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DAMPING AND VIBRATION ISOLATION OF STRUCTURAL ENGINEERING PROBLEMS

A

DISSERTATION

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CANDIDATE'S DECLARATION

I hereby certify that the work which is being presented in this dissertation entitled **"DAMPING AND VIBRATION ISOLATION OF STRUCTURAL ENGINEERING PROBLEMS"** in partial fulfilment of the requirements for the award of the degree of Master of Technology with specialisation in Structural Engineering to Jawaharlal Nehru University, is an authentic record of my own work, under the supervision and guidance of Dr.V M Inamdar of Faculty of Civil Engineering, College of Military Engineering, Pune.

Dated: 30 November 1999

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CERTIFICATE

This is to certify that the dissertation entitled **"DAMPING AND VIBRATION ISOLATION OF STRUCTURAL ENGINEERING PROBLEMS"** that is being submitted by Major V Ramesh for the award of degree of Master of Technology in Structural Engineering to the Jawaharlal Nehru University, New Delhi, is a bonafide study carried out by him under my supervision and guidance and hence recommended for acceptance and approval.

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EXAMINER'S CERTIFICATE OF APPROVAL

The dissertation entitled **"DAMPING AND VIBRATION ISOLATION OF STRUCTURAL ENGINEEERING PROBLEMS"** submitted by **Major V Ramesh,** in partial fulfilment of the requirements of the degree of Master of Technology in Structural Engineering, to Jawaharlal Nehru University, New Delhi, is hereby approved for the award of the degree.

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Synopsis

Motions involving deformation of structures are caused by dynamic forces or dynamic disturbances. Dynamic forces may be induced by rotating machinery, wind, water waves or a blast. A dynamic disturbance may result from an earthquake during which the motion of the ground is transmitted to the supported structure..

There are numerous sources of vibration in an industrial environment: impact processes such as pile driving and blasting; rotating or reciprocating machinery such as engines, compressors and motors; transportation vehicles such as trucks, trains and aircraft; the flow of fluids and many others. The presence of vibration often leads to undesirable effects such as structural or mechanical failure, frequent and costly maintenance of machines, human pain and discomfort.

The analysis of vibration response is of considerable importance in the design of structures that may be subjected to dynamic disturbances. Under certain situations, vibrations may cause large displacements and severe stresses in the structure. The transmission of vibrations to connected structures may lead to undesirable results.

Vibration can be sometimes eliminated on the basis of theoretical analysis. However, the manufacturing costs involved in eliminating the vibration may be too high; a designer must compromise between an acceptable amount of vibration and a reasonable manufacturing cost. In case of lightly damped systems, it can be seen that even a relatively small excitation force can cause an undesirably large response near resonance.

In these cases, the magnitude of the response can be significantly reduced by the use of isolators and auxiliary mass absorbers.

Vibration control by the use of viscoelastic damping materials has in recent years grown from a specialised approach for the solution of difficult and expensive vibration problems in certain military aerospace systems into a widely used element in the package of structural and functional changes particularly needed to solve noise and vibration control problems in the field of general engineering such as automotive, diesel engine, office and transportation system production.

This work is intended to study damping and vibration isolation of structural engineering problems employing damping materials. We have used the properties of viscoelastic materials in the vibration analysis but a detailed analysis of their material behaviour is not our primary concern. This work addresses the vibration damping of structural elements and is not a materials-oriented work.

The first Chapter presents an elementary review of basic theory of damping. Chapters 2 and 3 describes the classification of damping and various vibration control techniques. Next chapter follows this by studying isolators for various types of equipment. In chapter 5, various seismic mountings are discussed. Chapter 6 touches briefly on acoustical control. The last chapter concludes the present study.

This study is compilation of research works done in the field of damping and vibration isolation of structural engineering problems. The material has been drawn from the vast wealth of available information.

CONTENTS

CHAPTER 1

INTRODUCTION AND BASIC PRINCIPLES

1.1 General

The response of physical objects to dynamic or time-varying loads is an important area of study in engineering. The physical object whose response is sought may either be treated as rigid body or considered to be deformable. The subject of rigid-body dynamics treats the physical objects as rigid bodies that undergo motion without deformation when subjected to dynamic loading. In many instances, however, dynamic response involving deformation rather than simple rigid-body motion is of primary concern. This is particularly so in the design of structures and structural frames.

Dynamic response involving deformation is usually oscillatory in nature in which the structure vibrates about a configuration of stable equilibrium. Such equilibrium configuration may be static or dynamic involving rigid-body motion. The approach of analysis could be either deterministic i.e. time variation of the loading is fully defined or non-deterministic i.e. time variation of loading is random.

Some of the instances where dynamic analysis is necessary are:-

- The structure that supports oscillating/reciprocating machinery.
- Bridge that supports moving loads.
- A structure that is subjected to a suddenly applied dynamic force such as blast and wind gust etc.
- Building foundation disturbed by the presence of some random activity like earthquake.
- Dynamic effect due to nuclear explosion.

1.2 Basics of Vibration Control

In engineering practice, controlling vibration and noise in structures and machines is partly art and partly science. This is because though one can in principle obtain from analysis or experiment the data needed to develop and optimise the appropriate control measures, in practice, one is constrained by factors of time, equipment and economics and is often obliged to make decisions concerning the control measures without having complete information. This means that a suitable guesswork is required, past experience must be drawn on and less than optimum measures must be sought which do the job without necessarily being the most perfect solution.

Vibration control exercise is basically of two types: -

a. To protect a building or structure from the vibration effects of equipment.

b. To protect equipment from the vibration effects of the building.

The first case covers the vibration isolation of equipment like fans, pumps, chiggers, etc., while the second case usually involves delicate items, such as electron microscopes, ultraprecision tools, etc. In both the cases, there is the additional requirement that the movement of the isolated equipment must be kept down to suitable limits.

1.3 Basic terms and definitions

Basic terminology that will be often used in the present work is as follows:

- Coulomb Damping Damping where the damping force is independent of velocity. Dry friction is a typical example.
- Hysteresis Damping Damping due to a non-reversible process, for example, when a rubber block is compressed and released it does not release all of the energy due to compression. If the process is repeated the rubber becomes hot due to the "lost" energy.
- Viscous Damping Damping in which the damping force is proportional to the velocity of movement. In theory, oil or air filled dashpots provide this form of damping.
- Damping Controlled System A system is said to be damping controlled when vibrating at its natural frequency. Under this condition, the effects of mass and stiffness cancel out and only damping controls the amplitude of the vibration.
- Stiffness Control A system is said to be stiffness controlled when it oscillates at frequencies below its natural frequency. Under these conditions it is the stiffness of the springs which controls the amplitude of vibration. Vibration isolators operated under these conditions will provide no isolation and may provide amplification.

- Vibration Isolator The preferred term to describe "antivibration mounts" etc. The devices do not prevent vibration, they isolate it.
- Transmissibility The proportion of the disturbing force which is transmitted through vibration isolators. It is normally expressed as percentage.
- Spring Constant or Spring Rate The force required to deflect the spring by unit distance, e.g. 1000kg/m. Stiffness is an alternative term for spring constant.
- Natural Frequency A frequency at which a system vibrates when disturbed from rest and then released. An idealised spring-mass system has one natural frequency. Real life systems have many natural frequencies.
- Shock It is generally used to describe a process in which an impulsive force produces a sudden change of velocity or position. It usually implies some form of impact which involves rapid transient transmission of energy.
- Mode Shape Deflected or displaced shape of a dynamic system when excited at one of its natural frequencies.
- Jerk Time rate of change of acceleration.

1.4 Basic Principles

The basic principle in both of the above cases is the same; the rigid connection between the vibrating system and the protected one is broken. To provide perfect isolation one would have to support the fan or the electron microscope on skyhooks. As this is not possible, some form of spring is used instead!

1.5 The Three Controlling Factors

The three controlling factors in a basic vibration isolation system are the stiffness of the springs, the mass of the suspended equipment, and the damping of the system. Their effects can be summed up as follows:

a. Stiffness

The springs provide the isolation; generally, the stiffer the springs, the less effective the vibration isolation. Rather surprisingly, the springs have little effect on the amplitude of motion for a properly operating vibration isolation system.

b. Mass

The mass keeps the suspended system still, the heavier the suspended mass, the smaller the movement for a given disturbing force. Since the heavier mass requires stronger springs to support it, for a given resonant frequency increasing the mass by means of inertia block does not reduce the transmitted force if the static deflection and resonant frequency are kept constant. It does, however reduce the movement of the suspended system.

c. Damping

Damping has three effects :-

i) It reduces the effect of resonance.

ii) It reduces the amplitude of motion of the suspended system at high frequencies.

iii) It tends to increase the spring stiffness by providing an additional connection short-circuiting the spring.

1.6 Angular Frequency

The equation of motion is merely an expression of the equilibrium of the inertia force, damping force and spring force and is given by:-

$$F_m + F_D + F_S = F(t) \qquad \textbf{(1.1)}$$

$$F_m = \text{Inertia force} = m \times \frac{d^2 y}{dx^2} = m\ddot{y}$$

$$F_D = \text{Damping force} = C \times \frac{dy}{dt} = C\dot{y}$$

$$F_S = \text{Spring force} = Ky$$

where,

m	=	mass of the system,	K	=	stiffness,
y	=	displacement,	C	=	damping coefficient,
$\dot{y}$	=	velocity, and			
$\ddot{y}$	=	acceleration			

Taking the simplest condition of undamped free vibration:-

In this case $C = 0$ and $F(t) = 0$;

$$\therefore \quad m\ddot{y} + k\dot{y} = 0$$

or, $\ddot{y} + \omega^2 y = 0$ **(1.2)**

Assume $\omega^2 = \frac{K}{m}$ where ω = Natural circular frequency (rad/sec).

General solution of Eq. (1.2) is

$$y = A_1 Sin\omega\, t + A_2 Cos\omega\, t$$

$$y = A_1 Sin\,(\omega\, t + 2\pi\,) + A_2 Cos(\omega\, t + 2\pi\,)$$

$$y = A_1 Sin\,\omega\left(t + \frac{2\pi}{\omega}\right) + A_2 Cos\,\omega\left(t + \frac{2\pi}{\omega}\right)$$

Period of vibration $= \dfrac{2\pi}{\omega} = T$,

Hence $T = 2\pi\sqrt{\dfrac{m}{K}}$ sec

Natural frequency (f) $= \dfrac{1}{T} = \dfrac{1}{2\pi}\sqrt{\dfrac{K}{m}}$ cps

1.7 Transmissibility and Isolation

The effectiveness of isolation can be expressed in terms of the proportion of the initial force, which is transmitted. This is known as the transmissibility

Transmissibility T= F_t / F_o **(1.3)**

where F_t is the transmitted force and F_o is the disturbing force. Vibration isolation occurs when T<1.

This is sometimes expressed in terms of efficiency as given below:

Efficiency η = (Fo-Ft)/Fo = 1-T **(1.4)**

Transmissibility is a better concept as it can be seen if the equivalent transmissibilities and efficiencies are set out side by side as below:

Disturbing Force(Fo) (kg)	Transmitted Force(Ft) (kg)	Transmissibility(T) %	Efficiency %
10	10	100	0
10	5	50	50
10	2	20	80
10	1	10	90
10	0.1	1	99
10	0.01	0.1	99.9

1.8 Causes of Vibration

There are two main areas which need to be examined. These are:-

- Causes of vibration in structures.
- Types of machinery producing vibration and their situation.

There are two main reasons for vibration in machinery both involving out-of-balance forces-directly rotating machinery which is statically or dynamically out-of-balance, such as motors and fans, etc., and reciprocating machinery which is inherently unbalanced, such as internal combustion engines and compressors etc. A third source is impact and shock from punch presses, etc.

All the above cause vibration as an unwanted by-product of their normal operation but there is a further category of machines which produce vibration deliberately as part of their operation. This includes vibrating screens, conveyors and bowl feeders for small parts in assembly.

1.9 Conclusion

All the above devices should be correctly isolated from the building structure and therefore cause no vibration problem. Normally, however, the isolation is not perfect - particularly if the support is steelwork and not rigid and they act as vibration sources. In practice, it is usually the first type and sometimes the second that create problems. In the next chapter, classification of damping is discussed in detail.

CHAPTER 2

CLASSIFICATION OF DAMPING

2.1 Introduction

Vibration damping is the process and techniques used for converting the mechanical vibrational energy of solids into heat energy. While vibrational damping is helpful under conditions of resonance, it may be detrimental in many instances to a system at frequencies above the resonant point. This is due to the fact that the relative motion between the base of the vibration isolator and the mounted body tends to become smaller as the isolator becomes more efficient at the higher frequencies. With damping present, the force transmitted by the elastic element is unable to overcome the damping force; this leads to a resulting increase in transmissibility.

All metal springs which include structural members such as brackets and shelves have some damping. However, such damping is insufficient for vibration isolators and must be augmented by special damping devices.

2.2 Classification of Damping

Damping can be classified into many different types. For example, comparing different materials A and B under exactly the same conditions (same boundary conditions, same geometrical dimensions), the same magnitude of periodic forcing function with the same frequency of excitation, material A may oscillate longer (or shorter) with larger (or smaller) amplitude than material B. This is primarily due to the difference in material properties. The damping force due to internal molecular friction is less (or more) than the damping force due to internal molecular friction in material B. This kind of damping is called *material (or structural) damping*.

Another type of damping encountered in a vibrating system is introduced through the surrounding medium in which the vibration takes place. For example, a vibratory structural system will oscillate much longer in air than in water. This kind of damping is called *viscous damping*. The viscous damping force depends on the property of the surrounding medium and the velocity of motion. In general, one can classify damping into two basic categories: nonmaterial damping and material damping.

2.3 Nonmaterial Damping

2.3.1 Viscous Damping

Probably, the most familiar example of this kind of damping is the shock absorber of an automobile. The system is shown in Fig 2.1.

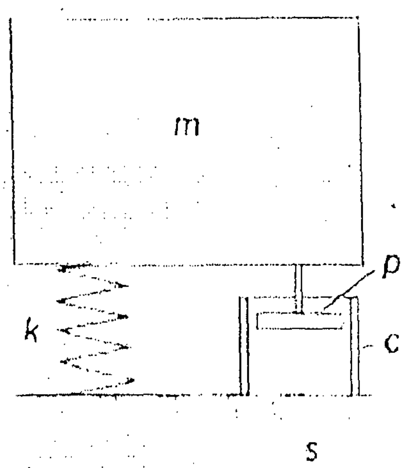


Fig 2.1. Automobile shock absorber

A piston p is attached to the body m and is arranged to move vertically through the liquid in a cylinder c which is secured to the support s. As the piston moves, the force required to cause the liquid to flow from one side of the piston to the other is approximately proportional to the velocity of the piston in the cylinder. The damping force is controlled by the viscosity of the liquid and the size of the orifice in the piston. There are several disadvantages of this type of damping; for example it is unidirectional; it is affected by temperature changes and because of the fact that the liquid is passed from one side of the piston to the other side through an opening it is time conscious. The opening, whether it is an orifice or the clearance between piston and cylinder, can pass only so much liquid in a given length of time. If the body to which the piston is attached is caused to displace faster than the liquid can transfer, a bottoming effect occurs. This effect is experienced by the riders in automobiles when a hole in the street is hit at too fast a rate; the springs may appear to bottom out, but actually it is the shock absorber or damper.

Some of the disadvantages of viscous damping may be overcome by using air instead of liquid as the damping medium. Air being compressible will add to the effective spring force with large displacements. If the air is housed within a flexible bellows, damping will be attainable horizontally as well as vertically. Such a system is illustrated in Fig 2.2.

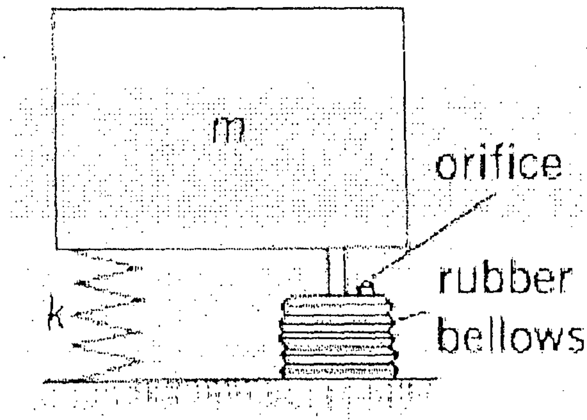


Fig 2.2. Shock absorbing system employing viscous damping with air

The viscous damping force is a function of the velocity of vibration, and is due to fluid resistance during oscillation. Generally, their mathematical description is quite complicated and not suitable for vibration analysis. Fortunately, a simplified version of a damping model was developed so that manageable mathematical solutions can be achieved for engineering purposes. This viscous damping model, designated by the dashpot, is very easy to use in analysing engineering vibration problems.

In this simplified viscous model, the damping force F_d is assumed to be linearly proportional to the velocity of a particle moving in fluid medium. Thus the viscous damping force (F_d) is expressed by the equation

$$F_d = c\, dx/dt \qquad \textbf{(2.1)}$$

where c is a proportional constant called the coefficient of viscosity, and dx/dt is the velocity of the particle relative to the fluid. One of the major objectives is to analyse the effect of material damping on the vibrational behaviours of the structural elements by using a constrained viscoelastic damping layer.

For systems with viscous damping force F_d, the energy dissipated (W_d) per cycle of vibration is equal to

$$W_d = \int F_d \, dx \qquad (2.2)$$

where F_d = Damping force

dx = Small distance over which velocity of the particle is measured

2.3.2 Coulomb damping

Coulomb damping, also called *dry damping* is produced from the sliding of two dry surfaces. The Coulomb damping force, F_c, is equal to the product of the normal force between the two surfaces, N, and the coefficient of friction μ:

$$F_c = \mu N \qquad (2.3)$$

The Coulomb damping force is assumed to be independent of the relative velocity of motion between the two surfaces. The sign of the damping force is always opposite to that of the velocity. The concept of coulomb damping is usually applied in structural joints. Damping force is introduced from slipping at the joint, and this gives rise to energy dissipation at the joint. Coulomb damping can also be applied for the two layer-beam.

A damper of this type is shown in Fig 2.3.

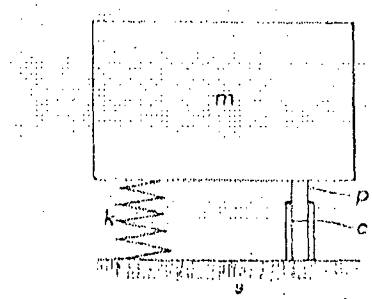


Fig 2.3. System employing Coulomb damping

A pin p is inserted in cylinder c and attached to body m is arranged to slide between two vertical spring members which are attached to the support s. The force exerted by the damper in opposition to the motion of the body m is the product of the normal force and the coefficient of friction, as mentioned earlier. The damping force is usually constant; however if the pin is tapered, a variable force may be obtained. Friction damping is used in several commercially available isolators because it provides a simple means to control the damping forces. If necessary, independent systems may be provided within the same unit for vertical and horizontal motions. Some frictional dampers are effective in both horizontal and vertical directions.

2.3.3 Inherent Damping

There are many applications where vibration dampers of an external type such as those discussed above cannot be used because of space limitations, economic considerations or the fact that the system needs very little damping. These applications make use of the inherent damping or internal hysteresis of such materials as rubber, felt and cork. Vibration isolation with inherent damping is most commonly used in applications with constant motor speeds such as air compressors, generators and grinders.

2.3.4 Magnetic Damping

This type of damping is attainable as a result of the electric current induced in a conductor moving through a magnetic field. The damping force can be made proportional to the velocity of the conductor moving through the field. Magnetic damping has not been used successfully in vibration isolators because of its effectiveness is limited in a single direction.

2.3.5 Other nonmaterial dampings

Many nonmaterial damping mechanisms have been very useful in engineering applications, one of which is *acoustic radiation damping*. The damping force is the force acting on the mass due to the acoustic medium, which is determined from the solution of motion of the acoustic medium.

The other well-known non-material damping device is the *linear air pump*. The linear air pump's nearly airtight volume is placed adjacent to the vibrating system. The entrapped air is alternately compressed and rarefied by the motion of the structure. This motion will produce a pressure increment that is proportional to the structure motion. In order to have energy dissipation, many small holes are built into the panel.

2.4 Material Damping

It is generally true that all engineering materials dissipate energy during cyclic deformation. Some materials such as rubber, plastics and elastomers dissipate much more energy per cycle of deformation than steel and aluminium. For conventional structural materials the energy dissipation per unit-volume per cycle is very small compared to certain high damping alloys, polymer matrix composites and rubberlike materials (viscoelastic materials).

2.4.1 Materials for Vibration Isolation

All materials exhibit some degree of resilience and since deflection under load is a prerequisite of a vibration isolator, it is theoretically possible in isolator design to consider any material for possible use. However, in practice, the following materials are used frequently:-

Material	Form in which used
Felt	Felt mat Felt composites
Cork	Cork blocks Cork composites
Composite materials	Rubber with non-metallic reinforcement Neoprene and Cork
Rubbers	Natural rubber Neoprene Silicone Butyl Nitrile
Wire mesh	
Metal spring	Coil spring Leaf spring
Air	Active Passive

2.4.2 Felt

Felt is a material about which there is comparatively little engineering information. It consists of a matrix of held together by mechanical or chemical action and is usually available in relatively thin section. It appears that felt has a certain amount of internal damping and does not therefore require the use of auxiliary dampers, but quantitative information is difficult to obtain.

Felt finds its application as a vibration isolation medium generally in the foundations of those machines which are intrinsically well balanced but can be expected to produce high frequency disturbances at acoustic frequencies. It is usually available as mats or pads and it is cheap and easy to install. It is best employed as a support medium for the concrete block to which the machinery to be isolated is bolted. This arrangement is primarily to ensure a relatively low loading per unit area – a condition necessary to ensure stability and long life.

2.4.3 Cork

The cork used as an engineering material is based on the substance which occurs naturally but treated or processed into a convenient shape and texture for industrial use. Cork has one important difference when compared with the rubber materials to be discussed later – it contains many minute pockets of air which compress under the action of applied loads. It is thus relatively easy to compress a given volume of cork pad material, whereas significant values of compression are not possible in the case of rubber, which merely deforms under the action of an applied compressive load.

Cork is capable of withstanding substantial compressive loads and is commonly used under large machines or foundation blocks, where its high load-bearing capability allows it to be employed in localised areas.

Cork finds wider application than does felt, primarily because its properties as an engineering material are better known and are more widely available. It is commonly used as the resilient support medium for the concrete foundation blocks which are used to support machinery.

2.4.4 Composite Materials

It is sometimes convenient to combine the properties of one or more materials in order to obtain a combination of one or more materials in order to obtain a combination of properties to fulfil specific technical requirements.

The two most widely used combinations are rubber like materials with fabric reinforcement, cork, and Neoprene. 'Neoprene' is the generic term for a number of polymers manufactured by the Du Pont Company. These are synthetic rubber-like materials which are easily mouldable into simple and complex shapes, are easily bonded to steel, and can provide approximately the same values of compressive strain (and therefore of isolation performance) as can comparable pieces of natural rubber.

2.4.5 Wire mesh

The resilient properties of a compressed pad of fine stainless-steel wire mesh can be exploited in vibration-isolation applications. Since stainless steel is impervious to attack from fluids normally found in a machine shop, isolators of these materials are suitable for machinery mounting. In addition they have very good performance at high and low temperatures and are thus suitable for use in aerospace applications where such extreme temperatures are found.

2.4.6 Air-suspension Systems

The requirement for a very high performance vibration-isolation system arises for one or more of the following reasons:-

- an extremely high value of vibration-isolation is required;
- high values of isolation are required where the disturbing frequencies are low; and
- the values of vibration amplitude are extremely small.

In these circumstances, air-suspension systems have been found to offer a highly successful alternative. The mountings usually take the form of an enclosed bag requiring an auxiliary supply of compressed air, which may be constructed either wholly of rubber or partly of rubber and partly of metal. The load-bearing capability of such devices is high since it is relatively easy to provide compressed air at normal factory airline pressure.

2.5 Conclusion

The most important step in isolation of vibrational forces is to select a suitable isolator. It is an advantage if a review of the types of isolators that are commonly used were made.

Perhaps one of the most common means of providing isolation is with some form of padding or matting material. The materials used in this type of isolator range from cork to rubber or neoprene.

The range of deflection for the isolators described earlier is approximately as follows:-

Isolation mats or pads	up to 5mm(0.2in)
Rubber-in-shear isolators	up to 15mm(0.6in)
Steel spring isolators	up to 175mm(7in)

Considering the range of deflection of these materials, it can be easily seen that steel springs are more likely to be used than rubber-in-shear isolators are, and this is the general tendency, as floor spans are becoming greater.

Although the cost of steel springs is higher than rubber-in-shear and other isolators having a lower deflection, the increase in cost taken over the project as a whole is minimal. One advantage with springs is that they will not wear out. Organic isolation materials such as cork, rubber, etc., may deteriorate with time and will not ensure continued performance as wear causes increased vibration levels.

Properly selected steel springs provide highly efficient isolation of low frequency vibrations. High frequency vibrations such as fan impeller blade impulses can be transmitted through the coils of springs and into the floor structure unless a noise insulation pad is used in series with the springs. In the next chapter, the various vibration absorbing techniques generally used in practice are discussed.

CHAPTER 3

VIBRATION CONTROL TECHNIQUES

3.1 Introduction

The objective of enhancing damping in structural elements is to control the response of the elements so that catastrophic failure due to excessive deformation can be avoided. This is particularly necessary at the condition of resonance. Without damping, the deflection of a structure will increase to infinity at resonance. Modifying the stiffness of the structure is usually not a feasible solution, since changing the stiffness changes the natural frequency and sooner or later the condition of resonance will be reached. Therefore, one of the effective ways is to improve damping characteristics.

Dynamic responses of structural elements depend on a number of factors. If the range of the frequency of excitation is known, then we can adjust the values of the mass and stiffness of the system. This will change the natural frequency so that the condition of resonance can be avoided. However, if the condition of resonance cannot be avoided with the given values of mass and stiffness, several different methods can be applied to control the dynamic responses. These vibration control techniques can be classified into two categories; passive and active.

3.2 Sources of Vibration

There are numerous sources of vibration in an industrial environment: rotating or reciprocating machinery, transportation vehicles, impact processes etc. The first thing to be explored to control vibrations is to try to alter the source of vibration so that it produces less vibration though this may not always be possible. Some examples of the sources of vibration that cannot be altered are earthquake excitation, atmospheric turbulence and engine combustion instability. On the other hand, certain other sources of vibration such as unbalance in rotating or reciprocating machines can be altered to reduce the vibrations.

It is well known that whenever the frequency of excitation coincides with one of the natural frequencies of the system, resonance occurs. The most prominent feature of resonance is large displacement. In structural systems, large displacements indicate undesirably large strains and stresses, which can lead to the failure of the system. Hence resonance conditions must be avoided in any system. In most cases the excitation frequency cannot be controlled as it is imposed by the functional requirements of the system or machine. Therefore, we must concentrate on controlling the natural frequencies of the system to avoid resonance.

The natural frequency of a system can be changed by changing its mass or stiffness. In many practical cases, however, mass cannot be changed easily and hence stiffness of the system is the factor that is most often changed to alter its natural frequencies.

3.3 Vibration Isolation

Vibration isolation is a procedure by which the undesirable effects of vibration are reduced. Basically, it involves the insertion of a resilient member (or isolator) between the vibrating mass (or equipment) and the source of vibration so that a reduction in the dynamic response of the system is achieved under specific conditions of vibration excitation. An isolation system is said to be active or passive depending on whether or not external power is required for the isolator to perform its function. A passive isolator consists of a resilient member (stiffness) and an energy dissipater (damping). Examples of passive isolators include metal springs, cork, felt, pneumatic springs, and elastomer (rubber) springs. An active isolator is comprised of a servomechanism with a sensor, signal processor and an actuator. As stated earlier, the effectiveness of an isolator is stated in terms of its transmissibility. The transmissibility (T_r) is defined as the ratio of the amplitude of the force transmitted to that of the exciting force.

Vibration isolation involves the control of the supporting structure, the placement and arrangement of isolators and control of the internal construction of the equipment to be protected. The simplest kind of mechanical vibration has the waveform of sinusoidal motion. Vibrations in structures also exist in waveform but are generally more complex in nature. Such movement may be caused, for example, by the engine in an automobile, by engines or wind buffeting in aircraft or by a punch press in a building. Delicate electronic equipment and precision instruments must normally be isolated from these motions if accurate measurements are to be obtained.

Isolation of a body can be basically achieved by making all natural frequencies lower than about two fifths of the lowest source of force frequency. But a force or torque may not excite all the normal modes, and then the natural frequencies in the modes that are not excited do not need to be considered except to ensure that they do not actually coincide with the force frequency.

All practical vibration isolation systems involve the use of an arrangement of resilient supports for installed machinery or equipment. In any practical system it is unlikely that the design requirements of the local vibration conditions will be the sole consideration in choosing vibration isolators. For example, an isolation system to be used in a typical factory would be required to withstand the effects of oil, cutting fluids and water, and would require to function under very dirty floor conditions. Again, an isolation system installed underneath a large forging machine, which would be normally below floor level, is unlikely to receive maintenance for long periods of time. The design of the isolation system must take into account this probability.

When a machine is supported by resilient mountings it will be free to move in a number of different ways (or modes) and will do so if forces are present to excite such motions. Some machinery (e.g. forging hammers and some air compressors) generates sufficiently large forces internally to cause unacceptably large movements when supported by resilient mountings. In these cases the total mass of the machine foundation must be increased, usually by the construction of a base block in concrete or steel.

It is also vitally important to recognise that resiliently mounted equipment will have a number of natural frequencies, some or all of which may be excited by the machine itself. Any reciprocating machinery, for example, will necessarily have to be started up and shut down during operation, and mounting system resonant frequencies will inevitably be excited, albeit momentarily, during this sequence. The resultant behaviour of the mounted machine will depend on the stiffness of the isolators and also on the damping present. Such damping may be incorporated into the isolators or added in the form of separate units.

It is clear, therefore, that isolator choice must be made with care and only after consideration of :-

- The vibration isolation required
- Dynamic properties of the mounted machinery
- Environmental conditions

3.4 Selection of Materials

The most commonly used engineering materials in practice as isolators have been listed before. The selection of a material for a vibration isolator application depends on both the performances required and also on the environmental conditions in which the isolator is required to operate. The performance required defines the stiffness of the isolator and therefore for a given load the deflection required. Since deflection is related to strain for an isolator of a given size, the maximum allowable strain for a particular material and size of isolator frequently defines the lowest practicable natural frequency at which an isolator can be designed in that material. Theoretically the material itself does not impose a limit to the practice, however, very stringent practical considerations make it impracticable to consider any but one or two materials for any application.

3.5 Location of Isolators

The vibration isolators may be positioned and arranged in many different ways. There are basically three types of variations:

- Isolators attached underneath equipment
- Isolators located in the plane of the centre of gravity of the equipment
- Radius of gyration system i.e. mountings arranged four on each side in the plane of the radius of gyration known as double side-mounted system.

The vibration isolators should be considered as only one part of the isolating system, the other parts being the supporting structure that lies below the isolator and the internal structure of the equipment that is above the isolator. When isolators are selected for use where the period of resonance is critical, it should not be forgotten that the flexibility of the isolators are in series so that the resonant frequency of the loaded system will therefore be inversely proportional to the square root of the sum of these two flexibilities. The additional flexibility of the structure will lower the natural frequency of the system and will also result in increase displacement during resonance, caused by the presence of the undamped structure.

3.5.1 Underneath mounting system

For an underneath mounting system, the most efficient for vibration isolation is one with a low natural frequency in both the vertical and horizontal axes. The most stable system is one with a low natural frequency in both the horizontal and vertical axes. The tests show that the isolator spacing should not be less than twice the height of the centre of gravity from the mounting plane. This condition is illustrated in the Fig 3.1.

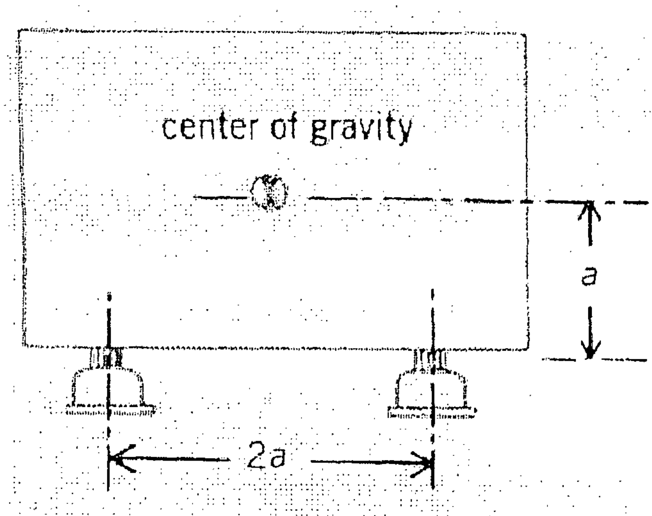


Fig 3.1. Underneath mounting system

3.5.2 Centre -of -gravity system

Locating the isolators in the plane of the centre of gravity has generally been considered the ideal system because of its ability to decouple the rotational modes of vibration. The primary conditions are :

- that the isolators be located in a plane passing through the centre of gravity.
- that the distance between the isolators be twice the radius of gyration of the body .
- that the horizontal-to-vertical stiffness of the isolators be equal.

3.5.3 Radius of gyration system

The third system that may be used when the limit of the centre of gravity system has been reached is a double side-mounted system or radius-of-gyration system as illustrated in Fig 3.2.

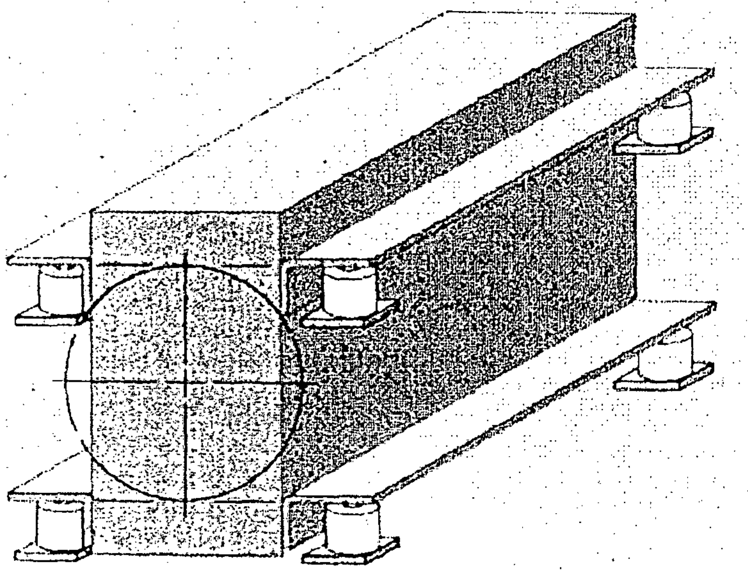


Fig 3.2. Radius-of-gyration System

Two sets of isolators are arranged on each side. For optimum results, the isolators should be located in the plane of the radius of gyration. However, since it is difficult to determine the exact radius, acceptable results will be obtained in most instances by assuming the body to be of uniform density. Satisfactory results have been obtained with bodies having height-to-width ratios up to 5. The limitation of this system will be reached when structural rigidity of the body is such that excessive bending occurs between the upper and lower isolator locations.

The next consideration in vibration isolation is the structural rigidity of the body to be isolated. This step is of importance in that the use of incorrect structure, particularly supporting brackets for component parts, can render the other two steps useless. Supporting brackets act as springs under a vibratory condition and become resonant at their natural frequency. Should resonance of the brackets coincide with the isolator resonance, damage may occur. The resonance point for the brackets and internal component should occur when the isolators are approaching their maximum efficiency so that the input vibration to the internal structure is at a low level. This level is normally reached at four times the natural frequency of the isolator.

3.6 Special Cases

3.6.1 Coupled Modes

Standard selection charts normally consider one mode of vibration, along the axis of the isolator. Some isolators are actually designed to prevent or limit movement in any other direction. However, these also give little or no isolation in any other direction and are therefore limited in use. In most cases this simple approach is adequate but for extreme cases where plant is very tall for its width or if large sideways forces are present, a more complicated approach is necessary. In most cases it is merely necessary to be aware of the complications so as to be able to recognise them if they occur.

There are six possible modes of vibration of any free body as shown in Fig 3.3.

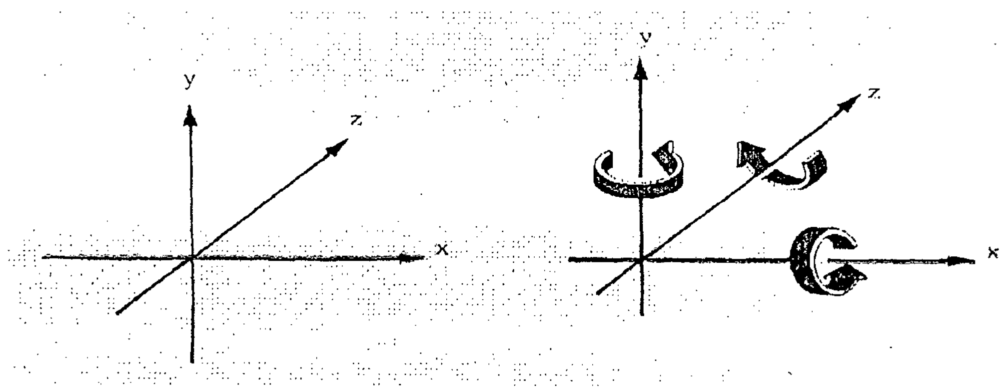


Fig 3.3. Modes of Vibration

There are three translatory modes along three normal axes and three rotational modes around these three axes. Modes of vibration may become coupled i.e. movement in one mode may cause movement in other modes. Whether this occurs depends on the location and stiffness of mounts and the distribution of the mass of the supported body.

A simple test to establish whether coupling exists is to consider or apply a static force directed through the centre of gravity of the system and examine the resultant motion of the body. If the body moves in two or more modes these modes will be coupled. In such a case there will be two resonant frequencies for each pair of coupled modes.

There will also be the natural resonant frequencies of the vertical translatory mode and the rotational mode around the y axis.

3.6.2 Isolation of sensitive equipment

It is sometimes necessary to isolate very sensitive equipment from the building structure as vibration from many sources within and outside the building will always be present. Some examples of sensitive equipment are electron microscopes, lasers and certain electronic equipment. It is often desired that the best possible isolation should be provided. This requires as low a natural frequency as possible. This may be achieved with springs but for example a resonant frequency of 1.6 Hz requires a static deflection of approximately 100 mm and 1 Hz requires 250 mm. Such high deflections will present installation and levelling problems; also the total length of spring will be very high. We must, therefore, turn to air mounts. There are a number of types of air mounts ranging from the simplest type which is pumped up to the required pressure after installation to the servo-level controlled mount with extra chambers to provide the correct damping. Resonant frequencies can be in the range of approximately 1.2 Hz to 3 Hz. Servo level mechanics can be used to correct the level of the isolated body to very low deflections despite high variations in load and as the static deflection remains constant so does the resonant frequency. Theoretically, the high frequency transmissibility of an air mount is inversely proportional to the square of the forcing frequency (ω^2) whereas a spring is inversely proportional to ω i.e. the high frequency isolation of an air mount is better than that of a spring. The ratio of horizontal to vertical stiffness can be made low, e.g. 0.5, thus offering good isolation in all modes.

The natural frequency of any system can be decreased by:-

- Increasing the radius of gyration
- Decreasing the stiffness
- Increasing the weight
- Decreasing the static deflection of the mount

This last point shows that if the mounts are moved very far apart, the natural frequency in this mode increases and therefore the transmissibility increases. A balance must therefore be struck between the requirements for as low a natural frequency as possible and for control of rocking motion due to the forces applied directly to the body.

3.6.3 Impact Isolation

In the case of impact, the system must be designed to absorb the energy of each impact before the next impact occurs. If this is not the case, the resultant transmission and deflection can build up and become very large. The time taken for an isolator system to absorb energy is dependent on damping, but it is common practice to assume for a fairly lighted damping system that 6 cycles of free oscillation should be permitted before the next impact i.e. the natural frequency of the isolators should be 6 times the frequency of impact. In addition, if the resonant frequency of the system were lower than the impact frequency, then at some point during run up and run down, the impact frequency will coincide with the resonant frequency. In this case, the force could be as high as during normal running when this point is reached and could cause severe problems. As mentioned before, the displacement is stiffness controlled below resonance and becomes mass controlled above resonance. This applies in impact isolation and therefore it is again desirable that the isolated body has a high mass. This implies that for a given deflection the stiffness of the isolators is also high.

3.7 Conclusion

This chapter presented an introduction to the basic elements of vibration control techniques including some special cases. All practical vibration isolation systems involve the use of an arrangement of resilient supports for installed machinery or equipment. The use of such resilient material is of fundamental importance to the theory and practice of vibration isolator design and selection. In the next chapter, isolators for different cases viz. for a symmetrical equipment, for an unsymmetrical equipment, for impulsive loading and for non-impulsive loading are discussed.

CHAPTER 4

ISOLATORS FOR VARIOUS TYPES OF EQUIPMENT

4.1 Introduction

A vibration isolator in its most elementary form may be considered as a resilient support for equipment. The function of an isolator is to reduce the magnitude of the force transmitted from the equipment to its support or alternatively to reduce the magnitude of motion transmitted from a vibrating support to the equipment. The effectiveness of an isolator in bringing about such a reduction is defined by the transmissibility of the isolator system. In the force-excited system, transmissibility is the ratio of the force experienced by the support to the force originating within the mounted equipment. In the motion-excited system, transmissibility is the ratio of the displacement amplitude of the mounted equipment to the displacement amplitude of the support.

4.2 Isolators for Symmetrical Equipment

When vibration isolation is considered with respect to a single-degree-of-freedom system having only one possible mode of motion, the concept of transmissibility may be expressed in simple numerical terms. In most practical systems, the mounted equipment has freedom in several modes of vibration which may exist concurrently. The displacement at each isolator may thus differ from that at other isolators. The method of analysis yields numerical values for the vibration amplitude at each isolator and makes it possible to calculate numerically the transmissibility for each isolator location. Except in special circumstances, it is difficult to justify the time required to make such a calculation. In general, an acceptable and conservative approach is to select a natural frequency on the basis of the desired transmissibility, and to design the isolator system to have a maximum natural frequency in any mode not exceeding the selected value.

The most common type of problem in vibration isolation concerns an equipment mounted upon four isolators located beneath the equipment. This type of problem reduces to its simplest form if the equipment and the isolator pattern may be considered symmetrical with respect to two vertical co-ordinate planes passing through the centre of gravity of the equipment. Assuming such symmetry, and viewing one face of the mounted equipment as shown in Fig 4.1, the equipment is free to move vertically, horizontally, and in rotation about an axis perpendicular to the face being viewed.

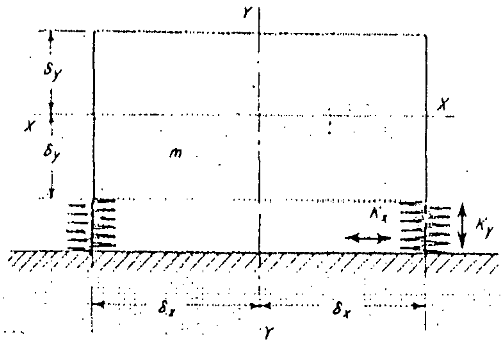


Fig 4.1. Elevation view of equipment supported upon isolators located at four lower corners.

If the vertical stiffness of each isolator is directly proportional to the dead-weight load which it carries, vibration in the vertical translatory mode is decoupled from vibration in other modes, and the natural frequency is determined by the mass of the mounted equipment and the vertical stiffnesses of the isolator . This natural frequency constitutes a reference for defining the natural frequencies in other modes, using Fig 4.2. It is necessary in using this figure to know the radius of gyration of the mounted equipment, the locations of the isolators with reference to the centre of gravity of the equipment, and the stiffnesses of the isolators in the several co-ordinate directions. The results obtained are the natural frequencies in the coupled rotational and horizontal translatory mode presented as a dimensionless ratio involving the natural frequency in the vertical translatory mode.

The above procedure for determining the natural frequencies in coupled modes is generally applicable and represents a rigorous analysis where the assumed symmetry exists. The procedure is somewhat laborious, however, because the dimensionless ratio ρ/δ_x appears in both ordinate and abscissa parameters and because it is necessary to determine the radius of gyration of the equipment. The relations set forth in Fig 4.2 may be approximated in a more readily usable form if (1) the mounted equipment can be considered a cuboid having uniform mass distribution, (2) the four isolators are attached precisely at the four lower corners of the cuboid, and (3) the height of the isolators is negligible.

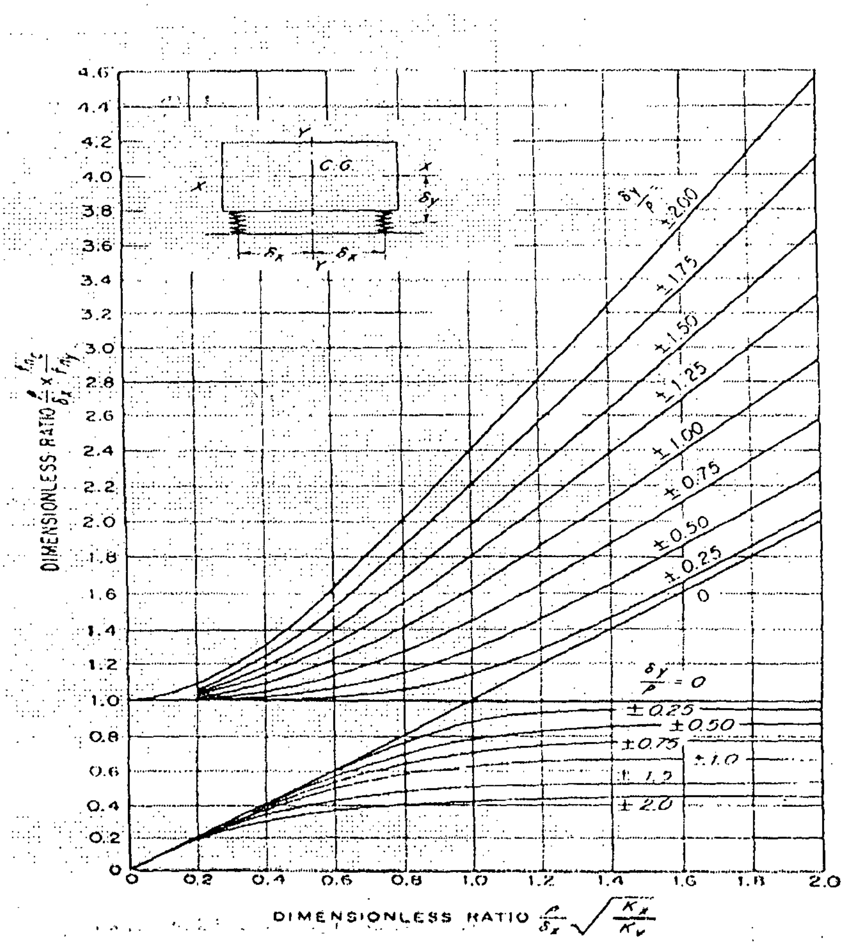


Fig 4.2. Curves showing ratio of two coupled natural frequencies f_{nc} in XY plane to decoupled natural frequency f_{ny} in translation along Y axis. The stiffnesses of the resilient supports in the X and Y directions are indicated by K_x and K_y respectively and the radius of gyration with respect to the Z axis through the centre of gravity is indicated by ρ.

The ratio of the natural frequencies in the coupled rotational and horizontal translatory modes to the natural frequency in the vertical translatory mode then becomes a function of only the dimensions of the cuboid and the stiffnesses of the isolators in the several co-ordinate directions.

Results obtained by making these assumptions are given by the following equation :

$$\frac{f_{n_c}}{f_{n_y}} = \frac{1}{\sqrt{2}}\sqrt{\frac{4\eta\lambda^2+\eta+3}{\lambda^2+1} \pm \sqrt{\left(\frac{4\eta\lambda^2+\eta+3}{\lambda^2+1}\right)^2 - \frac{12\eta}{1+\lambda^2}}} \tag{4.1}$$

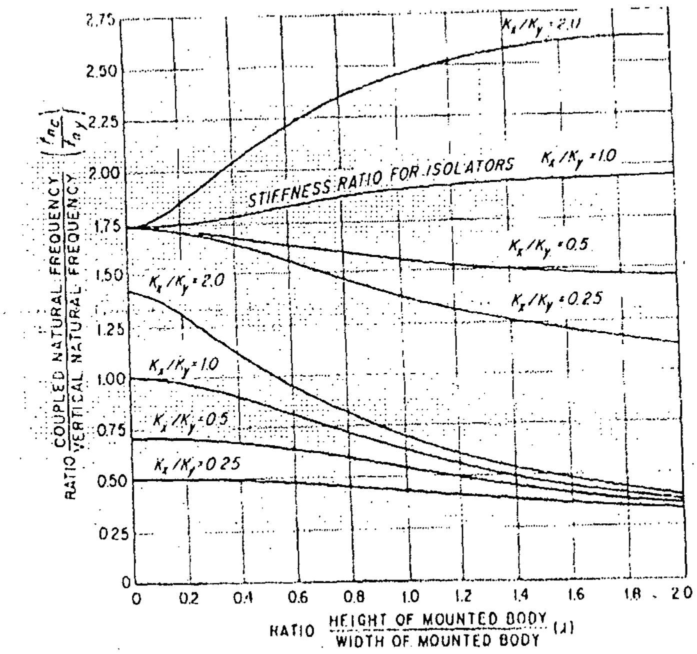


Fig 4.3. Curves showing ratio of coupled natural frequencies f_{nc} in XY plane to decoupled natural frequency f_{ny} in translation along Y axis for body supported as shown in fig 1.1 and having uniform mass distribution. The body is assumed to be a cuboid with the isolators attached precisely at the lower corners.

where $\eta = K_x/K_y$ designates the ratio of horizontal to vertical stiffness of the isolators and $\lambda = 2\delta_y/2\delta_z$ indicates the ratio of height to width of mounted equipment. The relation given by Eq.(4.1) is shown graphically in Fig 4.3. The curves included in this figure are useful for calculating approximate value of natural frequencies and for indicating trends in natural frequencies resulting from changes in various parameters.

The following important trends are worthy of emphasis:

1. Both the coupled natural frequencies tend to become a minimum, for any ratio of height to width of the mounted equipment, when the ratio of horizontal to vertical stiffness K_x/K_y of the isolators is low. Conversely, when the ratio of horizontal to

vertical stiffness is high, both coupled natural frequencies also tend to be high. It is thus apparent that, when the vibration isolators are located underneath the mounted body, the generally favourable condition of low natural frequencies is obtained using isolators whose stiffness in a horizontal direction is less than the stiffness in a vertical direction. A low horizontal stiffness may be undesirable, however, in applications requiring maximum stability. A compromise between natural frequency and stability may then lead to optimum conditions.

2. As the ratio of height to width of the mounted equipment increases, the lower of the coupled natural frequencies decreases. The trend of the higher of the coupled natural frequencies depends on the stiffness ratio of the isolators. It is evident that one of the coupled natural frequencies tends to become very high when (1) the horizontal stiffness of the isolators is greater than the vertical stiffness and (2) the height of the mounted equipment is approximately equal to or greater than the width. When the ratio of height to width of mounted equipment is greater than 0.5, the spread between the coupled natural frequencies increases as the ratio K_x/K_y of horizontal to vertical stiffness of the isolators increases.

4.3 Isolators for Nonsymmetrical Equipment

In some applications, the centre of gravity of the mounted equipment is so eccentric to the pattern of isolators that the assumption of two vertical planes of symmetry discussed above is not justifiable. In other instances, it may be necessary to employ more than four isolators underneath the equipment or to employ additional isolators near the top of the equipment for stabilising purposes Assuming the equipment to be symmetrical with respect to a single vertical plane through the centre of gravity, the system may be analysed as a three-degree-of-freedom system having motion in the plane of symmetry. If the system meets the test for decoupling the differential equation of motion for the decoupled mode may be omitted and the remaining two equations may be solved simultaneously for the natural frequencies in the two remaining modes. If the system does not meet the test for decoupling, the three equations of motion must be solved simultaneously to yield three natural frequencies in three completed modes of vibration.

When the centre of gravity of the mounted equipment in a plan view is not symmetrical with respect to the pattern of isolators, the distribution of the dead-weight load among the isolators is unequal. If the equipment is supported on three isolators, the load carried by each may be determined simply by applying the principles of statics. If four isolators are employed, as shown by the plan view in Fig 4.4, the analysis becomes considerably more involved because a change in the deflection of any one isolator causes a redistribution of the load among the other isolators.

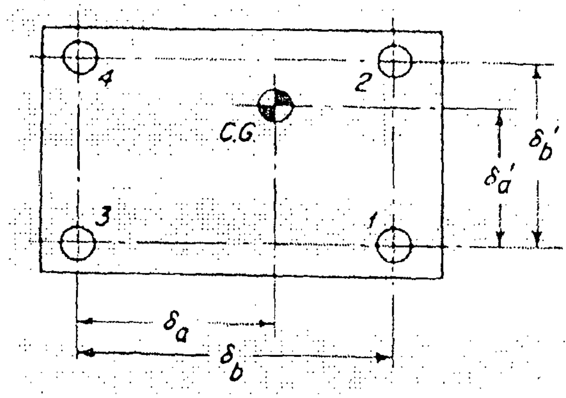


Fig 4.4. Plan view of equipment supported by four isolators wherein centre of gravity of equipment is not located above centre of pattern of isolators.

The following three equations are obtained from a summation of vertical forces and from summations of moments of these forces about two perpendicular horizontal axes :

$$K_1d_1 + K_2d_2 + K_3d_3 + K_4d_4 = W$$

$$\delta_b K_1d_1 + \delta_b K_2d_2 = \delta_a W \qquad (4.2)$$

$$\delta_b K_2d_2 + \delta_b' K2d2 = \delta_a' W$$

where the subscripts to K and d refer to the linear stiffness and static deflection of isolator designations shown in Fig 4.4. Assuming the equipment to be rigid, the isolators to be of the same free height, and the supporting surface to be flat and rigid, the following equation defines the rigidity of the mounted equipment :

$$d_1 + d_4 = d_2 + d_3 \qquad (4.3)$$

If the four isolators are identical, there is a unique solution for the load carried by each isolator which may be obtained from Eqs. (4.2) and (4.3) by setting each value of stiffness equal to K. This gives the following set of equations defining the load carried by each isolator :

$$\begin{aligned}
\frac{F_1}{W} &= \frac{1}{2}\left(\frac{\delta_a}{\delta_b} - \frac{\delta_a'}{\delta_b'} + \frac{1}{2}\right) \\
\frac{F_2}{W} &= \frac{1}{2}\left(\frac{\delta_a}{\delta_b} + \frac{\delta_a'}{\delta_b'} + \frac{1}{2}\right) \\
\frac{F_3}{W} &= \frac{1}{2}\left(\frac{3}{2} - \frac{\delta_a}{\delta_b} - \frac{\delta_a'}{\delta_b'}\right) \\
\frac{F_4}{W} &= \frac{1}{2}\left(\frac{\delta_a'}{\delta_b'} - \frac{\delta_a}{\delta_b} + \frac{1}{2}\right)
\end{aligned} \qquad \textbf{(4.4)}$$

The relations given by Eq (4.4) are shown graphically in Fig 4.5. In this figure, the dimensionless ratio $\delta a/\delta b$ and δ_a'/δ_b' are plotted on the horizontal and vertical axes, respectively, while the dimensionless ratios F/W of load carried by each isolator to total load is the parameter of the family of diagonal lines. The isolator locations are designated by the encircled numerals at the four corner, in accordance with corresponding designations in Fig 4.4. The load carried by each isolator is indicated by the diagonal line, where the appropriate numerical value is determined from the scale on the side of the line facing the particular isolator designation. Negative values indicate the force on the isolator is upward whereas positive values indicate a downward force.

Example : Assume that the arrangement of equipment and isolators is such that both dimensionless ratios δ_a/δ_b and δ_a'/δ_b' are numerically equal to 0.8. This point is adjacent to the upper right-hand corner of Fig 4.5. The portion of the load carried by isolator 1 is determined from the diagonal lines extending from lower left to upper right. The applicable numerical values, being on the same sides of the lines as 1, are along the upper and right margins. Interpolating between diagonal lines, isolator 1 is found to carry 25 percent of the load; i.e. $F_1/W = 0.25$. The load carried by isolator 4 is determined from the same lines, using the co-ordinate scale along the left and lower margins. Thus F_4 / W is also 0.25. The loads carried by isolators 2 and 3 are determined from the opposite set of diagonal lines. For isolator 2, the load as indicated by values extending along the lower and right margins is determined from the ratio $F_2/W = 0.55$.

The load on isolator 3 is determined from the same set of diagonal lines, using the scale extending along the left and upper margins. The ratio $F_2 / W = -0.05$; i.e. the force on isolator 3 is an upward force equal to 5 per cent of the weight of the equipment.

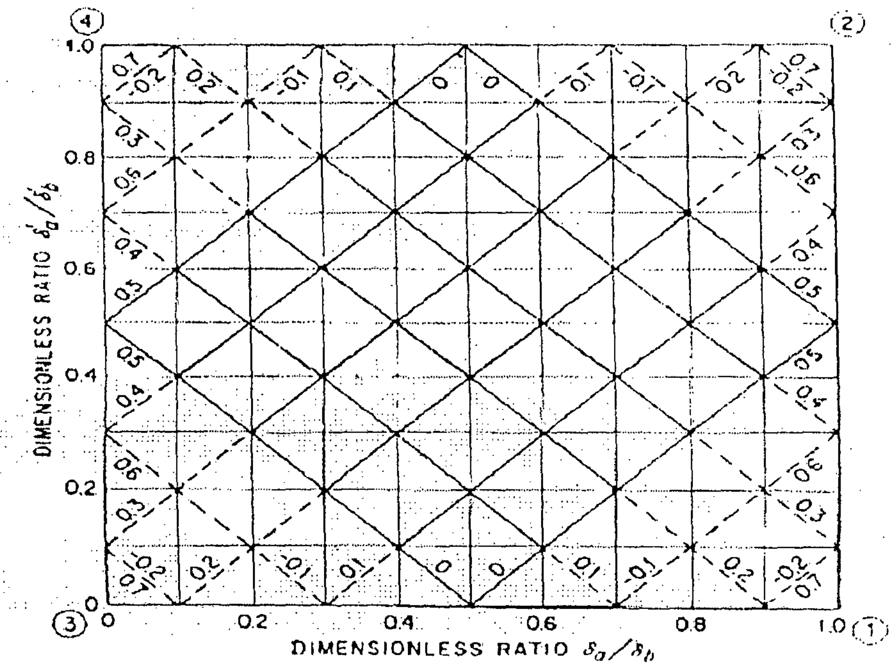


Fig 4.5. Diagram for determining deflection of isolators and loads on isolators for system shown in Fig 4.4. If all the isolators are of the same stiffness, the proportion of the total load carried by each isolator is indicated by the numerical value on the appropriate diagonal lines. If all isolators experience the same static deflection, the required stiffnesses of the isolators are obtained from the numerical values on the appropriate *solid* diagonal lines.

It is a requirement in decoupling vibration in the vertical translatory mode that the stiffness of each isolator be proportional to the dead-weight load which it carries. This tends to occur when the static deflections of all isolators are equal. Equations (4.2) define three necessary conditions for static equilibrium, and a fourth may be assumed at will. The fourth condition assumed here is that diagonally disposed isolators between them carry 50 percent of the weight of the supported equipment. This eliminates the possibility that a pair of diagonally opposite isolators will be excessively stiff, with the result that the supported equipment tends to pivot about a line connecting these isolators. This assumption gives the following equation :

$$(K_1 + K_4)d = (K_2 + K_3)d \qquad (4.5)$$

Setting $d_1 = d_2 = \ldots\ldots\ldots = d$ in Eq (4.2) and solving simultaneously with Eq (4.5),

$$\frac{K_1}{W/d} = \frac{1}{2}\left(\frac{\delta_a}{\delta_b} - \frac{\delta_a'}{\delta_b'} + \frac{1}{2}\right) \qquad \text{etc.} \qquad (4.6)$$

where the equations are similar to Eq (4.6)

The relations given by Eqs (4.6) are also shown graphically by Fig 4.5, wherein the parameter of the family of diagonal lines is K/(W/d). Fig 4.5 may thus be used to determine the stiffnesses of the respective isolators required to maintain the equipment level. The appropriate scale to use in determining the stiffness of any isolator is that appearing on the same side of a diagonal line as the encircled isolator designation, in accordance with the example in this section. When Fig 4.5 is employed to determine the stiffnesses required to attain equal static deflection at each isolator, the diagonal lines which are dotted must not be used. The dotted lines indicate a value of K greater than $\frac{1}{2}$(W/d), a condition which is incompatible with the initial assumption that diagonal pairs of isolators between them carry 50 per cent of the weight of the equipment. In the above example, the equipment cannot be mounted to attain equal static deflections because the co-ordinate point $\delta_a/\delta_b = 0.8$, $\delta_a'/\delta_b' = 0.8$ is in a region of dotted lines.

4.4 Isolators for Low-speed Machines

From the known operating frequency of a machine, the natural frequency required of the isolators to attain a desired transmissibility may be determined form Fig 4.6. This natural frequency must be considered a maximum; it is divided by the ratio f_{nc}/f_{ny} obtained from Fig 4.6 to obtain the maximum acceptable natural frequency in vertical translation. The latter is converted to static deflection by reference to Fig 4.7. The relatively great static deflection which appears necessary in mounting low-speed machinery introduces many problems.

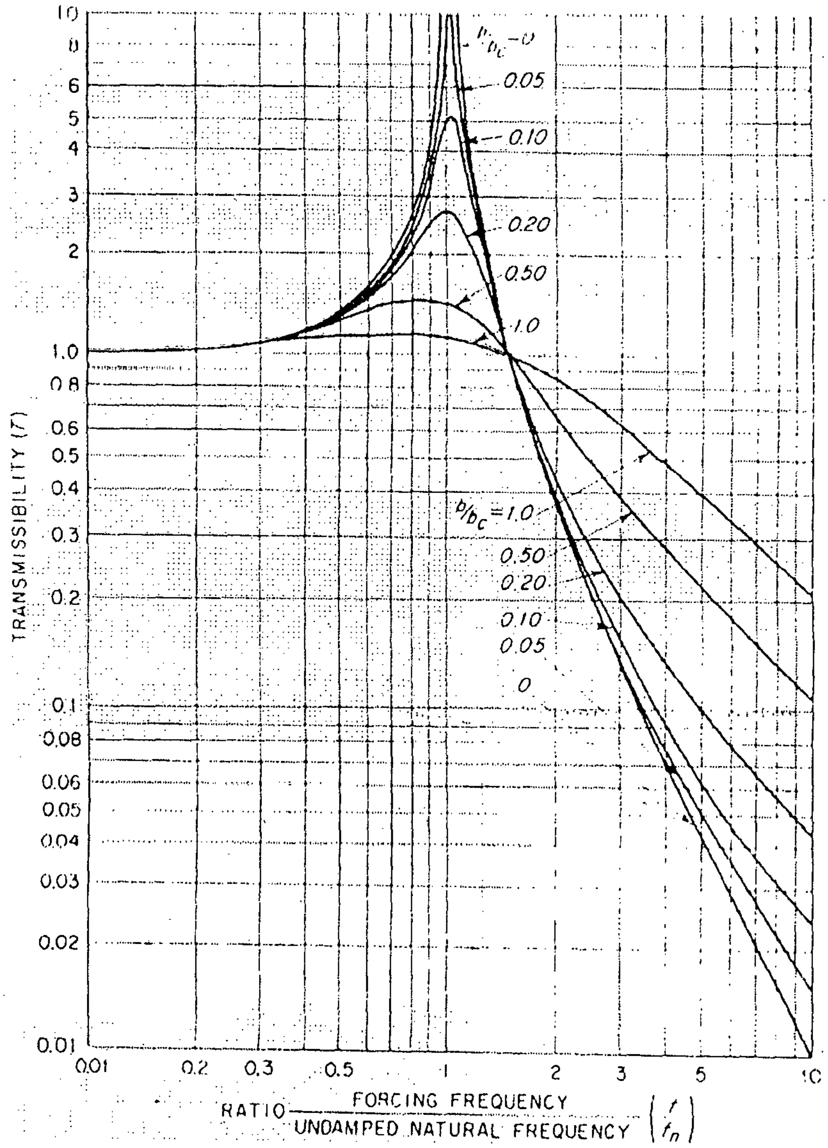


Fig 4.6. Force and displacement transmissibility curves for a viscously damped single - degree - of - freedom system. Force transmissibility is the ratio of maximum transmitted force to maximum applied force F_0. Displacement transmissibility is the ratio of displacement amplitude y_0 to applied displacement amplitude s_0.

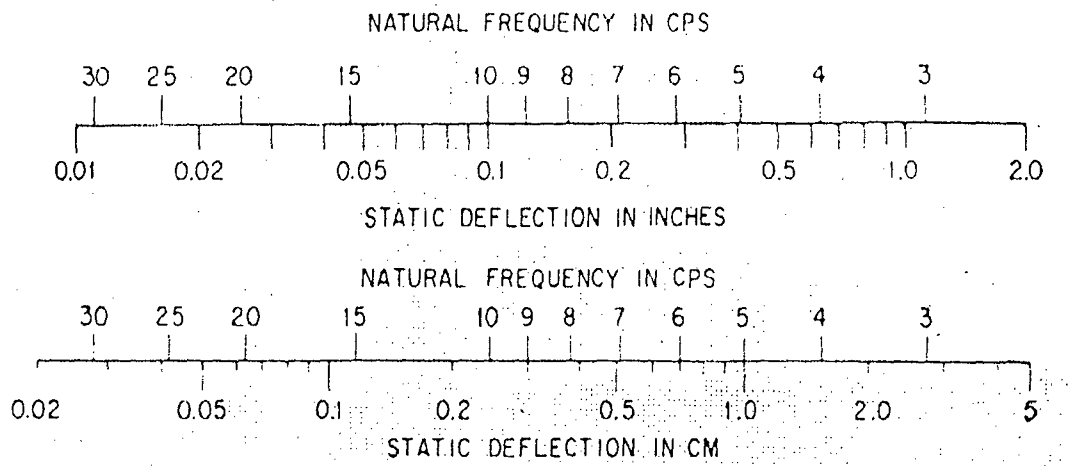


Fig 4.7. Relation between natural frequency and static deflection for a linear single-degree-of-freedom system.

In general, it is advantageous to maintain the static deflection at minimum by the application of either of two somewhat different methods as follows :

1. The horizontal distance between isolators may be made relatively great compared with the dimensions of the mounted equipment, thereby setting the natural frequencies in the coupled rotational and horizontal translatory modes either substantially equal to or much greater than the natural frequency in the vertical translatory mode. The high natural frequency then falls above the forcing frequency, and there is no isolation in the common sense of the word. However, there is force reduction in rotational modes of vibration, because the isolators are disposed at the ends of long arms. This approach can be used only for constant-speed machines, because the forcing frequency is interposed between two natural frequencies and variation of the forcing frequency may lead to resonance.
2. The equipment may be rigidly mounted upon a concrete inertia block which in turn is supported by isolators. The inertia block makes it possible to attain a centre of gravity for the machine and block combination which is low relative to the machine. The isolators may then be placed in the same horizontal plane as this centre of gravity without being above the level of the base of the machine, and the maximum natural frequency may be substantially equal to the natural frequency in vertical translation. The addition of the inertia block is also advantageous in that it increases the mass of the

machine and thereby decreases its vibration amplitude. This approach is satisfactory for use with variable-speed machines because all natural frequencies are below any designated operating speed for the machine. The use of a concrete inertia block is often the best solution to the isolation of low-frequency vibration but usually results in a relatively elaborate installation.

The following two examples indicate generally the methods of analysis used in the application of isolators and illustrate in the same order the above two procedures for mounting low-speed machinery.

Example 4.1.

The machine to be considered, as illustrated schematically in Fig 4.8, is relatively long in the direction of the Z axis and relatively narrow in the direction of the X axis. The force that is to be isolated is harmonic at the constant frequency of 8 cps. It is assumed to result form the rotation of an unbalanced member whose plane of rotation is

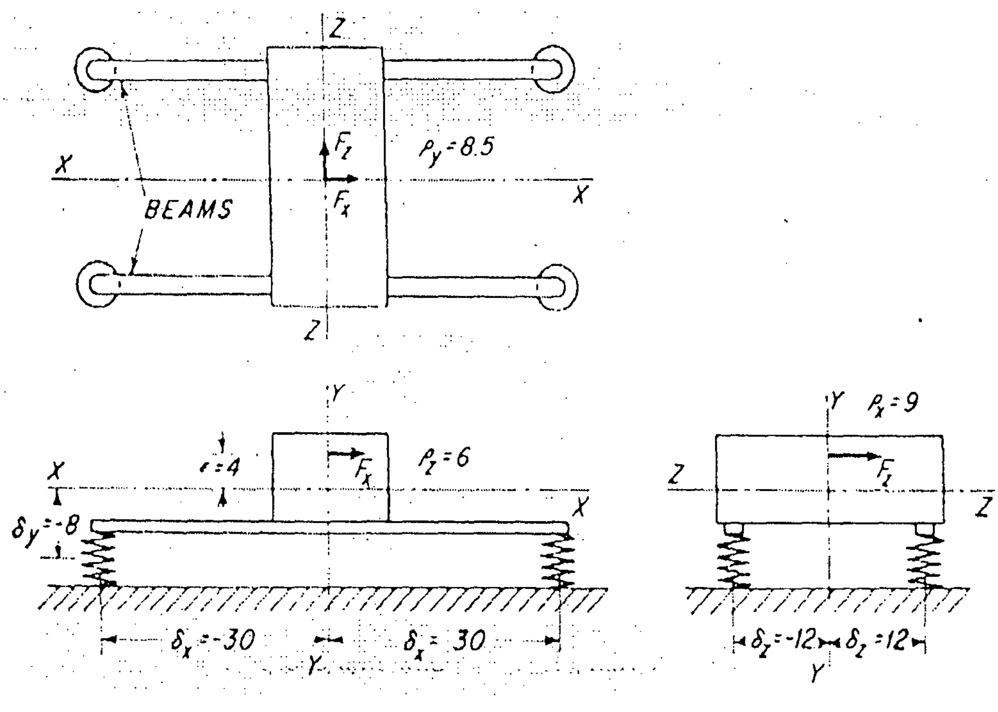


Fig 4.8. Mounting method in which isolators are placed at ends of long beams extending in direction of short dimension of mounted equipment.

taken, in the first instance, as a plane perpendicular to the Z axis and in the second instance, as a plane perpendicular to the X axis. The machine is set upon beams which extend parallel with the X axis and engage the isolators at their opposite ends. The distance between isolators is thus 60 in (152.4 cm) measured in the direction of the X axis and 24 in (61 cm) measured in the direction of the Z axis. The centre of co-ordinates is taken at the centre of gravity of the support body, i.e. at the centre of gravity of the machine and beams assembly. The total weight of the machine and supporting beam assembly is 100 lb(45.4 kg) and its radii of gyration with respect to the three co-ordinates axes through the centre of gravity are $\rho_x = 9$ in (22.9 cm), $\rho_y = 8.5$ in (21.6 cm), $\rho_x = 6$ in (15.24 cm). The isolators are of equal stiffnesses in the directions of the three co-ordinates axes ($\eta = K_x/K_y = K_z/K_y = 1$).

The following dimensionless ratios are established as the initial step in the solution :

δ_y / ρ_z	=	- 1.333	δ_y / ρ_x	=	- 0.889
δ_x / ρ_z	=	± 5.0	δ_z / ρ_x	=	± 1.333
$(\delta_y / \rho_z)^2$	=	1.78	$(\delta_y / \rho_x)^2$	=	0.790
$(\delta_x / \rho_z)^2$	=	25.0	$(\delta_z / \rho_x)^2$	=	1.78
$\eta (\rho_z / \delta_x)^2$	=	0.04	$\eta (\rho_x / \delta_z)^2$	=	0.561

The various natural frequencies are next determined in terms of the vertical natural frequency f_{ny}. Referring to Fig 4.2, the coupled natural frequencies for vibration in a plane perpendicular to the Z axis are determined as follows :

$$\frac{\rho_z}{\delta_x}\sqrt{\frac{K_x}{K_y}} = 0.2\sqrt{1} = 0.2$$

For $\delta_y/\rho_z = -1.333$, $(f_{nc} / f_{ny}) (\rho_z / \delta_x) = 0.19; 1.03$. The signs of the dimensionless ratios δ_y/ρ_x and δ_x/ρ_x require an explanation. The natural frequencies are independent of the sign of δ_y/ρ_z. The frequency ratio f_{nc} / f_{ny} then becomes positive. Dividing the above values for $(f_{nc} / f_{ny}) (\rho_z / \delta_x)$ by $(\rho_z / \delta_x) = 0.2$; $f_{nc} / f_{ny} = 0.96; 5.15$.

Vibration in a plane perpendicular to the X axis is treated in a similar manner. It is assumed that exciting forces are not applied concurrently in planes perpendicular to the X and Z axes, and vibration in these two planes is independent. Consequently, the example becomes two independent but similar problems, and similar equations apply:

$$\frac{\rho_x}{\partial_z}\sqrt{\frac{K_x}{K_y}} = 0.75\sqrt{1} = 0.75$$

For $\delta_y/\rho_x = -0.889$, $(f_{nc}/f_{ny})(\rho_x/\delta_z) = 0.57; 1.29$. Dividing by $\rho_x/\delta_z = 0.75$, $f_{nc}/f_{ny} = 0.76$; 1.72.

The natural frequency in rotation with respect to the Y axis is calculated from Eq (4.7) as follows, taking into consideration that these are two pairs of springs and that $K_x = K_y$.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K\partial^2}{m\rho^2}} \quad (4.7)$$

The six natural frequencies are now tabulated:

1. Translatory along Y axis : fny.
2. Coupled in plane perpendicular to Z axis : 0.96 fny.
3. Coupled in plane perpendicular to Z axis: 5.15 fny.
4. Coupled in plane perpendicular to X axis: 0.76 fny.
5. Coupled in plane perpendicular to X axis: 1.72 fny.
6. Rotational with respect to Y axis : 3.80 fny.

Considering vibration in a plane perpendicular to the Z axis, the two highest natural frequencies are in the translatory mode along the Y axis and in the coupled mode in which the natural frequency f_{nc} is 5.15 times the vertical natural frequency f_{ny}. In a similar manner, the two highest natural frequencies in a plane perpendicular to the x axis are the natural frequency increment which is void of natural frequency f_{ny} in translation along the y axis and the natural frequency 1.72 f_{ny} in a coupled mode. The natural frequency in rotation about the y axis is 3.80 f_{ny}. The widest frequency increment which is void of natural frequencies between 1.72 and 3.80 times the vertical natural frequency. This increment is used for the forcing frequency, which is taken as 2.5 times the vertical frequency. In as much as the forcing frequency is established at 8 cps, the vertical natural frequency is 8 divided by 2.5 or 3.2 cps. The required vertical stiffness of the isolators are calculated to be 105 lb per in (18.8 kg per cm) for the entire machine, or 26.2 lb per in(4.67 kg per cm) for each of the four isolators. This results in a static deflection of 0.95 in (24 cm) if the static and dynamic stiffness of the isolators are equal.

Example 4.2.

The following example illustrates the procedure for designing an inertia-block installation. The required weight of the block may first be estimated from the requirement that the isolators be approximately in the same horizontal plane as the centre of gravity of the machine-and-block combination, and from limitations on the permissible vibration amplitude of the machine and block as supported by isolators. In the example which follows, the inertia block appears T-shaped when viewed from one end, as shown in Fig 4.9.

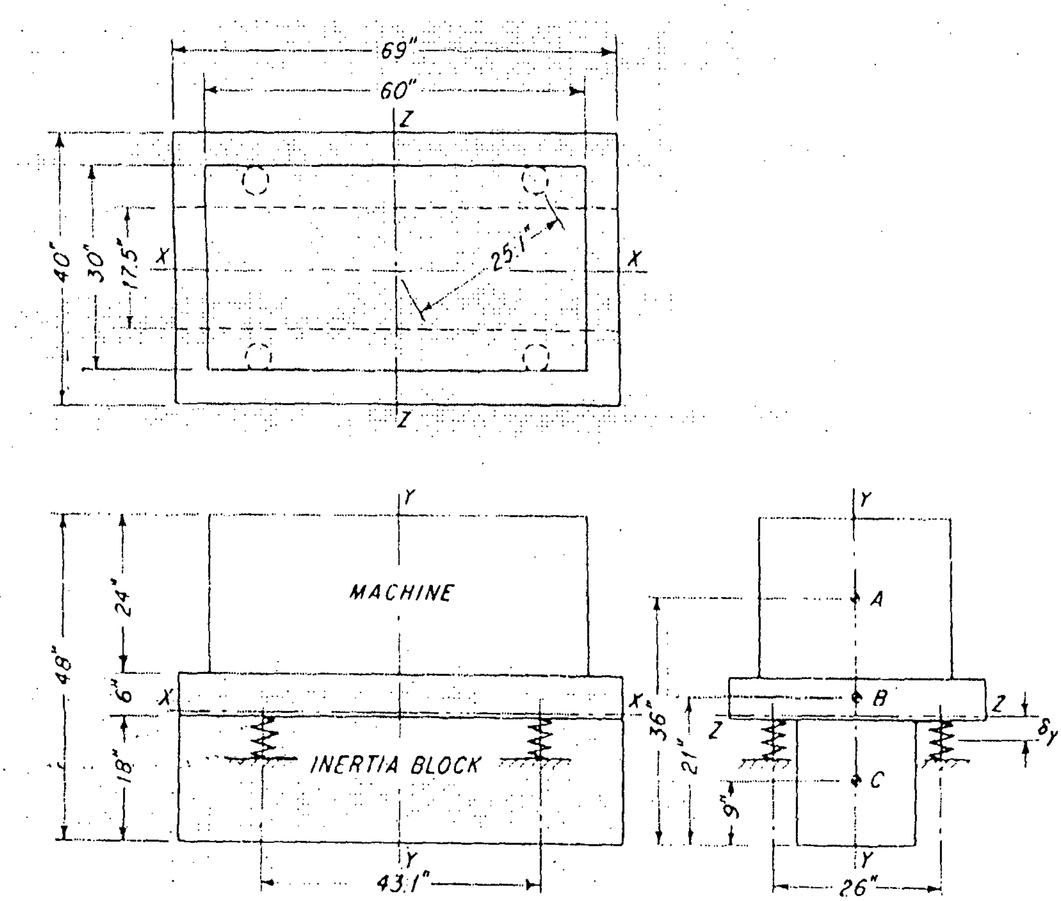


Fig 4.9. Mounting method in which machine is secured to inertia block whose end view is T-shaped and which is supported by isolators.

Table 4.1. Method of Calculating Natural Frequencies of Machine Secured to Concrete Block and Supported by Isolators as shown in Fig 4.9

		Properties with respect to axis		
		X-X (1)	Y-Y (2)	Z-Z (3)
1	Weight of part, lb	800 1,380 1,820	800 1,380 1,820	800 1,380 1,820
2	Mass m of part, lb sec^2/ in (A) (B) (C)	 2.07 3.58 4.72	 2.07 3.58 4.72	 2.07 3.58 4.72
3	Moment of inertia of parts, lb in sec^2 (A) (B) (C)	 258 486 251	 720 1,428 2,004	 775 1,896 2,004
4	Height of centre of co-ordinates above base	18.6	18.6	
5	Height h from each centre of gravity to centre of co-ordinates (A) (B) (C)	 17.4 2.4 9.6	 17.4 2.4 9.6	
6	mh^2 for part (A) (B) (C)	 624 21 491	 624 21 491	
7	Total moment of inertia, lb in sec^2	2,131	5,288	4,675
8	Radius of gyration, in	14.8	22.6	21.2
9	Dimensionless ratios: δ_y/ρ_x [Col (1)] or δ_y/ρ_z [Col (2)] ρ_x/δ_z [Col (1)] or ρ_z/δ_x [Col (2)]	 0.23 0.568	 0.15 0.523	
10	$\rho_x/\delta_z\sqrt{K_z/K_y}$ [Col(1)] or $\rho_z/\delta_x\sqrt{K_x/K_y}$ [Col(2)]	0.50	0.46	
11	$(\rho/\delta)\ (f_c/f_n)$	0.56; 0.44	0.52;0.44	
12	(f_c/f_n)	1.00; 0.77	1.00;0.84	
13	Natural frequency in coupled modes, cps	2.5; 1.94	2.5 ; 2.1	
14	Natural frequency in rotation, cps			2.6

This permits the spacing between isolators in both horizontal directions to be selected for optimum conditions and makes effective control of vibration possible. Although the example is based on this type of construction, the procedure is general and may be followed for any design of inertia bock. The first step is to assume dimensions for the block and to divide the mounted combination into parts which may be assumed regular. This includes the machine A and the parts B and C of the inertia block. The weights of these parts are recorded as item 1, their masses as item 2, and their moments of inertia as item 3 in Table 4.1. The moments of inertia of each part are calculated with respect to the three co-ordinate axes through the centre of gravity, assuming it to be a cuboid of uniform density and using the expression $I = (l_1^2 + l_2^2) / l^2$ where l_1 and l_2 are the length and width of the face perpendicular to the axis about which the moment of inertia is calculated.

The centre of co-ordinates is next taken at the centre of gravity of the machine-and-block combination. The height of the centre of co-ordinates is determined with reference to the plane of the lowermost surface of the block by taking the moment of the centre of gravity of each part with respect to this reference plane, adding these moments, and dividing by the total mass of the combination. The resultant height of the centre of co-ordinates is set forth as item 4 in table 4.1, while the distance from this centre of co-ordinates to the centre of gravity of each of the three parts is included as item 5. If the distance in item 5 is designated h, the product of mass and h^2 for each of the parts is set forth as item 6. The total moment of inertia of the machine-and-block combination is the sum of quantities for all three parts in items 3 and 6, and is set forth in item, 7. The corresponding radius of gyration as included in item 8 is the square root of the quotient of the moment of inertia in item 7 divided by the sum of the masses in item 2.

The parameter δ_y in Fig 4.10, the vertical distance from the centre of co-ordinates to the mid-height of the supporting spring, is 3.4 in for the height of springs contemplated. The two dimensionless rations δ_y/ρ and ρ/δ are now evaluated using $\delta_y = 3.4$, $\delta = 13$ and 21.55 for vibration in planes normal to the X and Z axes, respectively, and the values for radius of gyration are calculated as item 8.

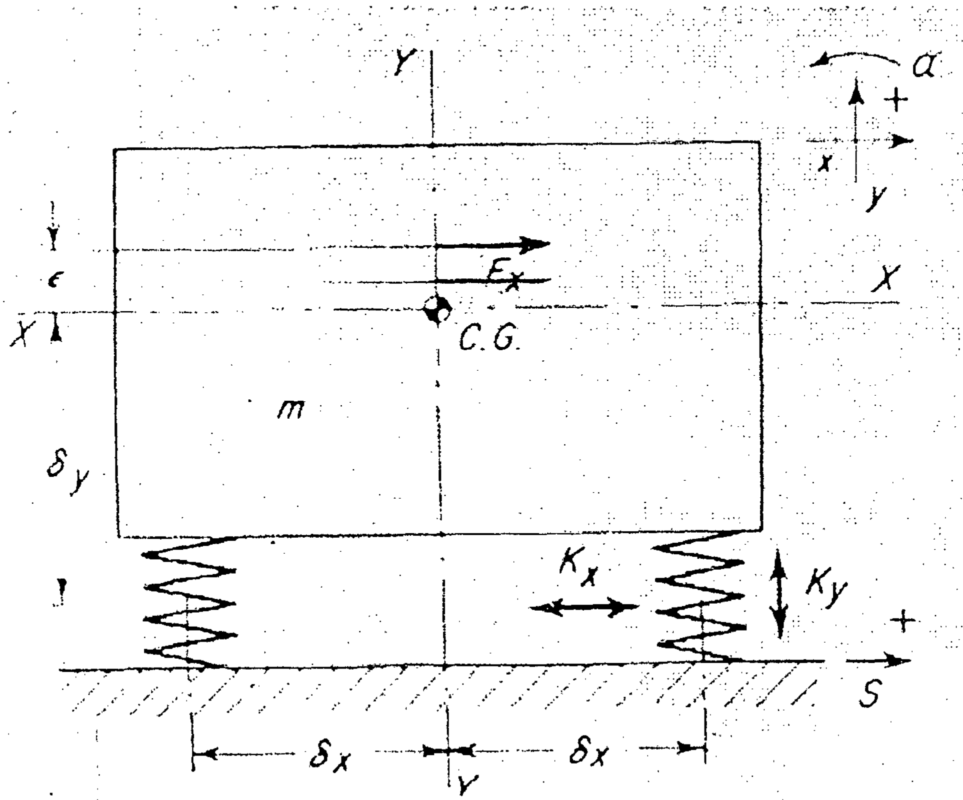


Fig 4.10. Elevation view of rigid body on resilient supports located at four corners.

The types of spring contemplated for use here is known to have a ratio of horizontal to vertical stiffness of 0.80. With this assumed value and the values for ρ/δ set forth in item 9, values for the abscissa scale, the values set forth in item 11 are read from the ordinate scale. Two values are available from the ordinate scale for each value on the abscissa scale because there are two curves for the dimensionless ratio δ_y/ρ. The ratios of the natural frequencies in the coupled modes to the natural to the natural frequency in the vertical translatory mode, as set forth in item 12, are obtained from item 11 by dividing the respective values by the appropriate values for the dimensionless ratio ρ/δ.

From the stated conditions of the installation, the minimum forcing frequency is 5 cps. To achieve a transmissibility of 0.33 for vertical vibration, a natural frequency of 2.5 cps is required in the vertical translatory mode (see Fig 4.6). Using the total weight of 4000 lb for the machine-and--block combination, the required total vertical stiffness of the isolators is calculated from Eq $f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W}} = 3.13\sqrt{\frac{1}{d}}$ cps to be 2600 lb per in., or a unit value of 650 lb per in, for each spring. Using the above value of 0.80 for the ratio of horizontal to vertical stiffness, the horizontal stiffness of each spring is calculated to be 560 lb per in.

The natural frequencies in the coupled modes, expressed in cycles per second, are obtained by multiplying the vertical natural frequency of 2.5 cps by the dimensionless ratios set forth in item 12. The resulting values for the natural frequencies on the coupled modes are set forth in item 13. The natural frequency f_{ny} in the rotational mode, with respect to the vertical Y axis, is calculated from Eq (4.7). This calculation employs the radius of gyration of 21.2 in set forth in item 8 and the diagonal distance of 25.1 in. shown in Fig 4.10, and takes cognisance of the four active springs. The resultant natural frequency is set forth as item 14 in Table 1.

4.5 ISOLATION OF SHOCK

The application of isolators to alleviate the effects of shock is difficult to discuss, partly because the term "shock" has no definite and accepted meaning. It seems to connote suddenness, either in the application of a force or in inception of motion. Because various categories of shock exist, it is not possible to formulate a single procedure that may be used universally in the design and application of isolators.

Two general types of problem are encountered in the application of isolators to reduce shock. In one type of problem, the applied force is of an impulsive nature, with the consequence that a massive member or members acquire additional momentum. The motion associated with this increase in momentum introduces a requirement that the isolator have adequate energy-storage capacity to arrest such motion. In a non impulsive application or force, the machine upon which such force acts often acquires a motion temporarily but arrests itself ultimately after experiencing some displacement. The optimum isolator is one which permits such motion while preventing the transmission of excessive force. Such an isolator is not required to have capacity to store energy but only to provide a suitable support for the equipment.

4.5.1 Impulsive Loading

For impulsive loading, the analysis refers to the schematic diagram illustrated in Fig 4.11. The body m_y represents a machine supported by an isolator K_y which in turn is supported by a foundation represented by the system m_s, K_s. The machine m_y suddenly acquires a downward velocity when acted upon by the impulsively applied force F. This may occur in a forging hammer, for example, where m_y represents the anvil of the machine and its downward velocity results from hammer impact. The effectiveness of the isolator K_y is indicated by a reduced deflection of the spring K_s. A simplified analysis may be carried out by making the following assumptions :

The downward velocity of the mass m_y is acquired instantaneously.

1. The isolator K_y is linear within the limits of travel sustained during the shock motion.
2. The natural frequency of the machine and isolator system m_y, K_y is small relative to the natural frequency of the support m_s, K_s.
3. The mass m_y of the machine is negligible relative to the mass m_s of the support.

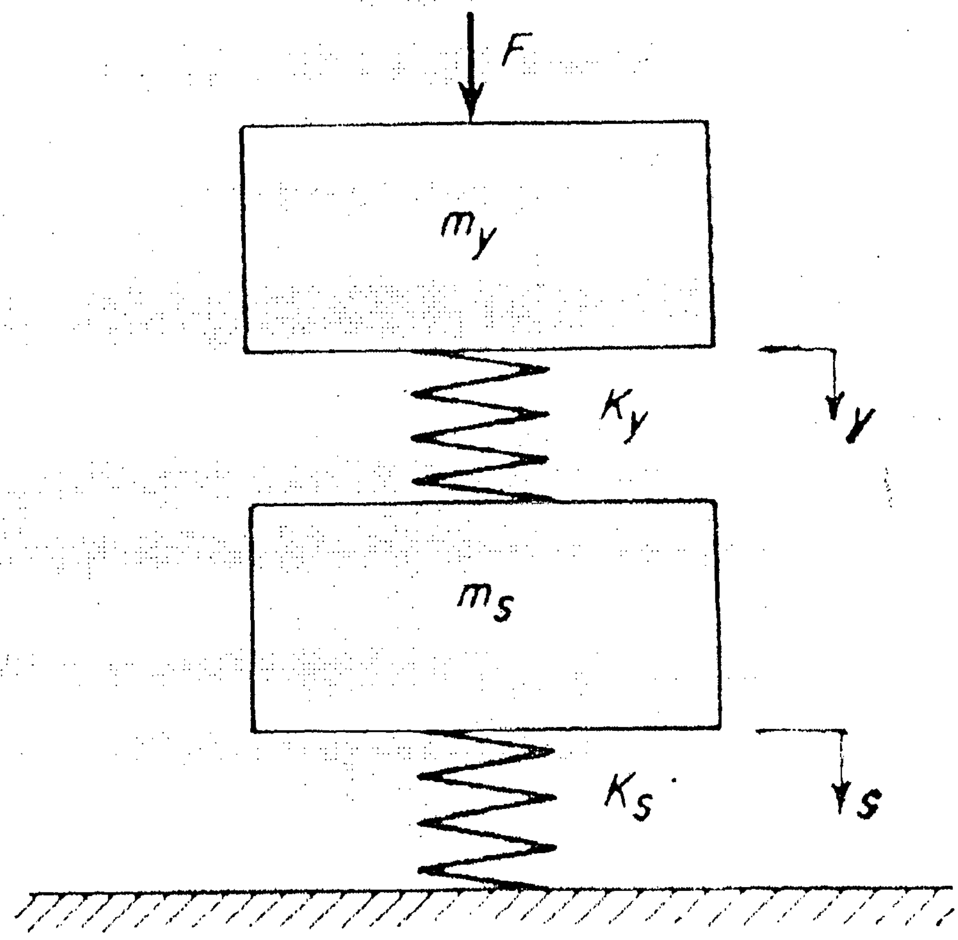


Fig 4.11. Schematic diagram to illustrate effectiveness of isolator K_y in protecting foundation m_s , K_s from impulsively applied force F acting on mounted machine m_y.

When the machine is rigidly mounted to the floor; i.e. the isolator K_y in Fig 4.11 is made infinitely stiff, an expression for maximum deflection s_0 of the foundation may be derived by letting f approach infinity and substituting $y_0 = s_0$, ;

$m = m_s + m_y$ and $f_n = \frac{1}{2\pi}\sqrt{K_s/(m_y + m_s)}$:

$$s_0 = \frac{J}{2\pi f_{n_s} m_s \sqrt{1 + m_y / m_s}} \qquad (4.8)$$

where $f_{n_s} = 1/2\pi\sqrt{K_s/m_s}$ in cycles per second, m_s is in units of pound second squared per inch, J is the impulse of the force F in pound seconds and s_0 is expressed in inches.

The influence of the mass m_y of the machine may be determined by letting $m_y = 0$ in Eq(4.8). This gives the following expression for maximum deflection of the foundation:

$$s_0 = \frac{J}{2\pi f_{n_s} m_s} = \frac{J}{\sqrt{K_s m_s}} \qquad \textbf{(4.9)}$$

The influence of the mass m_y of the machine in reducing the maximum deflection s_0 of the foundation is obtained from a comparison of Eqs. (4.8) and (4.9).

The result of the comparison is illustrated by the dotted line in Fig 4.12, where in unity on the ordinate scale is the dimensionless deflection obtained from Eq (4.9).

The effectiveness of an isolator is investigated by letting K_y be relatively small in accordance with above assumptions 3 and 4. This creates two independent systems which are uncoupled because m_y is assumed much smaller than m_s . The maximum displacement yo of the machine is then determined from the following expression:

$$y_o = \frac{J}{2\pi f_{ny} m_y} = \frac{J}{\sqrt{K_y m_y}} \qquad \textbf{(4.10)}$$

where m_y is in units of pound seconds squared per inch, f_{ny} is in units of cycles second, influence of the impulse J depends largely upon operating conditions. If the machine operates automatically, substantially more over-all motion often is permissible than if manually operated. When the machine is supported by isolators, assuming $m_y << m_s$ the natural frequency of the machine-and-isolator system is $f_{ny} = \frac{1}{2\pi}\sqrt{K_y m_y}$ and the maximum deflection of the foundation is $s_o = K_y y_o$ where y_o is given by Eq (4.10):

$$s_o = \frac{J}{2\pi f_{ns} m_s} x \frac{f_{ny}}{f_{nx}} \qquad \textbf{(4.11)}$$

The relation given in Eq (4.11) is shown by the solid line in Fig 4.12.

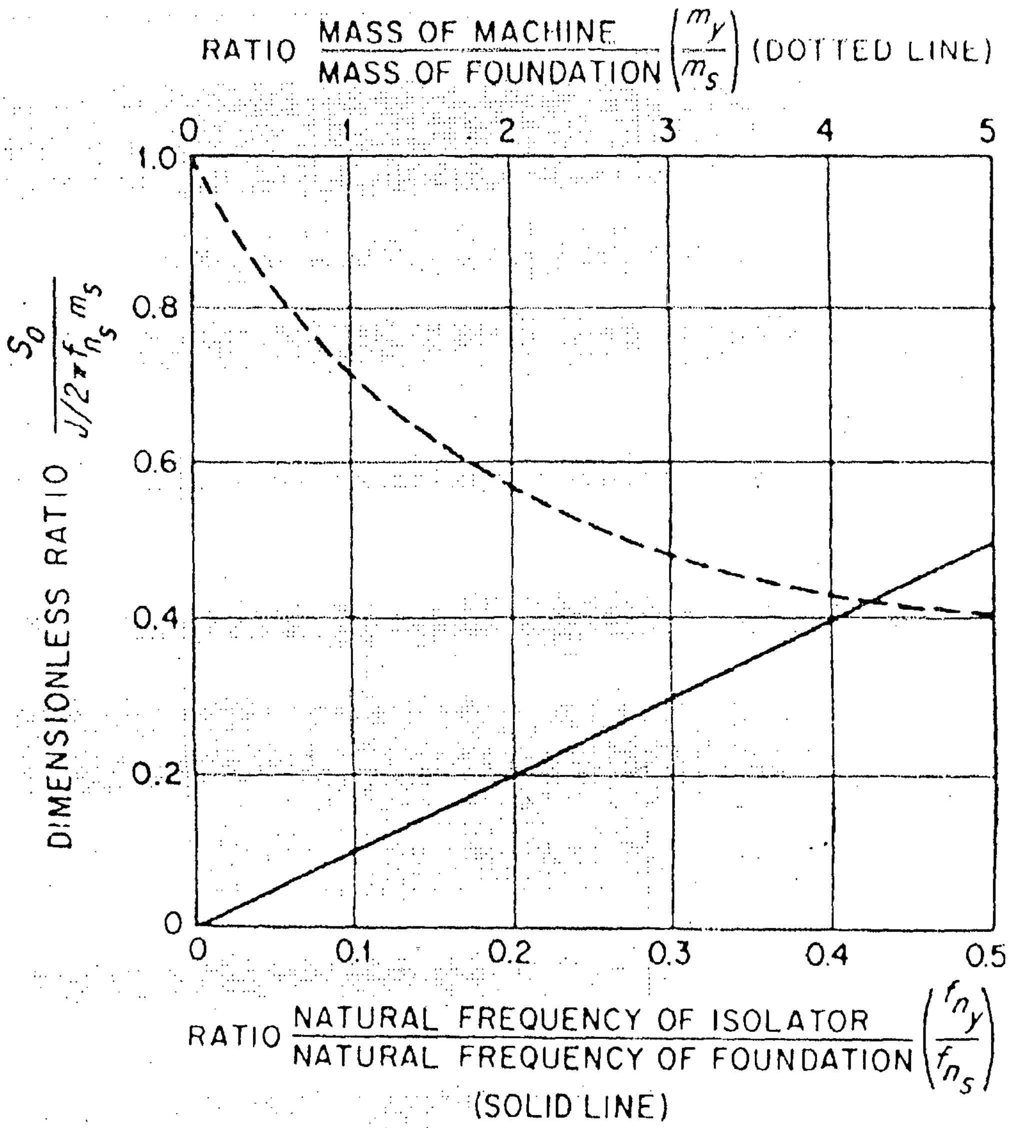


Fig 4.12. Curves showing effect of means to reduce deflection s_0 of foundation in Fig 4.11. The dotted curve applies to the scale on the upper margin and indicates deflection of support with machine rigidly attached thereto wherein mass m_y of the machine is varied. The solid line applies to the scale on the lower margin and indicates effectiveness of the isolator K_y in reducing deflection of support.

Inasmuch as the maximum deflection of the foundation is directly proportional to the natural frequency of the isolator system, the value of s_0 may be decreased by either increasing m_y or decreasing K_y. The latter may be undesirable because it increases the motion of the machine, as indicated by Eqs. (4.10) and (4.11), decreases both the deflection of the support and the movement of the mounted machine. This is commonly accomplished by mounting the machine on a concrete block supported by isolators. The combination of isolators with an inertia block for the machine is an optimum solution to problems requiring energy storage within the isolators where there is an impulsive addition of energy.

4.5.2 Nonimpulsive Loading

If there is no change in the over-all momentum of the system but only a change in the distribution of momentum within the system, it is not essential generally that the isolators have energy-storage capacity but rather only freedom to permit motion as dictated by the momentum transformation. In the punch press shown in Fig 4.13 (a), for example, the total momentum remains constant. The angular momentum of the flywheel is suddenly reduced during the working stroke, but the angular momentum of the entire press is increased correspondingly. Conversely, the flywheel momentum is increased as its velocity is gradually stored by the motor, and the momentum of the entire press is decreased.

The moment applied to the flywheel by the metal working operation is indicated by pulses A in Fig 4.13 (b), while the moment applied to the flywheel by the motor is indicated by pulses B. An equal and opposite pattern of moments reacts upon the body of the press. Assuming the press to be supported to rotate freely about an axis normal to the paper, its angular position at any instant is indicated by the solid line in Fig 4.13 (c). As the press periodically acquires and then loses its velocity, it gains a step of displacement at each cycle, as indicated by the successive angular-displacement increments f in Fig 4.13 (a) and the time history shown by the solid line in Fig 4.13 (c).

If the punch press is supported by isolators, it cannot continue to rotate indefinitely as suggested by the solid line in Fig 4.13 (c). The isolators take command of the press after each stroke and restore it to its initial position, when the next metal working strokes occurs. If the isolators are adequate, they interfere but little with the motion of the punch press during the metal working stroke and permit it to rotate in response to the moments applied during the metalworking operation. The isolators should have permissible deflection at least as great as suggested by the maximum displacement of the dotted line in Fig 4.13 (c) and adequate stiffness to return the press to its normal position before the inception of the next working stroke.

In general, satisfactory isolation will be obtained by noting the duration of the metalworking force as indicated by the width of pulse A and applying isolators accordingly. If the isolators are selected so that their natural period is substantially greater than the duration of the metalworking force, the effect is similar to maintaining a large ratio of forcing to natural frequency for conditions of vibration isolation. Shock isolation is then attained the a similar manner analogous to vibration isolation.

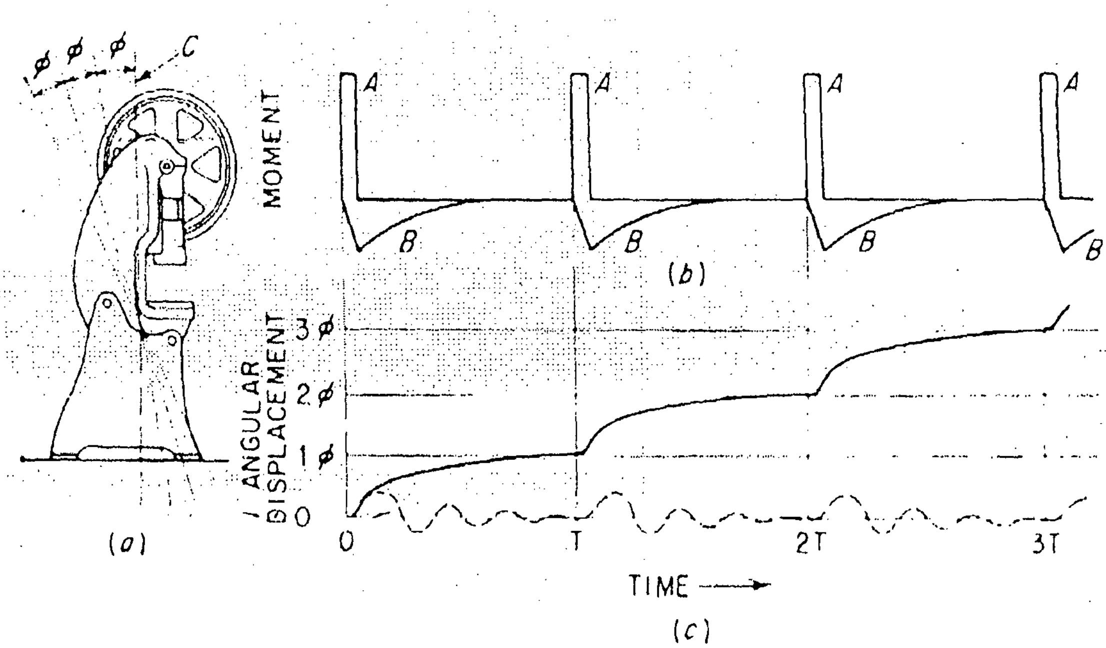


Fig 4.13. Typical punch press (a) and diagram of its movement under influence of non impulsive forces involved in its operation. The curves A and B at (b) indicate the couples acting upon the press. The solid line at (c) indicates the displacement that the press would experience if freely supported in space, whereas the dotted line indicates the corresponding displacement when supported by vibration isolators.

4.6 SPECIAL PROBLEMS IN ISOLATION OF HIGH FREQUENCY VIBRATION

In the classical theory of vibration isolation upon which the above discussion is based, it the assumed that the resilient elements of the isolator is massless and that the support for the isolator is infinitely rigid. If the frequency of the forcing vibration is relatively low, the assumption leads to a good approximation and the results usually are acceptable for practical purposes. When the frequency of the forcing vibration becomes relatively high, standing waves tend to occur in the isolator and the transmissibility may becomes relatively great at the standing -wave frequencies. Further more, resonance may occur in the support for the isolator with consequently large vibration amplitudes.

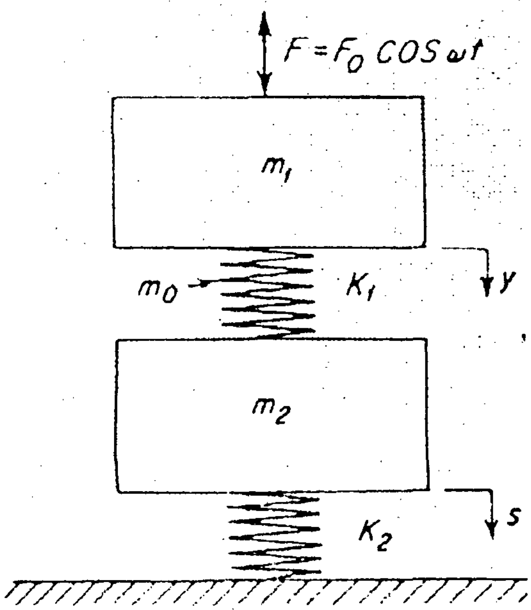
4.6.1 Standing Waves

It is difficult to determine standing-wave frequencies in rubber elements of irregular shape by analytical means. The applicable principles, however have been demonstrated on the basis of simple rubber elements. The analysis refers to the system illustrated in Fig 4.14 and includes a rubber cylinder of mass m_o and stiffness K_1, a rigid massive support m_2, and a rigid body m1 arranged to load the rubber cylinder along its axis of symmetry. The expression for transmissibility is

$$|T| = \frac{1}{\sqrt{\left[1+\left(\frac{B_2}{B_1}\right)^2\right]\sinh^2 \xi_b B_2 + \left(\cos B_2 - \frac{B_2}{B_1}\sin B_2\right)^2 + \xi_b \frac{B_2}{B_1}\sinh 2\xi_b B_2}} \tag{4.12}$$

where $B_1 = m_o/m_1$ and $B_2 = f/f_n \sqrt{m_0/m_1}$. The forcing frequency in cycles per second is indicated by f, while $f_n = \frac{1}{2\pi}\sqrt{\frac{K_1}{m_1}}$ is the natural frequency assuming the rubber cylinder to be massless. The damping parameter ξ_b is a function of the viscosity and the velocity of sound in the resilient material of the isolator. For small values of damping and to a first approximation, $\xi_b \approx b/b_c$.

Transmissibility curves calculated from Eq (4.12) are shown in Fig 4.15 for values of the damping parameter ξ_b equal to 0.05 and 0.3., when the ratio of mass of mounted equipment to mass of resilient elements is equal to 10. The ordinate in Fig 4.15 is dimensionless transmissibility, while the horizontal scale is the product of the dimensionless frequency ratio f/f_n and the square root of the dimensionless mass ratio m_o/m_1. For comparison, classical transmissibility curves for a massless resilient element and $b/b_c = 0.05$, 0.03 are shown in dotted lines in Fig 4.15 it is evident that the effect of the mass of the resilient element is negligible at low values of the forcing frequency.



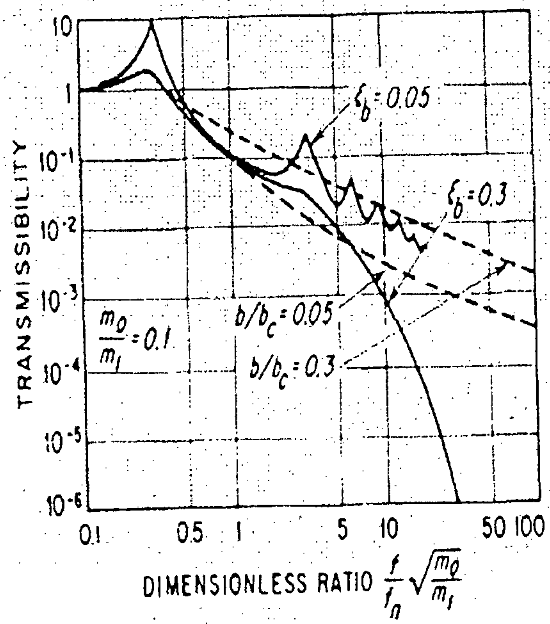


Fig 4.14. Schematic diagram of systems to show effect of standing waves in isolator K_1 or resilience in support K_2 when mounted equipment is acted upon by force F.

Fig 4.15. Transmissibility curves for the isolator K_1 in Fig 4.14 when K_2 is infinitely rigid and when standing wave effects are considered in the isolator K_1.

When the forcing frequency becomes greater, the length of the rubber cylinder includes one or more half wavelengths of the forcing frequency and a standing wave develops. A peak then occurs in the transmissibility curve. The heights of successive peaks, as the forcing frequency increases, tend to decrease and the effect of standing-wave resonance ultimately disappears. The analytical results shown in Fig 4.15, as calculated from Eq (4.12), has been substantially confirmed by experimental investigations. The height of the peaks resulting from standing-wave resonances increase as the damping parameter ξ_b decreases and as the mass ratio m_o/m_1. increases.

The peaks are most likely to be high where the resilient element is a metal spring because the damping of metal is relatively small, and the mass is relatively large. For resilient elements made of rubber, the mass ratio m_o/m_1 tends to be small, and the damping parameter ξ_b is substantially greater than that for metal. Referring to Fig 4.15, the damping parameter $\xi_b = 0.025$ is relatively small for rubber, and the mass ratio $m_o/m_1=10$ is relatively large. As a consequence, the upper solid curve in Fig 4.15 represents approximately the most unfavourable circumstances likely to be encountered with the use of rubber isolators. If metal springs are used and if high transmissibility occurs as a result of a standing wave resonance, the difficulty often can be eliminated by interposing a pad of rubber in series with the metal spring. Experience shows that this usually eliminates any noticeable peak in the transmissibility curve.

4.6.2 Resilience of Support

The effectiveness of vibration isolators may be decreased by resilience of the foundation. This is illustrated with reference to Fig 4.14 wherein the system m_2, K_2 is an idealisation in lumped parameters of any structure, such as a steel beam. The

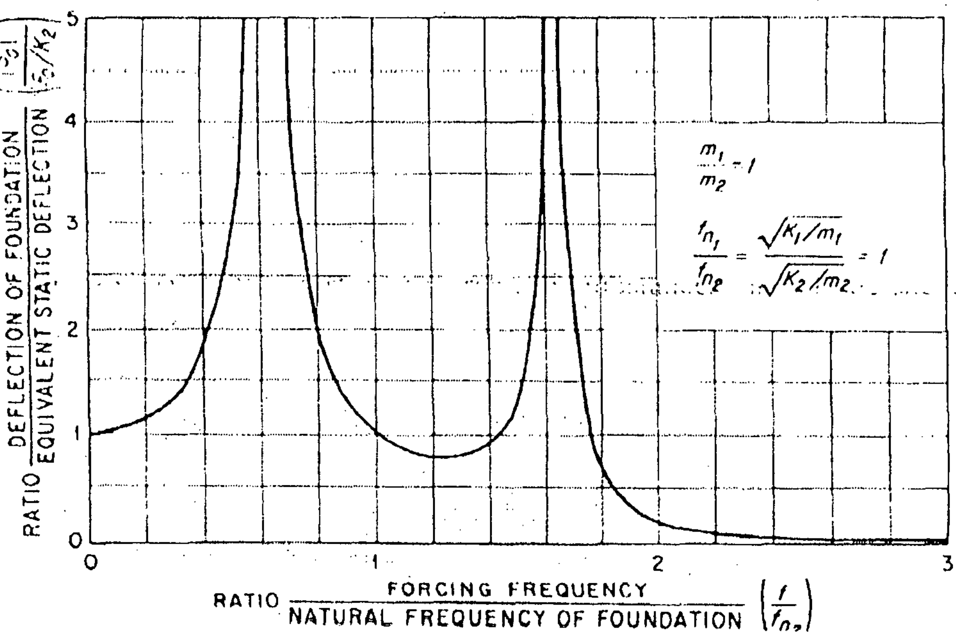


Fig 4.16. Curve showing ratio of displacement of system m_2 , K_2 to equivalent static deflection F_0/K_2 for system shown in 4.14. These curves are for a mass ratio m_1/m_2 =1 and for a ratio of natural frequencies f_{n1}/f_{n2} = 1.

mounted equipment is represented by the rigid body m_1, and the isolator is represented by the massless linear spring K_1. Excitation of the system results from the force F applied to the mounted equipment. If the structure K_2 is infinitely stiff, the foundation is rigid and the previously developed theory for isolators as given by the transmissibility expression is applicable.

If the structure K_2 is not rigid, the equation for the maximum displacement s_o of the support m_2, is

$$\frac{|S_o|}{F_o/K_2} = \frac{\left(f_{n_1}/f_{n_2}\right)^2}{\left[\left(m_1/m_2\right)\left(f_{n_1}/f_{n_2}\right)^2 - \left(f/f_{n_2}\right)^2 + 1\right]\left[\left(f_{n_1}/f_{n_2}\right)^2 - \left(f/f_{n_2}\right)^2\right] - \left(m_1/m_2\right)\left(f_{n_1}/f_{n_2}\right)^4}$$

(4.13)

Where $f_{n1}^2 = 1/4\pi^2\ (k_1/m_1)$, and $f_{n2}^2 = 1/4\pi^2\ (k_2/m_2)$ and both branches of the system are undamped. The relation given by Eq (4.13) is illustrated numerically in Fig 4.16 for a ratio m_1/m_2 of unity and a ratio f_{n1}/f_{n2} of unity.

The curves of Fig 4.16 show two resonant frequencies because the system illustrated in Fig 4.14 is a two-degree-of-freedom system with coupling between the two branches of the system. The vibration amplitude of the foundation tends to become great when the frequency of the forcing vibration equals one of the natural frequencies of the coupled system. The natural frequencies of the system are determined by equating the denominator of Eq (4.13) to zero. The resulting expression for the two natural frequencies in the coupled modes is as follows :

$$\frac{f_{n_1}}{f_{n_2}} = \frac{1}{\sqrt{2}}\sqrt{\left(1+\frac{m_1}{m_2}\right)\left(\frac{f_{n_1}}{f_{n_2}}\right)^2 + 1 \pm \sqrt{\left[\left(1+\frac{m_1}{m_2}\right)\left(\frac{f_{n_1}}{f_{n_2}}\right)^2 + 1\right]^2 - 4\left(\frac{f_{n_1}}{f_{n_2}}\right)^2}} \qquad \textbf{(4.14)}$$

where f_{nc} represents natural frequencies in the coupled modes and $f_{n2} = \frac{1}{2\pi}\sqrt{\frac{K_2}{m_2}}$ is the natural frequency of the foundation considered as a decoupled system. The relation given by Eq (4.14) for the natural frequencies in coupled modes is shown graphically in Fig 4.17 for several values of the mass ratio m_1/m_2.

It may be noted from Fig 4.17 that one natural frequency of the coupled system is always equal to or greater than the natural frequency of the foundation considered as a decoupled system. If the frequency of the forcing vibration becomes equal to this coupled natural frequency, the foundation may be expected to vibrate with relatively large amplitude. An actual structure foundation has distributed mass and, consequently, natural frequencies in many modes. The relations set forth in Fig 4.17 apply to any one of such natural frequencies, and resonances may occur between the

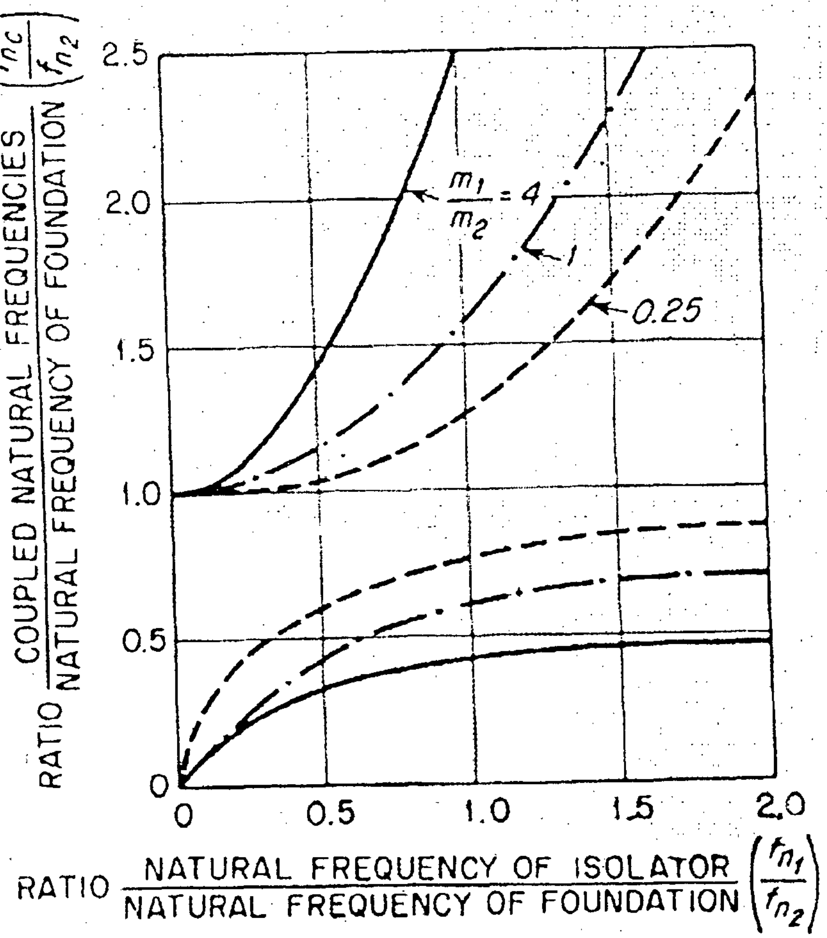


Fig 4.17. Curves showing the ratio of the coupled natural frequencies f_{nc} to the natural frequencies of the foundation in Fig 4.14.

forcing vibration and any one of the natural frequencies of the coupled system. This effect cannot be completely eliminated by isolation because the coupled system has relatively large values of natural frequency, even when the natural frequency f_{n1} of the isolator branch approaches zero. This may account for the failure of vibration isolators to isolate vibration of relatively high frequency under certain conditions. The damping materials can be used to reduce the effect of resilience of the foundation.

4.7 Conclusion

The advantage attained through the use of vibration isolators may be lost if the mounted equipment or machine is attached to rigid piping, electrical conduit or shafting. Conduits for supplying electrical power or for conducting fluids should have flexible portions to prevent transmission of vibration. Rubber hose often is preferable but if the temperature is too great or if chemically active fluids must be transmitted, metal hose or tubing must be used. In the next chapter, seismic mountings of various typed are discussed.

CHAPTER 5

SEISMIC MOUNTINGS

5.1 Introduction

Vibrations of earth's surface caused by waves coming from a source of disturbance inside the earth are described as earthquakes. Earthquakes may be caused by:-

(a) Tectonic activity

(b) Volcanic activity

(c) Land-slides and rock-falls

(d) Rock bursting in a mine

(e) Nuclear explosions

As engineers, we are interested in earthquakes that are large enough and close enough (to the structure) to cause concern for structural safety–such earthquakes are usually caused by tectonic activity.

The design force is lower for a more flexible structure. Thus, one can increase the natural period of the building to reduce the seismic demand. One way to make a structure more flexible is to replace the stiff partitions with flexible ones. However, this is not desirable because a more flexible structure undergoes a large deformation and thus an earthquake may cause more severe damage to the non structural elements. The most effective way to increase flexibility of a structure, and yet not make its non-structural elements more vulnerable to earthquake damage, is to provide a base isolation system. This is done by providing seismic mountings, bearing pads or some other isolation device under the building by first underpinning the structure. Seismic mountings if properly designed and provided can prevent catastrophic damage to buildings/ equipment.

In designing a seismic mounting, it is necessary to decide what resilient material to use and where to place it in relation to the seismic mass. The obvious and normal place is under the mass, an arrangement that is referred to as a "base mounting". The mounting may consist of a massive block formed directly on the ground, or resting on unit or area isolators standing on a structural support. Alternatives to the base mounting are the "centre-of-gravity". (c.g.) mounting in which isolators are located around the sides of the seismic mass, nominally in the plane containing the c.g., and the "pendulum" mounting in which the seismic mass is suspended so that its c.g. is below the plane containing the points of attachment of rods or springs that support it.

5.1.1 Base Mounting

Before unit and area isolators came into general use the common practice was to isolate equipment on an independent block, that is, a block not in contact with the surrounding building structure(Fig 5.1).

An independent block is formed directly on or in the soil, sand, or other site materials, which for brevity we shall refer to as the "ground". The installation is, of necessity, located at ground or basement level. An independent block may be used to isolate an engine or other vibration source, or a vibration-sensitive equipment or process.

The vibratory behaviour of a massive block resting on the ground is complicated by the fact that the ground functions as both the resilient material and the support. Where does the resilient material end and the support begin? A simplified analysis assumes that some part or "bulb" of the ground adjacent to the block can be regarded as additional seismic mass, and that the material outside this bulb provides the resilience. More advanced theory regards the ground as a material having distributed elastic and damping properties, and deals with the propagation of elastic waves, the dynamic properties of soil, clay, sand, other site materials and combinations of materials, and the way these properties change under the influence of vibration.

The mounting most commonly used today in buildings and industry is the base mounting on unit isolators. The installation may be above the floor, or in a concrete box or pit (Fig 5.2), which accommodates part of the seismic mass below the floor level so that the equipment remains at a convenient working height.

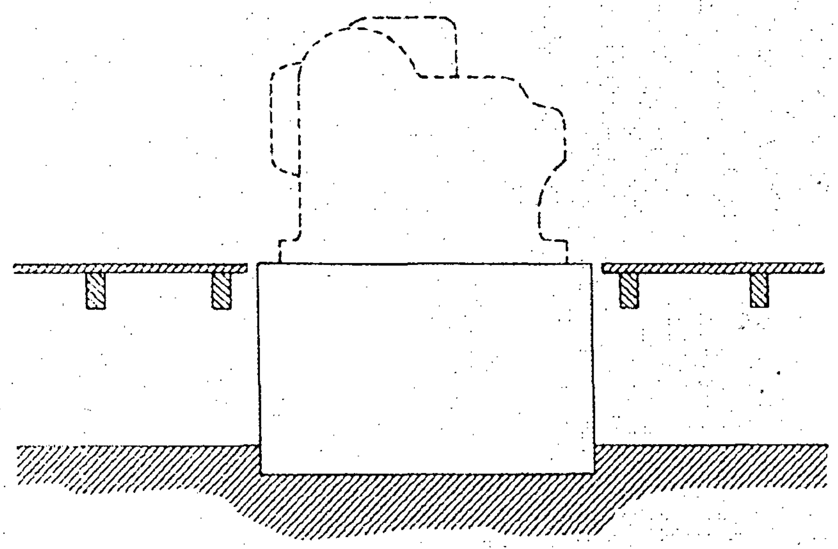


Fig 5.1. Independent block

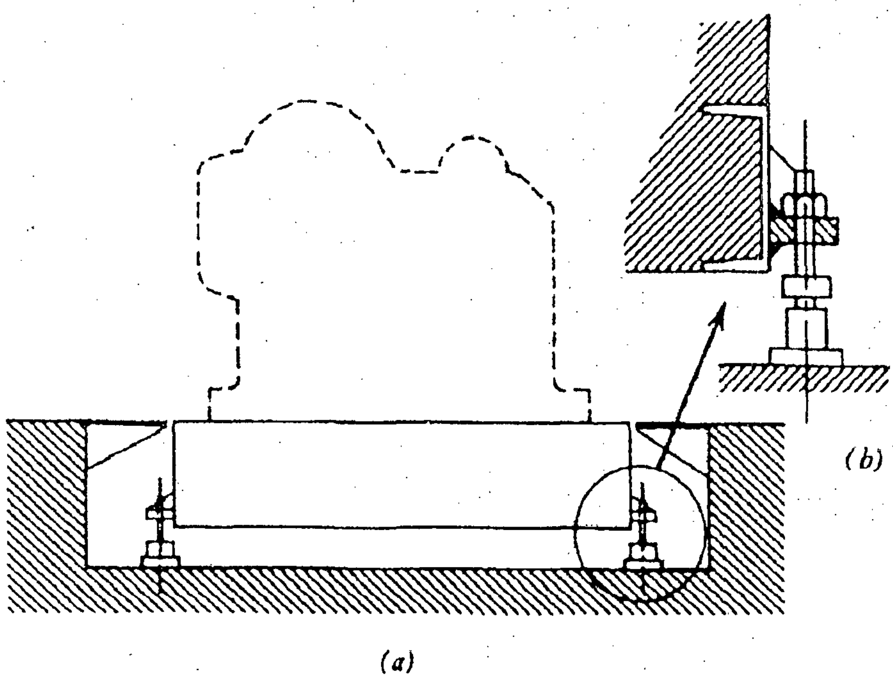


Fig 5.2. Base mounting on rubber isolators

Example 5.1

A base-type mounting designed to isolate a large roll-grinding machine from vibration generated by cold-rolling mills in a sheet metal manufacturing plant is sketched in Fig 5.2. The 40-tonne machine, which can accommodate a 10-tonne roll, is installed on a 67-tonne inertia block. The total seismic mass of 117 tonnes is carried on 54 rubber-in-shear isolators in two rows as illustrated, with a static deflection of 5 mm. The load is applied to the isolators through screws (Fig 5.2 b) so that load sharing among the isolators can be adjusted, and any isolator removed or replaced if necessary.

5.1.2 Centre of Gravity Mounting

The isolators are sometimes located beside the seismic mass in such a way that when the c.g. of the seismic mass is displayed horizontally the restoring force at each isolator acts in the horizontal plane through the c.g.

Example 5.2

In preparation for the installation of a jig boring machine at an engineering works, the vibration was measured at the site of an existing machine that was to be replaced. The vibration resulting from the operation of a plate-shearing press in the vicinity was excessive. Some modification of the foundation of the press made no improvement, so the company designed a seismic mounting for new machine.

The 20-tonne machine was installed on a 30-tonne inertia block (Fig 5.3) and the whole supported on 16 helical springs in four groups of four arranged symmetrically in plan about the vertical axis through the c.g. of the seismic mass. The load was transferred to the springs through housings at the ends of crossbeams (Fig 5.3 b). The springs, which were made of 38-mm diameter steel, in coils 229-mm mean diameter, were designed for a static deflection of 54 mm, and located in the manner defined above for c.g. mounting.

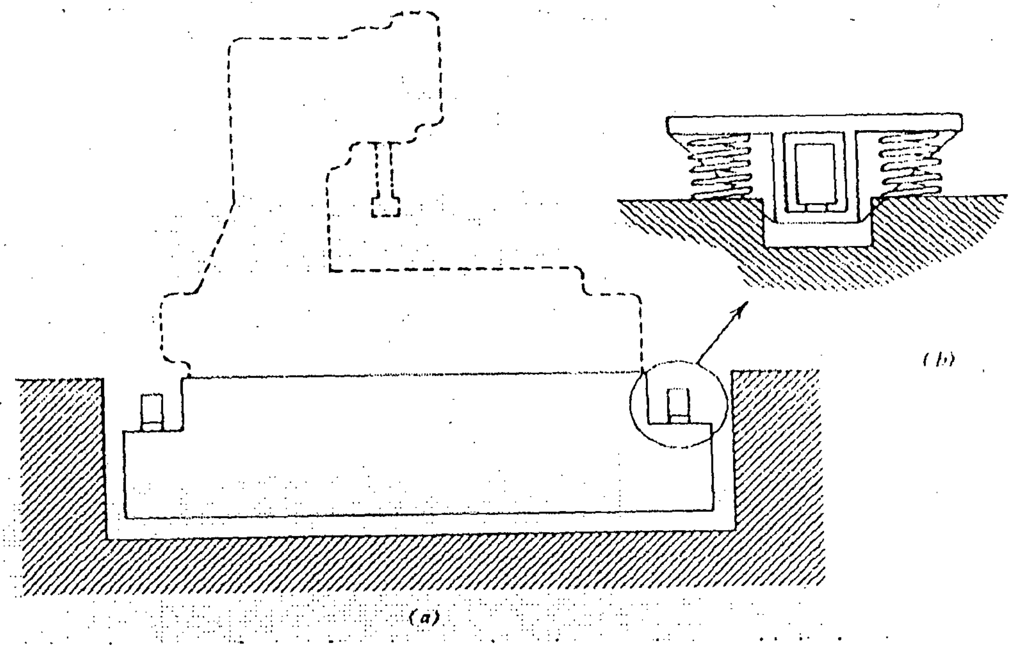


Fig 5.3. Centre-of-gravity mounting using cross beams and helical springs

Example 5.3

This example relates to an installation devised for the calibration of accelerometers in terms of the fundamental standards of length and frequency. Basically a calibration involves the measurement of the electrical output of the accelerometer while it is subjected to sinusoidal vibration of known displacement amplitude and frequency. As the range of calibration that is required in practice involves displacement amplitudes of only a small fraction of a millimetre, special methods – in this example optical interferometry must be used to determine the amplitude. Unfortunately, a method that is sensitive enough to measure the small calibrating vibration that is imparted to the accelerometer is sensitive also to unintentional and unwanted vibration of elements of the interferometer system; therefore, special attention must be given to the isolation of the calibrator from site vibration.

The mounting consists of a 1-tonne seismic mass on four air springs Fig (5.4). The mass is T-shaped so that the air springs are positioned to satisfy the requirements of a c.g. mounting. With this installation on a basement floor where vertical vibration was about 0.1 μm at 12 Hz; the vibration of the seismic mass was less than 0.002 μm.

Calibration by interferometry is costly and time consuming. The interferometric calibrator is used to calibrate only high quality standard accelerometers, which are then used to calibrate other accelerometers (reference accelerometers) by comparison: the standard and the reference accelerometer are subjected to the same vibration and their outputs compared. The reference accelerometer in turn is used in a similar way to check the calibration of "working" accelerometers.

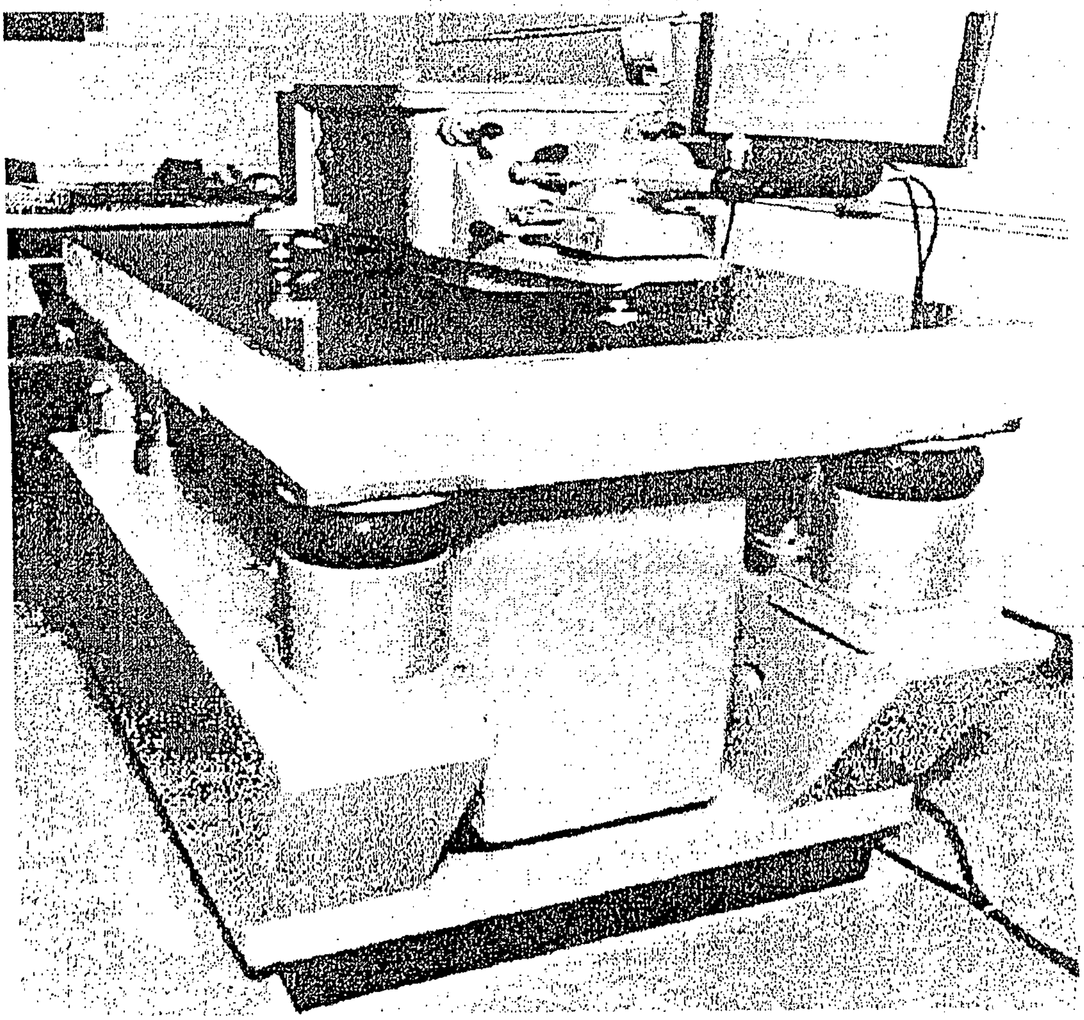


Fig 5.4. Centre-of-gravity mounting using T-shaped block on air springs.

5.1.3 Pendulum Mounting

A type of mounting for small instruments such as galvanometers, which was commonly used during the earlier decades of this century but is rarely used today because it is too cumbersome, consisted of a platform suspended as a pendulum. The platform was attached to wires or springs handing from the ceiling or from a tall tripod or a wall bracket. Perhaps the best known of these was the Julius suspension in which the platform was provided with a means of adjusting the c.g. of the suspended mass and of damping the free vibration of the system following manipulation or adjustment of the instrument.

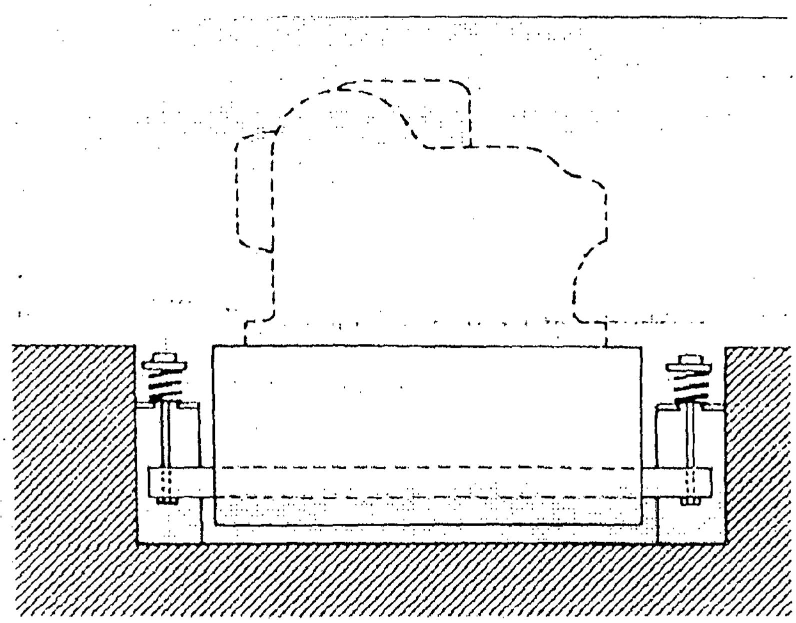


Fig 5.5. Pendulum mounting on helical springs.

A form of pendulum mounting that is sometimes used for large items of equipment is shown schematically in Fig 5.5. The equipment is fixed on an inertia block which is suspended on tension rods. The upper ends of the rods rest on helical compression springs and the lower ends are located in spherical seatings or the equivalent at the ends of through-beams in the inertia block. The pendulum mounting is suitable for large, low-speed engines having unbalanced, horizontally reciprocating parts, because of its low natural frequencies of horizontal vibration.

5.2 Isolators

The most important unit isolators are helical springs, rubber isolators and air springs, and units combining metal spring, rubber, and pneumatic elements. Area isolators currently used include ribbed or embossed rubber carpet, pads or mats of glass fibre, cork, or felt, and composite and layered combinations of such materials.

For a given seismic mass and geometric layout of isolators, the natural frequencies of the mounting are determined by the stiffness of the isolators. The damping requirements are satisfied by selecting a type of isolator that has adequate damping, or by using separate dampers such as those mentioned below under the heading of helical Springs, which have negligible inherent damping. In the discussion to follow about particular types of unit isolator we give special attention to these two properties, the stiffness and the damping.

A word of caution is necessary about the significance of the term "stiffness" as applied to vibration isolators having non-linear load/deflection characteristics. In the course of installation of a mounting each isolator receives a share of the weight of the seismic mass and deflects a certain amount, say 10 mm. When the mounting is in service the vibratory displacement of the loaded isolator below and above the mean position is very much smaller, say 0.1 mm. The stiffness that determines the performance of the isolator is not that associated with the relatively large static deflection from the unloaded condition, but that associated with the small oscillatory deflection of the loaded isolator in service.

Furthermore, the stiffness concerned is not the static stiffness in this range of small deflection but the dynamic stiffness, which is the restoring force per unit displacement that the isolator exerts on the seismic mass under the particular conditions of a frequency and amplitude of the vibration. For some isolators this dynamic stiffness is significantly different from the static stiffness value that is found by dividing the working load on the isolator by the deflection of the isolator under that load. The dynamic stiffness can be found only by making tests under dynamic conditions in testing machines designed for the purpose. Data on the properties of isolators under dynamic conditions should be obtainable from the manufacturers of the isolators.

5.2.1 Helical Springs

Helical springs are in such general use in engineering that there is no need to reproduce the design formulas or graphs that are used to design a spring to have a specified deflection under a given load. Design data for helical springs intended for use in seismic mountings, and examples of spring isolator units, can be found in Baker (1975), Church (1976), Crede (1976), Spring Research Association (1974), and in technical publications of the major spring manufacturers.

Helical springs can be designed to carry a load of a small fraction of a kilogram or more than a tonne. The springs shown in Fig 5.6 were used to isolate a 7-tonne camera.

Example 5.4

The 7-ton camera was carried on three groups of four springs made of 16-mm square-section steel in coils 117-mm mean diameter. The largest helical spring is reported to be made of material of diameter 76 mm mean coil diameter almost 0.5 m, and free height 1.2m; its static deflection was 25 mm under a load of 1.2 tonne. These *springs* were designed to support an entire building in an underground cavern, for use as a command centre to isolate it from shock and vibration in the event of a nuclear explosion.

Helical spring mountings can be designed to have a natural frequency as low as about 2 Hz for which the static deflection is about 60 mm. To attain significantly lower natural frequency as required static deflection of the spring would be excessive. For example, a compression spring for a 1Hz vertical natural frequency would deflect some 250 mm under the working static load.

The inherent damping in a helical spring isolator is negligible which is a disadvantage in applications where there is a possibility of unwanted resonance. Any necessary damping must be provided by separate means such as friction pads or fluid dampers. In fluid dampers a vane or plunger fixed to the seismic mass is immersed with large clearance in a highly viscous fluid in a pot fixed to the base or support. Dampers of the kind used as shock absorbers in vehicles are not used in stationary mountings because usually displacements of one end of the damper relative to the other would be too small to overcome threshold static friction.

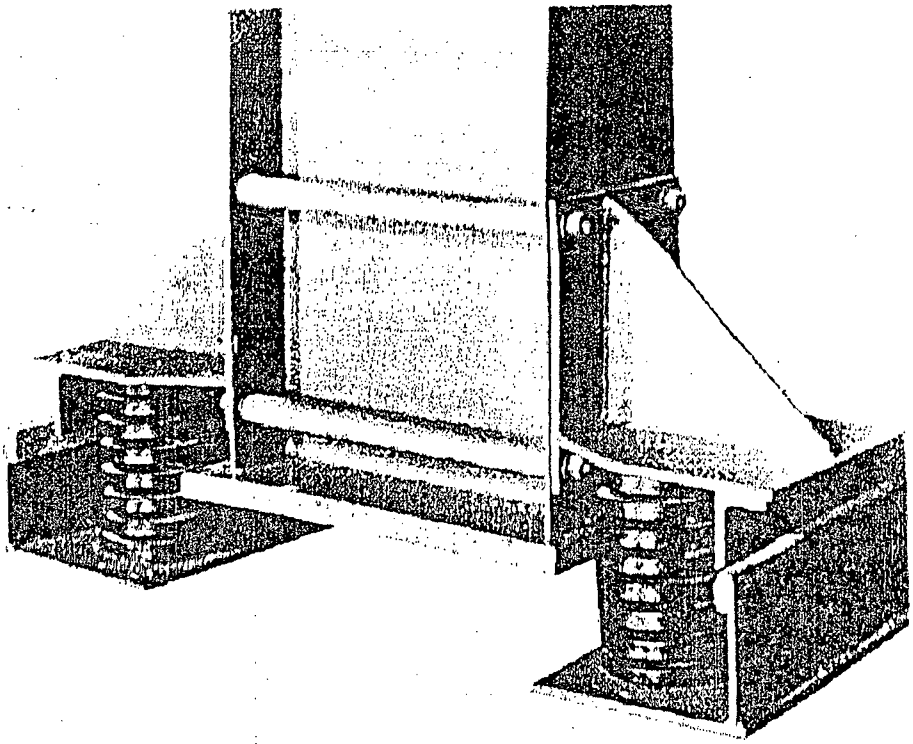


Fig 5.6. Helical springs used to isolate a large camera.

5.2.2 Rubber Isolators

Rubber has been used as a vibration isolating material for at least a century. The major developments in rubber isolators followed the discovery that metal could be chemically bonded to rubber, for this made possible the design of unit isolators in which the rubber could be loaded in shear or combined compression and shear while bonded to the metal parts necessary for holding the isolator in the required position.

Rubber isolators are commercially available in a very wide variety of shapes and sizes. Perhaps the most common type is the rubber-in-shear.

The design of a rubber isolator is a matter for the specialist involving the choice of material, natural rubber or a particular composition of synthetic rubber and the determination of the shape and size of the rubber to achieve the desired load rating and stiffness characteristics.

Example 5.5

A gear grinding machine sited about 75 mm from a forging hammer was installed on a 38-tonne inertia block and the total seismic mass of 54 tonnes mounted on 32 rubber-in-shear isolators with a static deflection of 5 mm. The vertical natural frequency corresponding to this static deflection for an installation on linear isolators is 7 Hz. The designer of the installation made some allowance for the effects of nonlinearity and dynamic stiffness and expected the natural frequency to be about 10 Hz.

The actual value was found to be about 17 Hz which unfortunately happened to be about the same as the predominant frequency of ground vibration transients resulting from operation of the forging hammer. Vibration measurements made simultaneously on machine and floor showed that the vertical vibratory displacement of the block was three to four times that of floor.

5.2.3 Bridge Bearings

Today rubber and synthetic rubber isolators are commercially available in compositions that are suitable for wide ranges of environmental conditions and in load ratings suitable for isolating a small instrument, a large machine, a bridge or a whole building. The largest rubber isolators which were developed as bridge bearings are in the form of a block of natural rubber containing a number of steel plates arranged horizontally and bonded to the rubber to form a multiple sandwich as shown schematically in Fig 5.7. The outer plates with their relatively thin protective coating of rubber are the bearing surfaces between which the unit is loaded in compression. The internal steel plates have the effect of increasing the compressive but not the shear stiffness.

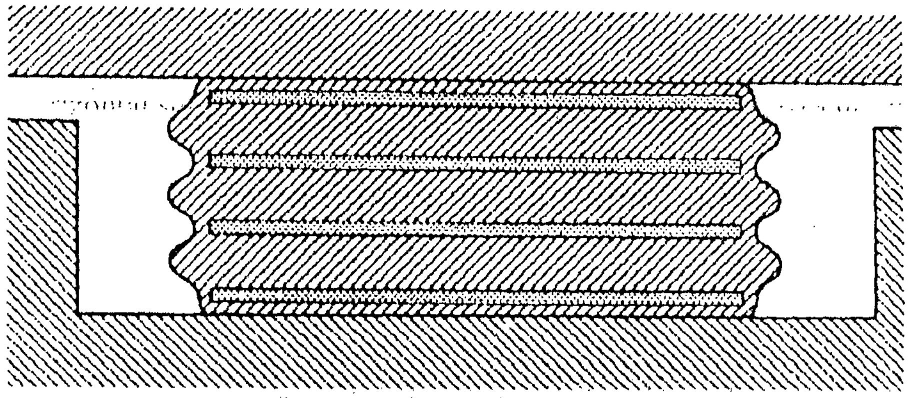


Fig 5.7. Bridge-bearing type of isolator.

5.2.4 Air Springs

An air spring is simply a container of compressed air, which supports a load. Air springs have been in use and development as vibration isolators over the past 50 years. In an early application, devised to isolate a small instrument such as a galvanometer, the air spring was an inner tube of a motor vehicle tire. The partly inflated tube was placed on a horizontal support and a platform carrying the instrument was placed on the tube. In another early application, the floor of a broadcasting studio was supported on an array of rubber bags connected to a compressed air supply. The system was designed to isolate the floor from the vibration of other parts of the building and to permit the room acoustics to be adjusted by controlling the pressure in the groups of air bags supporting separate sections of the floor.

5.2.5 Bellows Type

Fig 5.8 (a) is a cross section of a two-convolution air spring. The envelope is made of nylon-reinforced rubber, sealed to hold pressure typically up to about 700 kPa, and there are metal plates at top and bottom for locating and loading the air spring and through which the air is supplied. A small air spring is suitable for a design load of about 50 kg, and a large one, having an outside diameter of say 0.5 m, about 15 tonnes.

Example 5.7

A 320-kg seismic table 1.8 X 1.5 m for optical holography is supported on four air springs of the two-convolution bellows type, about 200 mm diameter. A high quality seismic table was essential for this work because the only room available had a suspended concrete floor which vibrated vertically with 0.5 μm p–p displacement due to activities elsewhere in the building, and much higher if an observer moved about the room. On this table highly satisfactory holograms were made which required exposure times up to 2.5 min, during which time the relative displacement of critical parts of the optical system must not exceed about 0.1 μm.

In the design of a mounting on air springs, as on isolators of any other kind, it is necessary to know both the axial stiffness and the ratio of lateral to axial stiffness of the isolator. The axial stiffness can be found from the manufacturer's data graph relating load and height for a given pressure. From a given initial condition (height, load, pressure) the change in height for an assumed small increase in load is read from the graph and the axial stiffness calculated as the rate of change of load with height.

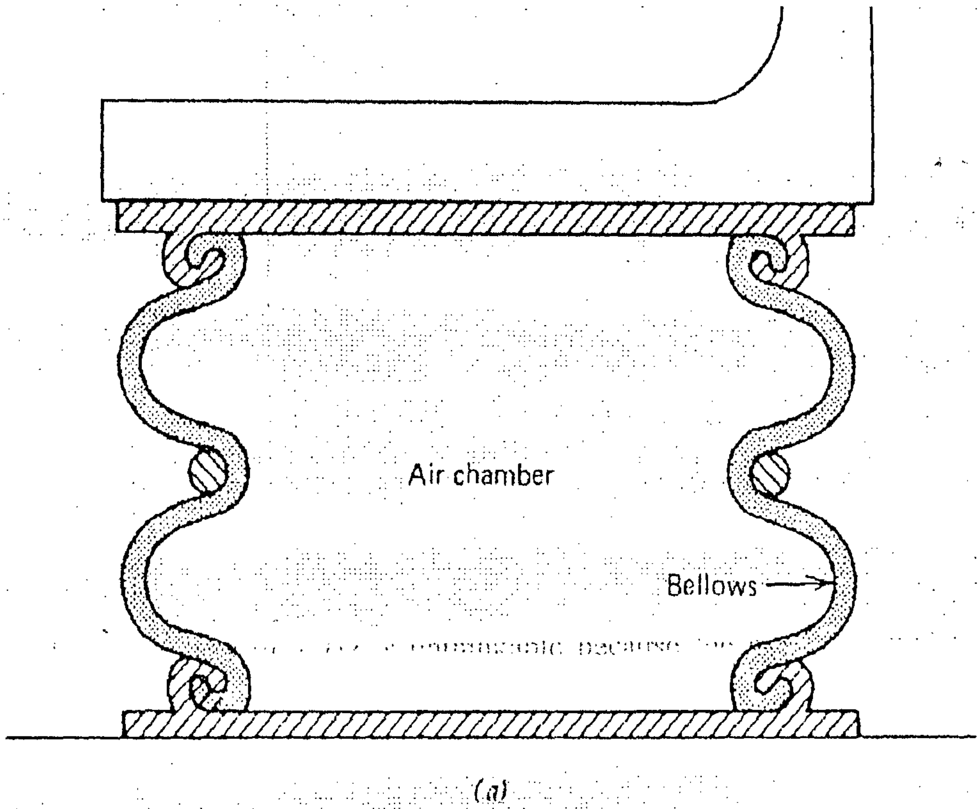
5.2.6 Air Spring with Height Control

The vertical load on an air spring is equal to the product of the "gauge" pressure of the contained air and the effective area of the air spring. During a vertical vibration, when the mass moves down a small distance from the rest position, the height of the air spring decreases by a small amount. This reduces the volume, increases the pressure, and provides an upward force tending to restore the mass to its rest position. The relationship between restoring force and deflection varies with the shape and material of the air spring envelope, and can be estimated by using the theory of air compression, or determined by making tests on the air springs.

An important conclusion can be drawn from a simplified theory based on the assumption that the effective area is constant; that is, the air spring is assumed to behave as a piston/cylinder element. By considering a small downward displacement of the piston and using the laws of gas compression to derive the increase in pressure associated with the reduction in volume, it turns out that the stiffness of the air spring, and hence the natural frequency of a given mass supported on the spring, depends only on the height of the air spring. A consequence of this is that if the load is changed and the pressure adjusted to keep the height the same for all loadings the vertical natural frequency is the same for all values of the load.

Automatic height control, which is obviously advantageous in vehicle suspensions, is valuable also in mountings for stationary equipment; for example, machine tools and weighing equipment which must be kept horizontal when the load and/or its distribution is changed. Air isolators have been developed which incorporate a height sensing device to control the air supply so that when the load is increased air is admitted to the air spring, and when the load is reduced air is released, thereby maintaining the spring at a constant height. The basic from of this type of isolator, shown schematically in Fig 5.8 (b) which consists of a metallic body or cylinder, which acts as an air chamber, and can support the load when it is nor supported by the air pressure. There is a flexible seal or diaphragm between the cylinder and the "piston" that carries the payload. The diaphragm operates nominally at zero deflection because the load is maintained at a constant height by a valve that admits air when the piston is below and releases air when the piston is above the mean position.

With air springs a natural frequency of a few hertz can be attained. Air springs offer the further advantage that by increasing the volume of air by means of a supplementary or "surge" tank, the natural frequency can be made lower than that attainable if the volume is limited to that of the air spring alone. In this way a vertical natural frequency of about 1 Hz is practicable. By contrast, a mounting on helical springs having a vertical natural frequency of 1 Hz is unthinkable because the springs would have a static deflection of about 250 mm, and correspondingly excessive height and diameter.



(a) Two convolution bellows type

Fig 5.8. Schematic diagram of air springs

With air springs a natural frequency of a few hertz can be attained. Air springs offer the further advantage that by increasing the volume of air by means of a supplementary or "surge" tank, the natural frequency can be made lower than that attainable if the volume is limited to that of the air spring alone. In this way a vertical natural frequency of about 1 Hz is practicable. By contrast, a mounting on helical springs having a vertical natural frequency of 1 Hz is unthinkable because the springs would have a static deflection of about 250 mm, and correspondingly excessive height and diameter.

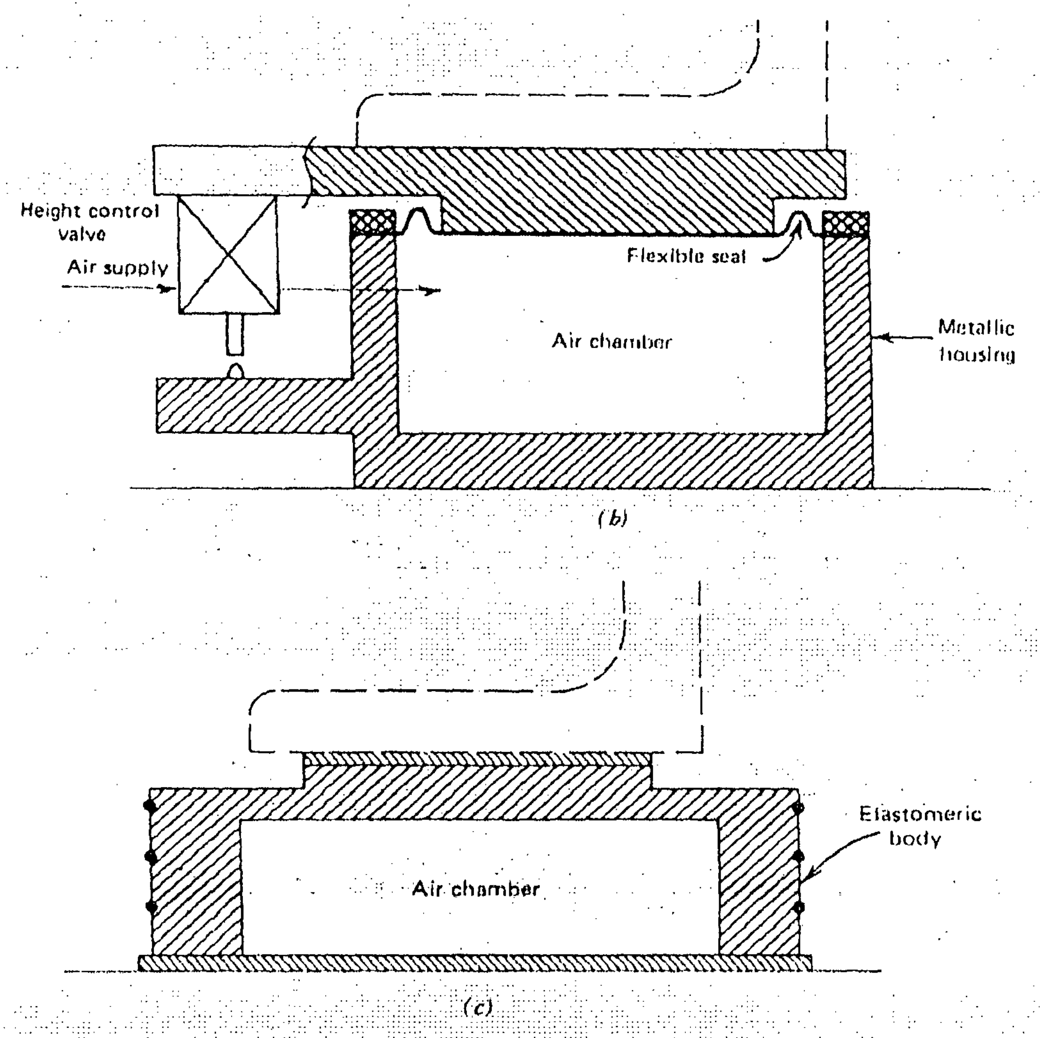


Fig 5.8 (continued). (b) Air spring with automatic height control

(c) Pneumatic-Elastomeric isolator

5.2.7 Pneumatic-Elastomeric Type

The theoretical and laboratory investigation of the prototype isolator of this type is described as below:-

The main features, shown schematically in Fig 5.8 (c), are an elastomeric thick-walled cylindrical body and a top shaped as a diaphragm coupling the body to the top plate. Unlike the thin wall of the bellows type, which serves only to contain the air, the thick wall of the pneumatic-elastomeric type can act as a temporary mounting pad during installation of the equipment, or at any other time when the isolator is not inflated. The thick wall also acts as a snubber if a vertical shock causes a downward displacement greater than that of the compliant top. The transverse stiffness is about equal to the axial stiffness, providing good transverse isolation and stability. Steel rings around the outside prevent bulging, and improve the deflection and stability characteristics. Isolators of this type are now commercially available in a wide range of load ratings.

5.2.8 Damping of Air Springs

There is some inherent damping in the bellows and the pneumatic-elastomeric types, due to straining of the flexible air chamber. Damping of an air spring can be increased by making use of the surge tank mentioned earlier as means of attaining a lower natural frequency. This additional damping is generated by restricting the flow of air between the air chamber and the surge tank. By making this restriction variable, the amount of damping can be adjusted to the optimum for the particular application.

To sum up, clearly the air spring has many attractive features. It offers natural frequencies of 1 Hz and lower which are impracticable with any other form of passive isolator; it is readily adaptable for use with a height sensing valve to maintain the height, level, and natural frequency of the mounting when the load or its distribution changes. On the debit side, air springs require regular inspection to ensure that the air pressure is maintained. The mounting is designed so that if for any reason the air spring becomes deflated the seismic mass will descend only a small distance and come to rest on a structural or other support, for example, the screw jack.

5.3 Performance Testing

The logical sequel to the design and installation of a seismic mounting is its performance testing to compare its actual behaviour with that intended by the designer.

Commonly, an installation, of which the mounting forms part, is accepted if it functions satisfactorily and without objectionable vibration.

Thus, for a source mounting, the criterion is that the vibration produced by a machine be acceptable to the operator of the machine and to those working or living in the vicinity. The following example (Baxa and Ebisch, 1978) is of interest for its application of this criterion in relation to an extremely severe source of vibration and for the subtle timing of the acceptance test.

Example 5.8

A company engaged in the crushing and shredding of scrap metal for recycling planned to install a 4000 hp hammer mill-type of automobile shredder. On learning that another company in this business was facing costly litigation arising from the operation of a similar plant, the company provided a carefully designed seismic mounting for the shredder. The hammer mill and its motor were installed on a 363-tonne inertia block, and the total load of 500 tonnes supported on 16 air springs of the two-convolution bellows type, with a vertical natural frequency on 1 Hz. The system was designed to isolate the very severe unbalance that would result from the loss of two hammers, each 190 kg at 1.2 m radius, normally rotating at 600 rev/min, and the shock resulting from the occasional explosion in the shredder of an automobile gasoline tank.

When opponents of the project protested at the city council meeting, that if the shredder were allowed to operate it would shake the neighbourhood apart, the company representative had the pleasure of informing the meeting that the shredder had been in operation for 2 weeks and nobody had noticed the vibration.

However effective an acceptance test of this kind, which is concerned only with the effects of the vibration transmitted through the mounting, it is not, in a technical sense, a performance test of the mounting, for which it is necessary to compare the transmitted (output) amplitude of force, or displacement, with the corresponding applied (input) amplitude. The experimental evaluation of a mounting on this basis presents some difficulty, which we now discuss.

5.3.1 Source Mountings

The purpose of a source mounting is to attenuate alternating forces, so that the force amplitude transmitted into the supporting structure is appreciably smaller than that generated by the source. The experimental determination of a "force transmissibility" as the ratio of the amplitudes of two alternating forces is not practicable. Although it may be feasible, for the purpose of experimental research or investigation to measure the forces generated by a machine (e.g., with force transducers designed into bearings), and the force transmitted by the mounting (e.g., with dynamic load cells under the isolators), the use of such methods in normal practical applications is unthinkable. This situation appears to have caused little concern because normally it is not the force but the resulting vibration that matters.

The performance of a source mounting can be determined by making only vibration measurements, if the vibration at a specified vibration-sensitive area can be measured with and without the mounting. This is obvious and attractive in principle, but in practice can be done only with installations of trivial size, such that the isolators can be removed and the equipment fixed to the floor for the test. For larger installations this is impracticable not only because of the labour and cost involved but also because of the difficulty of providing temporarily, for the purpose of the test, an effectively "rigid" fixing, in place of the isolators.

To sum up, a source mounting is designed on theory relating to force transmissibility, but it cannot be assessed experimentally on this basis.

5.3.2 Mountings for Sensitive Equipment

As with vibration sources, sensitive equipment installations are usually assessed only on overall performance. The installation is accepted if the equipment attains the required standard of performance or output, for example, a particular grade of surface finish produced by a grinding machine, a desired sharpness of the image in an electron microscope, or simply freedom from vibration of the reference line or scale in a measuring instrument.

The function of a seismic mounting is to minimise the "internal" vibrations of the equipment that deteriorate the quality of the output. Therefore, the performance testing of the mounting should involve a comparison of the vibration of the critical parts of the equipment with that of the site.

Examples 5.9

In tests on a jig borer (Example 5.2) the relative vibration between tool, spindle and worktable, with machine idle, resulting from normal and test excitations applied to the site, was measured with an air-gap capacitive transducer and associated displacement meter. One plate of the transducer was supported in the tool holder in the spindle, and separated by a small air-gap from the other plate which was attached to an angle plate on the worktable.

In tests on a roll grinder (Example 5.1) an air-gap capacitive transducer was set up to measure the relative vibration between wheel and roll, with machine idle, in response to site excitations. The plate was attached to the surface of the wheel, separated by a small gap from a similar plate attached to the surface of the roll.

In principle, the performance of the mounting can be determined by measuring the response of a critical element of the equipment, to a given site excitation, with and without the isolators. The author (J A Macinante) tried this method during the tests on the jig borer referred to in Example 5.2. Hardwood blocks were fixed with steel wedges between the inertia block and the pit, and the motion of the tool spindle relative to the worktable of the machine was measured with a certain test excitation (dropped weight) and compared with the response to the same excitation with the inertia mass "floating" on the isolators. However, vibration measurements made on the wedged block and the adjacent floor, with the test excitation, showed that the displacement of the block was appreciably less than that of the floor. Therefore, the test was abandoned because, if the wedging had been effective, the vibration of the block would have been the same as that of the floor.

In the tests of the gear grinder mounting referred to in Example 5.5, an attempt was made to nullify the isolators by using steel screw jacks to clamp the seismic mass. With 14 jacks tightened around the sides of the block and 17 under the block, the displacement of the block in response to a forging hammer blow was appreciably greater than that of the adjacent floor. Again the test was abandoned because the wedging was inadequate.

While the wedging in both of these cases may have been satisfactory if displacements of the order of one or two millimetres were involved, it was ineffective in these cases where the displacements to be compared were well below 0.1 mm. At such small displacements the blocks or jacks, in effect, merely function as stiffer isolators on which the seismic mass can vibrate.

5.4 Conclusion

To sum up, in normal practice it is likely that the acceptance testing of sensitive equipment installations will continue to be based on the satisfactory performance of the equipment and not on a specific evaluation of the mounting. This is reasonable from the viewpoint of the owner or user of the equipment, but provides no "feedback" for the designer of the mounting. The experimental determination of the quality of the mounting, expressed as a ratio of quantities representing the unwanted response and the site vibration, is quite practicable with existing vibration measuring techniques and instrumentation. However, the cost of testing of this kind may not be justified in normal practice, for its only purpose would be to verify the design of the mounting.

CHAPTER 6

ACOUSTICAL CONTROL

6.1 Introduction

The definition of noise as being unwanted sound cannot be bettered. Noise destroys efficiency. It is said that the more efficient a machine, the less noise it makes and it is predicted that the ultimate in machine efficiency precludes noise and vibration completely. The ultimate, may be a dream of the future, but today, it is certain that action to reduce noise and vibration at source, or to confine and insulate it, is an urgent necessity. In this section, we will consider various kinds of noises and means to control it.

6.2 Airport Noise

Noise associated with airport operation can be considered in a variety of ways, but a logical breakdown is (i) analysis of individual sources, and (ii) assessment of total noise. Even this leads to numerous different methods of treatment, particularly where subjective effects are concerned since these will vary with individuals, with time of day or night and even with the same individuals day by day. Frequency of noise occurrence, and nature of noise spectra can be as significant as actual noise levels on a subjective rating, whilst varying degrees of tolerance may be built up in individuals by familiarity with particular noises.

Individual aircraft will have different effects as noise sources when: -

(i) Taking off

(ii) Landing and

(iii) Running up engines on the ground.

The latter can also be considered as an 'airport noise' since the noise source is stationary and thus remains at a constant distance from any individual observer at a fixed point. Noise produced by conditions (i) and (ii) can readily be expressed by: -

(a) A single line curve representing overall sound pressure level at various distances from the sources.

(b) Spot values of overall sound pressure at a particular distance from the take off point, in line with the runway or otherwise co-ordinated.

(c) Sound spectra determined in octave bands at specific points related to landing and/or take off paths.

(d) Spot values of overall sound pressure, or octave band analysis during fly-over at specific height(s).

(e) Overall sound pressure contours establishing a complete fly-over pattern from the point of take off.

Fig 6.1 shows typical overall sound pressure levels for a multi-engine propeller driven transport aircraft and a jet airliner on take-off related to the fly-over path, i.e. distance from the point of take off. Three climb conditions are shown in each case, the value of a steep climb out being evident, particularly in the case of the jet aircraft. Similar curves are shown for a military jet fighter and jet bomber in Fig 6.2, the shallow climb being the lowest power consistent with safety after rotation.

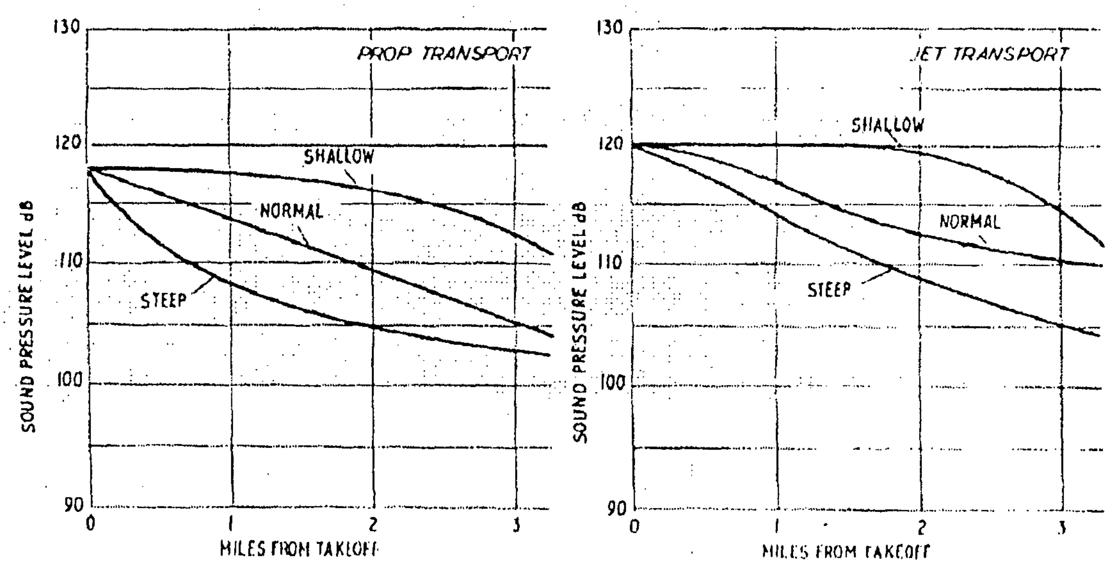


Fig 6.1. Overall sound pressure levels for a transport aircraft and a jet airliner on take-off.

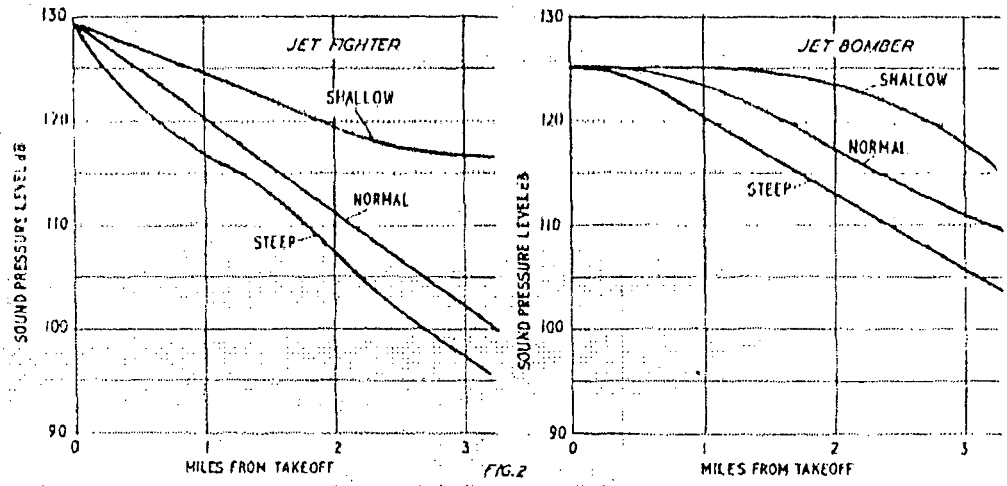


Fig 6.2. Overall sound pressure levels for a military jet fighter and a jet bomber on take-off.

Spectra measurements for a variety of aircraft taken in the fly-over path with the aircraft at a height of 2000 feet in each case are shown in Fig 6.3. In the case of the jet aircraft the thrusts are those commonly used following a power cutback for noise reduction at lower altitudes. The curve for the piston-engined airliner is for climb power.

A further, generalised, presentation of fly-over overall sound pressure level for various typical types of aircraft is given in Fig 6.4, the distance in this case being related to the slant range. Similar curves can be calculated directly for individual aircraft, although in all cases practical attenuation will be modified by changes in ambient air conditions, changes in the flight path and variations in power setting.

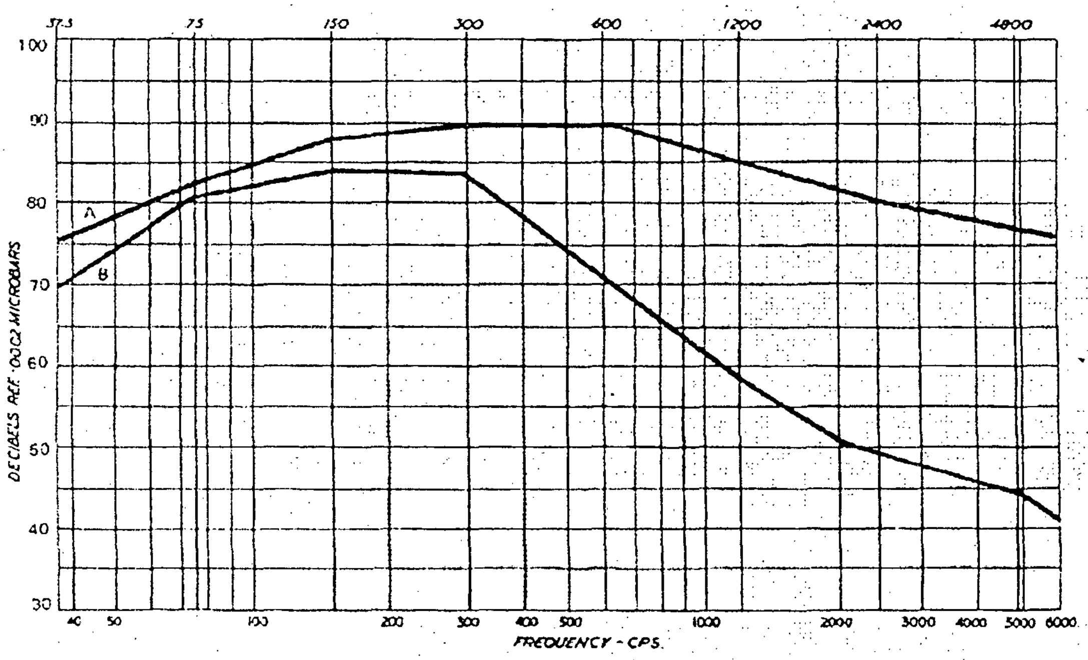


Fig 6.3. Typical Flyover sound spectra at a distance of 2000 feet perpendicular to flight path.

A. Four jet airliner. B. Four (piston) engine airliner.

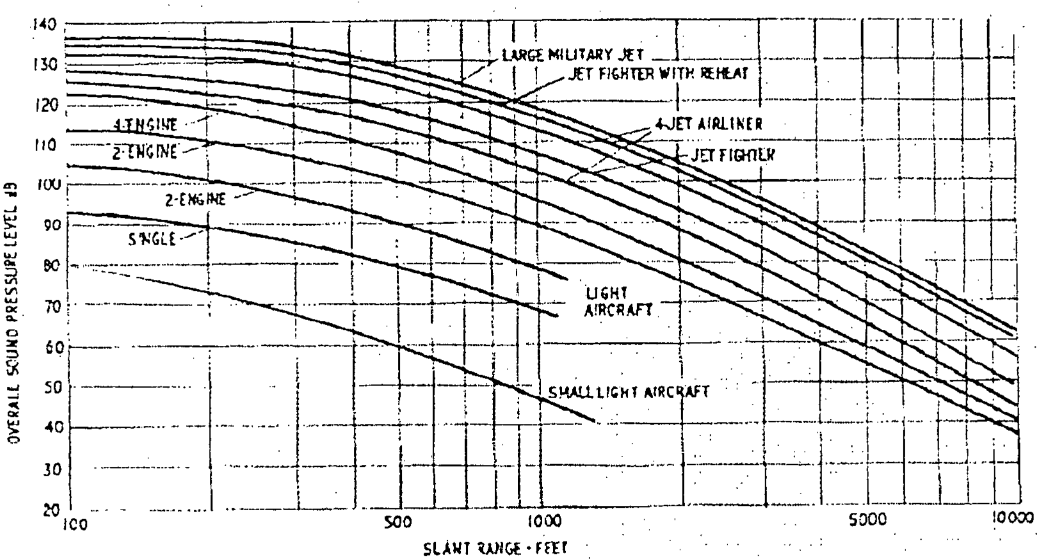


Fig 6.4. Overall sound pressure level for various typical types of aircraft.

Fig 6.5 shows two 'critical' overall sound pressure level contours plotted for a jet fighter and jet bomber aircraft. The 'critical' values are taken as 83 dB as typical of speech interference level with jet noise, and 40dB as representing the limit for sound, which will not interfere with sleep or rest. It will be seen that the latter extends upwards of two miles on each side of the runway and take-off path.

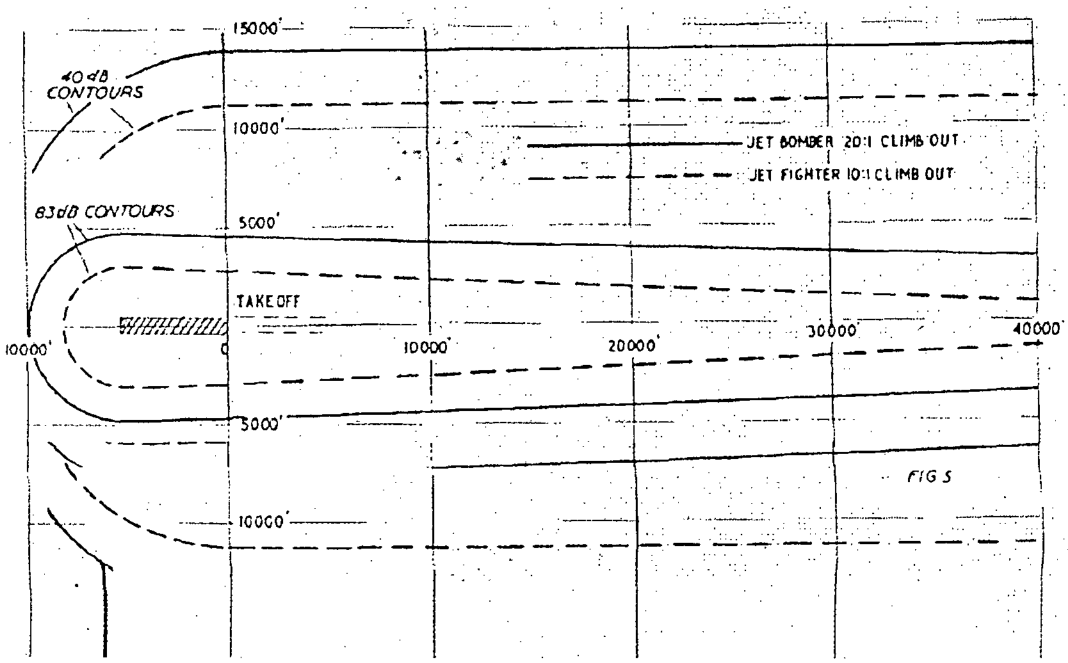


Fig 6.5. Two 'critical' overall sound pressure levels contours plotted for a jet fighter and a jet bomber aircraft.

Theoretically, at least, complete patterns could be calculated for the known frequency of operation of individual aircraft, runway utilisation, flight patterns, etc, when such data would establish a time pattern of total noise. In practice it is considerably easier, and generally more realistic, to determine such data statistically. The question then arises as to how best to present such data subjectively, since measurements have to be made objectively.

6.2.1 Airport noise control

The scale originally adopted by the Port of New York authority was the perceived noise level, quoted in PNdB, as giving a reasonable measure of the annoyance value (or subjective rating) of the noise from both piston and jet engined aircraft. The PNL is defined as the sound pressure level of a band of noise from 910 to 1090 cps that sounds as noisy as the sound under comparison.

Since PNdB takes into account all frequencies in the spectrum between 20 and 10000 cps there is no simple equivalent to overall sound pressure level measured in dB, or filtered sound pressure level measurement, e.g. dB(A). As a very rough approximation, however, the PNdB value for aircraft sound is about 14 to 15 dB higher than that given directly by dB(A) measurement.

Various surveys have justified the conception of a Noise and Number index (NNI), derived as

$$NNI = PNdB - 80$$

The figure 80 is taken as the PNdB value at which the subject rating of annoyance is zero (see fig 6.6).

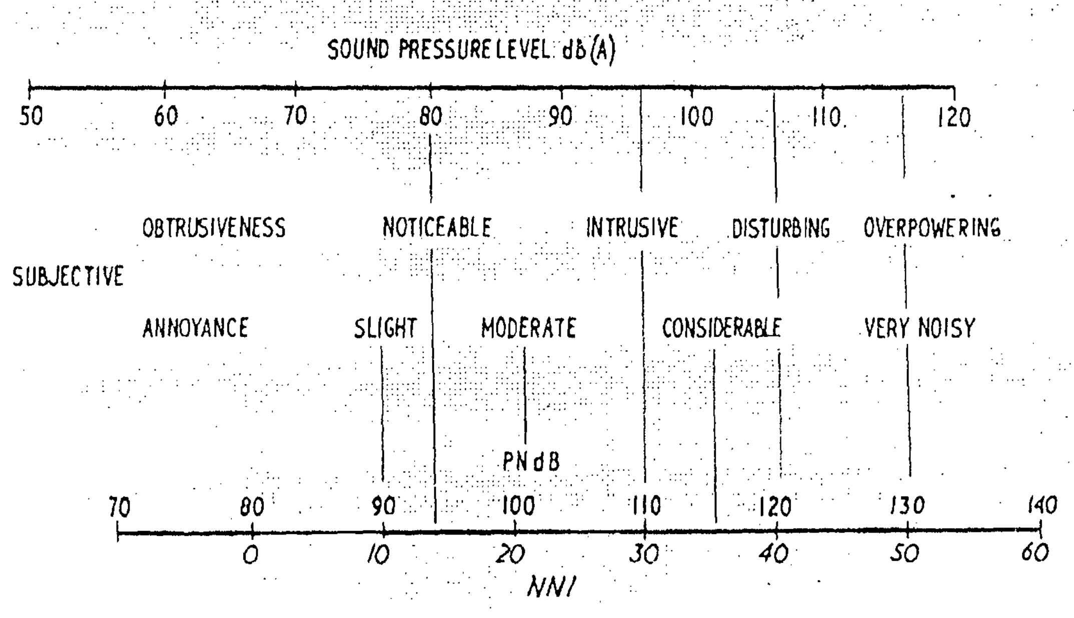


Fig 6.6. Rating of annoyance at various NNI levels.

Figures 6.7 – 6.12 show some of the measures taken in UK to suppress noise levels emanating from some typical kinds of aircraft.

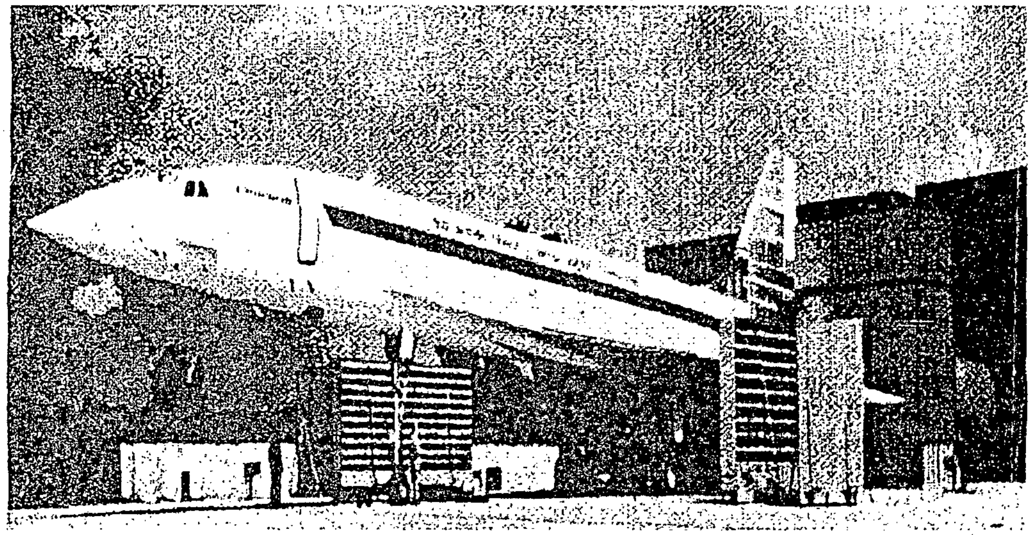


Fig 6.7. Ground run-up silencer for Concorde

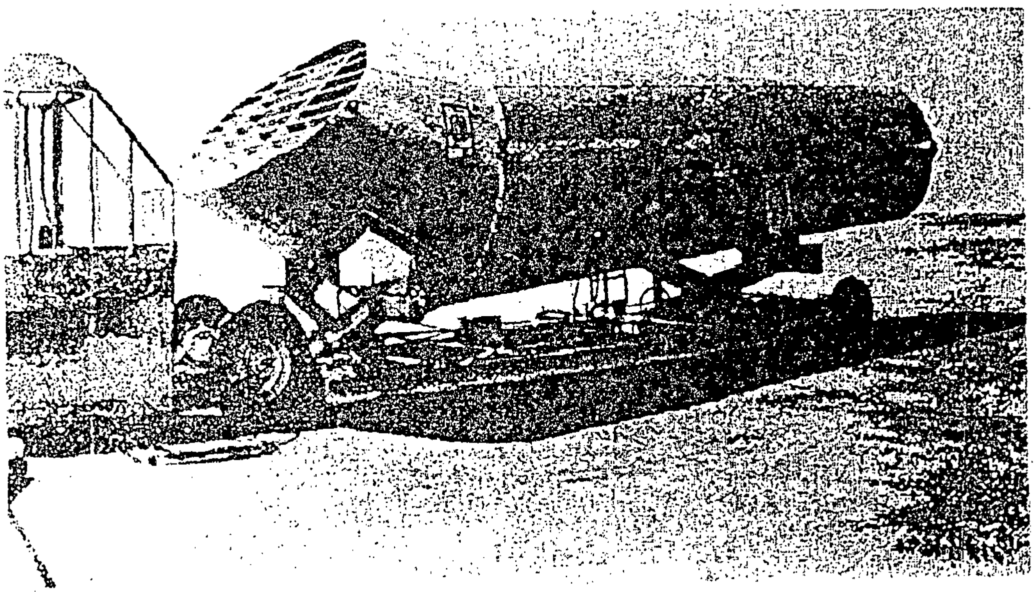


Fig 6.8. The IAC UDAC ground run-up suppresser for commercial jet transports is self propelled and enables the operator to position it in less than five minutes. A push-button controlled hydraulic system raises, lowers and tilts the suppresser for rapid attachment to engine.

Fig 6.9. Ground mufflers in operation at Avro Aircraft ltd., Malton, Ontario, Canada.

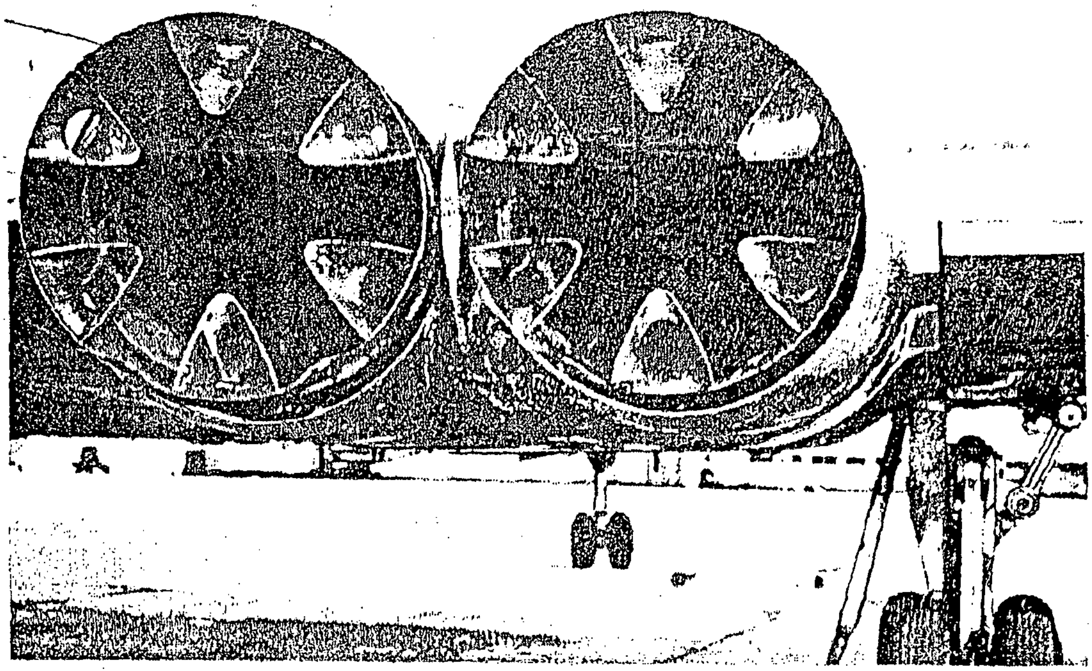


Fig 6.10. Jet silencers applied to Comet aircraft silencers.

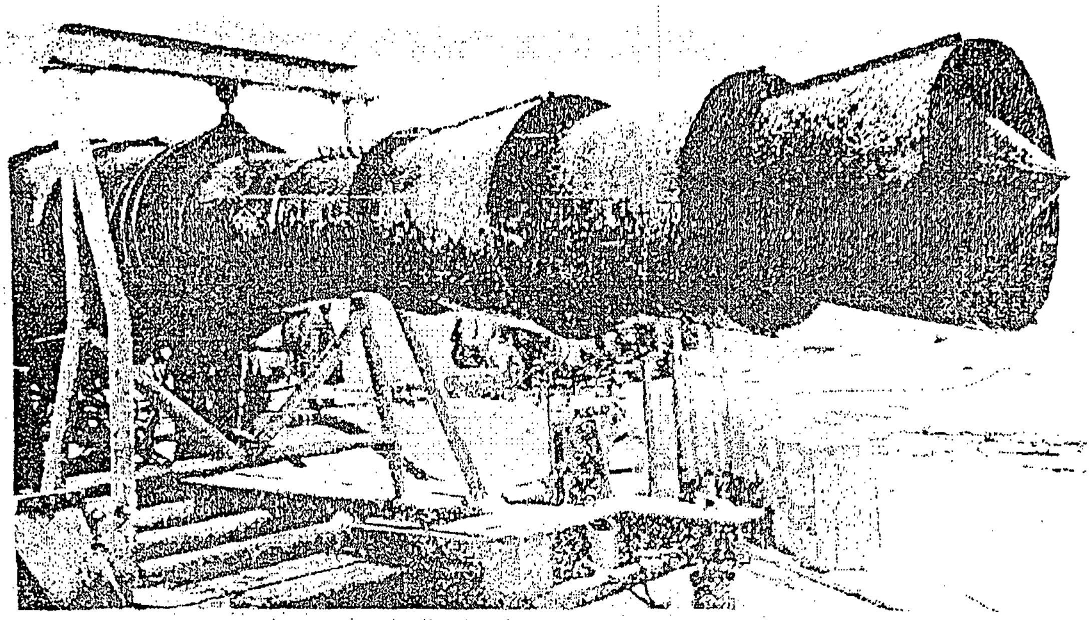


Fig 6.11. Typical Multi-jet noise suppresser with J-57 engine.

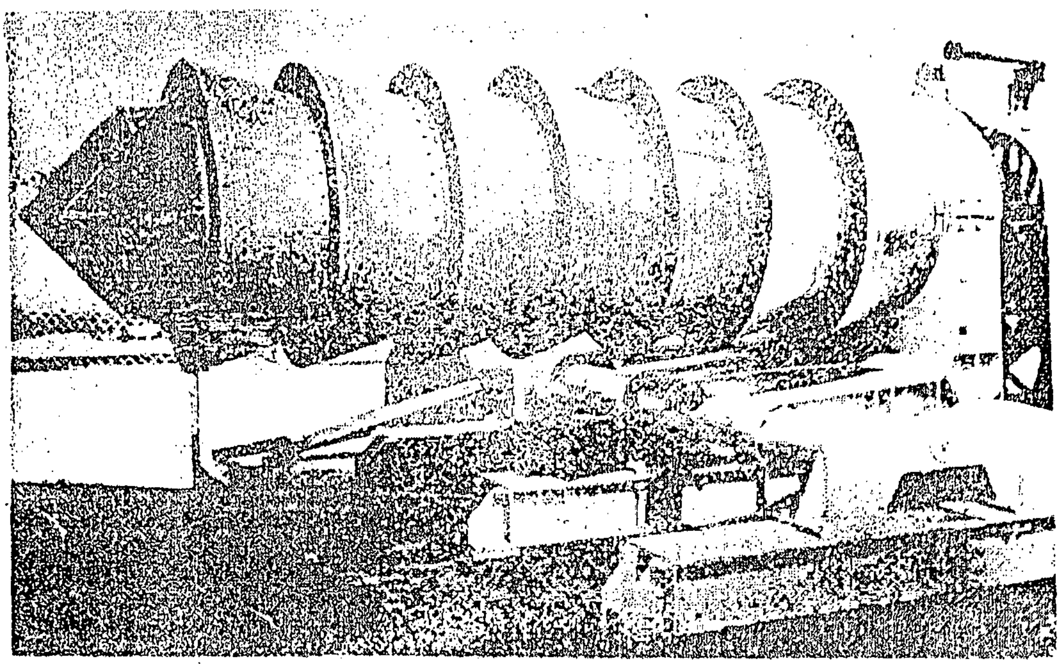


Fig 6.12. Industrial Acoustics Company (IAC) twin multi-jet suppressers, for engines of thrust up to 30,000 pounds.

6.3 Road Vehicle Noise

The only obligatory requirement as regards the noise level generated by a road vehicle is that it shall not exceed a specified legal limit for the country in which the vehicle is operated. Standards, both of measurement of overall sound levels and of the levels themselves, vary considerably from country to country – see Table 6.1. In effect this merely dictates that a silencer be incorporated in the design of such vehicles and in subsequent use the degree of silencing must be maintained within permitted maxima.

Table 6.1 – Summary of legal noise limits

Country	Sound Level Measurement	Mopeds	2-Stroke Scooters and Autocycles	2-Stroke Motorcycles	4-Stroke Motorcycles	Cars	Commercial Vehicles - Petrol Engine	Heavy Commercial Vehicles Petrol Engine	Diesel Engine Vehicles
United Kingdom	dB(A)	All 90							
Austria	dB(B)	80	80	85	90	85	85-90*	90	90
West Germany	dB(B)	75	75	80	80-82*	82	87	87	87
France	dB(A)	76	80	86	86	83	90	90	90
Italy	dB(B)	83	83	87-92*	90-92*	88-93*	88-93*	88-93*	88-93*
Denmark	dB(B)	All 73							
Holland	dB(A)	All 85							
Luxembourg	DIN phons	75	-	75-85*	75-85*	85	85	85	85
Switzerland	dB(B)	70	-	75-85*	75-90*	85	85-90*	90	90
Finland	DIN phons	75	75	82	84	85	85	85	85

* Depending on engine capacity

In practice, control of road vehicle noise as it affects the occupants is of primary importance in design as high noise levels are both objectionable and tiring to the majority of drivers and passengers. A low noise level, therefore, can be an important selling feature, except in the case of certain more specialised types where sheer performance is the major requirement. Even then noise cannot be ignored entirely, for excessive noise (excluding exhaust noise) can be an indication of excessive vibration which could detract from performance, or even be damaging.

It does not follow, however, that treatment of noise reduction from the 'interior' or occupancy aspect will automatically ensure a low level of 'exterior' noise. It is possible, by the suitable use of sound-deadening materials, isolation and suppression of vibration sources, to produce a vehicle which is extremely quiet for the occupants, but still has excessive airborne exhaust noise, engine noise or even road noise to the 'exterior' listener. In general, though, it can be anticipated that reduction of exhaust noise to low enough levels to be masked by engine noise will meet all statutory requirements and at the same time classify the vehicle as 'quiet' or 'reasonably quiet' to the external listener. Such a degree of silencing can usually be achieved without any appreciable reduction in power, although practical difficulties may be experienced in the case of some sports car with small ground clearance in accommodating the necessary silencer volume. Also with specific types of vehicles, such as diesel-engined commercial vehicles and two-wheeled and other vehicles powered by two-stroke engines, the actual engine noise may be regarded as 'objectionable', despite being well within permitted sound levels, because of the character of the noise spectrum.

6.3.1 Road vehicle noise control

As already indicated, certain generators of noise and vibration can be treated at source, e.g. by balancing, damping and silencing. The effect of remaining vibrations can then be reduced by isolation and damping as far as practicable. Solutions are not necessarily straightforward and depend very much on empirical work, particularly as a major source of noise is likely to be unanticipated resonance.

Isolation is applied particularly to the engine unit, which is mounted on resilient mounts. Unfortunately, complete isolation is not possible since the degree of resilience necessary for this would result in a mounting which would be much too flexible. The problem is further complicated by the fact that the support to which the resilient mounts are attached is itself relatively flexible and may itself be excited by road noise etc., or by feedback (of its own excitation) from the unsprung mass of the suspension. It is therefore necessary to work out the best possible compromise. Initially based on experience and such empirical data as are available and subsequently adjusted, or modified, as necessary in the light of experimental testing and final evaluation under true road conditions.

6.4 Railway Noise

Passenger Noise

Noise inside trains, or `passenger noise' is largely due to the vibrations produced by the rolling stock wheels rolling on the rails. Locomotive noise is largely negligible since this is unlikely to be transmitted back through the rolling stock, although airborne noise may be induced through openings (e.g. open windows). The primary source can thus be considered noise generated under the passenger carriage, which will vary appreciably with speed. A typical (generalised) sound spectrum is shown in Fig 6.13.

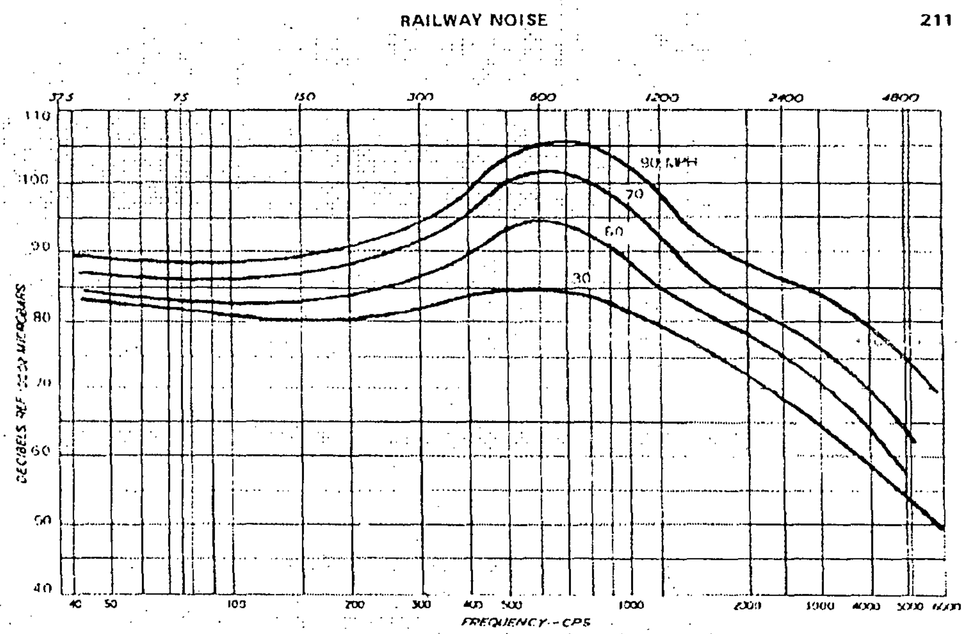


Fig 6.13. Typical coach underfloor noise levels (generalised levels)

Wheel/tract noise may be transmitted directly to the structure of the carriage, whilst airborne noise will strike the underside of the floor (and to a lesser extent the sides) and be transmitted through the floor to the structure, and directly to the interior through holes and cut-outs in the floor. The sound level inside the carriage will thus depend on how well transmitted vibrations are damped, and how well airborne sound is attenuated by insulation, etc. The noise level can also be expected to vary with position within the carriage.

On a subjective basis a noise climate at 80% or 90% level of 60 to 75 dB(A) would generally be regarded as acceptable, and indeed relatively quiet by railway passenger standards – see Table 6.2.

Table 6.2 – Subjective Noise Ratings

dB(A)	Rating
60	Extremely quiet
65	Very quiet
70	Quiet
75	Acceptable as reasonably quiet
80	Noisy
85	Very noisy
90	Extremely noisy
100	Objectionably noisy

Such levels are achieved on modern Pullman coaches, and some others, by the use of double glazing and sound deadening materials in the construction and a suitable degree of isolation applied to the suspension and sub-frame. Further treatment may also be necessary to ancillary equipment and ventilation to achieve such levels - see Table 6.3.

Table 6.3 – Passenger Noise

Source	Remarks	Treatment
Rail Joints	Primary source of running noise	(i)Replacement of jointed rails by welded track (ii)Double glazing of carriages (iii)Rubber (isolating) carriage suspensions (iv)Sound absorbing panelling in carriages
Rail corrugations	Developed by wear on hard steeled rails	Replace with modern carbon steel rails
Suspension	Related to rail joints and condition of track	(i)Vibration damping (ii)Sound absorption treatment (iii)Improve track (iv)Reduce unsprung weight
Ancillary Equipment	e.g. Vacuum brake compressor/exhauster	(i)Use isolating mounting (ii)Sound absorption
Ventilation	Provides entry for airborne noise	(i)Employ indirect ventilation (ii)Sound absorption treatment to carriage
Air conditioning		(i)Noise level controlled by design (ii)Ducts must be immune to vibration
Passing Train		(i)Double glazing (ii)Sound absorption panelling (iii)Damping of carriage slides
Couplings	Related to track condition etc.	(i)Suitable design of coupling system (ii)Suitable design of corridor connections

The use of sound-absorbing panelling and partitions will also reduce the intensity of airborne noise striking the sides of a carriage, such as when a train passes close to a wall or another train on the adjacent track. In this case additional airborne sound is directed specifically against the sides, which normally receive less insulation treatment then the bottom – see also Table 6.4.

A typical noise spectrum which might be achieved in a carriage with good isolation and sound insulation is shown in Fig 6.14. Noise reduction is more readily achieved at the higher frequencies and the reduction possible at the lowest frequencies may be comparatively small – see Fig 6.13. Another significant factor is that the noise level between 30 mph and 90 mph can be expected to rise by at least 5 dB, which is an appreciable difference subjectively. Also noise levels can vary much more locally within the carriage, or if a window is opened. Even a `quiet' carriage may thus be relatively noisy by other standards. The question of tunnel noise' poses particular problems with underground trains and this particular question has been studied since 1909 on the London Underground system. Sound levels have, however, tended to increase up until about 1940 when the use of acoustic tiles for tunnel lining up to about train floor level became standardised –see Table 6.5.

Further measures which have subsequently been adopted in an attempt to reduce noise include :-

(i) rubber carriage suspensions

(ii) rubber bushings for shoe gear

(iii) use of non-metallic brake blocks

(iv) quietening and improved isolation of compressors and ancillary equipment

(v) detail mechanical improvements in suspension and sub-frame design.

Table 6.4 – Sound deadening treatment to passenger carriages

Treatment	Result	Remarks
Double Glazing	Very Effective	Costly
Thickening Floor	Effective	Increases weight and cost
Sandwich Floor (Sub panel and Insulating Blanket)	Very effective in 200-1000 cps range of frequencies	Increases cost
Floating Floor construction	Quite effective	Less practical solution
Side Panel Insulation – Hardboard interior trim Hardboard facing	Very effective, especially at higher frequencies Moderately effective in controlling panel vibrator	Low cost treatment
Blanket Insulation Perforated Metal trim	Effective	Good strength
Blanket Insulation Perforated Hardboard trim	Very effective	
Blanket Insulation, Air space and trim panel	Effective at lower frequencies	Increased thickness required

Table 6.5 – Noise levels in London underground trains (windows open)*

Operating condition	1924 stock (dB)	1938 stock (dB)
Above ground, normal running	86-88	91-92
In tunnel	89-90	101-102
In tunnel with foam slag/asbestos board lining #	84-85	94-95
In tunnel with drilled asbestos felt tiles #	87-89	96-98

* London transport board
\# Up to coach floor level

The latest development in the design of 'quiet' carriages with double glazing, sound absorbent panelling, indirect ventilation and further suspension improvement.

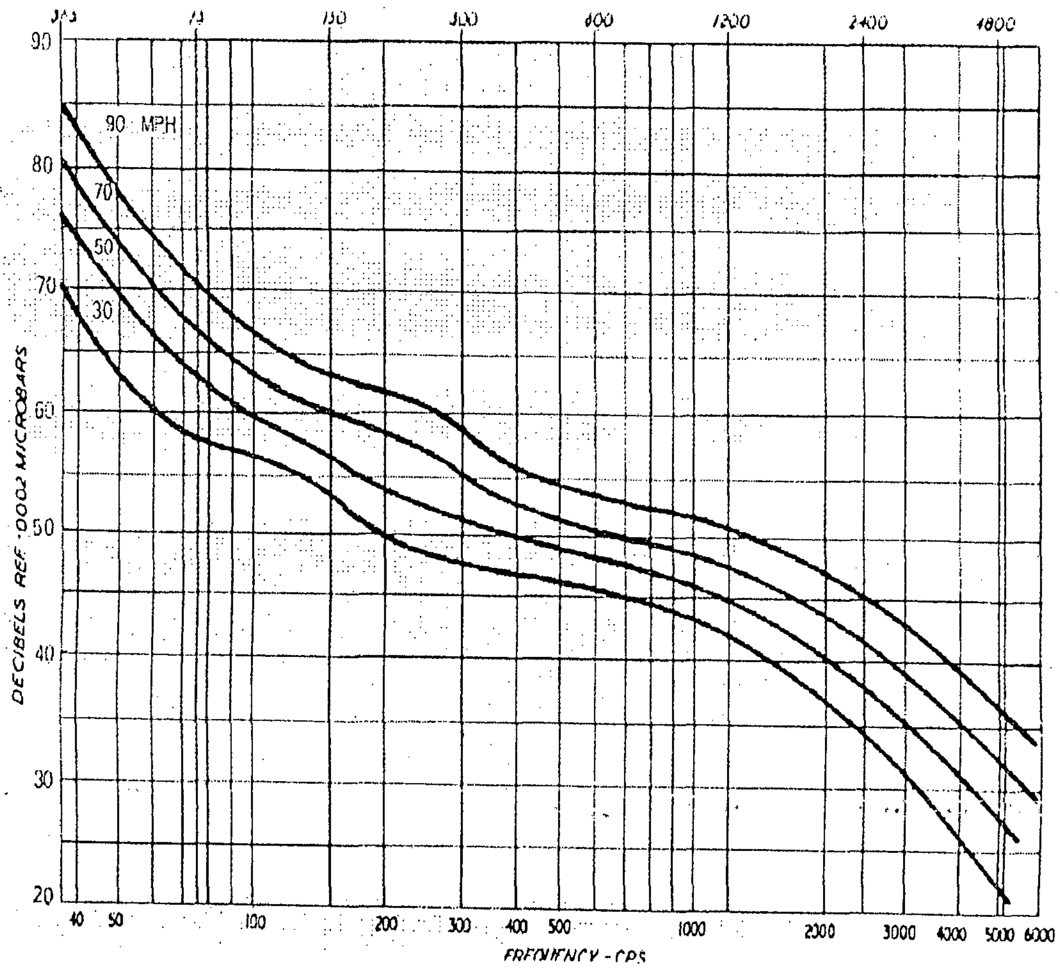


Fig 6.14. Typical coach inside noise levels (generalised curves, windows closed).

6.5 Building Noise

There are various ways in which plant noise could be reduced, but these are not necessarily applicable or practicable for specific work, and all place a major degree of responsibility on the contractors or plant manufacturers, or both. In a majority of cases, contractors tend to maintain that the problem of noise is largely out of their control; whilst manufacturers have, until recently, had comparatively negligible demand for quieter plant, although a number have commendably concentrated on this point, particularly in the case of pneumatic tools and compressors, and in the production of 'silent' pile drivers. Possible methods of noise reduction can be separated as follows :-

(i) Site Planning

This is an academic rather than a realistic solution in a majority of cases since necessary working points are usually specifically established regardless of surroundings However, intelligent planning can often increase the distance from adjacent buildings at which noisier plant work and thus effect some reduction in sound level reaching the building.

(ii) Use of acoustic barriers and screens

The use of temporary or portable barriers, screens and enclosures can be effective in reducing airborne sound generated by plant in use. Much of the desirable effect is lost, however, if the enclosure or barrier has gaps (particularly between enclosure or screen and the ground) or the enclosure is not intelligently employed (e.g. where directional sound is being generated it is not directed into an enclosure).

The use of enclosures and acoustic barriers is still very limited and much more attention could be given to this subject which offers what is an essentially practical and simple solution to noise control.

(iii)Silencing of tools

The use of silencers and sound insulation applied directly to the machine or plant can effect appreciable reductions in operating sound levels, particularly in the case of pneumatic tools – e.g. see Table 6.5. Further information on this subject is summarised in Table 6.6.

(iv)Proper tool maintenance and use

Whilst sound reduction can be achieved by silencing applied to the tool part or the whole of the gain can be lost if the equipment is not kept in good repair, or used in such a manner that the silencing is rendered ineffective. Silencers and mufflers fitted to tools as initial equipment may also be removed on site because they increase the bulk or weight of the tool and are thus regarded as decreasing the tool efficiency.

(v)Use of alternative tools

For a number of operations, appreciably quieter tools are available to do the same job as conventional drills, breakers and other percussive tools. Typical examples are electro-hammers replacing pneumatic demolition hammers and hydraulic drills operated by hydraulic generators. Currently the noise levels achieved by such alternatives are not necessarily appreciably lower than that of a silenced or muffled pneumatic counterpart, although the quality of the noise is generally more acceptable subjectively. They do, however, appear to lend themselves to further noise reduction, whereas conventional pneumatic tools are more restricted in what can be achieved in further silencing improvements.

The main limitation of such alternative tools is that they are nearly always appreciably more expensive than pneumatic tools, thus their practical attractiveness remains strictly limited at present. It is unlikely, too, that quieter tools would be chosen for general use unless (a) initial and maintenance cost became directly comparable; and (b) efficiency was higher, offering savings in operating costs.

Table 6.5 – Typical noise levels of pneumatic drills

Position of Observer	Tool without Silencer dB(A)	Tool with Silencer dB(A)
In line with exhaust,		
Distance 50 ft	80-84	70-76
Distance 25 ft	86-90	76-82
At right angles to exhaust,		
Distance 50 ft	82- 84	70-80
Distance 25 ft	88-90	76-86

(vi)Prime movers

Diesel engines are the most common source of prime movers providing power for construction sites and the silencing of such engines is not always as effective as it might be, since they are regarded as heavy-duty units. In many cases silencers may be omitted entirely. Even with suitable silencing, diesel noise can be objectionable subjectively, which is further aggravated by the fact that diesel engines on site are commonly left running continuously whether or not the plant is in use.

(vii)Alternative techniques

Certain constructional jobs may lend themselves to alternative techniques, avoiding the necessity of employing an inherently noisy plant. A typical example here is the alternative and quieter – constructional methods available to take the place of driven piles, such as the use of bored piles, timbering and shuttering, and concrete pouring in deep trenches. Again the main disadvantage with alternative methods is that they are usually considerably more costly. They are thus mostly applicable on construction sites where noise reduction is important or necessary and the additional cost can be justified

Table 6.6 – Silencing applied to compressed air plant

Tool	Noise Source(s)	Treatment
Percussive Tools	(i)Exhaust of compressed air (ii)Internal impact (iii)External impact	(i)Exhaust Silencers (ii)Padded jackets (iii)Screen enclosures
Compressors	(i)Compressor noise	(i)Intake silencers (ii)Jacketing
	(ii)Prime Mover	(i)Adequate silencing (ii)Reduction of engine noise

6.5.1 Silent Pile Driver

One of the most disturbing noises encountered on building and civil engineering construction sites is that produced by a piling hammer driving sheet steel piles where the instantaneous peak sound pressure may exceed a reading of 100 dB at a distance of 50 feet. This is appreciably higher than the typical noise level of other contractors plant. Various methods have, therefore, been investigated and designed to produce `silent' pile driving, or at least reduce the maximum noise level to a more acceptable figure.

On particular sites there may be a special need to maintain relatively low noise levels, e.g. building or construction work being carried out in the immediate vicinity of a hospital. Ins such cases the maximum desirable level of 76 dB(A) measured outside the nearest house or occupied building on the boundary of the site, as recommended by the 'Wilson Report' may still be undesirably (or even unacceptably) high.

The possibility of 'silent' pile driving can be tacked in two different ways, viz. :

(i) The use of a type of piling which does not require to be driven in place and can thus be erected in a more silent manner.

(ii) The use of a type of driver which does not rely on impact as a means of driving the piles.

Both methods yield practical solutions, but the use of special piling (i) is usually more costly and slower. The most promising solution, therefore, lies in the field of development of 'silent' pile drivers.

Here the choice would appear to lie between two basic methods :-

(a) A vibratory system where the pile is vibrated at a fairly high frequency. The motion set up in the pile is transmitted to the particles of the soil and under such conditions friction between them and the dead weight of the equipment is greatly reduced, causing the pile to move downwards through the soil under the dead weight applied.

(b) A ram system whereby a steady thrust is applied directly to the pile to force it downwards. A hydraulic ram is the obvious choice in this case since a very high thrust can be developed and delivered over the full stroke of the ram.

Both are practical systems. The vibratory method is particularly effective in the case of granular soils, but may be less so in other types. Also the rate of driving is directly dependent on the type of soil. Noise level is much lower than that with impact pile driving, but the basic disadvantage of the system is that the vibration effects set up in the ground can have disturbing effects on adjacent buildings or structures and this may prohibit the use of such equipment on certain sites.

6.6 Conclusion

Noise and vibration are ~~the~~ among the main disadvantageous bye-products of modern technological progress. Louder noise levels, more sources of nuisance, modern construction using lightweight building techniques and increased concern with amenity are reflected in requirements for sound insulation and noise control in many nations building regulations and codes. Utmost care should be taken to ensure that sound insulation in various industrial processes is properly adhered to as higher noise levels leads to annoyance and decreased efficiency.

CHAPTER 7

CONCLUSION

7.1 General

The following notes are intended to summarise the basic content of all preceding chapters. They are presented as an 'aide-memoir' to assist in any designs for vibration control.

The evolution of a practical solution to a vibration control problem involves five basic actions:

	Action	Procedure
1.	Define the vibration excitation characteristics	Analyse the vibration environment and develop a simplified mathematical model of the excitation characteristics
2.	Specify all vibration control performance requirements	Relate to subjective design criteria, vibration tolerance of machinery, stability and performance limits, structural integrity, etc.
3.	Select appropriate vibration control method	Review all applicable techniques and select one or a combination of methods to attain the performance objectives.
4.	Undertake analytical design of vibration control system	Proceed with theoretical analysis of systems considered under 3 above in order to quantify the static and dynamic performance of the selected system.
5.	Select necessary materials and / or fittings	Review appropriate products and control systems to meet constraints imposed by available space, loadings, methods c attachment, stability, visual appearance, etc.

7.2 Vibration Control

The following notes extend and develop some of the factors to be considered in the execution of the various stages of a design for vibration control.

7.2.1 Task 1

(a) Determine whether the objective of the exercise is to isolate the vibration source from structure (positive or active isolation) or to isolate a vibrating system from a potential recipient (negative or passive isolation). (Active vibration isolation normally involves consideration of both sub-audible (3-30 Hz) and audible (30 - 1000 Hz) frequencies. Passive vibration isolation normally involves the range of frequencies sensed by touch only (3-30Hz).

(b) If the vibration arises at rotating machinery, consider its degree of balance, rotational frequencies of all shafts, gear tooth meshing frequencies, out-of-balance forces in reciprocating machinery, compressor output pressure pulsations and all higher harmonics of fundamental frequencies and heat frequencies of all shafts and excitations mechanisms.

7.2.2 Task 2

(a) The limits for vibration transferred to buildings will be determined for audible frequency range by the appropriate background noise criterion (NC, NR or dB(A) ratings).

(b) For sub-audio ratio frequencies, the limits will be given by the Reiher-Meister or Deickmann systems of subjective ratings. Other systems can give advised limits for avoidance of structural damage, fatigue failure, etc., or the maintenance of machine stability, etc.

7.2.3 Task 3

Seven basic methods may be considered for the reduction of vibration amplitudes:

	Action	Procedure
a)	Reduce dynamic excitation	Improve inertial balancing; improve quality of manufacture; modify or redesign vibration source to reduce vibration acceleration (e.g. reprofile cams, reduce throw of cranks, reduce oscillating masses, etc.)
b)	Increase structural rigidity	The deflection amplitude of a vibrating member may be decreased by stiffening (which will increase its resonant frequency and its overall strength). Check that all fundamental and harmonic resonances of the stiffened structure occur or frequencies outside the range of vibration excitation.
c)	Detune resonant frequencies	Design equipment to ensure that resonant frequencies of members and components differ from one another and from predominant excitation frequencies. No natural frequencies of the assembly should coincide.
d)	Decouple vibration	Design equipment to ensure that coupled resonators are minimised in number or avoided altogether.
e)	Isolate vibration	Interpose a resilient element between the vibration source and the receiver. This is a 'broad band' control technique which can limit both the fundamental frequency and its harmonics. 'Stiffness mechanisms' used as resilient elements can consist of : Metal springs (strip or coil) Elastomers/rubber Elastomeric foam Cork, felt or composite materials Wire mesh Stranded mineral fibres Pneumatic/hydraulic systems Check that the selected system provides sufficient inherent damping for energy dissipation or add dampers as necessary.

f)	Absorb vibration	Attach an energy absorbing mechanism (a combination of a mass and a resilient element to function as an energy sink). This is a narrow band technique and can only be employed to control excitation at a single frequency. The level of vibration attenuation will depend on the degree of absorber damping and the degree of synchronism of absorber frequency and critical frequency.
g)	Damp structural resonances	Apply high energy dissipating coatings for fixtures to reduce the amplitude of vibration of structural resonances. These may take the form of viscoelastic coatings, visco-elastic-damped structural composites, built-up structural assemblies (slip-friction damping at rivets or bolts) or special proprietary dampers (slip-friction, viscous-shear or viscoelastic-shear damping).

7.2.4 Task 4

(a) For simple well-defined problems use 'short-form' charts or graphs to quantify isolated performance in terms of their static deflection or resonant frequency.

(b) For complex systems, a detailed analysis of the vibration response of all elements may be necessary, involving specialist design services and the use of a powerful computing facility.

(c) Quantify the damping performance of isolators or added treatment. Where necessary, establish the degree of compromise between resonant vibration control and high frequency vibration isolation.

7.2.5 Task 5

(a) Having established the preferred combination of techniques for vibration control, examine all potential proprietary equipment to determine their conformation with design requirements and their performance limits. Where necessary establish the degree of compromise between the desired and attainable targets.

(b) Check that selected equipment or treatments conform to requirements for resistance to fire, freezing, chemical attack, corrosion, high temperature working, etc.

(c) Ensure that final design makes provision (where necessary) for levelling the machine, mount compression by shipping bolts, access for servicing and inspection of treatment, etc.

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