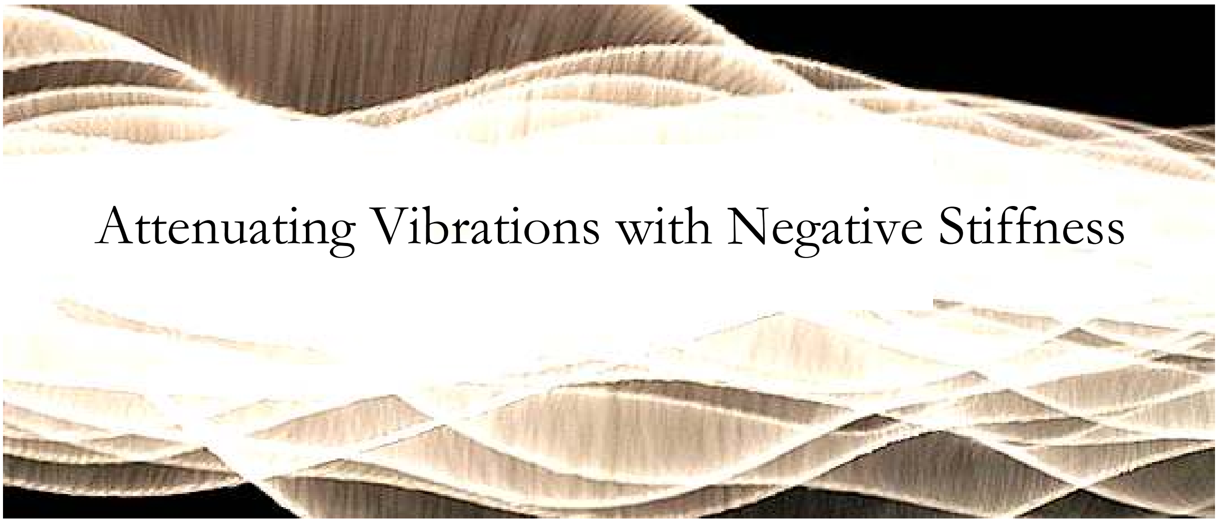
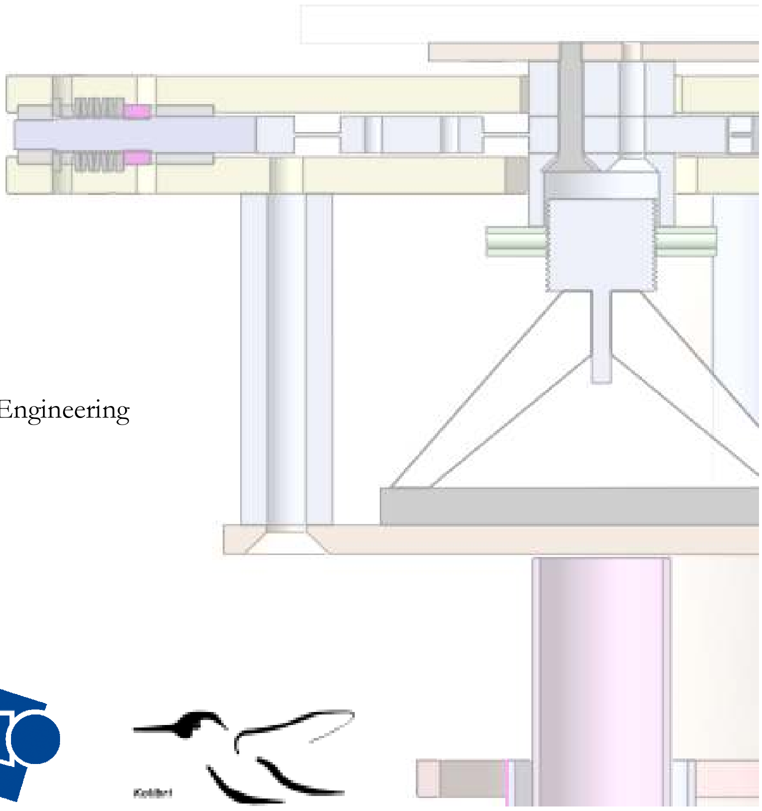
Attenuating Vibrations with Negative Stiffness



Improving active vibration isolation
by reducing suspension stiffness
with a negative stiffness mechanism.

Final Thesis report



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Preface

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Summary

Many high-tech equipment demands a vibration free environment and can only operate at vibration levels much lower than typical ground vibrations. This high level of vibration isolation is difficult to achieve with conventional suspension solutions, especially in the range of low frequencies in the order of 1 Hz.

The goal of the Kolibri project conducted at TNO is to develop an active vibration isolated tabletop system which excellent vibration attenuation properties at low frequencies: 60 dB attenuation at 1 Hz. The performance of the current Kolibri table does not meet this requirement by 30 dB and is to be improved.

The current Kolibri set-up is analyzed to find the performance limiting functional sub-systems. The conclusion of this analyze is that a redesign of the suspension will provide a significant improvement of the vibration isolation performance. Reducing the natural frequency of the suspention from 10 Hz to 0.5 Hz will result in the desired performance improvement of the total Kolibri system. This reduction of natural frequency is to be achieved by reducing the suspension stiffness.

The principle of negative stiffness is chosen to reduce the positive stiffness of the conventional spring. This will potentially result in a compact suspention mechanism at reasonable costs. Several concepts for this mechanism are analyzed. The configuration with three pre-tensioned radial flextures is chosen and completely worked out in detail.

After manufacturing, the negative stiffness mechanism assembled to be tested. The performance of the negative stiffness mechanism is measured; the natural frequency of the mechanism is found to be X Hz, which is higher/lower than the specifications.

Conclusions and recommendations are given for future developments on the Kolibri project, and for improving the negative stiffness mechanism.

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1

Introduction

1.1 Background

Vibration reduction techniques are well established and wide-spread, but over time the demands for vibration attenuation have continuously increased. Conventional vibration isolation systems existing of a linear spring and damper do not provide the performance needed for many current applications.

A solution for this problem is the use of active position control. With the Kolibri project, TNO Industrie en Techniek strives to develop a equipment table with active position control. The objective is to achieve a unequalled vibration isolation from floor vibrations in all 6 degrees of freedom, especially at low frequencies around 1 Hz.

The system specifications for the Kolibri Table are described by TNO in 2007: *Reduce disturbance vibrations of 1 µm at 1 Hz to a 1 nm level. In other words, achieve a 60 dB disturbance reduction at 1 Hz. This applies for all 6 degrees of freedom.*

The 1 nm rms level is chosen because this is needed for the most demanding applications. Note that the reduction itself is the primary goal for the Kolibri project, the absolute value of the residual vibrations is a secondary goal.

1.2 Problem statement

The current set up of the Kolibri table is not performing according to its specifications. An 30 dB attenuation of disturbing vibrations is achieved, which is not enough.

1.3 Research Goal

The performance of the system has to be improved to comply with the system specifications. The goal for this research may be stated as follws: *Improve the vibration isolation system by increasing the vibration attenuation at 1 Hz from 30 dB to 60 db.*

1.4 Research Approach

The Kolibri subsystems that act together towards the desired performance. There are many factors that can be limiting the overall performance of the Kolibri table and it would be undesirable to redesign the complete system, especially in the given time frame.

The first step in this research is to search and find the performance limitations and locate the subsystem involved. In other words, the most limiting aspects or 'bottlenecks' are to be identified.
Each of these limitaing factors are described and analysed. The focus of the subsystem analysis is to access the performance limiting process and accurately predict the effects of changing the key parameters. After gaining insight into these factors methods are developed to eliminate them. The system for which a new design wil potentially result in the best performance improvement is chosen to be further analyzed and redesigned. Recommandations will be given for further improvement of other performance limiting factors that are not chosen to be redesigned.

The next process step in this research is the design of the new subsystem. The goal of this phase is to completely define and realize the new subsystem according to its design specifications. This may involve programming, tuning, manufacturing and assembling new components.

The final steps is to test the new subsystem and compare the measured results with the design specifications. This can be done by directly testing the subsystem in the complete Kolibri set-up, or by seperately testing the subsystem outside Kolibri. In the latter situation, the measured sub-system performance has to be projected onto the total system, predicting the total system performance. This question is to be answered: how has the new design contributed to the overall system performance?

The whole process of this research is shown graphically in figure XXX below. Each process step has its own results which can be used in the following process step.

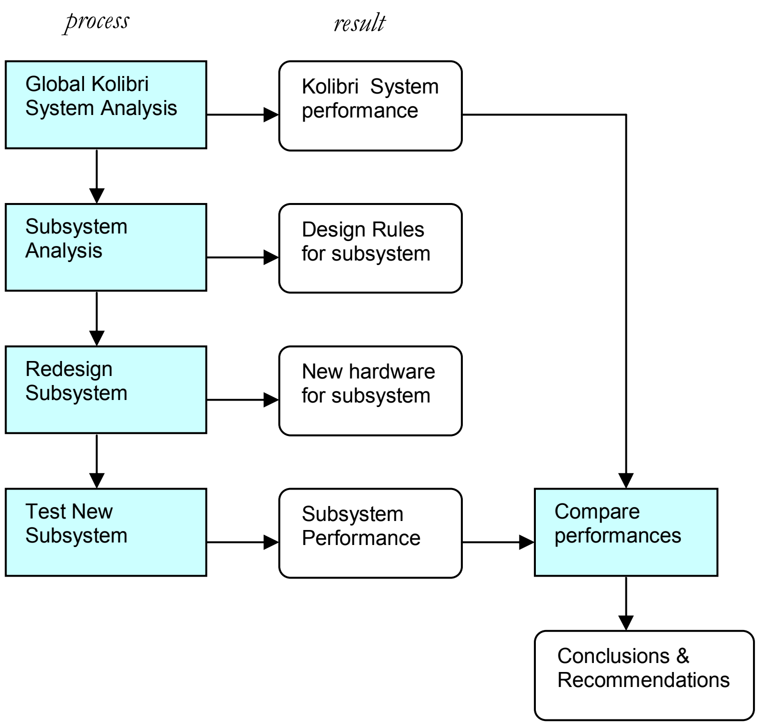


Figure 1-1 Process Chart of the Research Structure

1.5 Report Structure

The report structure roughly follows the same scheme as the research approach. Chapter two consists of background information on vibration isolation technology. In this chapter information is provided to understand the motivation and goals of the Kolibri project.

The analyzis of the total Kolibri system is described in chapter three, ending with a conclusion of which subsystem will be alalyzed in detail and redesigned. Chapter four describes the process of development of the chosen subsystem, resulting in a complete redesign. The test results of the new design are shown in chapter five.

Wrapping up the report, chapter six contains the conclusions of this research and presents recommandations for future research.

2 Introduction to Vibration Isolation

As we stand on the floor, the ground under our feet may seem motionless, but is in fact moving. Because these motions are mainly periodic and stationairy, they are mostly refferd to as ground vibrations. A wide range of sources produce these vibrations, some are natural phenomena, others are induces by human activity. An every-day example is the passing of a heavy bus through the street, resulting in a tremor through the office floor. These vibrations can be the main error source for may scientific fields of research.

When a totally movement-free environment is desired, a vibration isolation system is needed to eliminate the vibrations. Vibration isolation comes in many flavors, and the final application of the vibration isolation system dictates which type op isolation performance is desired.

It is important to understand there is no such thing as 'the vibration isolation performance'. There are many variables which can be taken into consideration when designing a vibration isolation system. Some examples are shock absorbance capability (important for resistance against large impacts), low accelerations (when inertia forces are to be minimized) and position accuracy (in optical applications).

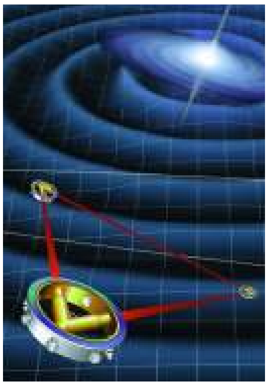
2.1 Demand for vibration free environments

The demands for vibration isolation have increased in time. The process towards smaller tolerances, less disturbances and smaller physical scales in many scientific and commercial applications forces have resulted in a strong demand for vibration free environments.

An less trivial process is pushing the demands of vibration isolation: the sources of vibrations have become stronger and more ambivalent. The quantity of traffic, building sites, aviation and people has increased during the last several decades.

Some examples of high end equipment demanding a low level of disturbing vibrations are:

- Scanning Electron Microscopy (SEM) [REFS]
- Optical microscopy
- Lithography Stage Positioning
- Laser Interferometry
- Microgravity in gravity-free environments
- MRI scanning position control
- Mass spectrometry
- (Optical) Telescopes



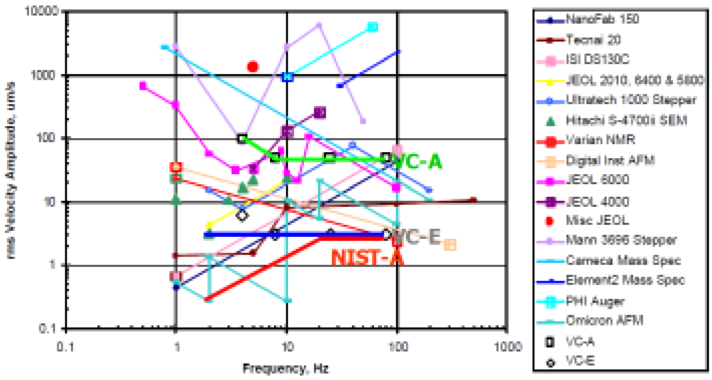


Figure 2-1 An artists impression of Laser Inter-ferometry Space Antenna (LISA)

Figure 2-2 Specified maximum vibration levels for several high end systems

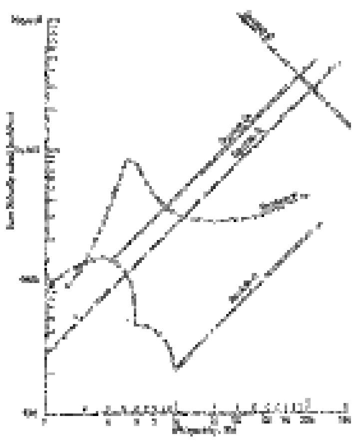
The maximal allowable vibration levels for some high end equipment are plotted in figure XXXX. The explanation of the used units will follow briefly in chapter XXXX, *Classification of vibration levels*. The typical level of ground vibrations in a office are in the order of 500 µm/s rms, and facilities with other running equipment have even higher levels.

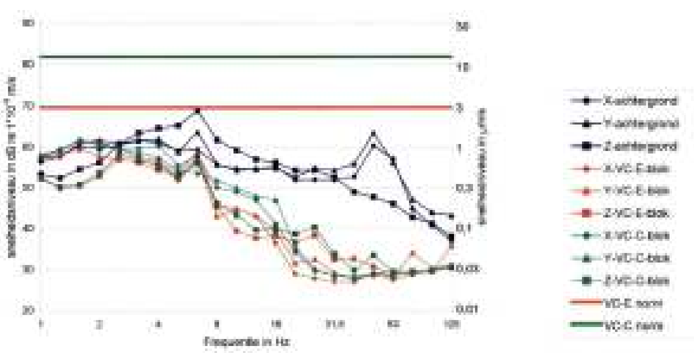
The conclusion is that most equipment listed in figure XXX, the vibration criteria is not met without additional vibration isolation. Different levels of vibration attenuation ask for different approaches in suspension design.

2.2 Classification of vibration levels

There are several quantities that can be used to specify the level of vibrations. Some commonly used quantities are position, velocity, acceleration and power. These quantities can be specified for one frequency, or for a frequency range.

Equipment and machinery sensitive to vibrations are designed with certain vibration specifications; the level of allowable disturbing vibrations. For some time, every manufacturer employed their own vibration specifications that seemed most convenient. Needless to say this practice resulted in much confusion. Figure X illustrates this, showing the vibration specification of 5 equipment manufacturers. Note the vertical axis is RMS velocity [micro-inch/second] and the horizontal axis is frequency, both in log scale. The need od a general standard for vibration levels was evident (see Colin **G. Gordon REF 2.1 x1).**





The International Standard Organization (ISO) has established vibration criterion (VC) guidelines focused on the effects of environmental vibrations on the health of people in **buildings [REF ISO 2.1 x3].** Four levels are defined: Workshop, Office, Residential Day and Theatre. An other widely accepted standard is the Bolt Beranek & Newman (BNN) criteria [**Hal Amick REF 2.1 x2, see appendix C**].

Note the tri-axial lay-out, in which horizontal lines represent constant velocities. At frequencies below 8 Hz, the criteria curves follow a line of constant acceleration. In **appendix XXX**, the vibration criteria are shown in a table, with a brief description of the nature of the vibrations at the criteria levels. The successive VC criteria are all one-third-octave apart (1/3 interval between two successive powers of ten).

The most recent addition is to the VC-standards is presented by Hal Amick from Colin Gordon & Associates in 2005. In addition to the existing VC levels, two more levels of vibration free criteria were added: VC-F and VC-G.

Figure XXXXX shows a typical measurement of ground vibrations, represented in a frequency spectrum, together with two VC levels. For a more detailed description of all vibration criteria, see the table in appendix C

2.3 Vibration isolation performance criteria

The position accuracy achieved with any vibration isolation system depends on the level of disturbing vibrations. Therefore it is not convenient to define the performance by a absolute measure. A suitable relative performance measure is the transfer of ground vibrations tot the table H_x, from now on referred to as **the transmissibility**.

$$x_p = x_f \cdot H_x \qquad \text{or} \qquad H_x = \frac{x_p}{x_f} \tag{1}$$

The transfer of disturbing forces (excited directly on the payload) to the payload position, from now on referred to as **the compliance.**

$$x_p = F_p \cdot T_F \qquad \text{or} \qquad H_F = \frac{x_p}{F_p} \tag{2}$$

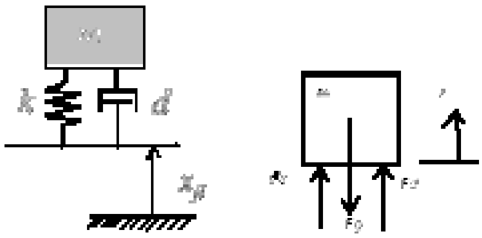
In equation XXXX, H_F is the compliance, or the force sensitivity.

2.4 Passive Vibration Isolation Systems

Passive vibration isolation in one degree of freedom

Passive isolation is capable of isolating a mass from a moving environment, but has a characteristic behavior with inherent advantages and limitations. In order to achieve more insight in the passive behavior, ne of the most fundamental concepts of dynamics is used: the one-degree-of-freedom (1DOF) mass-spring-damper system.

The approach of vibration isolation systems in this research consist of a suspension and a rigid body mass. The mass *m* is called 'payload' because this mass is carried by the suspension. In general, the suspension consists of a spring and a damping component (resp. *k* and *d*), as shown in **Error! Reference source not found.**.



with F the force excited on the spring in Newton, and δ the deflection of the spring in meter, δ' the velocity of the deflection of the spring in Newton-seconds per meter.

Equations of motion

Using Newton's second law, and the free body diagram of figure XXXX, the equation of motion of the mass is constructed. Separating the floor position and table position results in

$$m\ddot{x} + d\dot{x}_p + kx_p = d\dot{x}_f + kx_f \tag{3}$$

The system dynamics are on the left, the forcing function on the right side of the equation.

Frequency response

This equation describes the motion of the mass in the time domain. As stated before, the behavior in the frequency domain is of great interest. The question may be asked ENZ ENZ

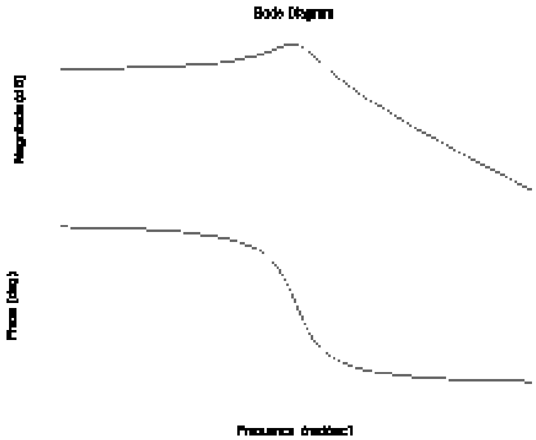
Using the Laplace transform [REF] takes the equation to the frequency domain and results in

$$ms^2x_p + dsx_p + kx_p = dsx_f + kx_f \tag{4}$$

This equation can be rewritten by collecting the terms for floor and payload position, that the position transfer is easily derived

$$H_x = \frac{x_p}{x_f} = \frac{ds + k}{ms^2 + ds + k} \tag{5}$$

The transfer calculated here describes the transfer of the floor position to the payload position, with ω as an independent variable. For any given angular velocity ω, the transfer has a certain value. This value is the result of the ratio between two complex values. This implies that transfer H_x can be expressed as a magnitude $|H_x|$ and a phase φ, the angle between xp ans xf.



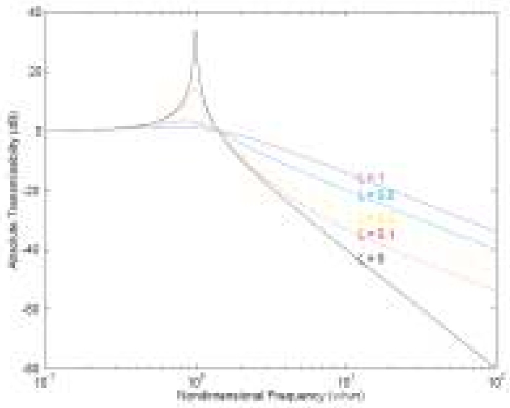


Figure 1 transmissibility of floor vibrations, shown in a magnitude and a phase.
Figure 2 The effect of different levels of damping on the vibration transmissiblity

The natural frequency of the suspended mass ω_n depends on the mass *m* [kg] and the spring stiffness *k* [N/m] as described in the wel-known equation

$$\omega_n = \sqrt{\frac{k}{m}} \tag{6}$$

Roughly speaking, the natural frequency of the suspended payload is the frequency at which the magnitude is at its maximum and the phase is exactly halfway its transition from one asymptote to the other.

Without damping, the magnitude drops down with increasing frequency above the natural frequency (in this example 1 rad/s). This decline is exponential, approaching the asymptote with a slope of -2 in the log/log scale, a decline proportional to $1/\omega^2$. If damping is involved, the decline of the magnitude approaches a slope of -1, proportional to 1/w.

The conclusion is that passive vibration isolation is achieved at all frequencies above the natural frequency of the suspended payload. For a optimal passive vibration isolation at low frequencies, the natural frequency of the passive suspention should be low too. A numerical example illustrates this.

Given a required attenuation of ground vibration swith a factor 1000 at 1 Hz, the natural frequency of the mass/spring system has to be $1/\sqrt{1000}$ Hz. This is also shown in figure XXXX. If damping is involved, the magnitude line will decrease faster with incresing frequency, making a even lower natural frequency needed.

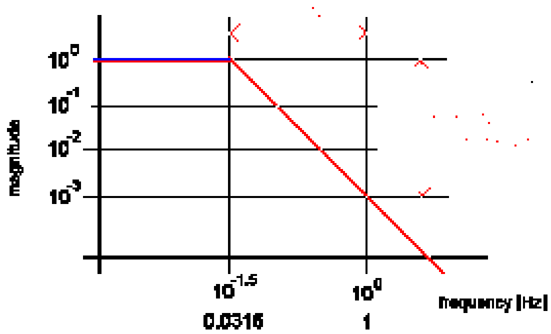


Figure 3 The natural frequency of linear springs needed for a 60 dB vibration attenuation at 1 Hz

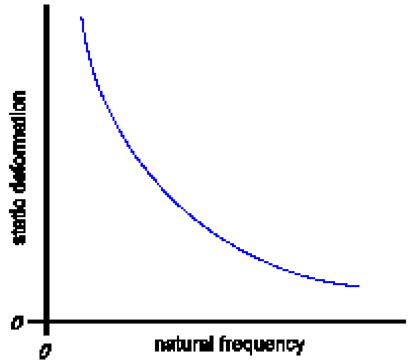


Figure (XXX) The inherent relation between natural frequency and static deflection for linear springs

Low natural frequency with linear springs

According to beforementioned relations, if the payload mass and spring stiffness are not limited and can be chosen freely, every level of vibration isolation can be achieved passively. Many applications use this design freedom and are equipped with large masses and low stiffness springs. [REF XXX].

However, there are practical limitations. A suspended payload may be very heavy, but the springs should be able to bear the high load. Either the spring will experience a large deflection, or the payload is chosen very heavy, combined with stiff springs. Large spring deflections are limited by available physical space, and by the material properties of the spring.

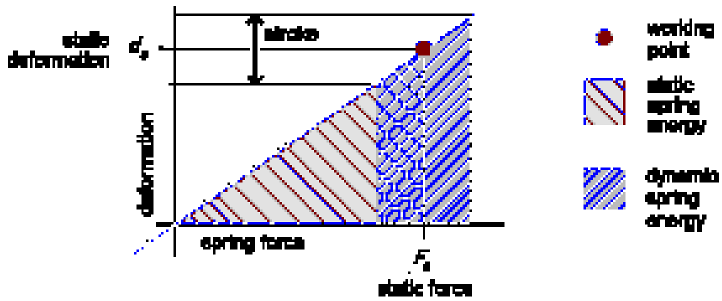


Figure 2-3

Vertical spring deflection and natural frequency

There exists a direct relationship between the deflection of a linear spring carrying a payload mass and its natural frequency together with the same payload mass **[REF 2.2 1].**

The deflection of the payload i:

$$\delta = \frac{F_s}{k} = \frac{mg}{k} \tag{7}$$

with g the gravitational constant with units [m/s]. The natural frequency of the spring-mass system given in equation XXXXX, and can can be evaluated for the spring stiffness k, resulting in

$$k = \omega_n^2 \cdot m \tag{8}$$

substituting equation XXX in equation XXX gives

$$\delta = \frac{g}{\omega_n^2} \tag{9}$$

If designing a system with a given mass and a very low natural frequency is desired, this relation shows a inherent dependency between spring deflection and natural frequency. Note that the deflection is not dependant on the payload mass. The conclusion is that for linear springs, a low natural frequency can only be achieved with a large static spring displacement.

A example illustrates this: when the desired natural frequency is 0.0316 Hz (approximately 0.2 rad/s), the static deflection will be:

$$\delta = \frac{9.81}{0.5^2} = 39.24 \text{ [m]} \tag{10}$$

It is not hard to imagine that hanging the mass from a XXXX meters high construction is far from practical. The height needed will be even larger due to the unloaded length of the spring.

Very long or large springs are not only unconvenient, but have an other inherent limitation: internal resonances. If the system is excited with a force equal to any of the internal resonant frequencies of the suspention, the vibration isolation may not meet specifications or amplify the excitation force. A frequently used solution to overcome the dimensional limitations is the use of a active vibration islolation system.

2.5 Active Vibration Isolation Systems

Vibration isolation has two main functions. The first is carrying the mass of the payload; this function will be called 'gravity compensation'. The second is the control of the position of the payload, or more briefly 'position control'. A passive vibration isolation system

Vibration isolation can be split in two main groups, passive and active vibration isolation. Both groups can be split again by considering the method of gravity compensation. For every one of the four resulting combinations some examples of existing systems are given in Table 1 below. In this report, the expression 'active vibration isolation' is used for active position control and does not refer to the method of gravity compensation.

Table 1

		Gravity compensation	
		Passive	**Active**
Position control	**Passive**	- Conventional Car suspension - Rubber mounts for stereo sets	- Air stream levitation with mechanical springs as horizontal position control.
	Active	- Adaptive optics - Active vibration isolated breadboards	- Magnetic levitation with electronic magnets

Many everyday vibration isolation applications use passive vibration isolation for both gravity compensation and position control. Some examples are conventional car suspensions, rubber mounts for machine frames and dampers mounted on large bridges.

In most applications with active vibration isolation, this is combined with passive vibration isolation because the active vibration isolation is designed to control the position, and a passive suspension is used for the gravity compensation. Some examples are microscope bases, vibration isolated optical tables and high speed train suspension.

Applications with all-active vibration isolation tend to be more advanced because there is no inherent stability otherwise guaranteed by the passive suspension. Examples are the control of the plasma cloud in a fusion tokomak, or magnetic levitation of a desk-top globe (see Figure 2-4) and a wafer stage for the computer chip production.

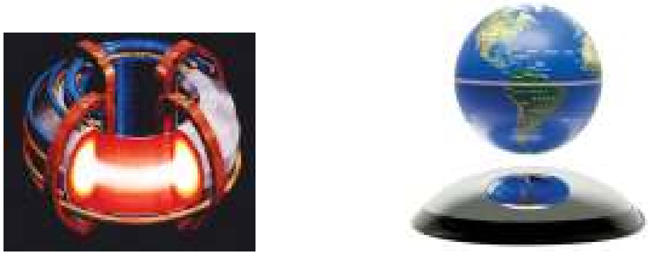


Figure 2-4

Feedback Control Systems

In general, the active position control of a payload is achieved by exciting an actuator force on the payload in such a way that the absolute movement of the payload is minimized. In order to know which force to apply, sensors are needed to measure the current position compared to the desired position, resulting in a position error. This error is fed into a controller; the controller output is used as the actuator input. Together with the suspension and inertia dynamics, the ‘control loop’ is closed. The obtained closed loop system, or ‘feedback control system’, is shown schematically in Figure 2-5.

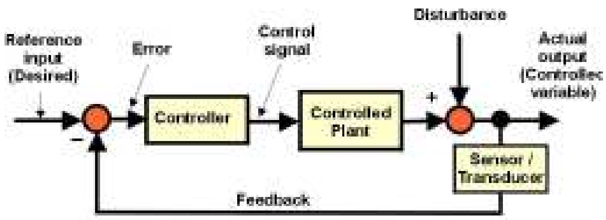
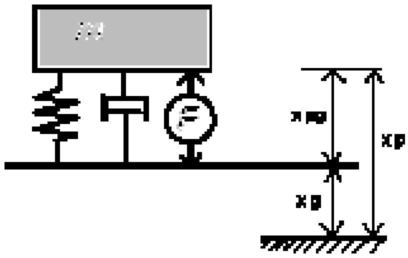


Figure 2-5

Active position control strategies

There are several strategies for both force actuation and sensing method. In most control systems, the actuator is chosen to act parallel to the suspension. In this way, the suspension and actuation can act independently, assuring no extra dynamics are introduced into the control loop.

Ideally, the sensor measures the absolute position of the payload with respect to a external, fixed reference frame. Unfortunately this is not possible, because the only present reference – the floor – is moving itself. A alternative is to measure the position of the table indirectly by measuring relative positions, such as the relative position of the payload with respect to the floor, as shown in figure XXX below.



x pg relative position of the table to the ground
x g absolute position of the ground
x p absolute position of the payload

2.6 The Kolibri Project

Many solutions for low frequent vibration isolation systems include either large masses or large overall dimensions. These approaches may perform very well, but are not very practical in a laboratory or other testing environment.

An optical telescope, for example, is likely to be built in a rural environment and will not be moved or modified within its expected life-span. This relaxes the demands for weight and size. This is in contrast to a flexible testing environment, e.g. a clean room. Here the most favorable configuration of a vibration isolation system has the form factor of a desk or workbench. The advantages are many:

- Relative small dimensions and weight for transportation and repositioning
- Tabletop form factor provides a flat working surface, and space under the working surface for extra equipment.
- The table form factor is compatible with other workbenches and tables.
- An off-the-shelf breadboard can be used as the tabletop, with all the advantages of a breadboard.
- General/universal applications.

These advantages are the focus of the Kolibri project at TNO [ref?]. The goal of the Kolibri project is to design and make a vibration isolated table with excellent vibration reduction in the low-frequency range of 0.01 Hz to 1 Hz.

The table has a passive suspension made of polymer dampers and an actuator system in parallel. The passive dampers and the tabletop mass have a natural frequency of approx. 10 Hz. The sensors have a high-pass characteristic, and have a crossover frequency at approx 3 Hz.

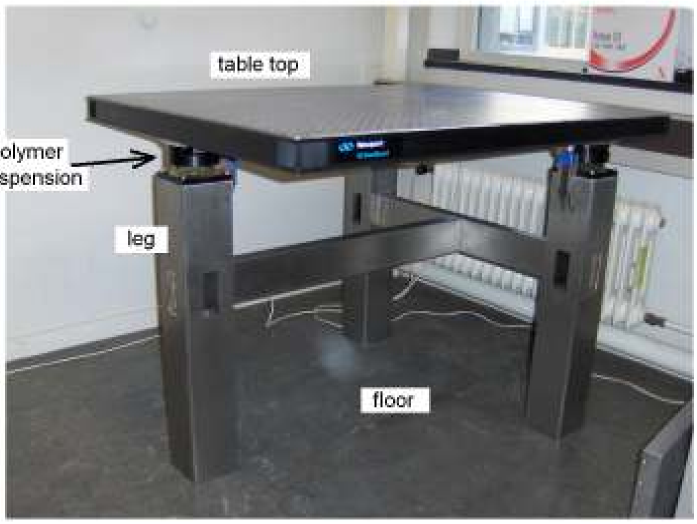


Figure 2-6 table with passive suspension

The current performance of the 1 DOF Kolibri table is shown in figure XXXX. the attenuation of vibrations is present typically between 10 and 0.1 Hz.

Figure 2-7 The Kolibri Project Logo

3 System Analysis of the Kolibri Table

3.1 Introduction

This chapter will describe the search for performance imitating factors of the Kolibri system. In the introduction a brief introduction is given to the system set up, here a more detailed description is given, in order to be able to identify all subsystems and their function

The active control system will get extra attention because this system consists of many components, all with their own characteristic behavior.

In paragraph 6.3, the Kolibri system is divided in functional subsystems. These subsystems are analyzed and assessed to find both performance limiting properties as potential solutions to improve the vibration isolating performance.

3.2 Basic working principals of the Kolibri control

The Kolibri table has a relative simple working principle: the passive suspension is combined with a active position control. What makes the Kolobri project stand out is the method op position error measurement. The relative position of the reference mass with respect to the breadboard is measured with a laser interferometer. Above the natural frequency of the reference mass, the measured value can be considered the absolute position of the breadboard. Physically, the reference mass does not move while the table is vibrating.

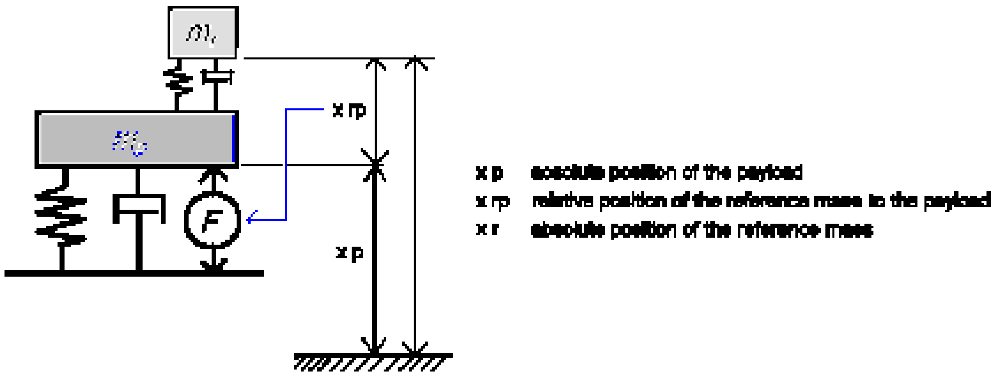


Figure 3-1

Feeding xrp into the controller results in the control scheme showed in figure x below. Compared to the more general control scheme of figure x, this control scheme has some differences. First of all the floor vibrations are a disturbing input between the actuator and the table. The ground vibrations are transferred into forces by the passive suspension. The controller, actuator, table dynamics, and sensor dynamics are modeled separately. The symbols used are C, A, T and Se respectively.

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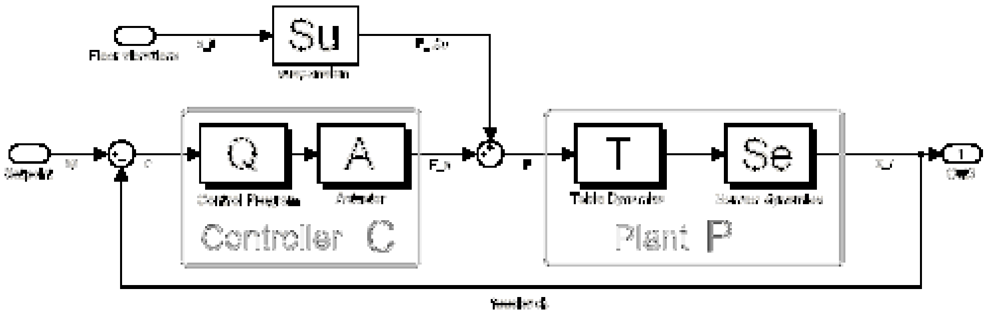

Figure 3-2

C = controller dynamics
A = amplifier transfer
T = table dynamics
Se = sensor transfer
Su = suspension transfer

3.3 System lay out

The Kolibri table has a six degree-of-freedom active position control. There are six translational sensors and six translational actuators. The actuators are voice coils to enable a contactless and precise actuation. A current trough the coil exerts a force on the metal core. The geophone has a inverse working principal: the mass moves trough a coil, producing a current which can be measured.

Figure 3-3

The sensors and actuators are used in pairs and mounted co-linear to the table, as shown in figure XX and XX. This co-located arrangement ensures a better controllability of the table, because there is virtually no mechanical dynamics between the sensor and actuator.

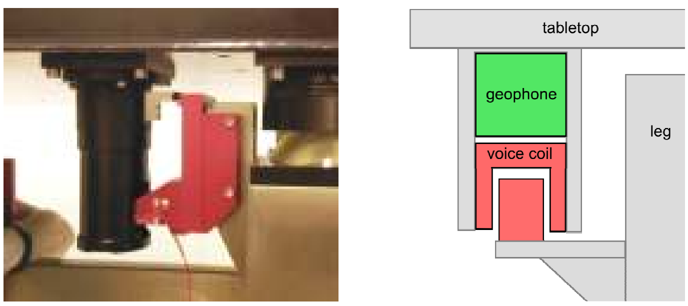

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Figure 3-4

In addition to the overview of the Kolibri table in figure X, a schematic overview of the working principal is given in figure X. One f the six sensor/actuator pairs is shown, together with the table and the computer hardware.

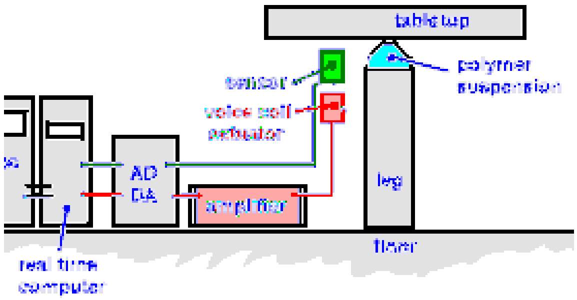


Figure 3-5

The location of all six sensor/actuator pairs is shown in figure XX. This lay-out shows how six translational sensor/actuators are used to control the table in its six degrees of freedom. The six degrees of freedom are defined with respect tot the table center of gravity, assumed to be exactly in the geometrical center of the breadboard. This assumption is not very accurate, because the control hardware mass shifts the table center of mass to a different location.

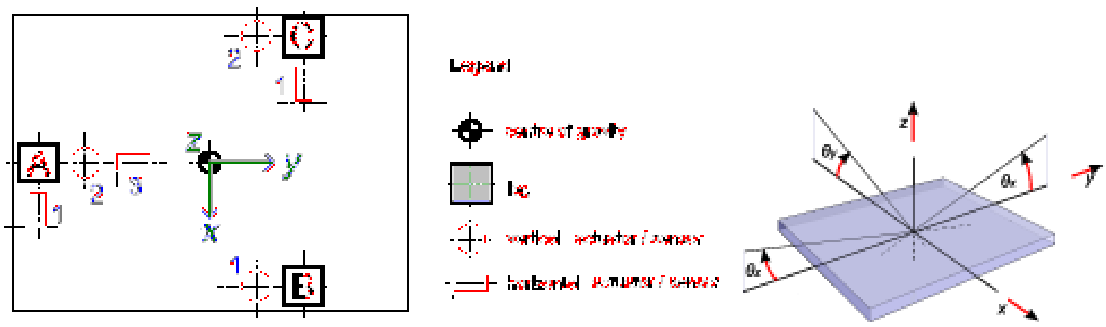


Figure 3-6 Dimensions of the Kolibri components with respect to the COG, in mm.

3.4 Kolibri Table Subsystems

The Kolibri table is divided into functional subsystems which can be analyzed separately. Two groups are distinguished: Mechanical subsystems and subsystems used for the active control. The behavior of each subsystem, and the effect this of this behavior on the total system performance will be investigated.

The choice of subsystems is somewhat arbitrary, but provides a basis for the system analysis.

Table 2: The Kolibri system divides into subsystems

Kolibri Subsystems	
Mechanical	Active control

Table top Legs & Frame Suspension	Sensors Actuators Amplifiers Controller

3.5 Current Performance of the Kolibri table

The performance of the current Kolibri table is tested by Measuring the movement of the table with respect to the ground with a capacitive sensor. During all measurements, an electromagnetic actuator (voice coil) is used to excite the table in the same direction as the capacitive sensor. There are two measurements carried out, with and without the active control turned on (open and closed loop resp.). Figure 3-7 shows the 1 DOF set up for this measurement.

The excited force is measured, together with the relative table position. From these two time-domain signals a transfer is calculated for each frequency in the frequency domain. The results are plotted in Figure 3-7. A reduction of 30 dB at 1 Hz is achieved. The reduction is smaller for the regions below 0.8 Hz and between 10 and 30 Hz.

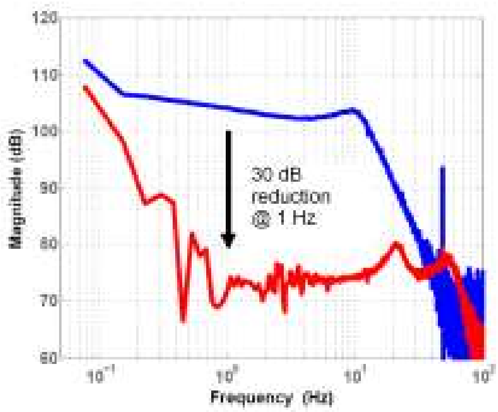
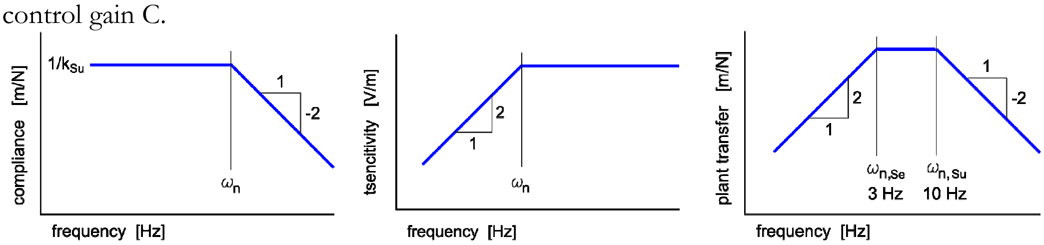


Figure 3-7 Overall Kolibri compliance reduction

3.6 Vibration transmission measurements

Measuring the open loop plant transfer is a very effective method to gain direct insight into the total system behavior. As can be seen in figure XXX, the plant transfer is the transfer between the actuators and the sensors. This transfer, by itself, can not be influenced by the controller and is a result of all the hardware dynamics. The plant transfer dictates the potential proportional action K of the control gain C.



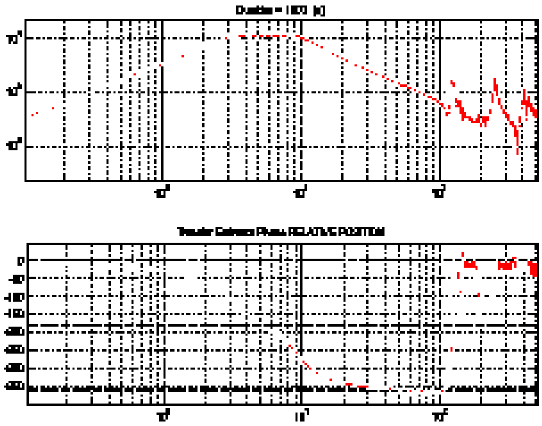
(a) (b) (c)

Figure 3-8 (a) The table compliance, (b) The sensor sensitivity and (c) The total plant transfer

1 DOF plant transfer measurement

Because the table is 6 DOF controlled, there are 6 possible transfers between actuator and sensor. The total number of possible transfers is $6^2 = 36$. The co-located transfer in the vertical direction is chosen, at position A2 in figure X (X_{A2}/F_{A2}).

The plant transfer for this sensor/actuator DOF is measured by exciting the tabletop with noise and measuring the response with the sensor. This is equivalent to system identification. The methodology used is describes in appendix XXX.



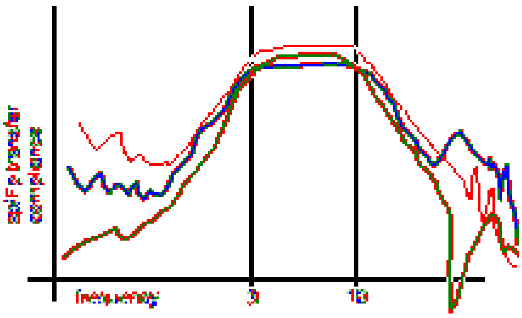


Figure 3-9 **Figure 3-10**

Figure XX shows the plant transfer xp/Fp aka the compliance. The transfer has some distinct features:

- A cross over frequency at 2 Hz
- A cross over frequency at 10 Hz
- fluctuations in magnitude and phase at 140 Hz and above
- resonance/anti-resonance at 250-300 Hz

From a plant transfer pint of view, the control gain is limited by

- The low frequent '1/f' noise
- The high frequent fluctations in magnitude and phase
- The high and low 0 dB crossings, frequency dependant on the gain.

6 DOF

In a similar way as for the 1 DOF plant identification, the 1 DOF input / 6 DOF output is measured. The same voice coil is used, but this time all sensors provide a output signal. The 6 transfers, x_i/F_{A2} with i = A1 up to C1, are plotted in figure XXX.

Some observations are:
- the 3 Hz cross over frequency is present at all transfers. This illustrates that for all transfers, the natural frequency of the sensors are the same.
- all transfers have a cross over frequency around 10 Hz, but they are not exactly at the same frequency. This implies that the natural frequency of the rigid body modes are all in the rande of 10 hz.
- At low frequencies, the transfer of x_{A3}/F_{A2} is exceptionally high. This is probably due to the 'horizontal tilting effect': the horizontal sensor A3 measures a tilting around the x axis because the gravity force on the sensor mass has a component in the measuring direction of the sensor.
- A large depression in the XXX transfer is measured at approx XXX Hz. This #####

3.7 Plant transfer as a measure of control potential

For the Kolibri table, control potential can be defined as *the expected difference between open and closed loop compliance using the estimated maximal proportional control gain.* In this way plant transfers can be directly related to the control potential of the active control system.

In figure XXX, the open loop gain CP of an arbitrary system is shown. Using the control scheme of Figure 3-2, some basic transfers can be derived:

Open loop Plant Compliance $$\left.\frac{x_p}{F_p}\right|_{OL} = P$$

Open loop gain: $$\left.\frac{x_p}{sp}\right|_{OL} = CP$$

Closed loop plant compliance $$\left.\frac{x_p}{F_p}\right|_{OL} = \frac{P}{1+CP}$$

For the regions A and C in figure XX, we have CP >> 1. Because of the large difference in absolute value, we can also choose to neglect the imaginary part of CP, thus assuming CP ≈ |CP|. These assumptions are not valid around de 0 dB-crossings (marked with 'X'). With these assumption and recalling that Su = k_{Su} , the following transfers can be derived:

The closed loop compliance $$\left.\frac{x_p}{F_{Su}}\right|_{CL} = \frac{P}{CP} = \left.\frac{x_p}{F_{Su}}\right|_{OL} \cdot \frac{1}{CP}$$

The closed loop transmissibility $$\left.\frac{x_p}{x_{fl}}\right|_{CL} = \frac{SuP}{CP} = \left.\frac{x_p}{F_{Su}}\right|_{OL} \cdot \frac{1}{CP}.k_{Su}$$

A similar derivation of compliance and transmissibility can be preformed for the regions under the 0 dB line, except with the assumption CP ≈ 0.

Closed loop compliance $$\left.\frac{x_p}{F_{Su}}\right|_{CL} = \frac{P}{1} = \left.\frac{x_p}{F_{Su}}\right|_{OL}$$

Closed loop transmissibility $\left.\frac{x_p}{x_{fl}}\right|_{CL} = \frac{SuP}{1} = \left.\frac{x_p}{F_{Su}}\right|_{OL} \cdot k_{Su}$

As can be seen from equation XXX, the closed loop compliance is equal to the open loop compliance divided by CP. In other words, if CP >> 1, *the area between the CP-curve and the 0 dB line is a measure for the control potential of the complete system.* A complementary conclusion is: *compared to the open loop situation, no improvement in compliance or transmissibility is made if CP ≈ 0.*

If applied to the Kolibri system, these conclusions result in the statement that area C in figure XXX is to be maximized for maximal vibration isolation performance.

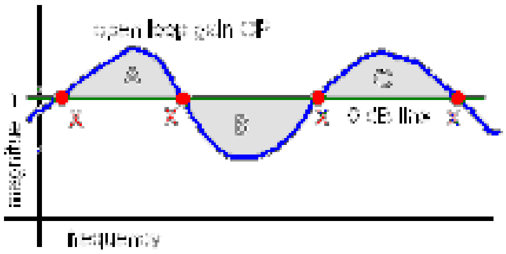


Figure 3-11
Arbitrary open loop gain

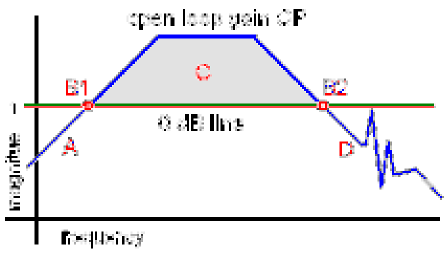


Figure 3-12
Kolibri Open Loop Gain

3.8 Controller design

MOET HIER UBERHAUBT WEL EEN VERHAAL?

3.9 Internal resonance modes

Observations from the bode plot

One of the limiting factors for achieving a higher control gain are the peaks in the magnitude plot of figure XXXX. One of the possible causes is the presence of resonance modes in the breadboard.

Literature for resonance modes

One f the most known scientists in the field of internal resonances is the German Physicist Chladni, who invented a technique to show the various modes of vibration in a mechanical surface. Chladni showed that a infinite number of natural frequencies excist in a thin plate. For a rectangular plate, the modes can be identified by the number of standing waves in fixed directions such as the length, width or a diagonal of the plate.

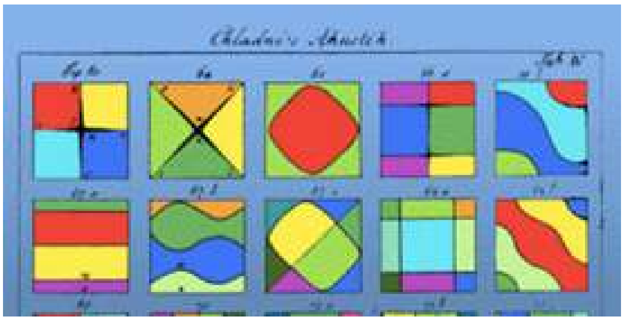


Figure 3-13

Analytical solutions for the natural frequencies of vibration modes are available if the boundary conditions are conveinent

The Breadboard tabletop

The current tabletop is a off-the-shelf breadboard optical table. This kind of breadboards

http://www.newport.com/SG-Series-Scientific-Grade-Breadboards/139753/1033/catalog.aspxModel: M-SG-34-2, SG Series Breadboard
Dimensions:1200 x 900 x 59 mm
Mass: 45 kg
Skin material: Stainless steal

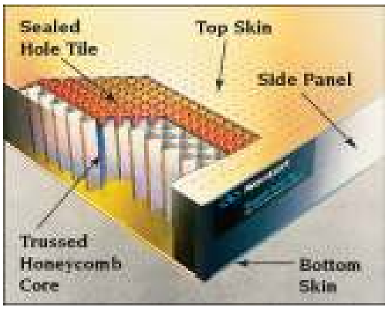
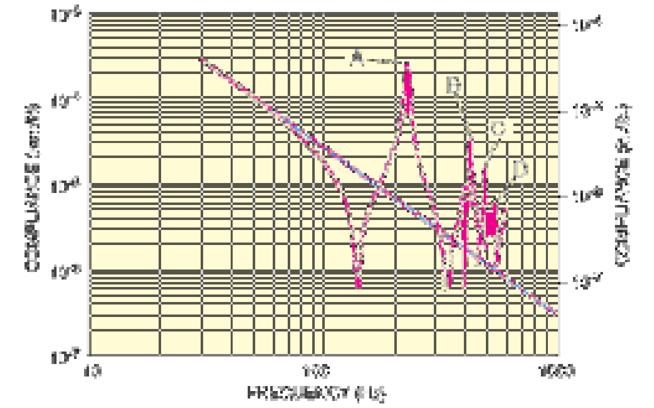
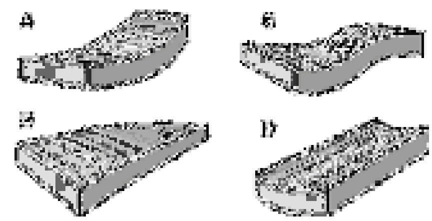


Figure 3-14

Figure 3-15

According to the manufacturer, the internal modes have a fixed sequence and a typical effect on the mass line.





Theoretical modal calculations

Simple supported
First harmonic

$$\omega_1 = \pi^2 \sqrt{\frac{EI}{ml^4}}$$

Both sides clamped
First harmonic

$$\omega_1 = 2.27 \cdot \pi^2 \sqrt{\frac{EI}{ml^4}}$$

The frequency of the modes for thin square plate (or rectangular) should be :

$$f = \sqrt{\left(\frac{n}{w}\right)^2 + \left(\frac{m}{l}\right)^2}$$

which comes from the solution to the wave equation in Cartesian coordinates.

FEM modal Analysis

Used element type, real constants and material properties

Experimental Modal Analysis

For random signal theory
REF 9.4.2
REF 9.4.3

Locations of sensors

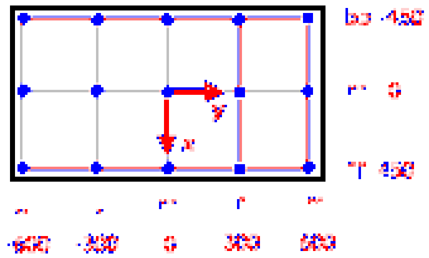
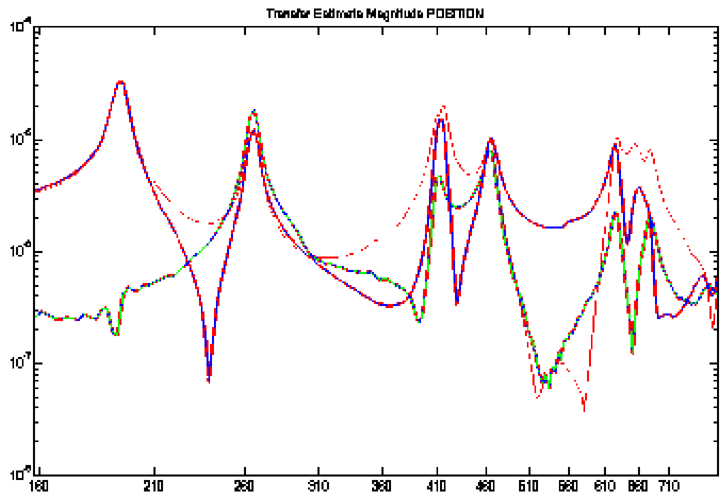


Figure 3-16 Locations for the geophones



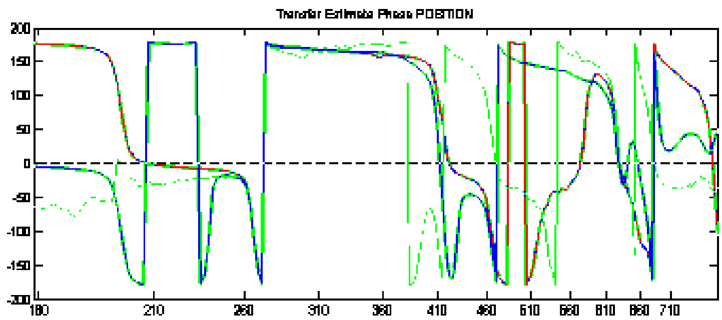


Figure 3-17 Bode diagram of three different sensor locations (bb,ll), (m,ll) and (ff,ll)

Overview of Modal Analysis

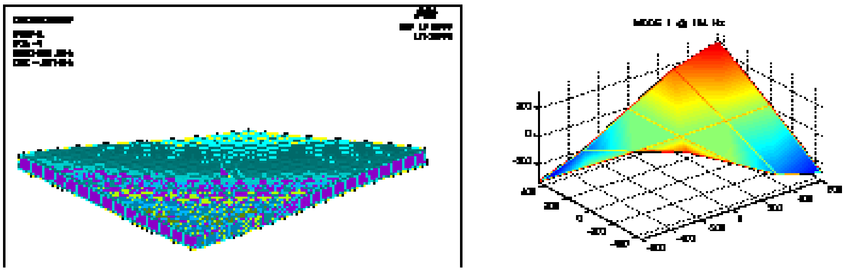


Figure 3-18 Plot of the first internal flexible resonant mode; derived from the FEM model (left) and the mesurements (right)

Table 3

Mode Number		Frequency		Error
-		[Hz]		[%]
All	Flex	FEM	Experi	
1-6	/	0	0	/
7	1	198	194	-2.0
8	2	257	266	3.5
9	3	405	/	/
10	4	428	413	-3.5
11	5	493	465	-5.7
12	6	560	624	11.4

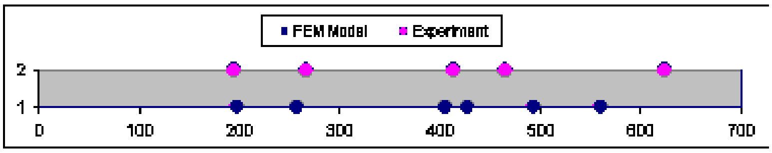


Figure 3-19

Possible solutions

Scaling
Table with higher specific stiffness

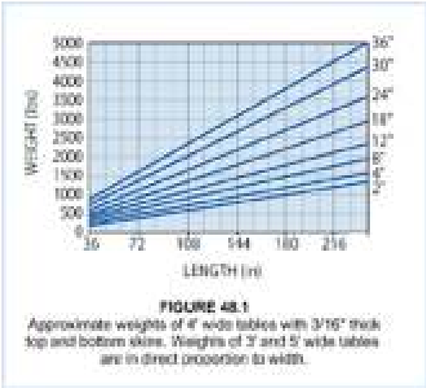


Figure 3-20

3.10 Cross coupling issues

positie van poten / sensoren / actuatoren (riget cross coupling)
crosscoupling in sesoren (+het verhaal van rens)

3.11 Passive suspension

zie VERHAAL MET "PERFORMANCIE ISSUES"

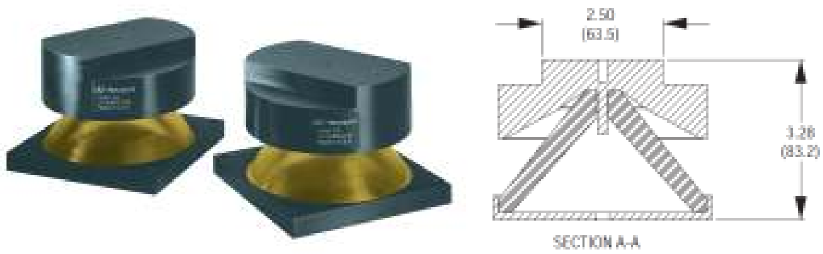


Figure 3-21 The 'NewDamp' suspention

Current Open loop transfers

hoe de passieve suspentie is terug te zien in de metingen

Effect of low stiffness open loop

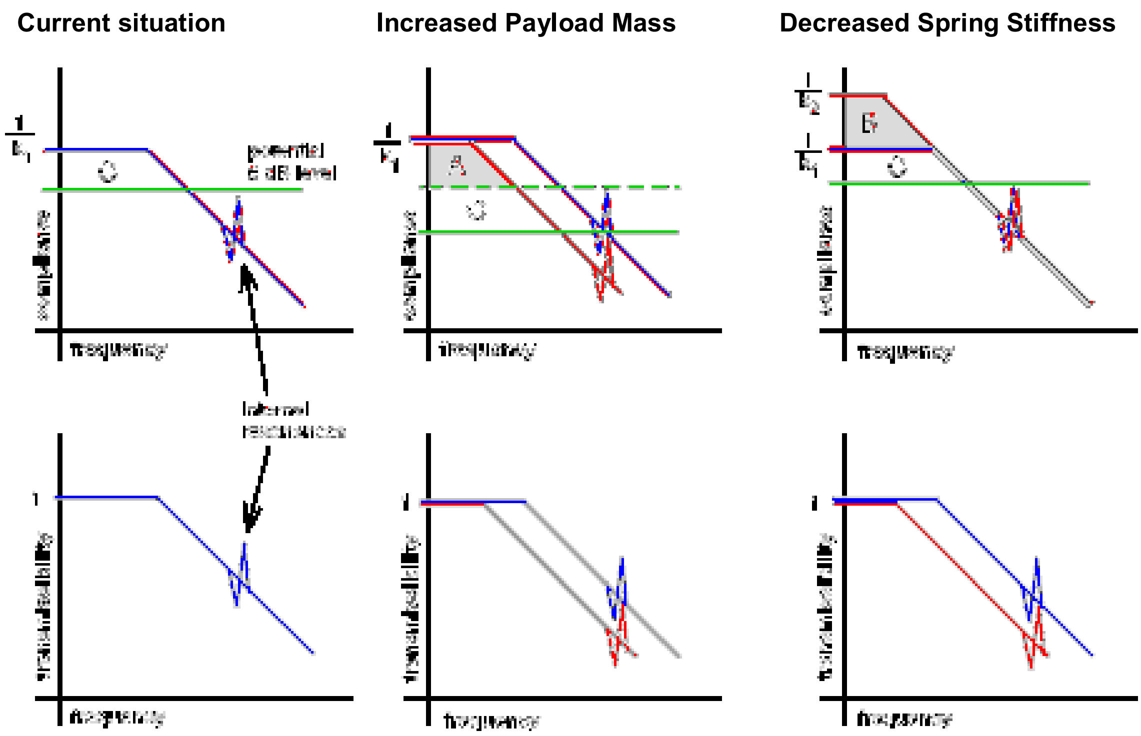
wat is het effect op de vibratie isolatie in open loop?
F>x OL

x>x OL

Low stiffness in closed loop

F>x CL
x>x CL
extra winst voor gain

Overzicht



een paar snelle conclusies overgenomen uit deelrapport

Compliance = GEBRUIKT VOOR control!

Increase mass results in:

For trans AND compliance
- natural frequency decreases
- the "mass line" shifts down, together with the internal resonances peaks

For compliance
- gain potential increases, area A is extra

Decrease Stiffness results in

For trans:
- natural frequency decreases
- the "mass line" shifts down, together with the internal resonances peaks

For compliance
- no shift of “mass line”, only more compliance at lower frequencies
- gain potential increases, area B is extra

NB:
A = B
C = C = C
Assumption: Internal resonances are coupled to the “mass line”. This is reasonable, because the internal resonance modes may be considered “decoupling” of mass from the rigid-body behaviour

DUS:
Het maakt niet echt uit of je nu meer massa toevoegd of minder stijfheid. BEHALVE
- minder stijfheid maakt apparaat gevoeliger voor directe stoorkrachten. Dit wordt in principe opgevangen door de control,. maar de control moet wel harder werken. Daarnaast is het zo dat
meer massa maakt hogere actuatie-krachten nodig. Je kan inderdaad de gain omhoog gooien (resonanties zijn lager), maar je hele systeem moet dit wel aankunnen!!!!!! (versterkers, actuatoren, ruis problemen etc)

In andere woorden: All extra compliance will be advantagous for the achieveable gain, because it is completely compensated for with the assive suspention.

3.12 Conclusion Choice of Subsystem to improve

The internal resonance modes

The issue of the internal resonances of the breadboard table are very

4 Design of the Negative Stiffness Mechanism

4.1 Design specifications

In general: decrease the natural frequency of the suspention from 10 Hz to 0.5 Hz

Frequencies

- All translational natural frequencies
  - < 0.5 Hz
- Rotational natural frequencies < 50 Hz (fixed)
- Lowest internal resonance frequency 1 kHz (fixed)
- Damping ratio 0.5 – 0.7

Stroke

- All translations +- 20 um
- Rotation +- 1 urad (?)

Dimensions

- Outer Dimensions:
  - Height: 60 cm
  - Horizontal diameter: 16 cm
- Mass: < 20 kg (?)

Table

- Mass: 100 kg
- Lowest natural frequency 200 Hz
- Dimensions: 1200 x 900 mm

Other desirable features

- Small number of parts
- Ease of manufacturing
- Ease of operation
- High scalability (different sizes)
- High generality (different application)
- No special or unconventional materials
- Clean room / vacuum compatible (?)

Design goals of the improved Kolibri table

Comparing the original design specification with the achieved performance leads to the conclusion that the design specifications are not met by 30 dB. The new design goal is:

Design a vibration isolation table with an attenuation of ground vibrations with 60 dB, maintaining the table form factor.

DIT BETEKENT DUS DAT stijfheid omlaag met een factor (k_oud/k_nieuw)^2

→ (10/0.5)^2 = 400
oftewel 99.75 % van de stijfheid moet eraf.

4.2 Solutions for low frequent suspension in literature

lineair

A straight stroke

van staal bijvoorbeeld

helical coil steel springs
flat wave cylindrical springs / steel
with or without extra damping

Figure 4-1

But: large deflexion needed, internal resonances, materiall spanningen, geometry shange, wrijving etc

Van staal met demping materiaal

bladveren met en zonder denping
rubber met grote vervormingen

minimal natural frequency apporximately 7 Hz, which is 8 cm deflexion?

omdat deflextion ← → natural frequency : Lange slag nodg

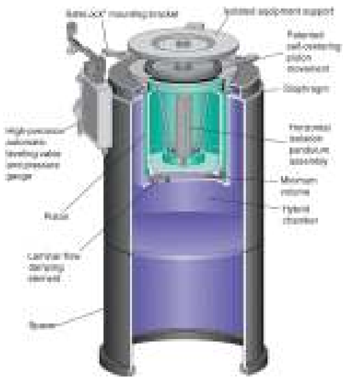
bestaat wel, bijvoorbeeld

advantage: long stroke problem can be overcome
con: extra internal resonances
extra friction

luchtpot is goed, maar…….

- extra lucht systeem
- groot

valt dus niet binnen de scope van dit project.



non linear

onderverdelings-schematje

geometrical truukjes

doorbuigen
negatieve stijfheid
lucht

Basic concepts for non linear springs

gravity compensation (see intro uitleg)
Verhaal van die euler-springs winterflood
energie
spanningen
schaling
Interne resonanties

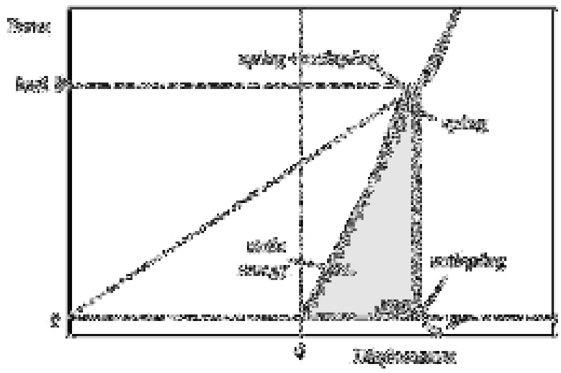


Figure 4-2

4.3 Negative stiffness overview

inspiratieverhaal van MINUSK en van Alabuzhev +patenten zut

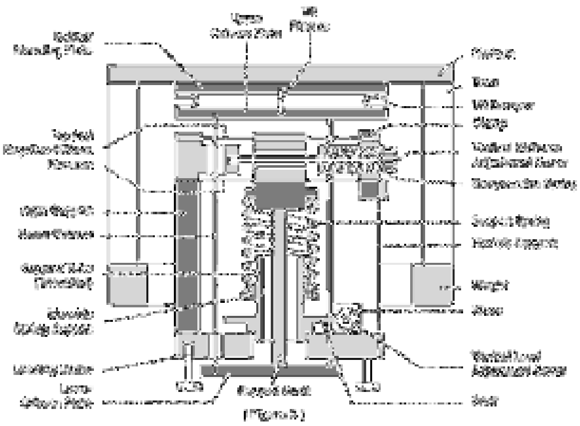


Figure 4-3

werkingsprincipe

uit boek sterkteleer

4.4 The GCM gravity compensating mechanism spring

helical springs

leaf springs

polymer springs

4.5 The Radial aanspannings mech

4.6 De hybride oplossing – proberen!

continues flexures

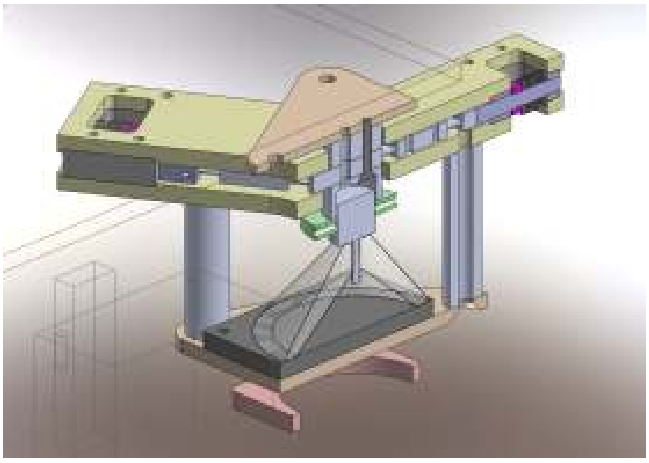
uitrekenen uitrekenen ---- kan niet!!!

4.7 Het uiteindelijke ontwerp

exel in bijlage

alle finale berekeningen

overzichtsplaatje
werktekeningen in bijlage



5 Testing the Negative Stiffness Mechanism

5.1 Test set up

breadboard
Kinematic Systems Vibraplane

Thickness	2	in
	50.8	mm
Dimensions	24 x 48	in
	609 x 1219	mm
weight per in^2	0.11	lbs/in^2
weight total	126.72	lbs
	57.5	kg

Equivalent table for natural frequeny

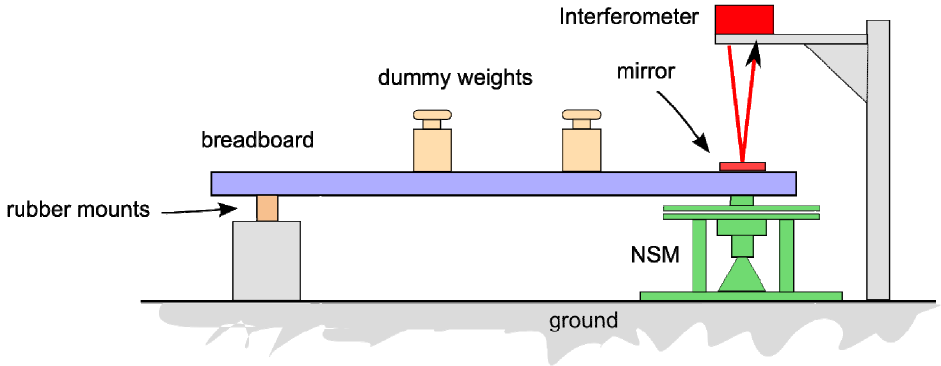


Figure 5-1
Figure 5-2

5.2 Measurements

6 Conclusions and recommendations

Literature

REFS part 1

9.4.1

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Peter Avitabile, University of Massachusetts Lowell, Lowell, Massachusetts
SOUND AND VIBRATION/JANUARY 2001

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Paul M.J. Van den Hof
Lecture Notes wb3250 - Random Signal Analysis
Course 2008 / Version 21 May 2008
Delft Center for Systems and Control
Delft University of Technology

10.1

SOME RESULTS OF AN INVESTIGATION OF A VIBRATION
PROTECTION SYSTEM WITH STIFFNESS CORRECTION
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NETI, Novosibirsk.
Iskopaemykh, No. 3, pp.
1977,
Translated from Fiziko-Tekhnicheskie Problemy Razrabotki Poleznykh
146-149, May-June, 1977. Original article *submitted* Ja nuary 10,

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A Control Strategy Using Negative Stiffness for Active (RADIAAL mechanisme)

Vibration Isolation
Tao Zhang1,2,3
Proceedings of the 3rd IEEE Int. Conf. on
Nano/Micro Engineered and Molecular Systems
January 6-9, 2008, Sanya, China

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Eupopean Patent EP
1241371 B1
Vibration Isolation system
14-07-2004 David L. Platus
(MINUSK system)

REFS part 2

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AO1 [1]

http://www.acticutinternational.se/web-content/Partners/History%20ANVC.html

OVERZICHT PAPERS

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Eric H. Anderson
ELITE-3 Active Vibration Isolation Workstation
newpor coorp

REQUIREMENT PAPERS

[2.1 1z]

Hal Amick
Buildings for Advanced Technology Workshop II
Vibrations ... Schould i worry?
Januari 2004

[2.1 1]

Low Frequency Active Vibration Isolation for Advanced LIGO
W Hua
19??

[2.1 1a]

D. B. Newell
An ultra-low-noise, low-frequency, six degrees of freedom active
vibration isolator
© 1997 American Institute of Physics
1997

[2.1 1b]

John F. O'Brien,
Six-axis vibration isolation technology applied to spaceborne interferometers

[2.1 1c]

G. Losurdo, Winterflood
An inverted pendulum preisolator stage for the VIRGO suspension system
REVIEW OF SCIENTIFIC INSTRUMENTS VOLUME 70, NUMBER 5 MAY 1999

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CarlosM. Grodsinsky
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JOURNAL OF SPACECRAFT AND ROCKETS
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M Jaensch and M U Lamp´erth
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SMART MATERIALS AND STRUCTURES, 2007

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Colin G. Gordon
Generic Criteria for Vibration-Sensitive Equipment
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San Jose, CA, November 4-6, 1991, pp. 71-85

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Hal Amick
On Generic Vibration Criteria for Advanced Technology Facilities
with a Tutorial on Vibration Data Representation
Journal of the Institute of Environmental Sciences
September/October 1997, v. XL, no. 5, pp. 35-44

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Proceedings of SPIE Conference 5933: Buildings for Nanoscale Research and Beyond
San Diego, CA, 31 Jul 2005 to 1 Aug 2005
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Hal Amick*,

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International Standards Organization, "Guide to the Evaluation of Human Exposure to
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KOLIBRI

[REF 3.1 tno 1]

Report IMC-RPT-DTS-2007-01428
Picosilent Platform, Proof of Principal

OTHER

[2.1 0a]

Ibrahim
Recent advances in nonlinear passive vibration isolators
Journal of Sound and Vibration 314 (2008) 371–452

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[REF 2.1 0] Eric H. Anderson workstation design

A partial list of applications demanding ultra-still platforms includes

• Reticle inspection and metrology
• Defect inspection and classification on patterned wafers
• Critical dimension measurement
• Scanning probe/atomic force microscopy
• Defect inspection on unpatterned wafers
• Overlay metrology
• Ellipsometer, thin film measurement
• Scanning/transmission electron microscopy

[REF 2.1 0a] Ibrahim , Recent advances in nonlinear passive vibration isolators

1. Reducing line spectra in the radiated acoustical signature of marine vessels.
2. Isolating equipment mounted in ships navigating in extreme sea waves.
3. Reducing the magnitude of the high launch loads across all frequency bands acting on spacecraft.
4. Reducing severe vibrations due to impact loads.
5. Protecting buildings, bridges, liquid storage tanks, oil pipelines, and nuclear reactor plants against the damaging effects of earthquakes.
6. Isolating laser interferometers of gravitational wave detectors.
7. Isolating electronic equipment, automotive vehicle front-end-cooling systems, and passengers from road roughness excitation.
8. Isolating automotive power-train system, engine through proper design of rubber and hydraulic mounting systems.
9. Protecting operators of hand-held machines.

State of the art applications

[REF 2.1 1] LIGO interferometry

"…the seismic isolation system is required to isolate the interferometer mirrors from ground motion above 0.1 Hz. The dominant source of motion above 0.1 Hz is the microseismic peaks near 0.15 Hz. The system needs to isolate the payload from this motion by at least a factor of five in all three translational degrees of freedom."

[REF 2.1 1a] D. B. Newell, INTERFEROMETRY earth besed

To extend the operating frequency band of earth-based interferometric gravitational wave ~GW! detectors down to 1 Hz, an unconventional system is required that provides approximately a factor of one million vibration isolation for the horizontal and more for the vertical, starting at 1 Hz.

[REF 2.1 1b] John F. O'Brien, interferometry 6 DOF stuart platform

RELEVANT verhaal niet 123 kunnen vinden

[REF 2.1 1c] G. Losurdo, interferometry, inverted pendulum

The design of a new preisolator stage for the VIRGO superattenuator is presented. The device is essentially a 6 m high inverted pendulum with horizontal resonant frequency of 30 mHz. An isolation of 65 dB at 1 Hz has been achieved. Very low forces are needed to move the whole superattenuator acting on the inverted pendulum. For this reason, the system is a suitable platform for the active control of the mirror suspension.

[REF 2.1 2] KNOPSE - microgravity and protein cristal grow

"For example, microgravity materials scienceinvestigationssuch as proteincrystal growth require a quiescent environment at frequencies as low as 0.1 Hz (Ref. 1), which is a signi. cantly lower frequency range than terrestrial vibration isolation applications."

[REF 2.1 3] CarlosM. Grodsinsky¤ Microgravity in the ISS

In region 1, the isolation system must directly transmit the very low-frequency quasi-steady accelerations(below 0.01 Hz) to prevent the isolated elements from bumping into the vehicle. The requirement for isolation from base motion implies that a rattle space must exist around the isolated elements to allow them to remain inertially stationary with respect to the vibrating vehicle.Obviously, it is undesirable for the isolated elementsto bump into themoving base becausethis not only negates the vibration isolation but also transmits an impulsive acceleration to the isolated element. In region 2, between 0.01 and 10 Hz, the amount of attenuationmust increase one order ofmagnitude for every decade of frequency. Three orders of magnitude attenuation is required in region 3 above 10 Hz.

[REF 2.1 4] M Jaensch MRI scan: accutate positioning and keeping it there

The imaging sequence of such a system can be described as follows. The pre-polarizing magnet is ramped up to full power within 100 ms, remains activated for a few hundred milliseconds and is subsequently actively ramped down within less than 50 ms. As the magnetic field generated by the prepolarizing magnet decays very quickly, the actual image-taking procedure, which relies on the collection of electromagnetic signals emitted by the polarized body, has to commence immediately after the ramping down has started. Furthermore, it is essential for the imaging process that the pre-polarizing magnet be at exactly the right, predefined location and does not deviate from this position by more than ±2 μm throughout the imaging period.

[REF 2.1 5] Deng Xi-shu Simulation of 6 DOF lithography stage

In term of research requirement, vibration reduction equipments should have six DOF and can reduce vibrations under six directions. Its working frequency range is from 0.5Hz to 250Hz. From 0Hz to 10Hz, the total PSD(power spectrum density) of the acceleration of the simulated working table is less than $10^{-7}(m^2/s^4.Hz)$, 10Hz to 40Hz, less than 10^{-7}-$10^{-6}(m^2/s^4.Hz)$, 40Hz to 250Hz,less than $10^{-6}(m^2/s^4.Hz)$. Horizontal displacement: < ±4μm; Horizontal acceleration: <±1mm/s^2

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Appendices

A1 Drawings of the Negative Stiffness Mechanism

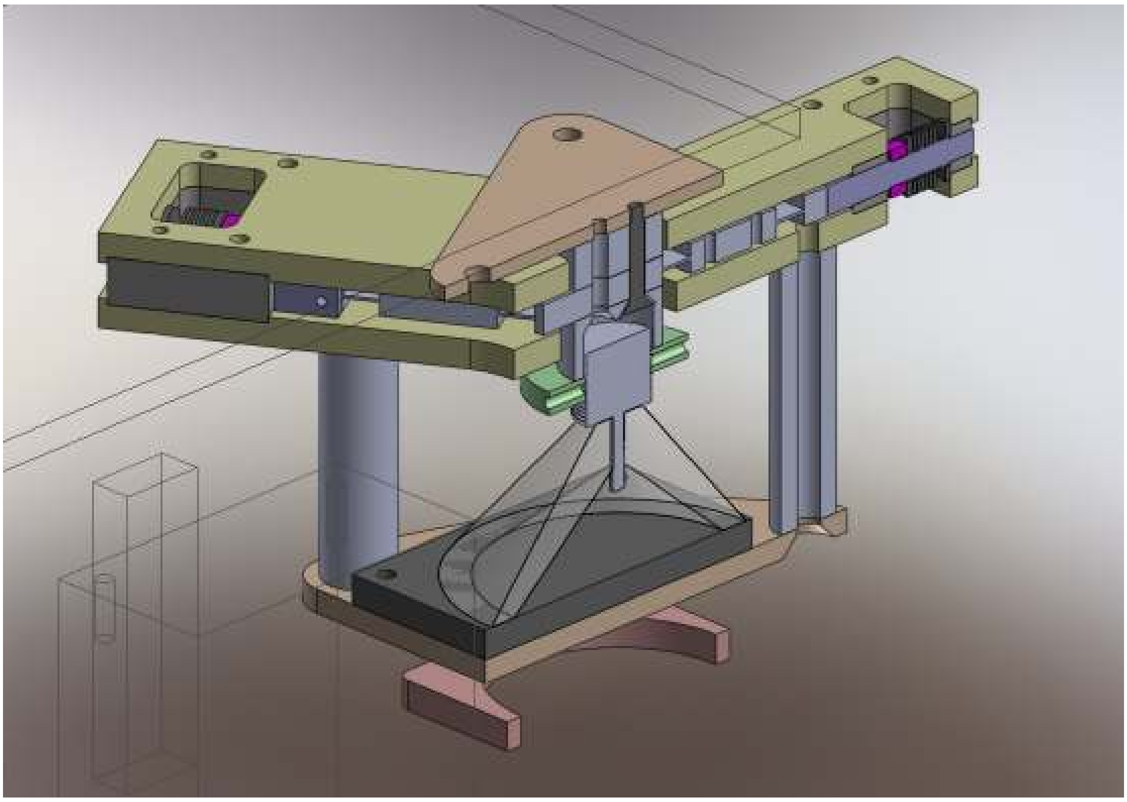
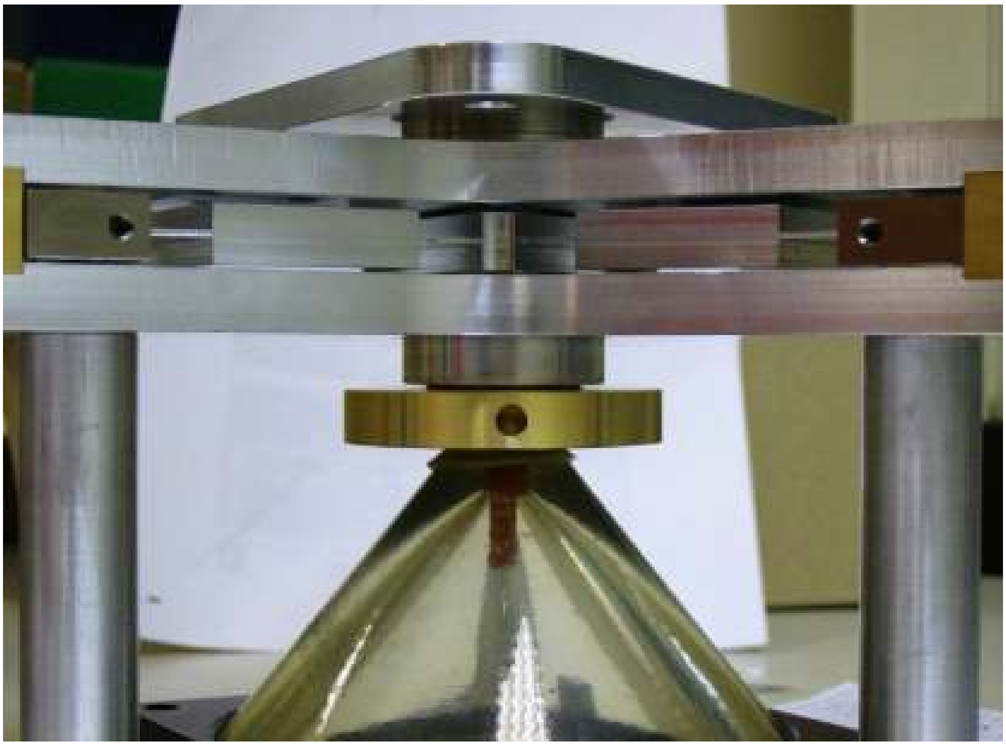
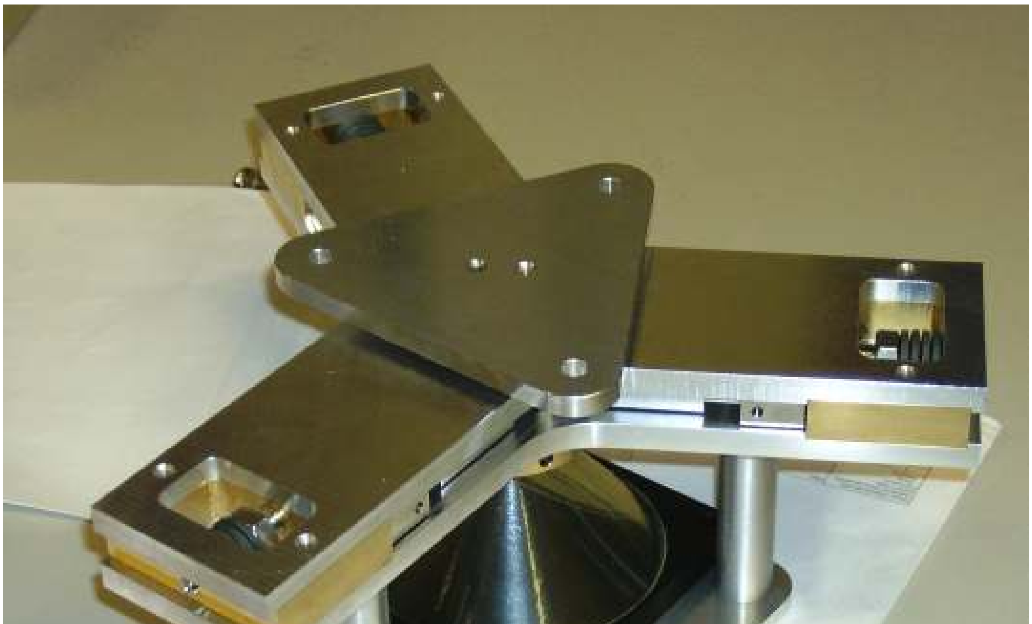


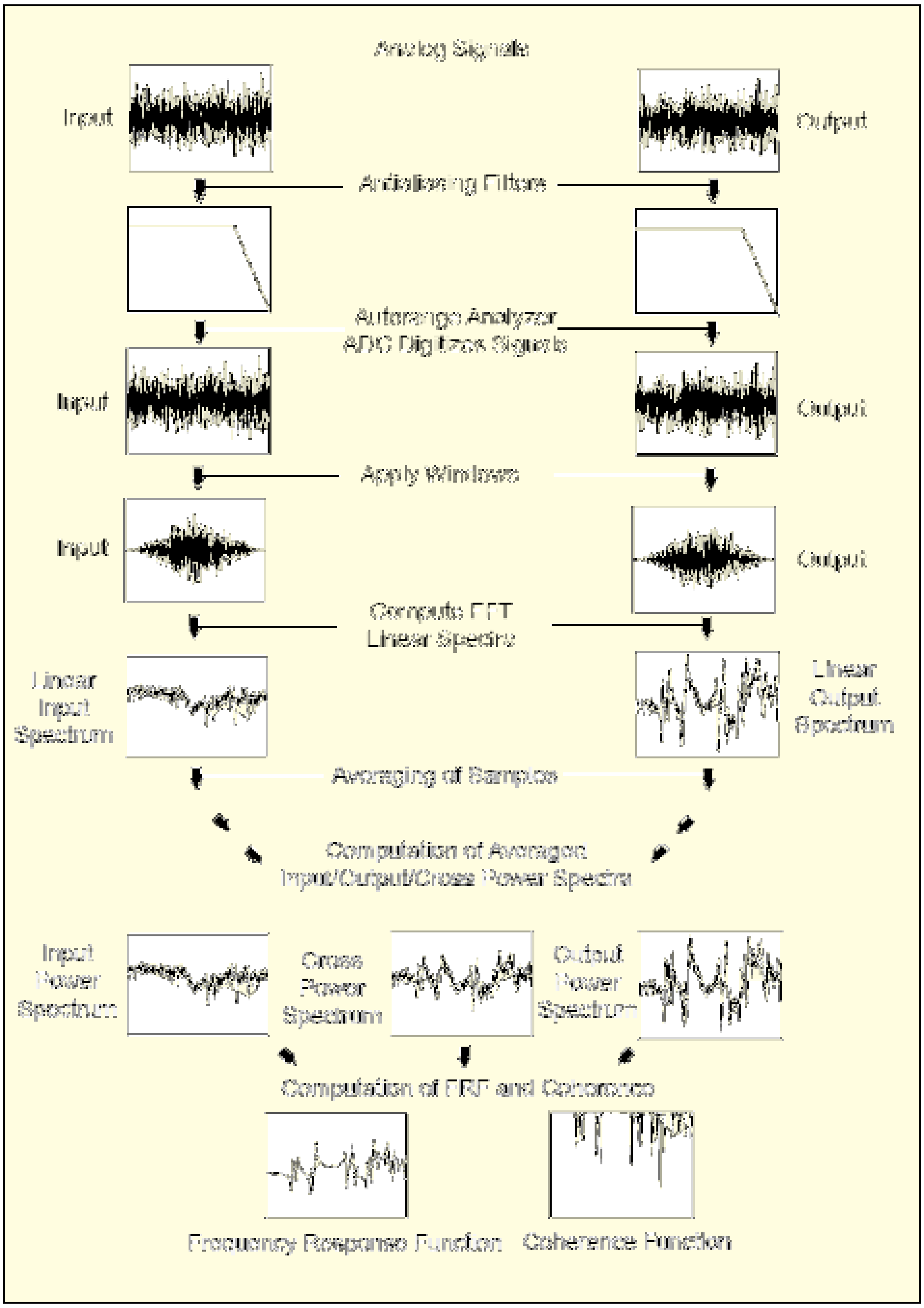
Figure 0-1

A2 Photos of the negative stiffness mechanism





B Random vibration Analysis



C Description of Vibration Isolation Criteria

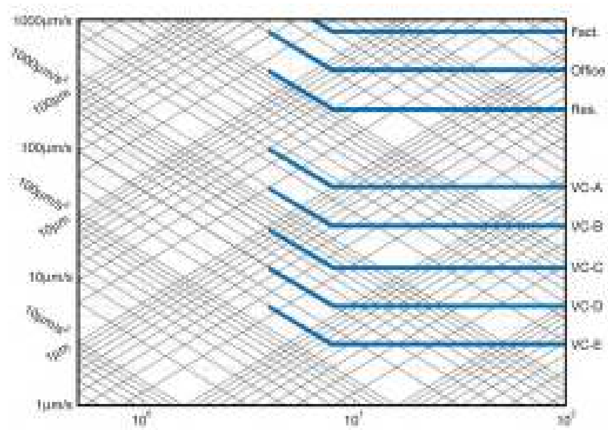
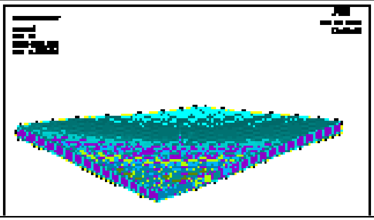
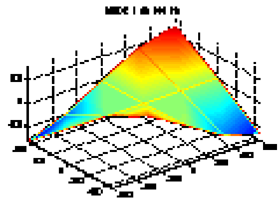
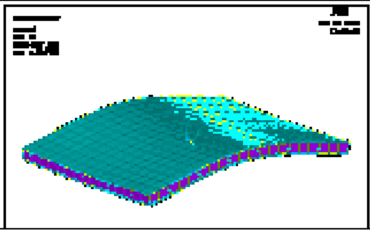
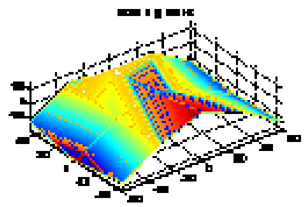
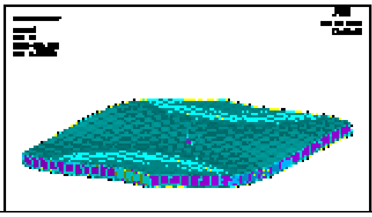
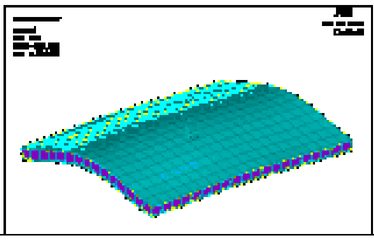
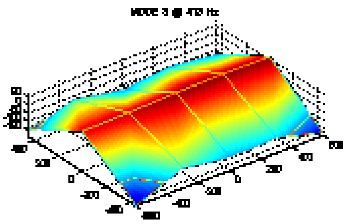
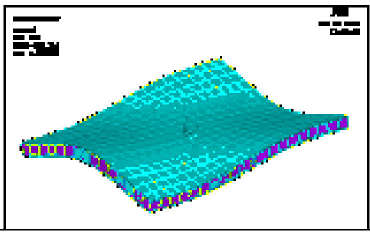
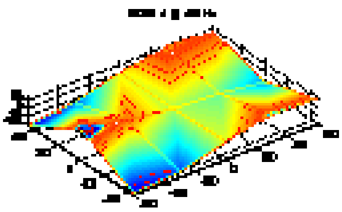
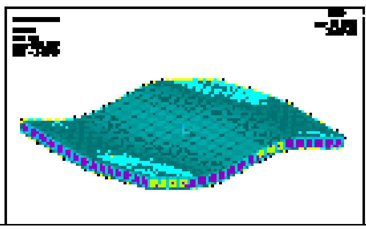
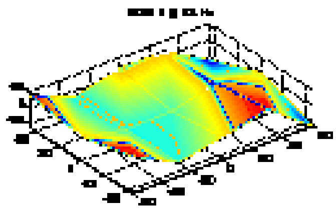


Figure 0-2

Criterion Curve	Max Level (1) micro-in./sec (dB)	Detail Size (2) microns	Description of Use
Workshop (ISO)	32,000 (90)	N/A	Distinctly felt vibration. Appropriate to workshops and non-sensitive areas.
Office (ISO)	16,000 (84)	N/A	Felt vibration. Appropriate to offices and non-sensitive areas
Residential Day (ISO)	8000 (74)	75	Barely felt vibration. Appropriate to sleep areas in most instances. Probably adequate for computer equipment, probe test equipment and lower-power (to 20X) microscopes.
Op. Theatre (ISO)	4000 (72)	25	Vibration not felt. Suitable for sensitive sleep areas. Suitable in most instances for microscopes to 100X and for other equipment of low sensitivity.
VC-A	2000 (66)	8	Adequate in most instances for optical microscopes to 400X, microbalances, optical balances, proximity and projection aligners, etc.
VC-B	1000 (60)	3	An appropriate standard for optical microscopes to 1000X, inspection and lithography equipment (including steppers) to 3 micron line-widths.
VC-C	500 (54)	1	A good standard for most lithography and inspection equipment to 1 micron detail size.
VC-D	250 (48)	0.3	Suitable in most instances for the most demanding equipment including electron microscopes (TEMs and SEMs) and E-Beam systems, operation to the limits of their capacity.
VC-E	125 (42)	0.1	A difficult criterion to achieve in most instances. Assumed to be adequate for the most demanding of sensitive systems including long path, laser-based, small target systems and other systems.

D Overview of internal resonance modes

i	Experimental	Measurement
1		
2		
3		
4		
5		
6		

E Brief history of active vibration isolation

Vibration control has a long history and may be considered a mature science [AO1]. The famous artist and scientist Leonardo Da Vinci (1452-1519) was one of the first to search into the vibrations in hydraulics, and described the phenomena that two waves can "cancel out each other", given that the timing (phase) is correct [AO2]. A good example of an early application of active vibration reduction is the prevention of "ship rolling" by Thorncraft and Yarrow in the 1850's. In 1954, Olsen and May present a solution to accomplish active noise control in a headset, as shown in figure (xxx). A similar "anti vibration" solution was used to eliminate the disturbing sounds produces by a large transformator.

The car manufacturer Nissan was the first to integrate a noise cancellation system in a car (1994). A well-known application of vibration reduction can be seen in many digital still camera's. The Japanese engineer Tetsuo Yoshida was an early inventor of this technology at NEC-Tokin in 1998 [XXX].

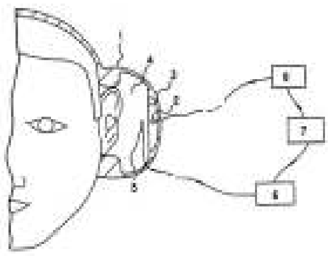


Figure 0-3

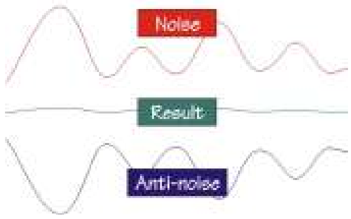


Figure 0-4

Figure 0-5

Figure 0-6

F RMS value of vibrations

Comparing equipment specifications with ground vibration levels is very tedious without a general accepted quantity for vibration levels. It is often convenient to represent an oscillating vibration signal in a form that does not involve positive and negative amplitude. One cannot generally use a time-varying average, as commonly defined, because in many cases the average of an oscillating signal is zero. Instead the average energy is used, or root-mean-square (rms), defined in the following equation, since it is based upon the square of the amplitude, which is always positive.

$$\text{rms amplitude} = v_{rms} = \left[\frac{1}{T}\int_0^T x^2(t)\,dt\right]^{1/2} \tag{11}$$

with T the integration interval and x the amplitude. If the average is taken over a short period, such as T = 1s, the rms value is calculated every second and can be plotted as a function of time. If the average is taken over a significant time period, then it is quite common to refer to this as the equivalent power level. Because ground vibrations are typical stationary (as opposed to transient), the equivalent rms value is best for representing the vibration levels.

Equation XXX can be used to calculate any rms value of an oscillating stationary signal (for example position and acceleration), but the rms velocity value has an extra advantage. The velocity is a single step of integration and differentiation removed from displacement and acceleration, respectively. It is also easier to plot velocity on the "tri-axial" paper that is often used to display displacement, velocity and acceleration values on a single graph (see **Error! Reference source not found.**).