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Active Control at Lord Corporation – A Reality

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ABSTRACT

This paper discusses various applications of active noise and vibration control that Lord Corporation has developed over the years and introduced to various markets and industries. For each application, the problem to be resolved is discussed in both its technical and, to some extent, economic context. The particular methodology with which active control is implemented is also discussed. Finally, product characteristics and achieved performance are described. The various examples discussed here span the aerospace, machining and precision instrument industries. They add up to a demonstration that for certain market niches, active control has now become a reality.

INTRODUCTION

Three hundred years after Huyghens formalized the principle of wave interference and 70 years after the invention of active noise control by Lueg in Germany, asking whether active control is a myth or reality seems like a valid question. The initially supposed pervasiveness of the technology and its predicted appearance in all areas of one's everyday life have failed to materialize. Many hi-tech startups have gone through boom and bust trying to bring the technology to market.

With the exception of a still limited market for active noise reduction headsets, applications of active control in consumer products are inexistent. There are, however, several industrial applications where active control has found a home. Lord Corporation has been able to develop and commercialize many of them. This paper presents a selection of active control products developed and commercialized by Lord for the aerospace, machining, and precision instrument industries, respectively. Each application will be described through a problem statement, with details of the active control strategy and the presentation of the related active control product's typical performance.

The applications discussed in this paper are evidence that while active control is not as widely used as was predicted in the early 80s, it has become a reality to many industries where it does bring significant value.

ACTIVE NOISE CONTROL THROUGH TARGETED ACTIVE STRUCTURAL CONTROL

Problem Statement

The golden rule for any noise or vibration control strategy is to tackle the disturbance at its source [Rossetti 95]. A practical situation where this is applicable in a very effective manner is the case of jet aircraft whose engines are mounted directly to the fuselage, typically toward the rear of the aircraft. The Douglass DC-9 (Fig. 1), McDonnell Douglass MD80 (Fig. 2), and most regional and business jets are examples of this configuration.

Fig. 1: McDonnell Douglas MD80 (jetliner)

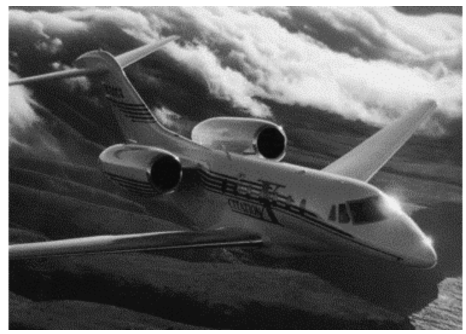


Fig. 2: Cessna Citation X (business jet)

Indeed, with such an engine-fuselage interface, the engine-induced noise is caused for the most part by the engine vibration at the N1 (fan) and N2 (spool) frequencies. This vibration is transmitted to the fuselage through the engine mounting system (Fig. 3). The fuselage in turn vibrates, creating noise inside the cabin.

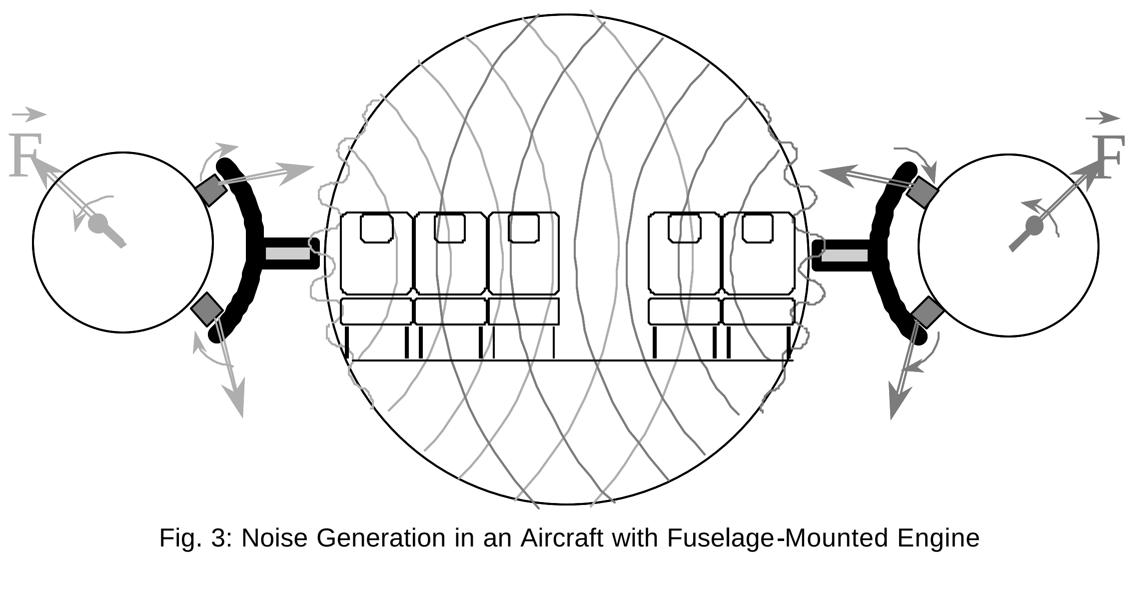


Fig. 3: Noise Generation in an Aircraft with Fuselage-Mounted Engine

This noise can sometimes be extremely annoying (up to 106dBC), and Douglass Aircraft Corporation had to install Tuned Vibration Absorbers on the engine yoke to try and reduce noise at least at the cruise frequencies (Fig. 4).

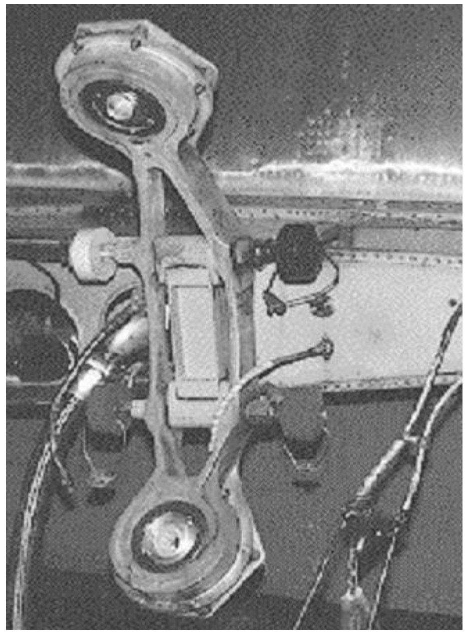


Fig. 4: DC9 Engine Yoke (Engine Removed) with 4 TVAs

From an “economic” standpoint, the presence of loud tones in the cabin noise spectrum is objectionable to the flying passenger. Many airlines are eager to control such components of the noise in order to avoid taking the passenger through a very inconsistent “experience” on a multi-leg route where he or she would fly on an aircraft with fuselage-mounted engines just after having flown on an aircraft with wing-mounted engines. In addition, when an airline refurbishes the interior of older aircraft (like the DC9), it is concerned that the objectionable noise quality of the aircraft will undermine the effort and money put into improving the cabin appearance. For these reasons, several airlines have opted to use Lord’s active control systems.

Active Control Strategy

In such a situation, the implementation of active control is based on a simple strategy: prevent the vibration from propagating from the engine to the aircraft fuselage. This is the most effective way of controlling noise at the typical frequency values of N1 and N2: 120 Hz and 170Hz respectively. Indeed, once disturbance at such frequencies enters the fuselage structure and cabin air volume the size of a DC9 or even a business jet, the complexity of the field (acoustic or vibratory) becomes extremely high, and controlling it becomes weight and cost prohibitive. While it has been demonstrated that strategies tackling the vibratory field in the fuselage structure [Mathur 95] or the acoustic field in the cabin [Finck 93] can be effective, they are both impractical and inefficient.

Active Control Product Description and Performance

The active systems developed by Lord in the late ‘90s for these applications are described in the block diagram in Fig. 5. They utilize eletromechanical actuators located directly on the vibration path from the engine to the structure (Fig. 6). A central controller drives these actuators so that they input counter-acting forces intended to cancel the vibration before it reaches the structure. Microphones are located in the cabin to provide the controller with a measure of the noise reduction performance that is used to optimize the actuator forces in real time. The central computer also uses reference signals for frequency and phase locking purposes. Engine vibration signals or tachometer signals have been used to provide this reference.

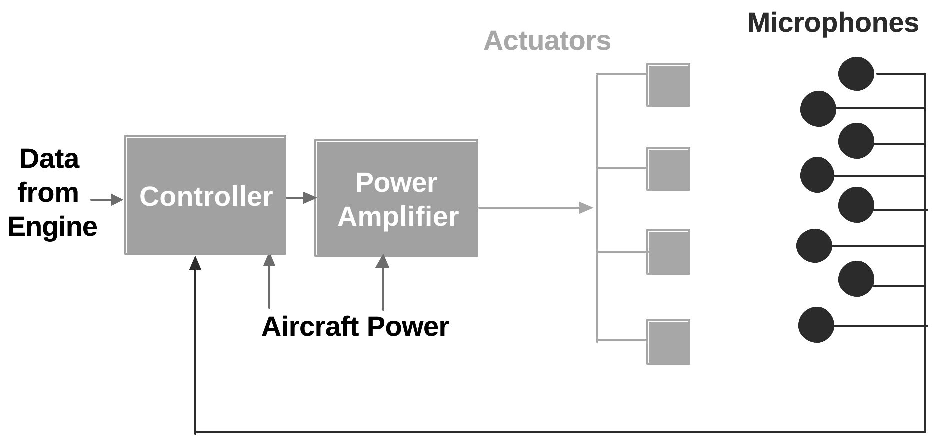


Fig. 5 Active System Block Diagram

While the algorithm is based on the now-classic time-domain filtered-U algorithm, several implementation issues need to be carefully looked at and addressed to ensure reliable long-term operation. These include the potential long-term divergence of the algorithm due to ill-conditioned transfer function matrices, the monitoring of each system component's health, and the actuator life management. The latter issue is key to maintaining system reliability. Actuator life management relies on guaranteeing that the actuator is not provided power beyond its thermal limit and that its stroke is always below the limit that ensures infinite life.

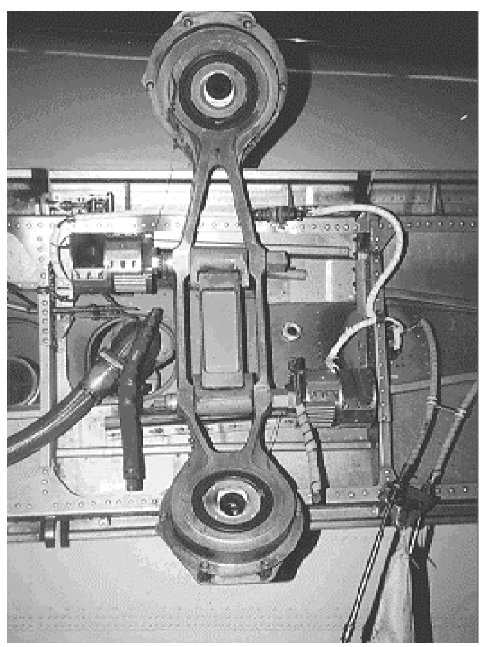


Fig. 6: DC9 Engine Yoke Showing Active System's Actuators

Products based on this principle have been developed and certified by Lord Corporation for the Cessna Citation X and the McDonnell Douglass DC9/MD80 series. They have been in production since 1995, and 1998 respectively. The main components for the DC9 system are shown in Fig. 7.

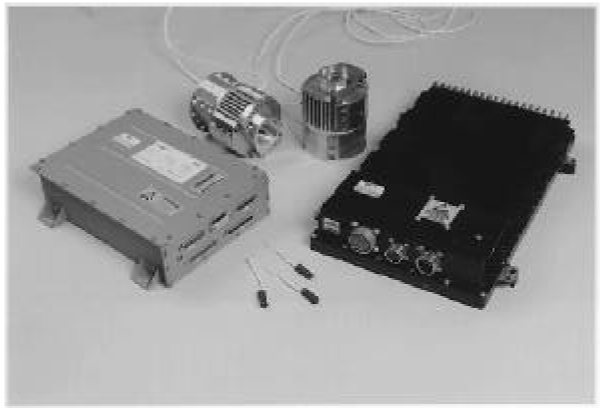


Fig. 7: Main Components of the DC9 ANC system

The cabin noise maps for the Cessna Citation X (Fig. 8) and the DC9 (Fig. 9) show how effective controlling the noise at its source (engine vibration in this case) can be. The noise reduction is effectively global. In both cases, the data was measured at head height in the cabin while the system's microphones were located in the cabin headliner in the case of the Citation X, and above the aisle ceiling panels in the DC9.

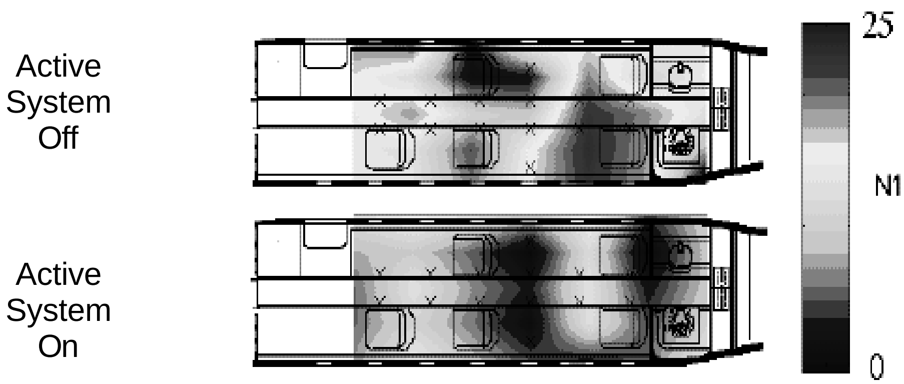


Fig. 8: Noise Levels in Cessna Citation X at N1 (dB arb. Scale)

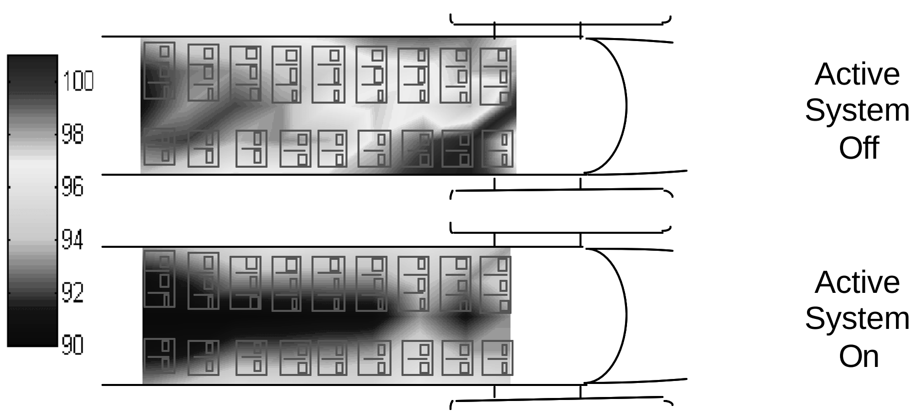


Fig. 9: Noise Levels in DC9 (45 aft seats) in dBC (Attenuation is up to 8dBC)

ACTIVE CONTROL OF STRUCTURAL MODES IN GANTRY ROBOTS

Problem Statement

Gantry robots are often used in machining operation on large pieces as in the case in the boat hull mold milling machine shown in Fig. 10. These structures are quite slender and are thus prone to significant oscillations at relatively low frequencies. Such vibrations are obviously due to poorly damped modes in the structure. In the case of the milling robot, the milling head oscillations have several undesirable effect: the motion of the milling head is directly imprinted on the machined surface, leading to poor surface quality; the speed at which machining has to be done to prevent excessive vibration is much lower than the gantry robot's capacity.

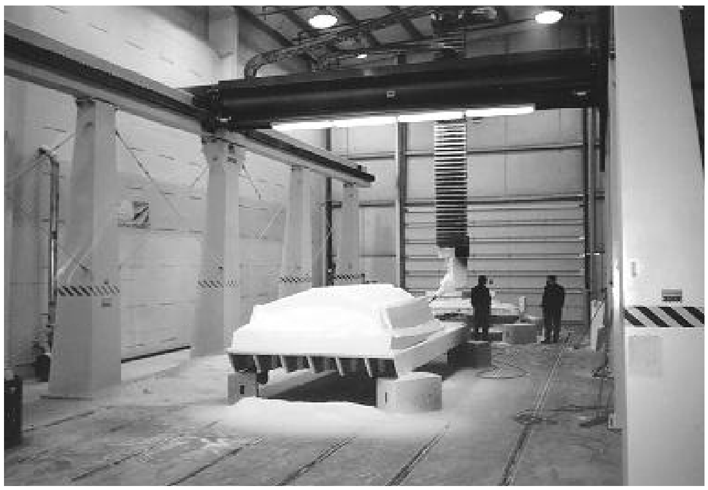


Fig. 10: Gantry Robot Milling a Boat Hull Mold

From an economic standpoint, these effects are directly measurable in monetary terms. Indeed, the poor surface quality makes it necessary to sand the hull molds manually. Further, the throughput of the expensive capital immobilized in the robot is much less than it could be, leading to high amortization costs in each mold.

Active Control Strategy

The dynamic situation is rather simple in this case: a vertical cantilevered beam exhibiting modes of vibration in both horizontal directions. One could imagine that passive damping could be implemented in the beam. However, this is impractical as the largest deflections are seen at the end of the beam where no fixed frame reference can be found to attach the other end of a damper. One could also think of using two Tuned Mass Dampers at the end of the beam to provide damping of the main modes. This would not work, however, as the beam can be retracted or expanded, thus changing the resonance frequency of the mode to be controlled. In addition, the weight of such devices would be prohibitive with respect to the design limits of the structure.

Active control can be advantageously brought to bear in such a situation. Using a pair of dynamic actuators, one can emulate viscous damping at the very tip of the main arm, thereby actively augmenting the modal damping of the modes contributing to the oscillations of the milling head. Whereas in the preceding case one was relying on feedforward control schemes based on vibration reference signals or tach signals, the active system for this situation is based on velocity feedback control, a simple way of emulating natural viscous structural damping.

Active System Description and Performance

A block diagram of the system implementation is provided in Fig. 11. For simplicity only one direction is represented. It is understood that the control system in the other direction in the place orthogonal to the main beam has exactly the same architecture. For the most part, the control scheme relies on velocity feedback into the actuators at the end of the beam so that the active system behaves as a virtual dashpot between mechanical ground and the beam. Interestingly in this particular situation, the control laws are slightly more complicated than pure velocity feedback since the control system has to account for the intrinsic dynamics of the actuators on the beam. Also, one should note that the use of two accelerometers per direction is needed to avoid mistaking acceleration of the rigid gantry body in its normal displacements for beam vibration.

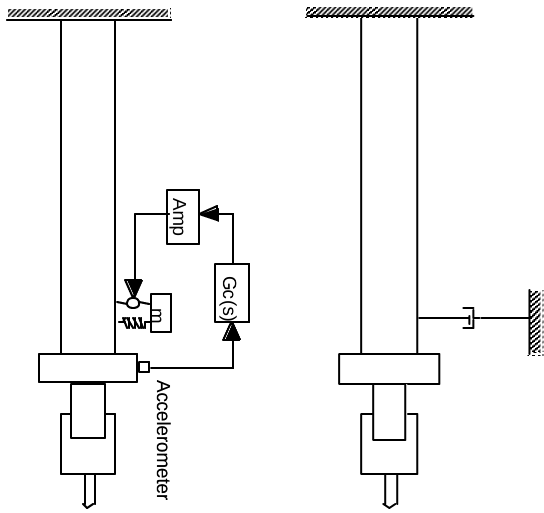


Fig. 11: Block Diagram of the Active System (left) – Mechanical Equivalent (right)

A picture of an early prototype implementation (Fig. 12) shows the actuators at the end of the main vertical beam. The production version of this system has the actuators fully integrated within the structure. The system uses off-the-shelf electronics for the controller and power amplifiers. The actuators provide 300N of force in the 5 to 10 Hz range (range of interest for the operational envelope of the target robot).

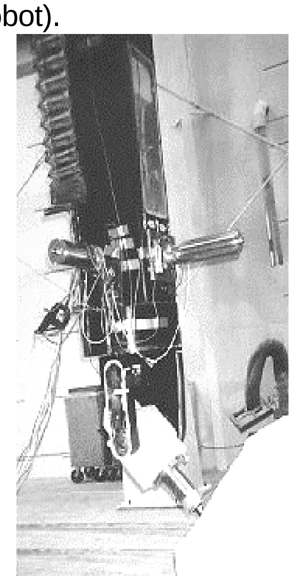


Fig. 12: Actuators on the Main Vertical Beam

The improvement provided by such a system is significant. The spectral plots in Fig. 13 show the impulse response of the structure in one particular direction before, and after the active system was implemented. As can be seen, the quality of the machined surface is greatly influenced by the resonant characteristics of the structure. The surface machined without active damping would need to be hand finished (by sanding) whereas the surface with active damping would go through quality control with flying colors. One other fundamental benefit from the higher structural damping resulting from the use of the active system is that the traveling speed of the tool could be increased dramatically, resulting in a 66% increase in productivity. Overall, the implementation of the active system paid for itself in less than 6 months of production.

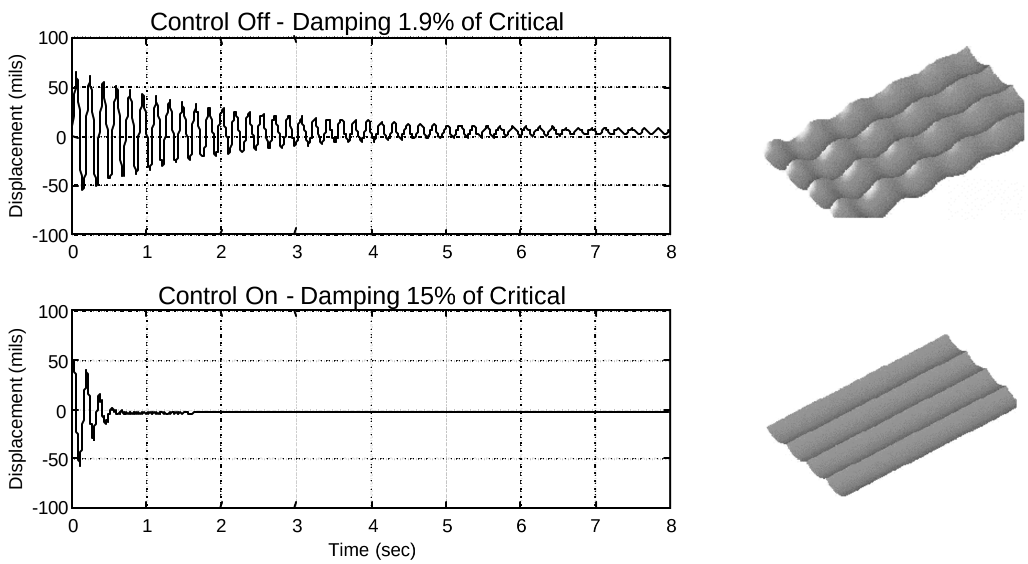


Fig. 13: Impulse Response of the Beam (left) – Machined Surface Quality (right)

ACTIVE DAMPING IMPROVEMENT IN PRECISION ISOLATION TABLES

<u>Problem Statement</u>

One of the classical dilemmas in the design of an isolation system is related to the amount of damping to be used in the isolation system. On one hand, damping is beneficial to prevent the isolation system from transmitting too much force to the isolated structure at its resonance frequency. On the other hand, high damping leads to poorer isolation capabilities above the resonance frequency. One area where this issue is particularly painfully felt is in precision isolation tables. While fully active tables have been on the market of “high-end” isolation tables for a few years, “low-end”, relatively inexpensive tables often rely on self-leveling air suspension systems. These systems are designed to be extremely soft, with resonance frequencies of only a few Hertz. They often exhibit a large amount of vibration amplification at their resonance frequency, as damping in the isolation system is kept at a minimum to ensure appropriate isolation at higher frequencies. This low-frequency amplification limits the range of delicate tasks that can be achieved on such a table.

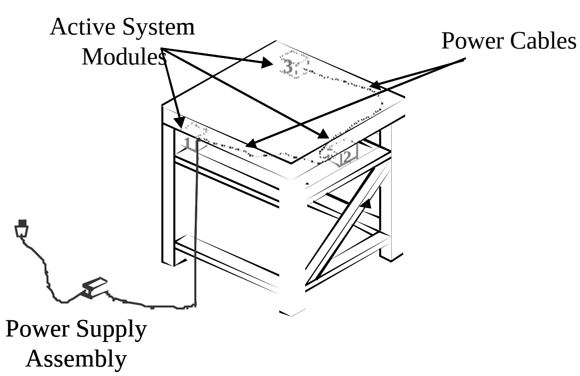
Fig. 14: Passive Isolation Table

Active Control Strategy

The focus of an active solution for such a situation is to implement active damping in a certain frequency range without degrading isolation capabilities above that frequency range. This system requirement needs to be accounted for along with the unique constraints of the isolation table. A noteworthy one among these is the extremely low vibration levels that must be reduced. An active system has to rely on a measurement through a sensor. Extremely low-level vibration can be measured with high-sensitivity sensors. The cost of such sensors however is prohibitive with respect to what the market would accept to pay for a low-frequency improvement in addition to the table's price.

System Description and Performance

The active system developed for this situation is an inexpensive, simple, modular system that fits between the tabletop and the supporting frame (see bloc Fig. 15).



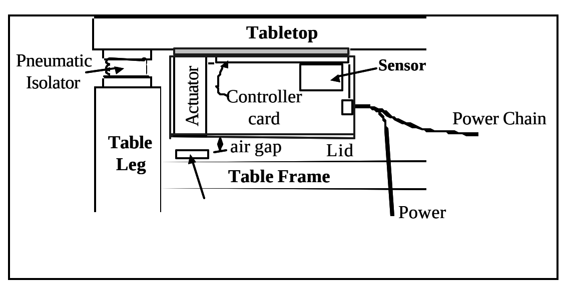


Fig. 15: System Installation (left: general view; right: close-up on one module)

An important feature of the active system implementation is that it is completely contact-free, i.e., it avoids any contact between the tabletop and the legs/frame assembly. This has the obvious goal of avoiding vibration flanking paths that would undermine the effectiveness of the isolation system. Each module has a complete control loop in it. A sensor provides a signal representative of the table's "micro-vibration" to a controller card that generates dynamic forces between the tabletop and the frame through the action of an electromagnet reacting on a permanent magnet on the frame. The control methodology relies on the so-called “sky-hook damping” scheme, which is based on absolute velocity feedback, whereas a passive damper put between the frame and tabletop would mechanically implement the equivalent of a relative velocity feedback. As the module is completely self contained (see Fig. 16), it must be attached to the table top, which is simply done in practice using a magnetic strip between the module’s bottom and the bottom face of the tabletop.

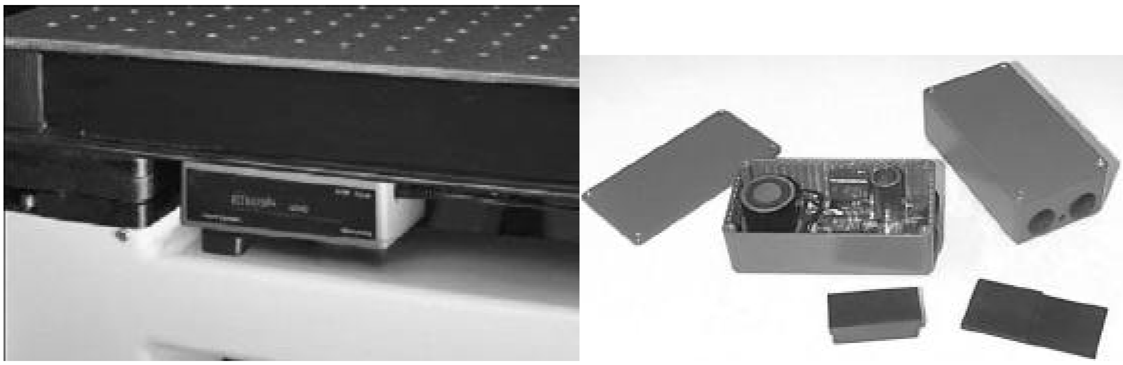


Fig. 16: Precision Isolation Table Active Damping Module
(left: installed underneath tabletop; right: details)

With the active system as described, one can implement “focused active damping” by restricting the damping to the frequency range of interest [Fowler 2000]. One can thus modify the transmissibility curve for that table (transfer function between the floor and the tabletop vibration) as shown in Fig. 17 where the resonant peak that exists without the active system implemented can be completely eliminated with very little, and often no, adverse effect on the higher frequency isolation.

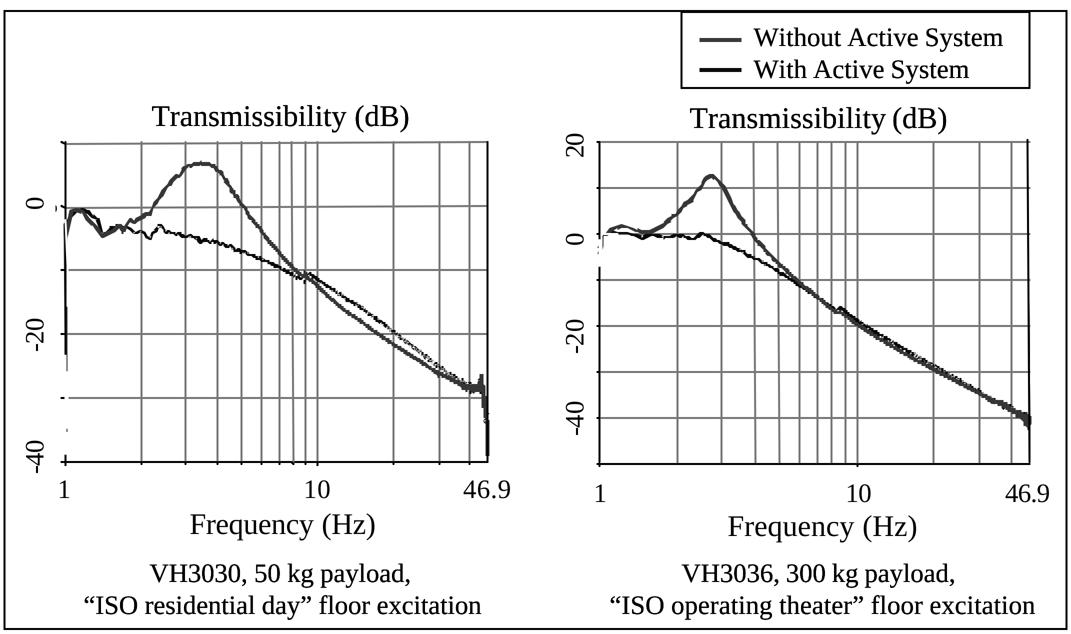


Fig. 17: Standard Transmissibility Measurements on Isolation Table

With such a system, which represents an increased table cost of less than 5%, the low-frequency amplification due to the table's passive isolation system that can reach more than 12dB (factor of 4) is completely eliminated. The isolation table starts reducing vibration from a very low frequency value, whatever the payload or excitation type as seen in Fig. 17.

SUMMARY

Concepts of active noise and vibration control have been around for a long time. The advent of inexpensive digital signal processing technology, propelled by the boom in digital telecommunication made it feasible. Nevertheless, active noise and vibration control has remained limited to certain applications due to both technical and economical limitations.

Through the description of a wide variety of applications, this paper has demonstrated that active control has become a reality in many aerospace and industrial situations. Interestingly (if not surprisingly), all these real-life applications are found in areas where the reduction of noise or vibration offers benefits and value of a scope that is much larger than the reduction of vibration or noise for their own sake.

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