Design of a Vibration Isolation System for a Cycle Ergometer to be Used Onboard the Space Shuttle

Submitted to:

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ERRATA SHEET

1. Page vi, line 12	Replace Detailed Analysis with Development
2. Page vi, line 13	Replace Vibration Attenuation Results with Embodiment
3. Page vi, line 14	Add 4.1.3 Interpretation
4. Page vi, line 15	Add 4.1.4 Vibration Attenuation Results
5. Page 14, line 7	Replace *third* with *sixth*
6. Page 14, line 13	Replace *fourth* with *third*
7. Page 14, line 17	Replace *fifth* with *fourth*
8. Page 14, line 20	Replace *sixth* with *seventh*
9. Page 75, line 8	Replace *4.1.2* with *4.1.4*
10. Appendix A	Replace *Transmissibity* with *Transmissibility*
11. Appendix E2	Figures mentioned are in *Ref 10.*
12. Appendix E3	Ignore *change to metric*
13. Appendix G10	Replace *Ref* with *10.*
14. Appendix J16	Replace *on page* with *on following pages*

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Executive Summary

Low frequency vibrations generated during exercise using the cycle ergometer onboard the Space Shuttle are disrupting sensitive microgravity experiments. The design team is asked by NASA/USRA to generate alternatives for the design of a vibration isolation system for the cycle ergometer. It is the design team's objective to present alternative designs and a problem solution for a vibration isolation system for an exercise cycle ergometer to be used onboard the space shuttle.

In the development of alternative designs, the design team emphasizes passive systems as opposed to active control systems. This decision is made because the team feels that passive systems are less complex than active control systems, external energy sources are not required, and mass is reduced due to the lack of machinery such as servo motors or compressors typical of active control systems.

Eleven alternative designs are developed by the design team. From these alternatives, three active control systems are included to compare the benefits of active and passive systems. Also included in the alternatives, is an isolation system designed by an independent engineer that was acquired late in the project. The eight alternatives using passive isolation systems are narrowed down by selection criteria to four considered to be the most promising by the design team. A feasibility analysis is performed on these four passive isolation systems. Based on the feasibility analysis, a final design solution is chosen and further developed.

From the development of the design, the design team has concluded that passive systems are not effective at isolating vibrations for the low frequencies considered for this project. Recommendations are made for guidelines of passive isolation design and application of such systems.

KEY WORDS: VIBRATION ISOLATION SYSTEM, PASSIVE ISOLATION SYSTEM, ACTIVE CONTROL SYSTEM, EXERCISE CYCLE ERGOMETER, LOW FREQUENCY VIBRATIONS.

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Introduction

This report discusses the need for exercise and problems that occur during exercise periods. The purpose of the project, limitations to the design process, specifications, and design methodology are also presented. Alternative designs for the isolation system are discussed including advantages and disadvantages. A feasibility analysis of the alternative design concepts are included to determine the design solution.

1.1 Sponsor Background

The design of an isolation system to reduce vibration of a cycle ergometer is sponsored by the National Aeronautics and Space Administration (NASA). In cooperation with NASA, the Universities Space Research Association (USRA) sponsors design projects nationwide. USRA was organized to foster interest in space related topics at the university level. The team worked directly with contacts in the Man-Systems Division at Johnson Space Center in Houston, Texas.

1.2 Problem Background

During orbit of the Space Shuttle, vibration from various machinery causes problems with astronomical and material science experiments. Astronomical experiments use highly accurate telescopes to view stellar bodies for extended periods. Vibration caused by the exercise equipment propagates through the orbiter, causing movement of the telescope. The telescope has a vibration control system that consists of accelerometers attached to the base of the telescope [1]. As vibrations propagate to the telescope, the accelerometers sense the vibration and activate counter-reaction servo-motors that maintain proper alignment of the telescope. Figure 1 shows an example of an accelerometer package and its position on the telescope. Unfortunately, when the telescope experiences vibrations created by the exercise equipment, the temperature of the counter-reaction motors increases significantly. Also for some vibrations, the motors are unable to react quickly enough.

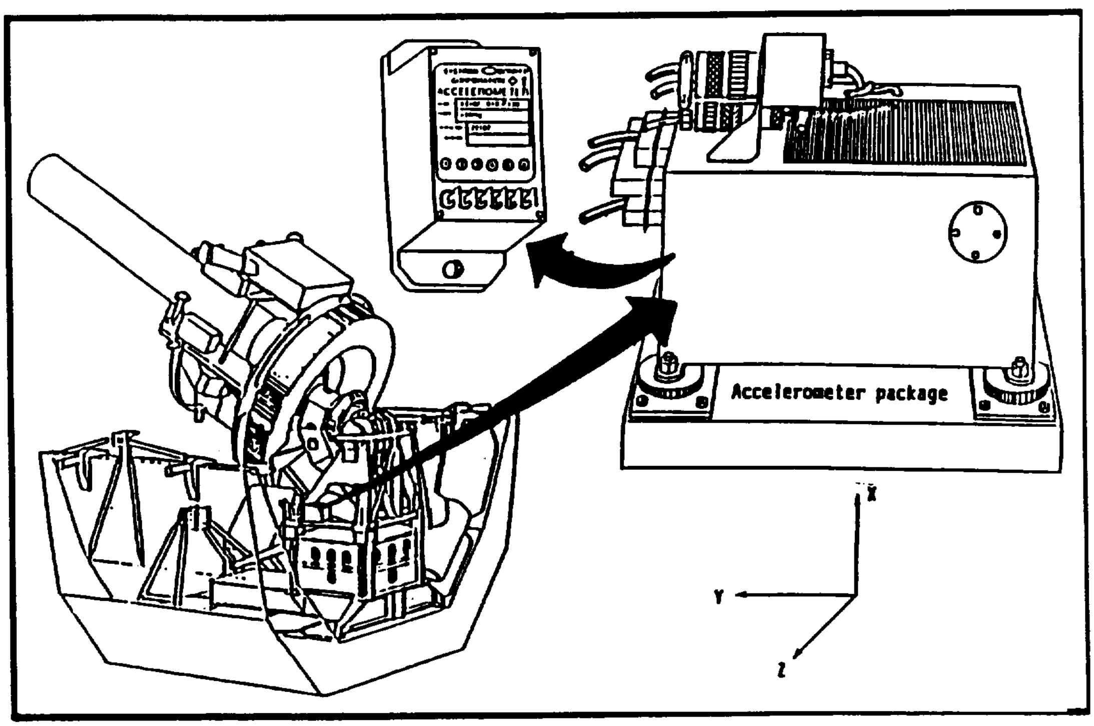


Figure 1. The accelerometer package senses vibrations to the telescope [1].

In addition to astronomical investigations, space flight provides the opportunity for many advances in material science. The microgravity environment associated with space flight in low earth orbit provides an opportunity to study physical processes which are difficult to investigate in the gravitational field of the earth [3]. Although the environment provides for excellent conditions for physical processes, acceleration frequency and amplitude due to vibration of equipment on board is not as steady or low as needed. For instance, when an astronaut exercises, the vibration imparts acceleration to the orbiter, thus affecting the experiments.

A solution to these problems may be to eliminate the exercise routine on the orbiter. However this is not a feasible solution since exercise is a necessity for the astronauts in space flight. After a prolonged period in space, the human body experiences muscle atrophy and bone decalcification [2]. These effects can prevent astronauts from performing important tasks including flying and landing the orbiter upon re-entry and performing escape procedures in case of an emergency. By exercising, these effects can be reduced . To perform the required exercise, a treadmill has previously been used. However, the cycle ergometer will be used for future flights. Figure 2 shows two configurations of the cycle ergometer: one in an upright position, the other in a recumbent or reclined position. Coupling clamps are provided for the attachment of the two configurations. These couplings consists of threaded clamps that are attached to the mid-deck floor and a clasp that grips the ergometer base to the floor.

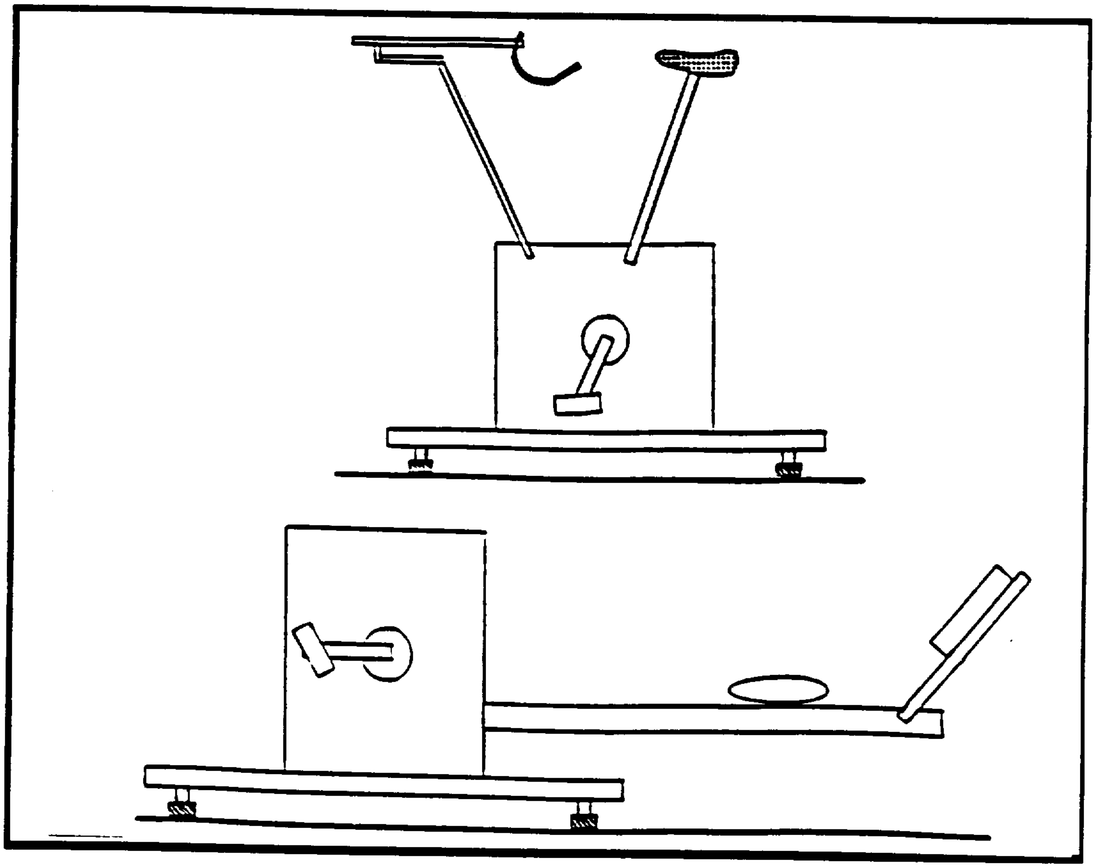


Figure 2. An example of the cycle ergometer. Two riding positions are shown, upright (top) and recumbent (bottom).

1.3 Purpose of the Project

The objective of the project is to develop a vibration isolation system for a cycle ergometer to be used on the orbiter mid-deck of the Space Shuttle (see Figure 3). Reducing the vibration caused by the cycle ergometer will permit astronauts to exercise during ongoing microgravity experiments and astronomical observations.

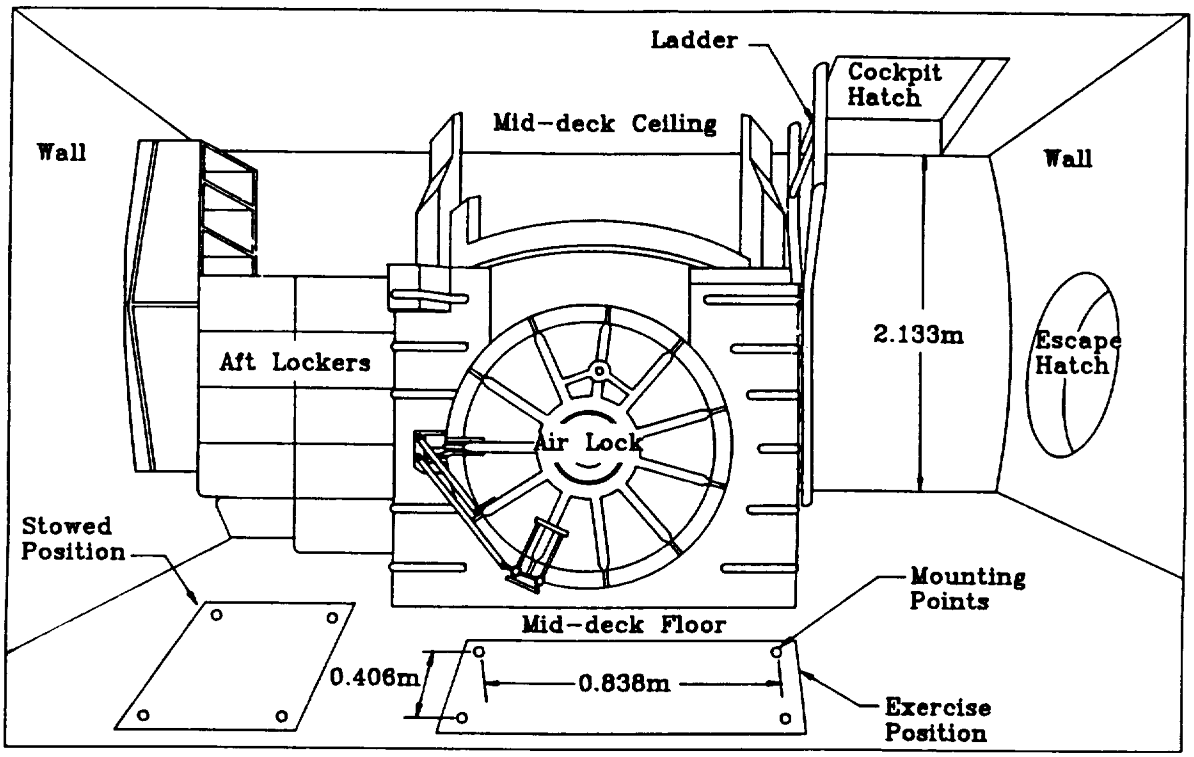


Figure 3. View of the orbiter mid-deck showing the position of the cycle ergometer mounting points.

1.4 Scope and Limitations

The design team focused on vibration due to the ergometer and the rider as opposed to other vibration sources in the space shuttle. Since only the ergometer structure is detailed and an ergometer cycle has not been developed, the team will not recommend or address any changes to the cycle ergometer. The sponsor conveyed that contracting companies are currently bidding on the design of the cycle. Prototypes of the cycle ergometer consist of planetary gearing or sprocket and chain systems. Information on the actual cycle ergometer was not given by the sponsors since majority of the vibration is from the motion of the rider.

The design team research types of isolation systems and found two types: passive and active. A passive isolation system is a system that is independent of sources to dampen input forces. A force is input to the isolator, and it is capable of reducing the vibration and give a reduction in the output force. An active system is dependent on sources, such as sensors. A signal is received and a controller sends a message to a reaction mechanism such as motors, valves, or switches.

The team decided to emphasize passive systems for several reasons. Passive vibration isolators are simple and reliable mechanisms isolating the vibration from the surroundings. Active systems are advantageous in that these systems use sources that can react quickly to an excitation and response. However, active controls use external energy sources and sophisticated controlling schemes. Also, weight and cost is increased with an active system due to typical machinery associated with active systems such as generators and servo-motors. The design team believed that passive systems are an effective solution to the problem.

1.5 Constraints

NASA specified that $1x10^{-4}$ g's is the maximum acceleration that may be input to the shuttle (see Appendix A). The mass of the shuttle is 106,625 kg (235,000 lbf). Applying Newton's 2nd law, the maximum force to be transmitted to the shuttle floor is 11.3 N (2.5 lbs). The transmissibility ratio, which is the proportionality of the output force of the isolation system to input force of the rider. The experimental input data gives an approximate maximum force of 445 N (100 lbf) that is input to the isolation system. This gives a transmissibility ratio of 0.25. Therefore, the isolation system must give a transmissibility ratio of 0.25 over the range operating frequencies.

1.6 Specifications

The design team was given several requirements and specifications for the design of the isolation system. The specifications, given in Appendix B, include standards given in the Interface Definition Document (IDD). The IDD contains requirements by NASA for equipment designed for the space shuttle [4]. Also, the specification list gives detailed requirements of the orbiter mid-deck in Figure 3. The major requirement considerations for payloads onboard the shuttle are as follows:

- Isolation System
- Cycle Ergometer Including the Isolation System
- Push-Off Loads
- Acoustics
- Heating and Cooling
- Electrical Energy
- Limitations on Mid-deck Payload Utilization of Electrical Power

1.7 Design Methodology

The team developed the design of the isolation system using the Pahl and Beitz design methodology [5]. This methodology consists of four phases: clarification of the task, conceptual design, embodiment design, and detail design.

Clarification of the task involves using the given information about the design and developing a list of detailed specifications. The conceptual design phase involves the development of a function structure, establishing solution principles, and combining these principles to create concept variants. A function structure determines the overall function of the problem using the requirements of the design. By defining the flow of energy, materials, and signals with the use of a block diagram, the function structure expresses the relationship between the inputs and outputs of the system. The sub-functions of the structure contribute to the development of conceptual variants for the overall design. These concept variants are then evaluated using a decision matrix developed by the design team. This decision matrix involves comparing the alternatives using economical and technical criteria and weighting each alternative accordingly. By weighting each alternative, the decision matrix will produce the most feasible design for the project with respect to the given specifications.

Embodiment design is the development of a preliminary alternative design with economic and technical considerations. This includes selection and feasibility of each component and preliminary sketches. Detailed design includes the arrangement and form in conjunction with production drawings, manufacturing processes, and assembly of the design.

To aid the development of design alternatives, the team further defined the isolation system from the project statement in terms of energy flows of the system. This was necessary to understand the important vibration components of the cycle and isolation system. The energy flow diagram, given in Appendix C, details the energy transmitted from the rider, to the cycle ergometer, isolation system, and the mid-deck floor. The team used this energy flow diagram to develop conceptual variants for the overall design.

Alternative Designs

The design team focused on passive systems when developing alternatives for the isolation system; however, active isolation systems are evaluated. Our main objective is to determine if passive systems can be used for our application. Since the mass and energy need to be minimized for the zero-g environment, the design team believed the use a passive system is appropriate. The active controls are presented to give a general perspective of both systems.

The design team will dampen vibration by coupling an isolation system to the cycle ergometer. This system will filter vibrations created by the ergometer from the space shuttle. This can be accomplished by placing the isolation device at several positions in the system. Two example positions are shown in Figure 4. One position of the isolator, shown as Position A, is between the ergometer and coupling. Another position, shown in Figure 4 as Position B, is between the coupling and the mid-deck floor.

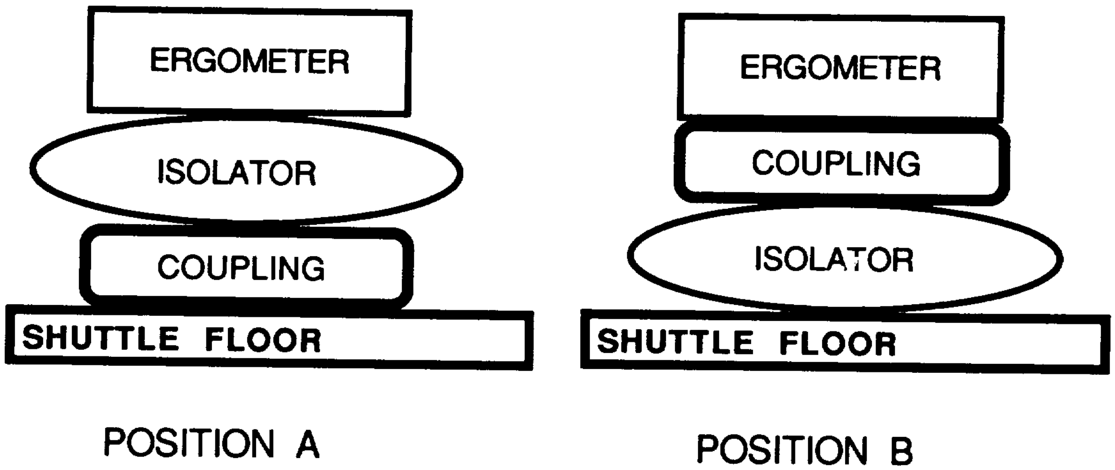


Figure 4. Positions of the Isolation System

The team developed six categories of alternatives for the design of the isolation system. The alternatives are based on several design considerations. Energy domains considered include electrical, hydraulic, mechanical, electromagnetic, and pressurized air. The isolation systems in each category are passive systems except the pressurized air and electromagnetic isolation systems. Active controls are not pursued to the extent of passive systems due to the increased complexity of these systems and constraints of the environment. The isolation system alternatives are:

2.1. Resilient Absorbers

- 2.1.1 Intermediate Absorber
- 2.1.2 Coupling Isolator
- 2.1.3 X-Support Isolation System
- 2.1.4 Ball and Socket Isolator

2.2. Spring/Damper Suspended Structures

2.2.1 Suspended Structure Isolation System

2.2.2 Isolation Box

2.3. Fluid Bladder Isolation System

2.4. Wedge Block Isolation System

2.5. Platus Vibration Isolation System

2.6. Air Cushion Isolation Systems

2.6.1 Air Bearing Isolator

2.6.2 Fan Isolation System

2.7. Electromagnet Isolation System

The first design concept uses resilient material to isolate the vibration between the mid-deck floor and the ergometer. Four different alternatives are developed using this concept: the intermediate absorber, the coupling isolator, the x-support isolator, and the ball and socket isolator. The intermediate absorber is a resilient material where the geometry of the absorber can be modified to obtain the most efficient damping system [6]. The coupling isolator dampens vibrations by isolating the entire system including the mounting clips of the ergometer. The X-support isolation system consists of two X-brackets and a resilient material. This system is mounted between the ergometer support structure and the shuttle floor. The ball and socket isolator consists of a sphere surrounded by a resilient material encased in a rigid spherical casing.

The second design concept for dampening vibration uses springs and dampers within a suspension structure. The suspend structures is a general concept in that different types of springs, such as helical or compression, and dampers, such as pneumatic can be used. One alternative using this concept is the suspended structure isolation system. The cycle ergometer is mounted in a rectangular support structure which is suspended between the mid-deck ceiling and floor with springs and dampers. Another alternative is an isolation box in which the ergometer and support structure are contained and suspended at the corners of a box that is coupled to the mid-deck floor.

The third concept makes use of an active air damper isolator where air is provided by pumps driven by the ergometer or with external power. The two air isolation systems, air bearing isolator and fan isolation system, create an air cushion upon which the ergometer is elevated. The system is active in that for different heights of the ergometer, a controller determines the airflow rate of the air cushions.

The fourth alternative, fluid bladder isolation system,dampens vibration by the exchange of liquids in flexible chambers. The fluid passing between the chambers must be of a non-petroleum base because of the required specifications from NASA.

The fifth alternative, wedge block isolator, dampens vibration through Coulomb (sliding) friction created by the movement of wedge blocks in response to the motion of the rider.

The electromagnet isolation system, sixth alternative, is an active control system. Damping is accomplished by actively varying the magnetic fields to cancel the forces that produce the vibrations.

The following sections describe the alternatives in greater detail. The advantages and disadvantages are also discussed for each concept based in part on how the alternative meets the specification list.

2.1 Resilient Absorbers

Two alternatives were developed using resilient material: an intermediate and resilient pad damper.

2.1.1 Intermediate Absorber The intermediate absorber consists of a resilient material placed at each foot of the support structure. The geometry of the absorber and the material can be altered to maximize the damping effect [6]. An example of an intermediate absorber is shown in Figure 5. The resilient material has air holes or voids that provide stress concentration areas within the material. These stress-concentrated areas dissipate the vibration by making it difficult for the vibration energy to be transmitted through the material around the voids to the mid-deck floor [7].

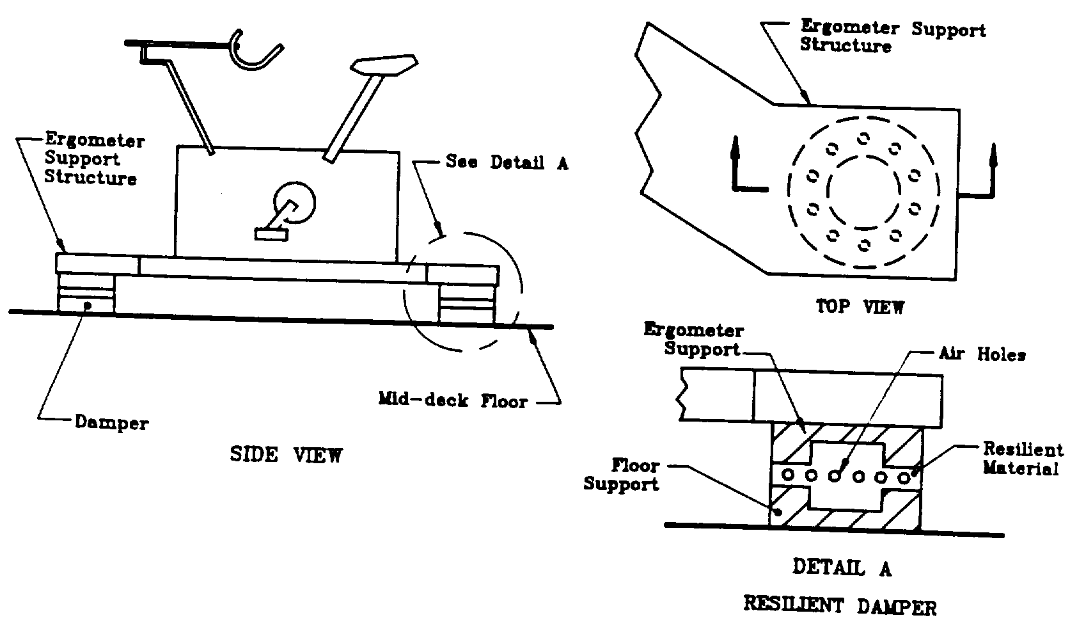


Figure 5. Intermediate Resilient Absorber

2.1.2 Coupling Isolator By examining the standard clamps that attach equipment to the floor of the mid-deck, the design team found that the mushroom shaped studs can move freely, to an extent, within the coupling clamp. This vibration of the stud within the coupling clamp can impart vibration to the mid-deck floor. The coupling isolator, shown in Figure 6, prevents these vibrations from occurring by isolating the entire ergometer and mounting clamps from the mid-deck floor. The isolator consists of a screw that is mounted into a pad of resilient material. Bonded to the opposite side of the pad is a rigid plate with an attached mushroom stud.

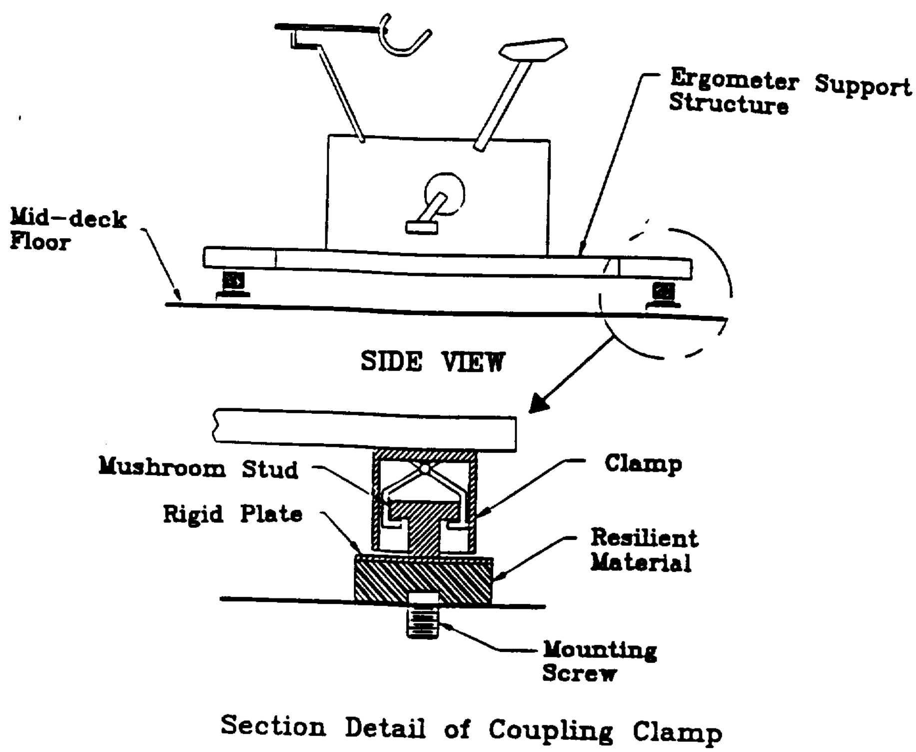


Figure 6. Coupling Isolator

The intermediate absorber and coupling isolator have several advantages. Using resilient material is a straight forward and proven approach to damping vibrations [6]. Materials such as elastomers and plastics are used to dampen vibrations of support structures that are attached to a fixed surface. Also, the geometry and material of the absorber can be altered to enhance the performance of the damper [8]. Another advantage of the resilient absorbers is that the absorbers require minimal usage of tools. In most cases, the absorbers can be attached either by threaded joints or simple clamps. Since the absorbers consist of a resilient material, storage space of the dampers is minimized. The storage space occupied by the absorbers can be stored into one locker, thus allowing the other locker to be used for other important equipment.

A disadvantage to the absorbers is wear of the material after extensive usage. Methods of attaching resilient material to a rigid surface might involve adhesive bonding or fasteners molded into the material. In both of these cases, extended use can weaken adhesive bonds and parts molded within the material can be wrenched from the material [6].

2.1.3 X-Support The X-support system, shown in Figure 7, is positioned between the ergometer support structure and mid-deck floor. Mounted to the bottom of the support structure and to the the mid-deck floor are X-support brackets. At the cross point of the brackets, a resilient material is used to couple the two X-brackets. Additional resilient dampers are placed at or near the ends of the brackets to dampen vibration due to extreme impact loads.

The X-support system was developed on the principle that as the distance vibrations have to travel to a point increases, the magnitude of vibrations encountered at that point decreases [6]. As the rider moves, the vibrations are transmitted along the lengths of the brackets to the resilient damper. The vibrations are damped along the length of the brackets when the resilient material compresses in reaction to the vibration encountered. The dampers placed at or near the ends of the brackets are used to prevent contact of the brackets due to high amplitude vibrations.

The X-Support system has several advantages. The absorber is positioned in the support such that it will dampen multidirectional inputs. As forces are input in the x,y, and z directions, the X-structure moves accordingly encountering the resilient dampers. Also, the support system can be mounted to both the recumbent and upright positions of the ergometer. However, with the upright position, the X-support is limited to the 0.3 m (10 in.) height that the cycle can be raised from the mid-deck floor (see Appendix B). Another advantage of the X-support is that the path for which the vibration will be transmitted is increased. Vibration must travel from the ergometer through the X-supports and damper to the mid-deck floor.

A disadvantage to the X-support is that due to the size of the system, exceeding the maximum storage constraint must be considered. The two mid-deck lockers provided consist of storage trays with the largest tray having a volume of 0.05 m^3 (1.8 ft^3). One method of eliminating this storage constraint is to permanently attach the X-support to the ergometer.

Unfortunately, this configuration gives rise to other strict launch constraints. Specifically, the weight of the ergometer cannot be increased by 1.82 kg (4 lbs) during launch. Additionally, the ergometer must be rigidly supported to prevent any movement of the ergometer structure during launch. Another disadvantage to the X-support isolation system is that instability due to moments created at the center of the cross brackets may occur. Careful considerations must be given to the design to prevent this rocking and possible resonance effects. Another disadvantage is that the assembly time to install this system may be high.

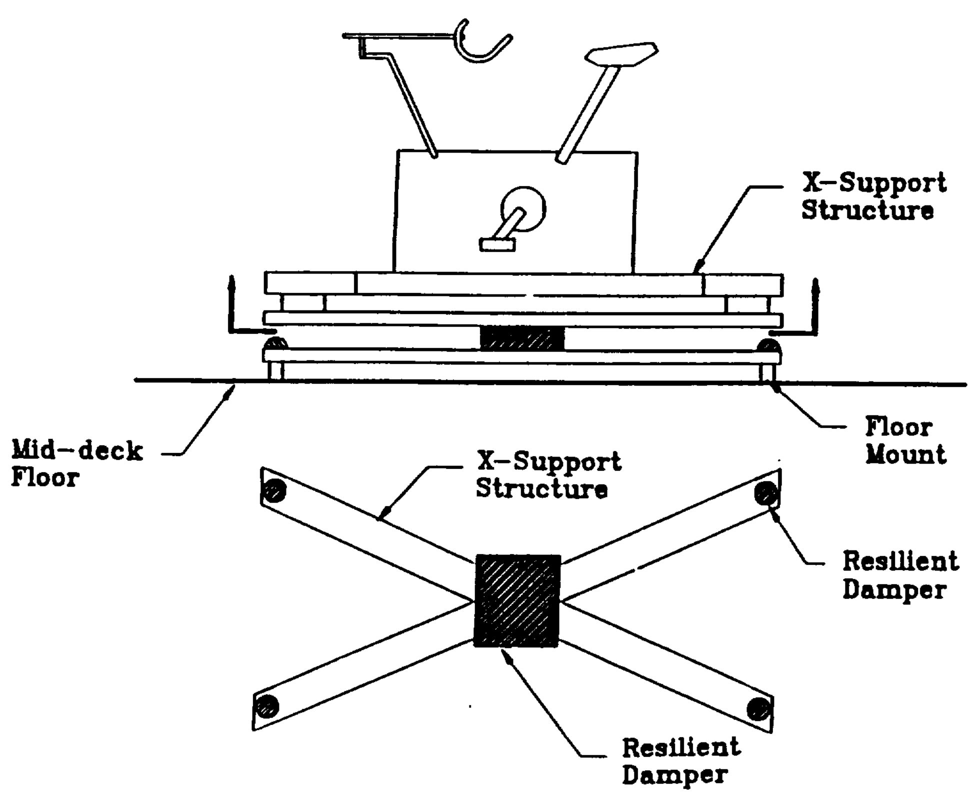


Figure 7. X-Support Isolation System

<u>2.1.4 Ball and Socket Isolator</u> The Ball and Socket isolator, shown in Figure 8, enables damping of vibrations from the motion of the rider in the all directions. The damper consists of a ball on the end of a rod both made of rigid material. This ball is mounted inside another hollow sphere. Filling the volume between the centrally mounted rigid ball and spherical shell is a resilient material. The ball and socket isolator is mounted between the ergometer and the mid-deck floor. As the cycle vibrates, the vibrations are transmitted to the rigid rod and ball. As the ball vibrates, the resilient material absorbs the vibrations by compressing and retracting.

An advantage of the Ball and Socket isolator is that the damping action is multidirectional and nearly uniform. Since resilient material lines the sphere, the isolator dampens vibration in the x, y, and z directions. Another important advantage of the ball and socket damper is low mass which is important for spaceflight. The ball and socket also requires minimal tools usage. The dampers are pre-assembled and pre manufactured. In orbit, the isolators can be simply screwed on to the ergometer support structure and clamped to the floor using the coupling clamps.

A disadvantage of the ball and socket is the difficulty of manufacture. The ball and socket has two components (the resilient material and spherical casing) that must be bonded which may require specialty manufacturing. Another disadvantage of the ball and socket is that spares are necessary. Failure of an isolator would be difficult to repair while the shuttle is in orbit.

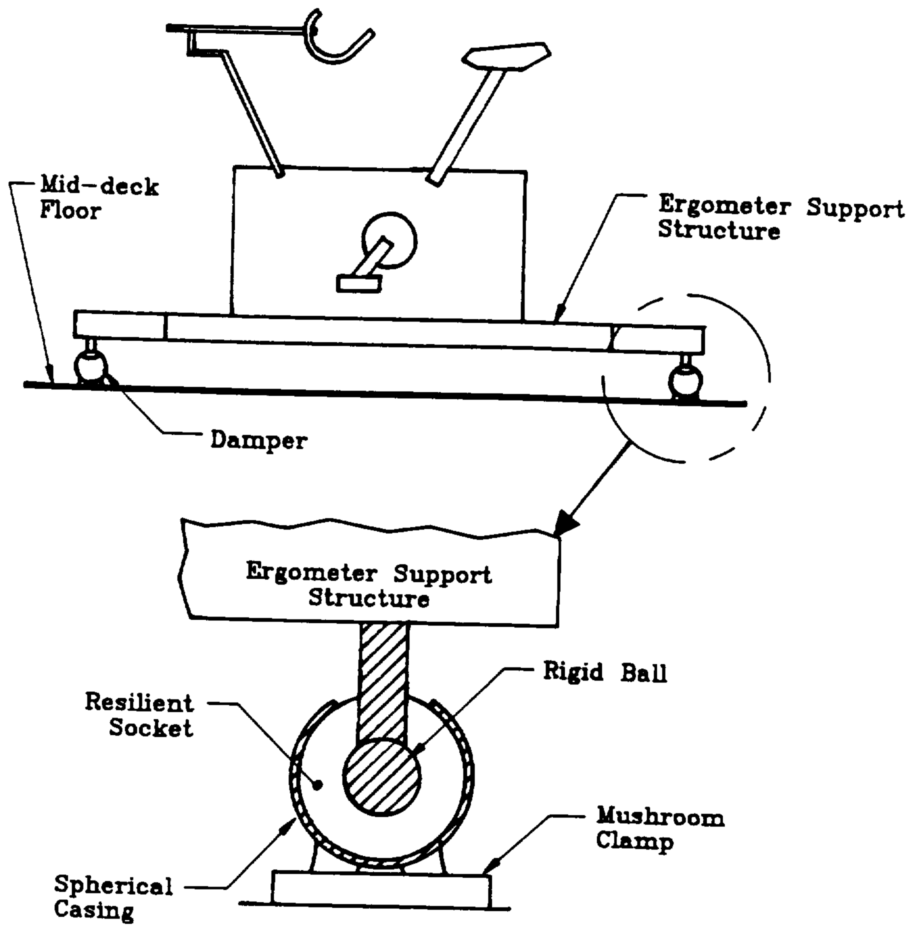


Figure 8. Ball and Socket Isolation System

2.2 Spring/Damper Suspended Structures

Another method of isolating the cycle from the shuttle is to develop a suspension system for the ergometer using springs and dampers. Two methods are developed using this concept. The first method involves suspending the cycle ergometer from the ceiling and floor of the orbiter. The other method involves suspending the ergometer from a box containing spring and damper elements.

2.2.1 Suspended Structure The suspended structure allows an opposing force to dampen the input force in the x, y, and z directions. To suspend the ergometer from the ceiling and the floor of the orbiter, the cycle, shown in Figure 9, is mounted on a rectangular box structure open on all sides. At each corner of the box is attached a spring and damper. These spring/damper components are then attached to tie points on the ceiling and floor of the orbiter. The attachments of the spring/damper components are configured such that movement of the components are allowed in all directions. Possible attachment joints are ball and socket joints or simple hooks.

Advantages to the suspended structure include tne low mass of the suspender. Also, the suspended structure dampens multidirectional inputs. Since springs and dampers are located at every corner of the structure, the symmetrical orientation of the isolators dampens any vibrational forces supplied by the rider.

Disadvantages of the suspended structure includes the amount of space needed for the system. This will interfere with movement of the astronauts, thus also making the system unsafe. If any of the cords break during orbit, spares are also necessary, thereby increasing the mass carried aboard the orbiter. Another important disadvantage of the suspended structure is assembly time. The astronauts must check to see if all of the cords are hung and the cycle ergometer elevated properly. Another possible disadvantage is the issue of stability. It is possible that residual motions would lead to additional motion of the ergometer.

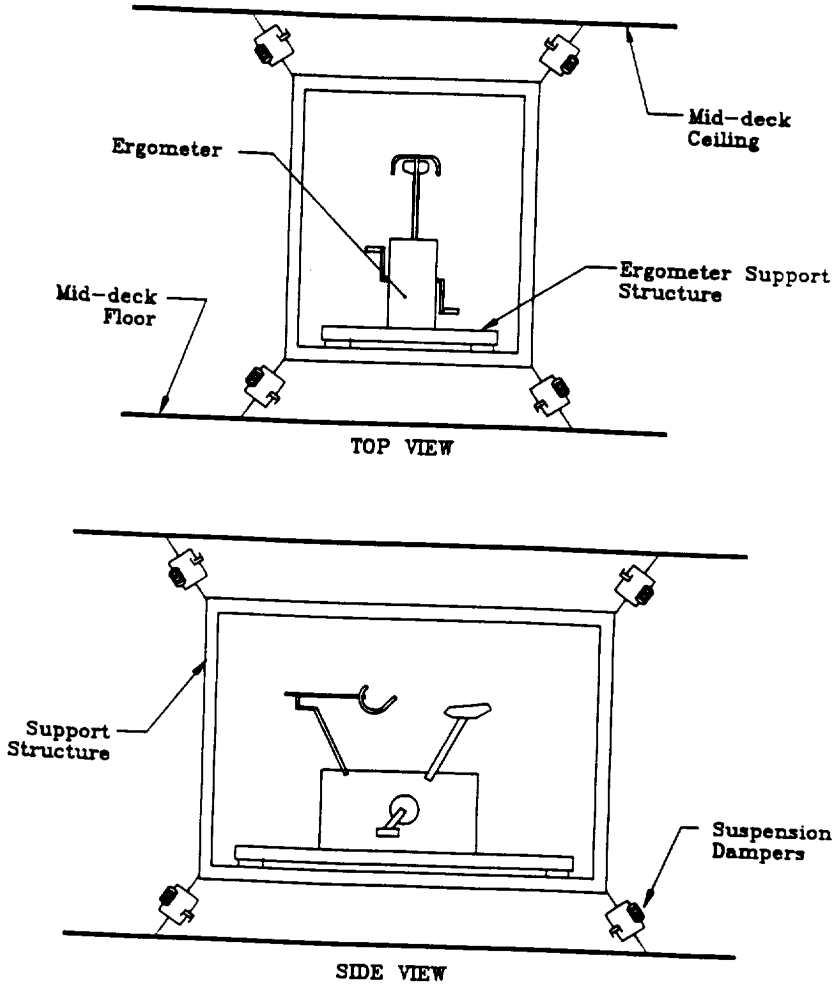
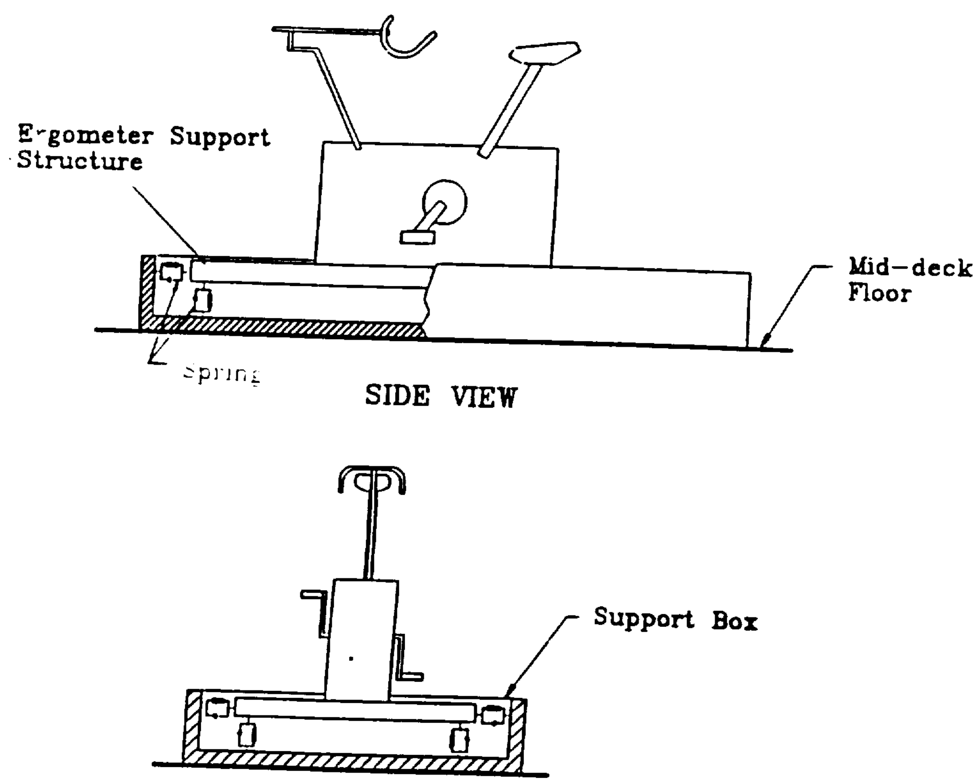


Figure 9. Suspended structure

<u>2.2.2 Isolation Box</u> Another method of suspending the ergometer is the use of an isolation box, shown in Figure 10. This system consists of the ergometer suspended in a box using spring/damper components. These components are arranged, so that in each corner a spring and damper are aligned in the x, y, and z directions. The spring/damper components are attached to the ergometer support structure and the sides of the boxes in the same manner as the suspended structure.

An advantage of using spring/damper components is the elimination of external sources of energy such as electricity. Also, the isolation box dampens multidirectional inputs of the rider. Since the spring/damper components are placed at each corner of the ergometer support structure, the symmetry within the system allows for an isolator to dampen the vibration in all directions.

A disadvantage to the use of spring/damper components is that minimal storage space is required for the implementation of the system. The spring/damper components will be added during orbit and the box will be attached before launching. However, the system requires more assembly time to provide for maximum damping. Also, as in the suspended structure alternative, the issue of instability may be an important disadvantage.



FRONT VIEW

Figure 10. Isolation Box

2.3 Fluid Bladder Concept

The fluid damper concept uses two fluid chambers to obtain a spring rate for the damping effect. The fluid flows through these chambers by an orifice to obtain the fluid inertia effect needed for the cancelling of the input force. The vibration isolator, shown in Figure 11, is comprised of two chambers, an upper (C1) and lower (C2) liquid chambers [8]. The two chambers are able to change volumes by elastic deformation of an elastomer with longitudinal vibration capabilities. Also, two other chambers (C3 and C4) are able to change volumes by elastic deformation of an elastomer with lateral vibration capabilities. The flow of liquid between the chambers in both directions is restricted by an orifice which is partially bounded by an elastomer material. The elastomer material has two functions: to carry the static and dynamic load caused by the rider and to act as a piston to pump liquid through the orifice into the second chamber.

In the process of pumping the liquid through the orifice, the elastomer material bulges slightly to such that not all the liquid is displaced through the chambers. The compliance of the first chamber is so stiff that the resisting motion of the orifice is high too oppose the fluid. The bottom chamber accumulates the liquid passing through the orifice. When the first chamber expands, the compliance of the bottom chamber allows for the liquid to be sucked through the orifice.

Multidirectional damping is accomplished by two fluid bladder dampers in one. The damping effect of the bladder isolator is developed by the communication between the upper and lower chambers (C1 and C2) with the orifice in the longitudinal vibration. Lateral vibration is dampened by the two chambers (C3 and C4) with the exchange of liquids through another orifice.

Advantages of the bladder system include the ability to dampen multidirectional inputs. The fluid bladder will dampen in the longitudinal and lateral direction because the exchange of liquid between the chambers is in all directions. Since the bladder system consists of an incompressible liquid to dampen, no external power is needed. Also, minimal tools are needed. The coupling of the bladder isolator to the ergometer and mid-deck floor can be achieved by the provided coupling clamps.

The hydraulic system has several disadvantages. The most important disadvantages is servicing and maintenance. If a leakage or rupture occurs, liquid discharges to the mid-deck area which can effect shuttle operations. Thus special consideration must be taken into account for containment of the fluid. Also for sudden shock loads, bypass valves may be necessary to prevent vibration transmission or worse, damage to the isolator.

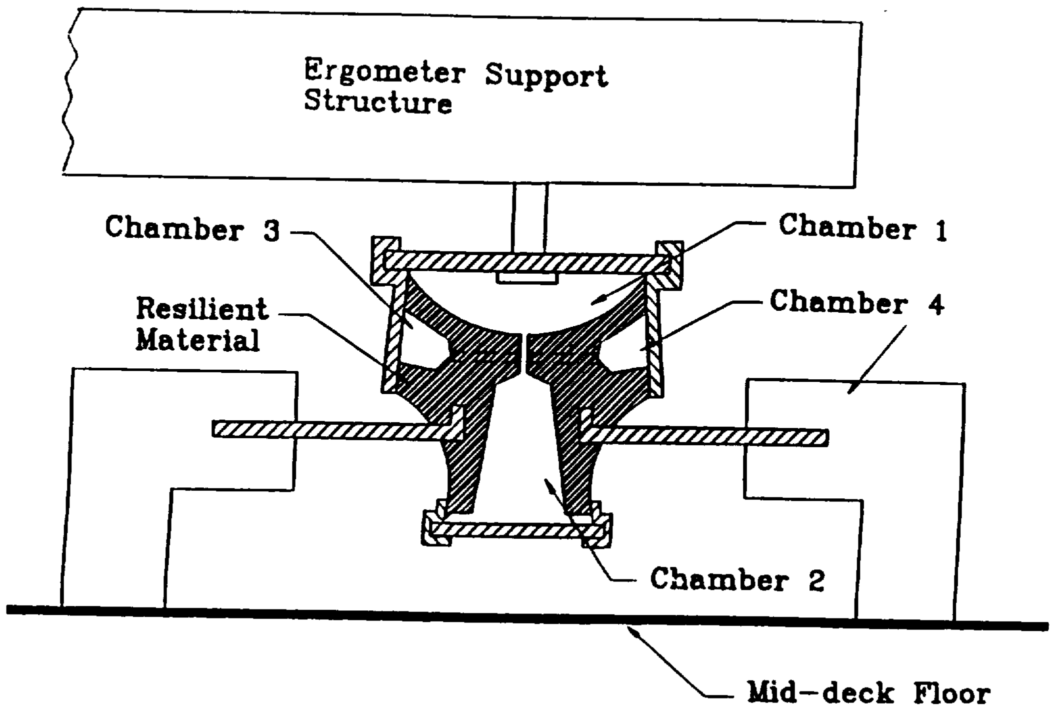


Figure 11. Fluid Bladder Isolation System Cross Section

2.4 Wedge Block Damper

The wedge block alternative is a purely mechanical isolator which relies on friction to perform the damping action. An example of the wedge block system is shown in Figure 12. Vibration transmitters are attached to the four corners of the ergometer support structure. The transmitters are wedged between elastically bound blocks supporting the ergometer. As vibration occurs, the transmitters slide along the contacting surfaces of the blocks. The retaining cords tend to return the transmitter to the centered position of the blocks again damping through friction at return motion. Friction between the transmitters and the blocks during this motion provides vibrational damping.

The wedge block has several advantages. Since the isolation system consist of springs and blocks, the device can be easily assembled. Also, minimal maintenance is required. The wedge block isolator can be stored in one of the provided lockers. Cost of the isolation system is small compared to the other alternatives since the components can be built in-house. A disadvantage of the wedge block is that the springs may need to be replaced after wear. Also in consideration of the specifications, the friction heat may need to be addressed.

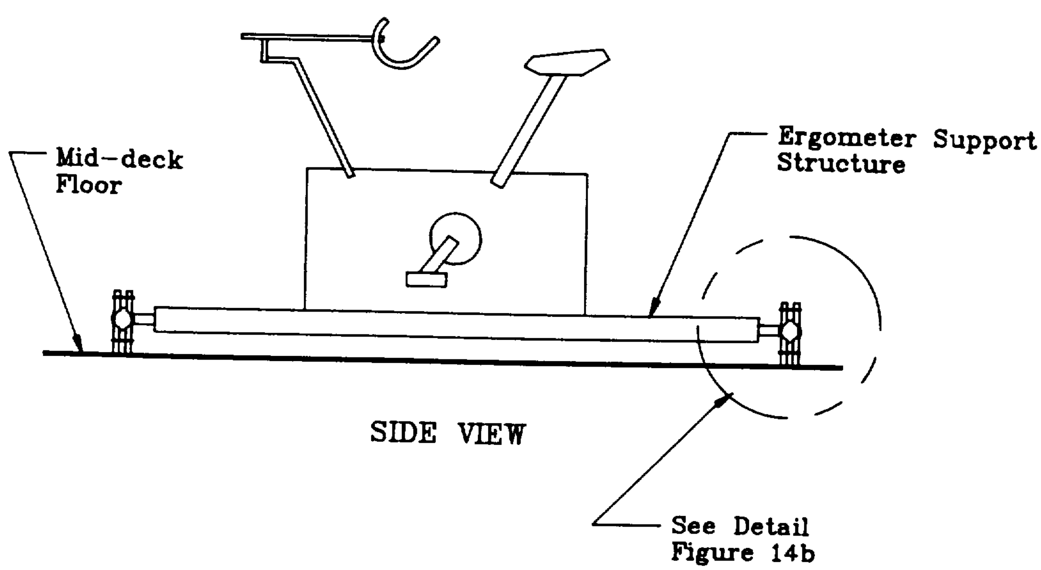


Figure 12. Wedge Block Isolation System

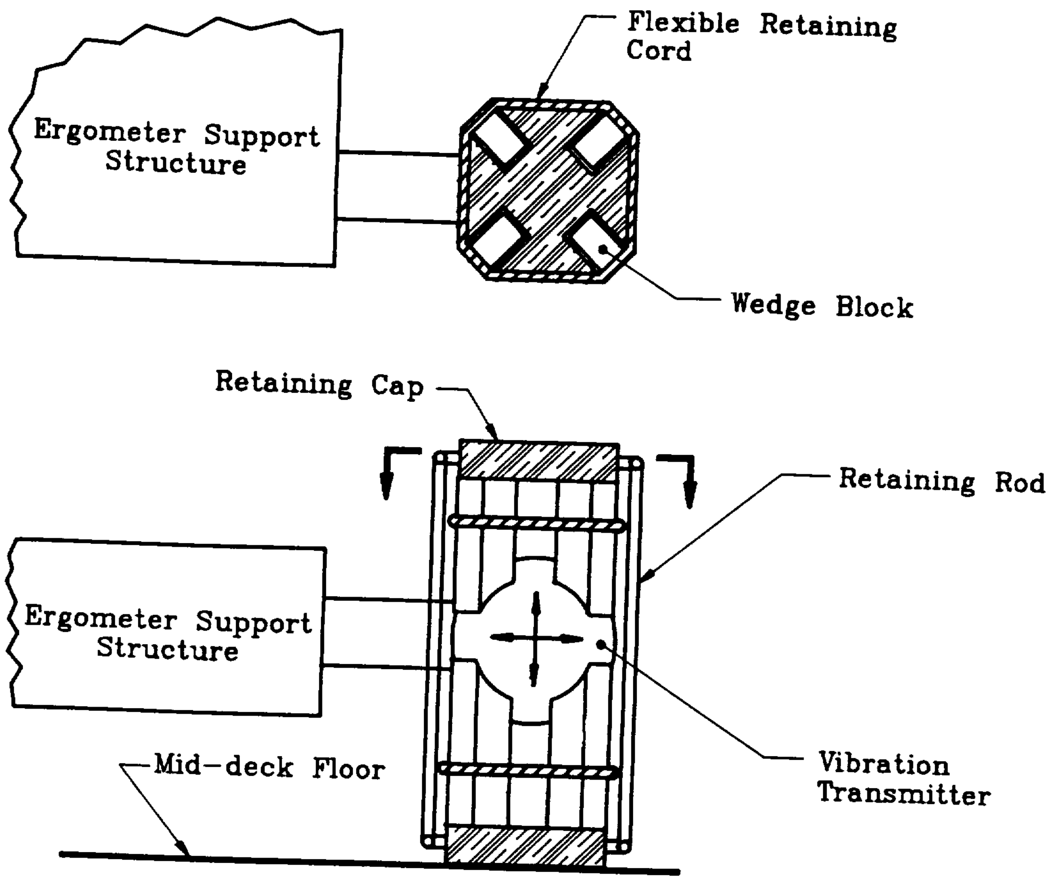


Figure 12a. Wedge Block Detail

2.5 Platus Vibration Isolation System

The design team contacted an engineer who has designed an isolation system to be used for low frequency vibrations [10]. Dr. David L. Platus of Minus K Engineering designed an vibration isolation system that uses negative-stiffness mechanisms to cancel the stiffness of a spring suspension. The system is a passive 6 degree-of-freedom isolator capable of system resonant frequencies as low as 0.2 Hz. The system provides isolation in both the vertical and horizontal motions with basic concepts of springs.

For the vertical motion isolation, the design uses a simple spring to support the weight of the payload and a negative stiffness mechanism to cancel the stiffness of the spring. The negative stiffness mechanism (NSM), consists of two bars hinged at the center, supported at their outer ends on pivots which are free to move horizontally, and loaded in compression by opposing forces P. When unloaded, in Figure 13a, the bars are aligned and in an unstable state of equilibrium. When loaded, in Figure 13b, the center of the hinge is displaced and in a state of equilibrium by F_N which opposes the displacement. For small displacements (δ), a linear relationship exists between F_N and δ expressed by the stiffness K_N. Figure 14 shows a schematic of a vertical motion isolator, designed by Platus, that uses flexures and loading screws. The center position of the isolator is maintained with the uses of translators to raise and lower the springs. This translator can be controlled actively such as a switch or signal.

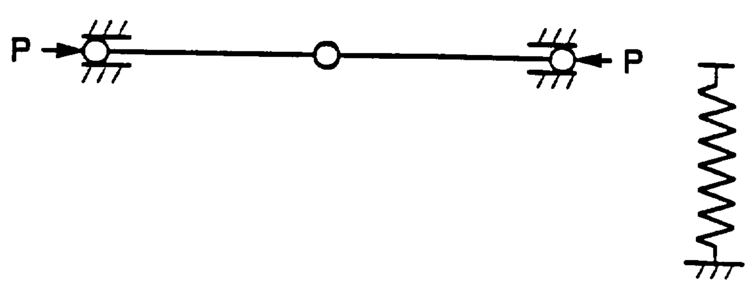


Figure 13a Unloaded Position [10]

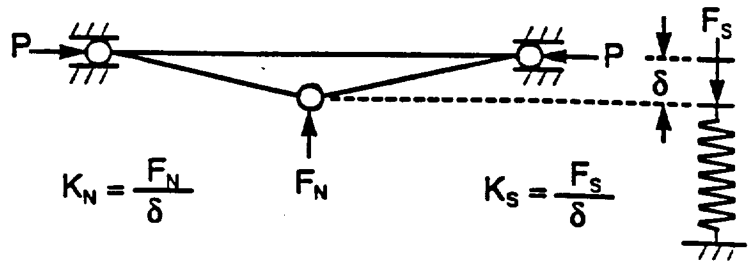


Figure 13b Loaded Position [10]

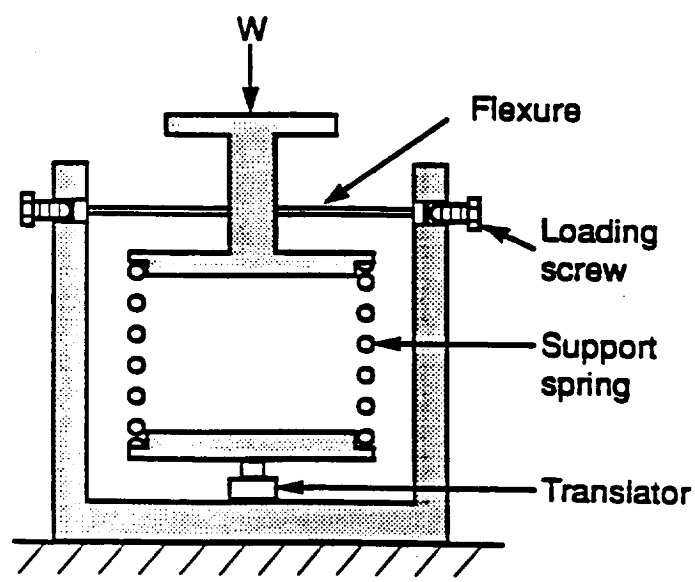


Figure 14 Schematic of Platus Vertical Motion Isolator [10]

Flexible columns that act like springs in combination with NSM's provide for horizontal-motion isolation. The payload displaces without significant rotation because the columns are very stiff in the horizontal direction. If the columns are loaded such that their buckling load is approached, then very low natural frequencies for the horizontal vibrations are obtained. This concept uses the reduction in bending stiffness of a beam column that is axially loaded. In Figure 15, the fixed-free columns act like a cantilever beam with horizontal end load, and acts like a spring with horizontal spring stiffness K_S. With the weight of the payload, bending moments are produced proportional to the deflection δ so that the horizontal force is required to produce δ. Figure 16 shows a payload that is supported by two preload flexible columns with an axial load Q.

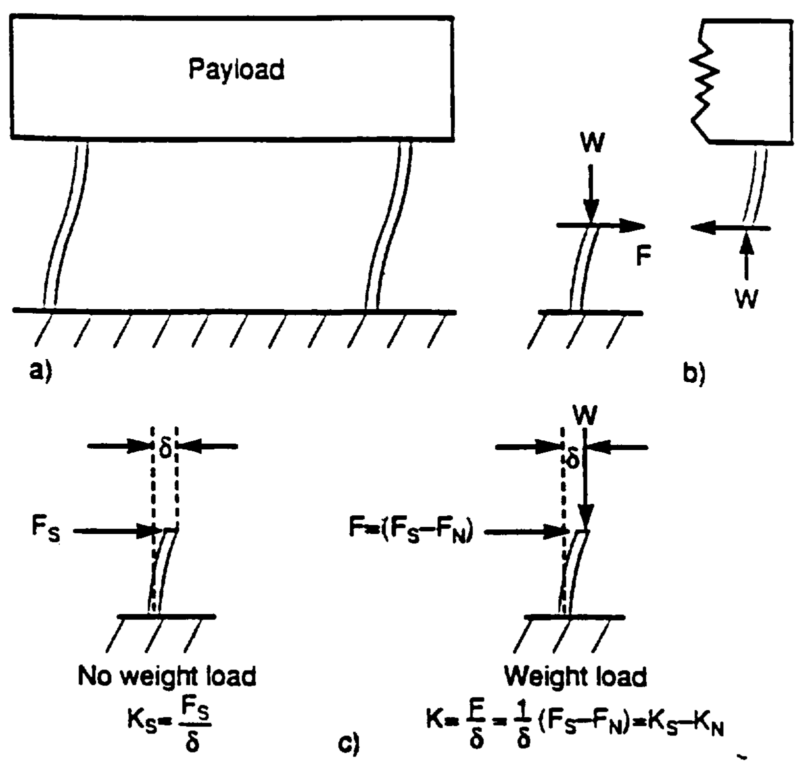


Figure 15 Fixed-Free Columns [10]

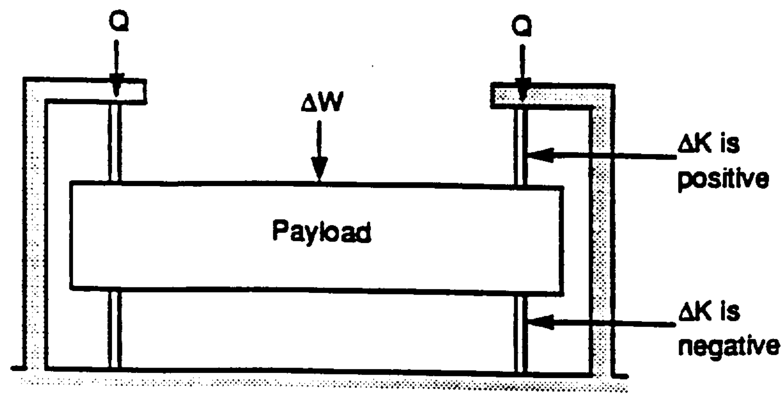


Figure 16 Payload Supported by Flexible Columns [10]

NSM isolation systems have been built and tested that will give resonances as low as 0.2 Hz or lower. Systems tested include passive systems, active and automatic leveling systems, and retrofit devices. These devices have proven to have several advantages. The NSM isolation systems are passive and compact. The systems can support small and large payload. Also, the isolators can provide low resonant frequencies (0.2 Hz) and transmissibilities below 1.4.

The design seems advantageous and applicable, however a feasibility study on the design was not performed. A transmissibility of 0.25 Hz is needed for the isolation system, which the NSM is capable of providing. Also, the system is passive and compact.

The NSM was referred to the team by a vendor. The design team contacted an engineer on feasibility of the other alternative designs. The engineer was familiar with the idea, and he suggested that the team contact David L. Platus, a contract engineer. Platus sent a report explaining the idea of his design. Since the design team received the report toward the end of the semester, a feasibility analysis on the idea was not accomplished. However, the NSM isolation system does seem feasible based on capabilities of the system.

2.6 Air Cushion Dampers

The basic principle of using air to dampen vibrations involves creating an air cushion that elevates the ergometer away from the shuttle floor. The energy needed for the damper is provided by the rider when riding the cycle ergometer or from an external power source. The design team has two air damper concepts: fan isolation system and air bearing.

2.6.1 Fan Isolation System The fan isolation system consists of small fans mounted in the base of the ergometer as shown in Figure 17. When the fans are engaged, a force is imparted to the mid-deck floor and the cycle is elevated on a cushion of air. To prevent the ergometer from floating away, flexible retaining cords are attached to the corners of the ergometer support structure. The fans can be powered by external electric power sources or mechanically from the cycle. Problems may arise when uneven flow from the fans is encountered due to the varying speeds of the cycle.

An advantage to the fan isolation system is the active air cushion. This is important because the system can be adjusted or altered for the impacts and forces provided by riders with different weights. The disadvantages of this system include the requirement of storage space. Also, assembly time is increased because of the many components of the system (i.e. compressor connections, retaining line attachment, etc.). Another important disadvantage is the power requirement and the potential high mass of the system due to air pumps, fan motors and the like. Minimal mass must be maintained for the shuttle.

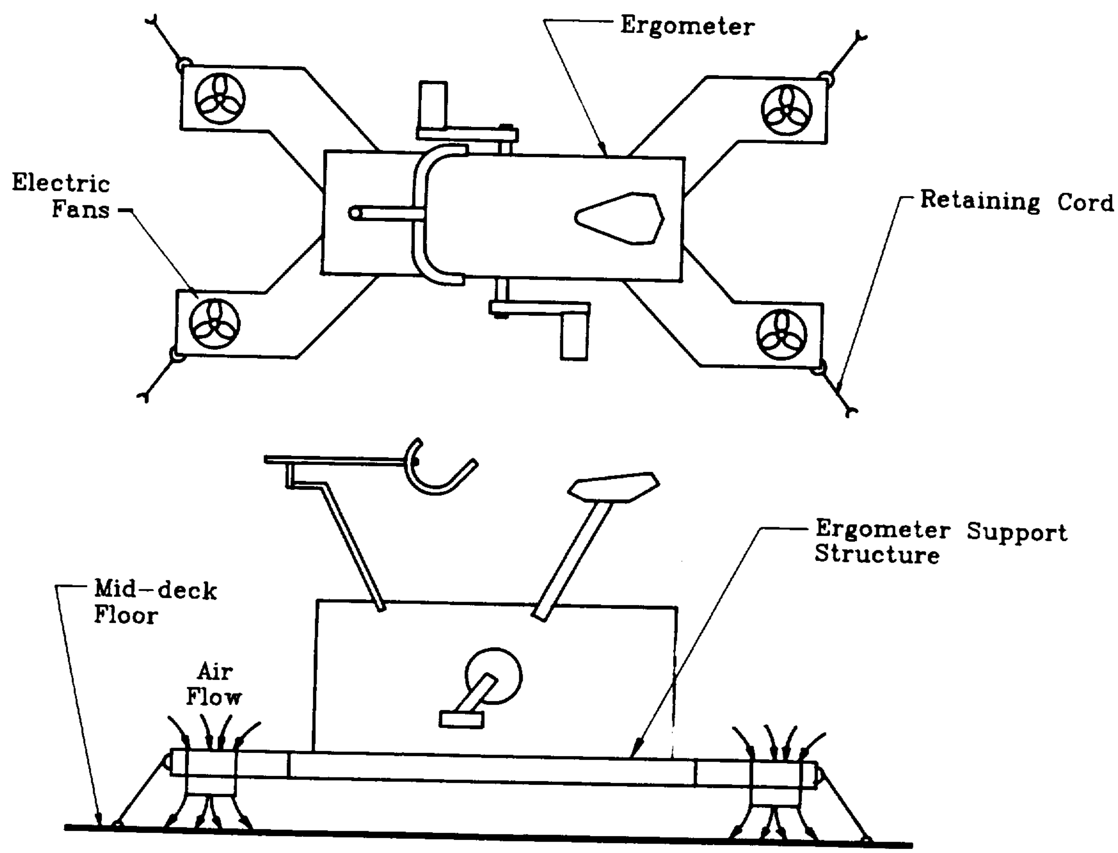


Figure 17. Fan Isolation System

2.6.2 Air Bearing Another method using air as the damping medium, is an air bearing, shown in Figure 18. This system consists of a well attached to the mid-deck floor using the provided coupling clamps. Inside the well sits a cylindrical air plug that has holes positioned radially and at the bottom surface for an even airflow. An air line is run from a compressor, through the base of the ergometer and to the air plug. As the person rides the ergometer, the compressor sends air through the plug and a localized air cushion is at each corner of the ergometer support structure. Again, to anchor the ergometer, flexible retaining cords are attached at the corners of the ergometer support structure.

Similar to the advantages for the fan isolation system, the air bearing isolator can be adjusted for different riders. The damping can be altered by controlling the air flow.

The air bearing system has several disadvantages. External power is necessary to supply the air, which adds more mass and extraneous components to the shuttle. Another disadvantage is the system components may be difficult to stow in the provided lockers. Also, the complexity of the air bearing system does not lend itself to quick installation during orbit .

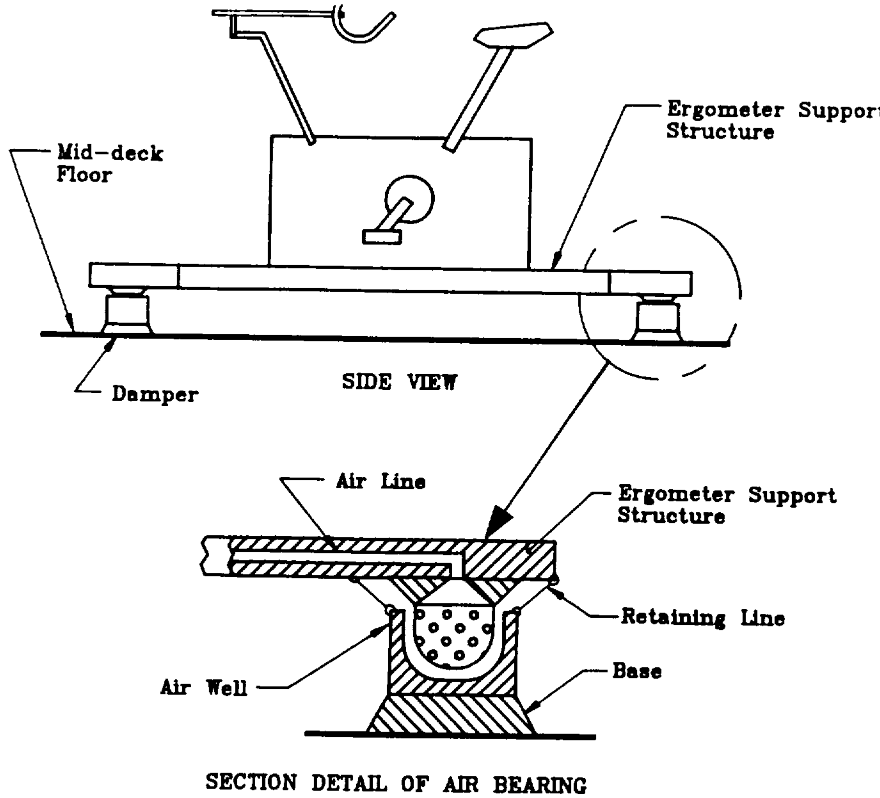


Figure 18. Air Bearing Isolation System

2.7 Electromagnet Support

In Figure 19, the top electromagnet is fixed to the ergometer while the bottom electromagnet is attached to the mid-deck floor. If the electromagnetic fields are activated, the magnets will repel. This repelling action is limited by the retaining lines running through the center of the electromagnets. As vibration is encountered, a transducer, such as a linear variable-differential transformer (LVDT), senses the displacement created by the vibration. The transducer sends a signal to a control system varying the current to the electromagnets. The change in current alters the magnetic field moving the ergometer in the direction to compensate for the displacement of the vibrations.

An advantage to the electromagnet isolation system is that the system is an active control system. The isolation system allows a wide range of dampening ability. If the rider inputs a large impact load to the ergometer, the vibration between the ergometer and the mid-deck floor will be minimized [6].

An important disadvantage to this isolation system is that the damper requires electrical energy. The current and voltage required for the isolator must meet specifications required by NASA. Also, the isolation system requires setup time for the positioning of the rider and setup time of the control system to obtain the maximum capabilities of the system. An additional disadvantage of the system is potentially poor damping characteristics in horizontal and angular directions [6].

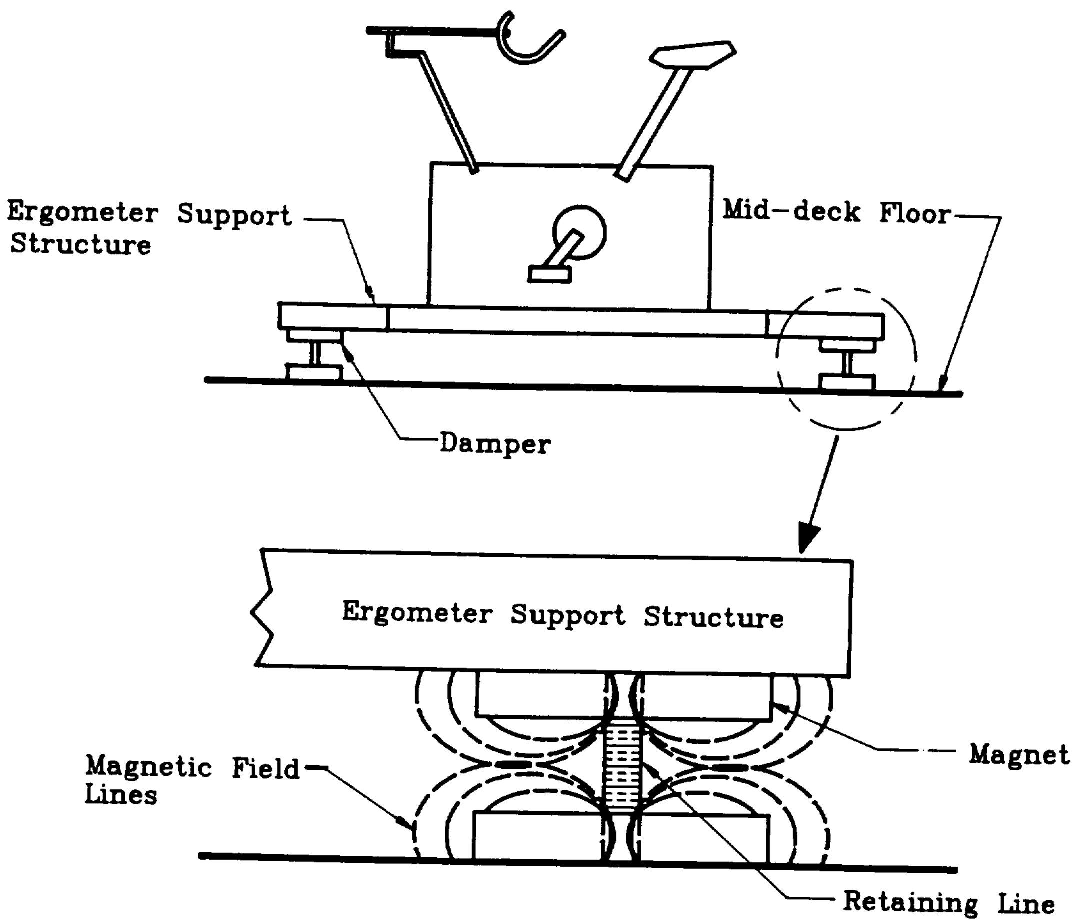


Figure 19. Electromagnet Isolation System

Evaluation of Alternative Designs

3.1 Preliminary Decision Matrix

The alternative designs were initially evaluated using an approach based on engineering knowledge and intuition of the design team. The approach used a preliminary decision matrix using a design criteria. Weighting factors were assigned to each of the criteria. These factors were developed using the method of pairs (see Appendix D1). This method involves systematically comparing each design criterion against other criteria. Each time a design criterion was considered to be better than another, that criterion received a tally mark. After all of the possible permutations of the criteria were evaluated, the tallies awarded for each criteria were summed and then divided by the total tallies for all of the criteria. The resulting number was the weighting factor.

The decision matrix, in Appendix D2, enabled the design team to select the most feasible alternatives. For instance, the wedge block was eliminated from the decision process because the design team thought that the damping due to friction would not be sufficient to isolate the ergometer. Additionally, the team believed that the wedge block alternative would not be as reliable as other alternatives because of the complex nature of the structure. The alternatives that were eliminated using this approach were not proven to be infeasible, but the process was based on engineering ability. The chosen alternatives for the feasibility analysis were resilient absorbers, isolation box, hydraulic damper, air bearing system, and electromagnetic systems.

The following section defines the parameters used for the feasibility analysis and the decision matrix:

Mass - This parameter is important because of the weight and space constraints of the space shuttle. Mass must be limited to allow for other important equipment to be stored onboard.

Energy Domains - Each alternative's energy domain is a function of the damping process. For example, the wedge block damper uses friction effects to dampen whereas the electromagnetic damper uses controlled magnetic fields to create the damping effect. This parameter was used to rate the relative complexity of the energy domain to perform damping from the standpoint that less complex domains are preferred (see Appendix D2).

Space Constraints - The space on the mid-deck occupied by the isolation system must be minimized. This is important so that daily activities and mobility of the astronauts will not be hampered.

In-orbit Assembly - The astronaut's time is valuable during orbit. The isolation system should provide for easy assembly so time is not wasted to complicated equipment set-up.

Stability - This parameter refers to the ability for the isolation system to remain stable during operation. The safety of the astronauts and reliability of the system are functions of stability.

Multidirectional - This parameter is critical to the design solution. The isolation system must be able to isolate vibrations in the x, y, and z directions. This parameter rates the ability of an alternative to dampen in these directions.

Storage - Since the space shuttle has limited storage space, the alternatives can be rated on the extent to which they are able to be stored in this space.

Time to Produce - This parameter determines the difficulty of the isolator to be manufactured. Considerations of the manufacturability include the complexity of machine work.

Number of Parts - The reliability of the system is a function of the number of parts. As the number of parts increases, the more complex the system becomes increasing the probability of failure of the system.

Wear- This parameter is important because the isolation system will be subject to extensive use. An important design consideration is how well the design responds to repeated usage.

Cost - Though cost is not a critical design constraint, it is still an important design criteria to the development of the alternative.

Material - This parameter rates the alternatives on whether special materials need to be implemented into the isolation system. This parameter is important because it is related to the manufacturability and the cost of the design.

Active or Passive - This parameter defines the complexity of the design. The continual interaction of the control system and the isolation system can lead to highly complex systems with regard to external energy sources and highly sensitive controlling equipment.

3.2 Feasibility Study of Alternative Designs

To determine the feasibility of the isolators, the team performed various preliminary calculations. Included in these calculations are the transmissibility ratios of a isolation system. The transmissibility ratio is output of the isolation system over the input. In this case, the input are forces imparted by the rider. Used in the feasibility analysis are calculations that pertain to specific type of damping system. For example, the fluid bladder must meet pipe flow requirements to sustain the movement of the fluid between the chambers. The following sections discusses the assumptions and results of the feasibility analysis for the chosen alternative designs.

3.2.1 Resilient Absorbers One of the conceptual ideas implemented in the alternative designs is the use of resilient absorbers. The application of resilient materials in the practice of vibration isolation is very common. This can be seen from the many configurations of resilient absorbers on the market, such as motor mounts and industrial equipment [9].

Mechanics of Resilient Materials The principle reason resilient materials are used for vibration isolation is the high mechanical damping inherent in the material. This can be exemplified when comparing a plastic beaker and a crystal glass after being struck. A plastic beaker emits a dull note of short duration while the crystal glass emits a long ringing note. The inherent mechanical damping of resilient materials is a manifestation of the viscoelasticity of the material. The viscoelasticity of polymeric materials is characterized by the fact that the solid is viscous in that it will creep. Also, polymeric materials are elastic in that it will recover after being strained [12].

The design team developed four alternatives using resilient materials: namely, the intermediate damper, the coupling absorber, the ball and socket absorber, and the x-support structure.

Feasibility Development To develop the feasibility of these absorbers, the design team focused on the simplest case.. The coupling absorber was the simplest means of implementing resilient absorbers because this absorber can be thought of as a resilient pad sandwiched between two plates. Beginning with a simplified system, the design team can develop its feasibility and use this as building a block to develop the more complicated alternatives such as the ball and socket absorber.

For feasibility purposes, the intermediate absorber was further simplified by considering a single degree-of-freedom system rather than the required six degree-of-freedom system. Once the feasibility of a one DOF system is determined, the analysis could then be extended to six DOF.

To begin performing the feasibility on the one DOF system, the force-time histories of ergometer exercise were analyzed to determine which direction (x, y, or z) was experiencing the greatest force magnitude. It was found that the forces in the z-direction were exerting force magnitudes of approximately 445 N (100 lbs) summed over the four mounting points (see Figure 20). Therefore, the one DOF system using the coupling absorber was considered for the z-direction.

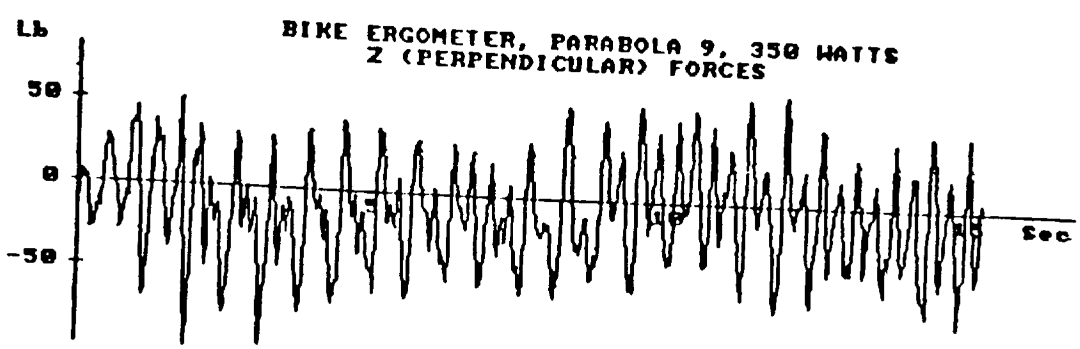


Figure 20. Force Time History of Z-Direction Showing Maximum Forces.

The simplified system considered is often referred to as a bonded rubber spring. These systems are commonplace when considering vibration isolation. For example, motor mounts are often made of rubber springs. After conducting a literature search, it was determined that there is an abundant amount of information concerning the response of resilient materials to vibration. Much of this information is in the form of response curves or transmissibility of the material.

By examining the response curves of rubber springs, it was determined that resilient materials behave as low pass filters. This is to say that low frequency vibrations are passed while high frequency vibrations are attenuated. A typical response curve for natural rubber is shown in Figure 21.

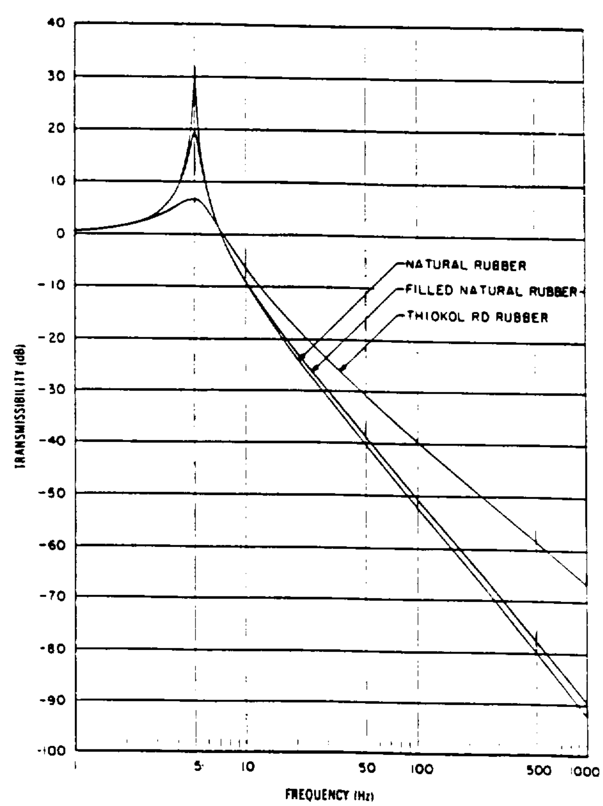


Figure 21. Transmissibility of Natural Rubber [23].

The response for resilient materials is a second order system with resonance occurring at the natural frequency of the material. The resonance of the system is shown by the spike in transmissibility on the response curve.

Feasibility By examining data produced by tests of the cycle ergometer in a zero-g environment, the problem vibrations were found to be occurring from approximately 1-5 Hz. Since resilient materials act as low pass filters, the problem arises as to whether the resonant frequency can be lowered enough to allow for attenuation at 1 Hz.

The analysis, in Appendix E, was performed by assuming that the isolation system is required to have a natural frequency of approximately 0.5 Hz. By choosing a very low natural frequency, the attenuation frequencies are also lowered.

The natural frequency of a resilient material is dependent on the stiffness of the material [11]. For resilient materials the stiffness is not simply expressed as a spring constant, k, as it would be for a material such as steel. The dynamic stiffness of resilient material is expressed as a dynamic stiffness with a real and imaginary term in Equation (1)

$$K^{*} = K(1+i\eta)] \qquad (1)$$

The first term, K, represents the stiffness of the material. The second term, the imaginary part, represents the damping of the system. This is a result of the viscoelasticity of the material. The complex stiffness can be related to the complex compression modulus given by Equation (2).

$$E^* = E(1+i\eta) \tag{2}$$

The complex compression modulus is a function of frequency. As frequency increases the complex modulus increases. The relation between the stiffness and the complex compression modulus is given by Equation (3)

$$K^* = E^*S/L \tag{3}$$

where S is a shape factor associated with the applied load area and L is the thickness of the material. From these equations, the natural frequency can be found for a the system being considered in terms of the thickness of the material.

For the feasibility analysis of resilient absorbers the thickness to obtain a natural frequency of 0.5 Hz was determined. This was done by determining the complex modulus at this frequency, assuming an appropriate load area for the system, and then backing out the thickness of the material.

Conclusions By performing the analysis described above, it was determined that the dimensions needed to achieve a natural frequency of 0.5 Hz were very large. For example, the thickness is approximately five times as large as the load area. A load area of two square inches would require a thickness of approximately ten inches. Though it is possible to achieve the required natural frequencies, because of space constraints and the need to minimize mass on the space shuttle this approach to vibration isolation was deemed impractical.

3.2.2 Isolation Box Another system chosen for development was the isolation box. After evaluating the system including the ergometer cycle and the isolation box, the team believed the best approach to the problem was to (1) find the equations of motion, (2) find the transfer function of the system and (3) determine the frequency response of the system.

Model of the System The first step was to model the system (see Appendix F for details). The free body diagram, given in Figure 22, of the system includes the rider, ergometer structure, and the isolation system located relative to a global reference frame. The rider, a rigid body, has horizontal, vertical and rotational movement. The ergometer structure, another rigid body in the system, also has movement in these directions. The spring-damper systems attached to the ergometer base also have x, y, and z positions. The systems have separate coordinates since they respond and act relative to each other.

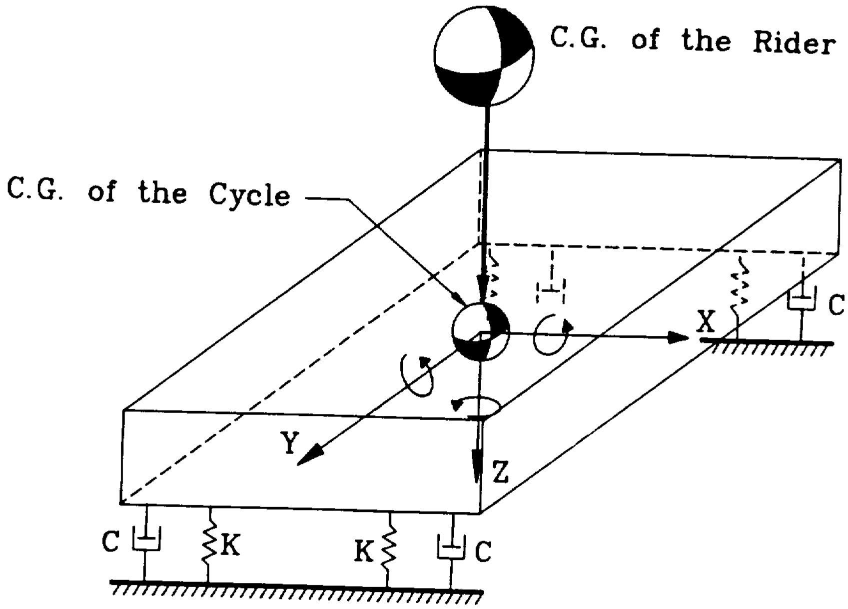


Figure 22. A Model of Exercise Cycle Including the Rider and Isolation System.

The dynamic model is a six degree-of-freedom three dimensional system. The isolation system at the corners of the ergometer structure consist of 12 coordinates at the corners of the structure which move relative to each other and the global reference frame. The rider has six coordinates because motion is allowed in the x, y, and z positions and yaw, pitch, and roll orientations.

Equations of Motion Equations of motion can be found for this system. It involves location the center of mass of the individual components relative to each other and the global reference frame. If the mass of the rider and the forces that the rider exerts when in motion is known, the inertia of the rider can be found. These variables, which are constantly changing as the position of the rider changes, must be related to the global reference frame of the system. Also, the positions of the spring-damper systems are constantly displaced relative to the displacement of the ergometer structure and to each other. This displacement is located relative to the position of the rider and global reference frame of the system.

Simulation of Model Once the equations of motion are known, the frequency response can be determined. The design team believes this problem can be solved with simulation of the dynamic model [13]. This approach would be the simplest because the matrices obtained for components in the system are complicated. The major elements of the model consist of two rigid bodies which represent the rider and the ergometer structure. The major compliant elements are the springs attached to the corners of the ergometer structure. Dissipative elements or dampers are also attached to the four corners of the structure. Each component of the free body diagram is represented as a dynamic model element.

Due to the complexity of the system, the design team did not use this free body diagram to model the system. However, a simple two dimensional rigid-body vibrational model was formed, shown in Figure 23. Since the input data given by NASA was taken from the ergometer structure, the rider and ergometer structure could be taken as a lumped mass. For simplicity, the rider and structure is represented as a rigid slab of total mass m with the mass center C at distances c and d from the springs and a and b from the dampers. The body has mass moment of inertia about the center C.

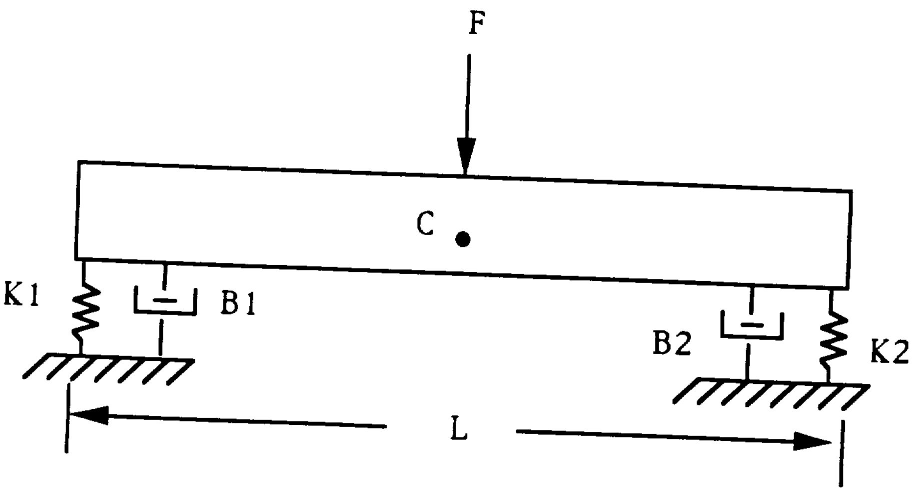


Figure 23. Two Dimensional Model of the Isolation Box

Figure 24 shows a free body diagram corresponding to the body in a displaced position, where the displacement consist of the vertical translation x(t) of the center C and rotation q about C.

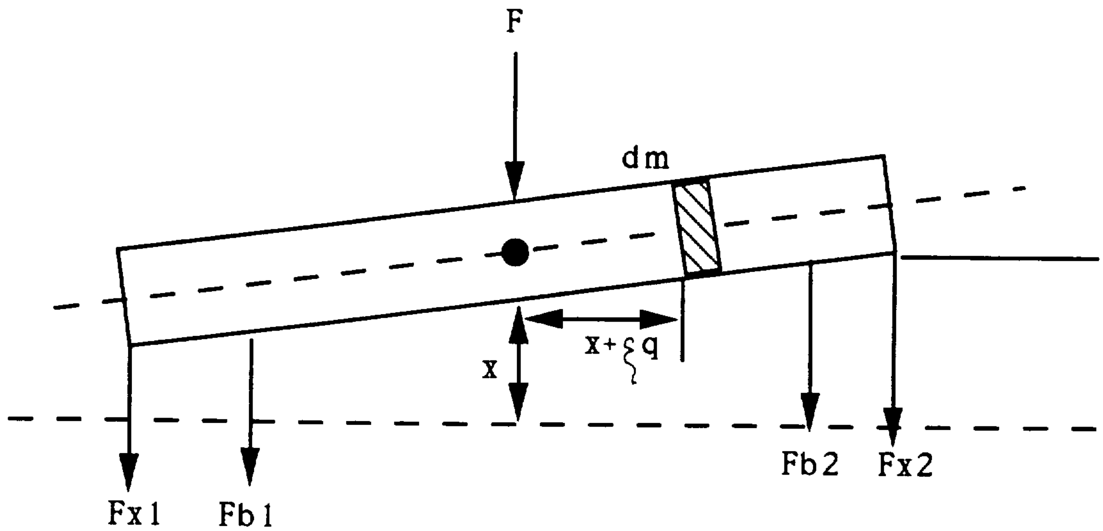


Figure 24. Displaced Position of the Two Dimensional Model of Isolation Box

There are two equations of motion, force equation for the translation in the vertical direction and a moment equation for the rotation about the mass center C [14]. Small angle approximations were assumed since the springs displace more in the y direction and not the x direction. To develop the equations, the team considered a differential element of mass *dm* at a distance from the center C. For small angles of acceleration q, the mass element is in the vertical distance and equal to $x + \zeta q$. The equations of motion are given in Appendix F.

From the equations, the team determined the transfer function for the system

Of course, this model will not represent the actual system, however the system is a good first approximation [15]. The team believed that if the system is capable of damping out 5 Hz, it is probable that the actual dynamic model is feasible.

3.2.3 Fluid Bladder Damper Another concept to dampen the vibration of the exercise cycle is the fluid bladder. The fluid bladder was found to be useful to dampen vibrations in industry. The primary application of the hydraulic damper is for engine mounts for automobiles and light trucks [13]. The damper has three tasks depending on the frequency and amplitude of the excitation. The damper can act as a spring providing basic load and motion capability to the system. Also, the damper provides restricted motion at or near the resonant conditions. The damper also acts as a tuned absorber providing isolation at a specified frequency. The team believed that for these reasons, the fluid bladder can be applied to the vibration isolation needed for the cycle ergometer.

The mounts are tuned to match precisely most application requirements better than conventional rubber mounts. The damper allows the use of a softer mount for better isolation of vibration and uses internal fluid damping to reduce motion at resonant conditions.

To determine if the fluid bladder is feasible for the application, a model of the system must be developed to determine the transfer function. The model can then be used to form equations of the system using state variables. State variables are needed to define the system response at any time, t. With the system of equations, the transfer function can be determined.

The team determined the model of the fluid bladder with the use of bond graphs [18]. Bond Graphs depict the energy flow and storage in physical systems using basic elements and bonds. The graphs use elements that store energy, transform energy, dissipate energy, and join or add sub-systems. There are three ideal energic elements: storage of potential energy (capacitor), storage of kinetic energy (inductor), and dissipation of energy (resistor). The elements which represents the physical system are connected with ports. A port is a place at which power can flow between subsystems. The ports are connected to other basic elements with the use of junctions. There are two types of junctions: 0 - junction representing the common effort and 1-junction representing the common flow. Effort variables refer to "pushing" such as forces and torques. Flow variables refer to the "motion" such as linear and angular velocity. The bond graph model is used to characteristically represent the physical system.

The team modeled the fluid bladder with the use of a bond graph, shown in Figure 25 and Figure 26. Using the prescribed concepts of a bond graph, the force provided by the rider is an input or effort in the damper model. The force supplied by the rider is transformed into a pressure of the first chamber. The pressure causes expansion of the chambers and stores potential energy (C1) of the fluid. When the pressure drops in the chamber, the potential energy in the chamber is reduced as the resilient material causes the fluid to be forced through the orifice. The orifice acts as a resistor because the fluid will only transmit through the chambers when pressure of the chambers is decreased and the resilient material compresses. The second chamber is similar to the first chamber in that potential energy (C2) is stored. The resilient material surrounding the two chambers adds mass or inertance and capacitance to the system.

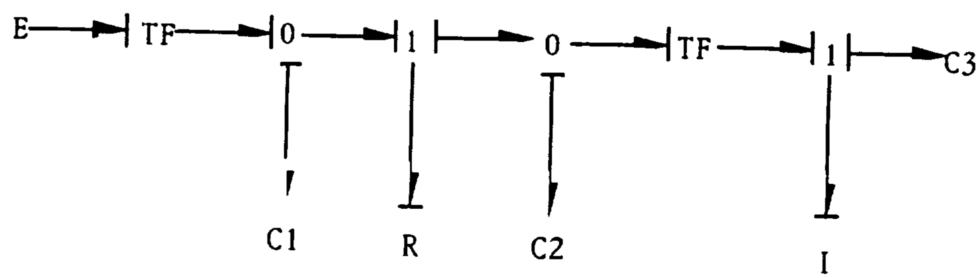


Figure 25. Bond Graph Model of the Fluid Bladder

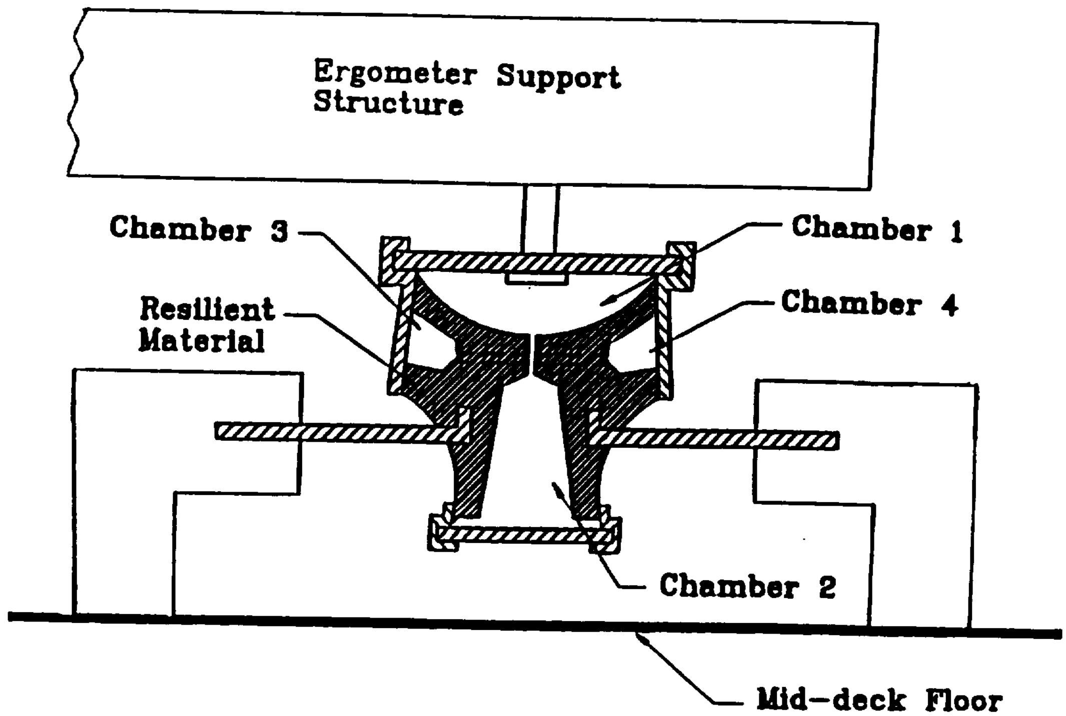


Figure 26. Cross Section of Fluid Bladder Damper

With this model, the team found the systems of equations. The systems of equations can then be used to determine the response of the system by knowing the transfer functions [18]. Generally, n-th order systems can be written as

$$y = [\mathbf{A}]\ y + [\mathbf{B}]\ u$$

where

y - column vectors of system variables

[**A**] - matrix whose elements depend on the system

[**B**] - matrix whose elements depend on the forcing function

u - column vector of the forcing function

With manipulation of the n-first order equations with n-unknowns, the equations can be formulated into one first order equation with one unknown. Since the n-th order equations are linear, the output of the system is assumed to be in the same form and the input. We then are able to find the transfer function, ratio of the output and input of the system.

The state equations and transfer function are given in Appendix G. To find the dimensions and parameters of the fluid bladder, values were assumed to determine the output of the system. These values were determined using a marketed fluid bladder with the specifications and requirements of the cycle ergometer. A transfer function of the fluid bladder was found to be 22.8 Hz, which is not even close to the low frequencies needed.

The fluid bladder is not a feasible solution to the problem. The marketed fluid bladders will dampen vibrations at frequency as low as 14 Hz and not the desired 1 to 5 Hz needed for the cycle ergometer. The team thought that if the material of the resilient absorber was changed so that when the chambers expand, the deformation of the resilient material increases. However, the material would need to be very thick (greater than 10 in.) and could not be used due to the space constraints of the project.

3.2.4 Active Air Bearing Feasibility The air bearing isolation system shown earlier in Figure 12 features an air plug which rests within an air well. To perform vibration isolation, it is proposed that compressed air be fed through the air plug within the air well. In concept, vibration transmission is reduced by controlling the cushion formed between the air well and the air plug. Based on the geometry of the air plug and air well, the original concept is designed to isolate in six degrees-of-freedom. However, after reviewing literature on air suspension systems it is clear that to fully describe this system requires analysis beyond the time frame of this project. Therefore a simpler representation of the system is analyzed.

Problem Simplification The proposed air bearing concept calls for a system that will develop a cushion capable of damping input forces from the ergometer. The geometry of the cushion is such that the input force was damped regardless of the direction the force originated from. Also, the cushion was to be actively controlled to accommodate a wide range of loads.

Based on the research of the team, two areas of simplification are made to better understand the feasibility. The first simplification involves the cushion geometry and the second involves active control analysis.

Cushion Simplification To perform feasibility on the air bearing, a model of the air cushion system is needed to understand the parameters of operation for the system. The simple model chosen to clarify the system parameters is a rigid plenum [16]. The simple plenum consists of a rigid wall chamber which floats on a cushion developed between the chamber and the ground surface (see Figure 27). The arrows indicate the airflow directions during operation. The model also assumes that forces and displacements are limited to one degree of freedom normal to the ground.

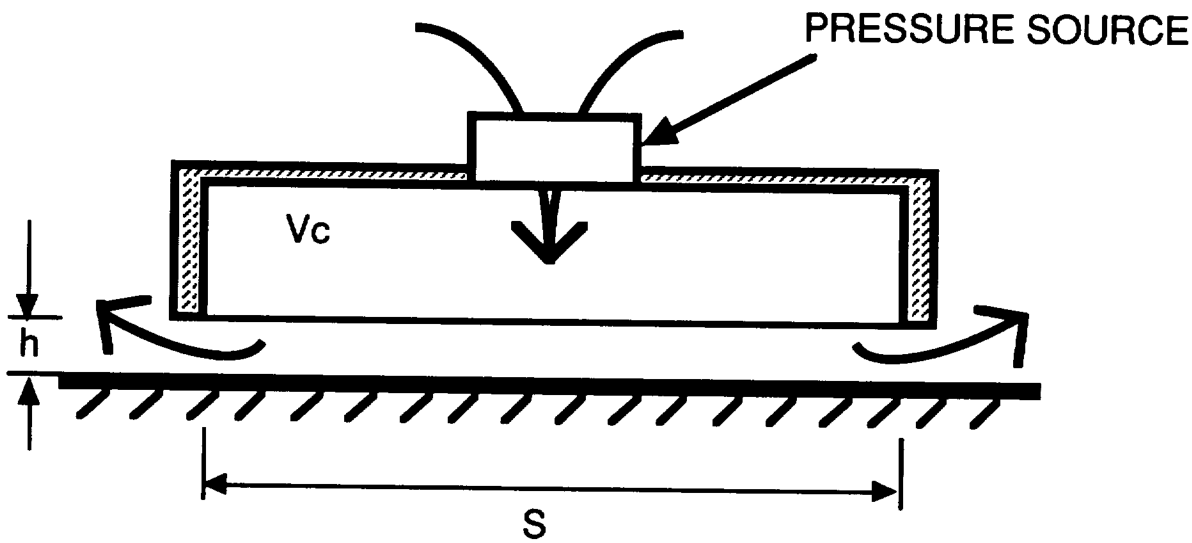


Figure 27. Simple Rigid Air Plenum Model

Active Control Simplification Due to the varying force developed during exercise an active control system is proposed. After some investigation of active control systems it is clear that a wide variety of controlling schemes exist. In the interest of determining the relative feasibility of the air bearing concept, the team will present simplified control principles based on the research of air cushion systems. These simplified principles include the types of control damping algorithms used and the types of components necessary such as valves and sensors.

Fundamentals of Operation Two areas of interest are discussed below. The first area discussed is air cushion operation followed by active control operation. The objective is to present fundamentals of operation that can be extended to apply to the air bearing concept as originally proposed. A second goal is to provide a foundation for future analysis of this alternative.

Air Cushion Operation The basis of the air bearing system is the formation of an air cushion between the ergometer and the orbiter floor. As mentioned earlier, the air bearing system is simplified by modelling it as a rigid air plenum (see Figure 27) with forces and displacements acting in one direction. With this assumption, the following parameters are important in the development of the air cushion: P, the pressure developed underneath the plenum, h, the height between the lip of the plenum and the ground, and S, the bottom area of the plenum over the surface. All three of these parameters must be considered when designing for a given load. In the simple steady state case, equilibrium is achieved when h reaches an upper limit at which point, for a constant P, the plenum will be balanced above the surface. It can be seen then, that an increase in h requires an increase in P. Variations to S result in changes to the required pressure source. An increase in S results in a decrease in the required pressure source for a given h. Similarly for a decrease in S, the required pressure source increases.

To assess feasibility, the performance of the damping action must be determined. The literature review of air cushion systems shows that most of the published information deals with the performance of air cushion vehicles or Hovercraft. Cushions as proposed for the 6 degree-of-freedom damper will require additional analysis since no model was found for these type of cushion geometries.

Active Control System Operation Since the ergometer forces vary with time, an active control system is proposed to vary the pressure of the cushion between the ergometer and the shuttle floor. A literature review revealed two possible methods of providing varying cushions. The first type of control method is the type used to reduce the heave effects of air cushion vehicles [16] The control of these vehicles is based on input signals which pass through a controller which regulates the position of plenum vent valves. By venting the plenum, the pressure developed can be regulated and in turn the cushion is controlled.

The second control method proposed is an extension of a payload isolation system developed for the Johnson Space Center [17]. This system used controlled air jets to blow on a free floating payload. This system however was designed to isolate a vibration free payload from external vibrations. Because the ergometer develops vibrations and the force levels are much more significant, the application of these concepts will require further analysis.

Feasibility Conclusions for the Air Bearing The air bearing system was not chosen as the design solution based on the amount of additional analysis required to realize the design. Detailed modeling of the air cushion is required for the 6 degree of freedom isolation geometry. In addition, a pressure source (compressor) along with routes for air distribution are required to form cushions at each of the four corners. Future work is also needed to properly resolve the control issue. This requires evaluation of the optimum high-pass controlling algorithm along with selection of motion transducers for a six degree of freedom system. In summary, although the air bearing isolator will not be developed based on the amount of required analysis time, the information given here addresses some of the more important aspects of the air bearing isolator and should provide a good base for future work.

3.2.5 Electromagnet Isolator Feasibility The use of electromagnets to reduce the vibrations transmitted by the ergometer to the floor was originally conceived as a system using two electromagnets, one mounted to the ergometer the other mounted to the floor. Vibration reduction is performed by varying the opposing force between the floor and ergometer. This section will describe the results of feasibility research on active electromagnetic suspension systems. The results are grouped into two areas of concern, magnetic field generation and active control design. Based on these results a discussion of the overall feasibility will be discussed.

Magnetic Field Generation To perform the vibration isolation, the electromagnet isolator must create forces that counteract those created during exercise. This section will deal with design issues based on literature reviews and discussions with magnetic field technology researchers.

The proposed isolator used two opposing electromagnets to repel the ergometer away from the floor. The distance between the two electromagnets was fixed to some maximum value to prevent the ergometer from being pushed away and to keep the fields in proper position for interaction. Using densely wound wire proves to be the most effective means of creating strong magnetic fields [19]. In addition, the field strength can be greatly increased by using a strong permanent magnet as the core for the wire windings. Research also reveals that the strength of the magnetic field is also a function of the winding geometry and current. For maximum opposing force, the windings should be oriented so that the magnetic flux interaction between the two electromagnets is maximized (see Figure 28). Also as the current increases in a given winding the strength of the magnetic field increases.

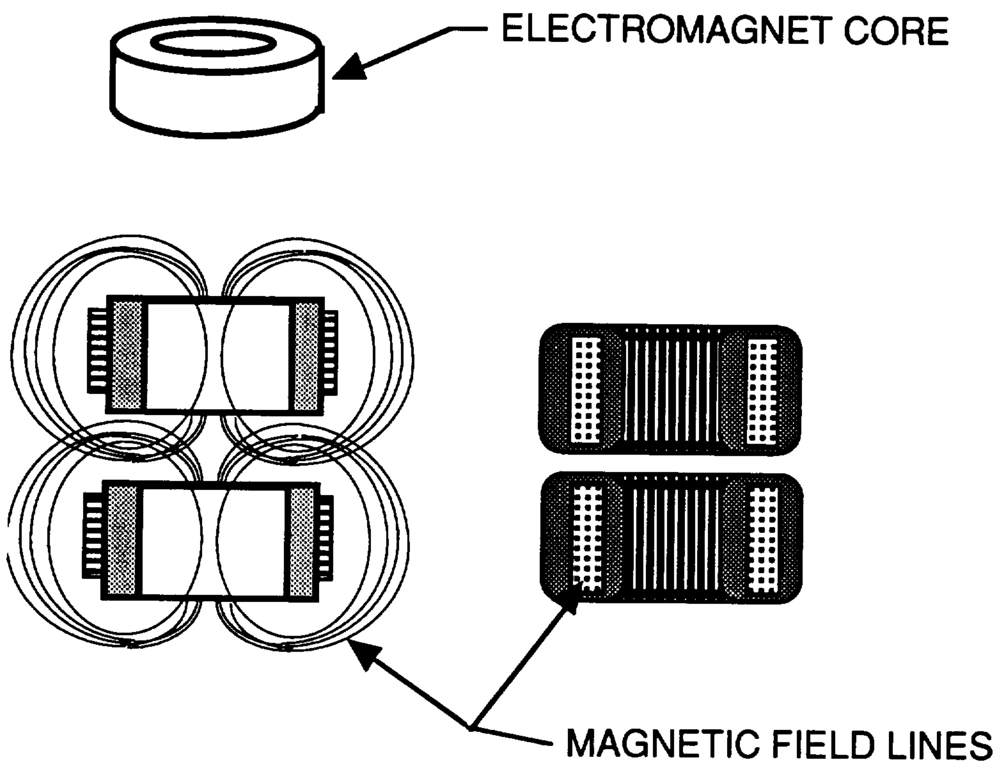


Figure 28 Cross Section of Cores and the Interaction of Fields Generated by External (Left), and Torroidal (Right) Windings

However, for increased currents, the temperature of the windings becomes an important design issue. Appendix H gives additional information dealing with electrical current and temperature limits onboard the shuttle.

In addition to the issues of power and temperature specifications is an effect that will be refereed to as "repulsion reversal." This effect, in short, is the behavior of two opposing magnets that are displaced into a position where the fields no longer repel each other and begin to attract each other. Figure 29 shows this effect for a simple bar magnet.

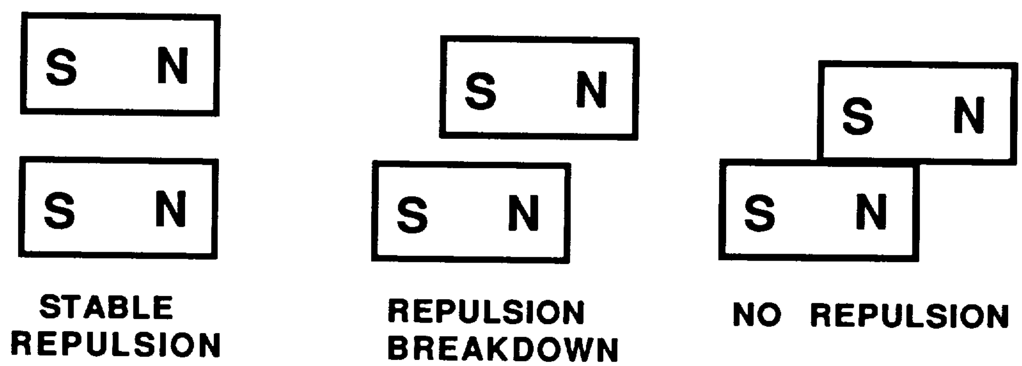


Figure 29 Repulsion Reversal Effect

Given the six degree of freedom ergometer system, the repulsion reversal effect is a design issue that must be resolved by study of different electromagnet positions and their field interaction during operation. In the current concept, the dampers are located at each of the corners of the ergometer base. For planar translation of the ergometer base (i.e. parallel to the plane of the floor) the effects of repulsion reversal is seen as a potential problem.

Active Control Design To produce time varying forces to counteract those from the ergometer, active control of the magnetic field is required. Magnetic field control is developed through the control of current passing through the electromagnet [20]. A wide variety of works dealing with active controls are available and in lieu of design time constraints, the team will present only a brief summary of two control issues that appear relevant to the ergometer vibration problem. They are: loop stability problems [21], and sensor configurations.

Given that a system can be described accurately by a transfer function, a control system can be developed in terms of feedback variable gains and control variable gains [20]. Different forms of these gains (proportional, integral or derivative) appear to have important effects on the stability of the system. Further study is possible to determine the best form of these feedback/control loop gains to ensure stability of the ergometer system.

The sensor configuration issue is important to the cycle ergometer problem because six degrees of freedom are possible and ideally a thorough control system monitors all six to form the proper control response. It has been seen in other works that sensor configuration can interfere with the normal operation of an isolation system [17]. Therefore, consideration in the selection of non-intrusive motion transducers is left as part of future study. One possible development is the technique of magnetic flux density measurement [22]. This process uses non-contacting measurements of magnetic flux to determine to position of a suspended load. This type of measurement system would be ideal in the cycle ergometer if developed further.

Electromagnet Isolator Feasibility Conclusions Clearly the potential for this type of system exists given the level of prior work dealing with active control design. However, the team did not believe this type of system was feasible given the overall complexity of the design and the time given for this design team. Future work however may build on issues presented here.

3.3 Technical Decision Matrix

The alternative designs were evaluated again using a decision matrix based on technical feasibility (see Appendix I). The technical feasibility matrix was developed by a more quantitative approach rather than the qualitative approach used for the intuitive decision matrix. The decision criteria for this decision matrix is based on the specifications and requirements of NASA at the beginning of the project. Again the design criteria were assigned weighting factors using the method of pairs.

The parameters used for this decision matrix are based on the technical feasibility analysis performed by the design team. The parameters included for comparison of the alternatives were mass, energy domains, space constraints, multi-directional damping, wear, isolation damping ability, and time constraint. Many of the parameters are similar to the intuitive decision matrix, however isolation damping ability and time constraint needs to be quantified. The isolation damping ability is important in choosing a design solution because the isoıation system must be able to dampen frequencies below 5 Hz. This parameter is quantified in the feasibility analysis as finding the transmissibility or the dimensions necessary for the isolator to work. The design team also included time constraints because active controls and the air bearing system are complicated systems which could not be proven during a semester.

From this decision matrix the isolation box was chosen as the best design solution. The next feasible alternative was the resilient absorber.

The parameters that separated the two alternatives were wear and energy domains.

Design Solution

This section presents the final design solution developed for a cycle ergometer to be used onboard the space shuttle. The section discusses first the solution assumptions and system model. Next governing equations are developed for the model and analyzed as part of the design embodiment. Finally, the vibration attenuation results are discussed for the isolation box design solution (see Appendix J for detailed calculations).

4.1 Kinematic Model of the Isolation Box

4.1.1 Development

The embodiment of the isolation box continued from the feasibility analysis of the simplified two-dimensional isolation box presented in the feasibility section. After further investigation of the applied forces, the isolation box may now be extended to a three dimensional problem with six degrees-of-freedom. The six degrees-of-freedom are a result of forces and moments in the x, y, and z directions supplied from the rider. This system, shown in Figure 30, can now be defined by making some simplifying assumptions:

1. Translations displacements in the x, y, an z directions are small.
2. Angular displacements q_x, q_y, and q_z are small.
3. All springs are linear.
4. Ideal viscous damping.

5. Lumped mass system (ergometer mass and rider mass).

6. All forces and moments act through the center of the lumped mass.

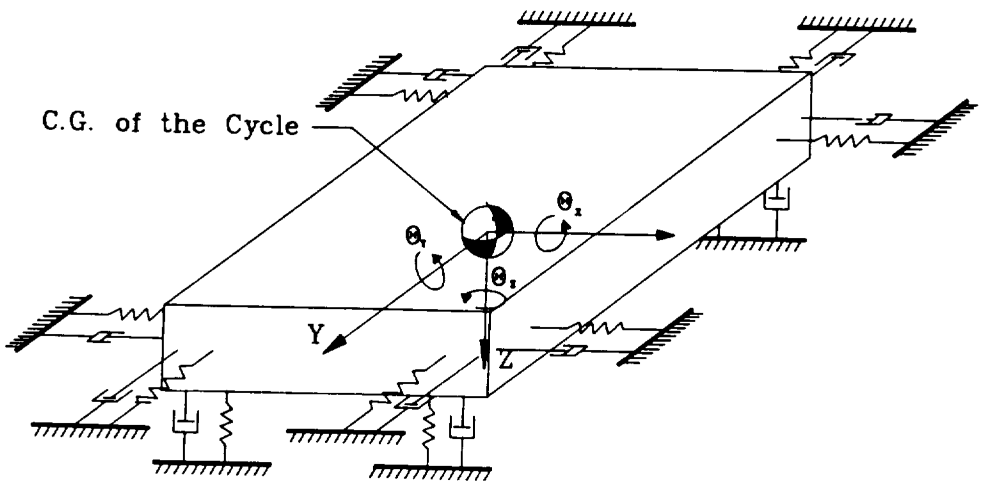


Figure 30. System Configuration for the Isolation Box

The kinematic model was used to determine the values of spring stiffness (K) and damping coefficients (B) needed to achieve a vibration isolation system that will transmit no more than 2.5 lbf to the floor of the orbiter. This value is chosen based on the worst-case load (assumed to be 100 lbf from sample data) and the acceleration limit of 1×10^{-4} g's. This force reduction requires an isolation system that has a characteristic transmissibility of 0.25.

4.1.2 Embodiment The first step in the embodiment process is to determine the state equations of the system. This can be done by considering free body diagrams for the x, y, and z directions and rotation about the x, y, and z direction. This process is shown in Appendix J. The resulting state equations (Equations 1- 12) are as follows:

Motion in:

X - Direction

$$\dot{P}_x = F_x - (4\ C_x P_x / m) - 4\ K_x X \tag{1}$$

$$\dot{X} = P_x / m \tag{2}$$

Y - Direction

$$\dot{P}_y = F_y - 4C_yP_y/m - 4K_zY \tag{3}$$

$$\dot{Y} = P_y / m \tag{4}$$

Z - Direction

$$\dot{P}_z = F_z - 4C_zP_z/m - 4K_zZ \tag{5}$$

$$\dot{Z} = P_z / m \tag{6}$$

H_x, θ_x - Direction

$$\dot{H}_x = M_x - (4C_z L^2/J)\ H_x - (4\ K_z L^2)\ \theta_x \tag{7}$$

$$\dot{\theta}_x = H_x/J \tag{8}$$

H_y, θ_y - Direction

$$\dot{H}_y = M_y - (4C_z W^2/J)\ H_y - (4\ K_z W^2)\ \theta_y \tag{9}$$

$$\dot{\theta}_y = H_y/J \tag{10}$$

H_z, θ_z- Direction

$$\dot{H}_z = M_z - 4(C_ya+C_xb)/J)\ H_z - 4\ (K_y\ a^2+K_x\ b^2)\ \theta_z \tag{11}$$

$$\dot{\theta}_z = H_z/J \tag{12}$$

where

P - indicates the momentun in the given direction

K - indicates the spring constant in the given direction

C - indicates the damping coefficient in the given direction

θ - indicates the angular displacement in the given direction

H - angular momentum in the given direction

Transfer functions can be developed for the forces and moments in the x, y, and z directions directly from the state equations. The six transfer functions represent the response of the system.

The next step of the embodiment process is to select spring constants and values for damping to be substituted into the transfer functions. The frequencies that are to be attenuated are fairly low. To achieve the desired attenuation, a low natural frequency is needed. The equation for the natural frequency for each direction can be readily obtained from the transfer function.

The denominator of the transfer function is called the characteristic polynomial of the system. The general form of this characteristic polynomial is shown below.

$$s^2+2\eta\omega_n+\omega_n^2 \tag{13}$$

This expression represents the order of the system being considered. The characteristic polynomial shown in (13) is that for a second order system. By comparing the denominator of the transfer function for our system to the general form of the characteristic polynomial, there is an apparent similarity. Because the characteristic polynomial of the system is quadratic, like in (13), the model of the isolation box is a second order system. From the general form of the characteristic polynomial, the equation for the natural frequency is obtained. This is shown in Equation (14) for forces in the x, y, and z direction.

$$\omega_n=\sqrt{4K_i/m} \qquad i = x, y, z \tag{14}$$

A further evaluation of transfer functions for moments in the x, y, and z directions yield different equations for natural frequencies. These equation are given below.

x-direction: $$\omega_n=\sqrt{4K_zL^2/J} \tag{15}$$

y-direction: $$\omega_n=\sqrt{4K_zW^2/J} \tag{16}$$

z-direction: $$\omega_n=\sqrt{(4K_xW^2-K_xL^2)/J} \tag{17}$$

For these equations, L is the length, W is the width, and J is the mass moment of inertia of the cycle. Equations (13) - (17) give the equations for the natural frequencies for each degree of freedom of the system.

4.1.3 Interpretation The model that is developed for the isolation box is restricted to one design parameter, which is the spring constant. Mass is not varied in the calculations for natural frequency for two reasons.

First, the assumption was made that the cycle was designed to minimize mass. Second, since the addition of mass results in an increase in shuttle mission cost, the addition of mass to the cycle was not considered.

By examining a general frequency response of a second order system, the following characteristics are inferred. At frequencies below the natural frequency, the response is flat. This means that for these frequencies, the transmissibility of the isolation system is constant. As the frequency approaches the natural frequency, a resonance peak occurs. At this point, the input to the isolation system is actually magnified. Moving further along the frequency axis, at frequencies greater than the natural frequency attenuation occurs. This means that the output of the system is being continuously decreased relative to a constant input.

By understanding the nature of second order systems, decisions are made as to approximate values of spring constants needed. If the system is excited at the natural frequency, the output becomes greater than the input to the system. For this reason, the operating frequency needs to be sufficiently greater than the natural frequency to assure attenuation of the signal to prevent resonance from occurring. Since the operating frequencies that we are concerned with range from 1 to 5 Hz, the natural frequency must he sufficiently less than 1 Hz.

By evaluating the equation for natural frequency, it becomes readily apparent that to achieve a low natural frequency, either the spring constant must be lowered since mass is constant as discussed earlier. In other words, the isolation system considered is made of soft springs.

The problem to solve is whether spring constants are available to drive the natural frequency sufficiently lowers that the operating frequency.

Using the transfer functions derived for the six degrees of freedom, frequency responses for each direction are developed. This process is simplified by using the computer software Design and Analysis of Linear Control Systems (DACS). DACS can be used to plot the frequency response for a given transfer function. By using very low values for spring stiffness, the responses are evaluated to determine if appropriate values of transmissibility are achieved for the operating frequencies.

4.1.2 Vibration Attenuation Results

The frequency responses for the system are developed for varying low values of the spring constant. The mass used is 113.5 kg (250 lbm). This is the lumped mass of the ergometer and the rider. The approach to the problem is to establish values of spring constant that yield a transmissibility of 0.25, as stated previously. By examining the frequency responses for values of spring constant in the range from 9117.6 N/m (52 Lb/in) to 10N/m (.05 Lb/in), the results concluded that attenuation of 0.25 is feasible for frequencies greater that 3.5. Unfortunately, this frequency is higher than the lowest operating frequency. Additionally, by considering the application of a static force, in the x-direction, the lumped mass encounters a counteracting force given by Equation (18).

$$F = 2KX \tag{18}$$

Using the spring constant of 175 N/m (1 Lbf/in), the spring encounters a deflection of 1.27m (4.16 ft). This value is not feasible because of space constraints of the orbiter mid-deck.

4.2 Configuration

The design of the spring box isolation system allows for low mass, and ease of installation. As shown in Figure 31 the isolation system in concept, consists of four isolator boxes attached at each corner of the ergometer base. The attachment of the isolator to both the ergometer and

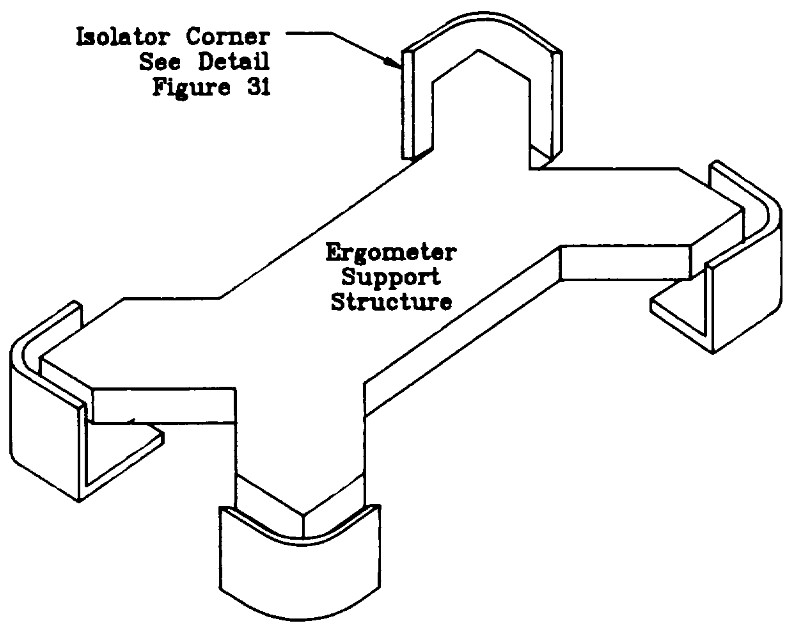


Figure 31 System Configuration for Isolation Box

the mid-deck floor uses the standard mushroom-type clips currently used to couple the ergometer to the mid-deck floor as shown, in Figure 31a.

Installation of the isolation system is designed to require minimal assembly time. After each box is removed from the locker, they are placed onto the mounting studs on the mid-deck floor. At this point, the ergometer is brought into place over the isolators and each corner is attached. Removal of the isolators is simply the reverse of installation.

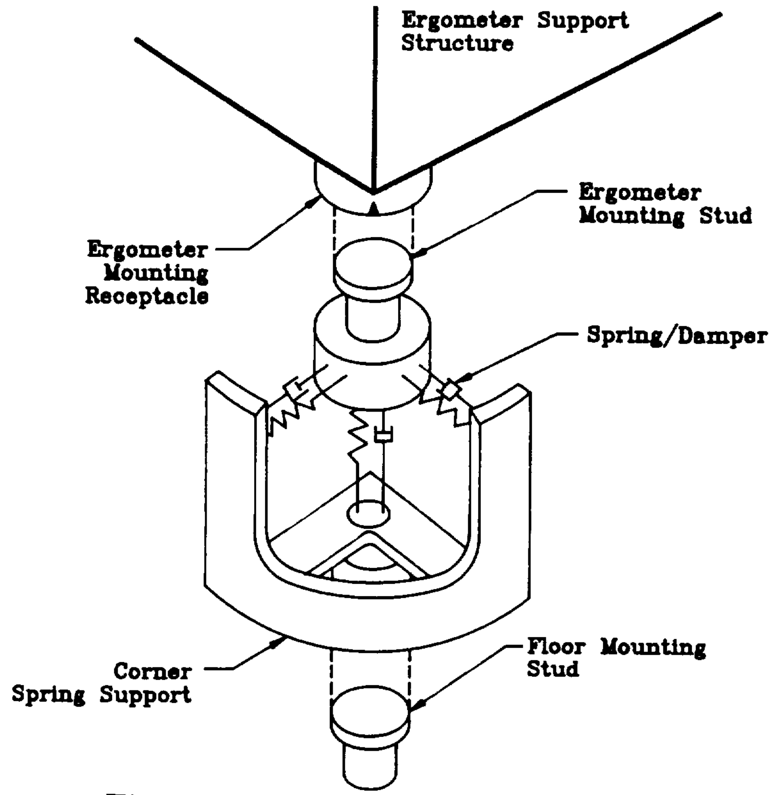


Figure 31a Isolation Box Coupling

Passive Isolator Application Guidelines

This section discusses the guidelines needed to solve passive vibration isolation problems. It is a result from the work performed on the feasibility of alternatives. Based on the results of the design solution, the passive systems are feasible for many applications of higher frequencies (> 5 Hz). For example, if the applications require isolations of vibrations at 25 Hz. with transmissibility of 0.2, passive systems, similar to the systems the team developed, will meet the requirements.

Passive systems have many advantageous. One advantage is that passive systems are not as complex as active systems. The design parameters do not change as the system operates. For example, a design parameter such as the spring constant does not vary with random input in a simple isolation box. Another advantage to the passive system is that the component mass is smaller in comparison to an active system. Active systems often require complex machinery to control the output of the system for a given input. Types of machinery commonly used are generators, servo-motors, and hydraulic pumps. Cost is another advantage to passive isolation systems. The cost of active systems are generally higher because of the components required for operation. If possible, passive systems should be used for vibration isolation.

Several parameters are important to determine if passive isolators can be used for a given application. This section gives a detailed description of the design process for vibration isolation systems.

When approached with a vibration problem, the first important consideration is to determine the cause of the problem. Vibrations of equipment can be a warning signal that the equipment needs repairing or should be replaced. Also, it is important to determine the source of the vibrations. Begin by trying to find a simple solution to the vibration problem. In other words, the vibration may be caused by neglect or misuse. If the vibration is an ongoing problem and not caused by natural interferences, the vibration must be reduced. The next step for a vibration problem is to determine important parameters necessary for the selection of an isolation system or motions that are exciting the system. Such parameters include the force that will be transmitted to the isolation system. Also, what mass will the isolation system have to support. Another parameter necessary for the selection of an isolation system is the constraints on the equipment and the environment. These constraints include geometry and space constraints.

The second important consideration leading to a solution to the vibration problem is to determine the parameters that will be important in the environment. For example, mass is important in the zero-g environment. Any payload mass that is to be used for spaceflight must meet the requirements of NASA. Also, research and becoming familiar with vibration and vibration control is an important step to find a solution. A decision of what types of isolation systems can then be determined.

From the specifications and requirements of the problem, the selection of a passive or active system can be determined. If a simple design and low cost is a requirement, a passive system more probable. If the problem requires enhanced damping effects and can make use of external power sources, an active system is applicable.

Based on the specifications, requirements, and research of applicable alternatives, the feasibility of these alternatives can be determined. This is accomplished by choosing an accurate approach to the problem based on the research, experience, and knowledge of isolation systems.

The next design consideration for a design solution of the vibration problem is to embody, or determine the feasibility of the passive system. Embodiment of passive vibration isolation problems begin with the development of an accurate model of the system. To develop a model the parameters of the system must be clearly defined. Such aspects as to the distribution of mass, the input-forces or motions of the systems and the output required are needed to clearly represent the system. With this information, a mathematical model of the system can be developed.

The first step is to identify the physical components of the vibrating system. Each component of the actual system is represented as an element in the model. Using these model elements, as a system, equations of motion are derived to determine the frequency response of the system. To find the frequency response of the system, the transfer function is needed. The transfer function will determine the output of the isolation system by knowing the input. The transfer function can then be plotted or used to find the natural frequencies of the system. The natural frequency will give the frequency at which resonance occurs. If resonance occurs, magnification of the input will be obtained. The transfer function vs. frequency, also called Bode Plots, are representations of the frequency response of the isolation system.

The next step is to perform a detail design on the system. Detail design involves a using the working mathematical model to develop a physical system. This process includes component selection based on the characteristics of the elements in the model. Also involved in the detail design process of the physical system. For instance, determining dimensions and clearances of the system to ensure the motion is not restricted. By applying these steps to any general vibration problem, passive elements can be proven feasible or infeasible.

Conclusions and Recommendations

The design team considered both active and passive isolation systems for the ergometer vibration reduction. Based on a design methodology that includes feasibility analysis and the decision matrix process, the team chose to passively isolate the ergometer with a tuned spring system. To embody the spring isolator the design team developed a lumped mass model of the ergometer and rider. State equations and transfer functions were developed for each axis of translation and rotation. Using the transfer functions, equations of natural frequency were developed for six degrees of freedom. To reduce the vibrations of the ergometer, an isolation system with a natural frequency lower than the operating frequencies was needed.

6.1 Passive System Abilities The design team chose to lower the ergometer natural frequency by lowering the spring constant of the system as opposed to increasing the overall mass.

The design team discovered that the low-frequency ergometer vibrations require very low spring constants to begin attenuation. It was found that for operating frequencies above 4 Hz, acceptable attenuation is possible. For lower frequencies the spring constants become so low that ergometer deflections exceed acceptable limits. In addition, these large deflections may lead to highly unstable operation. The spring isolator is not recommended to solve the ergometer vibration problem alone. However based on the analysis and research of the other alternative designs, several recommendations can be made.

6.2 Further Developments The goal of the design team was to develop a passive isolation system for the low frequency vibration of the cycle ergometer. Although the design solution discovered problems with low spring constants required, several recommendations can be made for additional research. These recommendations deal with modelling the ergometer system and secondly with combining the spring isolation system with an active isolation system.

The design team feels that a more descriptive model of the ergometer and rider is needed. In the interest of time, the design team modeled the cycle and rider as a lumped mass with all forces and moments acting through the center of the mass. The design team recommends that work be done on deriving a distributed mass model of the cycle and rider so that possible advantages of rider dynamics can be considered. Another recommendation is that the order of the system model be increased. Based on the analysis of second order systems, it is evident that an increase in system order will further reduce transmissibility. Modification of the ergometer model involves adding energy storage elements to each degree of freedom. The effect of this storage element addition is a downward shift in the plots of system transfer functions.

Another recommendation is additional research into the feasibility of a combined active and passive isolator system. In this configuration, the spring isolator developed in this project is used in conjunction with a high pass active isolator dedicated to the low frequency frequency vibrations the passive system cannot isolate. By using the passive isolator in this manner, the demands on the active system may be significantly reduced.

The design team also recommends that a simulation of the cycle ergometer be developed to aid analysis of the isolator . This simulator would allow for the full range of ergometer exercise settings as well as different riders.

A final recommendation deals with the mounting technique of the ergometer to the mid-deck floor. The team feels that vibration is developing at the mounting points at the base of the ergometer through the use of "loose" couplers. By making the connections tighter between the ergometer and shuttle floor the vibration magnitudes may be significantly reduced by eliminating the "bang-bang" behavior.

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APPENDICES

Appendix A
Transmissibity Requirements

Maximum acceleration = 1×10^{-4} g's

Mass of shuttle = 106,625 lbm

$F = ma$

$= (106,625 \text{ Kg})(1 \times 10^{-4} \text{ g's})$

$= 10.66 \text{ N} = 2.4 \text{ lb}$

$$\text{Transmissibility} = \frac{\text{Output Force}}{\text{Input Force}}$$

From input data (given by NASA)

Needed output force = 100 lbs

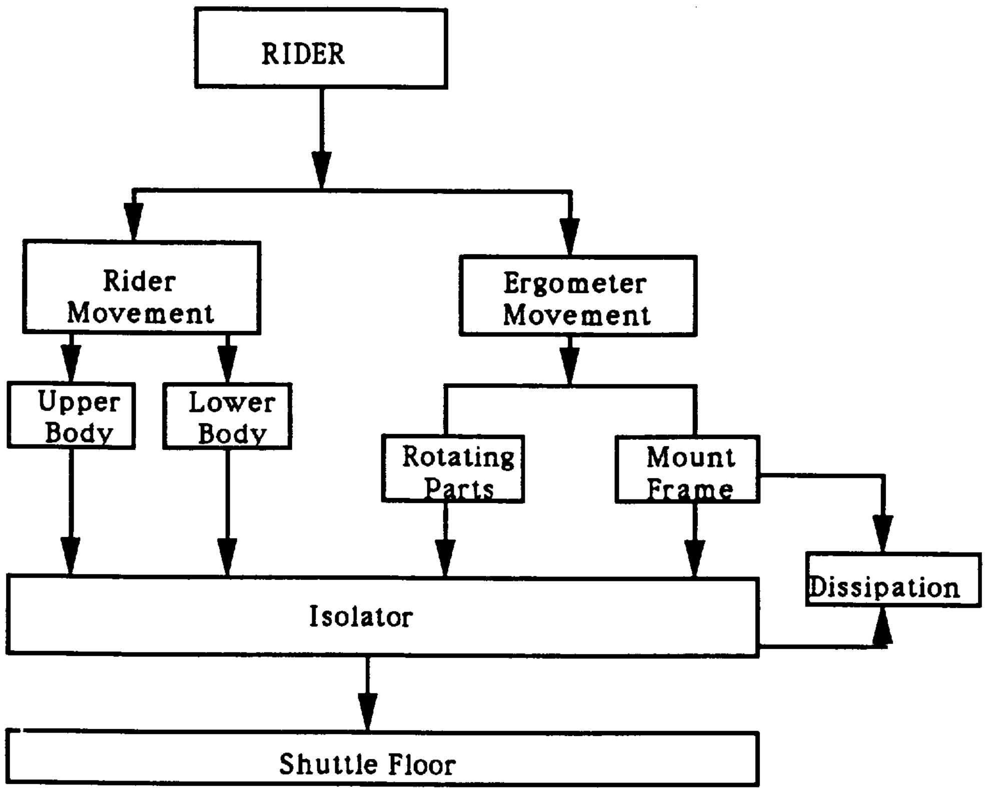
Transmissibility = $2.5^{lb}/100^{lb}$ = $\underline{\underline{0.25}}$

D/W	D = Demand W = Wish
	Isolation System
W	• No tool usage
D	• Orbiter mass: 106,625 kg (235,00 lbm)
D	• Maximum acceleration into the orbiter from the ergometer is 1×10^{-4} g's, with a goal of 1×10^{-5} g's
W	• No pertroleum products may be used
D	• Include a factor of safety of 1.4 based on yield strength
D	• Constrained to the forces, moments, and power spectrums provided to calculate the riders input into the exercise system
D	• Floor to ceiling middeck height: 1.9 m (76 in.)
D	• Maximum height cycle may be raised from middeck floor: 0.3 m (10 in.) (Upright position only)
D	• The payload shall provide for safe containment of any by-product of payload experiment-gas, liquid, or solid
D	• No gases except air be discharged into the middeck environment
D	• Any special illumination shall be provided by the isolation system
	Ergometer
D	• The ergometer will be mounted to the orbiter middeck floor

D/W	Specifications (Cont'd) Isolation System D = Demand W = Wish
D	• Total launch weight of the frame and ergometer is 31 kg (68 lbm) This weight cannot be increased above 33 kg (72 lbm)
D	• The vibration isolation equipment will be assembled to the ergometer/frame assembly in orbit and disassembled and stowed prior to deorbit
D	• Ergometer dimensions: 0.4 x 0.5 x 0.2 meters (16 x 19 x 6 inches)
	Middeck Locker
D	• Two middeck lockers can be used to stow the vibration isolation equipment
D	• An isolation system is defined as not exceeding 25 kg (54 lbm) when stowed in the lockers
D	•Contents of the middeck locker do not consume more than 115 W maximum continuous DC power for 8 hours
D	• Provides .06 cubic meter (2 cubic feet) of stowage volume
D	• Two sizes of standard stowage trays are available: Large trays: .05 cubic meter (1.8 cubic feet of volume) Small trays: .02 cubic meter (0.85 cubic feet of volume)
D	• The isolation system equipment shall be packaged in trays using foam inserts or non-structural plastic tray dividers dividing the trays into halves, quarters, eighths or sixteenths to prevent floating during orbit [4]
	Push-Off Loads
D	• Isolation system - provided middeck equipment shall be designed to sustain a 57 kg (125 lb) static load distributed over a .1 m x .1 m (4 inches x 4 inches) area
	Acoustics
D	• Individual payload elements shall not emit continous acoustic noise into the crew working/living spaces exceeding the levels speified in the IDD

D/W	
	Heating and Cooling
D	• Maximum heat load to the middeck locker: 60 W
D	• The maximum air outlet temperature allowed to discharge to the cabin (air circulation fan used by the isolation system): 49°C (120°F)
D	• External surface temperatures of the isolation system elements accessible to the crew shall not exceed 45°C (113°F); inaccessible external surfaces shall not exceed 49°C (120°F)
	Electrical Energy
D	• Power shall be available for periods up to 8 hours continously during in-orbit operations(see IDD) [4]
D	• Maximum voltage levels: 32 V DC (Nominal : 28 V DC)
D	• All isolation system's wiring connecting to orbiter power sources shall be sized to be consistent with appropriate circuit protection devices (see IDD)
	Limitations on Middeck Payload Utilization of Electrical Power
D	• For emergency purposes, the isolation system is able to sustain a safe condition with permanent loss of orbiter power
D	• Isolation system shall provide means for its power activation/deactivation by crew control
D	• All power must be removed when mating or demating electrical connectors [4]

Appendix C: Energy Flow Diagram



Appendix D

Decision Matrix Method

The decision matrix aided the team in selecting the most feasible conceptual alternatives designs. The first step for completing the decision matrix was to identify the most important parameters according to the specifications and requirements of NASA. After identifying these parameters, the team assigned weighing factors to each of the design parameters using the method of pairs. This method compares two parameters and assigns a mark to the most important of the two. This process is completed for all parameters and the marks are summed and divided by the total number of marks. The total number of marks, from this calculation, is the weighing factor for the parameter. After determining the weighing factors, the team rated each alternative accordingly. To determine the rank of each alternative, the team multiplied the weighing parameter by the rating and totaled the products.

Appendix D1: Weighing Factors- Method of Pairs

1. Mass
2. Energy Domains
3. Space Contraints
4. Stability
6. Multi-Directional Damping
7. Storage
8. Time to Produce
9. Number of Parts
10. Wear
11. Cost
12. Materials
13. Active/Passive

	1	2	3	4	5	6	7	8	9	10	11	12	13
1		1	3	1	5	6	7	1	9	10	1	1	13
2			3	4	5	6	7	2	2	10	2	2	13
3				4	5	6	7	3	3	10	3	3	3
4					5	6	7	4	4	10	4	4	4
5						6	7	5	5	5	5	5	5
6							6	6	6	6	6	6	6
7								7	7	7	7	7	7
8									9	10	8	8	13
9										10	9	9	13
10											10	10	10
11												11	13
12													13
13													

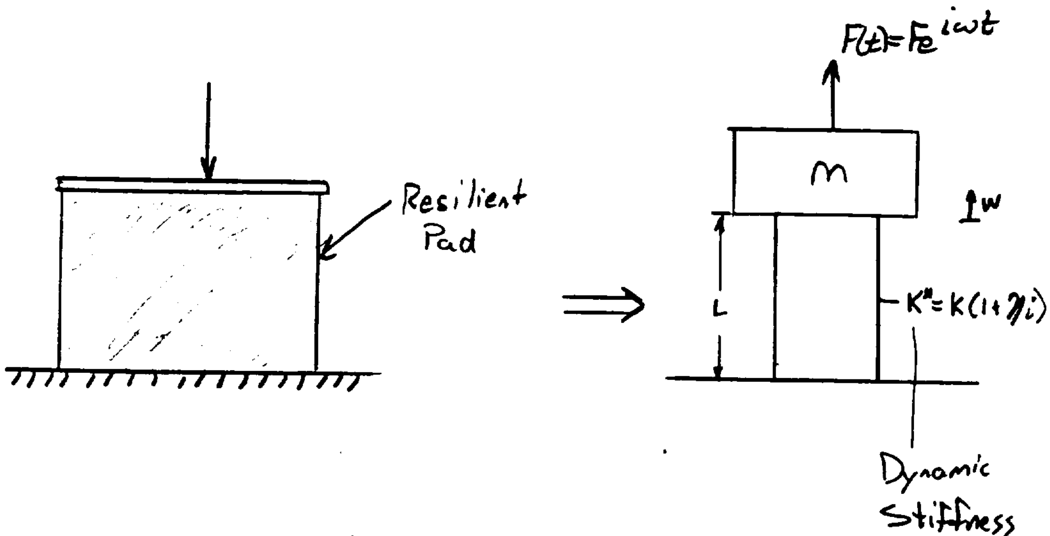
Appendix D2: Preliminary Decision Matrix

Parameters / Weighing Factors / Alternatives	1	2	3	4	5	6	7	8	9	10	11	12	Rate	Total
	.064	.051	.09	.09	.128	.154	.141	.026	.051	.115	.013	.077		1.0
Intermediate Absorber	8 / .512	9 / .459	9 / .81	9 / .81	9 / 1.15	7 / 1.08	10 / 1.41	8 / .208	9 / .459	6 / .69	8 / .104	8 / .616	3	8.308
Coupling Isolator	9 / .576	9 / .459	10 / .9	9 / .81	9 / 1.15	7 / 1.08	10 / 1.41	8 / .208	8 / .408	6 / .69	7 / .091	8 / .616	2	8.398
X-Suppport Isolator	6 / .384	9 / .459	7 / .63	7 / .63	6 / .768	8 / 1.23	7 / .987	6 / .156	7 / .357	6 / .69	6 / .078	8 / .616	6	6.987
Ball&Socket Isolator	8 / .512	9 / .459	10 / .9	9 / .81	9 / 1.15	10 / 1.54	10 / 1.41	7 / .182	7 / .357	7 / .805	7 / .091	8 / .616	1	8.834
Suspended Structure	5 / .32	8 / .408	4 / .36	4 / .36	4 / .512	9 / 1.39	6 / .846	8 / .208	6 / .306	8 / .92	6 / .078	8 / .616	8	6.32
Isolation Box	6 / .384	8 / .408	6 / .54	7 / .63	5 / .64	9 / 1.39	7 / .987	7 / .182	6 / .306	8 / .92	6 / .078	8 / .616	5	7.077
Fan Isolation System	5 / .32	6 / .54	7 / .63	6 / .54	3 / .384	6 / .924	6 / .846	6 / .156	5 / .255	7 / .805	4 / .052	5 / .385	9	5.603
Air Bearing	4 / .256	5 / .255	6 / .54	5 / .45	4 / .512	6 / .924	6 / .846	5 / .13	4 / .204	8 / .92	4 / .052	5 / .385	10	5.474
Fluid Bladder Isolator	7 / .448	7 / .357	8 / .72	9 / .81	8 / 1.024	8 / 1.23	9 / 1.27	6 / .156	7 / .357	5 / .575	5 / .065	6 / .462	4	7.475
Wedge Block Isolator	7 / .448	6 / .306	7 / .63	8 / .72	7 / .896	6 / .924	8 / 1.23	4 / .104	3 / .153	4 / .46	4 / .052	8 / .616	7	6.437
Electro-magnet	6 / .384	5 / .255	7 / .63	4 / .36	3 / .384	4 / .616	6 / .846	5 / .13	4 / .204	9 / 1.03	2 / .026	3 / .231	11	5.101

Appendix E

Resilient Absorbers Calculations

Modeling the system as a resilient block between two rigid plates.



Resilient Materials

Notes on Dynamic Stiffness

$$K^* = K(1+i\eta)$$

The first term, K, represents the stiffness of the material

The second term (Imaginary Part) represents the damping of the system, $iK\eta$.

The Dynamic stiffness can be related to the complex compression modulus, E^*, by:

$$K^* = \frac{E^* s}{L}$$

Where $E^* = E(1 + i\eta)$

Therefore

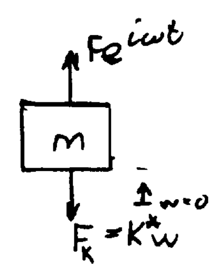
$$K^* = \frac{A}{t}\left[E(1 + i\eta)\right]$$

A - load area
t - thickness

Now solving for the motion of the system

This is a 1-DOF system

<u>Sum of forces in w-direction</u>



$$\sum F_w = m\ddot{w}$$

$$Fe^{i\omega t} - K^* w = m\ddot{w}$$

$$m\ddot{w} + K(1 + i\eta)w = Fe^{i\omega t}$$

$$\ddot{w} + \frac{K}{m}(1 + i\eta)w = Fe^{i\omega t}$$

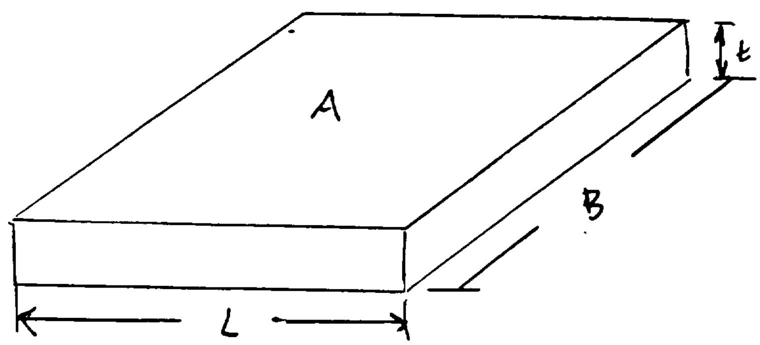
<u>Note</u>: The general form of a 2nd order differential equation

$$\ddot{w} + 2\eta\omega_n\dot{w} + \omega_n^2 w = 0$$

There is no $2\eta\omega_n$ because damping is part of the stiffness.

$$\omega_n = \sqrt{\frac{K^*}{m}}$$

$$\omega_n = \sqrt{\frac{\frac{A}{t}\left[E_c(1+i\eta)\right]}{m}}$$



The compression modulus, E_c, is dependent on ω

$$E(\omega) = \check{E} + \hat{E}\left[1 - \frac{1}{1+(\beta\omega)^n}\right]$$

$\check{E}$, $\hat{E}$, β, n can be obtained from the figures on the next page.

Since this is a second order system we know we want to have a low natural frequency. (See Chapter 4 of Report)

For $\omega = 2\,Hz$

$\check{E} = 100\ lb/in^2$

$\hat{E} = 10^5\ lb/in^2$

$\beta = 10^{-4}$

$n = 2$

$$E(\omega=0.5Hz) = 100\,lb/in^2 + 10^5\,lb/in^2\left[1-\frac{1}{(10^{-4})(0.5)}\right] \quad E.3$$

$$E(\omega=.5Hz) = 100\,lb/in^2$$

changing to metric

$$E =$$

Now

$$\omega_n = \sqrt{\frac{\frac{A}{t}(E(1+i\eta))}{m}}$$

$$\omega_n\sqrt{tm} = \sqrt{A[E(1+i\eta)]}$$

$$t = \frac{A[E(1+i\eta)]}{\omega_n^2 m}$$

what is η?

$$\eta \sim 0$$

$$t = \frac{AE}{\omega_n^2 m}$$

$$= \frac{(4in^2)(100\,lb/in)}{(2Hz)^2(7.83)}$$

$$\boxed{t = 12.77in}$$

This thickness is too big for our uses
The dimensions for a resilient isolator are
not feasible.

Appendix F

Three Dimensional Model of Isolation Box

Feasibility Analysis

must find:

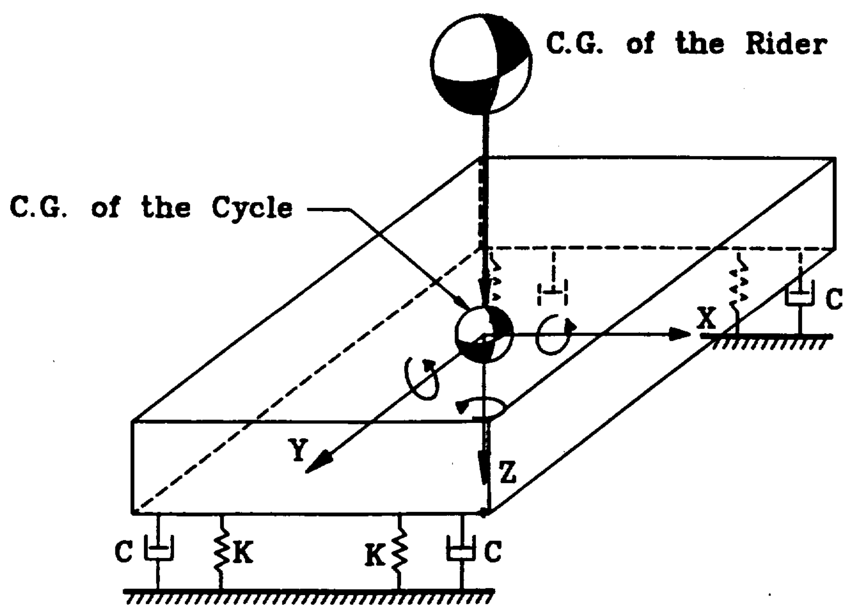
(1) equations of motion

(2) transfer function

(3) determine the frequency response

Model the System:

Free body Diagram



Rotational movement of the rider and ergometer structure. Spring-Damper systems x, y, z direction relative to each other. Obtain 6 DOF three dimensional system. F1

1) Equations of Motion

locating the center of mass

$P_1 = r_g + \rho_1$ where r_g = location of center of mass relative to the "fixed frame" at that time

$\dot{v}_1 = \dot{v}_g + \rho_1$ $\quad \rho_1$ = const, distance from the mass of rider to the isolator.

$\rho_1 = w \times \rho_1$

Center of Mass equations:

$$m r_g = \sum_{i=1}^{n} m_i r_i, \quad m = \sum_{i=1}^{n} m_i$$

$$r_g = \frac{1}{m} \sum_{i=1}^{n} m_i r_i$$

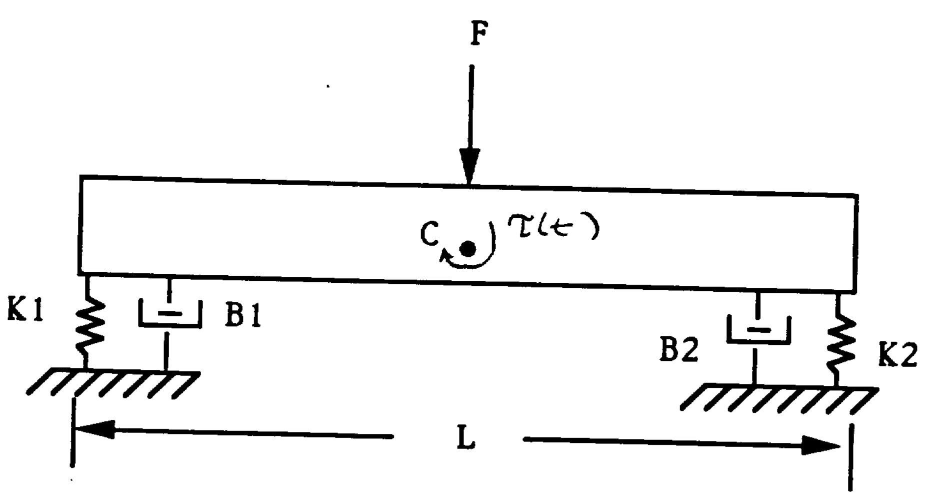
Need information on the motions of ricer at different times or frequencies, in the form of:

M, F => at the center of mass, r_g

Two Dimensional Model of Isolation Box

To determine the equations of motion for the two dimensional system of isolation box, used coordinate transformations coupling.

The system represents the ergometer structure, spring-damper system at corners. The system is one-dimensional, plane of symmetry.

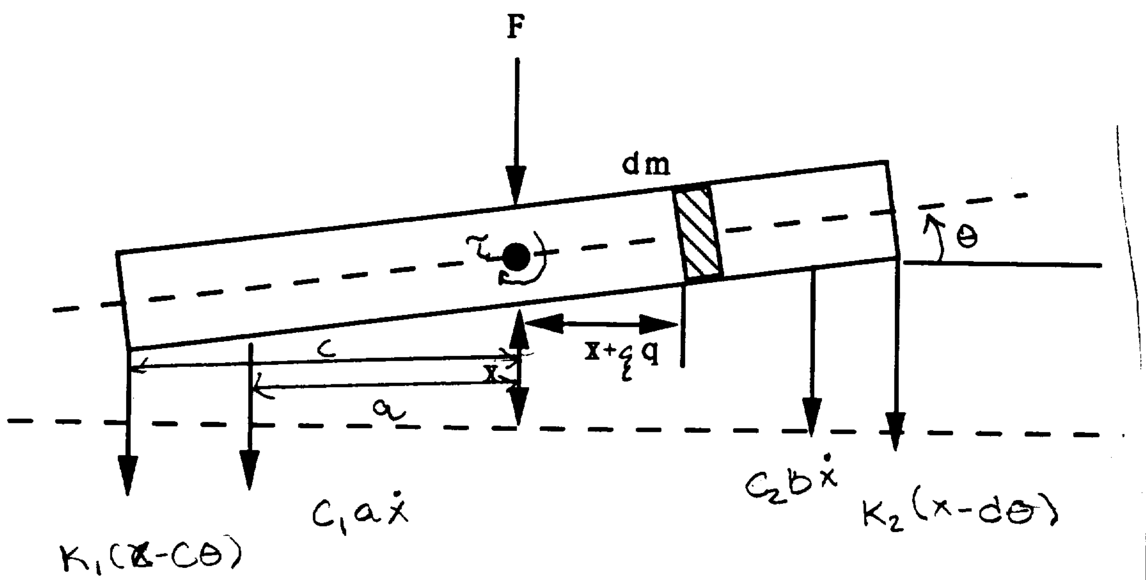


Ergometer structure represented by total mass with center of mass C. Distances a + b from center of mass to the springs. Body has mass moment of inertia I_c about C. Force applied by rider creates a torque about C. Free body diagram where displaceplacement consist of vertical motion and rotation about C. F3

Assume $\theta(t)$ to be small.

To determine equations of motion, find the displacement which consist of translation of the center C and rotation about C.

Using Free Body Diagram:



Force equation:

$$-K_1(x-c\theta)-C_1a\dot{x}-K_2(x-d\theta)-C_2b\dot{x}-F=\int_{body}(\ddot{x}+\xi\ddot{\theta})dm$$

$$=\ddot{x}\int_{body}dm+\ddot{\theta}\int_{body}\xi dm$$

$$=m\ddot{x}$$

Motion Equation:

F4

$$K_1(x-c\theta)c + C_1\dot{x}a^2 - K_2(x+d\theta) - C_2\dot{x}b^2 + \tau(t) =$$
$$\int_{body} \xi(\ddot{x}+\xi\ddot{\theta})dm = \int_{body} \xi dm + \ddot{\theta}\int_{body}\xi^2 dm = I_c\ddot{\theta}$$

Simplifying:

$$-K_1x - K_1c\theta - C_1\dot{x}a - K_2x - K_2d\theta - C_2\dot{x}b + F = m\ddot{x}$$
$$K_1xc - K_1c^2\theta + C_1\dot{x}a^2 - K_2xd - K_2d^2\theta - C_2\dot{x}b^2 + \tau(t) = I_c\ddot{\theta}$$

$$m\ddot{x} + (K_1+K_2)x - (K_1c - K_2d)\theta + (C_1a + C_2b)\dot{x} = F$$
$$I_c\ddot{\theta} - (K_1c - K_2d)x - (K_1c^2 + K_2d^2) - (C_1a^2 - C_2b^2) = \tau$$

State Equation Form:

$$\begin{bmatrix} m & 0 \\ 0 & I_c \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{\theta} \end{bmatrix} + \begin{bmatrix} K_1+K_2 & -K_1c-K_2d \\ -(K_1c-K_2d & K_1c^2+K_2d^2 \end{bmatrix}\begin{bmatrix} x \\ \theta \end{bmatrix} + \begin{bmatrix} C_1a+C_2b \\ -C_1a^2-C_2b^2 \end{bmatrix}$$
$$= \begin{bmatrix} F \\ \tau \end{bmatrix}$$

Taken this form:

$$m\ddot{z} + 2C\dot{z} + 2Kz) = F$$

State equations:

$$\dot{p} = F - \frac{2Cp}{m} - 2Kz$$
$$\dot{z} = p$$

In State Matrix Form,

$$\begin{bmatrix} \dot{p}_z \\ z \end{bmatrix} = \begin{bmatrix} -2c/m & -2k \\ 1/m & 0 \end{bmatrix} \begin{bmatrix} p \\ z \end{bmatrix} + \begin{bmatrix} 1 \\ 0 \end{bmatrix} F_z$$

$$SI - A = \begin{bmatrix} s + 2c/m & 2k \\ -1/m & s \end{bmatrix}$$

$$\det(SI - A) = s^2 + 4c/m\,s + 4k/m$$

$$C_{11} = \frac{\begin{vmatrix} 1 & -2k \\ 0 & s \end{vmatrix}}{s^2 + \frac{2c}{m}s + \frac{2k}{m}} = \frac{s}{s^2 + \frac{2c}{m}s + \frac{2k}{m}}$$

$$C_{21} = \begin{bmatrix} s + \frac{2c}{m} & 1 \\ -1/m & 0 \end{bmatrix} = \frac{+1/m}{s^2 + \frac{2c}{m}s + \frac{2k}{m}}$$

$$C_z = \begin{bmatrix} s \\ 1/m \end{bmatrix} \frac{1}{s^2 + 2c/m\,s + 2k/m} \Leftarrow \text{Transfer Function}$$

Using this transfer function, the team developed Bode plots. From Berg Precision components $K = 52 N/m$

$$\omega_n = \sqrt{2K/m} = \sqrt{\frac{2(52 N/m)}{113 kg}}$$

$$= 0.91 Hz$$

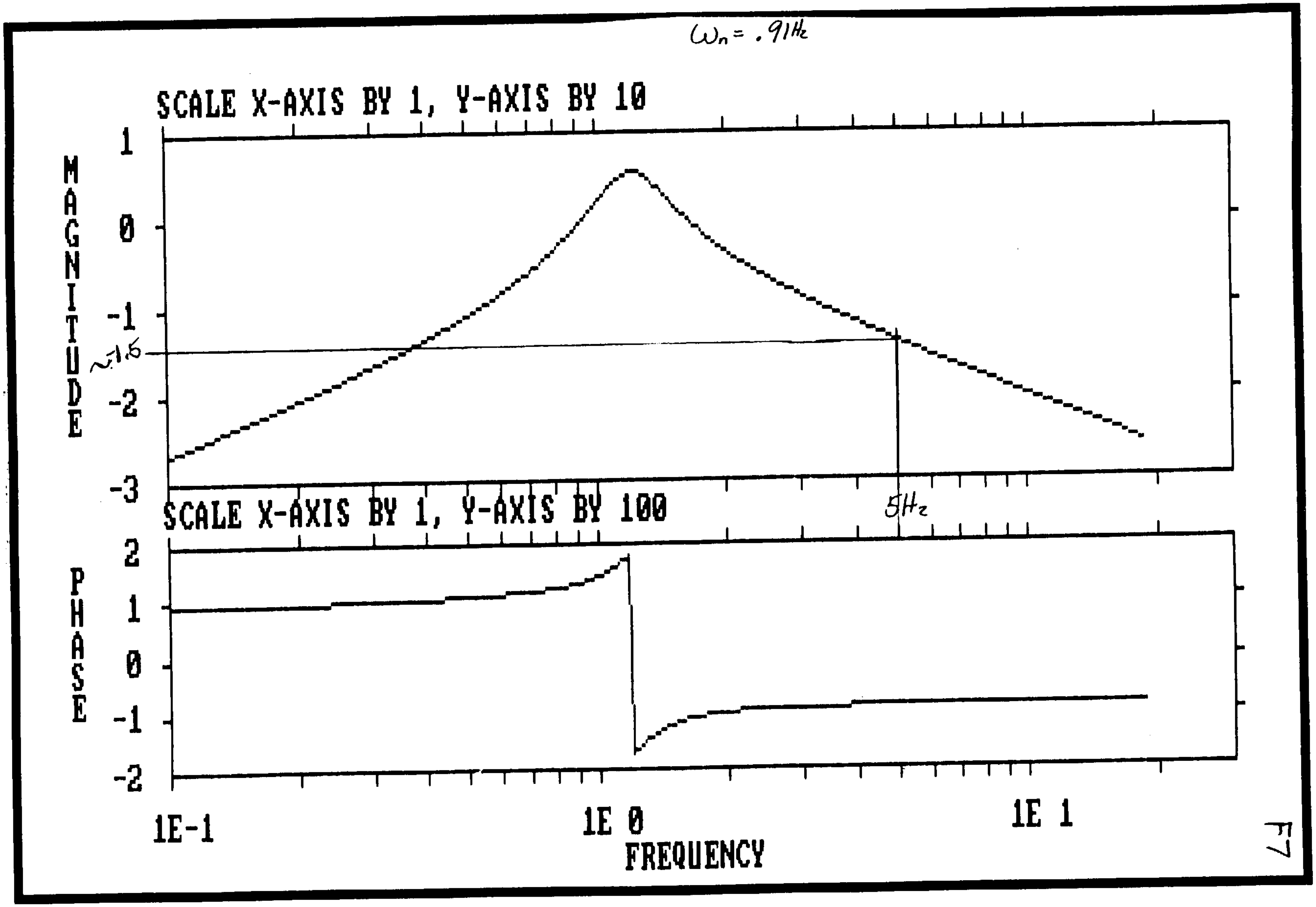
At 5 Hz, $20 \log_{10} |T(\omega)| = -15$

$$|T(\omega)| = .17$$

This system has the regulated transmissibility at 5 Hz.

Reference:

Berg, Precision Mechanical Component Catalog, Winfred M. Berg, Inc. East Rockaway, New York, p. 328, 1990.

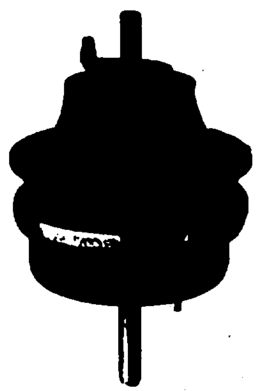


$\omega_n = .91Hz$
SCALE X-AXIS BY 1, Y-AXIS BY 10
MAGNITUDE
1
0
-1
~-1.6
-2
-3
SCALE X-AXIS BY 1, Y-AXIS BY 100
5Hz
PHASE
2
1
0
-1
-2
1E-1
1E 0
1E 1
FREQUENCY
F7

Appendix G

Example of Fluid Bladder Damper

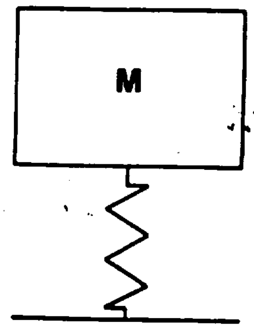
The Fluidlastic Mount is a marketed hydraulic damper. The mount is used for automobiles and light trucks. Fluidlastic Mount uses rubber and Fluids to provide for vibration isolation reduction better than most conventional dampers.



Fluidlastic Mount

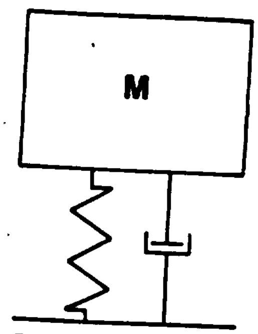
The damper is three mounts in one:

1) Provides for basic load and motion capability to the system.



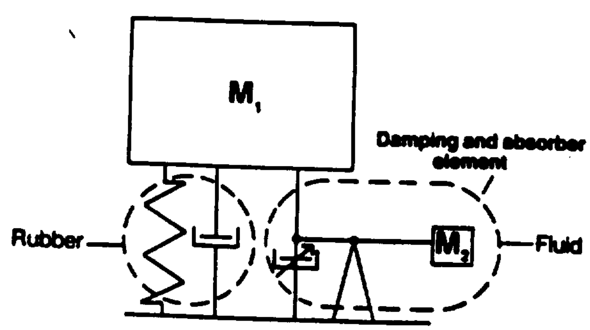
Rubber mount

2) Provides for restricted motion at or near resonant conditions (Damping)



Damped rubber mount

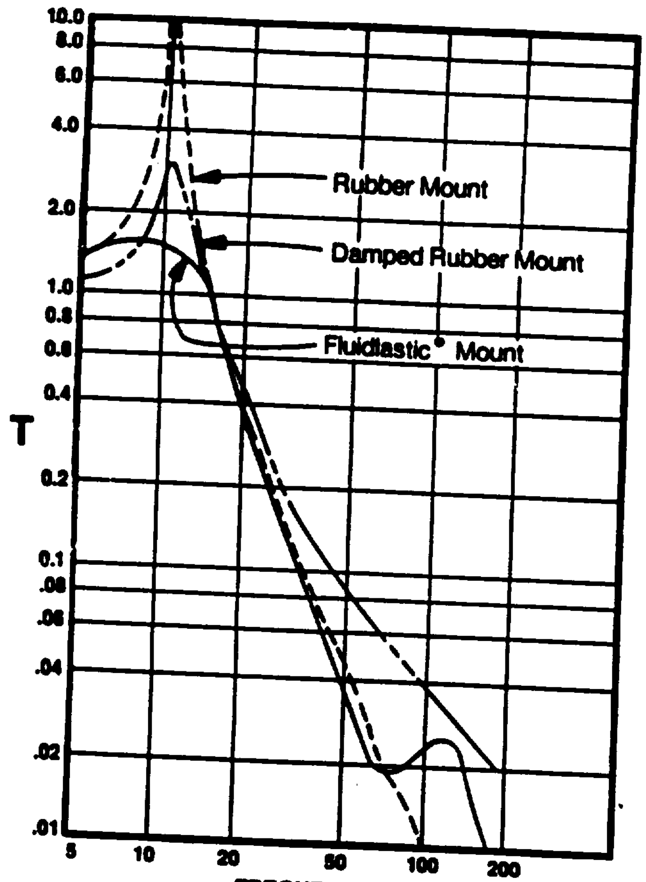
3) Tuned absorber providing for isolation at specific frequency.



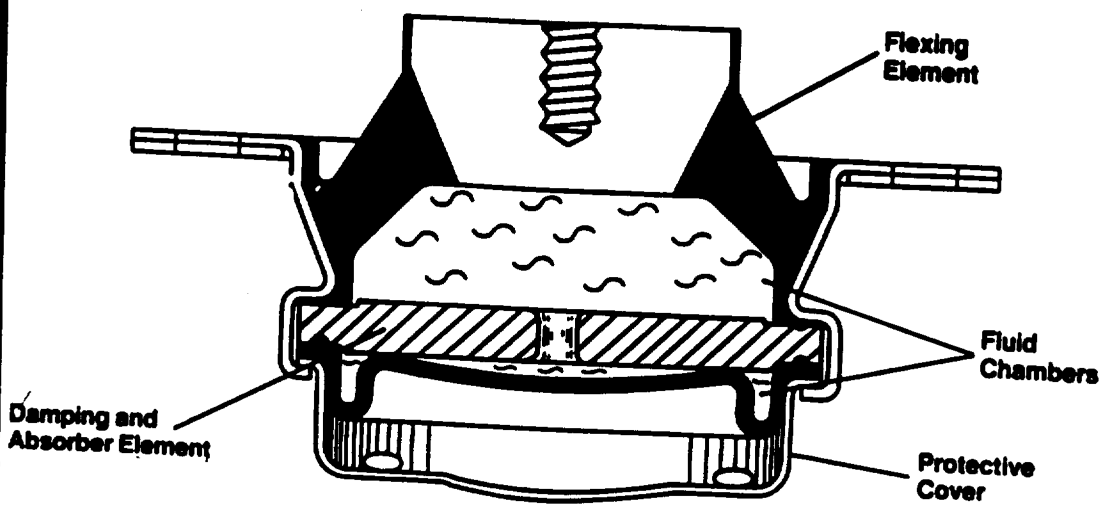
Fluidlastic Mount

G2

When researching market hydraulic dampers, the fluid bladder systems will dampen vibrations at high frequencies (approximately 14 Hz). In other words, the fluid bladder dampers will not dampen frequencies as low as 5 Hz. needed for the exercise cylce.



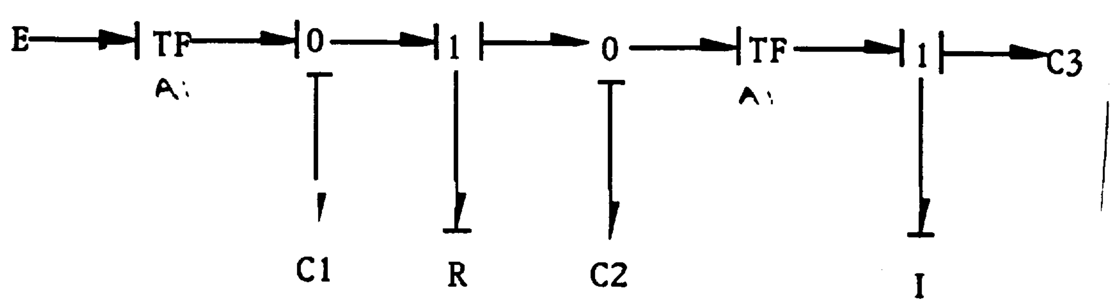
Fluidlastic Mounts outperform conventional isolators at resonance and in the isolation zone.



Fluidlastic Mount

The Fluidlastic Mount was used to determine the parameters need for the feasibility analysis.

Bond Graph of the Fluid Bladder Damper



E - Force input
C1 - capacitance chamber 1
R - orifice
C2 - capacitance chamber 2
I - inertiance of resilient material
C3 - capacitance chamber 3

Procedure for finding Transfer Function using Bond Graphs

Variables of Bond Graph:

e - effort, $e = dP/dt$

f - flow, $f = dq/dt$

q - Generalized displacement

$q = \int f\,dt$

p - generalized Momentum.

C - Capacitance (stores potential energy)

R - resistance (dissipates energy!)

I - mass (stores kinetic energy)

TF - transformer.

This system is a Nth order system because two capacitors and inertia has derivitive causality.

N-th order Transfer Function:

General equation:

$$\dot{y} = \underline{A}\,y + \underline{B}\,u$$

By forcing the system at a frequency at or arbitrarily close to natural frequency, response fails.

Assume $y = Ye^{st}$ $u = Ue^{st}$

where s - forcing frequency

Solve N'th order equation

$$(s\underset{\sim}{I} - \underset{\sim}{A})\underset{\sim}{Y} = \underset{\sim}{B}\,\underset{\sim}{u}$$

With Cramer's rule, we find

$$\underset{\sim}{Y} = \underset{\sim}{C}\,\underset{\sim}{u}$$

$$C = (sI - A)^{-1}\underset{\sim}{B}$$

where $C = \dfrac{Y_i}{U_j} = \dfrac{\det(sI - A \text{ w/ ith column replaced by ith colum of } B)}{\det(sI - A)}$

Gives transfer function characterized by the system of equations

Feasibility on Fluid Bladder

To find the transfer function, must ge the constitutive equations, then the state equations With the state equations, get the transfer function.

Constitutive equations:

$e_1 = E$

$e_1 = n_1 e_2, f_2 = n_1 f_1$

$f_3 = \dot{q}_3$

$f_5 = \frac{1}{R} e_5$

$e_7 = \frac{1}{C_2} q_7$

$e_8 = n_2 e_9, f_9 = n_2 f_8$

$f_{10} = \frac{1}{I} P_{10}$

$e_{11} = \frac{1}{C_3} q_{11}$

Connecting Equations:

$e_2 - e_3 = e_4$

$f_2 - f_3 - f_4 = 0$

$e_4 - e_5 - e_6 = 0$

$f_4 = f_5 = f_6$

$e_6 = e_7 = e_8$

$f_6 - f_7 - f_8 = 0$

$e_9 - e_{10} - e_{11} = 0$

$f_9 = f_{10} = f_{11}$

Systems of Equations:

$$\dot{q}_7 = f_7 = f_6 - f_8 = f_5 - \frac{1}{n} f_9 = \frac{1}{R} e_5 - \frac{1}{nI} P_{10} = \frac{1}{R}(e_4 - e_6) - \frac{1}{n_2 I} P_{10}$$

$$= \frac{1}{R}\left(\frac{1}{n_1} E - \frac{1}{C_2} q_7\right) - \frac{1}{n_2 I} P_{10} = \frac{1}{R n_1} E - \frac{1}{R C_2} q_7 - \frac{1}{n_2 I} P_{10}$$

$$\dot{P}_{10} = e_{10} = e_9 - e_{11} = \frac{1}{n_2} e_8 - \frac{1}{C_3} q_{11} = \frac{1}{n_2 C_2} q_7 - \frac{1}{C_3} q_{11}$$

$$\dot{q}_{11} = \frac{1}{I} P_{10}$$

Matrix Form

$$\begin{bmatrix} \dot{q}_7 \\ \dot{p}_{10} \\ \dot{q}_{11} \end{bmatrix} = \begin{bmatrix} -\frac{1}{RC_2} & -\frac{1}{n_2 I} & 0 \\ \frac{1}{n_2 C_2} & 0 & -\frac{1}{C_3} \\ 0 & \frac{1}{I} & 0 \end{bmatrix} \begin{bmatrix} q_7 \\ p_{10} \\ q_{11} \end{bmatrix} + \begin{bmatrix} \frac{1}{Rn_1} \\ 0 \\ 0 \end{bmatrix} E$$

$$sI - A = \begin{bmatrix} s + \frac{1}{RC_2} & \frac{1}{n_2 I} & 0 \\ -\frac{1}{n_2 C_2} & s & \frac{1}{C_3} \\ 0 & -\frac{1}{I} & s \end{bmatrix}$$

$\det(sI - A)$

$$= \left(s + \frac{1}{RC_2}\right)\left(s^2 + \frac{1}{C_3 I}\right) - \frac{1}{n_2 I}\left(-\frac{s}{n_2 C_2}\right)$$

$$= s^3 + \frac{1}{C_3 I} s + \frac{1}{RC_2} s^2 + \frac{1}{RC_2 C_3 I} + \frac{1}{n_2^2 I C_2} s$$

$$= s^3 + \frac{1}{RC_2} s^2 + \frac{1}{C_3 I} s + \frac{1}{n_2^2 I C_2} s + \frac{1}{RC_2 C_3 I}$$

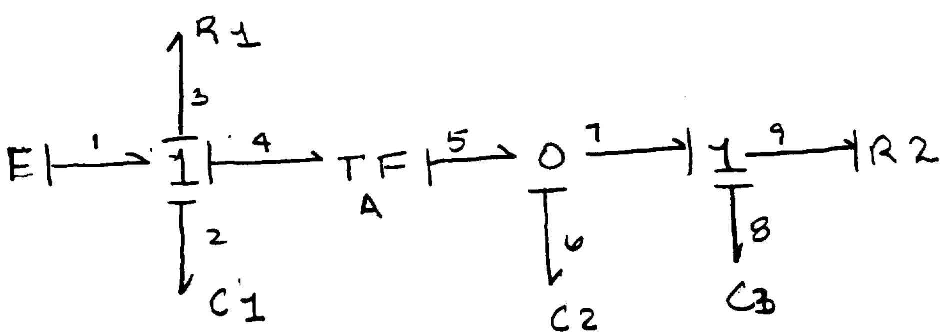
$$C_{11} = \frac{\begin{vmatrix} \frac{1}{Rn_1} & \frac{1}{n_2 I} & 0 \\ 0 & s & \frac{1}{C_3} \\ 0 & -\frac{1}{I} & s \end{vmatrix}}{\det(sI - A)} = \frac{\frac{1}{Rn_1}\left(s^2 - \frac{1}{IC_3}\right)}{\det(sI - A}$$

$$C_{21} = \frac{\begin{vmatrix} s+\frac{1}{RC_2} & \frac{1}{Rn_1} & 0 \\ -\frac{1}{n_2C_2} & 0 & \frac{1}{C_3} \\ 0 & 0 & s \end{vmatrix}}{\det(sI-A)} = \frac{\frac{1}{Rn_1}\left(-\frac{1}{n_2C_2}s\right)}{\det(sI-A)}$$

$$C_{31} = \frac{\begin{vmatrix} s+\frac{1}{RC_2} & \frac{1}{n_2I} & \frac{1}{Rn_1} \\ -\frac{1}{n_2C_2} & s & 0 \\ 0 & -\frac{1}{I} & 0 \end{vmatrix}}{\det(sI-A)} = \frac{\frac{1}{Rn_1}\left(\frac{1}{n_2C_2I}\right)}{\det(sI-A)}$$

$$C = \begin{bmatrix} \frac{1}{Rn_1}\left(s^2 - \frac{1}{IC_3}\right) \\ \frac{1}{Rn_1}\left(-\frac{1}{n_2C_2}s\right) \\ \frac{1}{Rn_1}\left(\frac{1}{n_2C_2I}\right) \end{bmatrix} \frac{1}{\det(sI-A)} \Leftarrow \text{Transfer Function}$$

The original Model for the Fluid Damper is Correct. However, the inertia and resistance of the material is negligible [reference below] A simpler model, which doesn't include inertia effects of the resilient material has been developed. [Ref]



$f_1 = F$

$e_2 = \frac{1}{C_1} q_2$

$e_3 = \frac{1}{R1} f_3$

$e_4 = n e_5$

$f_5 = n f_4$

$e_6 = \frac{1}{C_2} q_6$

$e_8 = \frac{1}{C_3} q_8$

$f_9 = R2 e_9$

$e_1 - e_2 - e_3 - e_4 = 0$

$f_1 = f_2 = f_3 = f_4$

$e_5 = e_6 = e_7$

$f_5 - f_6 - f_7 = 0$

$e_7 - e_8 - e_9 = 0$

$f_7 = f_8 = f_9$

$\dot{q}_2 = f_2 = F$

$\dot{q}_6 = f_6 = f_5 - f_7 = nf_4 - R2e_9 = nF - R_2(e_7 - e_8)$

$= nF - R_2\left(\frac{1}{C_2}q_6 - \frac{1}{C_3}q_8\right)$

$\dot{q}_8 = f_8 = f_9 = R_2 e_9 = R_2(e_7 - e_8) = R_2\left(\frac{1}{C_2}q_6 - \frac{1}{C_3}q_8\right)$

$$\begin{bmatrix} \dot{q}_2 \\ \dot{q}_6 \\ q_8 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & -R_1/C_2 & 1/C_3 \\ 0 & R_2/C_2 & -1/C_3 \end{bmatrix} \begin{bmatrix} q_2 \\ q_6 \\ q_8 \end{bmatrix} + \begin{bmatrix} 0 \\ n \\ 0 \end{bmatrix} F$$

Third order system, that gives natural frequency of:

$$\frac{A_p^2}{C2} = \frac{j\omega + 1/R2C2}{j\omega + \frac{1}{R_2}\left(\frac{1}{C_2} + \frac{1}{C_3}\right)}$$

where

A_p - piston (orifice)

I_f = inertia (resilient material)

$C_1 = 4.517 \times 10^{-7}$ N/m $R = 8342.7$ $N \cdot s^2/m$

$C_2 = 4.517 \times 10^{-7}$ N/m

Natural Frequency

$$\omega_N^2 = \frac{1}{R}\left(\frac{1}{C2} + \frac{1}{C3}\right)$$

$$= \left(\frac{1}{8342.7}\right)\left(\frac{1}{4.517 \times 10^{-7}}\right)\left(\frac{1}{4.517 \times 10^{-7}}\right)$$

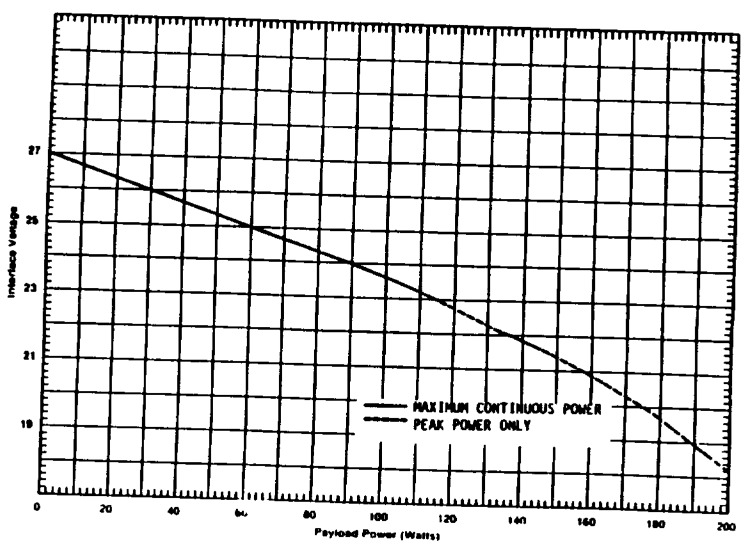
ω_N = 22.8 Hz. Not close to our range

From the marketed Fluidlastic Mount, explained earlier, the results conclude that the frequencies obtained from the damper are high frequencies. Therefore, the mount cannot be applied to the exercise cycle.

APPENDIX H
ELECTROMAGNETIC DAMPERS

As part of the feasibility of electromagnetic dampers, the following information from the Shuttle Payload Interface Definition Document is usefull in the design.

Concerning Current the following graph shows that max current (briefly) is 10A @ 19V and .37A @ 26V.



INTERFACE VOLTAGE VERSUS POWER

Additional Guidlines:

- Operating Temperatures
  - Max. External Accesable Surfaces <113°F
  - Max External INAccesable Surfaces <120°F

Electromagnetic Compatibility
  - Limitations on Electric Field Emissions
  - IDD section 8

Appendix ~~J2~~ I1: Technical Decision Matrix

Design Parameters / Weighing Factors / Alternatives	1	2	3	4	5	6	Rate	Total
	.1	.15	.2	.05	.25	.25		1.0
Resilient Absorbers	4 / .4	9 / 1.35	9 / 1.8	5 / .25	8 / 2.0	9 / 2.25	2	8.1
Isolation Box	6 / .6	6 / .9	10 / 2	9 / .45	10 / 2.5	9 / 2.25	1	8.7
Air Bearing	6 / .6	6 / .9	7 / 1.4	7 / .35	7 / 1.75	2 / .5	4	5.5
Fluid Bladder Isolator	6 / .6	7 / 1.05	7 / 1.4	7 / .35	2 / .5	6 / 1.5	5	0.5
Electro-magnet	8 / .8	7 / 1.05	7 / 1.4	8 / .4	7 / 1.75	2 / .5	3	5.9

Appendix ~~J2~~ I2: Weighing Factors- Method of Pairs

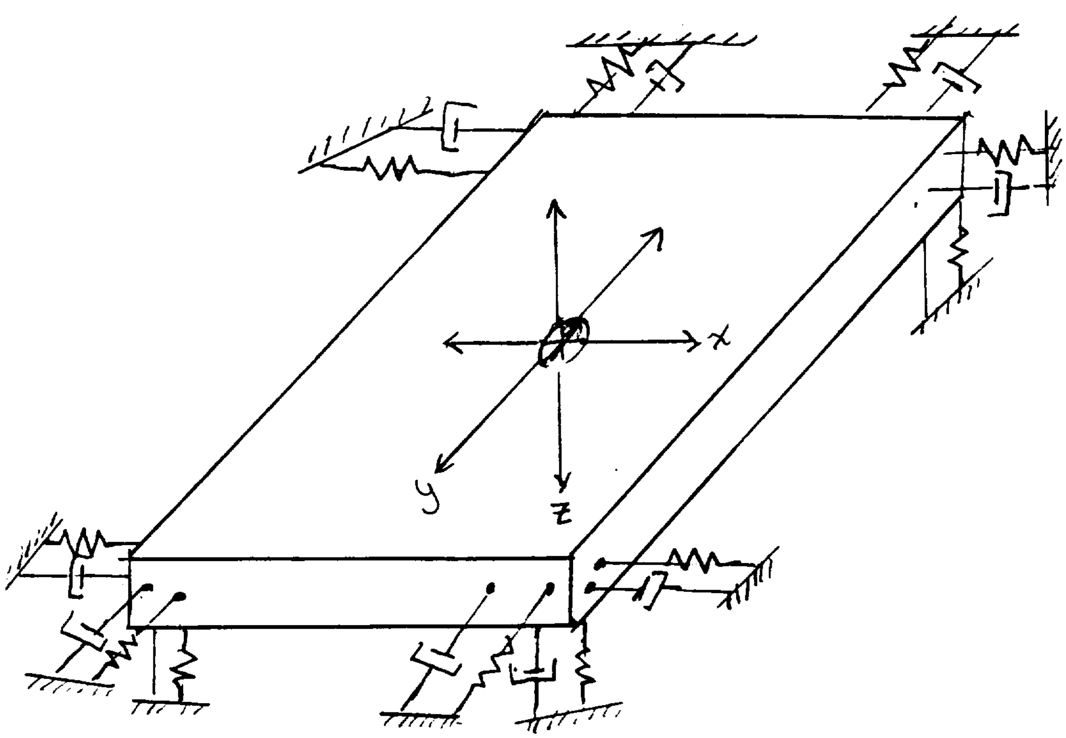
1. Mass
2. Energy Domains
3. Space Contraints
4. Multi-Directional Damping
5 Wear
6. Isolation Damping Ability
7. Time Constraint

	1	2	3	4	5	6	7
1		1	3	4	1	6	7
2			3	4	5	6	7
3				4	3	3	7
4					4	6	7
5						6	7
6							6
7							

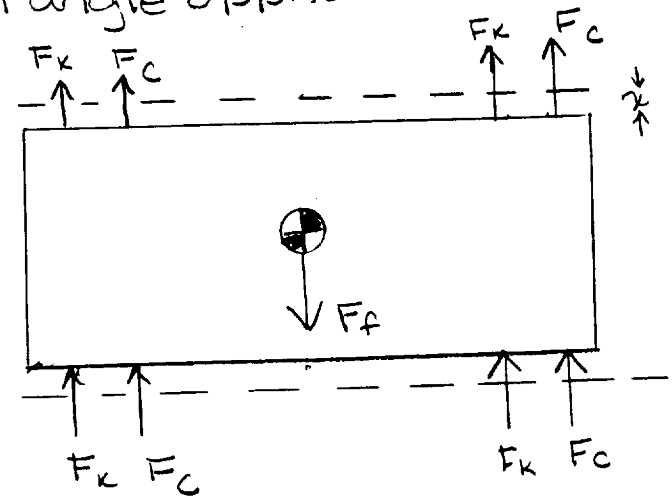
Appendix J

Three Dimensional Model of Isolation Box

Spring and Damper located at each corner of the lumped mass. Each system can be viewed separately.



Free body Diagram of Isolation Box using small angle approximations.



$$\Sigma F = m\ddot{x}$$

$$m\ddot{x} = -4F_k - 4F_c + F_f$$

$$m\ddot{x} = -4(K_x x) - 4(C_x \dot{x}) + F_f$$

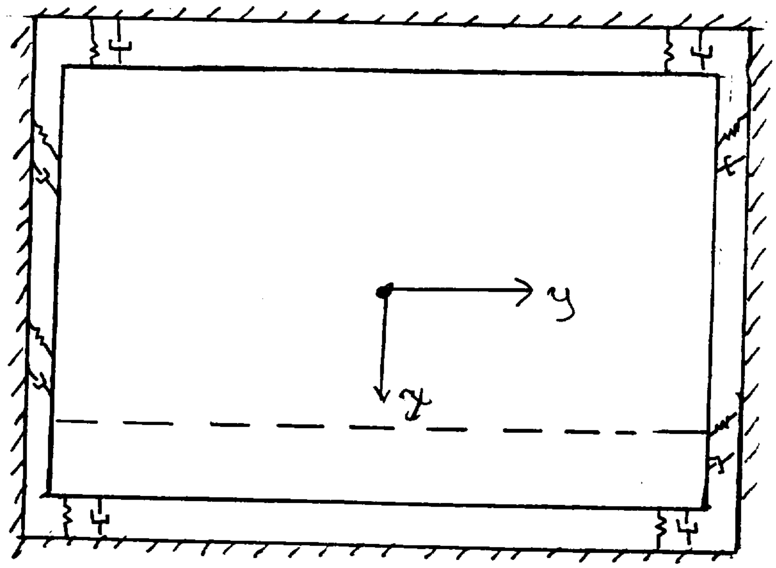
$$m\ddot{x} + 4C_x\dot{x} + 4K_x x = F_{f_x}$$

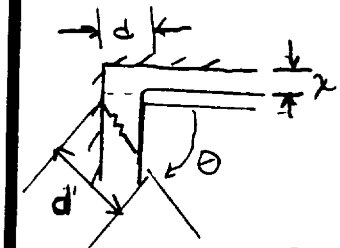
From symmetry, we can obtain equations for the y and z direction as follows:

$$m\ddot{y} + 4C_y y + 4K_y y = F_{fy}$$

$$m\ddot{z} + 4C_z z + 4K_z z = F_{fz}$$

Small Angle Approximation.





$$F_x = F_n \sin\theta$$
$$F_n = k(d' - d)$$

$$d' = x/\sin\theta$$
$$d = x/\tan\theta$$

$$\begin{aligned}
F_n &= k(d' - d) \\
&= k(x/\sin\theta - x/\tan\theta) \\
&= kx(1/\sin\theta - 1/\tan\theta) \\
&= kx(1/\sin\theta - \cos\theta/\sin\theta) \\
&= kx(1 - \cos\theta/\sin\theta)
\end{aligned}$$

$$\begin{aligned}
F_x &= F_n \sin\theta \\
&= kx\cancel{\sin\theta}(1 - \cos\theta/\cancel{\sin\theta}) \\
&= kx(1 - \cos\theta)
\end{aligned}$$

J4

Assuming θ to be small,

$\cos\theta \rightarrow 1$

$F_x = kx(1-1) = 0$

In other words $F_x = 0$ under small angle Approximations.

Neglect springs in directions normal to displacement direction for Isolation Box.

We can then easily obtain the state equations are:

$$\dot{p}_x = F_x - \frac{4C_x p_x}{m} - 4K_x x$$

$$\dot{x} = p_x/m$$

$$\dot{p}_y = F_y - \frac{4C_y p_y}{m} - 4K_y y$$

$$\dot{y} = p_y/m$$

$$\dot{p}_z = F_z - 4\frac{C_z p_z}{m} - 4K_z z$$

$$\dot{z} = p_z/m$$

Angular momentum:

$$\dot{h}_x = M_x - \left(\frac{4C_z l^2}{J}\right) h_x - (4K_z l^2)\,\theta_x$$

Angular displacement:

$$\dot{\theta}_x = h_x/J$$

$$\dot{h}_y = M_y - \left(\frac{4C_z w^2}{J}\right) h_y - (4K_z w^2)\,\theta_y$$

$$\dot{\theta}_y = h_y/J$$

$$\dot{h}_z = M_z - \left[\frac{4(C_y a + C_x b)}{J}\right] h_z - 4(K_y a^2 + K_x b^2)\,\theta_z$$

$$\dot{\theta}_z = h_z/J$$

In matrix form,

$$\begin{bmatrix} \dot{p}_x \\ \dot{x} \end{bmatrix} = \begin{bmatrix} -4k/m & -4k \\ 1/m & 0 \end{bmatrix} \begin{bmatrix} p_x \\ x \end{bmatrix} + \begin{bmatrix} 1 \\ 0 \end{bmatrix} F_y \quad x\text{-direction}$$

$$\begin{bmatrix} \dot{p}_{z} \\ \dot{y} \end{bmatrix} = \begin{bmatrix} -4C_z/m & -4_z \\ 1/m & 0 \end{bmatrix} \begin{bmatrix} p_z \\ p_z \end{bmatrix} \begin{bmatrix} 1 \\ 0 \end{bmatrix} F_z \quad z\text{-direction}$$

Moments:

$$\begin{bmatrix} \dot{h}_x \\ \dot{\theta}_x \end{bmatrix} = \begin{bmatrix} -4C_z w^2 & -4k_z w^2 \\ 1/J & 0 \end{bmatrix} \begin{bmatrix} h \\ \theta \end{bmatrix} + \begin{bmatrix} 1 \\ 0 \end{bmatrix} M_x$$

$$\begin{bmatrix} \dot{h}_y \\ \dot{\theta}_y \end{bmatrix} = \begin{bmatrix} -4C_z w^2 & -4k_z w^2 \\ 1/J & 0 \end{bmatrix} \begin{bmatrix} h_y \\ \theta_y \end{bmatrix} + \begin{bmatrix} 1 \\ 0 \end{bmatrix} M_y$$

$$\begin{bmatrix} \dot{h}_z \\ \dot{\theta}_z \end{bmatrix} = \begin{bmatrix} -\frac{4(C_y a + C_x b)}{J} & -4(k_y a^2 + k_x b^2) \\ 1/J & 0 \end{bmatrix} \begin{bmatrix} h_z \\ \theta_z \end{bmatrix} + \begin{bmatrix} 1 \\ 0 \end{bmatrix} M_z$$

for x, y, z directions, respectively.

Determine the transfer function from the force and moment state equations.

<u>In the x-direction:</u>

$$sI - A = \begin{bmatrix} s + 4C_x/m & 4k_x \\ -1/m & s \end{bmatrix}$$

$$\det(sI - A) = s^2 + 4C_x/m\, s + 4k/m$$

$$C_{11} = \frac{\begin{vmatrix} 1 & 4k_x \\ 0 & s \end{vmatrix}}{s^2 + {}^{4C_x}/_m\, s + {}^{4k_x}/_m} = \frac{s}{s^2 + {}^{4C_x}/_m\, s + {}^{4k_x}/_m}$$

$$C_{21} = \frac{\begin{vmatrix} s + \frac{4C_x}{m} & 1 \\ -\frac{1}{m} & 0 \end{vmatrix}}{s^2 + {}^{4C_x}/_m + {}^{4k_x}/_m} = \frac{{}^1/_m}{s^2 + {}^{4C_x}/_m\, s + {}^{4k_x}/_m}$$

$$C_x = \begin{bmatrix} \dfrac{s}{s^2 + {}^{4C_x}/_m\, s + {}^{4k_x}/_m} \\[2ex] \dfrac{{}^1/_m}{s^2 + {}^{4C_x}/_m\, s + {}^{4k_x}/_m} \end{bmatrix} \Leftarrow \text{Transfer Function}$$

<u>y-direction</u>

$$sI - A = \begin{bmatrix} s + {}^{4C_x}/_m & 4k_x \\ -{}^1/_m & s \end{bmatrix}$$

$$\det(sI - A) = s^2 + \frac{4C_y}{m} s + {}^{4k_y}/_m$$

$$C_{11} = \frac{\begin{vmatrix} 1 & 4k_y \\ 0 & s \end{vmatrix}}{\det(sI - A)} = \frac{s}{s^2 + 4C_y/m\,s + 4k_y/m}$$

$$C_{21} = \frac{\begin{vmatrix} s + \frac{4C_y}{m} & 1 \\ 1/m & 0 \end{vmatrix}}{\det(sI - A)} = \frac{-1/m}{s^2 + 4C_y/m\,s + 4k_y/m}$$

$$C_y = \begin{bmatrix} \dfrac{s}{s^2 + 4C_y/m\,s + 4k_y/m} \\ \\ \dfrac{-1/m}{s^2 + 4C_y/m\,s + 4k_y/m} \end{bmatrix}$$ Transfer Function

<u>Z-direction</u>

$$sI - A = \begin{bmatrix} s + \frac{4C_z}{m} & 4k_z \\ -\frac{1}{m} & s \end{bmatrix}$$

$$\det(sI - A) = s^2 + 4C_z/m\,s + 4k_z/m$$

$$C_{11} = \frac{\begin{vmatrix} 1 & 4k_z \\ 0 & s \end{vmatrix}}{s^2 + \frac{4C_z}{m}s + 4k_z/m} = \frac{s}{s^2 + 4C_z/m\,s + 4k_z/m}$$

$$C_{21} = \frac{\begin{vmatrix} s + \frac{4C_z}{m} & 1 \\ -1/m & 0 \end{vmatrix}}{s^2 + \frac{4C_z}{m}s + \frac{4k_z}{m}} = \frac{1/m}{s^2 + \frac{4C_z}{m}s + 4k_z/m}$$

$$C_z = \begin{bmatrix} \dfrac{s}{s^2 + \frac{4C_z}{m}s + 4k_z/m} \\[2ex] \dfrac{1/m}{s^2 + 4C_z/m\,s + 4k_z/m} \end{bmatrix}$$ Transfer Function

Moments

X-direction

$$\det(sI - A) = \begin{vmatrix} s + \frac{4C_z w^2}{J_x} & 4k_z w^2 \\ -1/J_x & s \end{vmatrix}$$

$$= s^2 + \frac{4C_z w^2}{J_x}s + \frac{4k_z w^2}{J_x}$$

$$C_{11} = \frac{\begin{vmatrix} 1 & 4K_2W^2 \\ 0 & s \end{vmatrix}}{\det(sI-A)} = \frac{s}{s^2 + \frac{4C_2W^2}{J_x}s + \frac{4K_2W^2}{J_x}}$$

$$C_{21} = \frac{\begin{vmatrix} s+\frac{4C_2W^2}{J_x} & 1 \\ -1/J_x & 0 \end{vmatrix}}{\det(sI-A)} = \frac{1/J_x}{s^2 + \frac{4C_2W^2}{J_x}s + \frac{4K_2W^2}{J_x}}$$

$$C_{mx} = \begin{bmatrix} \dfrac{s}{s^2 + \frac{4C_2W^2}{J_x}s + \frac{4K_2W^2}{J_x}} \\ \\ \dfrac{1/J_x}{s^2 + \frac{4C_2W^2}{J_x}s + \frac{4K_2W^2}{J_x}} \end{bmatrix}$$ Transfer Function

y-direction

$$\det(sI-A) = \begin{vmatrix} s+\frac{4C_2W^2}{J_y} & 4K_2W^2 \\ -\frac{1}{J_y} & s \end{vmatrix} = s^2 + \frac{4C_2W^2}{J_y}s + \frac{4K_2W^2}{J_y}$$

$$c_{11} = \frac{\begin{vmatrix} 1 & 4k_z w^2 \\ 0 & s \end{vmatrix}}{\det(sI - A)} = \frac{s}{s^2 + \frac{4C_y w^2}{J_y} s + \frac{4k_z w^2}{J_y}}$$

$$c_{21} = \frac{\begin{vmatrix} s + \frac{4C_y w^2}{J_y} & 1 \\ -1/J_y & 0 \end{vmatrix}}{\det(sI - A)} \qquad \frac{1/J_y}{s^2 + \frac{4C_y w^2}{J_y} s + \frac{4k_z w^2}{J_y}}$$

$$C_{My} = \begin{bmatrix} \dfrac{s}{s^2 + \frac{4C_y w^2}{J_y} s + \frac{4k_z w^2}{J_y}} \\[2ex] \dfrac{1/J_y}{s^2 + \frac{4C_y w^2}{J_y} s + \frac{4k_z w^2}{J_y}} \end{bmatrix}$$ Transfer Function

Z-direction

$$\det(sI - A) = \begin{vmatrix} s + \frac{4C_y a + C_x b}{J_z} & 4(k_y a^2 + k_x b^2) \\ 1/J_z & s \end{vmatrix}$$

$$= s^2 + \frac{4(C_y a + C_x b)}{J_z} s + \frac{4(k_y a^2 + k_x b^2)}{J_z}$$

J12

$$C_{11} = \frac{\begin{vmatrix} 1 & 4(k_y a^2 + k_x b^2) \\ 0 & s \end{vmatrix}}{\det(sI - A)} = \frac{s}{\det(sI-A)}$$

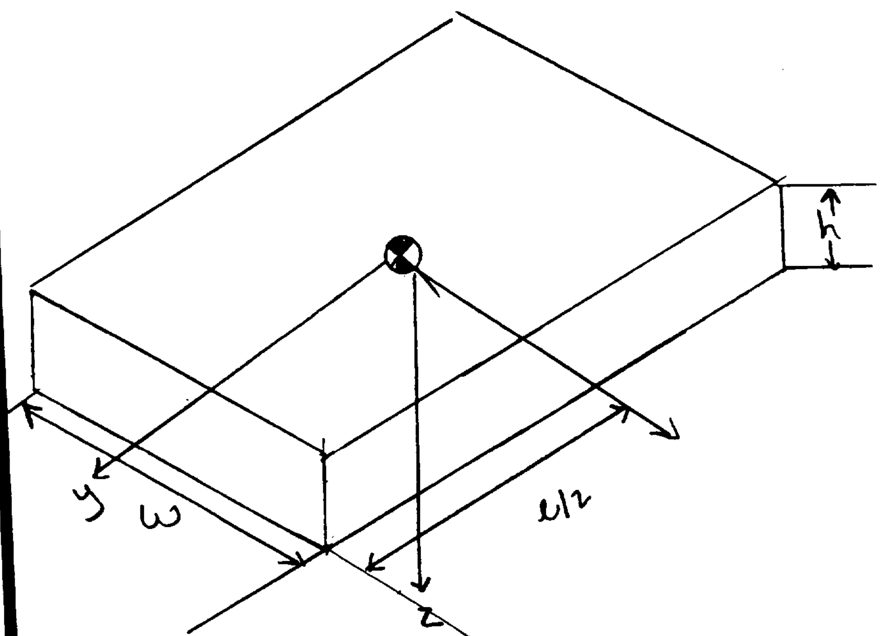
$$C_{21} = \frac{\begin{vmatrix} s + 4(c_y a + c_x b) & 1 \\ 1/J_z & 0 \end{vmatrix}}{\det(sI - A)} = \frac{1/J_z}{\det sI - A}$$

$$\begin{bmatrix} \dfrac{s}{s^2 + \dfrac{4(c_y a + c_x b)}{J_z} s + \dfrac{4(k_y a^2 + k_x b^2)}{J_z}} \\ \dfrac{1/J_z}{s^2 + \dfrac{4(c_y a + c_x b)}{J_z} + \dfrac{4(k_y a^2 + k_x b^2)}{J_z}} \end{bmatrix}$$

a: length
b= Width

This says that the system we're considering can't attenuate the frequency below 3 Hz. At frequencies near 5 Hz ($f > 4.8$), the system can isolate vibrations up to 80% with correct selection of k.
The response will be the same for each direction of forces because the transfer functions are same except different values of K

For the rotation



Assume some direction:

$\ell = 23.63'' = .6\,m$

$W = 15.74'' = .4\,m$

$h = 16.0'' = .4\,m$

$W = 250\,lbs = 1112.5\,N$
$= 113.5\,kg$

$J_x = \frac{1}{12} m(h^2+\ell^2) = 4.918\,kgm^2$

$J_y = \frac{1}{12} m(w^2+h^2) = 3.026\,kgm^2$

$J_z = \frac{1}{12} m(w^2+\ell^2) = 4.92\,kgm^2$

From characteristic polynomial:

$$w_n = \sqrt{\frac{4(k_y\ell^2 + k_x w^2)}{J_z}}$$

$K = 30\ N/m$

$K = 10\ N/m$

$K = 1\ N/m$

Assume $k_1 = k_x$
$\ell = 0.6\,m$
$w = 0.4\,m$
$J_z = 4.92\,kg/m^3$

	$K_y = K_x$	η	ω_n	ω_n^2	$2\eta\omega_n$	$T(\omega)$ $\omega = 1$ Hz	$\omega = 5$ Hz	$\omega = 3$ Hz
ROTATION ABOUT Z-AXIS	30 N/m	0.3	3.56		2.136	0.126	0.251	0.5
	10 N/m	0.3	2.05	4.23	1.23	0.447	0.178	0.398
	1 N/m	0.3	0.65	0.42	0.39	1.43	0.75	0.316
TRANSLATION X-direction	30 N/m	0.3	1.03	1.06	2.17	0.75	0.2	0.40
	10 N/m	0.3	.65	0.29	.598	0.778	0.178	0.316
	1 N/m	0.3	.19	0.35	0.11	1.0	0.178	0.316

Table 1 - Data from Bode Plots

From these figures in Table-1, The required attenuation is possible for some Frequencies but not all. The frequencies considered $1 < \omega < 5$ Hz represent our operating frequencies.

In addition, the spring constants needed to get the low natural frequency are unrealistically low. These spring constants are too low for our system. This is apparent by considering deflections resulting from a 445N force

For x-direction

$$|T(\omega)| = \frac{s}{s^2 + 2\eta\omega_n + \omega_n^2}$$

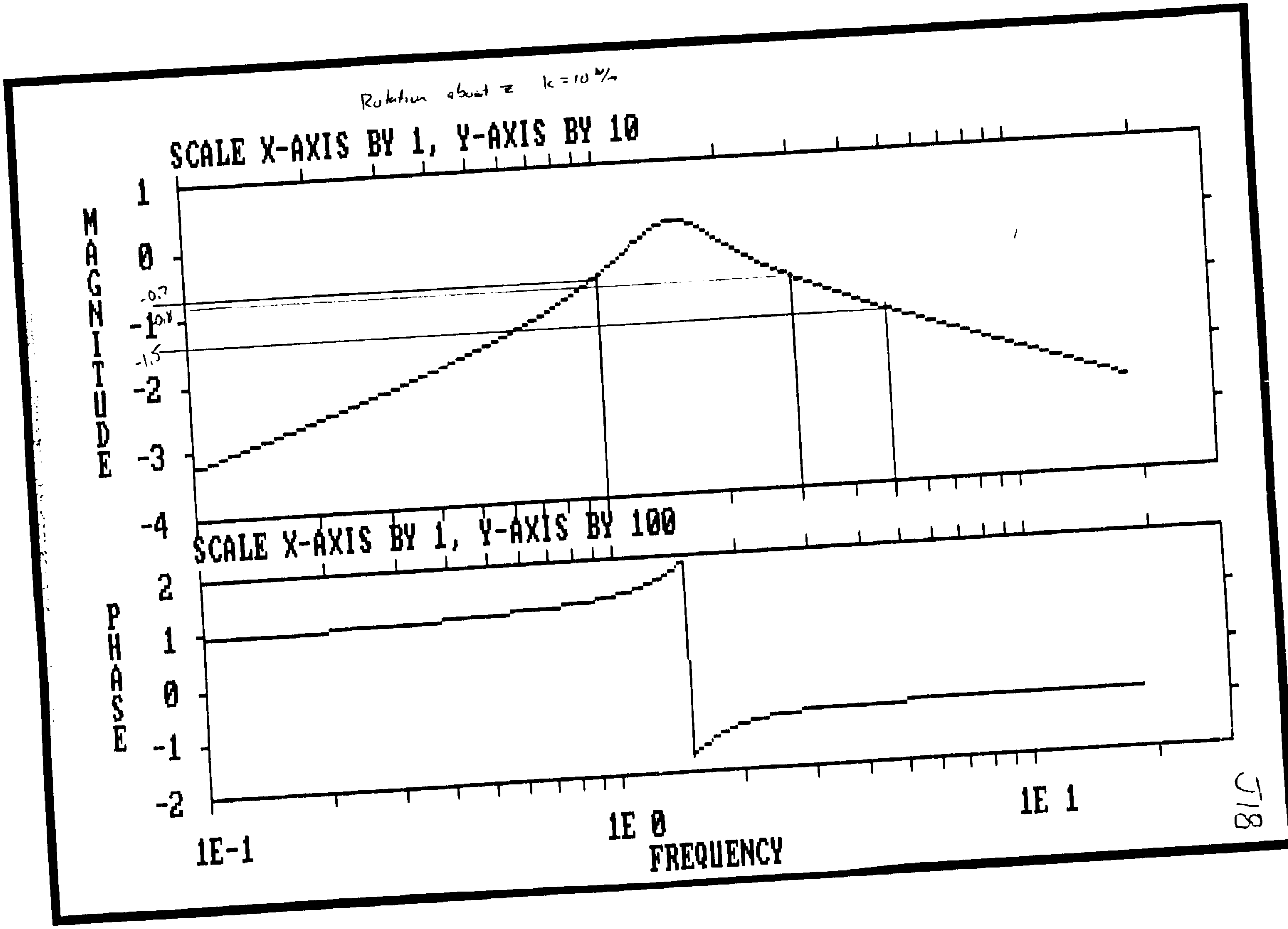
$$\omega_n = \sqrt{\frac{4K_x}{m}}$$

ω_n - natural frequency

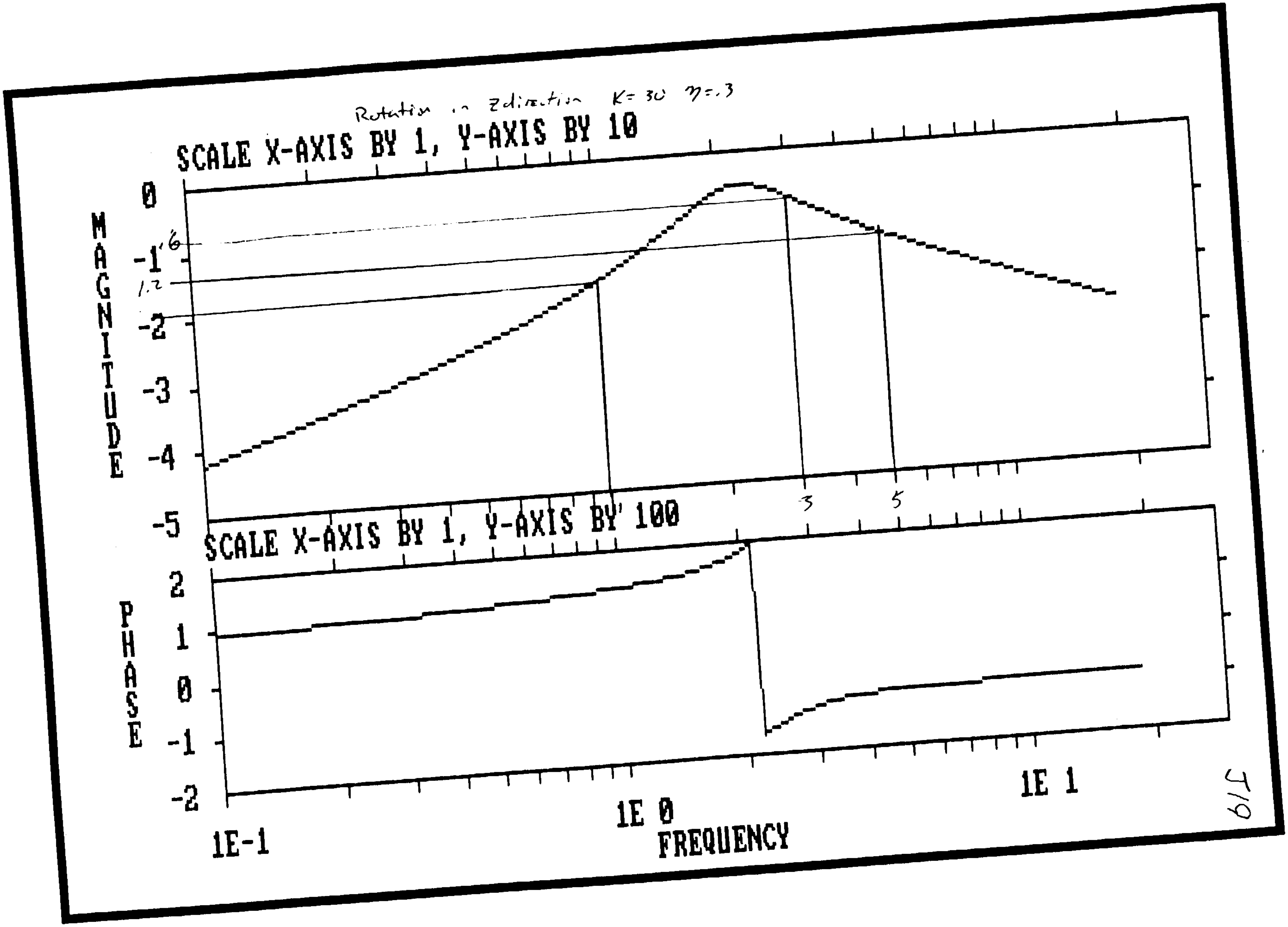
η - damping Ratio

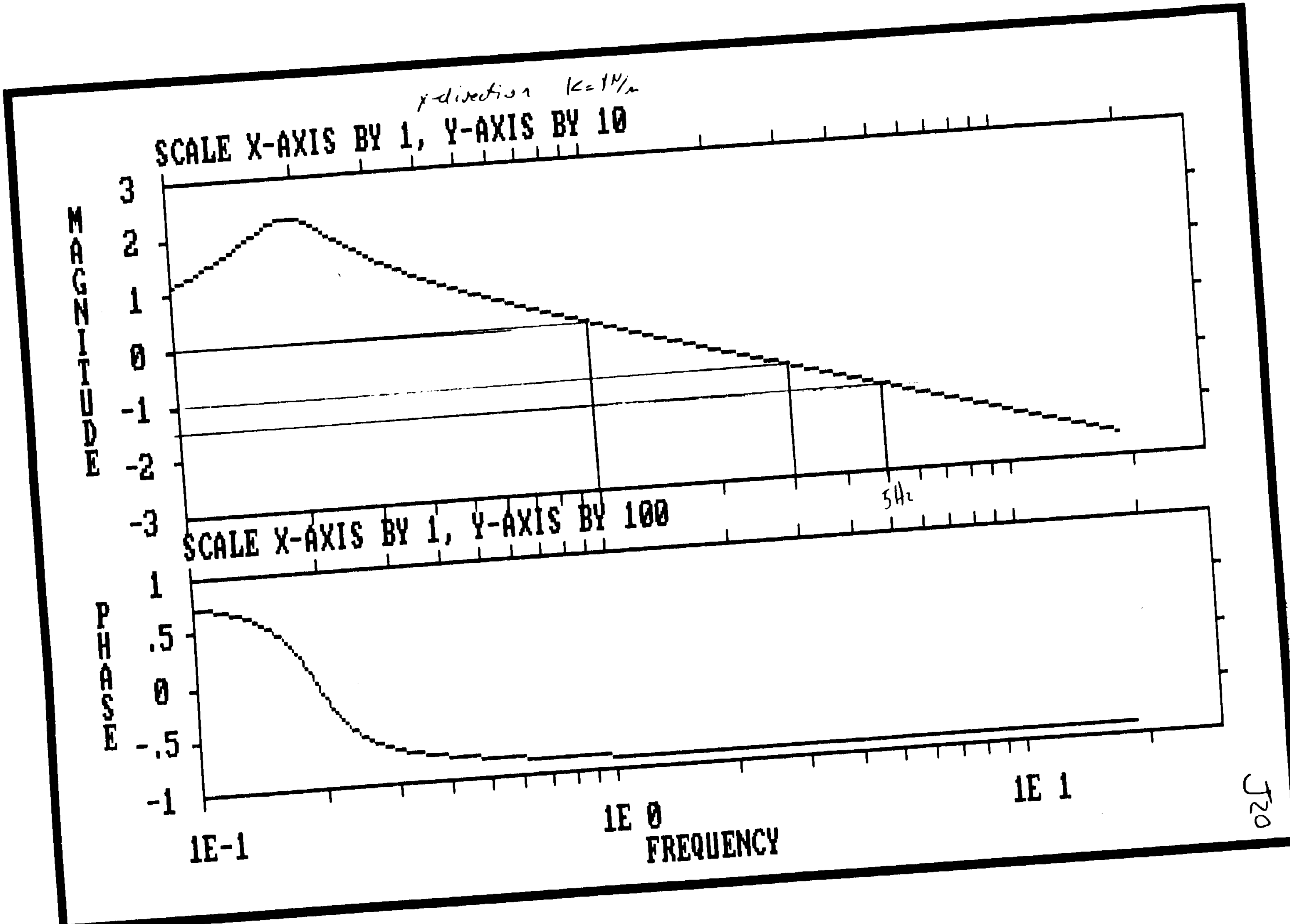
$m = 113.5\,kg$

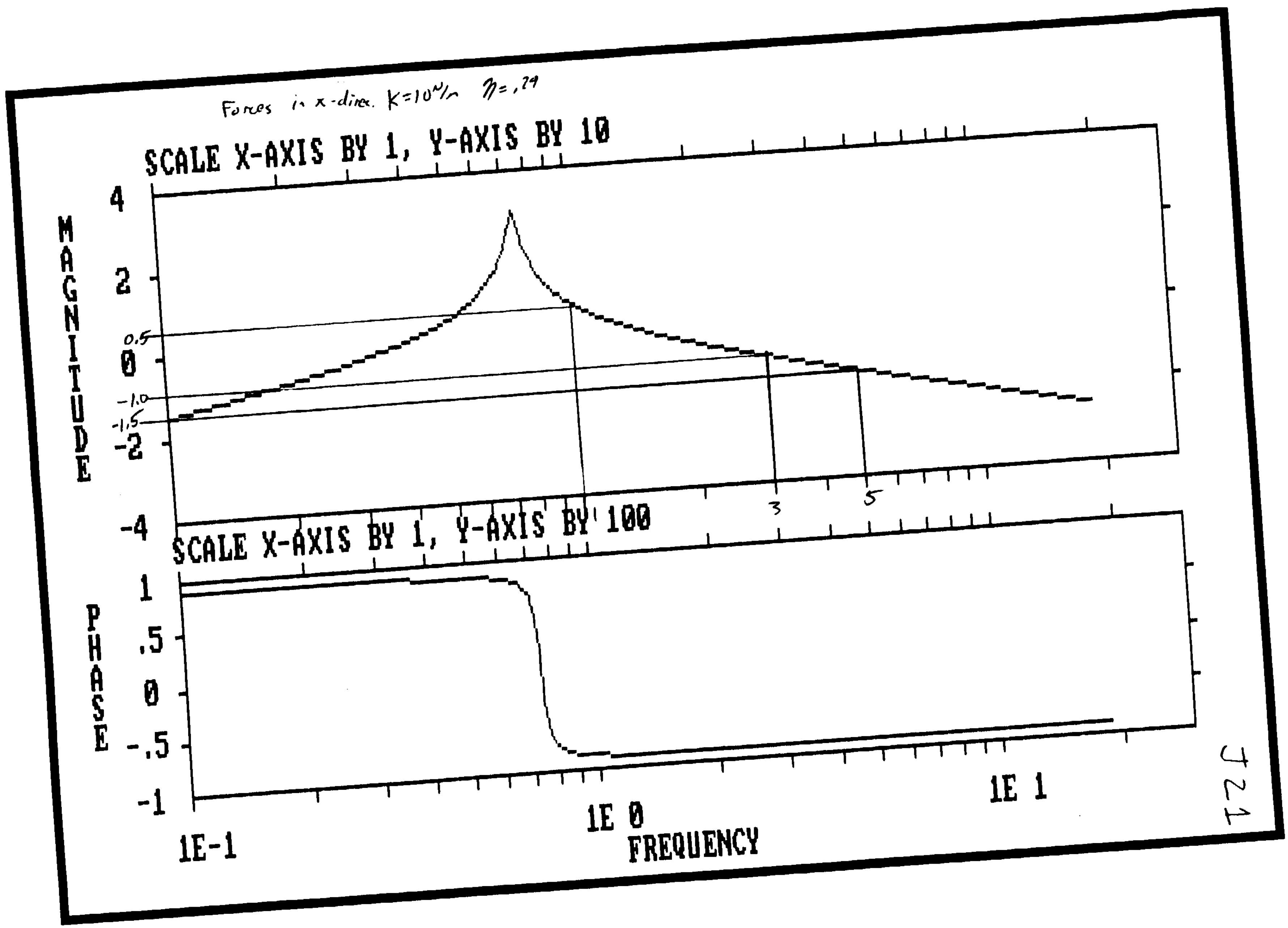
Representative figures are given on page

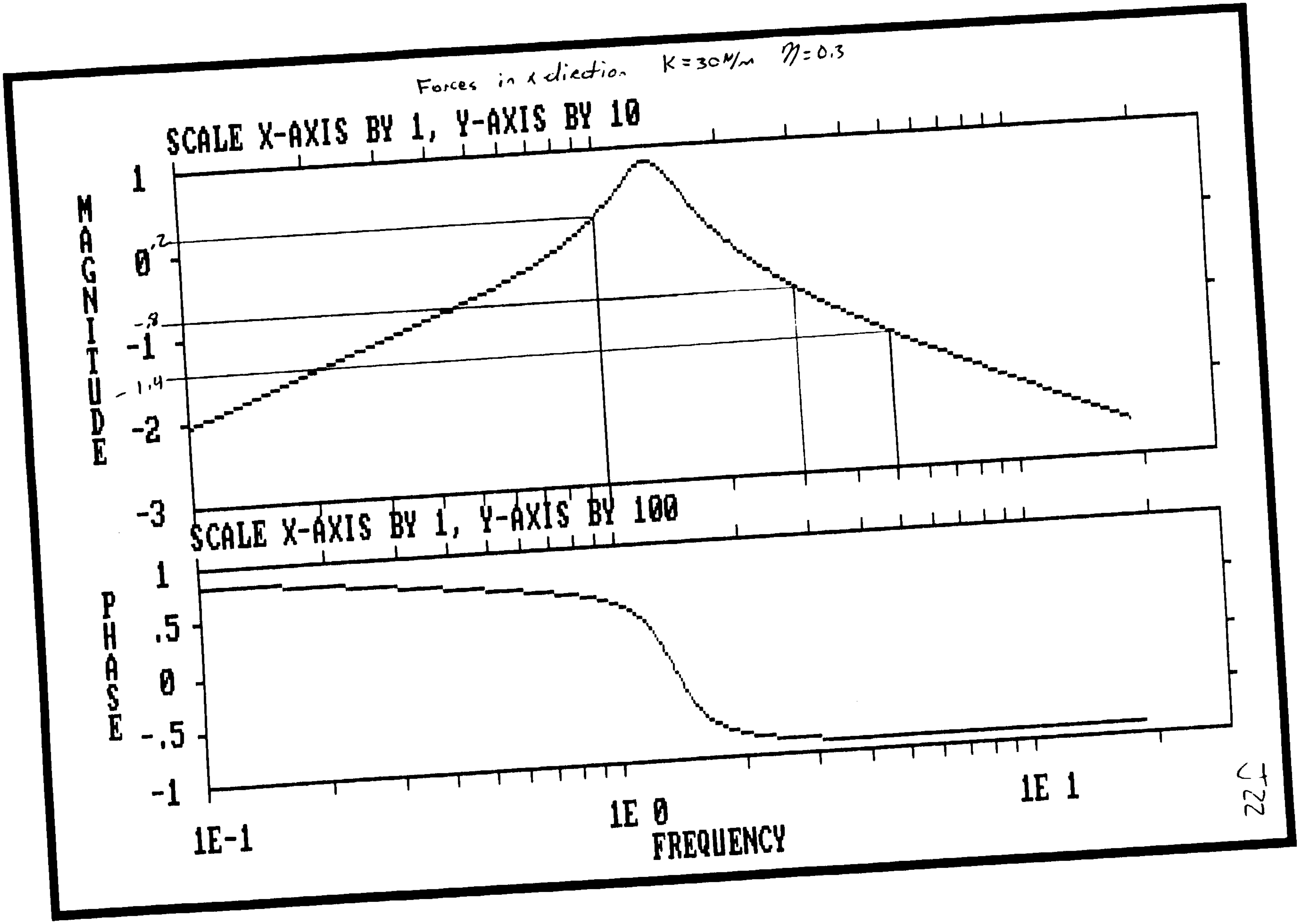


Rotation about z k = 10 N/m
SCALE X-AXIS BY 1, Y-AXIS BY 10
MAGNITUDE
1
0
-0.7
-0.8
-1
-1.5
-2
-3
-4
SCALE X-AXIS BY 1, Y-AXIS BY 100
PHASE
2
1
0
-1
-2
1E-1
1E 0
1E 1
FREQUENCY
J18


Rotation in Z direction K=30 η=.3
SCALE X-AXIS BY 1, Y-AXIS BY 10
MAGNITUDE
0
-1
-2
-3
-4
-5
1.6
1.2
3
5
SCALE X-AXIS BY 1, Y-AXIS BY 100
PHASE
2
1
0
-1
-2
1E-1
1E 0
1E 1
FREQUENCY
J19


x-direction K=1N/m
SCALE X-AXIS BY 1, Y-AXIS BY 10
MAGNITUDE
3
2
1
0
-1
-2
-3
5Hz
SCALE X-AXIS BY 1, Y-AXIS BY 100
PHASE
1
.5
0
-.5
-1
1E-1
1E 0
1E 1
FREQUENCY
J20


Forces in x-direc. K=10 N/m η=,29
SCALE X-AXIS BY 1, Y-AXIS BY 10
MAGNITUDE
4
2
0.5
0
-1.0
-1.5
-2
-4
3
5
SCALE X-AXIS BY 1, Y-AXIS BY 100
PHASE
1
.5
0
-.5
-1
1E-1
1E 0
1E 1
FREQUENCY
J21



Forces in x direction K = 30 N/m η = 0.3
SCALE X-AXIS BY 1, Y-AXIS BY 10
MAGNITUDE
1
0
.2
-.8
-1
-1.4
-2
-3
SCALE X-AXIS BY 1, Y-AXIS BY 100
PHASE
1
.5
0
-.5
-1
1E-1
1E 0
1E 1
FREQUENCY
J22