passive behavior of the serial mount in the contemplated z-direction is comparable to the behavior of the developed active mount in passive condition.

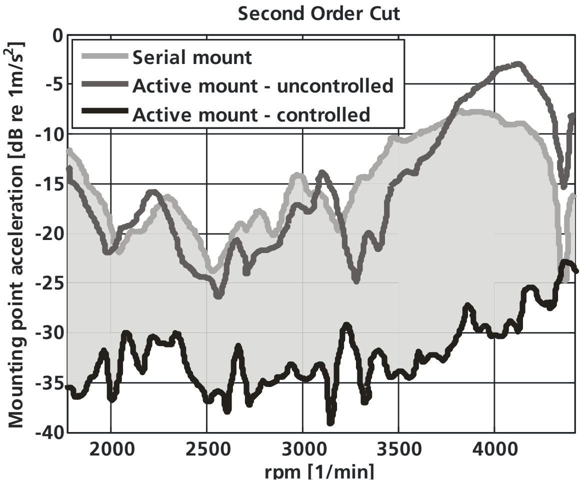

Fig. 14: *Second engine order acceleration during a run-up and an accelerator pedal position of 30%*

6. Conclusion

Within the framework of the EU project "HiperAct“ (High Performance Piezoelectric Actuators; CP-IP 212394) completed in 2012, an active engine mount was developed at Fraunhofer LBF. To overcome the disadvantages of existing active mounts, this mount incorporates a novel topology featuring the decoupling of static forces. Compared to active mounts without decoupling, the actuator only bears the significantly smaller dynamic loads. Hence smaller actuators with stroke amplification mechanisms can be used, which results in reduced power requirements. Additionally, the topology ensures an additional protection against mechanical overload of the actuator.

Tests under close-to-reality conditions have shown a significant reduction of the accelerations resulting from the dominating second engine order at the engine support by as much as 20dB, while the sound pressure level was reduced by as much as 10dB.

The authors expect that the proposed topology could be a major step forward in the development of resilient active engine mounts. Ideally, it is possible to replace or enhance existing passive solutions with active technologies without any need for other changes on the system.

7. Outlook

The ongoing work examines different control strategies by using the sound pressure or a combination of car body acceleration and sound pressure as controller error signal. In case of miniaturization requirements the control algorithm can be implemented on embedded control platforms with reduced calculation complexity [9].

Furthermore, the development of stroke amplification mechanisms and tailored piezo amplifier solutions for optimal efficiency is in progress.

8. Acknowledgements

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