UNIVERSITY OF ROORKEE (INDIA) ROORKEE

UNIVERSITY OF ROORKEE,
ROORKEE (U.P.)

Certified that the attached dissertation on "NOISE CONTROL IN AIR CONDITIONING SYSTEMS"

was submitted by

SRI T.L. SITHA RAMA RAO

and accepted for the award of Degree of Master of Engineering ~~in~~

APPLIED THERMO-DYNAMICS (REFRIGERATION & AIR CONDITIONING)

*vide* Notification No. Ex/39/P-65(Degree)/1962

dated June 29, 1962

*Dated* June 29, 1962

(S.S. SRIVASTAVA)
*Assistant Registrar (Exam.)*

29/6/62

PSUP(R) 126 Ch. 1961 -1,000

# NOISE CONTROL

IN

# AIR CONDITIONING SYSTEMS

CHECKED
1995

BY

**T.L. SITHARAMA RAO**

C.64.

DISSERTATION
for the Degree of
Master of Engineering

UNIVERSITY of ROORKEE CENTRAL LIBRARY
Library Acc. No: 62452
Date 6.11.62
ROORKEE.
C82

**DEPARTMENT OF MECHANICAL ENGINEERING**

**UNIVERSITY OF ROORKEE**
**Roorkee, U P. (India)**

1962

## - : CERTIFICATE : -

Certified that the dissertation entitled " NOISE CONTROL IN AIRCONDITIONING SYSTEMS " which is being submitted by Sri T.L. Sitharama Rao in partial fulfilment for the award of the Degree of Master of Engineering in APPLIED THERMODYNAMICS (REFRIGERATION AND AIR-CONDITIONING) of the University of Roorkee is a record of student's own work carried out by him under my supervision and guidance. The matter embodied in this dissertation has not been submitted for the award of any other Degree or Diploma.

This is further to certify that he has worked for a period of SIX months from December 1, 1961 to June 1, 1962 for preparing dissertation for Master of Engineering Degree at the University.

V. Narasimhan

Dated 4.6.62

(V.NARASIMHAN)
SENIOR SCIENTIFIC OFFICER,
CENTRAL BUILDING RESEARCH INSTITUTE,
ROORKEE, U.P.

## :- ACKNOWLEDGEMENTS :-

The author wishes to express his deep and sincere gratitude to Dr. V. Narasimham, Senior Scientific Officer, Central Building Research Institute, Roorkee for his able guidance and encouragement in the compilation of this dissertation.

The author is also indebted to Prof. M.V. Kamlani, Head of Mechanical Engineering Department, University of Roorkee, for providing facilities at the department to carry out this work.

June 4, 1962

(T.L. SITHARAMA RAO.)

********

# SYNOPSIS

It has been found that working efficiency of man increases when the environment in which he works is thermally controlled. In recent years emphasis has been laid in maintaining suitable temperature and humidity by air-conditioning the space, using the mechanical system of air-conditioning.

As is common with all systems where one attempts to reverse nature, air-conditioning systems also produce certain undesirable effects. The most important of these is the noise generated by air-conditioning systems. Noise produces deleterious effects on efficiency, with the result that the advantages gained by air-conditioning is offset by the noises that are generated. However, there are ways and means by which these noises are kept at reasonable levels so that they donot produce any ill effects.

Criteria for noise levels have been evolved in the West, based on extensive studies on normal human beings subjected to various high level noises. Based on these criteria one can design an air-conditioning system that produces the best result under the given circumstances.

This dissertation deals with the several aspects of hearing, noise sources, noise paths, noise criteria, vibration isolation and the methods by which noise control is effected.

As a typical example embodying the principles enumerated in the earlier chapters, the problem of noise reduction in an air-conditioning system serving a fairly large sized library has been discussed.

# CONTENTS

CONTENTS (continued).



Continued ........../

## Contents (continued)

and incorporating adequate noise control measures in the basic planning and design of a structure.

The control of sound, or noise, in buildings has become a necessity to a great extent by the advent of air conditioning. Indoor sounds that at one time would go unnoticed, because of the noise that entered the building through open windows and doors, now is a source of disturbance to occupants in an otherwise quiet area.

Most older buildings were built with an eye only toward substantial construction; thermal insulation was not a prime consideration nor were thermally insulated windows and entry doors. Cooling tower, blowers, fans, compressors are noises we all associate with a cooling system, but previous to the era of noise control we ignored them and were happy to be comfortable.

When entire buildings were air conditioned, windows were sealed, thermal insulation materials were used and outside openings were sealed to prevent the infiltration of untempered air. These practices are also recommended in sound control. No doubt this eliminated the outside noises due to people in the streets, the traffic noise, noise due to jets and aeroplanes, and numerous other types of exterior noises. But along with this noise due to the air conditioning equipment, itself became quite significant.

People got annoyed due to fan noise, rattle of the compressors, splash of water in cooling towers and the vibrations

## CHAPTER - I

## INTRODUCTION AND HEARING MECHANISM.

### 1.1 INTRODUCTION :-

People do not like noise. By definition, it is 'UNWANTED' sound. It may interfere with speech communication on jobs or in leisure activities, in certain respects it may affect behaviour, it may produce a temporary hearing loss, and if the noise level is high enough, it may cause permanent damage to hearing.

Noise control is, therefore, a matter of considerable social and economic importance. This has become increasingly true in recent years. In consequence it has brought together individuals of widely varying vocations who share a vital interest in the problem : acoustical engineers, physicists, electrical engineers, designers of military equipment, aeronautical engineers, mechanical engineers, air conditioning and ventilation engineers, builders, architects, city planners, public health officials, business executives, lawyers and compensation experts.

Manufacturers of house hold appliances are acutely aware that a quieter product means increased sales. Office managers are learning that the reduction of office noise may result in less personal fatigue and, possibly, increased efficiency. Jet engine manufacturers find it necessary to build engine test cells with complex acoustical treatment to insure against possibility of complaints or legal action from neighbouring residents. Architects are begining to realise the need for anticipating noise problems and incorporating adequate noise control measures in the basic planning and design of a structure.

The control of sound, or noise, in buildings has become a necessity to a great extent by the advent of air conditioning. Indoor sounds that at one time would go unnoticed, because of the noise that entered the building through open windows and doors, now is a source of disturbance to occupants in an otherwise quiet area.

Most older buildings were built with an eye only toward substantial construction; thermal insulation was not a prime consideration nor were thermally insulated windows and entry doors. Cooling tower, blowers, fans, compressors are noises we all associate with a cooling system, but previous to the era of noise control we ignored them and were happy to be comfortable.

When entire buildings were air conditioned, windows were sealed, thermal insulation materials were used and outside openings were sealed to prevent the infiltration of untempered air. These practices are also recommended in sound control. No doubt this eliminated the outside noises due to people in the streets, the traffic noise, noise due to jets and aeroplanes, and numerous other types of exterior noises. But along with this noise due to the air conditioning equipment, itself became quite significant.

People get annoyed due to fan noise, rattle of the compressors, splash of water in cooling towers and the vibrations due to the mechanical equipment installed. With the wide applications of air conditioning in places like Television and broad - casting studios, auditoriums, concert halls, hospitals and others, noise control became very significant.

The aim of noise control is not to eliminate all noise but to produce a balanced noise environment. Generally, noise should be reduced only to the extent needed to allow normal activity to proceed with reasonable comfort. The quieting of an air - conditioning system much below the level of noise created by normal activity is wasteful. Furthermore, excessive quieting is often detrimental since it reduces acoustical isolation of one area from another, some background noise masks conversation and other distracting noises in adjacent areas by over-riding these weaker sounds and rendering them inaudible or unintelligible. Frequently the only acoustical privacy found in large industrial office space, drafting rooms, purchasing departments, etc., results from the masking effects of the ambient noise. Hence the amount of noise reduction is dictated by the use of the space and the activity going on and this depends upon the noise criterion. Hence noise control is not the same as noise reduction.

The most satisfactory solution of a noise control problem in airconditioning systems requires the close cooperation of the architect, air conditioning engineer and the acoustical engineer.

## 1.2 THE HEARING MECHANISM :-

Noise has no meaning for a deafman. Hence, for a due to the mechanical equipment installed. With the wide applications of air conditioning in places like Television and broad - casting studios, auditoriums, concert halls, hospitals and others, noise control became very significant.

The aim of noise control is not to eliminate all noise but to produce a balanced noise environment. Generally, noise should be reduced only to the extent needed to allow normal activity to proceed with reasonable comfort. The quieting of an air - conditioning system much below the level of noise created by normal activity is wasteful. Furthermore, excessive quieting is often detrimental since it reduces acoustical isolation of one area from another, some background noise masks conversation and other distracting noises in adjacent areas by over-riding these weaker sounds and rendering them inaudible or unintelligible. Frequently the only acoustical privacy found in large industrial office space, drafting rooms, purchasing departments, etc., results from the masking effects of the ambient noise. Hence the amount of noise reduction is dictated by the use of the space and the activity going on and this depends upon the noise criterion. Hence noise control is not the same as noise reduction.

The most satisfactory solution of a noise control problem in air-conditioning systems requires the close cooperation of the architect, air conditioning engineer and the acoustical engineer.

1.2 THE HEARING MECHANISM :-

Noise has no meaning for a deafman. Hence, for a better understanding of many problems of noise control, some knowledge of the mechanics of hearing is helpful. The ear is a biological detector of sound, a physical mechanism that can be injured by too intense a sound. In the following pages the anatomy of hearing mechanism and the effects of noise on hearing are described.

The hearing mechanism shown in cross section in Fig.1.1, can be divided into three parts; the external ear, the middle ear and the inner ear. The external ear consists of an external appendage, called the pinna, and the ear canal, which is closed at the inner end by the ear drum. The middle ear contains three tiny bones or ossicles which transmit vibrations from the ear drum to the inner ear. These bones - the hammer, anvil, and stirrup-constitute a lever mechanism that communicates the vibrations of the drum to the membrane of the oval window, which is the entrance to the inner ear. Since the oval window is only about one twentieth as large as the ear drum, the pressure of the vibrations communicated through the oval window to the liquid in the inner ear is increased. Thus the action of the middle ear is that of an efficient mechanical transformer, coupling vibrations in the ear to the liquid in the internal ear. The inner ear has two distinct functions ; (1) the maintenance of body equilibrium, accomplished by the vestibular portion of the ear, which is made up principally of three semicircular canals, and (2) the perception of sound, which is accomplished by the cochlea and its associated apparatus.

The cochlea is a liquid - filled spiral canal, subdivided along its length into two canals by a bony structure and

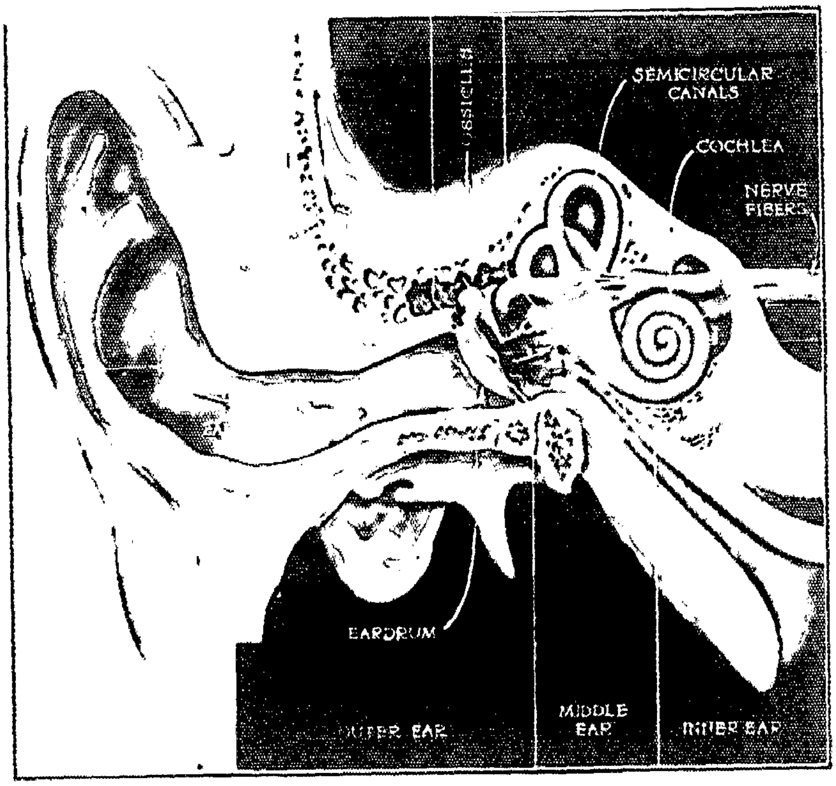



Fig. 1.1 Cross-sectional drawing of the hearing mechanism. (Courtesy Western Electric Co.)

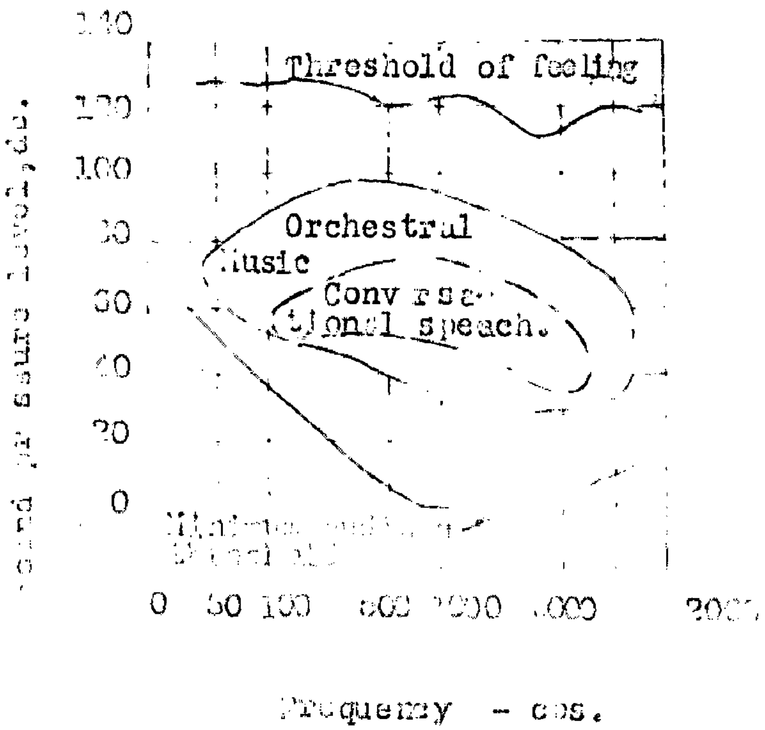



Fig. 1.2 [illegible] showing the [illegible] versus [illegible], and the [illegible].

a tough membrane. The end of one of these canals is closed by the oval window. It is through this window that the vibrations of the ossicles is transmitted to the liquid in the cochlea. Sensitive nerve endings, associated with tiny hair cells in a third canal, are excited by the vibration set up in the cochlear liquid and they send impulses to the brain by the way of the nerve fibers. It is believed that the rate at which the total number of these nervous impulses are communicated to the brain determines the loudness. This rate depends on the number and activity (pulsing rate) of the nerve endings stimulated. It increases with the sound pressure of the wave striking the ear. The pitch of the sound sensation is determined principally by the location of the nerve endings that are most excited by the resonant vibration of various sections of the basilar membrane; however, at low pitch the frequency of arrival of the nervous impulses at the brain may be the chief determinant. Tonal quality is determined largely by the number, location, and extent of the excited nerve endings and is complexly related to the wave form and the pressure of the sound wave striking the ear.

1.3 SENSITIVITY OF THE EAR(6)* :-

A sound wave must have a certain minimum value of pressure in order to be heard by an observer. This value for selected observers, who have good hearing, who are facing the source of plane progressive waves and listening with both ears, is called the minimum audible threshold for a free field. It is shown in the lower [illegible] of Fig. 1.2. Frequency is indicated along the horizontal

* [illegible] in parenthesis indicate the numbers in the list of

axis, and the pressure level of the plane progressive sound wave that is just barely audible is indicated along the vertical axis. It is clear from the figure that the sensitivity of hearing varies enormously for sounds of different frequencies. Fortunately our is most sensitive in the frequency range that is most important for the intelligibility of speech sounds. Since, in the evolution of man speech and music were developed later than the sense of hearing it appears that speech and music have developed in such a manner as to be well adapted to the sensitivity characteristics of the ear.

An observer in a free plane progressive wave will notice that, as the pressure of the wave is increased, the resulting sound becomes louder and louder until it attains a level at which the sound can be felt (a sort of tingling sensation) as well as heard. This level is called the threshold of feeling. Above this threshold the observer experiences a mixed sensation of sound, feeling, and pain. Fig. 1.2 shows that, unlike the minimum audible threshold, the threshold of feeling varies relatively little with frequency. The minimum audible threshold curve, if extrapolated at both ends, will intersect the threshold of feeling curve at two points which determine the lower and upper frequency limits of audibility, namely, at about 20 cycles for the lower limit and at about 20,000 cycles for the upper limit. These are the average values for young persons with good hearing. The upper frequency limits, along with the sensitivity for the higher frequencies, generally decreases with increasing age.

The ability of the ear to differentiate small changes of sound pressure or frequency is of importance in the hearing of speech and music. Anything that interferes with this function of the ear renders the understandability of speech or music more difficult. It is, therefore, of interest to inquire about the ear's capability in this respect. The minimum perceptible increment of sound-pressure level of a pure tone varies with both pressure and frequency, but, for levels greater than about 40 db above the threshold of audibility and for frequencies between 200 and 700 cycles, the minimum perceptible increment in pressure level varies from one quarter to three quarters of a decibel. The smallest perceptible change in frequency that the ear can detect is different for different pressure levels and frequencies, but, for pure tones more than 40 db above threshold and for frequencies greater than 500 cycles, it is of the order of 0.3 per cent for monaural (hearing with one ear) listening with an ear-phone. The variation of the frequency of a sound source in a room produces a marked variation in the pressure distribution within the room. Therefore, an observer can detect a smaller change in the frequency of a source in a room than in open air.

1.4 EAR RESPONSE TO SOUNDS AND DECIBELS :-

The intensity of sound is measured in the absolute value of watts per square cm or in the more basic units of ergs per second per sq. cm. A measurement of intensity in these terms means very little to us since we are interested in the physiological and not physical aspect of sound. Others in the field are also interested in the physiological aspect of sound and to this end the decibel scale has been set up.

The human ear is stimulated by sound according to the Weber - Fechner law which states, in effect, that the response of the human ear (and all the sensory organs) varies as the logarithm of the intensity - that is, a note of 10 units of power is 10 times louder than note of 1 unit, but a sound of 100units is only 20 times louder, while a note of 1000 units is 30 times louder than the 1 unit tone.

The unit used to express this ratio of power levels is the decibel. Mathematically the decibel (db) may be expressed as :-

$$db = 10 \log_{10} \frac{P_2}{P_1}$$

For example, if $P_2 = 1000$ and $P_1 = 100$,

$$db = 10 \log_{10} \frac{1000}{100} = 10$$

In this case relative numbers were used and the expression 10 db would mean little unless we have a reference power.

The reference power was set up by testing many thousands of young people with good hearing and determining at what intensity a 1000 cps note was audible. This level was determined to be $1 \times 10^{-16}$ watts per sq. cm. This level is termed 0 db and all db levels are in ratio to this level. Therefore, when a level is quoted at 30 db, it means that the intensity is 1000 times greater than the accepted base level.

The range of sound intensities encountered in practice is so very great - the ratio of some of the highest intensities (for example, the sound intensity of a rocket is about 100 watts per sq. cm) measured to the faintest intensity that the average ear can hear comes to about $100/1 \times 10^{-16} = 10^{18} : 1$ - that is would be extremely awkward and cumbersome to work with just sound intensity in watts per square centimeter. For this reason, it is very convenient to relate these powers on a logarithmic scale, which is the decibel scale. This enables one to handle such a tremendous range of values very conveniently.

## 1.5 PHYSIOLOGICAL AND PSYCHOLOGICAL EFFECTS OF NOISE (7):-

Noise is defined as unwanted sound. Undoubtedly a sudden violent noise such as an explosion causes change in the rate and regularity of heart beats and is a great shock to the whole nervous system. But such noises are not present in an airconditioning system. It is fair to assume that a persistent series of noises, even though less intense than an explosion, must have an accumulative effect, probably the first symptom of excessive and persistent noise is a violent head-ache. It disturbs the sleep, concentration and causes fatigue. The most direct physiological effect of noise is upon the ear, and temporary or even permanent deafness may result, prolonged exposure to noise day after day, say, in an office, certainly produces a steady deterioration of hearing. When temporary deafness does occur, the hearing loss is usually greater in the higher frequencies (2000 - 4000 cps) even though the predominant energy in the noise is in the lower frequency region.

A great deal of attention has been paid in recent years to the psychological aspect of the effect of noise, and although the factor involved in the human reaction to noise are difficult to assess, it has been shown in some cases that excessive noise adversely affects the output of workers and increases the liability to error. This is probably the result of distraction caused by the noise which brings about lack of concentration, though a contributory cause of the decreased output and increase in error, may be the effect of noise on the health of the workers. In some cases relief has been obtained by giving an absorbent treatment to the ceiling of the noisy room, but in this connection the conclusions of H.J. Sabine and Wilson are important. After a series of investigations they reported in 1943, that relief has been obtained by absorbents in some spaces even where loudness reductions were negligible, and they made suggestions relating to the phenomenon of reverberation and to what they called the "spreading effect" which might be very helpful. They concluded that reverberation is regarded as an irritant because it prolongs an already disagreeable sound and also because it interfere with conversation. They suggest that reverberation becomes an important factor when the noises consist of distinctive impacts, and that the addition of reverberation to the steady noise of rotary machines is not annoying.

The importance of the "spreading effect" which is described as the tendency of sound in reverberant rooms to remain at the same level, or to decrease slowly with increase in distance from the source, lies in the fact that it causes the other noises to interfere with the operator or occupant to hear the radio and other appliances. As far as the symptoms of strain and fatigue due to noise are concerned, the occupant will be affected more by other unwanted noises than the sound (say music) he wants to hear even though the latter, being closest to him, sets up higher levels at the ear.

In brief, noise causes fatigue and irritation, interferes with the speech communication, annoyance and in some cases, where there is continuous exposure to high level noises, temporary or even permanent hearing loss.

---

## CHAPTER - II.

## AIR-CONDITIONING SYSTEMS:

### 2.1 INTRODUCTION :-

Air conditioning is the simultaneous control of temperature, humidity, purity and distribution of air in a a space. Now-a-days comfort air conditioning is no longer considered a luxury, but a necessity. In industries also air-conditioning is largely adopted.

In laboratories, it provides comfortable conditions for the occupants so that their work and measurements will be precise. It may assist medical and biological personnel in studying the effects of temperature and humidity on living things. In textiles and printing industries, control of humidity being essential, necessitates air conditioning. Precision parts require conditioned spaces so that they are maintained in good condition. Air conditioning is largely adopted in steel manufacture, pharmaceuticals, photographic products and other industries.

Comfort airconditioning is applied in residences, restaurants, offices, automobiles, railway cars, airplanes and in a large variety of spaces. The occupants feel comfortable, fatigue will be lessened, and their efficiency is increased.

A space can be airconditioned either by a central system or by a self contained unit. The term, central, applied to an air conditioning system implies that the equipment such as fans, coils, filters, and their accessories are designed for assembly in the field rather than in the factory as a unit. As the central system usually serves several different rooms, or spaces, individual controls are required for each room.

2.2 FEATURES OF CENTRAL SYSTEMS :-

The central systems are generally located in the basements and only one apparatus serves all the rooms. Such systems are usually connected by ducts to all the rooms, and preferably have exhaust fans that may effect complete removal and disposal of any desired proportion of the air.

Central air-conditioning systems are served by heating and refrigerating equipment which may be located at some distance from the air-supply apparatus, and which may serve one or more central air-supply systems.

Fig. 2.1 shows a typical year round central system. The refrigerant circuit, which includes the refrigerant compressor condenser, evaporator and cooling tower, is not shown.

Outdoor air enters from an intake louvre preferably on that side of the building least exposed to solar heat, and not close to the ground or dust gathering roof. The return air duct or connection comes from the return air fan. The return air and outdoor air connections are so arranged that there is complete mixing of the two to minimise the possibility of stratification through the equipment. All the air passes through filters where the air is cleaned. The cleaned air passes to the equipment which changes its temperature and humidity. Except in very warm climates heating or tempering coils are required to warm the entering outdoor air to a temperature above freezing, the heat being supplied by means of steam or hot-water. Cooling coils provide the necessary sensible cooling and dehumidification. The coils may be chilled by direct expansion of the refrigerant within the tubes, or by a pump circulated liquid such as water or brine. The cooling and dehumidification can be obtained by having a spray chamber and these should be followed by eliminators to remove droplets of water if any.

For humidification in dry weather, water sprays are used either separately as shown or combined with the cooling coils. Then we have the fan, driven by an electric motor, to pump the conditioned air to different spaces. Reheating coils utilising steam or hot water reheat the air in warm weather for control purposes or bring the air to its final temperature in cold weather.

The system damper provided controls the amount of air entering the duct before the duct branches off. Before the air is supplied into the conditioned space there may be a plenum chamber (an air compartment maintained under pressure, and connected to one or more distributing ducts). Again another damper called diffuser damper is provided to control the volume of air entering the space. Finally the conditioned air enters the space through the diffusers located either in the walls or in ceilings. The vitiated air is drawn back to the conditioning equipment through the return air duct. This can be done by the same supply fan if its suction pressure is sufficient or else a separate fan has to be used.

## 2.3 UNITARY-CENTRAL SYSTEMS :-

Many different types of central air-conditioning systems with room units of various designs have been developed for multi-room buildings, such as office buildings, hotels and hospitals. The primary object in using these systems is to save space by reduction of duct sizes or by entirely eliminating ducts. These units are generally placed beneath the windows. There are three common unitary central systems as described below :-

### 2.3.1 Induction Convectors :-

Induction convectors, Fig. 2.2, employ nozzles that produce a high velocity air jet. The high velocity jets of primary air induce a flow of air from the room through coils located in the secondary air-stream and supplied with chilled water in summer and with hot water in winter. The chilled water removes a large portion of the sensible heat in summer and the hot water supplies the sensible heat loss in winter. The required flow of primary air is greatly reduced due to the fact that a portion of the sensible load is carried by the secondary air stream. Since the primary air quantity is small very high velocities can be maintained in the supply ducts without requiring fan power in excess for a conventional system. The primary air is treated in the usual manner to reach the required dew point and a surface or spray dehumidifier may be used. The primary air quantity is sufficient for ventilation purposes and frequently consists of entirely outdoor air.

### 2.3.2 All-Air high-Velocity Systems :-

All-air high -velocity systems are air-conditioning systems in which duct velocities and static pressures are such that special control and acoustic treatment is required for proper introduction of the conditioned air into the space to be served.

These systems are of either single duct or dual-duct types.

In the single-duct system, the supply temperature is established and variations in the rooms are compensated by throttling the air supply volume.

Dual-duct systems deliver the entire air supply to cold and warm air supply ducts, from which it is distributed to air mixing valves, (a mechanical device that serves to proportion air from cold air duct and warm air duct into a common outlet or duct) or acoustical termination device (an air distribution unit consisting of an air valve, acoustical attenuation chamber, and an air outlet).

The proportion of cold air and warm air is thermostatically controlled.

2.3.3 Fan and Coil Units:-

Another type of room convector used in connection with central cooling and heating plants is known as the fan and coil unit.

The units, like induction units, are located around the periphery of a building, usually under windows, and are equipped with fans and a water-type heating-cooling coil. They normally take air for ventilation directly from outdoors, and have a manual damper for adjusting the quantity of air within certain limits. Hot water supplied during the heating season frequently is varied in accordance with outdoor temperature. Cold water for use during cooling season should be supplied at a fixed temperature low enough to provide the proper amount of dehumidification.

2.4 EVAPORATIVE COOLING :-

In climates where on the hottest days, the outdoor wet bulb depression is relatively great, it may be possible to replace mechanical refrigeration, or other cooling sources and use the evaporative cooling effect.

2.5 PRECOOLING :-

When sufficiently cold water from wells or streams is available, a saving in refrigeration may be obtained by the use, in location ahead of dehumidifier, of precooling coils through which cold water is circulated. This results in decrease in load carried by the dehumidifier and refrigeration plant.

2.6 SELF-CONTAINED UNITS :-

These units are generally of smaller capacity than the central systems. The refrigerating compressor along with its motor, condenser, heating and cooling coils and the fan for circulation of air are housed in a sheet metal casing with angle iron frames and removable panels to give access to the different parts. Generally the inside of the casing is lined with sound absorbing materials.

A room air conditioner is also a self-contained unit. This is a factory made encased assembly designed primarily as a unit for mounting in a window or through a wall, for free delivery of conditioned air to an enclosure and without ducts for conditioned air supply or return. It includes a primary source of refrigeration and dehumidification and means for circulating and cleaning air, and may also include means for ventilating, heating, or performing other functions. In these units the components are so designed that the noise generation is low enough.

# CHAPTER - III.

## SOURCES OF NOISE AND THEIR APPRAISAL.

### 3.1 INTRODUCTION :-

In any air-conditioning system, the components generate as well as transmit the noise. Those that generate noise are the sources of noise and hence we must know the share of each component in the total noise as well as their noise spectrum reaching the conditioned space. This information is necessary to determine the proper noise control measures.

### 3.2 ACOUSTIC TRANSMISSION PATHS :-

Fig. 3.1 shows a typical air-conditioning system, of course, excluding the refrigerant compressor, motor, spray chamber, cooling tower etc. The sketch also shows the different paths by which the noise can be transmitted to the conditioned space.

Path A :- is through a thin concrete slab to spaces above and below the equipment room. These slabs, although satisfactory structurally, are inadequate from the view point of providing sufficient transmission loss to prevent the air borne transmission of the noise to the spaces above and below the equipment room.

Path B :- is through the partition separating the equipment room from an adjoining office. Again we have a sound barrier that is not adequate for the job.

Path C :- partially shown consists of a door in the partition separating the equipment room from the office.

Path D :- shows the direction sound may take over the partition and door through the acoustical ceiling into the office.

Path E :- Indicates how sound may be carried through the duct work and through the walls of a duct and suspended acoustical ceiling to the room below. This can occur through the supply duct as well as the return air duct.

Path F :- Shows the most common path of all, through the supply duct work and out of the supply grille.

Path G :- indicates the passage of sound through the return air duct work against the flow of air and out of the return air grille.

Path H :- is the sound transmitted through the supply duct from the adjoining room where there may be a noise source. This happens when different spaces are connected to a common supply duct.

In addition there may be noises transmitted to the space from outside due to the traffic, automobiles, aircraft and etc.

Hence we can group the primary noise source in an air-conditioning system according to the principal way, in which the noise from these sources reaches the occupied space.

1. Noise transmitted through or along ducts :
   a) Mechanical equipment noise such as fans, motors, compressors etc.
   b) Self noise from air motion and turbulence within the distribution system.
   c) Cross talk from one room to another.

d) Noise and vibration transmitted through duct walls.

2. Noise transmitted through building structure resulting from :

   a) Machine vibrations.

   b) Transmission through walls, ceilings etc. from mechanical equipment or other rooms.

3. Noise transmitted from sources external to the building.

The self noise includes the noise generated by grilles, pressure regulating valves, dampers, turning vanes, bends, branches and etc.

3.3 FAN NOISE :-

Before going into details about fan noise, we shall briefly discuss the types of fans.

In the broad sense, fans are generally understood to be air-moving devices using a centrifugal or axial flow type of air propulsion. Vane or lobe types of displacement blowers, usually, are excluded from the fan category and are very rarely used in air conditioning work.

3.3.1 TYPES OF FANS (11) :-

Fans may be divided into two general classifications, centrifugal and axial (see fig. 3.2) . In centrifugal fans the flow through the impeller is essentially radially outward from an axis of rotation; centrifugal force causes a flow and compression of the mass of air enclosed in the rotor. In axial flow fans the flow is essentially in an axial direction. Occasionally a combination of axial and centrifugal actions is used, which is referred to as a mixed flow fan. Axial flow fans that operate against little referred to as propeller fans. They are used chiefly for exhaust and circulating purposes.

(a) Centrifugal Multi-blade Fans :-

There are three general types of centrifugal multiblade fans, a division that depends upon whether the tip of the fan blade curves into or away from the wheel rotation or whether it is radial (see Fig. 3.3)

1. Forward-curved blade fans direct the air into the fan scroll with a velocity greater than the tip speed. The blade shape and velocity diagrams are shown in Fig. 3.3(a). For a given capacity and pressure, this operates at a lower speed and occupies smaller space than fans of the other blade type. Generally, the air velocity and turbulence are higher in this type of fan and the noise from this source is greater.

2. Backward -curved blade fans discharge the air backward so that its absolute velocity is less than tip speed. The blade shape and velocity diagram are shown in Fig. 3.3(b). For a given rate of airflow and pressure, the speed is nearly twice that of a forward curved fan. This is quieter than a forward curved fan.

3. Radial-blade fans discharge the air into the scroll with a velocity substantially equal to the tip speed. The blade shape and velocity diagrams are shown in the Fig.3.3(c). Generally the tip of the blade is radial, the leading or inner edge being curved well forward to meet the air with minimum shock loss. The noise from a well-designed fan of this type lies somewhere between that of the forward and backward curved types.

(b) Axial Flow Fans :-

Axial-flow fans are divided into two classes, one type with inlet or discharge vanes called vane-axial fans and one without such vanes called tube axial fans. All axial flow fans impart some spin to the air which passes through them, the amount of spin depending largely upon the pressure for which they are designed. These fans generally operate at higher speeds than centrifugal fans. In spite of this, when carefully designed, their noise characteristics are comparable to the best of other types.

3.3.2 AERODYNAMIC SOURCES OF FAN NOISE :-

In order to obtain an understanding of fan noise it is helpful to analyse the sound generated into its principal components; these are in general, all present, although in varying proportions.

1. Lift Noise :-

This is associated with the circulation around each blade of the impeller and therefore, with the lift and with the raise in total pressure across the impeller. The theory is due to Gutin (12) who related the forces applied to the air by the blades of an aircraft propeller to a system of acoustic dipoles. His analysis is in terms of the components of the force, as 'torque noise' and 'thrust noise', but a combined term such as 'lift noise' seems more appropriate in fan work. Lift noise is an inevitable accompaniment of the work done on the air by the impeller, and is the principal source of that part of the noise which has a recognisable pitch.

2. Displacement Noise :-

The origin of this component lies in the successive passage of fan blades having finite thicknesses. As the air passes through the impeller it is forced aside to give space for the thickness of the blade to pass, reuniting on the discharge side. Acoustically, displacement noise can be represented by a system of simple sources, as described by Deming (13) .

3. Rotational Noise :-

Rotational noise, commonly called blade noise, is basic to all types of fans. Every time a blade passes a given point, the air at that point receives an impulse. The repetition rate of this impulse - the blade passing frequency - determines the fundamental tone of this type of noise.

The intensity of the blade noise and the relative strength of the various harmonics are determined by the shape of the air impulse (14). Doubling the number of blades of a fan doubles the frequency of the fundamental. However, its effect is somewhat more important than that . Consider an element of area in the disk of a propeller fan. Each time a blade passes this area, the air will receive an impulse as indicated in fig. 3.6(a). This impulse may be resolved by Fourier analysis into a steady component which gives rise to the desired air flow, and a series of oscillating components whose frequencies are integral multiples of the impulse frequency. These oscillating components are the sources of blade noise and consist not only of the fundamental frequency but a large number of harmonics. If another blade is added to the one-blade fan, all of the odd harmonics of the one-blade fan will be cancelled, and the strength of the even harmonics will be doubled.

However the capacity of the fan has also been increased. For the same capacity and rpm a somewhat smaller size fan may be substituted. The net result is that all of the odd harmonics of the single blade are eliminated and the amplitude and frequency of the even harmonics are essentially unchanged, as indicated in Fig. 3.5.( ). In general, for symmetrically spaced blades, the fundamental frequency is determined by the product of the number of blades and rpm. Increasing the number of blades will reduce the number of audible harmonics, but, assuming constant capacity and rpm, the strength of the remaining tones will be essentially unchanged. Hence this will decrease the total rotational-noise - the sum of the sound powers being radiated at each of the discrete frequencies. Doubling the number of blades decreases the rotational noise about 3 db.

The above observations also apply to centrifugal fans having few blades. However, the number of blades of a centrifugal fan is governed by optimum airflow design, and the noise generation decreases, but slightly, for more than the optimum number of blades.

A shroud around a propeller fan may serve to reduce noise considerably if it is working properly. However, if the flow breaks down over part of the shroud, the noise may become considerably worse than for an unshrouded case.

In case of centrifugal fans, if the scroll clearance increases rapidly, the noise at the blade passing frequency increases.

4. Vortex Noise :-

(a) Generation by Blades :-

The blades are a major source of vortex noise. When a blade moves through the air a pressure gradient is built up across the blade in the direction of its thickness. If the air flow next to the blade is steady, or laminar, the pressure gradient is essentially constant and very little noise results. However, with an incorrectly designed blade profile, the flow may separate from the convex side of the blade, thus giving rise to rather large eddies. Moreover, this point of separation is variable. Hence, the pressure pattern and eddy formation fluctuate rapidly and cause considerable noise. Also Von Karman Vortices will be shed from the trailing edge of the blade since this edge must have a finite thickness. These vortices are shed alternately from opposite sides of the blade. Since they are random in size and point of release from the blade, a broad-band noise spectrum results. For axial fans the noise due to such vortices increases with the thickness of the trailing edge. For centrifugal fans, this is true only if the air completely fills the space between the blades. Also for axial fans, vortex noise increases sharply when the air flow is such that the eddies peeling from one blade are struck by the following blades. This effect may be minimized by the use of curved trailing edges so designed that most of the vortices are shed at the tips of the blades.

In centrifugal fans not having airfoil-shaped blades, vortices can be created at the inlet of the blade.

(b) Air-Stream Turbulence :-

Noise may also be generated in an air stream itself, for example in a mixing zone where the air stream enters relatively still air. Since velocity gradients must exist in the mixing region, vortices will arise. The strength, size, and motion of these vortices are a function of the velocity field in the mixing zone. In general, the vortices grow, decay, and move in a random fashion, thus giving rise to a broad-band noise spectrum. An obvious way of reducing this noise is to avoid high relative velocities, when possible, in such mixing zones.

Perhaps a more controllable source of turbulent noise is that due to obstructions in the air stream, either at inlet or outlet. The presence of any sharp edges or bends will cause increased turbulence and noise. Also, the placement of heat exchangers too near either the fan intake or exhaust will result in increased noise.

3. Interference Noise :-

Powerful tones of definite pitch may be generated in fans when the wakes from the impeller blades impinge on an obstruction-such as a badly located or shaped cut-off in a centrifugal fan or when the wake behind an obstruction is cut by successive impeller blades as in an axial flow or propeller fan with struts or motor supporting brackets which are badly shaped and too close to the impeller. Interference noise has the same frequency characteristics as rotation noise but does not show such marked directional effects unless the stationary obstructions are regularly spaced round the axis. In axial flow fans direct radiation from the localised pressure pulsations may be supplemented by pulses of intense vortex noise due to the momentary breakdown of air flow over the affected blades.

Particular cases of interference noise arise in axial-flow fans with upstream or downstream guide vanes. The steady force on each impeller vane is modulated at a frequency corresponding to the number of guide vanes, and the resultant interference and rotation noise contains fundamental and harmonic frequencies equal to the revolutions per second multiplied by the number of impeller blades, the number of guide vanes, their sum and their difference. If contra-rotating impellers were placed close enough together to cause interference noise, fundamental and harmonic frequencies would be present corresponding to each blade number with sum and difference terms. In addition interference patterns revolving round the axis would cause low frequency hunting effects if the impeller speeds were slightly unequal.

3.3.3 NON-AERODYNAMIC SOURCES OF FAN NOISE :-

There are several sources of noise that are not strictly due to the aerodynamics of the fan itself. As far as possible the fan builder incorporates the best over-all design to keep these noises to a minimum. Some of these may be in the driving equipment which may or may not be furnished with the fan proper.

1. Fan Unbalance :-

This may be a serious source of noise. Unbalance results in a complete vibration once per revolution. If the fan is rigidly mounted on supports that are good sound radiators, low frequency tones may be radiated efficiently and be difficult to suppress.

For quietest operation, vibration isolation mounts should be used (See Chapter VI.) as well as flexible connections to the duct work.

2. Bearing Noise :-

Good bearings are not generally a source of objectionable noise. Well lubricated sleeve bearings are somewhat quieter than ball or roller bearings. In larger units where the fan noise is higher, antifriction bearings are quite satisfactory.

3. Structural Resonance :-

A wide range of frequencies is present in most fan noises. If the energy in a given band is high and corresponds to the natural frequency of some part of the fan (generally flat panels) the resulting noise may be radiated efficiently. Added bracings can be used to raise the natural frequency of the part or damping material may be applied to it in order to reduce the noise radiation.

4. Motor Noise :-

Noise of magnetic origin may be radiated by the fan if the impeller is mounted directly on the motor shaft. In case of motors with built in fans for cooling, aerodynamic noise from the fan may be appreciable. Some consideration should be given to the design of motor cooling fan.

5. Couplings and Bolts :-

Generally these do not contribute much to fan noise. For couplings better alignments may reduce the squeaks. In case of chain and straight-spur-gear drives the noise may be quite large and should be avoided if possible. Pulleys for drive should run true and well balanced.

These noises are of small magnitude as compared to the aerodynamic noise. These are best reduced to a minimum by the proper choice of equipment, good design and mounting.

3.3.4 EFFECT OF FAN PARAMETERS ON NOISE VS PERFORMANCE :-

Several significant attempts to measure fan noise and relate it to fan operating characteristics have been reported in the literature.

In an attempt to derive an empirical relationship for the overall power level for fans in terms of the horse power and the number of blades of the fan, Poldtrop and Hosler (16) measured the noise characteristics of 5 fans, 3 vane axial and 2 centrifugal, one of which had forward curved blades and the other backward curved blades. The test was conducted at several speeds. They arrived at an empirical formula for the overall power level of the fan output.

$$PWL = 120.4 + 17.7 \log_{10} \left(\frac{HP}{N}\right) + 15 \log_{10} \frac{N}{6} \text{ db}$$

$$\text{re } 10^{-13} \text{ watts.} \quad \ldots\ldots\ldots(3.1)$$

where HP = Horsepower delivered to the fan rotor by the motor driving it,

N = Number of fan blades.

Beranek and his associates (16) measured the values of sound power level for centrifugal and vane-axial fans. The fan noise was reported as overall power level radiated from the fan instead of simply the sound pressure level existing at some distance. The noise was also analyzed on the octave band basis.

This contribution was quite significant in several ways :-

1. Since the acoustic power level of a noise source is independent of the distance of the observer from the source and substantially independent of the method of measurement, the introduction of acoustic power level to describe fan noise makes possible an agreement of measurements even when made with physically different arrangements from one laboratory to another.

2. The use of power level for the designation of fan noise also allows the rapid and simple calculation of sound pressure level at a listener's position whether he be near fan or in a remote location connected only by airconditioning ducts. (The method of calculating this is described in 5.11)

3. With an octave band analysis of noise spectra, an insight is obtained into the spectral quality of fan noise which enables the economical design of acoustical treatment of noise control.

The equation 3.1 is no longer used in its original form. It has undergone several unpublished variations which introduce additional characteristics of the fan such as wheel diameter, blade tip speed, etc., in an attempt to fit the measured data more closely. The equation became unwieldy from a practical stand point and frequently could not be used in entirety because all of the information required was not readily available to the engineer.

Centrifugal fans from 1.0 to 40 hp were tested by Beranek, Kamperman and Allen (17) to derive a general equation for the noise generation. The horse power was chosen as the main operating characteristic since this is readily available as the nameplate hp of the motor driving the fan. The overall power level was measured and plotted against hp. The overall PWL followed the law,

$$PWL = 100 + 10 \log_{10}(HP) \text{ db re } 10^{-13} \text{ watts.} \qquad (3.8)$$

The spectrum shape was also plotted. The spectrum slopes off at -5 db per octave and the range of levels in each band is about $\pm$ 4 db.

But these curves donot indicate that all fans of the same power are equally noisy, such is not the case. There is a possible spread of $\pm$ 4 db in the overall and a further possible spread of $\pm$ 4 db in the spectrum level of individual octave bands relative to the measured values.

The use of motor hp for calculating fan noise levels was criticised, since the motor is sometimes over-rated in order to avoid possible overloading. This is especially true in the case of fans with forward curved blades.

The form of the equation was also criticised, since it did not include a term accounting for operating pressure as required by the sound law for Fans formulated by R.D.Madison (10).

With the increased importance of high pressure systems, involving pressures of 10 in. of water or more, the question of pressure term became significant. The required equation (10) below applies when the operation pressure differs significantly from 1 inch of water.

$$PWL = 100 + 10 \log_{10} (HP) + 10 \log_{10} P \text{ in db re } 10^{-13} \text{ watts} \quad \ldots\ldots\ldots\ldots(3.3)$$

Here P is the measured pressure head across the fan in inches of water. Since this equation is intended for the use only when the fan is operating near peak efficiency, either static or total pressure may be used, the difference will be only a fraction of a decibel. The power level in each octave band relative to the overall power level is shown in fig. 3.6 for ventilating fans of two types. The shaded areas in both graphs give the expected variation due to various shapes of the fan blades and other details. The overall power level is calculated using Eq. 3.2.

The tests conducted by Daly (20) show that the sound spectra for axial flow fans are similar to that shown in Fig.3.7. The spectra differs much from the spectra shown in Fig. 3.6. The variation is quite large in the lower frequencies. This is due to the fact that the tests of Daly were free field tests. Hence, the end reflection loss in the duct is not included . If this is taken into consideration then the spectra shown by dotted line is obtained and now the ~~two~~ two spectra are almost identical.

The overall sound level measurements have produced one very valuable result. When attention is confined to one fan or to a series of geometrically similar fans, some very simple noise relations can be observed. These are called the sound laws for fans similar to the other fan laws.

3.3.5 SOUND LAWS FOR FANS (10,21) :-

The sound power level of a series of homologous fans was found to be a function of fan size, speed, capacity and pressure as shown in the columns labeled sound laws for fans Table 3.1. The corresponding fan laws are also given to enable the engineer to evaluate the other variables. These laws are intended to be applied to the overall sound power level, but may , with limitations be applied to octave band analysis. These limitations are concerned primarily with the blade frequency. Octave band analysis made under the conditions of these tests show that the peak sound power is in the octave band in which the blade frequency occurs. If the fan speed is changed to such an extent that the blade frequency is shifted to a different octave band., it is necessary to move the peak sound power to this octave in the frequency spectrum.

These laws have been found to hold for centrifugal and axial fans over operating ranges normally encountered in commercial and industrial use. Propeller fans have agreed with the aforementioned law on an overall sound power level basis, but have shown some deviation on an octave band power level basis.

## TABLE - 3.1.

Fan Laws and Sound Laws for Fans.

| Law No. | Dependent Variable | | Basic data | Independent variables | Density correction | Sound Laws |
|---|---|---|---|---|---|---|
| 1 | $Q_a$ | = | $Q_b$ | $\times (D_a/D_b)^3 \times (N_a/N_b)$ | $\times 1$ | |
| | $P_a$ | = | $P_b$ | $\times (D_a/D_b)^2 \times (N_a/N_b)^2$ | $\times (\rho_a/\rho_b)$ | $PWL_a = PWL_b +$ |
| | $HP_a$ | = | $HP_b$ | $\times (D_a/D_b)^5 \times (N_a/N_b)^3$ | $\times (\rho_a/\rho_b)$ | $70 \log(D_a/D_b) + 50 \log(N_a/N_b)$ |
| 2. | $N_a$ | = | $N_b$ | $\times (D_b/D_a)^3 \times (Q_a/Q_b)$ | $\times 1$ | |
| | $P_a$ | = | $P_b$ | $\times (D_b/D_a)^4 \times (Q_a/Q_b)^2$ | $\times (\rho_a/\rho_b)$ | $PWL_a = PWL_b - 80 \log$ |
| | $HP_a$ | = | $HP_b$ | $\times (D_b/D_a)^4 \times (Q_a/Q_b)^3$ | $\times (\rho_a/\rho_b)$ | $(D_a/D_b) + 50 \log (Q_a/Q_b)$ |
| 3. | $D_a$ | = | $D_b$ | $\times (HP_b/HP_a)$[illegible] $\times (Q_a/Q_b)$[illegible] | $\times (\rho_a/\rho_b)$[illegible] | $PWL_a = PWL_b +$ |
| | $N_a$ | = | $N_b$ | $\times (HP_a/HP_b)$[illegible] $\times (Q_b/Q_a)$[illegible] | $(\rho_b/\rho_a)$[illegible] | $20 \log (HP_a/HP_b)$ |
| | $P_a$ | = | $P_b$ | $\times (HP_a/HP_b) \times (Q_b/Q_a)$ | $\times 1$ | $- 10 \log(Q_a/Q_b)$ |
| 4. | $D_a$ | = | $D_b$ | $\times (HP_a/HP_b)$[illegible] $\times (P_b/P_a)$[illegible] | $\times (\rho_a/\rho_b)$[illegible] | $PWL_a = PWL_b +$ |
| | $N_a$ | = | $N_b$ | $\times (HP_b/HP_a)$[illegible] $\times (P_a/P_b)$[illegible] | $\times (\rho_b/\rho_a)$[illegible] | $10 \log(HP_a/HP_b)$ |
| | $Q_a$ | = | $Q_b$ | $\times (HP_a/HP_b) \times (P_b/P_a)$ | $\times 1$ | $+ 10 \log(P_a/P_b)$ |
| 5. | $D_a$ | = | $D_b$ | $\times (Q_a/Q_b)$[illegible] $\times (P_b/P_a)$[illegible] | $\times (\rho_a/\rho_b)$[illegible] | $PWL_a = PWL_b +$ |
| | $N_a$ | = | $N_b$ | $\times (Q_b/Q_a)$[illegible] $\times (P_a/P_b)$[illegible] | $\times (\rho_b/\rho_a)$[illegible] | $20 \log(P_a/P_b) +$ |
| | $HP_a$ | = | $HP_b$ | $\times (Q_a/Q_b) \times (P_a/P_b)$ | $\times 1$ | $10 \log(Q_a/Q_b)$ |

HP = Horse Power , D = Diameter or any significant dimension.

Q = Discharge ,

N = Revolutions/minute. $\rho$ = Density.

The sound law for fans can be used to make calculations for equivalent points of rating form only and, since the fan sound power level is not constant for all points of rating on the performance curve, it is necessary to determine sound power level at the point of rating at which the fan is to be used.

A double width fan is essentially two fans of the same size, speed and sound power level; therefore, its sound power level will be 10 $\log_{10}$ (2) or 3 db greater than the smaller one. Likewise, the sound power of a multi-stage fan is less than that of a single stage fan of the same capacity and pressure. Thus, if a single stage fan is 90 db, the sound power level of a six - stage fan to give the same air and pressure is 10 $\log_{10}$ (6) + 20 $\log_{10}$ (1/6) = 7.8 - 15.6 = -7.8 or 7.8 db less (Law 5).

3.3.6 FAN NOISE CHARACTERISTICS :-

When fans have a similar geometric shape they belong to a similar, or what is called a homologous series. Curves of the performance characteristics, pressure, efficiency, etc., of such a series have similar shapes when plotted against fan capacity. These curves can be combined by the use of dimensionless coordinates such as pressure coefficient, $\Psi$, and flow coefficient, $\Phi$, defined by the equations 3.4 and 3.5.

$$\Psi = \frac{\text{pressure head}}{\text{wheel peripheral velocity}} = \frac{P}{\left(\frac{\pi DN}{4005}\right)^2} \qquad (3.4)$$

$$\Phi = \frac{\text{outlet velocity}}{\text{wheel peripheral velocity}} = \frac{V}{\pi DN} \qquad (3.5)$$

where P = the pressure head across the fan in inches of water.

D is the fan wheel outer diameter in ft,

N is the fan wheel speed in rpm,

V is the outlet discharge velocity $= Q/A$

Q is the volume discharge in CFM

A is the outlet area of the fan opening in $ft^2$.

When the experimental values of $\psi$, is plotted against $\phi$, we get a single curve as in Fig.3.8(19). A set of curves such as these applies not only to one fan at various speeds but to all fans of a homologous series within close limits.

Each point of operation determined by a value on the abscissa of Fig.3.8 is a point of rating, it is a point of operation at which the capacity of a fan is a given fraction of the free discharge capacity. At a given point of rating the pressure developed will be a fixed fraction of the blocked pressure, and the efficiency of operation will be a constant for all fans of a homologous series and for all speeds within the normal operating range of each fan. In fact, at a given point of rating ... all the operating characteristics of a homologous series will be proportional according to the fan law.

Rewriting the sound law 5, we have

$$PWL = PWL_0 + 10\log_{10} (QP^2) \qquad \text{------------}(3.6)$$

where PWL = total overall sound power at same point of rating,

$PWL_0$ = constant (db) at the given point of rating,

Q = air volume discharge, CFM

P = Static pressure, in. of water.

The quantity $PWL_s$ is called the specific sound power level. It is defined as the sound power level produced by a (hypothetical) member of the series which when operating at a given point of rating has a capacity of 1 CFM at a static pressure of 1 in., of water. This is similar to the term specific speed (18) of a fan which characterises fans and is a fixed quantity for a given point of rating of a like series of fans regardless of the size and speed. The overall sound power level of any member of the series of fans at any speed or point of rating can be represented by the Eq. 3.6, if the capacity, operating pressure, and specific sound power level at the point of operation are known. The specific sound power level calculated from the measured overall PWL, is given in Fig. 3.6 plotted in dbs against flow coefficient (10 ). From this curve it is seen that the lowest value of specific sound power level occurs at the knee in the pressure characteristic, i.e. near the point of maximum fan efficiency. Fortunately, it happens to be generally true that the minimum amount of noise is generated near the point of maximum efficiency.

This point is better understood if we plot specific sound power level and the static efficiency against the specific speed. Specific speed is defined as the speed at which a (hypothetical) fan discharges 1 CFM against a pressure of 1 in. of water. Mathematically,

$$N_s = \frac{N\,Q^{\frac{1}{2}}}{H^{3/4}} \qquad (3.7)$$

where N = fan rotor, rpm,

Q = Air flow, CFM

H = Static pressure, in., of water.

The static efficiency curve will be an inverted V and the specific sound power level curve will be V shaped. Again, at the maximum static efficiency, the specific sound power is a minimum.

With such plots as these, we can readily calculate the overall power level at a given point of rating and thence the octave band spectra from Fig. 3.6.

3.4 NOISE FROM SMALL CENTRIFUGAL FANS :-

An interesting approach to the fan noise problem has been suggested by Goldman and Maling (22). Study on double inlet centrifugal fans having a single forward curved blade design and mounted in conventional scroll housings, was carried out to measure the noise power radiated from fan discharge as a function of octave-band, fan diameter, fan width, speed, head and flow for fans upto 6" diameter and 7" wide.

It was found that the overall power level had a linear relationship with the static pressure at shut off, the parameter being the aspect ratio. At shut off, all the fans which had the same aspect ratio radiated the same noise at the same static pressure.

The overall power level was plotted with the diameter of the wheel and the width of fan as variables at free discharge. The variation was linear. Since the plots were logarithmic and the fact that the data falls on straight lines implies that the noise power is proportional to some power of the diameter or width, the magnitude of the power being determined by the slope.

These measurements indicate that fan noise is separable into two parts, one associated with the head developed (from the sound power vs shut off pressure) and the other associated with the flow (from the sound power vs diameter or width at free discharge). The former is called the head noise and the latter is called the flow noise. The following relations were derived by the slope of the curve,

$$(PWL)_{Head} = \frac{C_1 H^3}{\alpha} \qquad \text{———} \quad (3.8)$$

where $C_1$ is the constant of proportionality

$= 0.0 \times 10^{-4}$ for these particular fans in reference 22.

$H$ = Static head in in. of water.

$\alpha$ = aspect ratio (width /diameter).

$$(PWL)_{flow} = \frac{C_2 Q^5}{\alpha^4 D^{12}} \qquad \text{———} \quad (3.9)$$

where $C_2$ is another constant of proportionality for these fans (22) $= 5.2 \times 10^{-10}$

$Q$ = discharge in CFM.

$D$ = Diameter of fan, inches.

At a condition between shut off and free discharge in the practical case,

$$(PWL)_{total} = (PWL)_{head} + (PWL)_{flow} \qquad \text{———} \quad (3.10)$$

The calculated values agree within +2 to -3 db with the measured values and at the point of maximum efficiency ± 1.5 db.

From fan laws we have :

Head varies $\propto$ (Speed)$^2$

Discharge varies $\propto$ speed.

Therefore we have,

$$\text{Head noise} \propto (\text{speed})^6$$

and

$$\text{flow noise} \propto (\text{Speed})^5.$$

The dependence of flow noise on the fifth power of the air flow can be correlated with the fact that vortex noise depends on the 5th and 6th power of the relative velocity between an air stream and an obstacle of which vortices are formed. Near shut off head noise dominates and near free discharge flow noise dominates.

## 3.5 SELECTION OF LOW NOISE FANS :-

Using such a plot as Fig. 3.9, it is possible to choose, from a homologous series, the fan size and speed for minimum noise output. Thus, from the minimum in the specific sound curve, the quietest point of rating is determined, this determines a value of $\psi$ and $\phi$ which can be used in equations and together with the required value of pressure and capacity to determine the optimum values of size and speed.

From the 'Noise from small centrifugal fans' we observe that flow noise decreases with in increasing diameter and aspect ratio (Eq. 3.9). The aspect ratio is limited to some maximum value for aerodynamic reasons. For a given head, the head noise is determined only by the aspect ratio and is smallest for this given aspect ratio (Eq. 3.8). Reductionsin flow noise obtained by increasing the fan diameter, are effective in reducing the total noise only if the flow noise is appreciable compared to the head noise.

The diameter of fan for which the head noise just dominates the total noise is, then, the optimum diameter. It turns out that the optimum fan operates at the maximum efficiency point on the flow curve near the knee.

Hence,

1. a large aspect ratio should be specified.
2. The optimum diameter should be found either through the use of noise power equations or by finding that fan of large aspect ratio for which the specified head and flow occur near the point of maximum efficiency.

It is apparent that the least noise which is possible in a particular ventilating system is determined only by the static pressure, since the noise may be made indefinitely small by increasing the fan size. The desirability of keeping the static head to a minimum is now apparent from noise considerations.

The minimum noise is also dependent on the type of fan. The optimum combination of aerodynamic and acoustic design for a fan leading to the quietest sounding output noise appears to be one having the best combination of numerous shallow, forward-curved blades and the lowest tip speed as is aerodynamically and practically possible. The backward curved blade fan emits less noise, but delivers less air unless operated at a higher tip speed than the forward curved blade. The composite-curvature (reverse curve) blade fan merits study as being a possible candidate for the least noisy blade profile.

3.6 COOLING TOWER NOISE :-

Cooling towers, although generally located outdoors, generate noise which may be troublesome because it may enter the building through windows or may create a community or neighbourhood noise problem. Results of measurements (23) on many cooling towers ranging from small fractional horse power sizes to over 50 hp units are summarised in equation which differs only slightly from equation.

$$\text{Over-all PWL} = 105 + 10 \log \text{ hp} \quad \text{db re } 10^{-12} \text{ watts} \qquad \ldots\ldots\ldots\ldots\ (3.11)$$

The output noise spectra are similar to those for centrifugal fans except that they fall with a slope of approximately 3 db instead of 5 db /octave.

3.7 COMPRESSOR NOISE :-

The compressor used in the refrigeration circuit is not a major noise source, Since the conditioned air does not come in direct contact with the compressor. But this may constitutes a major noise source in the equipment room and the vibrations may be transferred to air through the ducts.

The motor compressor assembly is usually resiliently mounted on the foundation. Hence a minimum of the forces of unbalance are transmitted to the outside. The compressors are generally equiped with exhaust and intake mufflers - which can be assumed to be satisfactory. However, there still remains a major source of noise i.e. the compressor valves themselves. Valve noise is mechanical in nature and can be minimised by proper design of the valves and valve springs. The valve noise possesses strong harmonics - and one or more of the harmonics almost invariably has a frequency near that of one of the resonances of the cavity within the compressor housing. Hence, standing waves are set up inside the casing and are transmitted through the shell.

Cavity resonance can be controlled by introducing sound absorption into the cavity. These materials introduced as filter in the liquid or gas lines, serve to reduce cavity resonance (14).

3.8 NOISE GENERATION IN ELBOWS :-

As the high velocity air passes through these components (and for that matter over any projection into "the air stream"), each in turn serves as an individual noise generator, due primarily to air turbulence and vortex shedding. As a result, a condition in duct system may be reached where the noise generation level of the components begins to exceed the attenuated sound level from the fan. With high velocity system becoming more prevalent, however, noise generation of the components must be known to predict accurately the expected levels in the ducts, and ultimately in the occupied space.

The noise generation levels, for each octave band has been studied as a function of velocity for vaned elbows(25). In general, the spectrum appears to be broad-band in nature (except for some pure tones) and fall of intensity in the higher frequencies. Similar to the grille noise, the noise generation in the higher frequencies increases more rapidly as the velocity increases.

The noise generation level showed a marked increase in the lower frequencies with an increase in elbow size for the same velocity. This is shown in Fig. 3.0.

For vaned elbows of the same size but of different angle, the noise generation level in the low and mid-frequencies range is about the same for a given air velocity.

The effect of aspect ratio on noise generated is shown in Fig. 3.10. The increase in the generated level in the lower frequency as the aspect ratio increases appears to be influenced more by the dimensions of the elbow and connecting duct than by the turning vanes themselves. Again a large volume bounded by the duct walls for the 24 x 6 (4:1) section would promote the excitation of more lower frequency modes. The relatively constant level in the mid frequency range with change in the aspect ratio is evident. There is slight rise in the generated level in the higher frequency bands with increase in aspect ratio. The general levels remain fairly constant over the low and mid-frequency ranges.

The noise generation in elbows is significant only in high velocity air conditioning system. In the ordinary systems the velocities in the ducts seldom go beyond about 1500 fpm. Hence the noise generated in elbows is very low compared to fan noise and hence neglected.

3.0 GRILLE NOISE :-

Grilles are the openings through which the conditioned air enters a space. A noise source occurs at the grille due to obstructions caused to the flow of air by the fins in the grille. This is an important source because it is located beyond the point where treatment of duct can have any effect.

3.9. (1) Velocity Dependence :-

The noise generated at outlet grilles is strongly velocity dependent. Fig. 3.11 shows the sound levels near the grille face for three types of grilles vs. grille face velocity. The wide angle grille C which gives a large spread (vertical vanes) generates the most noise. Grille B (a honeycomb type), with a smaller spread is less noisy. And grille A(perforated metal) with little spread, is still quieter. These curves were taken 6 ft., from grilles having a face area of 0.5 sq. ft.(28) The sound levels were determined using the A weighting net work. The grille noise per s.ft. of grille core area for each octave band has a linear variation with the logarithm of the grille face velocity (34). Expressed mathematically,

$$SPL_n = a \log 10 \ V_f + b \qquad \text{———} \quad (3.12)$$

where a and b are constants dependent upon the type of grille and octave band of interest.

$$SPL_n = SPL_c + 10\log_{10} \frac{A_o}{A_g} \qquad \text{———} \quad (3.13)$$

where $SPL_c$ = grille noise measured experimentally,

$10 \log_{10} \frac{A_o}{A_g}$ = correction to be applied to obtain grille noise per sq.ft. of core area.

$A_o$ = 1 sq.ft.

$A_g$ = core area of grille, $ft^2$.

The slope of these curves increase in the higher octave bands, indicating thereby that at higher frequencies, grille noise increases at a more rapid rate with increasing velocity. The number of fins per inch in a grille has a considerable effect on the noise spectra. A grille with smaller number of fins per inch have predominance of noise in the lower frequency bands, and with greater number of fins per inch, the predominance of noise shifts to higher frequency bands. In other words grilles with greater static pressure loss, contain greater acoustic energy in the higher frequency bands.

3.9·(2) Area Effects :-

If the face velocity of a grille is held constant, the total sound power radiated increases in proportion to the area of the grille. Each time the grille area is doubled, the sound level in the room increases 3 db. This behaviour is the result of the incoherent nature of the multiplicity of turbulent sources. If the volume of air supplied by a grille is held constant and the area of the grille changed, the sound level changes markedly. Fig. 3.12 shows the sound level versus grille area for three different volume flows.

Multiplicity of Grilles :-

If more than one grille supplies or exhausts the air in a room, the sound level in the room will be proportional to the acoustic power radiated by all the grilles in operation. The calculation of the noise level when the noise level of each of the grilles is known requires special calculation or the chart in Fig. 4.3 can be used.

Hence the best combination for producing a low loudness rating of grille noise, is a smaller number of fins per unit and a low static pressure loss. The velocities are to be kept low, but this is governed by the availability of space.

3.10 NOISE TRANSMISSION THROUGH DUCT WALLS :-

The transmission of noise through duct walls varies greatly with frequency and with duct size, shape and material. The transmission loss (TL) of duct side walls cannot be readily predicted at single frequencies but TL estimates can be made for octave bands of noise. At high frequencies (i.e. those for which the duct panel dimensions are larger than a wave length) the TL of the duct wall may be assumed to approach the field mass law (27) although there may be wide departures from these values. For lower frequencies the TL is controlled by stiffness and is therefore, greatly influenced by any bracing members.

The sound power level of the noise passing through the wall outward to the surrounding space may be estimated from the following approximate equation (27).

$$PWL_T = PWL_P - TL + 10 \log_{10} \frac{S_W}{A} \qquad (3.14)$$

where $PWL_T$ = sound power level of noise transmitted through duct walls in any octave frequency band.

$PWL_P$ = sound power level transmitted along the inside of duct at the beginning of section of duct considered (in the same octave band)

$S_W$ = total area of duct wall for section considered, ft$^2$

$A$ = Cross-sectional area of duct, ft$^2$

In case where $10 \log \frac{S_W}{A}$ approaches or larger than the TL of the duct wall this approximation no longer is valid, since the $PWL_T$ cannot exceed the $PWL_P$ which was originally inside the duct. For such a condition, a reasonable approximation is to assume that $PWL_T$ is 3 db less than $PWL_p$, i.e., half the sound is transmitted through duct walls.

Similar considerations apply for transmission from a room into a duct, but here the sound pressure level inside the duct ($SPL_i$) is calculated from the sound pressure level existing outside the duct wall ($SPL_o$). Then the sound power level of the resulting sound in the duct is calculated from the $SPL_i$ ;(28)

$$SPL_i = SPL_o - TL + 10 \log_{10}\left(\frac{1}{4} + \frac{S_W}{R}\right) \quad \text{-----} \quad (3.15)$$

TL = transmission loss of duct side wall, db

$A$ = duct cross sectional area, $ft^2$

$S_W$ = Duct wall area exposed to noise, $ft^2$

$R = \dfrac{\alpha(S_W + 2A)}{1 + \alpha}$

$= \dfrac{2A + \quad S_W}{2A + S_W}$

= absorption coefficient of duct walls.

= effective absorption coefficient.

R = and quantity analogous to 'Room constant',

The absorption coefficient    for the duct wall is made up of an absorption due to any transmission of sound back into room. The part of    due to lining can be found for the acoustical characteristics of the material. The part of    due to transmission is simply anti $\log_{10}$ (TL/10), It is usually negligible for lined ducts except at low frequencies, but it is not negligible for unlined ducts when $S_U$ is large compared with 2A. Wrapping the outside of a duct with thermal insulation increases the absorption at low frequencies. For large ducts, wrapping may produce an of 0.1 in the lowest two octave bands but produces negligible change at higher frequencies.

The sound power level entering a duct and travelling in each direction along the duct from the exposed area is computed from the $SPL_i$ by adding $10 \log_{10} A$. This section of duct is then treated as a noise source having the calculated power level. The noise-power level in each octave band of this source can be added to other noises at this point on a power level basis when both noises are of a random nature. However, if the transmitted noise is speech or other intelligible noise, it must be handled separately as a cross talk problem on the basis of privacy and speech intelligibility.

When ducts travel parallel to each other in close proximity, as in a duct chase, noise may pass from one duct to the other through the duct walls. This type of noise transfer may be particularly serious where large ducts that serve very quiet areas, such as an auditorium or broadcasting studio, pass next to ducts serving lobbies or other noisy areas.

3.11 DIFFUSER NOISE :-

Aside from the information furnished by diffuser manufacturers, to their users, on sound levels generated by air diffusers, few other data are available. Recently, however, some test results on several types of ceiling air diffusers have yielded a satisfactory correlation of the sound generating characteristics of these devices (29). The data, covering ceiling diffusers ranging in size from 4 to 18 inches neck diameter, show that the overall sound power level (from 75 to 10,000 cps) can be found within ± 2 db by the equation.

$$L_W = 10 \log_{10} A_{min} + 60 \log_{10} V_{max} - 2 \text{ db} \qquad (3.16)$$

where,

$L_W$ = overall sound power level in the seven octave bands from 75 to 10,000 cps, decibels re $10^{-13}$ watts.

$A_{min}$ = minimum flow area for the air passing through the diffuser, square feet. For undamped flow the minimum area will normally occur in the diffusor neck. Where for damped flow it will occur at the damper position (See Fig. 3.15).

$V_{max}$ = maximum air velocity ($Q/60\ A_{min}$), ft per sec.

For ceiling diffusers of cone-type construction used in conjunction with deflecting vanes and a damper, Fig. 3.13 presents the approximate octave band spectrum relative to the overall PWL as given by the above equations. The curves of Fig. 3.15 agree within approximately 3 db of the octave band data collected on 6-12- and 18-in. neck diameter ceiling diffusers.

# CHAPTER - IV.

## NOISE CRITERIA.

### 4.1 INTRODUCTION :-

A criterion is defined as a standard of judging. Noise control criteria provides standards for judging the acceptability of noise levels under various conditions and for various applications of space.

The criteria establishes the acceptable noise levels for the space being quieted. The purpose of quieting may be to provide an environment favourable to those occupying them, in concert halls favourable to the listening of music, in bed rooms for comfortable sleep, and so forth.

4.2. In most noise problems the measurement of noise is essential in providing information which will aid in the determination of proper noise control measures. The specific noise measurements will vary from problem to problem. However, regardless of what measurements are needed (and they can only be determined by consideration of the specific problem), it is important that they can be obtained for the simple reason that adequate noise control measures cannot be sensibly effected unless information about the noise to be controlled is available.

The basic noise or sound measuring instrument and the one most commonly used is the sound level meter. Essentially, it consists of a microphone, a calibrated attenuator, an electronic amplifier, and an indicating meter (Fig. 4.1).

The sound waves striking the microphone are

impulses.

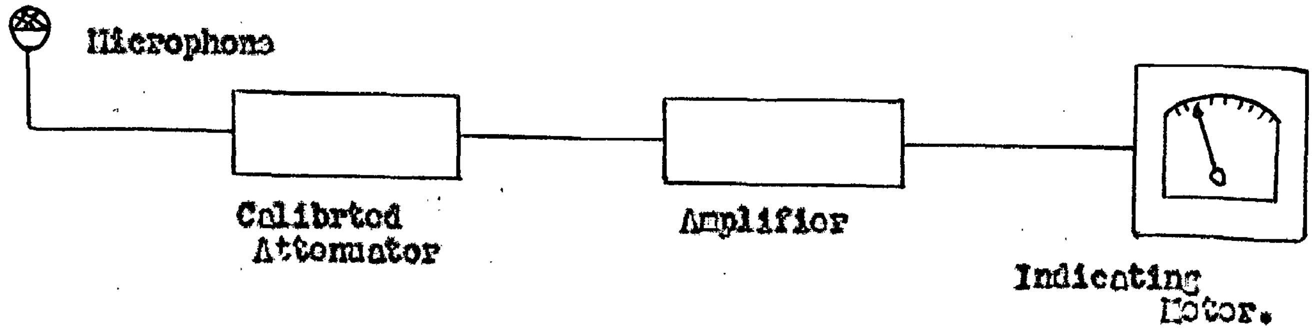



Fig. 4.1. Simplified block diagram of a sound level meter.

These electrical signals are amplified and fed to an indicating meter which is calibrated in decibels. The sound level meter reading is an indication of the total sound pressure level in decibels re 0.0002 dynes per sq. cm., in the frequency range from 20 to 10,000 cps. These instruments include an electronic weighting network which de-emphasises certain frequency portions of noise. There are three weighting or frequency response characteristics provided in the instrument (35). These are obtained by weighting networks designated as A,B, and C. They are also referred to as "40 db", "70-db" and flat respectively. Responses A, B and C selectively discriminate against low and high frequencies in accordance with certain equal-loudness contours of Fig. 4.4. Ordinarily, A net work is used for sound levels below 55 db, B net work between 55 and 85 db, and C network for levels above 85 db. For octave band analysis the C or flat scale is used where the response until frequencies is the same.

Although the sound level meter is an useful instrument, it is limited in that it gives only a single reading covering the frequency range of 20 to 10,000 cps. This reading is called the "overall" reading. This reading gives no clue to the frequency distribution of the noise. Complex sounds and noises normally contain frequency components at several or all frequencies within this range. In addition, the actual distribution of the sound pressure with respect to frequency differs for different noise sources.

To obtain the frequency analysis, one employs an instrument known as a frequency analyser. There are several kinds of analysers available, such as the octave band analyser, the third - octave band analyser, and several types of narrow-band analysers. The octave ban analyser divides the frequency range of 20 to 10,000 cps into eight frequency bands, the third octave band is composed of 24 bands in this range, and a narrow band analyser allows for readings in a comparitively narrow frequency band centered at any specified frequency over the entire frequency range. But a frequency analyser in wide-spread use today is the octave band analyser. It is suitable in the majority of applications where continuous wide-band noises are found. For certain noise problems or where more detailed information is required, it may be necessary to employ one of the analysers mentioned above, or more specialised equipment.

In Fig. 4.2 are plotted two noise spectra as read on an octave band analyser. These spectra are examples of typical noises wich one might encounter in practice. Spectrum A is predominantly low frequency in nature, that is , it has its highest octave band levels in the low frequency bands. In contrast, spectrum B, has its highest band levels in the high frequency bands. These two spectra have been chosen for this example because they represent entirely different kinds of noises. Spectrum A might well represent a "roar" of a diesel truck and spectrum B the "hiss" of a compressed air jet.

What would the sound level meter read for each of these two noises? To determine this it is necessary to calculate the overall levels for spectrum A and B. The overall is obtained by adding logarithmically the levels in each of the octave bands. The adding process can be performed easily by use of Fig. 4.3. For instance, for spectrum A, the levels in the last two bands 57 and 59 db, add to give 61 db. The next two lower bandlevels, 62 and 66 db, total 67.5 db Adding these two totals results in a level of 68.5 db for the upper four octave bands and soon.

Carrying this calculation to completion, we find that the overall for Spectrum A is 84 db. A similar calculation would reveal that the overall for spectrum B is also 84 db. Therefore, although these two noises differ audibly, have octave band spectra in direct contrast to one another, and would probably require very different noise control measures, the sound level meter or overall reading for each would exactly be the same. It should be clear then, at this point, that one can be reasonably certain that a noise control study based on a frequency analysis, of the noise sources involved will result in a much more well balanced solution than one utilising only overall levels.

## 4.3 ACCEPTABLE AND UNACCEPTABLE NOISES :-

Noises seem to fall into two categories, those which, because of their character, are acceptable, and those which are unacceptable. Some examples of the former are : the rustle of wind in trees, the patter of rain on the roof, the rumble of distant traffic, the blur of a large number of voices, such as a concert audience at intermission time. One important example of an "unacceptable" noise is speech intruding into what is ostensibly a private office, home or dormitory room. Other examples are the roar of the neighbour's power lawnmower, the whine of the neighbours air conditioning compressor, and the sound of the neighbour's exhaust fan.

Often, diffuser noise and similar mechanical equipment sounds are classified by the listener as "acceptable". In such cases the diffuser noise is usually blended with distant traffic noise or office activity noise. Other times, the diffuser noise is definitely unacceptable.

When noises in a space are of an "acceptable" character, the listener is usually not aware of their existence. Annoyance from such noises is usually small unless their level is great enough to interfere with conversation or in some way force themselves upon the listener's attention. When an unacceptable noise intrudes a space having a certain level of acceptable noise, the resulting annoyance is often determined not by the level of the unacceptable noise, but rather by the magnitude of the difference between the acceptable and unacceptable noise levels.

The acceptable amount of air conditioning noise varies greatly depending upon the uses of the space and the people occupying it. Music: A very low noise level can be tolerated where music or speech is to be reproduced, to avoid having the noise obtrude into the dynamic range of music. The requirements are only slightly less exacting where music is to be played live, as in auditoriums, churches, music rooms and home living rooms. Speech Communication:- Speech communication is an important requirement of most room uses. But, the background noise level which can be tolerated varies greatly with different applications. Since people are different, no single method of rating noise will suit all. Age of the listener and his occupational and medical history affect his hearing. Fluctuations in hearing can be influenced by lack of sleep, psychological state, and the degree of momentary alertness. Then too, the quality of noise is an important factor in annoyance. Discordant pitch combinations sound unpleasant nearly to all ears. Furthermore, the masking effect of the background or ambient noise and the method of listening also affect the amount of annoyance felt by the ears.

Obviously, not all these factors can be measured and presented as a single figure annoyance rating. Several people have made investigations and have proposed criteria based upon the evaluation of different properties of sound including loudness level, speech interference and annoyance. Beranek has presented two related critera, which he has called "noise criteria" (10). These criteria combine a speech communication criteria and a loudness level criterion. Now each of them will be discussed in the following pages in detail.

4.4 LOUDNESS OR LOUDNESS LEVEL CRITERIA :-

During the past two decades several methods have appeared in the literature for calculating the loudnessof sounds directly from their sound pressure spectra. In their classical work on loudness, Fletcher and Munson (36) developed a method of calculation suitable for determining the loudness of complex tones from the frequency spectrum. The loudness of a continuous spectrum noise may be calculated by the procedure developed by Fletcher and Munson. Each of these methods is suitable for the particular kind of noise for which it was developed, and each is somewhat cumbersome to use.

In engineering applications a single objective procedure for determining the loudness of any type of noise with reasonable accuracy is of great value. Several years ago an attempt was made in this direction by L.L.Beranek and APG Peterson (37) based on the early works of Fletcher and Munson and a suggestion contained in a paper by Churcher and King (38).

4.4.1 Equivalent tone method :-

The method suggested by Beranek (39) is very cumbersome since it needs special filters for sound analysis. Since the octave band analysis of noise is readily available, the method of calculating the loudness level was modified by Tysser and Mintz (40).

For this a loudness chart is made use of. The loudness chart with contours of equal pressure level in octave bands is shown in Fig. 4.4. The pressure level contours are plotted at 10 db intervals. The vertical grid coordinates are loudness values in sories arranged in a logarithmic scale. The contrours of equal pressure level were derived from the Churcher -King equal loudness contours by finding the loudness level corresponding to the selected values of level of the band at the mean frequency of each band. The corresponding values of loudness level in phons is also shown on the right of the same chart.

The procedure for determining the loudness of sound by use of the loudness chart as follows :-

1. Measure the SPL in db in each octave band from 37.5 - 75 cps upto and including 4800 - 9600 cps.

2. Plot the observed octave band sound levels on a contour graph, shown in Fig. 4.4 using the pressure level contours to determine the vertical coordinates for each octave band.

3. Using the loudness grid, read off the loudness values corresponding to each point plotted.

4. Add the individual values of loudness to obtain the total loudness in sones. If desired, the corresponding level in phons may be obtained on the right hand scale.

The chart in Fig. 4.4. has an additional feature which is of great value to the accoustical designer. Noise analysis plotted on such a chart indicate the distribution of "loudness" on the audio spectrum in an easily understood graphical form. The highest point of the graph is the loudest point, the dominant frequency band is identified at a glance. In contrast, the graph of an octave band noise analysis plotted in the conventional manner- db vs frequency - does not generally have the loudest frequency at the highest point of the graph. The picture of loudness distribution with such a graph is misleading.

Phons and Sones compared :-

The phon loudness gives a true rank order rating to sounds but it does not give a true relative rating. An increase in loudness level of 10 phons might mean a tenfold increase in loudness or it might mean slightly more than doubling of loudness, depending upon whether the sound is in the low level or the high level range. Fig. 4.4. also shows the relation between the loudness and loudness level in phons. The loudness units are named "sones" Sones are simple units to deal with because their value is such that 20 sones sound is twice as loud as 10 sones, and 5 sones sound one half as loud as 10.

LIMITATIONS OF EQUIVALENT TONE METHOD :-

The octave band equivalent tone method is subject to error on certain types of noise which are too large to be neglected. If the noise has a component in the first octave band, its loudness can vary as much as 100 percent if the frequency of this component changes from one side of the band to the other, although the computing procedure shows no change in loudness for this frequency shift.

Another source of error becomes important when the first two or three octavescontribute heavily to the loudness and the remaining octaves are relatively unimportant. Under these conditions the total loudness will be overestimated, on the other hand, the total loudness will be under-estimated if the last three octaves contribute heavily to the loudness of the noise and the other octaves are of lesser importance. In such cases a correction is usually applied or the equivalent noise method is used.

4.4.2 EQUIVALENT NOISE METHOD (41,42) :-

Another method of computing loudness is based upon measurements of the loudness of bands of noise and is due to Stevens.([illegible]). The result of such measurements are plotted in Fig. 4.5 which shows loudness as a function of pressure level of octave bands. The computational procedure is applicable to spectra that are reasonably continuous. The procedure is as follows:-

1. Tabulate the sound pressure levels in the various bands.

2. Find the loudness corresponding to each band from Fig. 4.5.

3. The loudness of the band having the greatest number of sones is left unchanged in value and the loudness values of the remaining bands are multiplied by a correction factor of 0.3.

4. The total loudness of the noise, the sum of the loudness values of the various band multiplied by their correction factor.

Expressed mathematically,

$$S_t = S_m + 0.3\,(\Sigma S - S_m) \qquad \text{———} \qquad (4.1)$$

where $S_t$ = total loudness,

$S_m$ = ~~main~~ loudness of the loudest band.

$\Sigma S - S_m$ = Sum of the loudness of the remaining bands.

This method of calculating loudness by Steven's procedure is universally accepted as the standard. The International Business Machines (IBM) of U.S.A., have published data on the loudness of various categories of machines and equipment and the calculations of loudness are based on the Stevon's method.

The criteria based on the allowable loudness values are given in Table 4.3 for various applications of the air conditioned space (49).

4.5 A - SOUND LEVEL :-

This is an approximate measure of loudness. This measuring method has the advantage of simplicity since the A-sound level is obtained by a single reading on the sound level meter. Since the A-scale weighting de-emphasises the low frequency portions of the noise spectrum, this automatically compensates for the lower sensitivity of human ear to low frequency sounds.

For sounds having a broad frequency Spectrum (such as air flow noise), an estimate of loudness level can be obtained from A-sound level measurements by the approximate equation.

Loudness level in phons = A-Sound level in db + 12 ± 2 ----------(4.2)

Measurements of the A-sound level can also be used to estimate the approximate $N_c$ level (see later), if no octave band analyser is available (8).

$N_c$ level in db = A-sound level in db - 8 ± 2 ---------------- (4.3)

Design criteria based on the A-sound level is given in Table 4.3 for different applications of the space. The use of these values are intended only for judgement of the acceptability of the noise in the space. For the purpose of proper system design and for the analysis and results the use of sound power level or sound pressure level on an octave band basis is imperative.

4.6 SPEECH INTERFERENCE LEVEL (44,45) :-

One of the most serious consequences of noise is that it prevents us from understanding what other people are saying, whether they are talking to us directly or talking over a telephone or public -address system. The understanding of spoken words is defined as intelligibility.

Intelligibility is a psychological factor and psychological techniques are required to measure it, but a speech communication consists of equipment on which many physical measurements can be made. The noise level that can be tolerated varies greatly with different applications, depending upon the distance between the speaker and the listener, the acoustical characteristics of the room and the degree of facility of communication required.

The frequency range which is important to speech intelligibility is approximately from 300 to 5000 cps. When the lower end of the dynamic range of the speech sound drops below the background level, the intelligibility is decreased, and the amount of this decrease depends upon the portion of the dynamic range which is masked by the background noise. Additionally, speech intelligibility is limited by the sustained reverberation of the speaker's own voice from the preceding syllables. The effect on speech intelligibility is the result of the addition of background noise and the [illegible] reverberant speech noise. Calculations of the reverberant speech level have been made for a number of sizes of rooms with the speaker and listener near opposite ends of the room according to the method of Bolt and MacDonald.

For reasons of simplicity, a method has been proposed for estimating maximum tolerable noise levels for satisfactory speech intelligibility in which acoustical measures of the noise are made by means of octave band filters and sound level meter. In this procedure the arithmetic average of the decibel level overall of the noise in each of the octave bands 600 to 1200 cps, 1200 to 2400 cps and 2400 to 4800 cps is found. The resulting number in decibels is a handy guide to the interfering effect of the noise on speech and is called the speech interference level.

With table 4.1 and Fig.4.4 we may arrive at criterion curves giving the maximum permissible noise levels in any frequency band for rooms of various types (46).

For example, in a large assembly hall with a good speech reinforcement system, we see from Table 4.1 that the 25-db criterion curve should be selected. If the noise is concentrated in a portion of the frequency range, the appropriate solid curve of Fig. 4.6 is used. If the noise levels principally lie near the appropriate curve in bands between 300 and 4800 cps, then, because of the masking effects, the noise in the lowest three frequency bands may be higher. For this case, the dashed curve A may be used in the design. In other words, more noise is permissible at low frequencies if there is high frequency masking noise than if there is not.

TABLE - 4.1

Nc Criteria for noise control.

| Type of room | Nc criterion. |
|---|---|
| Concert halls | 20 - 25 |
| Broad cast studios | 15 - 20 |
| Legitimate theatres (500 seats, no amplification) | 25 |
| Music rooms | 25 |
| Assembly halls (amplification) | 25 - 30 |
| School rooms (no amplification) | 25 |
| Homes (sleeping areas) | 25 |
| Conference room for 50 | 25 |
| Conference room for 20 | 30 |
| Movie theatres | 30 |
| Hospitals | 30 |
| Churches | 30 |
| Court rooms | 30 |
| Libraries | 30 |
| Small private office | 40 |
| Restaurants | 45 |
| Coliseums for sports only | 50 |
| Secretarial offices (typing) | 55 |
| Factories | 40 - 65. |

## 4.7 COMBINED SPEECH COMMUNICATION AND LOUDNESS CRITERIA FOR OFFICE SPACES :-

The results of a series of noise ratings made by office workers (47,48) are plotted against SIL and LL in Figs. 4.7 and 4.8, Fig. 4.7 for executive offices and small conference rooms, Fig. 4.8 for stenographic and large engineering drafting rooms where many people are conversing simultaneously. The subjective rating did not correlate as well with the overall sound pressure level as did either the SIL or LL. The correlation of the subjective ratings was generally better with SIL. The left hand ordinate of Fig. 4.7 shows the rating scale used on the questionnaires given to the office workers. On the average the personnel in the executive offices thought that the noise should not exceed a rating just below "moderately noisy" if their work was not to suffer. More than half of those questioned said that if the room were more than moderately noisy, their ability to telephone and converse would be affected. Nearly all of them said that they used the telephone "often" to "very often" and that they conversed "often" with co-workers as far away from them as 8 to 10 ft. It follows, then, that the SIL in executive offices should not exceed about 40 db if conversations are to be considered as in a more or less normal tone of voice over distances upto 8 to 10 ft.

In some offices the subjective ratings correlated better with the computed LL than with measured SIL. Examination of the octave band spectra measured in these offices revealed that the low frequency noise levels were higher relative to the levels in the speech range than they were in offices in which the subjective ratings correlated well with both the SIL and the LL.

Where the correlation was good with both LL and SIL, the LL in phons was not more than 22 units higher than the SIL in db. Whenever the difference betwen LL and SIL became substantially greater than 22 units, say 30 units, the subjective rating was higher than would be predicted by the SIL alone. In addition, when the LL minus the SIL difference exceeded 30 units, there were usually complaints, even when the SIL met the requirements of being lower than 40 db.

For the reasons just given the acceptable LL as plotted in Fig. 4.7 is about 22 units to the right of SIL. If the difference between the two curves exceeded about 22 units, the noise level is likely to be found less acceptable than would be predicted by the SIL alone. These studies, therefore, show that neither the SIL nor the LL alone is sufficient to characterize the criterion spectrum; both are needed.

The curves in Figs. 4.8 are like those of Fig. 4.7 except that the ratings of the noise were made in stenographic pools or large drafting rooms. In these areas, the average noise usually has SIL of 50 to 55 db. As we can see, the personnel say that if the SIL were to exceed about 55 db, (or the LL exceed about 77 phons) their work would suffer. An SIL of 55 db permits them to converse at distances of 3 to 4 ft., in a more or less normal tone of voice. The degree to which noise interferes with telephoning bears the same relation to SIL in both figures.

4.7.1 OCTAVE BAND CRITERION LEVELS :-

Figs. 4.7 and 4.8 present SIL-LL criteria for the two types of office space. Because there is not in existence today either an LL meter or an SIL meter that can be read directly, the information in Fig. 4.7 and 4.8 must be converted into octave band levels, for which there is a measuring apparatus. The results are given in Fig. 4.9 and 4.10. In Fig. 4.7 the computed LL for each curve is 22 units higher than SIL. The Nc (noise criteria) number shown beside each curve equals the SIL. The assumption made in the conversion was that the curves slope off monotonically. This automatically means that the highest frequency band does not carry a large portion of the loudness.

The curves of Fig. 4.9 and Fig. 4.10 are called noise-criterion curves. In Fig. 4.10 an alternate family of curves is presented where LL is greater than the SIL by 30 units. The 'A' in NCA means "alternate". The NCA curves are recommended only where a max. compromise due to economic factor is necessary.

In a particular room the eight octave band levels should not exceed the values indicated by the NC or NCA curves selected in the specification for that office.

In choosing between the NC and NCA families of curves the quality of the space being quieted and the character of the low frequency noise should be considered. Both groups of office personnel prefer a noise spectrum that is shaped like the NC curves i.e. an LL-SIL difference equal to 22 units, and experience indicates that negligible improvement in personnel comfort is achieved by still further reducing LL-SIL difference. A majority of the personnel objects to a noise where octave band levels at the low frequencies exceed for a given SIL, the values given by the NCA curves, even when the SIL is low enough for voice communication to be satisfactory.

The two families of curves in Fig. 4.9 and Fig. 4.10 represent, therefore, the lower and upper limits of the LL-SIL difference that is generally acceptable to office personnel. In writing specifications either an NC or NCA number (to the nearest 5 units) or the corresponding sound pressure level (to the nearest db) in the eight octave bands can be given. It is possible, of course, to interpolate between the NC and NCA curves.

Recommended Office Criteria :-

Based on the results and the studies described above and on observations made recently in other industrial buildings, it is recommended that the NC curve for a particular office space be selected with the aid of table 4.2. As was stated in detail above, an NCA curve should be substituted for an NC curve only in instances calling for extreme accuracy.

4.7.2 APPLICATION TO OTHER INTERIOR SPACES :-

Although a detailed study has not been made of the reaction of people to noise in other spaces, fairly dependable conclusions can be drawn from general experience gained in studying noise control problems generated by Ventilating systems and exterior sources. One must also recognize that local situations and local attitudes affect the criticalness of people. What seems adequate in one location may evoke complaints in another. For example, in a convention hall with a near perfect sound system, Table - 4.2.

Recommended noise criteria for office (CB)

| NC curve. | Communication environment. | Typical applications. |
|---|---|---|
| NC-20 to NC-30 | Very quiet office-telephone use satisfactory- suitable for large conferences. | Executive offices and conference rooms for 50 people. |
| NC-30 to NC-35 | "quiet" office, satisfactory for conferences at a 15 ft. table, normal voice 10 to 30 ft, telephone use satisfactory. | Private or semi-private offices, reception offices and small conference rooms for 20 people. |
| NC-35 to NC-40 | Satisfactory for conferences at a 6 to 8 ft table, telephone use satisfactory, normal voice 6 to 12 ft. | Medium sized offices and industrial business offices. |
| NC-40 to NC-50 | Satisfactory for conferences at 4 to 5 ft tables, telephone use occasionally slightly difficult, normal voice 3 to 6 ft, raised voice 6 to 12 ft. | Large engineering and drafting rooms etc. |
| NC-50 to NC-55. | Unsatisfactory for conferences of more than 2 to 3 people, telephone use slightly difficult, normal voice 1 to 2 ft, raised voice 3 to 6 ft. | Secretarial areas(Typing) accounting areas(Business machines)blue print room etc. |
| Above NC-55 | "Very noisy" office environment unsatisfactory, telephone use difficult. | Not recommended for any type of office. |

a designer may get away with an air conditioning system that is 6 to 10 db noisier in the speech bands than that required in a hall with a mediocre sound system.

The greatest variability in requirements for noise control exists in homes. Here the habits and temperaments of people are paramount. Except for the need to converse over the telephone - which requires only that the SIL be lower than 60 - the activities in a home vary. In many homes the noise of children, radios, TV and outside traffic create so much din that noise at an LL of 80 phons would pass unnoticed. In other homes where there are no children or where a writer or a scholar engages in creative thought, an LL may need to be as low as 40 phons to pass unnoticed.

Table 4.3 presents the NC or NCA curves which are believed to provide acceptable noise levels for a variety of interior spaces. The noise levels referred to are to be measured with a particular room in question unoccupied but with other activities inside and outside the building proceeding normally. For comparison with the NC curves, the sound level meter A-scale readings in dbA, computed from the appropriate NC curve are given. It is not recommended that A-scale readings be used in specifications because the same A-scale reading may be obtained for a wide variety of shapes of spectra. Furthermore, the eight octave bands are necessary in the engineering design of noise control measures and no single number can substitute.

The architect or consultant will have to use his own judgement in selecting a curve for a particular specification

Table - 4.3

RANGE OF DESIGN CRITERIA FOR AIR CONDITIONING SYSTEM NOISE CONTROL.

| Type of area. | Total loudness sones (45) | | | A-sound level decibels. | | | NC level decibels | | |
|---|---|---|---|---|---|---|---|---|---|
| | Low limit | Average | High limit | Low limit | Average | High limit | Low limit | Average | High limit |
| **Auditoriums and Music Halls.** | | | | | | | | | |
| Concert and opera halls<br>Studios for sound reproduction | 0.8 | 1 | 1.5 | 25 | 30 | 35 | 20 | 22 | 25 |
| Legitimate theatres<br>Multi-purpose halls | 1 | 1.5 | 2 | 30 | 33 | 40 | 25 | 27 | 30 |
| Movie theatres<br>Semi outdoor amphitheatres<br>Lecture halls, planetarium<br>TV audience studios | 1.5 | 2 | 3 | 35 | 40 | 45 | 30 | 32 | 35 |
| Lobbies | 3 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| **Churches and Schools:** | | | | | | | | | |
| Sanctuaries | 1.0 | 1.7 | 3 | 25 | 30 | 35 | 20 | 25 | 30 |
| Libraries | 1.5 | 2.0 | 3 | 35 | 40 | 45 | 30 | 35 | 40 |
| Schools and Class rooms | 1.5 | 2.5 | 4 | 35 | 40 | 45 | 30 | 35 | 40 |
| Laboratories | 2 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| Recreation halls | 2 | 4 | 8 | 40 | 45 | 55 | 35 | 40 | 50 |
| Corridors and halls | 2.5 | 5 | 8 | 40 | 50 | 55 | 35 | 45 | 50 |
| Kitchens | 4 | 6 | 8 | 45 | 50 | 55 | 40 | 45 | 50 |
| **Hospitals and Clinics:** | | | | | | | | | |
| Private rooms | 1.0 | 1.7 | 3 | 30 | 35 | 40 | 25 | 30 | 35 |
| Operating rooms, wards | 1.5 | 2.5 | 4 | 35 | 40 | 45 | 30 | 35 | 40 |
| Halls and corridors<br>Laboratories<br>Lobbies and waiting rooms | 2 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| Wash rooms and toilets. | 3 | 5 | 8 | 45 | 50 | 55 | 40 | 45 | 50 |
| **Residences :** | | | | | | | | | |
| Private homes (rural & suburban) | 0.8 | 1.3 | 2 | 30 | 35 | 40 | 25 | 30 | 35 |
| Apartment houses<br>Private homes (urban)<br>Two & three family units | 1.5 | 3 | 6 | 35 | 40 | 45 | 30 | 35 | 40 |
| **Restaurants, Cafeterias, Lounges.** | | | | | | | | | |
| Restaurants. | 2 | 4 | 8 | 40 | 45 | 50 | 35 | 40 | 45 |
| Cocktail lounges | 2.5 | 5 | 8 | 40 | 50 | 55 | 35 | 45 | 50 |
| Night clubs. | 2.5 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| Cafeterias | 3 | 6 | 10 | 45 | 50 | 55 | 40 | 45 | 50 |
| **Stores retail :** | | | | | | | | | |
| Clothing stores.<br>Department stores (Upper floors) | 3 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| Department stores (Main floors)<br>Small retail stores | 3 | 6 | 8 | 45 | 50 | 55 | 40 | 45 | 50 |
| Super markets. | 4 | 7 | 10 | 45 | 50 | 55 | 40 | 45 | 50 |

Table - 4.3 (Continued)

| Type of area. | Total loudness sones (49) | | | A sound level decibels | | | NC level decibels | | |
|---|---|---|---|---|---|---|---|---|---|
| | Low limit | Average | High limit | Low limit | Average | High limit | Low limit | Average | High limit |
| <u>Sport activities indoor.</u> | | | | | | | | | |
| Coliseums. | 2 | 3 | 4 | 35 | 40 | 45 | 30 | 35 | 40 |
| Bowling alleys, gymnasiums | 3 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 50 |
| Swimming pools | 4 | 7 | 10 | 45 | 50 | 60 | 40 | 50 | 55 |
| <u>Transportation (Rail, bus, plane).</u> | | | | | | | | | |
| Ticket sales office. | 2 | 4 | 6 | 35 | 40 | 45 | 30 | 35 | 40 |
| Lounges<br>Waiting rooms | 3 | 5 | 8 | 40 | 50 | 55 | 35 | 45 | 50 |
| <u>Hotels</u> | | | | | | | | | |
| Individual rooms or suites. | 1 | 2 | 4 | 35 | 40 | 45 | 30 | 35 | 40 |
| Ball rooms<br>Banquet halls | 1.5 | 3 | 6 | 35 | 40 | 45 | 30 | 35 | 40 |
| Halls and corridors<br>Lobbies | 3 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| Garages | 4 | 6 | 8 | 45 | 50 | 55 | 40 | 45 | 50 |
| Kitchens and laundries. | 4 | 7 | 10 | 45 | 50 | 55 | 40 | 45 | 50 |
| <u>Manufacturing areas.</u> | | | | | | | | | |
| Foreman's office. | 3 | 5 | 8 | 45 | 50 | 55 | 40 | 45 | 50 |
| Assembly lines<br>Light machinery. | 8 | 12 | 20 | 50 | 60 | 70 | 45 | 60 | 70 |
| Foundries, heavy machinery. | 10 | 20 | 40 | 60 | 70 | 80 | 55 | 65 | 75 |
| <u>Offices :</u> | | | | | | | | | |
| Board rooms | 0.5 | 1 | 2 | 25 | 30 | 35 | 20 | 25 | 30 |
| Conference rooms | 1.0 | 1.7 | 3 | 30 | 35 | 40 | 25 | 30 | 35 |
| Executive Office | 1.5 | 2 | 3 | 35 | 40 | 45 | 30 | 35 | 40 |
| Supervisor office.<br>Reception room. | 1.5 | 3 | 6 | 35 | 40 | 50 | 30 | 35 | 45 |
| General open offices<br>Drafting rooms | 2 | 4 | 8 | 40 | 45 | 55 | 35 | 40 | 50 |
| Halls and corridors | 2.5 | 5 | 10 | 40 | 50 | 55 | 35 | 45 | 55 |
| Tabulation & Computation | 3 | 6 | 12 | 45 | 55 | 65 | 40 | 50 | 60 |
| <u>Public buildings :</u> | | | | | | | | | |
| Public libraries.<br>Museums, court rooms | 2 | 3 | 4 | 35 | 40 | 45 | 30 | 35 | 40 |
| Post offices.<br>General Banking areas.<br>Lobbies. | 2.5 | 4 | 6 | 40 | 45 | 50 | 35 | 40 | 45 |
| Wash rooms & Toilets. | 3 | 5 | 8 | 45 | 50 | 55 | 40 | 45 | 50 |

because of the wide range of attitudes toward noise and because of local customs and expectations in different locations. In some cases lack of funds for quieting may require that a calculated risk of complaint be taken. In other, previous experience may have conditioned people exposed to the noise so that they are less, or even more, critical of their noise environment than table 4.3, would predict. Just as in other field involving human reactions, number on page do not alone replace careful analysis of each noise problem and the taking into account of local differences.

LIMITATIONS OF TABLE 4.3.

Regardless of the method used for an expressing it, the proper design criterion for noise level to be specified for any room in an air conditioned building must take into account the following factors :-

1. The activity for which the room is to be used, especially the type of speech communication required. Extraneous sounds with an NC level as low as 25 db may be disturbing to a person seated at the rear of a large room and listening to a speaker, addressing a group at normal voice. On the other hand, discussions between two persons at a distance of 3 ft., will hardly require raising of the voice, even if the background noise has an NCA level as high as 65 db.

2. The type and level of background sound, other than from the air conditioning system, which are normally present in the room. These include internal noise (from office machines etc) as well as external noise (from traffic etc). It is not economical to establish the design criterion lower than the normal background level.

3. The extent to which speech and other distracting sounds may intrude from adjacent rooms. Among other factors, this involves the degree of sound isolation provided by the building construction. It is not recommended that air-conditioning noise be used to mask noise coming through poorly built partitions. However, the acoustic privacy provided even by good construction does depend on a certain amount of neutral background sound in the receiving room. If all out door sounds are excluded, a slight amount of steady, wide band sound from the air distribution system is to be preferred, rather than complete silence.

The room classifications in table largely take into account the first factor listed above. The background and privacy factors, however, have to be evaluated by the architect or engineer for each individual case. For this reason, table cannot list specific recommendations, but only a broad range ( ) of design criteria levels. This does not mean that any level within that range will be equally acceptable. It may be expected, however, that the proper level, established after a thorough analysis of all factors, will lie somewhere between the indicated limits.

## CHAPTER - V.

## PERFORMANCE OF THE SOUND ATTENUATING ELEMENTS.

### 5.1 INTRODUCTION :-

From the study of the previous two chapters (Chapter 3 and 4) we can estimate the amount of acoustical energy that enters a conditioned space. This will be the sum total of the noises generated by the different sources. Depending upon the functional use of the space we can select a suitable NC curve and from this the acceptable noise level in each octave band can be determined. The difference between the generated and the acceptable levels of noise, gives the amount of noise reduction required in each octave band. This reduction of noise required can be affected in a number of ways, for example, by lining the ducts with acoustical materials, using prefabricated package attenuators and so on. The noise reduction can also be effected by suitable choice of the fan, and other mechanical devices.

### 5.2 THE SYSTEM CONCEPT :-

Noise control, in general, is basically a system problem. In general there are many different components which may be manipulated to achieve a particular end result. The system, which is the combination of all the components, contains three major parts, the source, the path and the receiver.

The source is that part of the system in which the vibratory mechanical (noise) energy originates. The important sources in air conditioning system are fan, compressor, grilles, damper or diffusers.

The sound energy from the source travels through the path. There may be a multiplicity of paths, both in solid structure and in air. Generally this takes the form of the air distribution ducts and walls of the room in which the equipment is housed.

The third major part of the system is the receiver. This may be a person disturbed in his sleep, a secretary trying to hear dictation, a lecture or music audience. The behaviour of the receiver is inherently a statistical quantity. The necessary noise control can be effected by suitably manipulating the three components of the system.

Fig. 5.1 shows the source, path and receiver. The noise control measures are shown by dashed lines.

5.3 REDUCTION OF FAN NOISE :-

Most of the noise generated by the fan is transmitted to the air conditioning duct as air borne noise. This determines the amount of sound energy which eventually emerges from the various outlets into the occupied spaces.

In practice, some of the acoustic energy is transmitted to the adjacent structure by way of radiation of sound from the impeller casing or as vibration of fan and motor assembly. If due consideration is given at the design stage and construction of the plant, room and the actual installation of the equipment, then the vibrations will not contribute to any significant degree to the noise of the duct work system.

There are very few ways of reducing the fan noise transmitted to the duct work. The choice of fan will generally be governed by factors other than the acoustic requirements, but as far as possible a fan with minimum noise has to be selected. One method of reducing noise from fan discharge is to fit a silencer to the fan. These units are available in range of sizes to suit a variety of fans. The manufacturers, normally provide with each silencer the information about the attenuation characteristics of the unit. A typical performance curve for one of these fan silencers is given in Fig. 52. When the attenuation values are given in the form of continuous curves, showing the variation with frequency, the values of attenuation in the octave bands are those at the geometric mean frequency at each of the eight octave bands.

5.4 ATTENUATION OF UNTREATED DUCTWORK :-

This is the natural attenuation provided by the duct work without any acoustical lining what so ever. This will be due to the straight duct portion, elbows, branches and end reflection loss.

5.4.1 STRAIGHT UNLINED DUCTS :-

The attenuation in straight ducts is a function of the lengths, size of duct and the frequency of sound. The earliest work has done by Wilbur and Simons (51). Their work consisted of tests on two sizes of unlined sheet metal ducts, one large (12" x 36") and one small (6" x 6") and they determined the respective rates of attenuation (db per foot length) at various frequencies over the greater part of the audio - frequency range. The characteristics obtained showed that in certain frequency ranges below 1,000 cps, both large and small ducts produced a fair degree of attenuation (upto 2 db / ft for the small duct). These frequency ranges were subsequently associated with conditions of panel resonance under which the vibrating duct walls can radiate sound power in the region of their resonant frequencies to the outside spaces, thus absorbing energy from the sound waves in the duct and apparently causing sound attenuation.

The work of Chaddock and Others (52) for square ducts has shown that the attenuation in unlined ducts is independent of the duct wall thickness above 500 cps, but reaches a maximum value at lower frequencies, depending upon the resonance frequency of the duct surfaces. With increased duct size the peak attenuation shifts to a lower frequency. Although the wall thickness increases with the duct size, the added mass of the metal and especially the increased length of the undamped surface from edge to edge, causes the resonance frequency to decrease.

With ducts of different aspect ratios, instead of only one peak attenuation value, as occured in square ducts, two such points appear (53). The peak in the lower frequency is evidently due to the resonance of the two longer duct surfaces, while the other is caused by the resonant vibration of the shorter sides. Table 5.1 gives the measured attenuation values per foot length of various sizes of uninsulated and unlined ducts by octave band analysis.

In general, it would seem that in majority of the the system, where the ducts of run 50 ft., or less, consist of medium to large sections, the noise attenuation provided by unlined sheet metal ducts is of such small magnitude that, without affecting the order of accuracy, it may be omitted from the evaluation of the noise characteristics of an air- conditioning system.

Table 5.1

No flow attenuation (db/ft) in straight unlined and uninsulated air conditioning duct by octave band analysis (63)

| Duct dimension inches | Frequency Band -cps 37.5 75 | 75 150 | 150 300 | 300 600 | 600 1200 | 1200 2400 | 2400 4800 | 4800 9600 | Wall thickness in. |
|---|---|---|---|---|---|---|---|---|---|
| 6 x 6 | 0.08 | 0.11 | 0.29 | 0.21 | 0.13 | 0.25 | 0.25 | [illegible] | 0.032 |
| 12x12 | [illegible] | 0.36 | 0.13 | 0.11 | 0.11 | 0.12 | 0.16 | 0.16 | 0.031 |
| 18x18 | 0.30 | 0.37 | 0.18 | 0.07 | 0.14 | 0.10 | 0.14 | 0.17 | 0.034 |
| 6x12[a] | 0.29 | 0.18 | 0.14 | 0.18 | 0.03 | 0.03 | 0.12 | 0.18 | 0.031 |
| 6x24[b] | 0.83 | 0.70 | 0.14 | 0.23 | 0.03 | 0.12 | 0.12 | 0.16 | 0.034 |

a - also applies to 12 x 6 duct

b - also applies to 24 x 6 duct.

5.4.2 UNLINED BENDS AND ELBOWS :-

In considering the attenuation characteristics of bends and elbows, it is perhaps unfortunate that a smooth multipiece bend which is more acceptable aerodynamically than the square corner elbow, is a poorer sound attenuator of the two.

Fig. 5.3 (64) shows the value of attenuation plotted against each octave bandfrequency for different sizes of elbows, With the increase in elbow size the peak attenuation shifts to a lower frequency band. Tests by Feistrup and Wooler (18) confirm the general trend of these curves and also show that the low frequency peak occurs at the point where the duct width equals half the wave length. This suggests that the mechanism of attenuation in this region of frequency is similar to that described for straight ducts.

The variation of attenuation of elbows with different angles has been studied (53). The attenuation decreases with decrease in elbow angle. This is obvious since there is less chance for the sound to be reflected as the elbow angle and hence the vane curvature decreases. However, the reduction of sound at elbows is much greater than in straight ducts and it should be included whenever appropriate in the calculation of duct work attenuation.

5.4.3 ATTENUATION AT BRANCHES :-

The attenuation provided by duct junctions is not strictly noise reduction at all, except in so far as bends and elbows which might be involved in their construction would produce attenuation as discussed above. It is actually due to a division of the sound energy travelling down the duct between the various branches (54). In determining the degree of attenuation in this case, it is generally assumed that the intensity of sound (power per unit area) is the same in all ducts on the downstream side of the junction so that the proportion of the total sound power fed into any one duct is equal to the ratio of its area to the total area of the branch ducts. The effective attenuation is then given by the equation :

$$\text{Attenuation} = -10 \log_{10} \frac{A}{\Sigma A} \quad \text{db re } 10^{-12} \text{ watts.} \qquad (5.1)$$

where $A$ = cross sectional area of branch duct.

$\Sigma A$ = total cross sectional area of branch ducts.

It is of course, assumed that this equation is just applicable to levels in particular frequency bands as it is to overall noise.

Table 5.2 gives the values of the attenuation for different values of the ratio $A/\Sigma A$.

Table 5.2

POWER LEVEL ATTENUATION AT DUCT BRANCHES.

| Ratio Branch duct area / sum of branch areas | Attenuation db | Ratio Branch duct area / Sum of branch areas | Attenuation db |
|---|---|---|---|
| 1.0 | 0 | 0.25 | 6 |
| 0.8 | 1 | 0.20 | 7 |
| 0.63 | 2 | 0.16 | 8 |
| 0.5 | 3 | 0.13 | 9 |
| 0.4 | 4 | 0.10 | 10 |
| 0.32 | 5 | | |

5.4.4 END REFLECTION LOSSES :-

There is another way in which noise may be reduced in the duct work system before it reaches the spaces to be air conditioned. When it arrives at the duct terminations some of the sound will be reflected back towards the fan. The proportion of acoustic power that will be reflected in this way is a function of frequency and outlet size, and the manner in which the degree of attenuation, or end reflection loss, varies with parameters is shown in Fig. 5.4 (54). Two characteristics have been drawn, one for the duct outlet flush with the wall and the others for a duct terminating in free space. It can be seen that rather more attenuation is produced by the former configuration than by the latter except for noise of very high frequency or for outlets of large dimension. If the duct is not square, the value of L is taken to be approximately equal to $\sqrt{L_1 L_2}$ . The frequency F is the geometric mean frequency of an octave band.

5.5 ATTENUATION OF LINED DUCT WORK :-

The attenuation of sound in an unlined duct is an insignificant quantity. However, a considerable reduction in the noise being transmitted along the duct work can be obtained by lining the duct walls with sound absorbing material and also by the use of configurations inserted in the duct run. The materials used for this purpose should be either of a rigid type or else the suitably faced or coated flexible type, in order to prevent their erosion by the moving air.

There are three places in an air conditioning system where this form of sound attenuation may be used with advantage. The first is in the main duct which feeds the branches that lead to the individual rooms. Acoustical treatment in this location is most effective in reducing noise transmitted along the duct from the plant room and furthermore the attenuation obtained in a single treated section before any branch take offs will be effective for the whole system where as a considerable number of attenuated sections will be required to achieve the same degree of quieting, further downstream after any take offs. Secondly, acoustical treatment can be very useful in the feeder ducts just preceding the system outlets, where it not only reduces the residual fan noise but also any aerodynamic noise generated in the duct work. When used in such positions, it has the added advantage of reducing sound transmission along the ducts from room to room. The third location where acoustical treatment is often valuable is in the duct to the fan inlet immediately behind the fresh air intake. In this position it attenuates the fan noise, which is being transmitted along the inlet duct and radiated into the open air, any outside noise which may be transmitted in the reverse direction into the system through the fresh air intake and any noise transmitted along the return air duct from a noisy place.

5.5.1 SOUND ABSORPTIVE MATERIALS :-

The sound absorptive materials have the ability to absorb a large fraction of the air borne sound which impinges on it, and its function is, generally to reduce the intensity of reflected air borne sound within an enclosure . Sound absorbers occur in many forms, and the relative importance of the fundamental physical properties which control their performance is somewhat a function of form.

The sound absorbent materials more commonly used in

62,452
CENTRAL LIBRARY UNIVERSITY OF ROORKEE.

noise reduction applications are in the form of porous semi-rigid tiles or soft blankets designed for the use directly in front of reflecting boundaries of rooms, enclosures, ducts, and so on. These materials are available in many variations. They may be simple porous masses, they may be covered with loose light weight membranes which can be moved freely by the impinging sound waves; or they may be used with relatively immobile facings, separate or integral, part of the area of which is opened by slots, holes, or fissures, to permit flow of air in and out of the porous body. The sound absorbing characteristics of these familiar materials depend on a number of physical properties such as the fraction of the volume represented by accessible air space within the material, called the porosity, the pressure required to produce a specified flow of air through the material, or the flow resistance, and the mechanical properties, such as the weight, mechanical resistance, and stiffness of the various structural elements of the material for the particular mounting used. Equally important are a number of geometric parameters such as the thickness of the material, the depth of material between the air space and reflecting backing, the thickness of open facings, the size and spacing of perforations in the facings, and the depth of their penetration into the porous body of the material. Sound absorption in these familiar materials occurs principally because the sound waves produce flow of air in and out of the porous body and give up much of their energy in overcoming viscous forces, but in some forms a significant amount of energy absorption occurs, generally at low frequencies, as a result of frictional losses in these mechanical elements of the absorber which are forced to vibrate by the incident sound waves. The sound absorbing characteristics are usually specified, at a number of frequencies encompassing the audible range, in terms of the fraction of incident sound energy which the materials will absorb in a diffuse sound field when they are used in large patches. These are the so called "reverberation room absorption coefficients" having dimensionless units, sabins per square foot. Generally, the value of absorption coefficient ($\alpha$) at a given frequency is the coefficient averaged over all angles of incidence. It has been a common practice to list the value of $\alpha$, usually at 128, 256, 512, 1024, 2048 and 4096 cycles. The present day tendency is to specify at 125, 250, 500, 1000, 2000 and 4000 cycles. For practical purposes, these two series of frequencies can be regarded as identical.

Table 5.3 gives the absorption coefficient of the more commonly used acoustical materials, at the frequencies mentioned above.

For most problems in noise control this information is sufficient for predicting the acoustical effects of a particular material. However, exact treatment of some problems, such as duct conduction, demands more fundamental specification of absorbent materials in terms of their acoustic impedance components as a function of frequency and angle of incidence.

A thin porous blanket has very little absorption at low frequencies, but the absorption efficiency increases with frequency to a maximum value determined by the thickness and specific

TABLE - 5.3. ABSORPTION COEFFICIENT OF SOME BUILDING MATERIALS.

The materials have been classified in three classes.
1. Acoustical plaster and other materials for plastic application.
2. Acoustical blankets, felts, loose materials etc.
3. Prefabricated units, tiles, fibre boards, panels etc.

| Manufacturer or importer | Materials. | Class. | Description. | Absorption coefft. 125 | 250 | 500 | 1000 | 2000 | 4000 | Mounting | Thickness | Wt. lb/ft³ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Cabots quilt | 2 | Mineral wool | - | 43 | 73 | 77 | 69 | - | 1" backing canvas cover | 1 | |
| Tualls Insulation Co.Ltd. | Pamfelt | 2-3 | Asbestos sheets | - | 60 | 55 | 65 | 70 | 75 | 1" air backing. | 1 | 8.00 |
| | Pamfelt) blanket) | 2 | Asbestos blankets. | - | 20 | 60 | 65 | 70 | - | 5/8" —do— | 3/8 | 8.00 |
| | Asbestos spray | 1 | Sprayed asbestos fibres. | -<br>- | 20<br>60 | 35<br>65 | 50<br>60 | 60<br>60 | -<br>- | | ½<br>1 | |
| J.W.Roberts Ltd. | Limpet | 1 | Sprayed ) asbestos ) fibres ) | 25<br>30<br>40 | 28<br>40<br>65 | 45<br>63<br>73 | 75<br>80<br>75 | 73<br>73<br>73 | 70)<br>73)<br>70) | Asbestos sprayed on scratch coat of ½" asbestos cement. | ½ )<br>1 )<br>1½) | 8.00<br>11.0 |
| Stillite Ltd. | Stillite scrim quilt | 2 | Fibrised calcium magnesia in scrim cloth. | 60 | 70 | 60 | 90 | 85 | 80 | Rigid backing | 2 | 3.03 |
| | Semirigid slab acoustic quality | 2 | —do—felt | 30 | 45 | 80 | 85 | 85 | 95 | 2" air backing | 1 | 13.4 |
| | Stilbat | 2 | —ditto— | 15 | 55 | 70 | 80 | 85 | 85 | Muslin covered | 2 | 4.50 |
| Thermacoustic Ltd | Standard thermacoustic | 3 | Selected woodwool with a mineral cementing agent | 15 | 20 | 45 | 80 | 65 | 70 | Rigid backing | 1½ | 3.83 |
| | Heavy Duty Thermacoustic | 3 | —ditto— | 20 | 30 | 60 | 75 | 75 | 75 | Rigid backing | 2" | 6.25 |

physical properties of the porous material (see A Fig. 5.5) Increasing the thickness lowers the frequency at which maximum absorption occurs (56) without greatly affecting the absorption at high frequencies (B Fig. 5.5) For thin blankets of the order of 1/2 in. thickness or less the maximum absorption occurs at so high a frequency that the average absorption is inadequate over the frequency region important for most noise reduction applications. The addition of a properly designed perforated facing to a porous blanket imparts resonant properties which increases the absorption in the desired frequency region, but at the expense of absorption at higher frequencies ( compare A and A', B and B' of Fig. 5.5).

Quite often an air space is left between the absorbing material and duct walls. In such cases, the absorption at the lower frequencies is slightly more but at mid and high frequencies, it is nearly the same as the one without any air space (57). It is important to recognise that effective absorption at low frequencies implies considerable bulk in the sound absorber. With practical perforated facings and porous fillers, peaking the absorption characteristics in the range of 100 - 200 cps requires a depth of 4 to 8 in. behind the facing. This essential bulk (and the consequent shipping cost) is one reason why integral units for low frequency absorption are not so prevalent commercially, as the thinner materials which are suitable for the more frequently encountered noise reduction problems.

5.6.2 ATTENUATION IN STRAIGHT LINED DUCTS :-

Ducts lined with porous material show a low

## TABLE - 6.3. ABSORPTION COEFFICIENTS OF SOME BUILDING MATERIALS.

The materials have been classified in three classes.
1. Acoustical plaster and other materials for plastic application.
2. Acoustical blankets, felts, loose materials etc.
3. Prefabricated units, tiles, fibre boards, panels etc.

| Manufacturer or Exporter. | Materials. | Class. | Description. | Absorption coefft. 125 | 250 | 500 | 1000 | 2000 | 4000 | Mounting | Thickness | Wt. lb/ft³ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Cafferata & Co. Ltd. | Thistle Plaster | 1 | Light weight porous aggregate with gypsum binder containing a foaming agent | 15 | 25 | 35 | 60 | 60 | 50 | Rigid backing | 10/32 | 3.37 |
| Fibre glass Ltd. | Superfine fibre | 2 | Glasswool | 10 | 30 | 60 | 60 | 85 | 85 | Covered with 10% perforated metal rigid backing. | 1 | 0.60 |
| | Resin bonded fibre | 2 | Bonded glass wool | 10 | 60 | 50 | 63 | 60 | 70) | Unfaced rigid backing | 2 | |
| | | | | 10 | 35 | 85 | 65 | 73 | 60) | | 1 | 0.80 |
| | | | | 20 | 50 | 70 | 60 | 75 | 60) | | 2 | |
| Celoplast Ltd. | Celoplast 1" | 3 | Double panel one-side perforated filled with kapok strips. | 20 | 30 | 65 | 90 | 70 | 60 | Rigid backing | 1 | 2.0 |
| | Celoplast 1 5/8 | 3 | Ditto, filled with "fibre-glass" strips | 25 | 45 | 60 | 60 | 35 | 20 | -do- | 1 5/8 | 3.31 |
| | Celoplast acoustic tile | 3 | ditto. with fibreglass mat behind 1/10 in. lining. | 15 | 65 | 60 | 60 | 50 | 45 | -do- | 1 | |
| Jones & Broadbent | J&B Loose Mineral wool | 2 | Inorganic fibres | 50 | 60 | 60 | 65 | 60 | 65 | | 2 | 9.00-11.00 |
| | J&B Felted mineral | 2 | felted & bonded inorganic fibres | 53 | 73 | 85 | 85 | 65 | 85 | | 2 | |
| | J&B Acoustic panels | 3 | Loose mineral wool between wire netting & one side muslin cloth | 50 | 60 | 60 | 65 | 60 | 65 | | | 20.77 |
| May Acoustics Ltd. | Sabinite | 1 | Gypsum plaster | - | 20 | 37 | 60 | 60 | - | Rigid backing | ½ | |
| | May acoustic asbestonite | 2 | Composed of asbestos fibres | - | 65 | 85 | 80 | 70 | - | Covered with muslin, rigid backing | | |
| | Cabots quilt | 2 | Mineral wool | - | 43 | 74 | 77 | 60 | - | 1½" backing canvas cover | 1 | |
| Newalls Insulation Co. Ltd. | Paxfelt | 2-3 | Asbestos sheets | - | 60 | 83 | 65 | 70 | 73 | 1" air backing. | 1 | 6.00 |
| | Paxfelt blanket | 2 | Asbestos blankets. | - | 20 | 60 | 63 | 70 | - | 5/8" -do- | 5/8 | 6.00 |
| | Asbestos spray | 1 | Sprayed asbestos fibres. | - | 30 | 35 | 50 | 60 | - | | ½ | |
| | | | | - | 60 | 65 | 60 | 60 | - | | 1 | |
| J.W. Roberts Ltd. | Limpet | 1 | Sprayed asbestos fibres | 23 | 25 | 45 | 76 | 75 | 70) | Asbestos sprayed on scratch coat of ½" asbestos cement. | ½ | 9.00-11.00 |
| | | | | 20 | 40 | 63 | 80 | 70 | 73) | | 1 | |
| | | | | 40 | 65 | 75 | 73 | 73 | 70) | | 1½ | |
| Stillite Ltd. | Stillite scrim quilt | 2 | Fabricated calcium magnesia in scrim cloth. | 50 | 70 | 60 | 60 | 65 | 60 | Rigid backing | 2 | 2.63 |
| | Semirigid slab acoustic quality | 2 | -do- felt | 50 | 45 | 60 | 80 | 65 | 65 | 2" air backing | 1 | 10.40 |
| | Stilbat | 2 | -ditto- | 15 | 65 | 70 | 80 | 85 | 83 | Muslin covered | 2 | 4.00 |
| Thermacoustic Ltd | Standard thermacoustic | 3 | Selected woodwool with a mineral cementing agent | 15 | 60 | 45 | 60 | 65 | 70 | Rigid backing | 1½ | 3.00 |
| | Heavy duty Thermacoustic | 3 | -ditto- | 50 | 80 | 60 | 75 | 75 | 75 | Rigid backing | 2" | 6.12 |

physical properties of the porous material (see A Fig. 5.5) Increasing the thickness lowers the frequency at which maximum absorption occurs (55) without greatly affecting the absorption at high frequencies (B Fig. 5.5) For thin blankets of the order of 1/2 in. thickness or less the maximum absorption occurs at so high a frequency that the average absorption is inadequate over the frequency region important for most noise reduction applications. The addition of a properly designed perforated facing to a porous blanket imparts resonant properties which increases the absorption in the desired frequency region, but at the expense of absorption at higher frequencies ( compare A and A', B and B' of Fig. 5.5).

Quite often an air space is left between the absorbing material and duct walls. In such cases, the absorption at the lower frequencies is slightly more but at mid and high frequencies, it is nearly the same as the one without any air space (57). It is important to recognise that effective absorption at low frequencies implies considerable bulk in the sound absorber. With practical perforated facings and porous fillers, peaking the absorption characteristics in the range of 100 - 200 cps requires a depth of 4 to 8 in. behind the facing. This essential bulk (and the consequent shipping cost) is one reason why integral units for low frequency absorption are not so prevalent commercially, as the thinner materials which are suitable for the more frequently encountered noise reduction problems.

5.5.2 ATTENUATION IN STRAIGHT LINED DUCTS :-

Ducts lined with porous material show a low attenuation at low and high frequencies with a more or less pronounced peak in between.

Low attenuation at low frequencies is due to the high acoustic impedance at low frequencies. At high frequencies where the wave length is less than the width of the air space in the duct, the transverse waves are rapidly absorbed at the beginning of the duct and the sound propagated along the du t consists mainly of a beam of axial waves. This beam tends to keep in the middle of the duct and is affected little by the absorbent walls.

The density of the porous material affects the attenuation to a considerable extent. At low densities, it is transmission along the lining and not along the air passages that determine the performance. It has been found (53) that in bulk specimens of rock wool within the density range of 5 to 25 lbs per cft., the attenuation of sound in decibels per foot is approximately proportional to the density. Knowing the attenuation at one density, the attenuation values at other densities can be derived. Table 5.4 (53) gives the attenuation values for a rock wool sample of density 10 lbs./cft. at different frequencies.

Table 5.4

ATTENUATION IN BULK ROCKWOOL AT DIFFERENT FREQUENCIES.

| Frequency cps | 100 | 200 | 400 | 800 | 1600 | 3200 | 6400 |
|---|---|---|---|---|---|---|---|
| Attenuation db/ft. | 24 | 31 | 40 | 53 | 71 | 82 | 116 |

If a duct is lined on four sides, the total attenuation may be obtained by adding, arithmatically, the attenuation of the 'sides' to the attenuation of the 'top and bottom' of the duct. A square duct lined on four sides will have about twice the attenuation of a duct lined on only two opposite sides.

At low frequencies $\left( \dfrac{\text{duct width } (l)}{\text{wave length}(\lambda)} < 0.1 \right)$

Sabine's approximation for attenuation (59) may be used for straight fully lined ducts. Sabine found that attenuation of a lined duct could be expressed as

$$NR \left(\text{db/ft}\right) = 12.6 \; \alpha^{1.4} \; \frac{P}{A} \qquad \text{------} (5.2)$$

where NR = noise reduction in db/ft.

$\alpha$ = absorption coefficient of the lining material

P = acoustically lined perimeter of duct, in

A = Cross sectional open area of duct, $in^2$

This formula was developed for a set of ducts ranging from 9 x 9 in., to 18 x 18 in. and for cross sectional ratios upto 2:1. Using materials having $\alpha$ between 0.20 and 0.60 and frequency ranges between 256 cps and 2048 cps. This formula becomes increasingly inaccurate as $\frac{l}{\lambda}$ increases, and its use must be restricted to small values of $\frac{l}{\lambda}$ (low frequencies).

An alternate form of equation for attenuation is (60)

$$\text{Attenuation} = K \frac{P}{A} \quad \text{db/ft} \qquad \text{------} (5.3)$$

where K is a number which depends upon the absorption coefficient of the lining and is given by Fig. 5.6.

5.5.3 LINED ELBOWS :-

The attenuation of sound in a bend lined with acoustical material is much more than that of an an unlined bend. At the bend multiple reflection of the sound waves takes place which increases the attenuation. Very little data exist concerning the attenuation, of bends greater or less than 90°. Some data are given by Doroack (61) and Brittain (62) for bends less than 90°. As a rough approximation, one may assume the attenuation to be proportional to the angle. For example, the attenuation of a 30° bend may be estimated to be one-third that of a 90° bend.

Fig. 5.7 (64) gives the value of attenuation for square corner bends with turnig vanes. The lining should extend several duct widths beyond the corner in the direction of sound transmission. Total attenuation due to the bend is the sum of the attenuation value given by the graph and the attenuation for a straight duct having the same total length of lining used.

5.5. 4 RECTANGULAR CELL-OR- SPLITTER-TYPE ABSORBERS :-

When the available duct run is short, added attenuation can be gained at the expense of increased pressure drop by dividing the duct into an aggregate type of sound-absorbing rectangular cells or by dividing the duct with sound absorbing plate splitters. To provide adequate sound absorption, all surfaces of the duct and splitters are lined as shown in Fig. 5.8. Eq. 5.2 can be used for calculating the attenuation.

As in the case of lined ducts, the attenuation increases steadily with frequency up to a point where the wave lengths of sound waves become less than about two thirds the width of air channel. Above this frequency, the attenuation fall rapidly owing to the increasing tendency of the sound to keep to a beam and not to flow laterally to the absorbent surfaces. The attenuation increases for a given frequency and thickness of lining as the width of air channel is reduced.

If the case of two similar lined ducts operating side by side is considered, symmetry would suggests that the two adjacent walls can be removed leaving two intervening linings back to back, without substantially altering the sound attenuation of each. Hence a splitter of thickness S is equivalent as regards attenuation to a lining of thickness $t = S/2$.

In fig. 5.9 (63,64) two arrangements A and B are shown which have the same amount of absorbing material, the same free area and the same outside duct dimensions, but the amount of sound absorbing surface and thickness of lining are different. The experimental attenuation curves for these arrangements are shown in Fig. 5.10. It can be seen that low frequency attenuation can be increased by having thicker and fewer number of splitters, so long as the resulting increase in absorption coefficient offsets the decrease in the ratio P/A .

## 5.6 PACKAGED SOUND ATTENUATORS :-

Packaged (prefabricated) attenuators are especially useful where high attenuation over a wide frequency of range is required and the available length is limited. Such attenuators (sometimes called sound traps, silencers or mufflers) are available in both rectangular and round forms. Essentially, they consist of an outer metal shell containing various air passage configurations The air passages have one or more surfaces made of a perforated metal, behind which is packed light weight fibrous, sound absorbent material. The free opening of the air passages, at their narrowest point, may be from 25 to 60 percent of the total body area (outside dimensions) of the attenuator, depending upon the required attenuation and allowable pressure drop. The air passages may be formed with either, (a) curved passages with relatively large free openings or (b) straight through passages with relatively small openings.

Rectangular packaged attenuators are available in units with both full size and half size duct connections. A full size unit has duct connections of the same size as the outside dimensions of the attenuator body itself. A half size unit has connections for duct only half as big as the attenuator body in either width or height. Full size connections are commonly used when attenuators are assembled in banks for large ducts. Half size connectors are used mostly in small ducts, especially when duct velocities are relatively high.

6.6.1 NOISE REDUCTION MECHANISMS :-

Low frequency noise can be reduced by changing the duct section rapidly. In order to reduce pressure drop, the duct is made uniform in cross section by lining the sides of the expanded section with porous blanket.

Roughly, the noise reduction at low frequencies depends almost entirely on the ratio of the total cross section of the attenuator to the open cross section through which air flows - provided, of course the length is fixed.

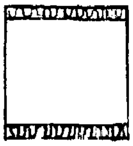
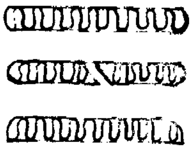
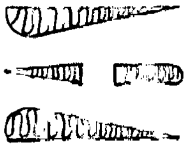
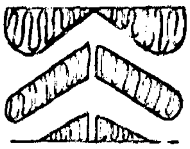
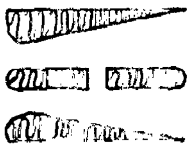
At high frequencies, sound waves behave much like a beam of light. If line of sight through the muffler is prohibited, the noise reduction will generally be quite large. High frequency noise depends much more on the bends than on the lining of the duct walls.

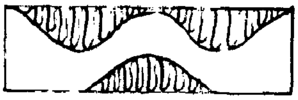
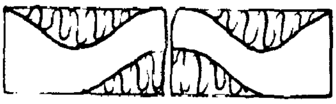
With the cross section, open area and geometry held constant, the noise reduction generally increases with increasing length of the unit, but it is definitely not directly proportional to the length. But the percentage of high frequency noise reduction varies little as a function of length. The noise reduction of the combination of two units in series is not equal to the sum of the noise reductions of each, but significantly less than the sum.

5.6.2 RATING OF PACKAGED ATTENUATORS :-

Prefabricated attenuators are rated in terms of net noise reduction (NR) in each octave band. The ratings should include air flow resistance, noise regeneration, and the net noise reduction of the attenuator itself, exclusive of such factors as end reflection, beaming, flanking, standing waves, and room absorption.

Typical ratings of some commercial prefabricated package attenuators are given in Table 5.5 (65). The data is not complete, in that it does not include data on regeneration of sound,

| [illegible] | [illegible] | [illegible] section | 20<br>75 | 75<br>150 | 150<br>300 | 300<br>600 | 600<br>1200 | 1200<br>2400 | [illegible] |
|---|---|---|---|---|---|---|---|---|---|
| [illegible] | 24 |  | 2 | 6 | 6 | 8 | [illegible] | 23 | [illegible] |
| [illegible] | 34 |  | 4 | [illegible] | 5 | 8 | 16 | 22 | 12 |
| [illegible] | 34 |  | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] |
| [illegible] | 37 |  | | [illegible] | [illegible] | 13 | [illegible] | 35 | [illegible] |
| [illegible] | 34 |  | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | 29 | [illegible] |
| [illegible] | 23 |  | 6 | 11 | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] |
| [illegible] | 46 |  | 7 | 13 | 13 | 18 | 30 | 47 | [illegible] |
| [illegible] | 60 |  | [illegible] | 13 | [illegible] | 30 | [illegible] | [illegible] | [illegible] |
| [illegible] | [illegible] |  | 14<br>14 | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] |

| Unit | Length [illegible] | Cross section | [illegible] 75 | 75 150 | 150 300 | [illegible] | 600 1200 | [illegible] |
|---|---|---|---|---|---|---|---|---|
| A-5 | [illegible] |  | 16 | 24 | 39 | [illegible] | [illegible] | [illegible] |
| C-1 | [illegible] |  | 9 | [illegible] | 11 | 15 | 25 | 37 |
| [illegible] | 36 |  | 11 | [illegible] | 13 | [illegible] | 30 | [illegible] |
| [illegible] | [illegible] |  | 7 | [illegible] | [illegible] | [illegible] | [illegible] | 40 |
| [illegible] | [illegible] |  | 11 | 12 | 20 | [illegible] | 45 | [illegible] |
| [illegible] | [illegible] |  | 10 | 10 | 17 | 25 | 45 | [illegible] |
| C-[illegible] | [illegible] |  | 11 | 14 | 25 | 33 | 56 | 52 |
| C-7 | [illegible] |  | 13 | 15 | [illegible] | 34 | [illegible] | [illegible] |

Note :- Units of A series are 24 in. deep.
Units of [illegible] series are [illegible] in. deep.
Units of C series are 14 in. deep.

pressure, drop and etc.

5.6.3. SELECTION OF A UNIT :-

The selection of a prefabricated attenuator for a given system involves three steps. :-

1. Select the attenuator length necessary to provide a noise reduction(NR) in each octave band at least 3 db above the required attenuation calculated, less the attenuation provided by any duct lining or lined plenum.

2. Select the face area of the packaged attenuator on the basis of the pressure drop allowed. If a pressure drop of approximately 15 percent of the total static pressure in the air handling system is allowed forthe attenuators, the outside body dimensions of the attenuator will generally be approximately the same as the connecting duct work.

3. Check for regeneration of noise at the air flow velocity corresponding to the selected face area, and if necessary increase the face area.

Air flow through the attenuator passages produces turbulence which, in turn, generates sound. The sound power leaving the attenuator cannot be less than that generated by the air flow. Regenerated sound power levels, therefore, limit the net attenuation obtainable . The next larger size unit should be substituted if it is found that, in any octave band, the power level of the regenerated sound is not at least 5 db lower than the power level ahead of the attenuator minus the required attenuation.

5.7 PLENUM CHAMBERS :-

A plenum chamber is a large box connected by small openings to two or more ducts. These are generally used after the fan discharge and also in the return air circuit. Plenum chambers treated with sound absorptive materials often provide very effective means of noise control. They are used in ventilation system to reduce noise. A sound treated corridor may b used to reduce cross talk between near by rooms with open doors, the corridor acts as a plenum chamber.

Plenum chambers can be single or multiple i.e. two or more chambers in series. Increased attenuation can be obtained in case of multiple plenums.

Fig. 5.11 shows the sketch of a single plenum chamber. The chamber is a box with dimensions L x W x H and the entrance and the entrance and exit, assumed to be equal , are slits with dimensions l x w in diagonally opposite positions. The ratio l/L which is denoted by n, is of particular interest as it represents the percentage open area at the plenum inlet and exit. It has been found (63) that with increasing values of n, the measured attenuation in decibels decreases for a particular frequency as shown by Table 5.6. At higher frequencies the attenuation increases. This is due to the fact that at higher frequencies the absorption coefficient of the lining material also increases.

In case of double plenums, the attenuation is not twice that of a single chamber of the same dimensions, but slightly less. This is due to the interaction between successive chambers.

Table - 5.6

MEASURED ATTENUATION IN DECIBELS FOR SINGLE PLENUM CHAMBERS.

L = 39 7/16 in, W = 25 in.,

H = 5½ in. ½ in. Glass Fiber lining.

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Value of $\alpha$ | 0.076 | 0.227 | 0.373 | 0.529 | 0.690 | 0.831 | 1.000 |
| Attenuation in 37.5 - 750 cps. | 13.6 | 13.2 | 9.3 | 10.4 | 9.4 | 5.0 | 2.5 |

5.7.1 CALCULATION OF PLENUM ATTENUATION :-

In order to estimate the attenuation to be expected from any plenum design, a method of calculation is necessary. No completely rigorous solution to this problem has yet been advanced. The following gives an approximate method of estimation (66,67)

$$\text{Attenuation in decibels} = 10 \log_{10} \left( \frac{\text{PWL in}}{\text{PWL out}} \right)$$

$$\frac{\text{PWL in}}{\text{PWL out}} = A \left\{ \frac{\cos\theta}{2\pi d^2} + \frac{1}{R} \right\} \quad \text{———(5.4)}$$

WHERE PWL - stands for sound power

$A$ = lw, the area of the exit,

$d^2 = (L-l)^2 + H^2$, d is the slant distance from entrance to exit.

$\cos\theta = H/d$

$R = a/1-\alpha$ is the plenum room constant,

$\alpha$ = is absorption coefficient of treatment

a is the total absorption, sabins, in plenum.

For large openings the agreement between the actual attenuation and the attenuation as given by Eq. 5.4 is rather poor. This is due to the fact that as the entrance and exit openings are of the order of three fourths of the chamber lengths the concept of a plenum begins to disappear. At low frequencies, (where the wave length approaches or exceeds chamber dimensions) a correction factor of about 5 to 10 db should be added to the computed value of attenuation.

5.7.2 MODIFIED PLENUM FOR HIGH ATTENUATION :-

By preventing direct transmission of sound from entrance to exit, a high attenuation is obtained du to multiple reflections of the sound waves. This can be achieved by means of lined baffles. Fig. 5.12 shows one of the designs. Here there is no direct path from entrance to exit. In fact, all path with less than three or four reflections are eliminated. The measured attenuation inthis cased compared well with two or three conventional chambers having approximately the same resistance to air flow.

5.8 OUTLET SOUND ABSORBERS :-

Outlet sound absorbers are rectangular or plate cells installed directly behind an outlet or they may be the lining of the pan or plaque outlet. They are particularly effective in the elimination of high frequency whistles which are generated by air flow in the ducts. They are also employed in large systems with long runs whereonly a few outlets near the fan require treatment. . Frequently, outlet cells are the only means of correcting existing noisy installations, a s the duct sections directly behind the outlets may be the only sections accessible for treatment (Fig. 5.13)

5.0 SELECTION OF NOISE CONTROL DEVICES :-

The choice of any one or more of the wide selection of noise control devices depends upon the amount and frequency range of noise reduction required.

Duct lining is the most common noise control device and is often used wherever the lining can serve the double purpose of noise reduction and thermal insulation. The transfer of thermal insulation from outside to inside the duct may in some cases provide all the needed noise reduction at little additional cost if the transfer is made in the building design.

Package units, splitters, or egg-crate units may be used where insufficient noise reduction results from the lining required for thermal insulation or where a large amount of reduction is required in a short duct run. Larger freedom in the choice of the frequency region of high noise reduction exists with these units than with simple lined ducts. Since costs are roughly proportional to noise reduction and volume, any excess noise reduction provided in frequency regions where it is not needed is generally wasteful. Furthermore, excess noise reduction in some frequency regions may lower the background levels excessively and decrease privacy.

Where fibrous attenuating materials are used, suitable facings should be chosen to give adequate protection against erosion at the anticipated stream velocities. Where large amounts of noise reduction are required and when the resulting noise levels are low, it is always necessary to consider for the attenuator the self noise of the attenuator due to air turbulence within its passages or at its termination. For low noise levels, such as are required for television studios or comparable spaces, the air velocities in ducts and in open areas of attenuators should not exceed 1,000 ft/min, duct outlet velocities should be even lower. In case the velocities are very high, of the order of 2,000 ft/min, and above, the noise generated at elbows, package units etc., are quite appreciable and they have to be considered.

5.10 PREVENTION OF CROSS TRANSMISSION BETWEEN ROOMS & THROUGH DUCT WALLS

Cross transmission as applied to an airconditioning system refers to direct transmission of sound through ducts from one duct opening to another or through unsealed cracks. A grille or opening not only allows air and noise to enter the room, but it allows sound from the room to enter the duct. Specially, if the duct length between two rooms is short, and one of the rooms is very is very noisy, then this noise will be transmitted to the other room through duct. The other room will also be equally noisy and all the noise control measures applied will be quite useless, unless the short lengths of duct between the two is also acoustically treated. The duct should be lined with acoustical material. If this is not sufficient, then the splitter type of duct construction has to be adopted.

When ducts originate at the fan in the equipment room and pass through this room on the way to the space being air-conditioned or ventilated, the ducts need lagging on the outside. Unless the ducts are lagged, some of the mechanical noise from air in the

the air stream, and thereby carried to the room. In such cases, that portion of the duct which is exposed to the sound in the equipment room be lagged with material, such as cork, pipe covering or other sound damping material to prevent the sound from entering the duct. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense or unless it is surface with another sound impervious layer such as metal or board. The lining material inside the duct also reduces the sound transmission from outside to air stream.

5.11 DETERMINATION OF ROOM LEVEL :-

The preceding pages have presented the means for estimating the sound power level produced by various sources normally encountered in air conditioning systems. Also procedures have been given for estimating the reduction in the sound power level provided by various types of duct elements between the source of noise and a room. By subtracting in any one frequency band all the values of noise reduction from the sound power level of the source, the net sound power level entering the room due to any source is obtained. This net sound power level radiated into the room is used in the following calculations to determine the noise levels at a listener's position in a room.

The sound pressure level at any point in a room due to a source, say a grille, is made up of two parts, (i) the direct sound from the source to the listener, and (ii) the reverberant sound arriving at the listener's position after one or more reflections from the surfaces of the room.

The direct sound is determined by the directivity factor of the source Q (see appendix) and the distance from the listener r . The reverberant sound is independent of the directivity factor and the distance and is determined only by the room constant R, which in turn is determined by the total absorption of the various surfaces of the room and its contents. The sound pressure level at a listener's position at a distance r and at an angle $\theta$ from the axis perpendicular to a grille face is given by the equation (5.5).

$$SPL = PWL + 10 \log \left( \frac{Q}{4\pi r^2} + \frac{4}{R} \right) + 0.5 \text{ db} \qquad (5.5)$$

where SPL = sound pressure level at angle $\theta$ and distance r from duct opening , db re 0.0002 microbar (see Fig.5.15)

PWL = Power level radiated from duct opening, db re $10^{-13}$ watts.

Q = directivity factor of duct opening, which varies with frequency, duct size, and angle $\theta$ (Fig.5.14)

r = distance from duct opening to listener , ft.

R = Room constant , ft$^2$ - varies with frequency.

The directivity factor depends upon the position of the duct opening with respect to walls, edges and corners of the room. Fig. 5.15 shows four methods for terminating an air conditioning duct in a room. In each of these cases, the noise power radiates into room. Approximate values of Q for these positions are given in the Fig. 5.14 for $\theta$ = 0 and $\theta$ = 45°

The directivity factor Q is of importance only when the listener is near the duct termination, i.e. when r is small. At large distances, i.e. when the first term in the parenthesis becomes negligible compared to the second, the listener hears only the reverberant sound.

The room constant R is defined by the equation :

$$R = \bar{\alpha} S/(1-\bar{\alpha}) \quad \text{------} (5.6)$$

where,

$\bar{\alpha}$ = average sound absorption coefficient (dimensionless) for the room at the mid frequency of the band of noise being considered.

S = total area of the boundary surfaces of the room, $ft^2$

Where a room contains several surfaces, each with different absorption coefficients, the average absorption coefficient may be determined by the equation :

$$\bar{\alpha} = \frac{S_1\bar{\alpha}_1 + S_2\bar{\alpha}_2 + \cdots}{S_1 + S_2 + \cdots} \quad \text{------} (5.7)$$

Where $S_1, S_2, \cdots$ = the areas of each type of absorbing material.

$\bar{\alpha}_1, \bar{\alpha}_2 \cdots$ etc = the absorption coefficients of each type of material at the mid frequency of the band being considered.

The room constant R at various frequencies can be computed approximately from known values for acoustical treatment, room furnishings, and occupants. Alternatively, the values of room constant R at 1000 cps can be obtained from Fig. 5.20 as a function of room size for five classes of rooms with proportions of about 1:1.5:2. These proportions give $S = [illegible]\ V^{2/3}$

The correction for variation of room absorption with frequency is given in Fig. 5.17 for large and small rooms. In order to get the room constant in other octave bands, the room constant at 1000 cps from Fig. 5.16 has to be multiplied by the multiplier given by Fig. 5.17.

The rooms are classified into five groups on the basis of reverberation time. This is the basic criteria of acoustical quality of the room and is closely related to the type of activity in the room. A short reverberation time means that sounds die out quickly. Dead rooms are good for conversation at close range and for studios for sound recording. Live rooms, on the other hand, have considerable echo and, therefore, are not suited for speech communication because words and syllables will blur into each other However, live rooms are desirable for certain types of music, especially music composed for large medieval churches.

Table 5.7 gives the typical reverberation ratings of the different classes for rooms.

Table - 5.7

Typical reverberation ratings.

| Use of room. | Sound absorbing surfaces. | Reverberation time, secs. | Rating for use in Fig.5.16 |
|---|---|---|---|
| Radio and T.V. studios music Room. | Special acoustic treatment | 0.2 to 0.25 | Dead |
| Department stores Restaurants | Acoustical ceiling, many people and | 0.4 to 0.6 | Medium |

Table 5.7 (continued)

| Use of room | Sound absorbing surfaces. | Reverberation time, T, sec. | Rating for use in Fig. 5.16 |
|---|---|---|---|
| Offices, libraries Homes, Hotels, Rooms, conference rooms, lecture halls | Acoustical ceiling (or soft furnishings) many people | 0.9 1.1 | Average |
| School rooms, Art Galleries, Building Lobbies, Hospitals, Small churches | Many people or some acoustical material | 1.0 to 2.2 | Medium Live |
| Large churches Gymnasiums Factories | People only | 2.5 to 4.5 | Live |

5.11.1 NOISE LEVEL FOR A SINGLE SOURCE :-

The value of sound pressure level (SPL) relative to the sound power level (PWL) of a source as calculated from Eq.5.5 has been plotted in Fig. 5.18 (C). To use the graph one must know the directivity factor (Q) for the duct opening in the specified direction and the room constant (R) for the room at the frequency of interest. The graph is entered from the bottom at the distance r. Follow diagonally upward and to the left until the proper value of Q is reached, then move vertically to the value of R for the room. Finally read the relative sound pressure level from the ordinate at the left . To obtain the actual sound pressure level add the sound power level to the ordinate readings.

An example is shown by the dotted line in Fig. 5·18 for r = 7 ft, Q = 2, and R = 2,000 ft². The relative sound pressure level is -28 db. Assume a sound power level of 120 db re $10^{-12}$ watts. Then 120 db is added algebraically to the relative sound pressure level to give the actual sound pressure level at the listeners' position. The result is 120 - 28 = 92 db re 0.0002 $\mu$ bar.

5.11.2 NOISE LEVELS FOR MULTIPLE SOURCES :-

Generally, in airconditioning installations, a number of grilles are provided in a room to supply the conditioned air. In such cases, the noise level computation has to be modified. The reverberant noise level is determined by using the total sound power level from all duct openings. To get the total sound power level perform the operation. Total

$$PWL = 10 \log_{10} \left( \text{antilog}_{10} \frac{PWL}{10} + \text{antilog} \frac{PWL}{10} + \right) \quad \text{——— (5.8)}$$

The noise level at a specified position near any particular duct outlet can be determined from Eq. 5.8 by replacing the room constant R by an "effective" room constant $R_o$ which applies for the duct in question. $R_o$ is given by

$$R_o = R \; \frac{\text{antilog } (PWL \text{ duct}/10)}{\text{antilog } (PWL \text{ total}/10)} \quad \text{——— (5.9)}$$

5.12 REDUCTION OF NOISE IN THE AIRCONDITIONED SPACES :-

If it has not been practical or for some reason it is not desirable to reduce the ventilation system noise to an acceptable level for distribution by one or more of the methods already described, further control of the noise can be achieved by acoustic treatment of the air conditioned spaces, although it should be appreciated that this is seldom the most economic approach to the general problem of controlling system noise. It is, however, the obvious method to use in those spaces where strict control of all background noise including that of the air conditioning system is required. This is especially true if the total volume of such spaces is very small in relation to a remaining air conditioned space, the acoustical requirements of which can be met without any treatment of the air conditioning system.

The method of determining the room attenuation has been already described and the design procedure for providing acoustic treatment of the air conditioned spaces is simply an extension of the same calculation. The basic principle is to increase the average absorption coefficient of each space requiring treatment, thereby reducing the degree of reverberation. Fig. 5.10 shows that this will increase the value of the room constant R and from Eq. 5.5 it may be seen that this will cause a reduction in the sound pressure level heard by the occupants of the room. The room absorption coefficient can be increased by lining one or more of its bounding surfaces with sound absorbing material. Some form of acoustic tiling is normally used for this purpose because of its pleasing architectural effect. A wide variety of acoustic tiles is available and the manufacturers of these usually provide data on the absorptive properties of these. By inserting the appropriate values of absorption coefficient and the surface area in Eq. 5.7, the average absorption coefficient of an acoustically treated room can be determined for each of the octave frequency bands. The actual sound pressure levels in the room are then obtained by procedure given previously. Comparison of the resulting noise, spectrum with that produced by the untreated system will indicate the degree of attenuation provided by the acoustic treatment of the room.

5.13 NOISE REDUCTION IN SELF CONTAINED AIRCONDITIONING UNITS (70)

The noise in the self contained units is due to compressor, condenser, evaporator and air flow.

5.13.1 COMPRESSOR NOISE :-

The compressor noise can be classed under (1) bearing noise, (2) motor noise (3) valve noise (4) gas noise and (5) lubrication system noise. The first three have already been discussed.

(a) GAS NOISE :-

The refrigerant gas stream is a prolific source of noise. In the reciprocating type compressor, where both suction and discharge gases are evidently disturbed by the rapid opening and closing of the valves, this is especially true. Suction and discharge mufflers are used to lessen the force of such impacts in the gas and to reduce the amount of emitted acoustic power.

The primary function of the discharge muffler is to convert the pulsating gas flow to a continuous stream. Any appreciable pulsation in the discharge gas stream will excite the refrigerant condenser and may result in a serious noise problem, since the gas condenser acts as an acoustic radiator of appreciable area. There are many possible discharge muffler designs. One, for example, employs a series of volumes separated by restrictions. This can be an effective design for a particular range of frequencies. There are others, such as a volume filled with small diameter shot and a resonant cavity type silencer. Compromises are necessary, when a muffler is selected for application to refrigeration compressors. These are generally caused by the limitations of space, effect on efficiency, and cost.

Suction muffler design is also influenced by the above factors. Hence a compromise has to be made in its selection.

(b) LUBRICATION SYSTEM NOISE :-

Noise in a refrigeration compressor caused by or associated with the lubrication system frequently is very troublesome. Some of the common noises associated with a compressor lubrication system are as follows :- If the compressor is splash lubricated, then churning, gurgling and splashing noises of the oil will be produced. Sometimes the elimination of these noises can be accomplished by strategic location of oil deflectors or baffles or by a design change in the oil slinger. If the compressor has pressurised oil system, assuming proper mechanical construction of the parts to the specified design, noise problems will most likely to be associated with the hydraulic behaviour of the system. The multiplicity of dilution ratios of refrigerant gas to lubricating oil contributes to difficulties in lubrication. Restrictions, hot spots, trapped volumes of liquid, and centrifugal forces are factors in the design which must be thoroughly analysed.

8.13.2 CONDENSER AND EVAPORATOR NOISE :-

A noisy condenser is usually caused by mechanical vibrations from the compressor or by unabsorbed gas pulsations in the discharge tubing. A forced convection condenser has a reduced noise level than a free-convection condenser. Isolation of the condenser from the mounting and also use of mufflers in the discharge line help to reduce the noise.

Evaporators are excellent acoustic radiators. The most probable source of the energy exciting the evaporator is the refrigerant liquid gas mixture leaving the capillary. The noises include intermittent hissing, squealing or whistling, and rumbling. In room air conditioners fan noise usually predominates, but capillary noises can be a very irritating part of the total noise.

Capillary noise can be minimised by mass loading of the capillary tube and liquid line in the immediate area of the capillary exit, and muffling of the liquid gas mixture as it leaves the capillary. A length of heavy wall tubing with a smaller inside diameter than the evaporator tubing between the capillary exit and evaporator entrance is effective in reducing the noise energy. Another method is to use a sintered, porous plug at the capillary exit. Changing the angle at which the capillary meets the evaporator tube or bending the liquid line before it reaches the main portion of the evaporator may also help to reduce the noise level. If the capillary entrance is flooded constantly, the noise generated will be less intermittent.

5.12.3 AIR FLOW NOISE :-

Appearance and installation considerations will probably determine the size and shape of the enclosure in case of room air-conditioners. These considerations may require an arrangement of components which do not produce best airflow circuit characteristics. Minimum pressure drop is conducive to low noise level, and therefore, it would be wise to when laying out the airflow circuits, to keep :-

1. Air velocities low.
2. A minimum number of flow area contractions and expansions.
3. A minimum number of bends.

Fan with minimum noise level has to be chosen for the given air flow and pressure drop.

Generally these units are lined inside and at the air passages with sound absorbent materials. The design of units with low noise level is by actual tests with various arrangements of the components taking into account the details such as appearance possibilities, performance, efficiency and cost.

---

## CHAPTER - VI

### ISOLATION OF MACHINE VIBRATIONS :

### 6.1 INTRODUCTION :-

Vibration is a continuing condition in which the oscillatory force or motion exists at a constant frequency, or combination of frequencies. This condition is designated steady-state, because the identical pattern of vibration amplitude is repeated during each cycle. It is also referred to as forced vibration, in as much as energy is supplied in sufficient quantity to compensate for any loss as a result of damping in the system.

In almost all the reciprocating and rotary machines, the vibrations are invariably present. The unbalanced forces due to the static unbalance or dynamic unbalance are transmitted to the foundations of the machines and this may set the floor and the adjoining building in vibration. This may cause excessive noise and vibration even in remote parts of the buildings. It is possible to have static balance in all machines. But in reciprocating machines, such as compressors, complete dynamic balancing may not be possible always. In fans, the initial balance can be upset by uneven accumulation of dirt during operation, or a slight damage.

Vibration and shock control for machinery in some cases provides virtually a complete solution to the problem of transmitted noise. This is particularly true if the problem is noise transmission to the floor below the machinery and to a lesser degree to the room adjoining a machinery room, or if the problem involves vibration and noise transmission through pipe lines. However, in air conditioning systems vibration control of the same equipment is only a part of the overall treatment required in the successful noise reduction programme. The resilient machinery mountings cost so low compared to the cost and inconvenience of the other types, of noise reduction treatment that it often pays to start with installation of resilient mountings before other treatments.

6.2 THEORY OF VIBRATION CONTROL :-

A very simple equation (75,73) applies to determine the transmission of steady state vibration, the constantly repeating sinusoidal wave form of vibration generated by such equipments as fans, compressors, engines and process.

$$\text{Transmissibility } T = \frac{F_t}{F_d} = \frac{1}{(f_d/f_n)^2 - 1} \quad \text{-------} \quad (6.1)$$

[This equation is exact for steel springs because they have straight line load deflection characteristics and negligible damping. When the equation is used for organic materials, the following correction will normally give conservative results: for rubber and neoprene use 50 percent of the static deflection when calculating $f_n$. For cork, use $f_n$ equal to one and onehalf times the natural frequency determined by actual test.]

where $f_d$ = frequency of disturbing vibrations, cpm.

$f_n$ = natural frequency of resiliently mounted system, cpm.

$F_d$ = unbalanced force acting on the resiliently supported system.

$F_t$ = force transmitted through the resilient mounting.

$$f_n = 188\sqrt{1/d} \qquad \text{------------} \quad (6.2)$$

where, $d$ = static deflection of the resilient mounting in inches.

$$d = W/K \qquad \text{------------------------} \quad (6.3)$$

where, $W$ = weight of the mounting

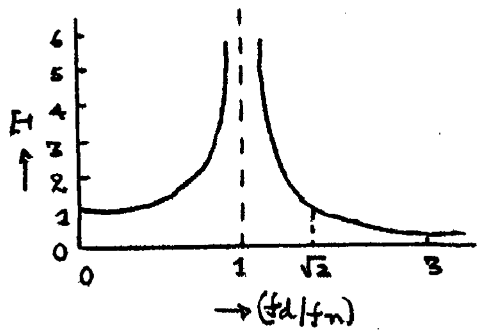
$K$ = stiffness factor of the mounting in lb/in. of deflection.

The natural frequency, $f_n$, of a resiliently mounted system is the frequency at which it will oscillate by itself if a force is exerted on the system and is then released. The more the deflection in the system, the lower is the natural frequency of the system. The importance of this can be seen by reexamining the equation 6.1 re-written in the following form ;

$$F_t = F_d \left( \frac{1}{(f_d/f_n)^2 - 1} \right) \qquad \text{------------} \quad (6.4)$$

Obviously, we want to minimise the transmitted force, $F_t$. Since the disturbing force $F_d$, is a function of the machine characteristics and cannot be reduced except by dynamic balancing of the machine ( or by reducing the operating speed, which is seldom practical) the transmitted force can be reduced only by minimising the function $1 / (fd/fn)^2 - 1$ .

This can be accomplished only by increasing the frequency ratio, $f_d/f_n$ . However, since the disturbing frequency $f_d$, is fixed for any given machine and is a function of the rpm, it can seldom be changed. The only remaining variable is the mounting natural frequency , $f_n$. Reducing $f_n$ by increasing the static deflection of the resilient mountings reduces the vibration transmission. This explains why the efficiency of machinery mountings increases as the resiliency and deflection increases.




$$T = \frac{1}{\left(\frac{f_d}{f_n}\right)^2 - 1}$$

Fig. 6.1 Transmissibility-frequency ratio. Applies for steel springs and to other materials with very little damping. The footnote to Eq. 6.1 gives an approximate correction for damping in organic materials.

Fig. 6.1 shows the effect of varying frequency ratios on the transmissibility . Note that for $f_d/f_n$ less than $\sqrt{2}$ the use of mountings actually increases the transmissibility above that which would result if no isolation were used and the machine bolted down rigidly. In fact, if careless selection of the mount, results in a mounting natural frequency equal, or nearly equal, to the disturbing frequency, a very serious condition called resonance occurs; in Eq. 6.4 the denominator of the transmissibility function becomes zero and the transmitted force $F_t$ , theoretically becomes infinite. As the ratio $f_d/f_n$ decreases beyond $\sqrt{2}$ , the resilient mountings reduce the transmitted force. Table 6.1 gives some suggested maximum design transmissibilities for different types of job conditions. Fig. 6.2(77) shows a chart which can be used to select proper resilient mountings when the following job characteristics are known, weight per mounting, disturbing frequency, and the design transmissibility. The chart shows the limitation of the various types of isolation materials, particularly helpful data in selecting the proper media.

TABLE - 6.1 (a) :-

Maximum allowable design transmissibilities :- (78).

In quiet surroundings, on upper floors, and in similar areas, more effective isolation is required. Installations near column footings or load bearing walls may transmit vibrations up into floors above. Below are given the general ranges of allowable vibration transmissibilities for different job conditions.

(a) Extremely critical conditions, Large heavy machinery in penthouses, directly over offices, libraries, hospital rooms, on resilient mezzanines, etc. maximum tolerable transmissibility, 10 per cent.

(b) Critical conditions : All upper floor installations except the extremely critical ones above. Also some ground floor installations near quiet areas in hospitals, broadcasting studios, libraries, etc. The bulk of installations requiring steel spring isolators fall into this category - maximum tolerable transmissibility between 10 and 20 percent.

(c) Non-critical condition: Installations in basements, on ground floors (except above) in industrial areas, where some vibration transmission can be tolerated because of greater building or ground mass to absorb it, greater ambient noise level, and distance from any critical areas. This is typical range generally covered by rubber and for cork - maximum tolerable transmissibility, between 40 and 60 percent.

TABLE - 6.1 (b). Recommended Maximum Transmissibilities for air-conditioning equipment installed in critical areas :-

| Type of Equipment | | Transmissibility $\beta$ |
|---|---|---|
| Absorption machines | | 5 |
| Centrifugal compressors. | | 2 |
| Reciprocating compressors | | |
| Upto 15 HP | | 15 |
| Upto 20 to 60 HP | | 10 |
| 75 to 150 HP | | 5 |
| Cooling Towers. | | 15 |
| Condensers | | 20 |
| Fan coil units. | | 20 |
| Centrifugal Fans. | | |
| Fan rpm | Fan size | |
| 800 and higher | All sizes. | 5 to 10 |

(Continued).......

Table 6.1(b) (Continued)

| Type of equipment | | Transmissibility % |
|---|---|---|
| Centrifugal fan. | | |
| Fan rpm | Fan size. | |
| 350 to 800 | All sizes | 10 to 20 |
| 200 to 350 | 48" dia. or smaller | Noise isolation only. |
| 200 to 350 | 54" dia. or larger | 20 to 30. |
| Piping, centrifugal pumps. | | 5 |
| Package airconditioners. | | 10 |
| Steam generators. | | 5 |

6.3 VIBRATION ISOLATION MATERIALS :-

Various materials are used as vibration isolators. Among them the important ones are cork, rubber and steel springs.

Cork :- Cork was the original vibration and noise isolation material and has been used for this purpose for at least a hundred years. The most widely used form of cork today is compressed cork made of pure granules of cork without any foreign binder, compressed and baked under pressure with accurately controlled density. Cork can be used directly under machines but its widest application is under concrete foundations (Fig. 6.3). It is not affected by oils, acids normally encountered, or temperatures between 0° F and 200° F, but it is attacked by strong alkaline solutions and will not rot under continuous cycles of moistening and drying. Cork under concrete foundation still giving good service after 20 years indicates a long useful life when properly applied. Cork is fairly good as a low frequency shock absorber, but its use as a vibration isolator is limited to frequencies above 1000 cpm. Because of the large amount of damping in cork, the natural frequency cannot be computed from the static deflection and must be determined in tests by vibrating cork under different loads to find the resonance frequency. The limiting values for cork given in Fig. C.2 were determined in this manner.

Rubber :- Rubber has very good isolation characteristics, is fairly good for low frequency shock absorption, and is useful as a vibration isolator for frequencies above 1200 cpm. Rubber is not affected by acids or alkalies, but is not recommended for use in sunlight. The temperature range for natural rubber is 50° F to 150° F, that of neoprene 0° F to 200° F. Neoprene rubber is recommended for applications where there is continuous exposure to oil. Special rubber compounds are available to meet conditions beyond those cited. Rubber tends to lose its resiliency as it ages. The useful life of rubber is about 7 years under impact applications, though it retains its sound insulation value for much longer time.

Several typical methods of loading rubber parts are shown in Fig. C.4.

(a) The rubber may be stressed in compression by placing it between two parallel plates, and loading it by a force, acting perpendicular to the plates. The distance between the plates decreases in response to the force, and the rubber bulges around the periphery or into holes cut through the assembly perpendicular to the plates.

(b) The rubber may be stressed in shear, if placed between and adhered to two parallel plates. The forces then are applied in a direction parallel to the plates, and the resultant deflection is approximately in the direction of the forces.

(c) Rubber is capable of carrying a considerable tension load. It is seldom used in this manner, however, because of the difficulty of obtaining suitable end connectors and because a member under severe tensile stress may fail if it experiences slight surface damage. It is considered poor engineering practice to load an adhered or bonded joint in tension.

(d) Rubber may be stressed in flexure, in which case a portion of the rubber member has a bending moment applied thereto. One surface, therefore, is subjected to tensile stress, and the opposite surface to compressive stress. It is particularly convenient to obtain large deflections in response to very light loads in this manner, and it is relatively easy to control the stiffness in various directions.

Individual molded rubber mountings are generally economical only with the light and medium weight machines, since heavier capacity mountings approach the cost of the more efficient steel spring isolators. Pad type rubber isolation has no such limitations.

Steel Springs :- Steel spring isolators provide the most efficient method of isolating shock and vibration, approaching 100 percent effectiveness. The higher efficiency is due to the greater deflections which they provide. Standard steel spring isolators such as those shown in Fig. 6.5 provide deflections upto 1½" compared to about 1/4" maximum for rubber and other materials, while special steel spring isolators can give deflections upto 10 inch. Since the performance of steel springs follows very closely the equations of vibrations control, their performance can be very accurately predetermined, eliminating costly trial and error which is sometimes necessary in other materials. Steel spring isolators are generally equipped with adjustable snubbers, since steel springs themselves contain no damping. (Damping is sometimes useful in limiting the movement of resiliently mounted machines, but damping reduces the isolation efficiency of the mounting). Most steel spring isolators are equipped with built in leveling bolts, which eliminate the need for shims installing machineries. The more rugged construction possible in steel spring isolator provides for a long life usually equal to that of the machine itself. Since high frequency noises sometimes tend to bypass steel springs, rubber sound-isolation pads are usually used under spring isolators to stop such transmission into the floor on critical installations.

Table 6.2 tabulates the useful range of cork, rubber and steel springs for different equipment speeds.

(Table 6.2 next page)

TABLE - G.2

The relative effectiveness of steel springs, rubber and cork in the various speed ranges.

| Range | RPM | Steel springs | Rubber | Cork. |
|---|---|---|---|---|
| Low | Upto 1200 | Required | Not recommended except for shock. | Unsuitable except for shock. |
| Medium | 1200-1800 | Excellent | Fair | Not recommended |
| High | Over 1800 | Excellent for critical jobs. | Good | Fair to good. |

G.4 APPLICATIONS :-

Figs. G.3, G.4 and G.5 illustrate the application of resilient machinery mountings to prevent transmission of structure borne noise and vibration. Properly designed mountings now permit installation of the heaviest equipment in penthouses on roofs directly over office and sleeping areas.

Such upper floor installations permit certain operating economies and release valuable basement space for garaging automobiles. When heavy machinery is installed on upper floors, great care must be taken to prevent vibration transmission which often shows up many floors below when machinery is installed on new concrete filled ribbed metal deck floors. They also permit installation of heavy machinery in old building which were not originally designed to accommodate such equipment.

Steel spring isolators resting on rubber sound pads are used on practically all anechoic room installations to prevent transmissionof structure borne vibration and noise into these sound rooms. Vibration and noise transmission through piping particularly in air conditioning installations, is a serious problem. When refrigerating compressors are installed on resilient mountings, provision should be made for the flexibility in the discharge and intake piping to reduce vibration transmission. This can be accomplished either through a flexible metallic hose or by providing for flexibility in the piping itself. This isoften accomplished by running the piping for a distance equal to 15 pipe diameters both vertically and horizontally before attaching the piping to the structure.

Additional protection is provided by suspending the piping from the building on resilient hangers or by supporting it from below on resilient mountings. Flexible canvas approximately 3 diameters long should be used on the intake and discharge of fans and such flexibility should not be nullified by subsequent rigid installation of duct.

Effective vibration control for machines is usually quite inexpensive, seldom exceeding 3 per cent of the equipment cost. In many cases resilient mountings pay for themselves immediately by eliminating special machinery foundations or the need to bolt the equipment to floor. It is much cheaper to prevent vibration and structural noise transmission by installing mountings. Resilient machinery mountings should not be considered a panacea for noise transmission problem. However, they have a definite use in the overall solution of noise problems, and their intelligent use can produce gratifying results at low cost.

6.5 SELECTION OF ISOLATORS :-

A reciprocating compressor rated at 50 hp and running at 800 rpm is to be isolated from the concrete foundation. It is to be mounted on four steel springs. Determine the stiffness of the spring if the compressor weighs 1000 lbs.

From Table 6.1(b) we can take the transmissibility to be 10 percent.

From Eq. 6.1,

$$T = \frac{1}{(f_d/f_n)^2 - 1}$$

$$0.10 = \frac{1}{\left(\frac{800}{f_n}\right)^2 - 1}$$

$$\therefore \quad f_n = 241.5 \text{ cpm.}$$

Natural frequency is, therefore, 241.5 cpm.

From Eq. 6.2,

$$241.5 = 188\sqrt{1/d}$$

$$\therefore \quad d = 0.606 = W/4 \times (1/K)$$

$$\therefore \quad K = \frac{1600}{4 \times 0.608}$$

$$= 657 \text{ lbs/in.}$$

Stiffness of tho spring is 657 lbs./in.

## CHAPTER - VII.

## SYSTEM DESIGN AND TYPICAL PROBLEM.

### 7.1 GENERAL PROCEDURE :-

The noise control problem is composed of three parts (1) The source (2) the path and (3) the receiver. The noise control can be effected at any or all of these parts.

A good procedure for attacking a noise control problem in an air conditioning system will be illustrated by reference to Fig. 7.1. Sound attenuation in the duct is composed of losses in the lined and unlined portions of the duct, losses at the bend and end reflection losses. Added attenuation of the system noise is achieved by adding additional lining to the duct or providing package attenuation units. Noise is also produced at the grille and in high velocity system, at the elbows. All losses are expressed in decibels. The procedure is as follows :-

1. Obtain sound power level in octave bands of the fan, using Eq. 3.1, and Fig. 3.6 . For this the horsepower and static pressure of the fan must be known. If the manufacturer's data is available , it is the best.

2. Determine the existing attenuations in the path, for example, natural duct losses, losses at bends, and reflection losses at the outlet. For this the information given in Chapter IV can be used.

3. Determine the net source power level by subtracting value of step (2) from step (1).

4. Determine the sound power level of the grille noise either from published data or from the data given in

Chapter XII.

5. Determine the sound pressure level in octave band at a specified listener's position due to grille noise alone which is radiated from the end of the duct, using the acoustical properties of the room and the position of listener (see Determination of Room Levels).

6. Determine the allowable sound pressure level in octave bands (the criterion) at the listener's position either from published standard data (NC curves) or from discussions with the architect or the ultimate user of the room under design.

7. Subtract the criterion levels from the sound pressure levels at the listener's position (in decibels) to determine that the levels at least meet the criterion in each octave band. If the grille noise is higher than the criterion in any octave band, a larger grille size must be used.

Note : The total noise level in the room is due to both the grille noise and the fan noise. If the grille noise level just meet the criterion, then the fan noise emitted in the room must be at least 5 db below the criterion. Alternatively, the grille noise levels and the fan noise levels in the room could each be 3 db below the criterion, because when two noises of the same intensity are combined, the total level is 3 db higher than either. As a third, choice, the fan noise could just meet the criterion, and the grille noise be at least 5 db below the criterion.

8. Determine the sound pressure levels in octave bands at a specified listener's position, using basic data the net source power level from the duct, the acoustical properties of room,

and the position of the listener as done in step (6).

9. Subtract the criterion levels from the sound pressure levels at the listener's position (in decibels) as done in step (7) to obtain the required amount of noise reduction in each of the eight octave bands.

10. Choose an economical means of providing the required noise reduction which usually must be obtained by treating the ducts between the source and the receiver either by adding an absorbing lining or by inserting package units of known attenuation as a function of frequency. Sometimes a combination of various attenuating elements described in Chapter XV will have to be used to meet the criterion. Because noise may be induced due to vibration, a consideration of the fan, compressor, condenser, evaporator and other equipment mounting may be necessary. Also, if the equipment room is adjacent to a quiet space, the construction of the equipment room walls to provide adequate noise reduction should require attention. It is usually good practice to provide slightly more acoustical treatment than is necessary, thus obtaining a safety factor. A factor of safety often used is 5 db greater noise reduction than would be indicated by the above procedure.

Frequently the controlling noise source in a building with a central station system is sound transmission through the walls of the apparatus room or from a duct plenum. Where such a room or plenum is near occupied rooms, the noise transmitted through the walls must be determined. The combined sound pressure level resulting at a listener's position should be compared with the criteria as done in step (7) for noise produced by grilles.

An extremely important matter to consider in this matter of isolation of sound by walls or acoustic barriers is the problem of cracks or small openings in the barriers. A small crack can seriously reduce the effectiveness of a heavy wall. A door that is poorly seated may be useless in providing the required sound isolation between rooms. Even though gasketings is provided, it seals poorly. The door must make contact with the gasket around its entire perimeter to be adequately sealed. It is not possible when such gasketing materials as felt, sponge, rubber, and hard rubber are used. Experience indicates that the best type of door gasket is a soft hollow rubber refrigerator-type gasket. This gasket will mould itself to the contours and the irregularities which normally occur in doors and provide, if properly applied, a continuous seal around the door perimeter.

It is not possible to lay down rigid procedure and rules for noise control design, but the design to some extent is influenced by the actual situation of the building with respect to its surrounding areas.

7.2 TYPICAL PROBLEMS :-

a) Part of a proposed scheme of airconditioning for a single storey building is shown in Fig.7.2. It is known that no space in the building will need to be any quieter than the particular library shown in the figure. The library is 150' x 50' with a 15 ft. high ceiling and is supplied with 7000 rpm cfm of conditioned air.

A central airconditioning plant is located in the basement. The centrifugal fan used handles a total quantity of air of 22,000 cfm at a static pressure of 3 in. water gauge. It is driven by a 25 hp electric motor. The main duct from the fan discharge has a horizontal run of 6 ft. and then branches off, where the air flow is divided into two parts as shown. The branch leading to the library has a vertical rise of 30 ft. and then a horizontal run of 10 ft. with a smooth bend in between. The distribution system has a number of branch take offs to various other offices and rooms. Air is discharged into the library through five grilles of size 12 in. x 10 in. located in the longer wall at a height of 12 ft. from the floor level. The return is through a central vertical riser located in the center of the library.

It is required to determine what acoustic treatment, if any, will be needed to ensure that the system noise heard by any occupant of the library will be acceptable.

SOLUTION :-

It is necessary to determine the noise level in the library as result of the air conditioning system for two positions of the occupants (i) nearest to a grille and (ii) remote from the grilles. The greater of the two has to be taken for design.

Since fan is the chief source of noise, the system design is based on the fan noise, only. To make an allowance for the noise generated due to equipment, the noise level is made 3 db lower than the criteria. The return air duct may also radiate sound into the space. To allow for this the noise level is made, further 3 db lower than the acceptable level and the same acoustic treatment is applied for return duct as for the supply duct. The mechanical equipment is mounted on suitable isolators to reduce transmission of vibrations through walls. Since the equipment room is remote from the conditioned space, the vibration transmission will be almost zero. The noise generation level at the system is negligible since the velocities are low.

NOISE CHARACTERISTICS OF THE SYSTEM :-

1. Fan Noise :-

From Eq. 3.3,

Sound power output of fan = 100 + 10 log 25 + 10 log 3 = 119 db

Referring to Fig. 3.6, and using the upper limit of the spectrum for centrifugal fans, the octave band data shown in Table 7.2(a) are obtained and entered as line 1 in Table 7.1.

2. Natural Attenuation of Untreated Duct work :-

(i) Two way junction at B:-

Area of branch = 24 x 40 sq. in.

Total area of branches = (36 x 40) + (24 x 40)

= 60 x 40 sq.in.

Attenuation at junction $= -10\log \dfrac{A}{\Sigma A}$ (Eq.5.1)

$= -10\log \dfrac{24 \times 40}{60 \times 40}$

= 4 db (in each band)

(line 2 of Table 7.1)

Attenuation in each band due to bend at B is read from Fig. 5.3, D = 36 in.

The values are entered in line 3.

(ii) Bend at C. - Since this bend is a smooth one, this is assumed to have no attenuation.

(iii) Branch take off at D.

Assuming that the velocities in the branch as well as the main duct to be as the same ,

$$\frac{A}{\Sigma A} = \frac{9500}{12000} = 1/1.26$$

$$\therefore \text{ Attenuation} = -10 \log \frac{1}{1.26} = 1.00 \text{ db in each band}$$

(line 4)

(iv) Branch take off at E :-

Similar to (iii) we have

$$\frac{A}{\Sigma A} = \frac{7000}{8800} = 1/1.26$$

$$\text{Attenuation} = -10 \log \frac{1}{1.26} = 1.0 \text{ db in each band}$$

(Line 5)

(v) Straight ducts :-

The total length of duct upto F is 6 + 30 + 10 = 46 ft. say 50 ft. The duct is covered with insulation and attenuation values per foot are given in Table 7.2(b) (From Table 13, page 218 of Ref. 9 ) The entire duct is assumed to be between medium and large size. The calculated values of attenuation is entered in line 6.

(vi) End Reflection Loss :-

This is taken from Fig. 5.4. For the 12" x 18" grille used here , $L = \sqrt{12 \times 18} = 14.7$ in.

Values of the mean frequency , F, $\frac{FL}{1000}$ and the corresponding end reflection loss are given in Table 7.2(c)(entered in line 7)

The total natural attenuation of the system in each band is the sum of the values from line 2 to line 7 and this is entered in line 8.

3. Power Level at the Grilles due to Fan :-

The attenuation for the length from F to K is neglected. The take offs to the grilles are assumed to be smooth, and hence the attenuation is negligible.

Hence the net power level just before the take off at F is found by subtracting from the fan power level (line 9). Since there are five grilles, the power is to be divided equally among them. This is obtained by subtracting 10 log 5 = 7 db from the total and this is entered in line (10).

4. Sound Power Level Generated by Grilles :-

Using grilles of vertical or horizontal bar deflection type, the power level in the speech interference bands is obtained by Fig. 8, page 359 of Ref. 8.

$$\text{Face velocity} = \frac{1500}{\frac{12 \times 18}{144}} = 1000 \text{ fpm}$$

The power level is 41 db per sq. ft. of face area. Since the grille, here is having an area of 1.5 sq.ft., the generated level for the grille is 41 + 10 log 1.5 = 43 db (Eq. 3.10).

5. Sound Pressure Level at the Occupant's position due to Grille Noise.

(a) Near field :- Assuming the occupant to be 5 ft. tall, the nearest occupant will be at 7 ft. from the wall for a sound wave with $\theta$ = 45 Deg. (Fig. 5.15 [illegible]) to reach him. Then the distance between the occupant and the grille is $\sqrt{7^2 + 7^2}$ = 9.9 ft., say = 10 ft. The directivity factor Q is obtained from Fig. 5.14 for $\theta$ = 45 deg. The curve B is used since the duct is flush with the wall and is neither at a corner or an edge.

The value of FL/1000 and Q are tabulated in Table 7.2(d). The value of L used is 12 in., the shorter side of grille F is the mean frequency of band.

The library is taken as an "average room" (Table 5.7) with $\bar{\alpha}$ = 0.15.
The value of room constant R at 1000 cps is,

$$R = \frac{S\bar{\alpha}}{1 - \bar{\alpha}} = \frac{(250 + 80)\ 2 \times 15 \times 0.15}{1 - 0.15}$$

$$= 1030 \text{ sq.ft.}$$

Fig. 5.10 cannot be used since it holds good for rooms with $S = 6.25\, V^{2/3}$ where S = surface area and V is the volume of the room and in this case the dimensions of the library donot satisfy the above equation.

The correction factor is obtained from Fig. 5.17 using the curve for large rooms. The corrected value is tabulated in Table 7.2 (e).

Since there are five grilles in the library, the value of effective room constant $R_o$ (Eq. 5.9) has to be used to calculate the relative SPL near the grille. Since all the grilles are identical,

$$R_o = \frac{1}{5}\ R \text{ total}$$

The value of $R_o$ is entered in Table 7.2(e).

11. General Design Considerations :-

The equipment chosen should have the minimum noise generation level so that they do not contribute much to the fan noise. The cooling tower is to be installed remote from the conditioned spaces, preferably on the roof and the fan exhaust duct treated acoustically. All the bends should be smooth and the cracks are to be eliminated. The return duct generally leads to a plenum and this can be treated with absorbing material to reduce noise transmission through return duct into conditioned space. The fresh air intake can be lined with absorbing material to prevent the outside noises entering the system. If the levels of the outside noise is determined by a noise survey, then the required treatment can be designed.

Table - 7.1

Summary of Calculations for the Example 7.2(a) given.

| No. | Item | Octave frequency bands - cps, | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 20<br>75 | 75<br>150 | 150<br>300 | 300<br>600 | 600<br>1200 | 1200<br>2400 | 2400<br>4800 | 4800<br>10000 |
| 1 | Actual band PWL of fan(db re $10^{-13}$ watts) | 119 | 117 | 112 | 107 | 102 | 97 | 92 | 87 |
| 2 | Attenuation at junction B | 4 | 4 | 4 | 4 | 4 | 4 | 4 | 4 |
| 3 | Attenuation at bend at B | 0 | 2 | 6.5 | 7 | 6 | 11 | 12 | 13 |
| 4 | Attenuation at junction D | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |
| 5 | Attenuation at junction E. | 1.3 | 1.3 | 1.3 | 1.3 | 1.3 | 1.3 | 1.3 | 1.3 |
| 6 | Attenuation in straight duct from A to F | 15 | 15 | 7.5 | 3.7 | 3.7 | 3.7 | 3.7 | 3.7 |
| 7 | End reflection loss | 12 | 7 | 4 | 1 | - | - | - | - |

ranges (up to 300 - 600 cps) and 1.75 times either, in the other higher frequency bands (Reference 65 ). The attenuation values are entered in line 19. It is seen that the treatment very well meets the criterion. To allow for the noise due to equipment and return grille, it was stated earlier that a further reduction of 6 db has to be provided. The package units provide this also.

Alternatively, one A-5 package unit (Table 5.5) can be used. This unit is 36 in. long and 24 in. deep and can be inserted between E and F. The attenuation values are entered in line 20. The balance of attenuation still required is obtained by subtracting line 20 from line 18. This is entered in line 21.

Further attenuation is obtained by duct lining. The 30 ft. length from D to C is lined with 1 in. thick layer of glass wool fibres on all sides (absorption coefficient is given in line 22) and then one 2 in. thick splitter (Fig. 5.8 c) is introduced and mounted parallel to the 43 in. side, making two 10 " x 43" cells.

$$\text{Then } \frac{\text{perimeter}}{\text{area}} = \frac{2 \times 2 \times 53}{2 \times 10 \times 43} = \frac{53}{215}$$

The attenuation is given by Eq. 5.2 as

$$\text{Attenuation} = 12.6\, l \frac{P}{A} \alpha^{1.4}$$

$$= 12.6 \times 30 \times \frac{53}{215} \times \alpha^{1.4}$$

$$= 93\, \alpha^{1.4}$$

The value of $\alpha^{1.4}$ is given in line 23 and the attenuation values in line 24. This treatment is satisfactory in meeting the criterion as well as the further 6 db reduction in all the bands.

used, then from Table 4.1, the SC-30 is suitable. Then from Fig. 4.6, we find that the sound pressure levels for SC-30 curve are very nearly the same as given by NC = 30.

7. The allowable pressure level in the speech interference bands is 30 db (line 15). The grille noise is 25 db which is well below the allowable and hence the grille size is satisfactory.

8. The sound pressure level due to fan noise at occupant's position 10 ft. from grille (near field) is found by adding the values in lines 13 and 10 and entered in line 16.

The sound pressure level due to fan noise in the farfield is found by adding lines 14 and 9 and entered in line 17.

9. Attenuation Required :-

The attenuation required is found by subtracting the acceptable pressure levels given by line 15 from values either in line 16 or 17. In this case the sound pressure level in the near field is greater and hence that is used. The required noise reduction in each band is entered in line 18.

10. Acoustic Treatment :-

A study of the values in line 18 and indicates that greatest attenuation is required in the 300-600 and 600-1200 cps bands and smaller values in the lower and higher frequency bands.

Two package units of the C - 8 type of Table 5.5 can be used in parallel. This combination can be inserted in the duct run between E and F. Each unit is 30 in. long and 12 in. deep and hence can be safely used. The total attentuation is assumed to be the sum of the individual values in the lower frequency

With the values of $R_0$ and Q the relative sound pressure levels for an occupant's position at 10 ft from grille is read from Fig. 5.10. This is entered as line 13.

The average value of relative SPL in the speech interference bands is -18 db. Consequently the sound pressure level of the grille noise at 10 ft., is 43-18 = 25 db.

(b) Reverberant Field :-

Now, the relative sound pressure level in the reverberant field has to be found out. The horizontal part of the room constant curve in Fig. 5.18 is used to read the relative SPL corresponding to the corrected value of R. This is entered in line 14.

The average value of relative SPL in the speech interference bands is -25 db. The noise level due to all the five grilles = 43 + 10 log 5 = 50 db.

Consequently the sound pressure level due to grilles in the reverberant field is 50 - 25 = 25 db. This value can be used in the calculation since the near field calculations also gave [illegible] 25 db.

C. Noise Criteria :-

The combined speech communication and loudness criteria is used. Table 4.3 shows that an acceptable noise level for libraries is given by the NC-30 curve of Fig. 4.7. The values of sound pressure level read from the NC-30 curve are entered in line 15.

If instead, the speech communication criteria is

Table - 7.1 (Continued).

| No | Item | Octave frequency bands - cps. 20 75 | 75 150 | 150 300 | 300 600 | 600 1200 | 1200 2400 | 2400 4800 | 4800 10000 |
|---|---|---|---|---|---|---|---|---|---|
| 8 | Total duct attenuation | 34.9 | 30.3 | 24.3 | 18 | 16 | 21 | 22 | 23 |
| 9. | Net power level radiated into space due to fan. | 84.7 | 86.7 | 87.7 | 89 | 86 | 73 | 70 | 64 |
| 10. | Power level at each grille outlet due to fan. | 77.7 | 79.7 | 80.7 | 82 | 79 | 66 | 63 | 57 |
| 11. | Directivity factor Q | 2 | 2 | 2.2 | 2.7 | 3.2 | 3.6 | 3.9 | 4.1 |
| 12. | Corrected value of room constant R sft. | 530 | 550 | [illegible] | 742 | 1060 | 1800 | 3120 | [illegible] |
| 13. | Relative SPL at 10 ft. fromgrille. | -14 | -14 | -13 | -15 | -16 | -17 | -20 | -21 |
| 14. | Relative SPL in reverberant field | -21 | -21 | -21 | -22 | -24 | -25 | -28 | -31 |
| 15. | Criterion SPL | 60 | 51 | 43 | 37 | 32 | 30 | 28 | 27 |
| 16. | SPL at 10 ft due to fan noise | 63.7 | 65.7 | 67.7 | 67 | 63 | 52 | 43 | 36 |
| 17. | SPL at far field due to fan noise | 63.7 | 65.7 | 63.7 | 67 | 63 | 51 | 43 | 39 |
| 18. | Attenuation required | 3.7 | 14.7 | 24.7 | 30 | 31 | 22 | 15 | 9 |
| 19. | Attenuation of two C-8 package units inparallel. | 22 | 22 | 32 | 39 | 53 | 77 | 63 | - |
| 20 | Attenuation of one A-5 package unit | 7 | 7 | 9 | 12 | 24 | 20 | 32 | - |
| 21 | Attenuation still required. | - | 7.7 | 15.7 | 18 | 7 | - | - | - |
| 22 | Absorption coefficient of lining material($\alpha$) | 0.23 | 0.20 | 0.4 | 0.52 | 0.53 | 0.23 | 0.21 | 0.20 |
| 23 | Value of $\alpha^{1.4}$ | 0.12 | 0.11 | 0.23 | 0.4 | 0.44 | 0.21 | 0.14 | 0.1 |
| 24 | Duct attenuation for 20 ft. lining. | 11 | 16.5 | 25.8 | 38.8 | 40.6 | 20.3 | 12.9 | 9.2 |

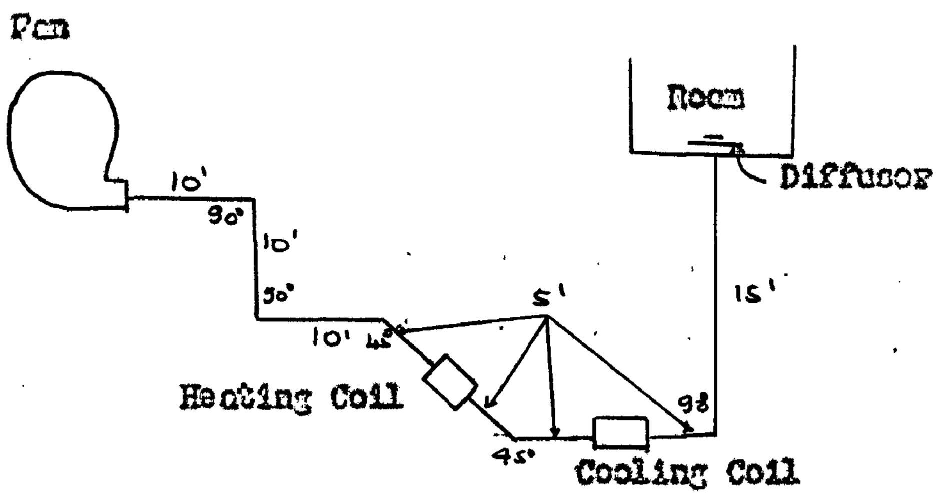
Table - 7.2

| No | Item | Octave frequency bands - cps | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 20<br>75 | 75<br>150 | 150<br>300 | 300<br>600 | 600<br>1200 | 1200<br>2400 | 2400<br>4800 | 4800<br>10000 |
| | **Fan Noise.** | | | | | | | | |
| a) | Relative Band PWL of fan noise db | 0 | -2 | -7 | -12 | -17 | -22 | -27 | -32 |
| | Actual PWL of fan noise | 119 | 117 | 112 | 107 | 102 | 97 | 92 | 87 |
| | **Straight duct attenuation.** | | | | | | | | |
| b) | Attenuation db /ft<br>Bare duct | 0.15 | 0.15 | 0.1 | 0.09 | 0.08 | 0.08 | 0.08 | 0.08 |
| | Insulated duct | 0.30 | 0.30 | 0.2 | 0.06 | 0.06 | 0.06 | 0.06 | 0.06 |
| | **End Reflection loss.** | | | | | | | | |
| | Mean frequency cps | 42 | 106 | 212 | 424 | 848 | 1696 | 3392 | 6784 |
| c) | FL/2000 cps.in. | 0.53 | 1.56 | 3.12 | 6.24 | 12.48 | 24.96 | 49.92 | 99.84 |
| | Endo reflection loss for grille in db | 13 | 7 | 4 | 1 | - | - | - | - |
| | **Directivity Factor.** | | | | | | | | |
| d) | FL/2000 cps-in. | 0.6 | 1.27 | 2.54 | 5.08 | 10.2 | 20.4 | 40.7 | 81.5 |
| | Directivity factor Q | 2 | 2 | 2.2 | 2.7 | 3.2 | 3.6 | 3.9 | 4.1 |
| | **Room constant R.** | | | | | | | | |
| | Correction factor | 0.5 | 0.5 | 0.53 | 0.7 | 1.0 | 1.5 | 3.0 | 6.0 |
| e) | Total room constant R Sq.ft. | 530 | 530 | 562 | 742 | 1060 | 1590 | 3180 | 6360 |
| | Effective room constant $R_0$, Sq.ft. | 106 | 106 | 112 | 160 | 212 | 318 | 636 | 1272 |

7.3 (b) Method of Calculation when there is also noise generation in the equipments:

Determine the acoustical performance of the duct system below :-

The calculations shown here are only for one frequency band and the same procedure will apply for other bands also.




Given : 600-1200 cps band.

PWL of fan re $10^{-13}$ watts, 83 db.
Duct size, 12 in. x 12 in.
Air velocity in duct, 5000 fpm.

| Component | Noise generation level db re 0.0002 $\mu$ b °° | No. flow attenuation |
|---|---|---|
| Duct | - | 0.11 db / ft. |
| 90° Elbow | 82 db°° | 3.5 db |
| 45° Elbow | 79 db | 1.0 db |
| Heating coil | 67 db° | 1.5 db |
| Cooling coil | 67 db° | 3.0 db |
| Diffuser | 69 db° | 10 db |

°° based on 1sq.ft. area.

° assuming that the free area of the coils and diffuser is greater than the duct, and hence the velocity much less.

Room constant, $R_0$ is 100 sq.ft.

Find the sound level in the room.

Starting with the fan, the power level input to the first section would be 83 db. Since the analysis will be made on the power level basis, the sound pressure levels are converted by equation.

$PWL = SPL + 10 \log S$ where $S$ = area of cross section. Moreover, since $S$ = 1 sq.ft. (Duct is 12 in. x 12 in.) in this example the power levels are numerically equal to the pressure levels. The results are tabulated in Table 7.4 and the procedure is also indicated there in.

The output power level, $PWL_o$, from the diffuser equals 73. With the room constant of 100 sq.ft. the sound level at 6 ft. from the diffuser would be 73 - 15 = 58 db.

If only the attenuation values as determined under no flow conditions, has been considered, the room level would have been estimated at 39.1 db, which is low. Any attempt to reduce the final level in the room by acoustical treatment of ducts preceding the diffuser section would be limited by the noise generation level of the diffuser.

In Table - 7.4,

$PWL_i$ = input power level.

$SPL_a$ = attenuation (db)

$PWL_g$ = noise generation level.

$PWL_o$ = output power level.

## Table - 7.4

Summary of calculations for example 7.2 (b) given.

| Duct component | $PWL_i$ | $SPL_a$ | Level | $PWL_c$ | $PWL_o$ | Remarks |
|---|---|---|---|---|---|---|
| | Attenuated | | | | | |
| Fan | | | | | 88 | |
| Duct | 88 | 1.1 | 86.9 | - | 86.9 | Atten. in 10 ft. |
| 90° Elbow | 86.9 | 3.5 | 83.4 | 82 | 85.7 | $SPL_o$ = addition of 83.4 & 82. |
| Duct | 85.7 | 1.1 | 84.6 | - | 84.6 | Atten. in 10 ft. |
| 90° Elbow | 84.6 | 3.5 | 81.1 | 82 | 84.5 | |
| Duct | 84.5 | 1.1 | 83.4 | - | 83.4 | Attend. in 10 ft. |
| 45° Elbow. | 83.4 | 1.0 | 82.4 | 79 | 84.0 | |
| Duct | 84 | 0.5 | 83.5 | - | 83.5 | Atten. in 5 ft. |
| Htg. coil | 83.5 | 1.5 | 82 | 67 | 82 | |
| Duct | 82 | 0.5 | 81.5 | - | 81.5 | Atten. in 5 ft. |
| 45° Elbow | 81.5 | 1.0 | 80.5 | 79 | 82.8 | |
| Duct | 82.8 | 0.5 | 82.3 | - | 82.3 | Atten. in 5 ft. |
| Cooling coil | 82.3 | 3.0 | 79.3 | 67 | 79.5 | |
| Duct | 79.5 | 0.5 | 79 | - | 79 | Atten. in 5 ft. |
| 90° Elbow | 79 | 3.5 | 75.5 | 82 | 82.8 | |
| Duct | 82.8 | 1.6 | 81.2 | - | 81.2 | Atten. in 15 ft. |
| Diffuser | 81.2 | 10 | 71.2 | 69 | 73 | |

Total 33.9 db

Room level without consideration of noise generation

= 88 - 33.9 - 15 = 39.1 db.

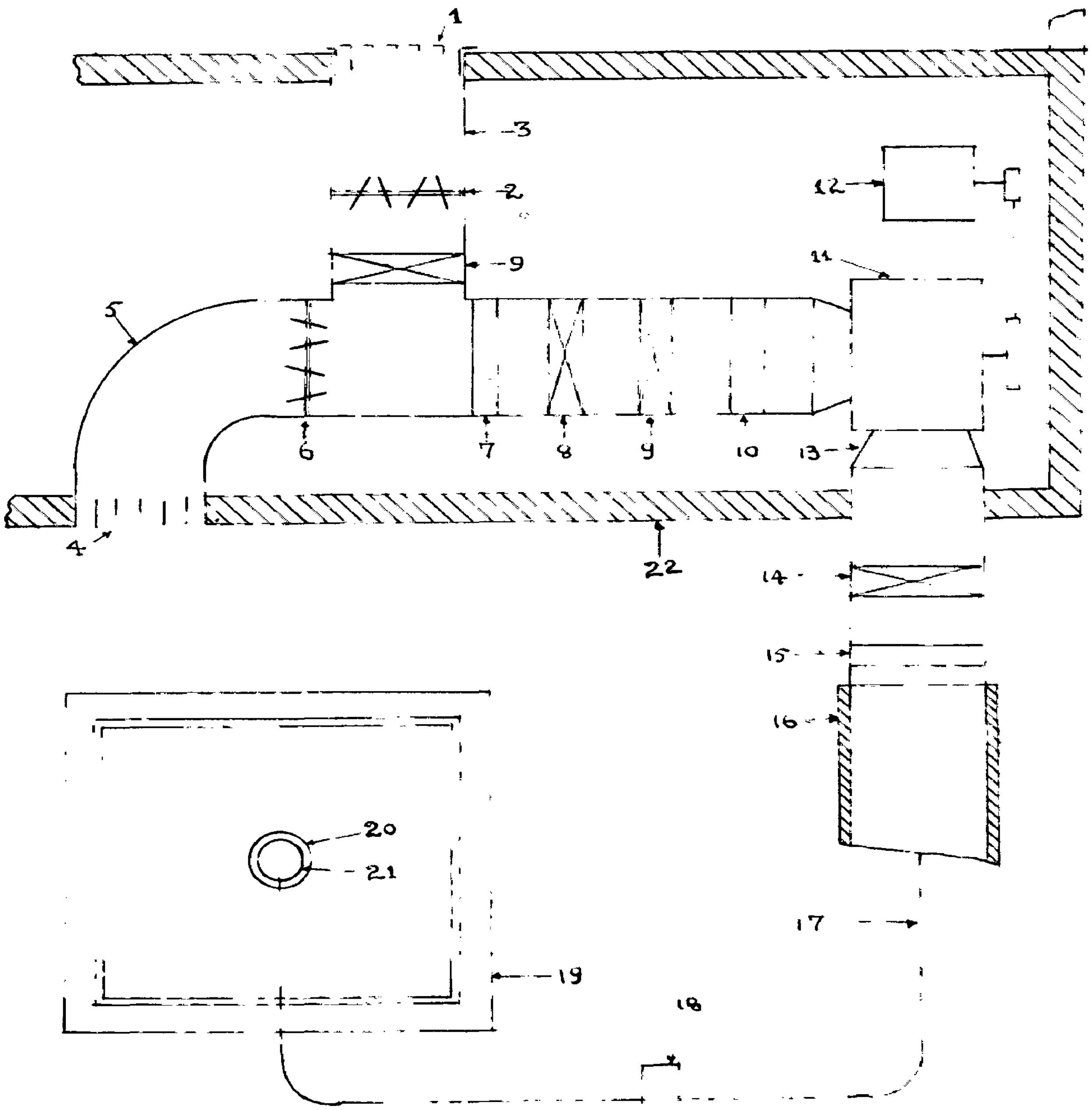
## CHAPTER - VIII.

### CONCLUDING REMARKS :

The treatment and control of noise in air conditioning systems is usually by trial and error. This design is not amenable to handbook solutions since the assessment ofthe effects of noise on man is still not fully understood. This is due to the fact that human beings react differently to noise and the problem is essentially psychoacoustical.

The data available regarding the noise due to mechanical equipments like fans, compressors and the like, is not complete. Noise spectra generated by components like cooling and heating coils, elbows, diffusers, are to be investigated for the proper selection of noise control devices.

While many of the criteria, data and the like, are available for easy reference to the engineer, he has to make full use of his discretion in arriving at the proper solution in any typical situation. Such a solution is possible only when there is mutual co-operation among the air-conditioning engineer, architect and the acoustician.



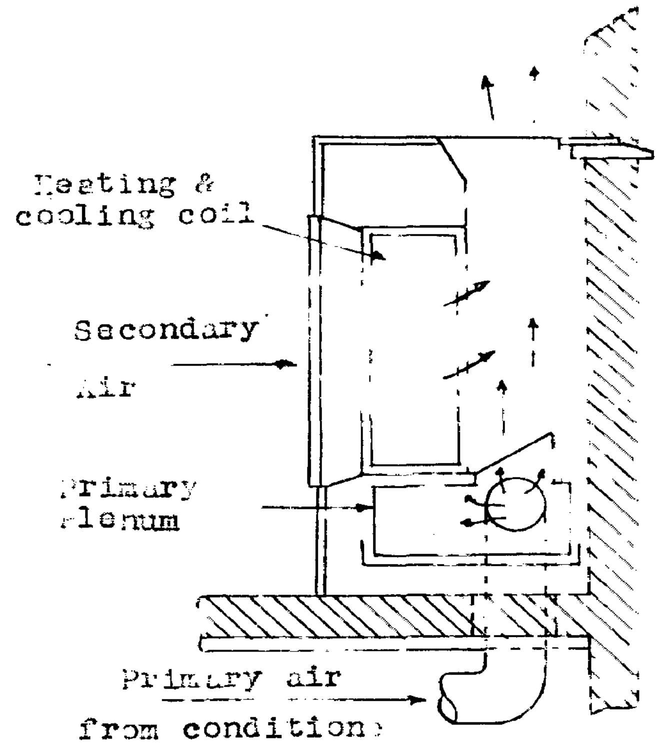
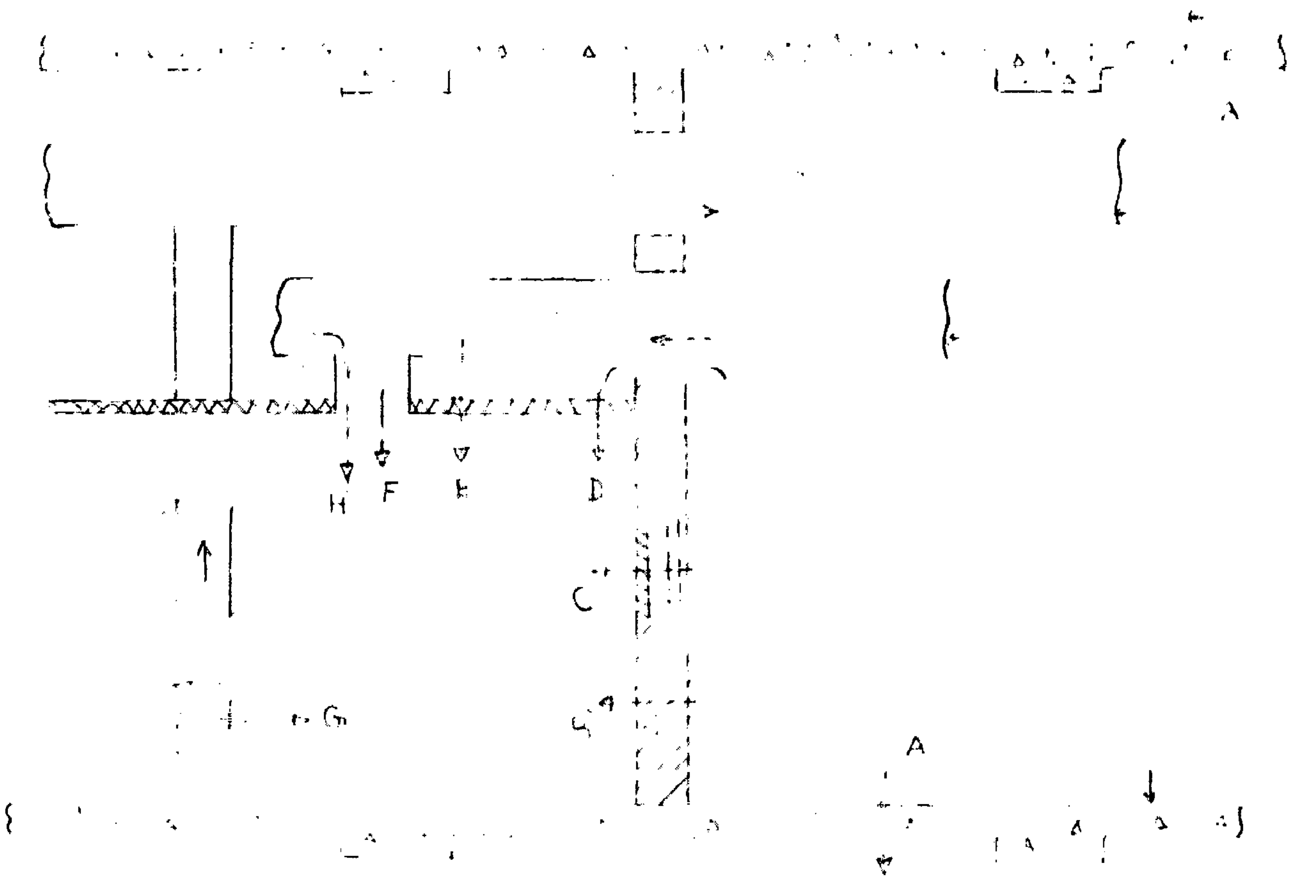
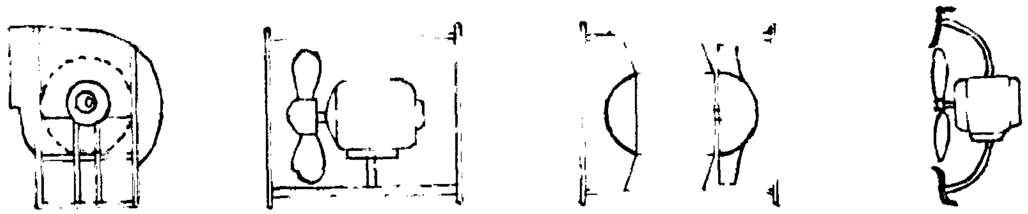




Fig. 2.2 Indu
(High-Pressur

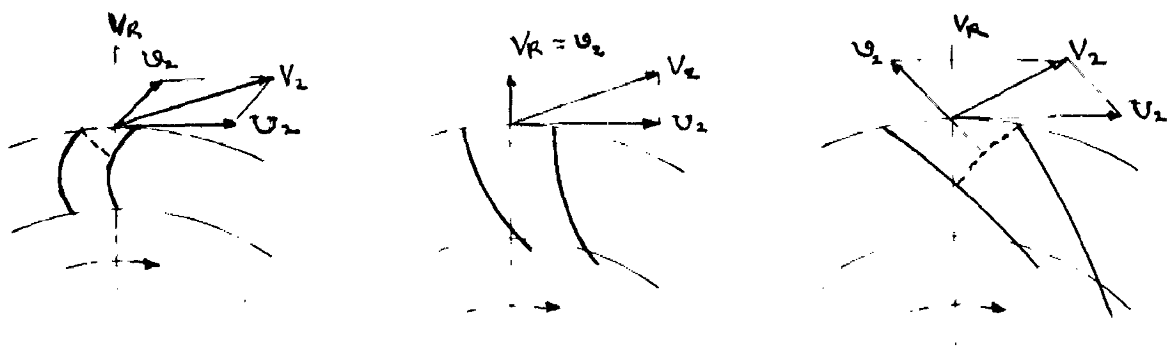

A
H
F
E
D
C
G
A



Centrifugal fan. Tubeaxial Vaneaxial Propeller fan.

Axial flow fans.

Fig. 3.2 Different types of fans.




Forward Radial Backward

$U_2$ = Velocity of wheel tip

$V_2$ = Absolute velocity

$v_2$ = Velocity relative to blade.

$V_R$ = Radial velocity.

Fig. 3.3 Blade shapes and velocity diagrams for different types of fan blades.

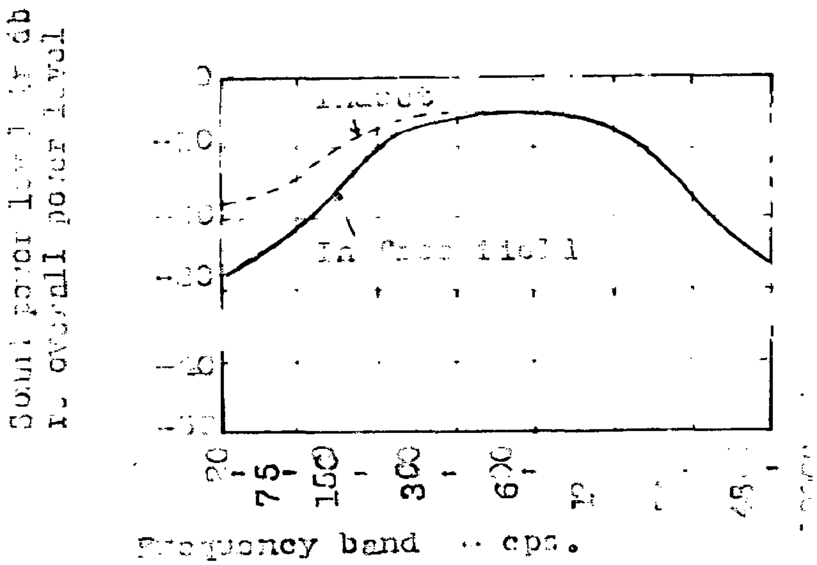
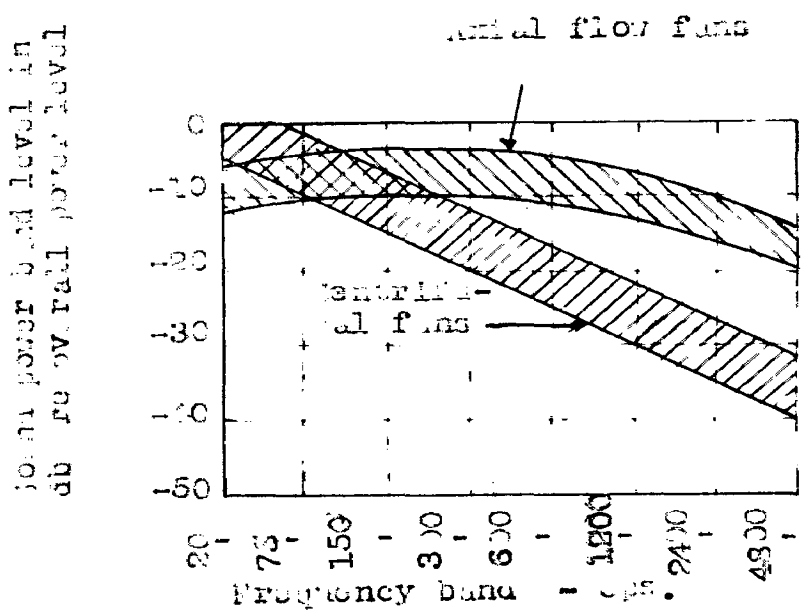



Fig. 3.7 Sound power spectrum [illegible]
single stage axial fans




Over all power level (all bands) [illegible]
= [illegible] + 10 log hp + 10 log P [illegible]
re $10^{-13}$ watts.

Fig. 3.6 Typical fan spectra f[illegible]

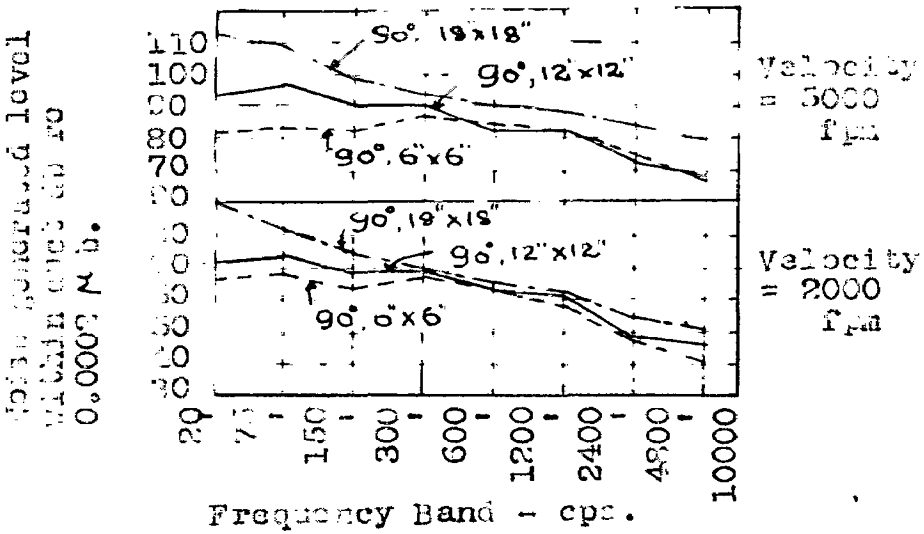
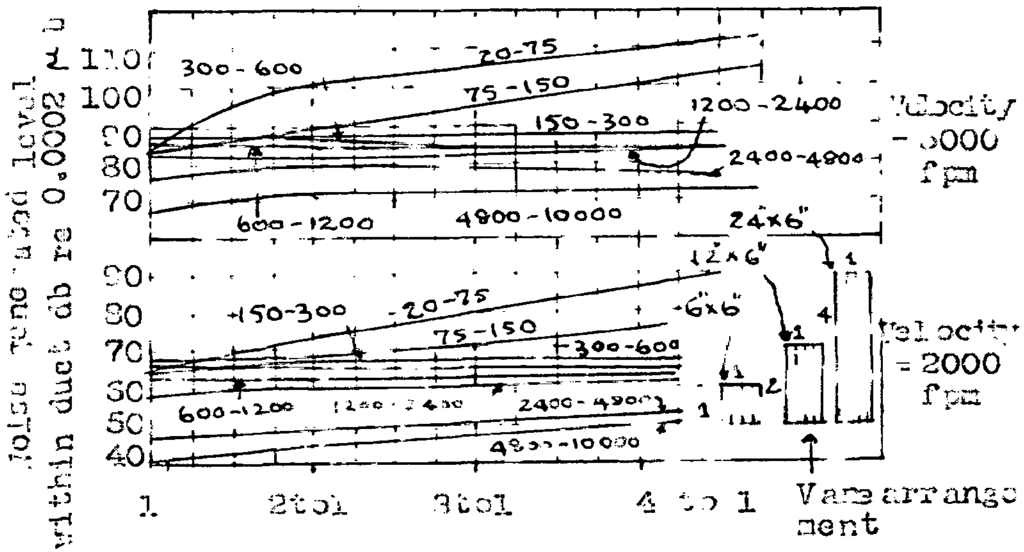
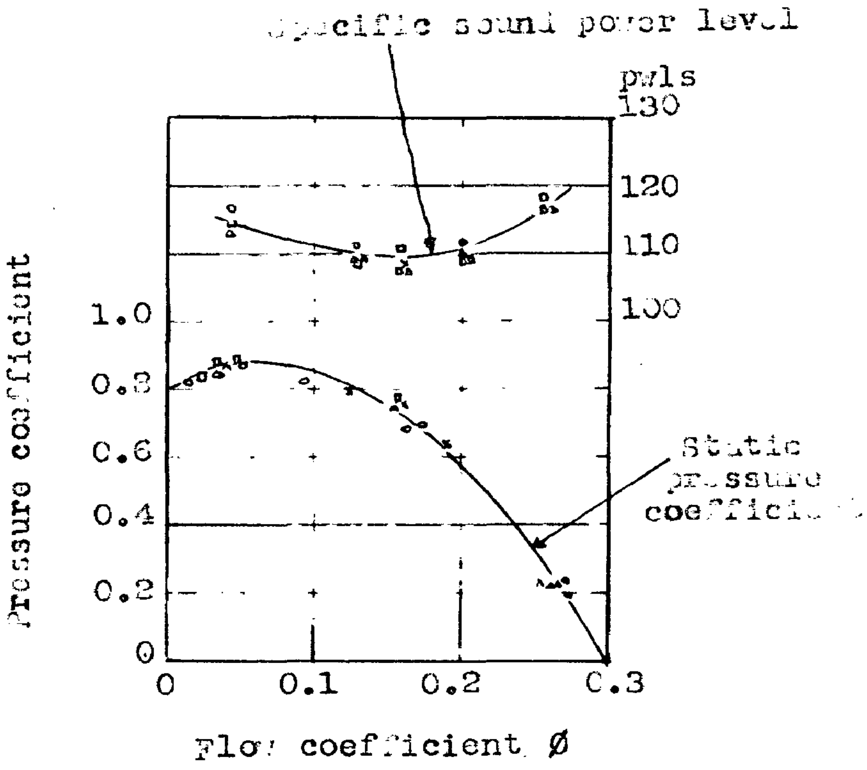



Fig. 3.9 Noise generation level by octave band analysis for three elbow sizes.




Aspect ratio (vane arrangement)
3.10 ... level vs aspect ratio.




0 670 rpm ▲ 500 rpm

◘ 600 rpm x 390 rpm

Fig. 3.8 Experimental values of static pressure and overall power level for a 27 in., diameter centrifugal fan with backward curved blades replotted in terms of and PWL against Ø (19)

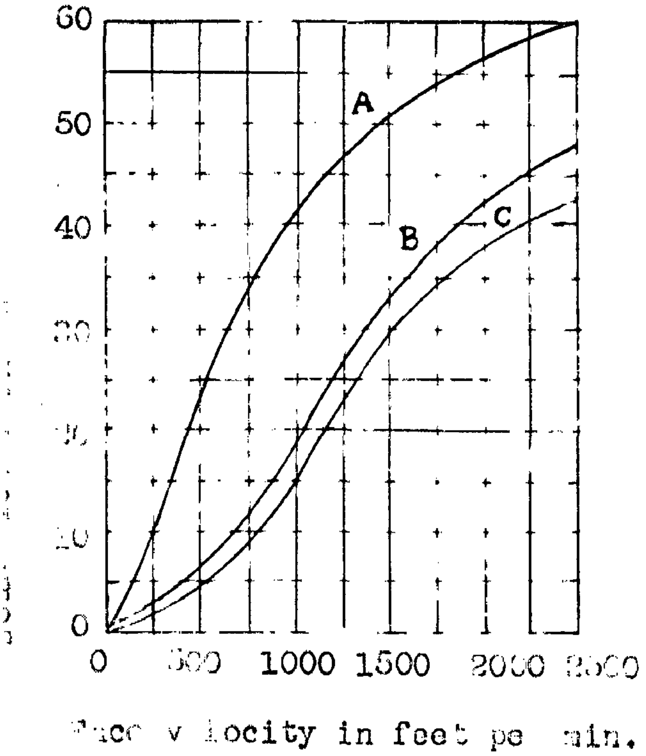



Fig. 3.11 Noise level near grille vs face velocity of air stream.

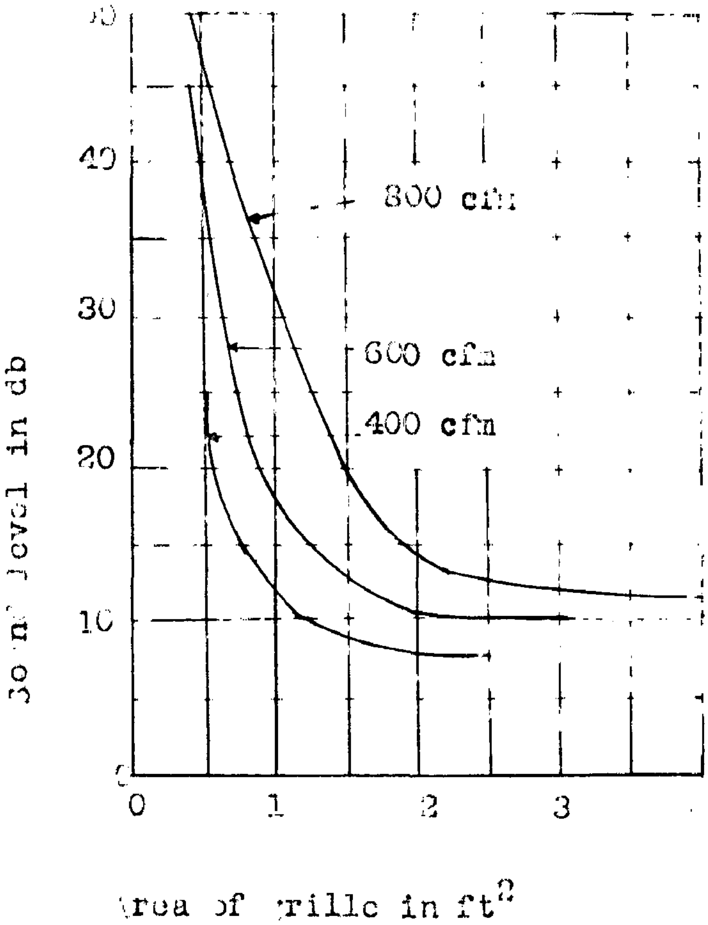
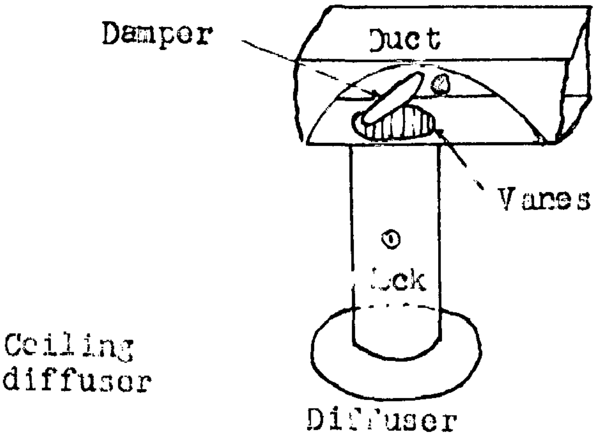
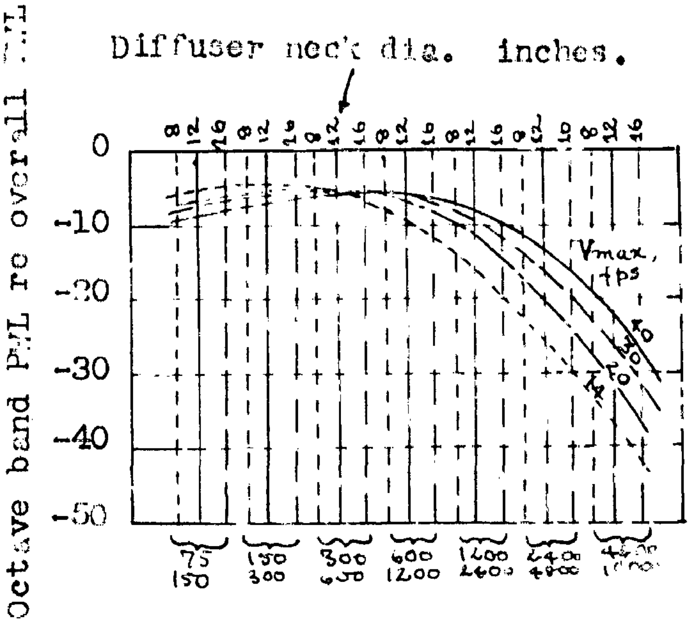



Fig. 3.12 Noise level at a grille vs area for constant volume of flow.

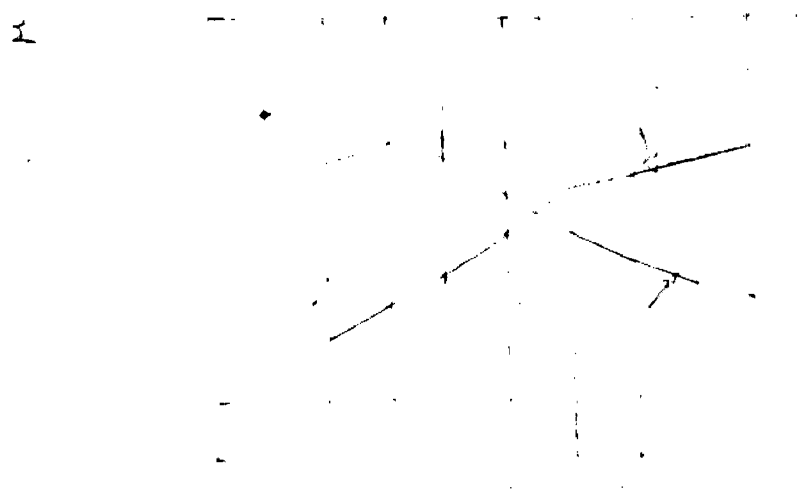


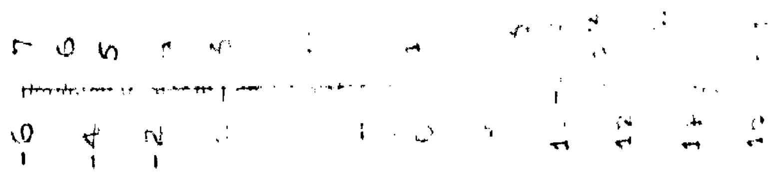

For wide open damper; min. area at (1)
For damper ... ; min area at (2)

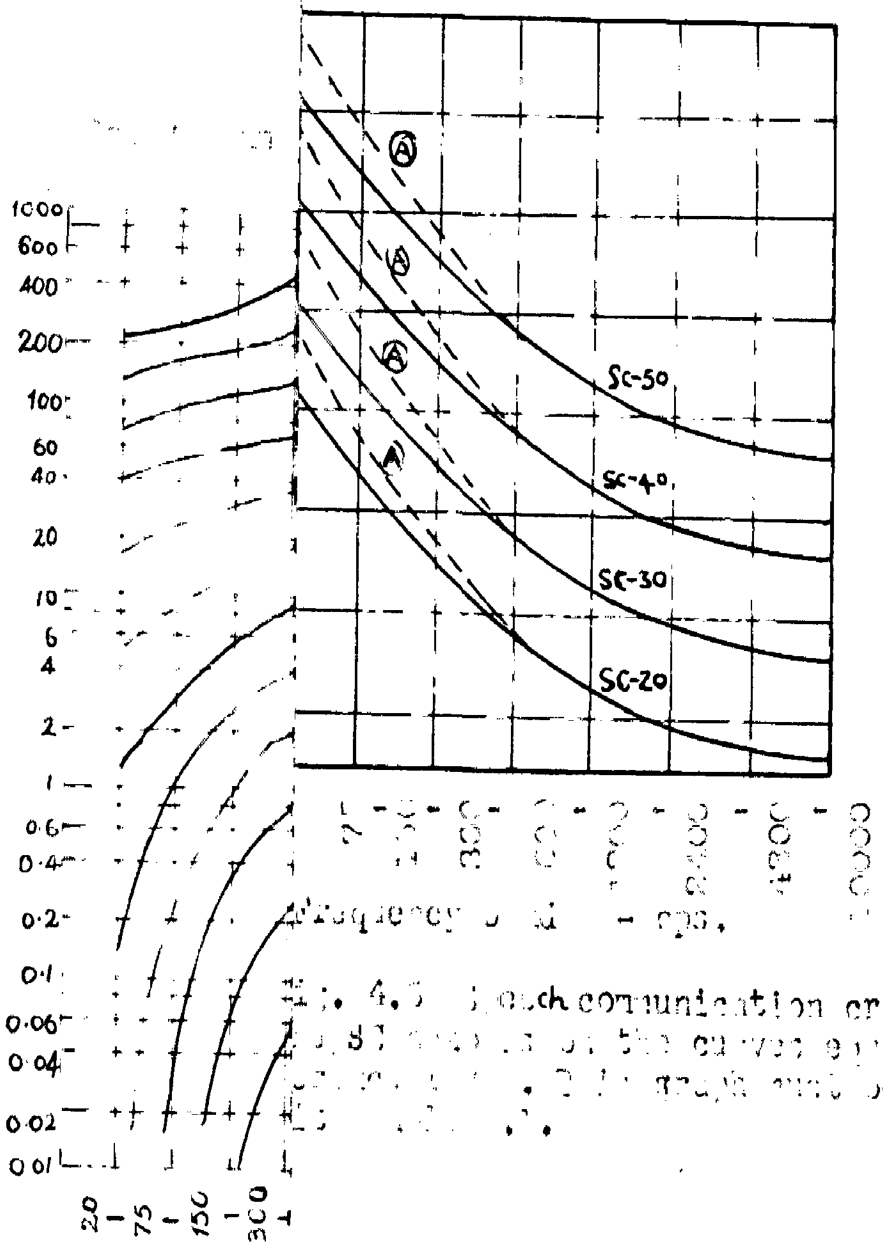



Frequency band -- cps.

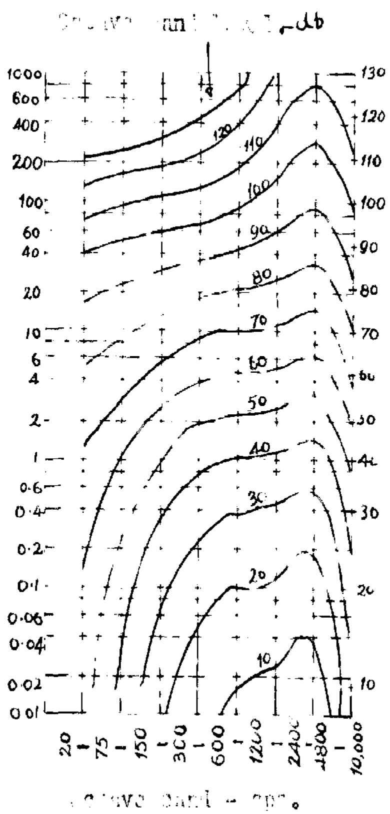
Fig. 3.13 Octave band spectrum of ceiling diffusers.



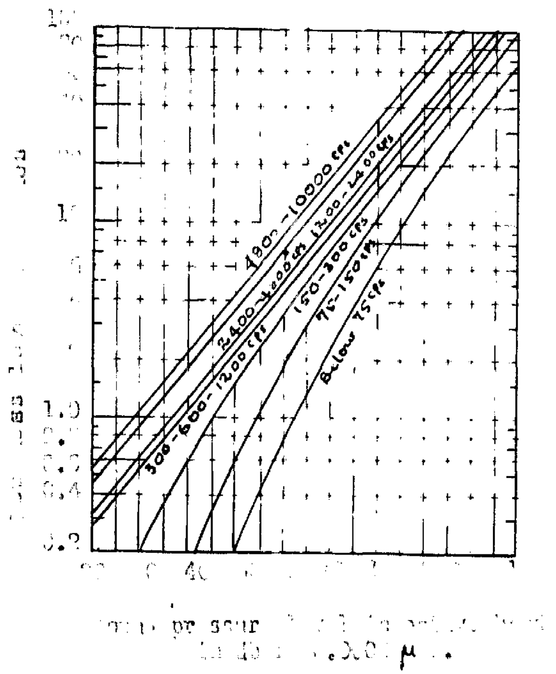





[illegible] 4.5 [illegible]ch communication criteria for ro[illegible]s.
[illegible] curves [illegible] to the speech in [illegible]
[illegible] graph [illegible] used in con[illegible]
[illegible]




... chart for converting ...
... pressure ... decibels




... pressure ...

Fig. 4. ...

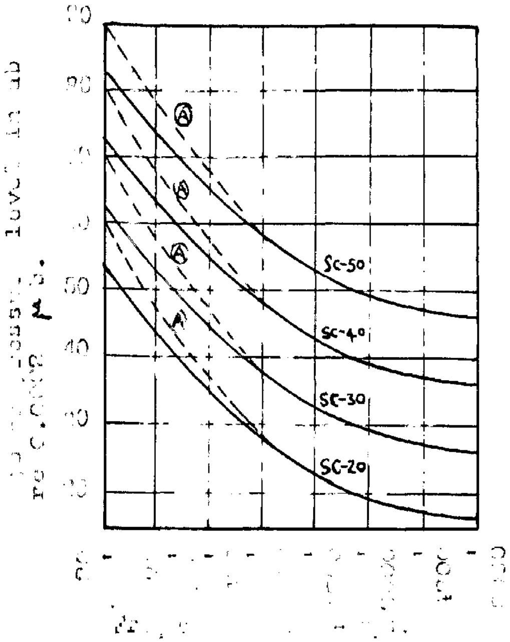
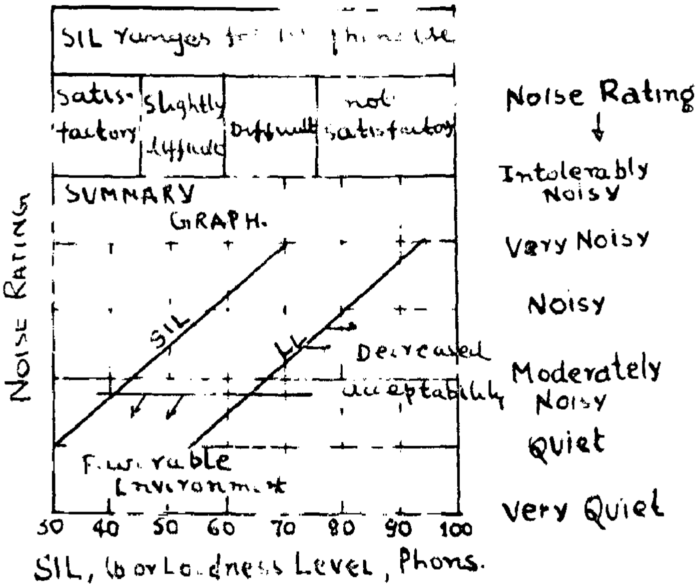
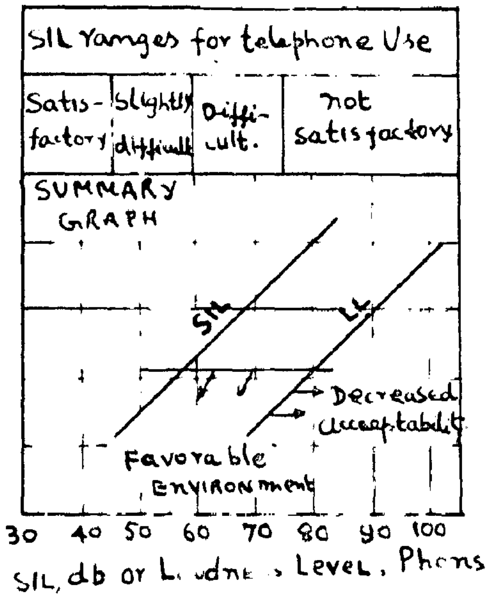
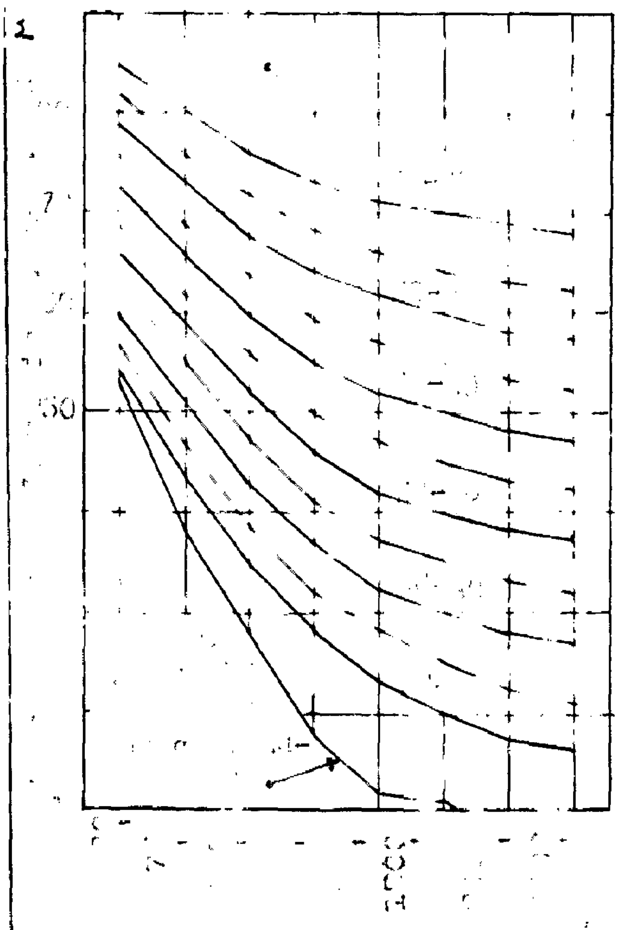
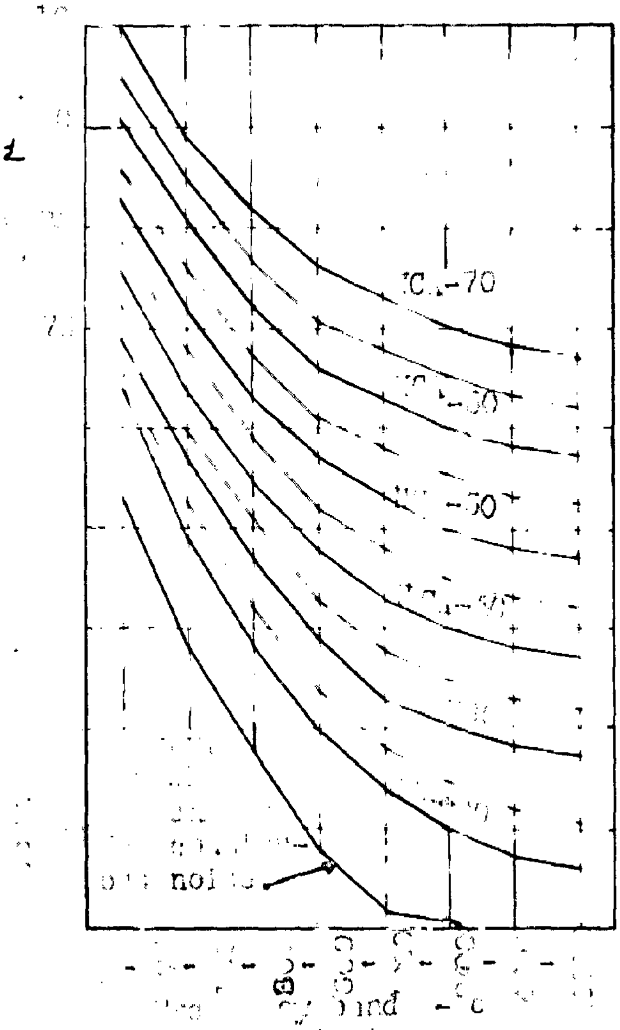



Fig. 5. ... communication criteria ...


SIL ranges for telephone use
Satisfactory
Slightly difficult
Difficult
not satisfactory
SUMMARY GRAPH
Noise Rating
SIL
LL
Decreased acceptability
Favorable environment
30 40 50 60 70 80 90 100
SIL, db or Loudness Level, Phons.
Noise Rating
Intolerably Noisy
Very Noisy
Noisy
Moderately Noisy
Quiet
Very Quiet


SIL ranges for telephone use
Satisfactory
Slightly difficult
Difficult.
not satisfactory
SUMMARY GRAPH
SIL
LL
Decreased acceptability
Favorable environment
30 40 50 60 70 80 90 100
SIL, db or Loudness Level, Phons





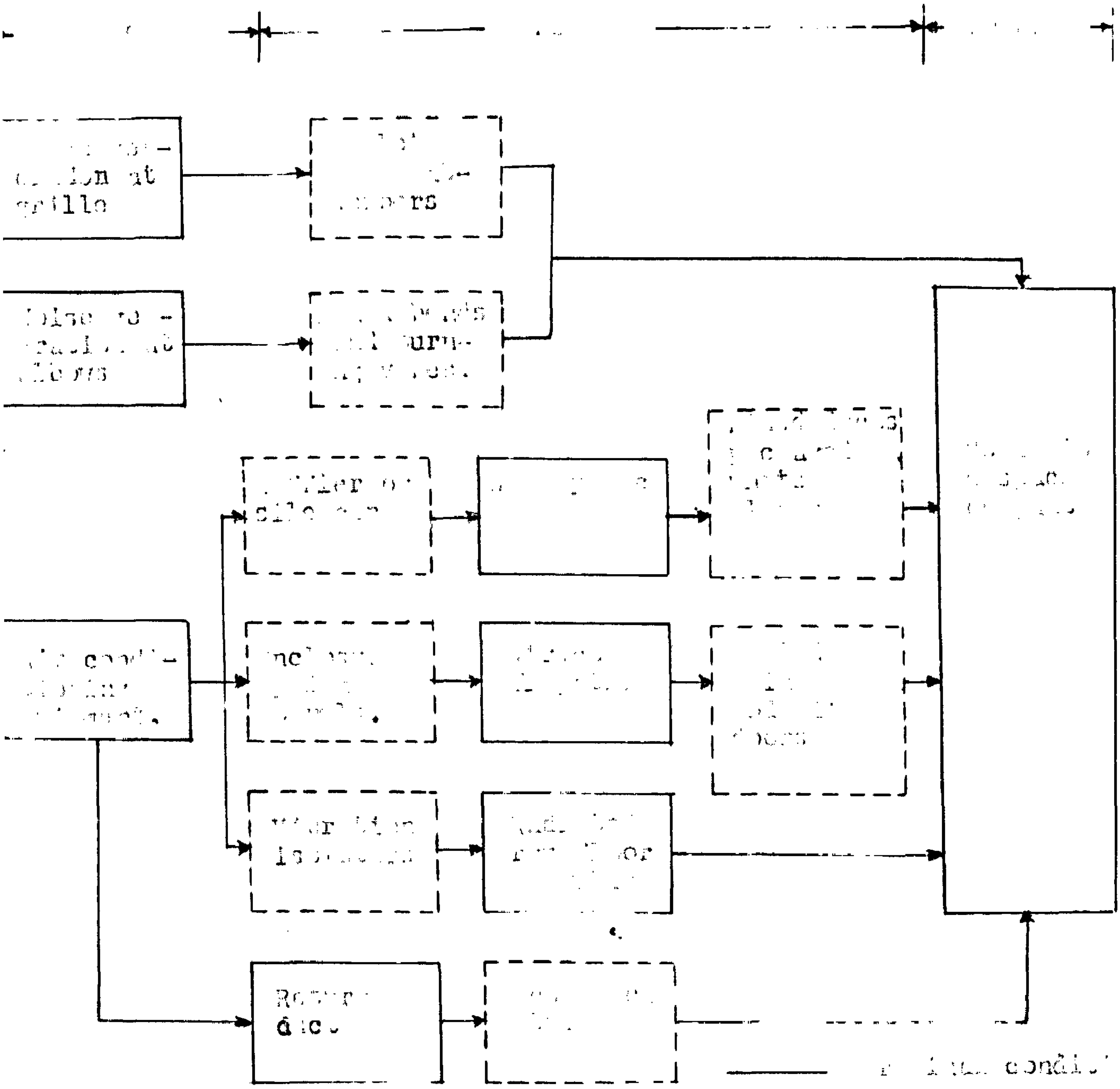
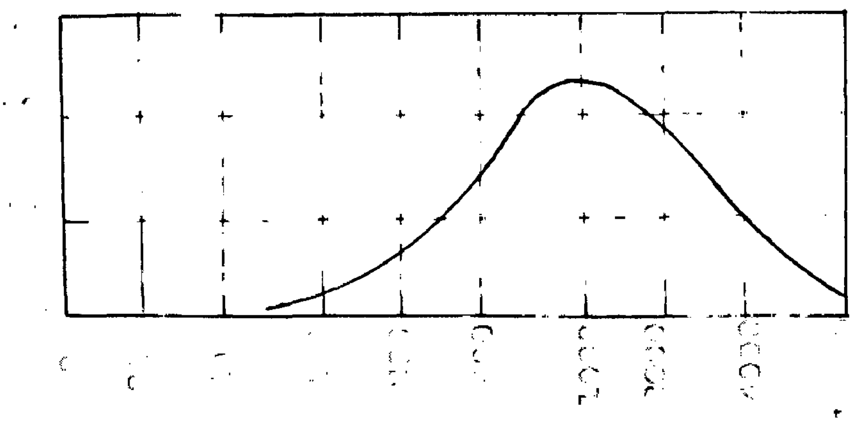
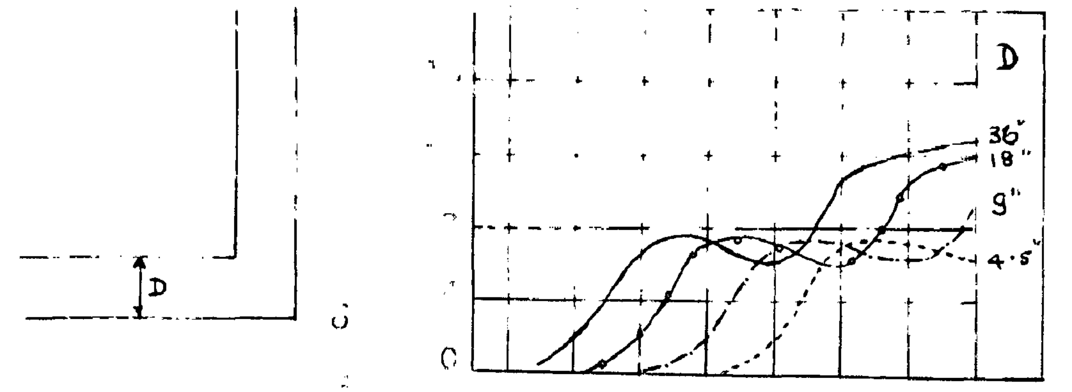


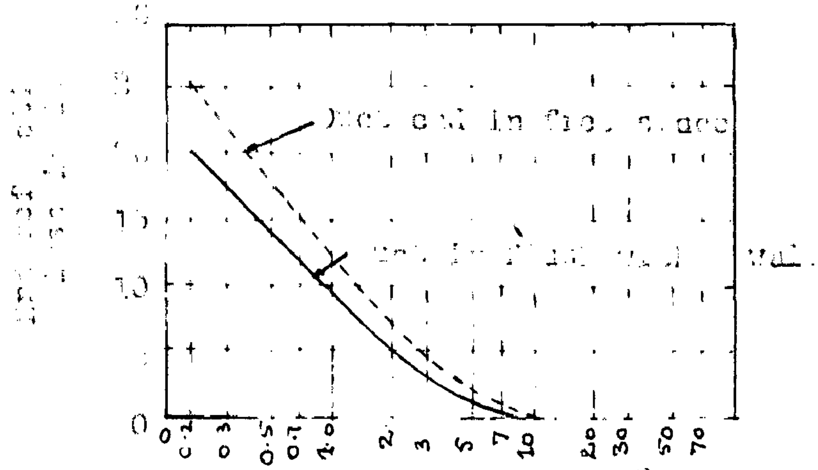

Fig. 5.1 Schematic [illegible] of [illegible] control system [illegible].





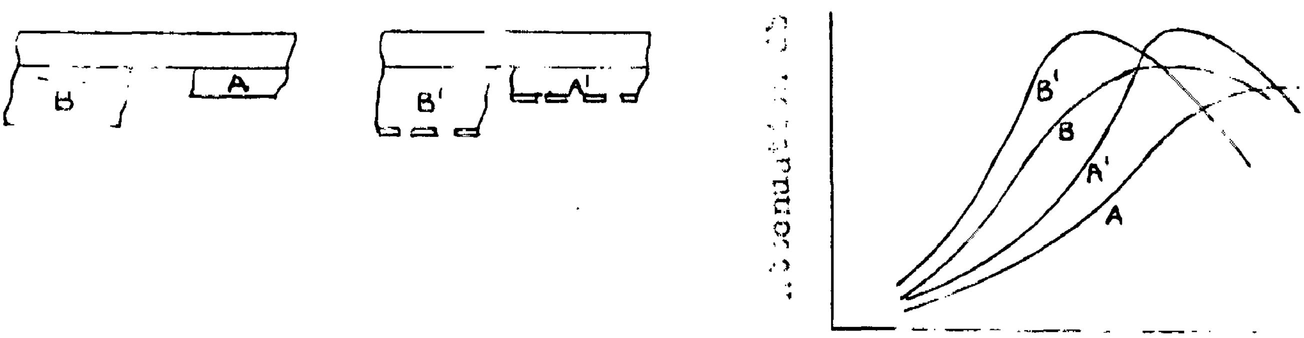

Frequency band – cps.

Fig. 5.7 [illegible] in decibels provided by [illegible] in a duct [illegible] (54)



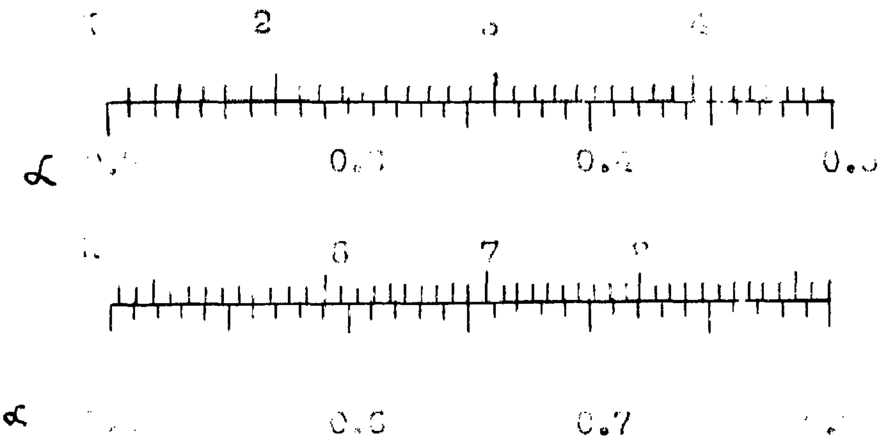

Frequency x length x $10^{-3}$ = [illegible]/1000 in cps.inches

Fig. [illegible] in decibels at the open [illegible] of area L

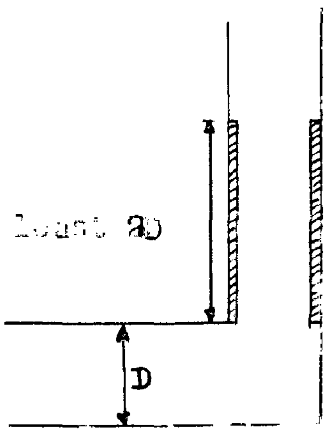



Frequency – cps.

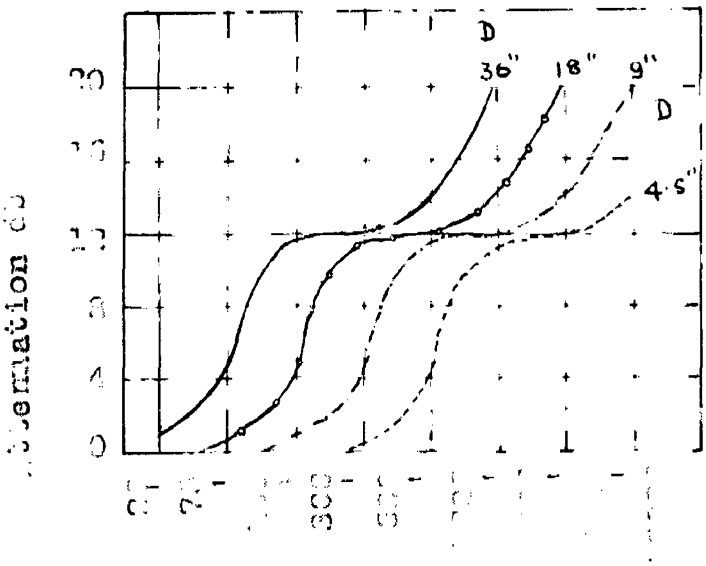
[illegible] Two types of sound absorbers and their frequency characteristics. [illegible]




[illegible] (in [illegible]) [illegible] function
of [illegible] (C2)

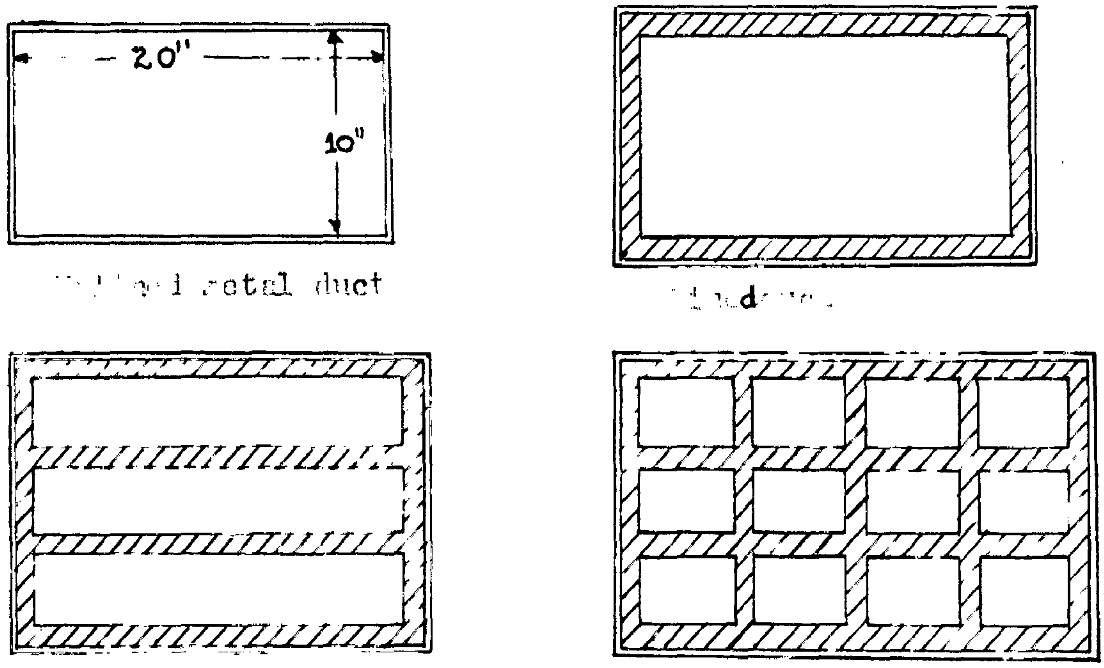



[illegible] -line [illegible] on




[illegible] (4)

The open area is same for each case shown
below for 1" thick absorption material



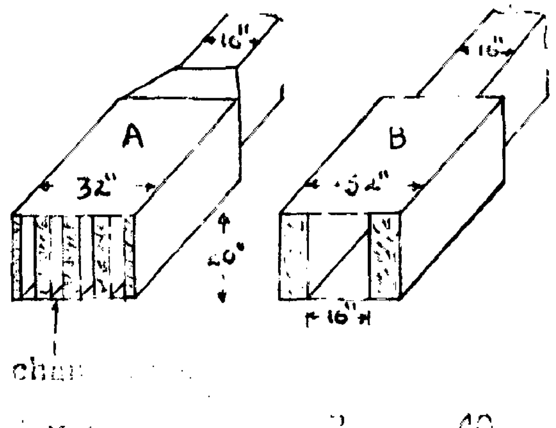

metal duct

lined duct

splitter plate type absorber

cell type absorber.

Fig. Acoustic treatment of ducts.

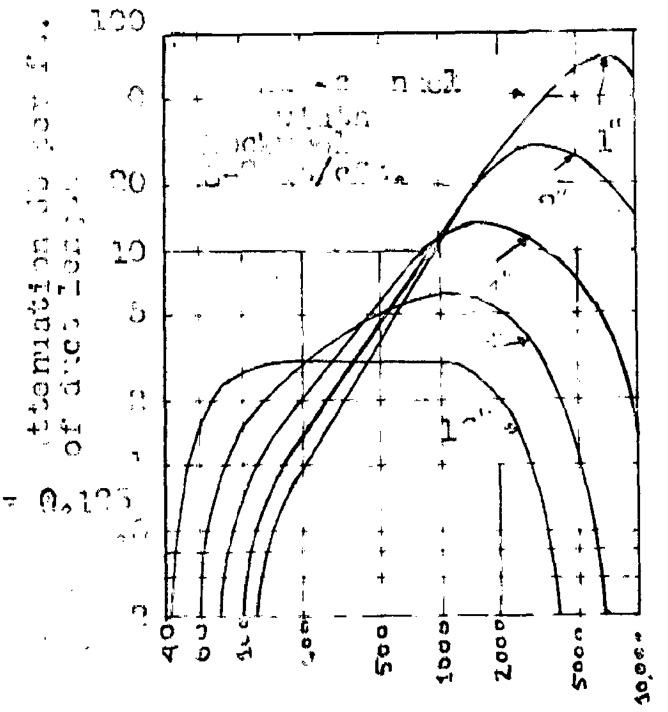



chan

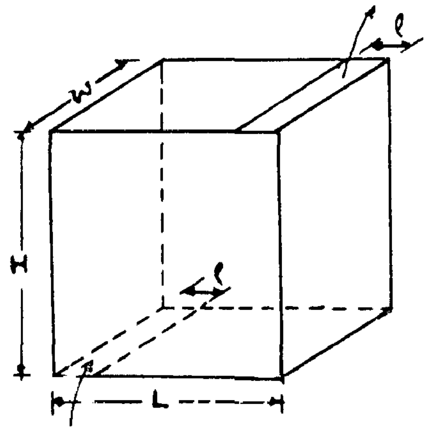
$$\frac{P}{A} = \frac{40}{320} = 0.125$$

sound

duct split




material (.8)




...etry of a single

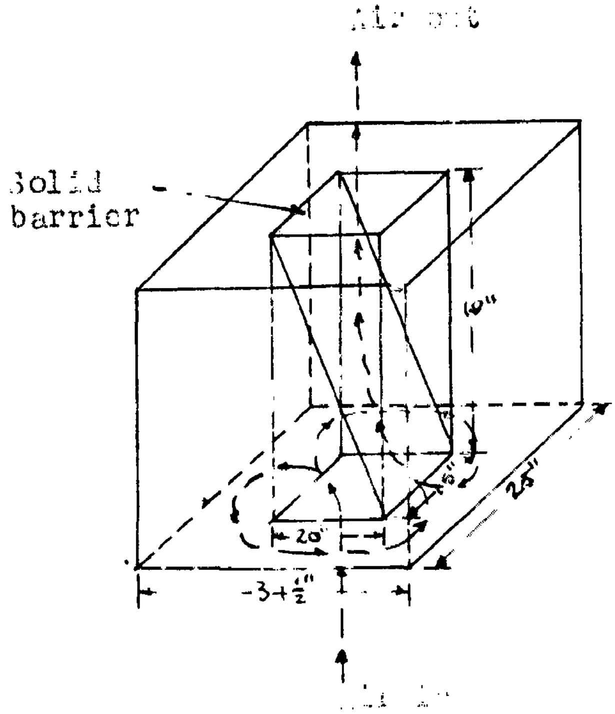
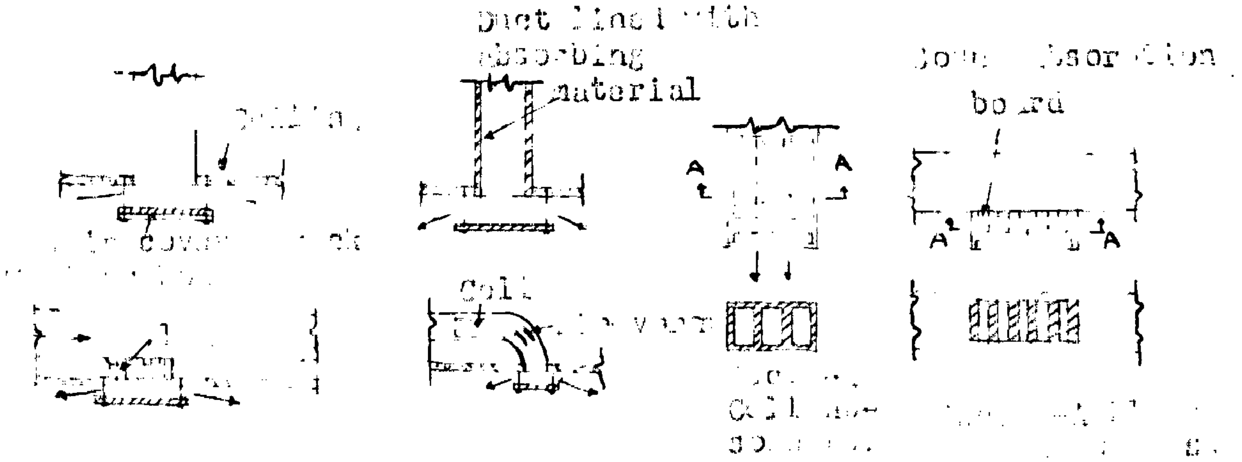
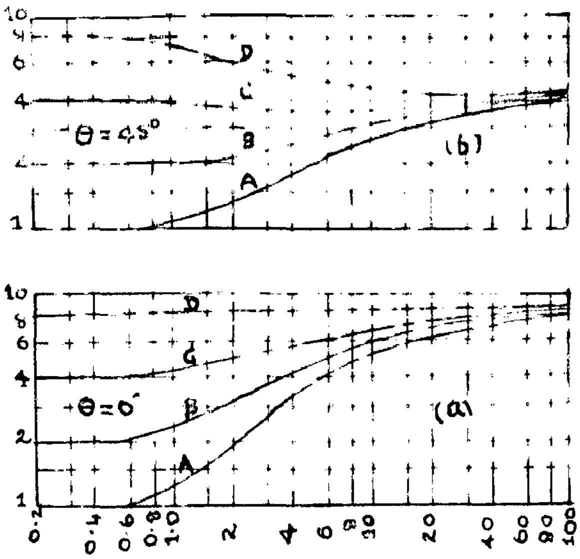



Fig. ... Sketch of a high attenuation plenum. Note there is no direct path from ...




... for ...




Frequency ... x 10^-3
= ... L/1000 ... inches.

Grille at

A - Centre of room

B - Centre of wall

C - Centre of edge

D - Room corner.

a) For a position straight in front of outlet.

b) For a position 45 deg. to side of outlet.

Fig. 5.14 Directivity factor for use in Fig. 5.1...

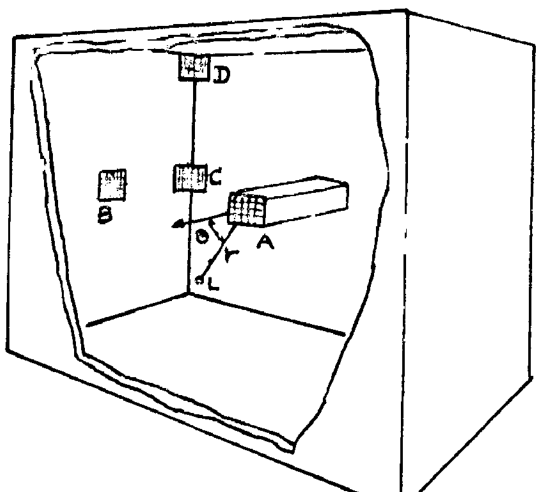


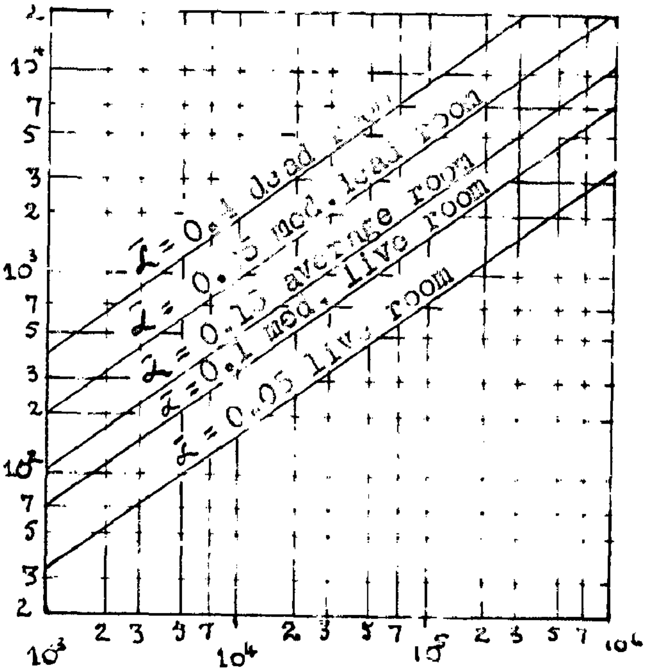

Fig. 5.15 Four typical means of terminating a ventilating duct in a room.

A - Centre of room

B - Centre of wall

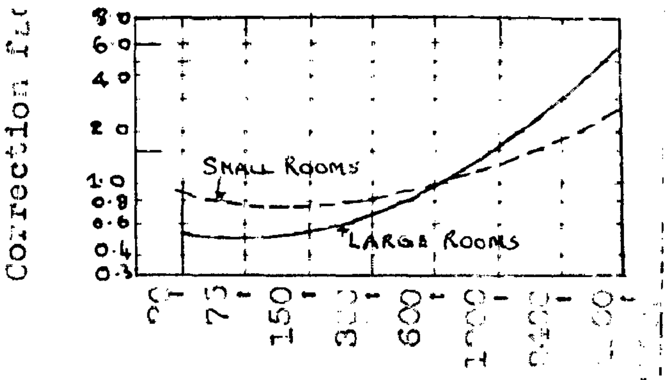
C - Centre of edge.

D - At corner.



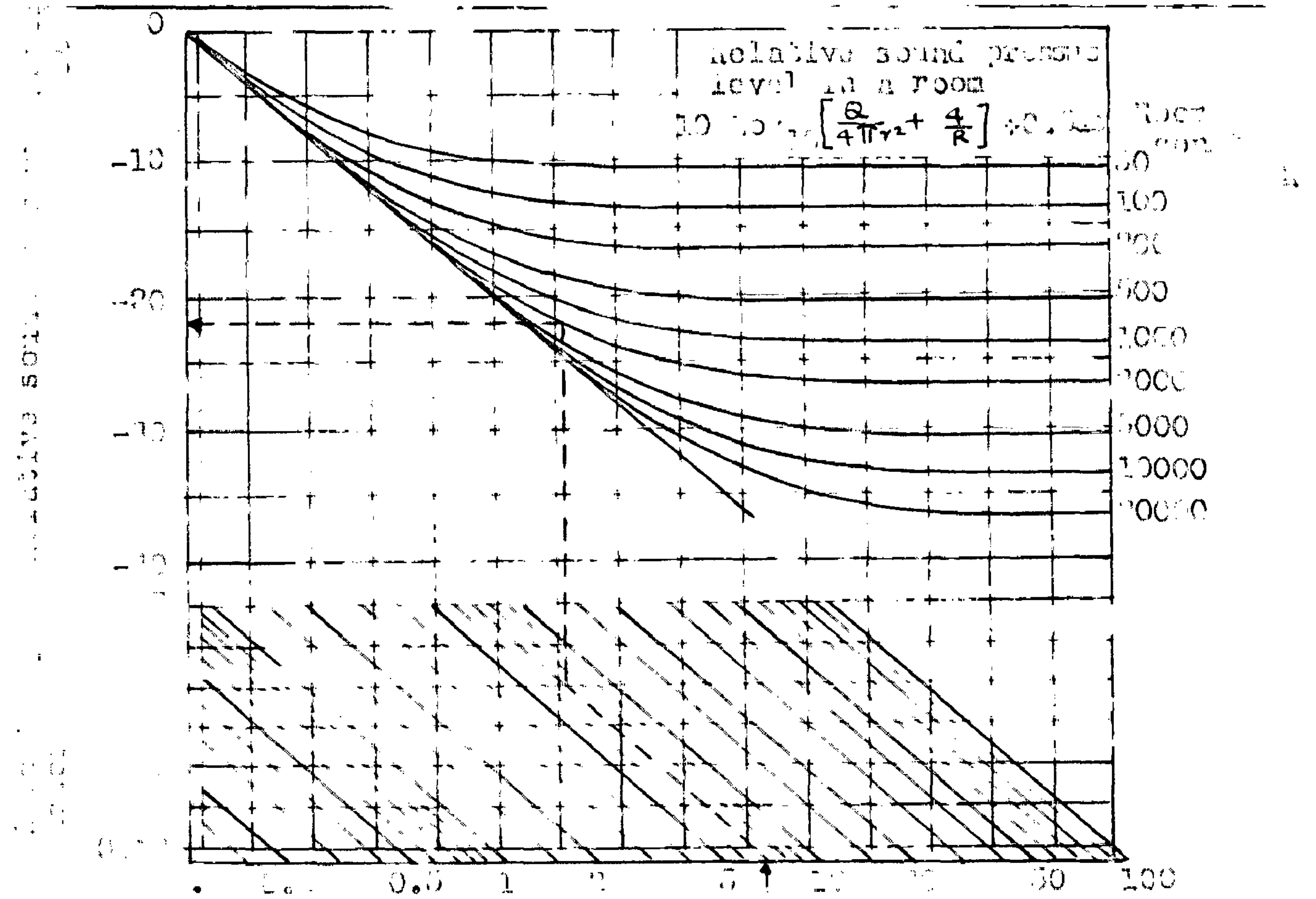

Room Volume V ft$^3$.

Fig. 5.13 Value of room constant as a function of volume of room V.




Octave band – c/s.

Correction factor for room absorption at various frequency bands




Distance r from source ft.

Curves for determining the SPL in a room relative to (S)

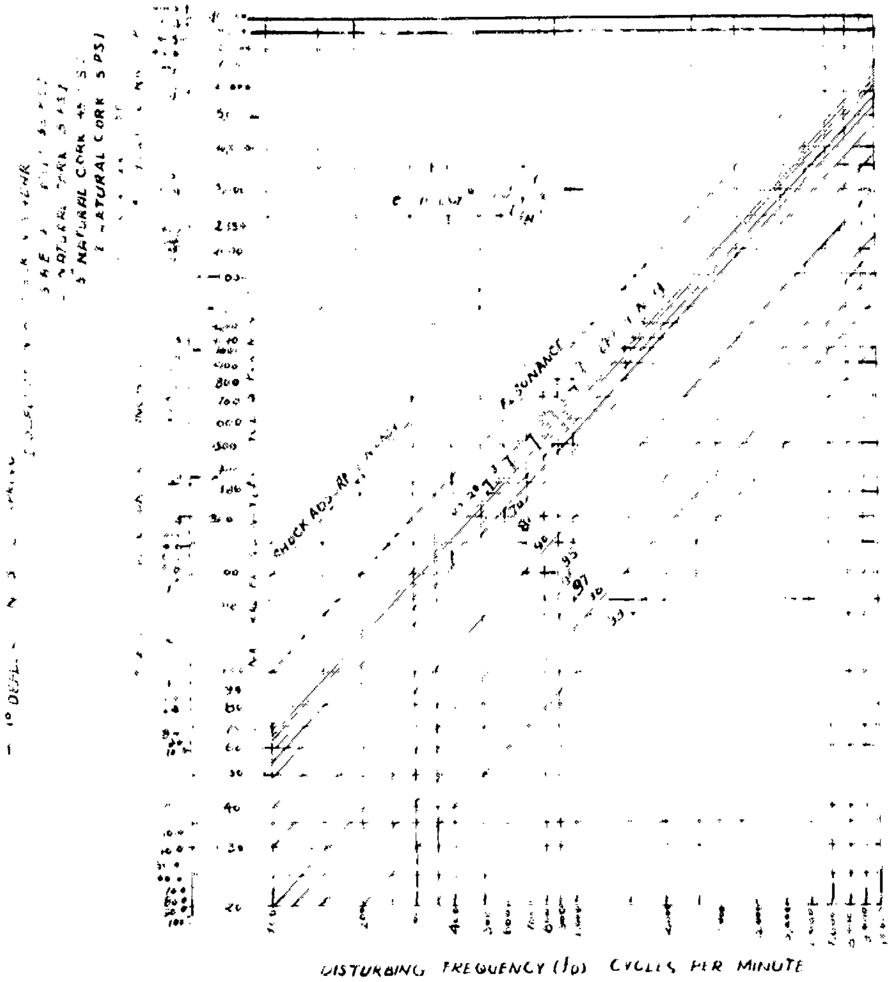
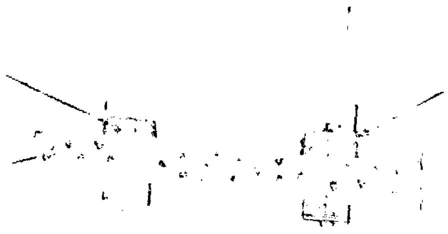
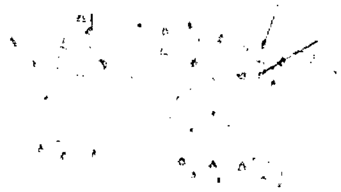
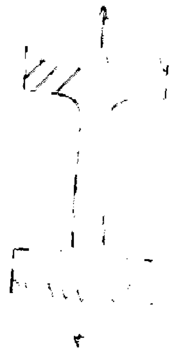
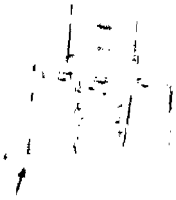
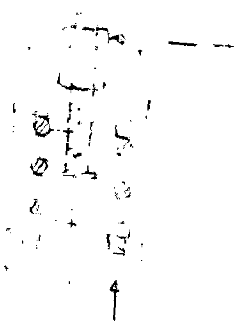


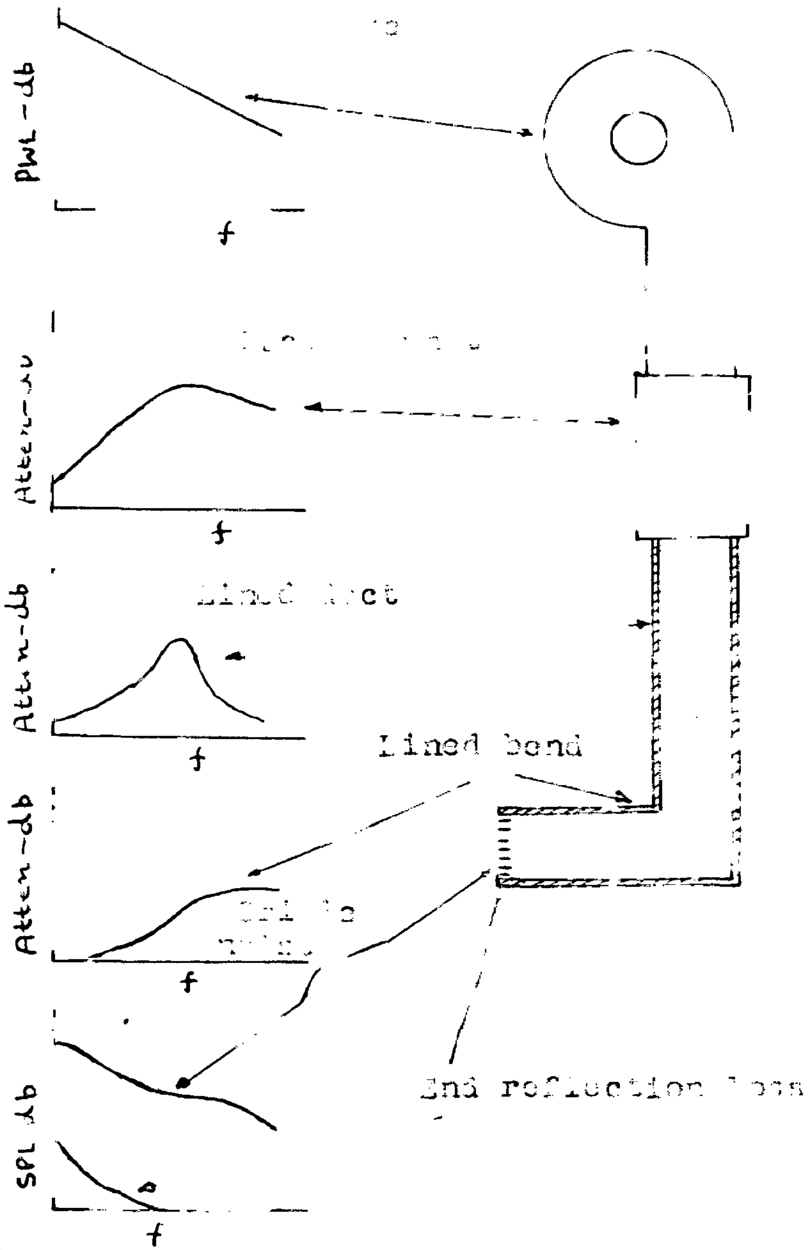

Fig. 6.? Chart for selecting the proper resilient mounting.

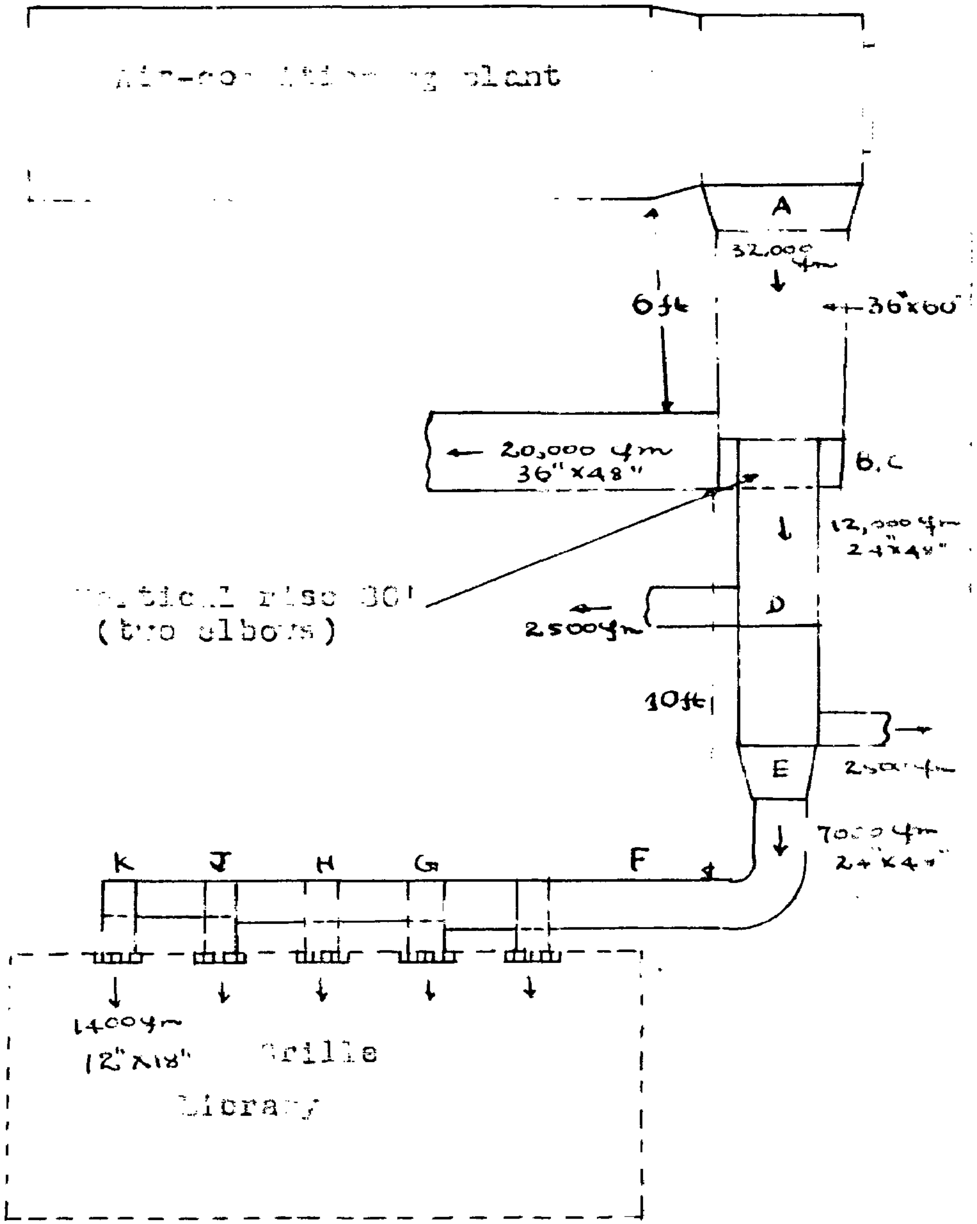
Enter the chart at the disturbing frequency [illegible] move vertically up [illegible] the desired transmissibility is reached and then move horizontally left to obtain the static deflection. Suitable material for mounting is also indicated. (77)


PWL - db
f
Atten-db
f
Lined duct
Atten-db
f
Lined bend
Atten-db
f
SPL db
f
End reflection loss


A
32,000 cfm
36"x60"
6 ft
20,000 cfm
36"x48"
B,C
12,000 cfm
24"x48"
Vertical rise 30'
(two elbows)
D
2500 cfm
10 ft
E
7000 cfm
24"x48"
F
K
J
H
G
1400 cfm
12"x18"
Grille
Library

## APPENDIX

### ACOUSTICAL TERMINOLOGY :-

Many of these definitions are quotations from American Standard Terminology ($Z$ 24.1 - 1951), American Standards Association.

### ABSORPTION COEFFICIENT (Acoustical Absorptivity), $\alpha$ :-

The sound absorption coefficient of a surface which is exposed to a sound field is the ratio of the sound energy absorbed by the surface to the sound energy incident upon the surface. The absorption coefficient is a function of both angle of incidence and frequency. Tables of absorption coefficient which are given in the literature usually list the absorption coefficients at various frequencies, the values being those obtained by averaging overall angles of incidence.

### ATTENUATION :-

The attenuation of a duct element is the difference between the sound power levels entering and continuing after the element.

$$\text{Attenuation} = PWL_e - PWL_c$$

where, $PWL_e$ = sound power level entering, db

$PWL_c$ = sound power level continuing, db

### BAND PRESSURE LEVEL :-

The band pressure level of a sound for a specified frequency band is the effective sound pressure level for the sound energy contained within the band. The width of band and the reference pressure must be specified. The width of the band may be indicated by the use of a qualifying adjective, e.g., octave band (sound pressure) level, half-octave band level, third-octave band level, 50 cps band level.

CYCLE :- A cycle is the complete sequence of values of a periodic quantity which occur during a period.

CYCLE PER SECOND (cps):- A unit of frequency. In many European countries the cycle per second is called the "Hertz".

DECIBEL (db) :- A decibel is a unit of level which denotes the ratio between two quantities that are proportional to power, the number of decibels corresponding to the ratio of the two amounts of power is 10 times the logarithm to the base 10 of this ratio (details next page )

DIFFUSE SOUND FIELD (Random-incidence sound field) :-

A diffuse sound field is a sound field such that the sound pressure level is everywhere the same, and all directions of energy flux are equally probable.

FREE FIELD :-

A free sound field is a field in a homogeneous, isotropic medium free from boundaries. In practice, it is a field in which the effects of the boundaries are negligible over the region of interest. The actual pressure impinging on an object (e.g., a microphone) placed in an otherwise free sound field will differ from the pressure which would exist at that point with the object removed. Unless the acoustic impedance of the object matches the acoustic impedance of the medium.

FREQUENCY (f) :-

The frequency of a function periodic in time is the reciprocal of the period. The unit is the cycle per unit time, e.g., cycles per second (cps) or kilocycles per second ($K_c$ or Kcps).

INTENSITY (I) :-

The sound intensity measured in a specified direction at a point is the average rate at which sound energy is transmitted through a unit area perpendicular to the specified direction at the point considered. Only in plane or spherical free progressive sound waves is the intensity related to the average pressure $p$ by the equation ;

$$I = p^2 / p_o c$$

where $p_o c$ represents the characteristic impedance of air.

INTENSITY LEVEL (sound-energy flux-density level)($L_I$) :-

The intensity level, in decibels, of a sound is 10 times the logarithm to the base 10 of the ratio of the intensity of this sound to the reference intensity. The reference intensity shall be stated explicitly; however, a commonly used reference is $10^{-16}$ watts per sq. cm. in a specified direction. In a plane progressive wave, there is a known relationship between sound-energy flux density and sound pressure so that sound-energy flux, density level can be deduced from a measurement of sound pressure level. In general, however, there is no simple relationship between the two and a measurement of sound pressure level should not be reported as intensity level.

LOUDNESS (L) :-

Loudness is the intensive attribute of an auditory sensation, in terms of which sounds may be ordered on a scale extending from soft to loud. Loudness depends primarily upon the sound pressure of the stimulus, but it also depends upon the frequency and wave form of the stimulus.

LOUDNESS LEVEL (LL) :-

The loudness level, in phons of a sound is numerically equal to the sound pressure level in decibels, relative to 0.0002 microbar, of a pure tone of frequency 1000 cps, consisting of a plane progressive sound wave coming from directly in front of the observer, which is judged by normal observers to be equivalent in loudness.

MEL :-

The mel is a unit of pitch. By definition, a simple tone of frequency 1,000 cps, 40 db above a listeners' threshold, produces a pitch of 1,000 mels. The pitch of any sound that is judged by the listener to be n times that of a 1 - mel tone is n mels.

MICROBAR, DYNE PER SQUARE CENTIMETER :- A microbar is a unit of pressure commonly used in acoustics. One microbar is equal to one dyne per sq. cm.

NOISE REDUCTION COEFFICIENT (NRC) :-

The noise reduction coefficient of a material is the average, to the nearest multiple of 0.05, of the absorption coefficients at 250, 500, 1,000 and 2,000 cps.

OCTAVE BAND PRESSURE LEVEL :-

The octave band pressure level of a sound is the band pressure level for a frequency band corresponding to a specified octave.

Electrical filters used for analysing noise reject signals of frequency below a lower "cutoff" frequency and above the upper "cutoff" frequency . Signals between these two frequencies are passed by the filter, so this intermediate region is called the pass band. The difference between the cutoff frequencies is the band width.

The following pass bands often are included in filters intended for noise measurement.

37.5 to 75 cps, 75 to 150 cps, 150 to 300 cps, 300 to 600 cps, 600 to 1200 cps, 1200 to 2400 cps, 2400 to 4800 cps, and 4800 to 9600 cps. Note that in each case the ratio of the cut-off frequencies is 2 : 1. This frequency ratio defines the interval called the octave in music, so these pass-bands are called octave bands.

PHON :- The phon is the unit of loudness level.

PURE TONE :- A pure tone is a sound wave, the instantaneous sound pressure of which is a simple sinusoidal function of the time.

REVERBERATION :- Reverberation is the sound that persists at a given point after direct reception from the source has stopped.

REVERBERATION TIME ($t_{60}$) :-

The reverberation time for a given frequency is the time required for the average sound pressure level, originally in a steady state, to decrease 60 db after the source is stopped. Usually the pressure level for the upper part of this range is measured and the result extrapolated to cover 60 db.

SABIN (Square -foot unit of Absorption) :-

The sabin is a measure of the sound absorption of a surface, it is the equivalent of 1 sq.ft. of perfectly absorptive surface.

SABINE ABSORPTION :-

The Sabin absorption in a room is the sound absorption ($\alpha$) defined by the sabin reverberation time Equation ;

$$t_{60} = 0.049 \frac{V}{a}$$

where $t_{60}$ is the reverberation time in seconds, V is the volume of the room in cubic feet, and a is the total (sabins) absorption in sabins (sq. foot units) In metric units ;

$$t_{60} = 0.161 \frac{V}{a}$$

where V is the volume of the enclosure in cubic meters, and a is the total sabin absorption in square meters.

SONE :-

The sone is a unit of loudness. A simple tone of frequency 1,000 cps, 40 db, above a listener's threshold, produces a loudness of 1 sone. The loudness of any sound that is judged by the listener to be n times that of the 1-sone tone is n sones. A milli sone is equal to 0.001 sone.

SOUND LEVEL :-

Sound level, in decibels, is the weighted sound pressure level obtained by use of a sound level meter whose weighting characteristics are specified in the latest revision of the American Standards Association Standard on sound level meters. The reference pressure is 0.0002 microbar, unless otherwise specified.

**SOUND POWER LEVEL** :- (PWL)

The sound power level of a sound source, in decibels, is 10 times the logarithm to the base 10 of the ratio of the sound power radiated by the source to a reference power. The reference power is 1 picowatt (1 $\mu\mu$ watt). To indicate the 1-picowatt reference power, the letter p is affixed to the abbreviation for decibels, that is , $db_p$

**SOUND PRESSURE LEVEL** (SPL) :-

The sound pressure level, in decibles, of a sound is 20 time the logarithm to the base 10 of the ratio of the pressure of this sound to the reference pressure. The reference pressure is 0.0002 microbar. In many sound fields the sound pressure ratios are not proportional to the square root of corresponding power ratios and hence cannot be expressed in decibels in the strict sense, however, it is common practice to extend the use of the decibel to these cases.

**TRANSMISSION COEFFICIENT** (Acoustical Transmissivity) :-

The sound transmission coefficient of a partition is the fraction of incident sound transmitted through the partition. The angle of incidence and the characteristic of sound observed must be specified, e.g., pressure amplitude at normal incidence.

**TRANSMISSION LOSS** :-

Transmission loss is the reduction in magnitude of the sound pressure level between the two stated points in the transmission system. If the levels are expressed in decibels, then the transmission loss is likewise in decibels.

**TRANSMISSION LOSS OF A PARTITION (T.L) :-**

The sound transmission loss of a partition, in decibels is -10 times the logarithm to the base 10 of the power transmissivity of the partition. It is equal to the number of decibel by which sound incident on the partition is reduced in transmission through it. It is thus a measure of air borne sound insulation of the partition. Unless otherwise specified, it is to be understood that the sound fields on both sides of the partition are diffuse.

**DIRECTIVITY FACTOR (Q) :-**

It is common for sound sources to radiate more sound in one direction than in another. The sound pressure in front of a speaker's mouth, for high frequency sound, is likely to be 10 times as great as that in the opposite direction. Low frequency sound from the mouth is more uniformly radiated in all directions. In general, a sound source that is small in comparison with the wavelength of sound it produces tends to be an omnidirectional source, a sound source large in comparison with the wavelength of the sound is apt to be directional. The directivity factor is often employed to describe how directional a source is. The directivity factor Q is defined as the ratio of the mean square sound pressure, at some fixed distance and specified direction, to the mean square sound pressure at the same distance but averaged over all directions from the source. The distance must be sufficiently great that the sound appears to come from a single point called the acoustic centre of the source. Thus if $p_d^2$ is the squared sound pressure in the specified direction and $p^2_{av}$ is the squared pressure averaged overall directions, the directivity factor for the specified direction is ;

$$Q = \frac{P_d^2}{p^2_{av}}$$

---

## -: BIBLIOGRAPHY :-


CHAPTER - I.

1. Beranek, L.L., 'Noise reduction' McGraw Hill Book Co., Inc. New York, Ch. 21, 1960.
2. Harris, C.M., " Hand Book of Noise Control" McGraw Hill Book Co. Inc. New York, Ch. 1, 1957.
3. Miller, R.L., How do you eliminate noise in airconditioning installations, Refrigerating Engineering, V.61, No.7, July, 1953, p. 738-41.
4. Pietrasanta, A.C., Some fundamental principals of noise control part I- Introduction to the problem of control, Refrigerating Engineering, V.62, no.4, April, 1954, p. 37-40, 100.
5. Tutt, R.B., and C.J. Hemond Jr. A new theory for the "Evaluation of noise components" in air-distribution systems. Noise control, V.3, no.1, Jan.1957, p. 10-13.
6. Knudsen, V.O., and C.M. Harris, "Acoustical Designing in Architecture" John Willey & Sons, Inc. New York, Ch. 2, (1950).
7. Same as Ref. 2, Ch. 4, 7 and 8.

CHAPTER - II.

8. "Heating, ventilating and airconditioning guide", American Society of Heating, Refrigerating and Airconditioning Engineers, Inc. New York, Ch.10, Vol.38, 1960.
9. "ASHRAE guide and data book" American Society of Heating, Refrigerating and Airconditioning Engineer, Inc. New York, Section VI, 1961.

CHAPTER - III.

11. Same as Ref. 8 Ch. 22.

12. Gutin, L., "Über das Schallfeld einer Rotierenden Luftschraube" Physik, Zeits Sowjetunion V 9, 1936, p 57-71, Also Air Research Council Paper No.3115.

13. Deming, A.F., Propeller Rotation noise due to torque and thrust JASA V12 No. 1, July 1940. p. 173-82.

14. Reigier, A.A., and H.H.Hubbard, Status of research on propeller noise and its reduction, JASA,V 25 no.3, May,1( 1953, p. 395-404.

15. Peistrup, C.F., and J.E. Wesler, Noise of ventilating fans, JASA V 25 no.2, March 1953, p. 322-26.

16. Beranek, L.L.,J.L.Reynolds, and K.E.Wilson, Apparatus and Procedure for predicting ventilation system noise. JASA V25 no.2 March,1953,p. 313-21

17. Beranek, L.L,G.W. Kamperman, and C.H.Allen, Noise in centrifugal fans, JASA V27 no. 2, Mar.1955,p.217-19

18. Madison, R.D., Editor "Fan Engineering" Buffalo Forge Co., Buffalo, N.Y. 5th edition, 1949.

19. Allen, C.H., Noise from airconditioning fans, Noise control, V 3, no. 1, January, 1957, p 28.

20. Daly, D.B., Noise level in fans, (with particular reference to axial, flow fans) IEVE, V. 23, May, 1958.

21. Madison, R.D., and J.B. Graham, Fan noise variation with changing fan operation, HPAC, V.30,no.1, Jan.1958. p. 207 - 14.

22. Goldman, R.B., and G.C.Maling, Noise from small centrifugal fans,Noise control V.1,no.6,Nov.1955,p.26-29,50.

23. Dyer, Ira and L.N. Miller, Cooling tower noise, noise control V.5 no.3, May 1959, p.44-47.

24. Wells, R.J., Noise control problems in airconditioning equipment. HPAC, V.29 no.8, Aug. 1957, p.138-43.

25. Morris, W.F., In high velocity systems, "Duct Parts Create Sound, as well as suppress it" J ASHRAE VII, no.3, March 1960 p. 40 - 54.

26. Geiger, P.H., How to avoid trouble from noise in airconditioning installations. Heating, Piping and Airconditioning V 8, 1936, p 601.

27. Same as Ref. 1., Ch. 13.

28. Same as Ref. 8, Ch. 29.

29. Chaddock, J.B., Ceiling air diffuser noise Bolt Beranek and Newman, Inc., Cambridge, Massachusetts, Tech. Information report No.45, August, 23,1957.

30. Ostergaard, P.B. Determination of sound power level of Airconditioning units, Noise control V.3, no.1, January 1957, p.26-29, 53.

31. Corbitt, R.A., Fan Noise Level, what does it mean? HPAC, V.23, no.6, June 1951, p. 99.

32. Hardy, H.C., and D.E. Bishop, Evaluation of equipment noise, HPAC, V.27, n.9, Sept., 1955, p. 137-47.

33. Kodaras, M.J., Suppression of ventilating noise. Noise control, V.4, no.2, March, 1958, p.42-46.

34. Harvet, D.H., Experimental study of Grille noise, characteristics, J ASHRAE V 1, no.7, July,1959, p.59-63.

CHAPTER-XI.

35. Peterson, A.P.G., and L.L.Beranek "Handbook of Noise measurement", General Radio Co., Cambridge, Massachusetts, 3rd edition.

36. Fletcher, H. and W.A. Munson, Loudness, its definition, measurement and calculation, JASA, V 5, 1933, p. 82-108.

37. Beranek, L.L., and A.P.G. Peterson, Determination of loudness of noise from simple measurements, JASA, V 20 no.4, July, 1948, p. 502.

38. Churcher, B.G., and A.J.King, The performance of noise meters in terms of the primary standard, JIEE (London) V.81, July-Dec., 1937, p. 57-90.

39. Beranek, L.L. J.L. Marshall, A.L.Cudworth and A.P.G.Peterson, Calculation and measurement of loudness of sound, JASA, V.23, no.3, May, 1951, p. 261-269.

40. Mintz, F., and F.G. Tyzzer, A loudness chart for octave band data on complex sounds, JASA V 24 no.1, Jan.1952, p. 80 - 82.

41. Stevens, S.S., Calculation of the loudness of complex sounds, JASA, V28 no.5, Sept., 1956, p807-32.

42. Stevens, S.S., Calculating loudness, Noise control V3 no.5, Sept., 1957, p. 11.

43. Hardy, H.C. Loudness limits for noise from heating and air-conditioning equipment, HPAC V 31, no.11, Nov.1959, p. 120.

44. Same as Ref. 2, Ch. 9.

45. Ashley, C.M. Criteria for room noise from airconditioning, HPAC, V 29, no.7, July, 1957, p. 145-50.

46. Beranek, L.L., J.L. Reynolds, and K.E. Wilson, Apparatus and procedures for predicting ventilation system noise, JASA, V 25, no.2, March, 1953, p.313-21.

47. Same as Ref. 1, Ch. 20.

48. Beranek, L.L. Revised Criteria for noise in buildings, Noise control, V.3, no.1, Jan., 1957, p.19-28.

49. Bryter, K.D. Noise control criteria for buildings, Noise control, V.3, no.6, Nov., 1957, p.14-20.

50. Purkins, H.J. Noise in ventilating systems, IHVE, V.20, no. 208, Oct., 1952, p. 255-63.

CHAPTER - V.

51. Wilbur, D.A., and R.F. Simons, Determining sound attenuation in airconditioning systems, Trans.A.S.H.V.E., V.49, 1943, P.267-273.

52. Chaddock, J.B., T.L.Moore, H.H.Bell, and J.K. Dunnely, Sound attenuation in straight ventilation ducting. Refrigerating Engineering, V.67, no.1, Jan.1959, p. 37 - 42.

53. Marks, W.F., In high velocity systems, duct parts create sound as well as suppress it. Journal ASHRAE V 2, no.3, March, 1960, p. 49-54.

54. Same as Ref. 8, Ch. 25.

55. Mariner, T., Control of noise by sound absorbent materials. Noise control V 3 no.4, July, 1957, p.11-19,54.

56. James, W.P., The attenuation of noise in ventilating duct-II, Journal IHVE V 26, April, 1958, p 22-28.

57. Delgado, P.D., and Sidney Baron, Estimating octave band levels of noise generated by airconditioning systems, HPAC V 29, no.9, Sept., 1957, p.160-63.

58. Brittain, C.P, C.H.Maguire, R.A.Scott, and A.J.King, Attenuation of sound in lined air ducts, Engineering V.168, no.4378,

59. Sabine, H.J., The Absorption of noise in Ventilating Ducts, JASA V 12, 1940, p. 53.

60. Same as Ref. 6, Ch. 13.

61. Beranek, L.L., J.L.Reynolds, and K.E.Wilson, Apparatus and procedures for predicting ventilation system noise, JASA V 25 no.2, March,1953, p. 313-21.

62. Brittain, C.P., Attenuation of sound in lined air ducts, Engg. V. 165, n. 4281, February 13,1948, p.145-47.

63. King, A.J., Attenuation of sound in lined air ducts, JASA V 30, no.6, June 1958, p 505 - 507.

64. King, A.J., Silencing air channels attenuation in Rockwool lined ducts, Engineering V 185 no 4791, Jan.3,1958 p. 20.

65. Doelling, N., How effective are packaged SOUND ATTENUATORS for airconditioning systems, Journal ASHRAE, V 2 no.2, Feb., 1960, p.46-50.

66. Wells, R.J., Acoustical plenum Chamber, Noise control, V 4, no.4, July,1958, p.9-15.

67. Wells, R.J., Noise control problems in airconditioning equipment, HPAC V 29, no.8, Aug.,1957, p.130.

68. Beranek, L.L, "Acoustics" McGraw Hill Book Co.,New York, 1954, Ch.10, Part XXXII, and Part XXXV.

69. Same as Ref. 6, Ch. 25.

70. Same as Ref. 2, Ch. 23.

71. Lobato, S., Porous materials for noise control, Noise control, V. 2, no.1, Jan.1956, p. 15-19,72.

72. Lemmerman, R.D., Air conditioning and ventilation noise reduction. Noise control, V 3, no.1, Jan.,1957, p. 47,- 51, 62.

73. HPAC Engineering Data File - some noise control fundamentals, HPAC, V 31, no.7, July, 1959, p. 41-56.

74. Allen, C.H., Control of airconditioning noise in advanced planning state, HPAC, V.31, no.10. Oct., 1959, p.103 - 108.

**[illegible]. CHAPTER - [illegible].**

75. Crede, C.E. "Vibration and shock isolation" John Wiley and Sons, Inc. New York, 1957, Ch.2.

76. Same as Ref. 2, Chs. 12, 13 and 14.

77. Miller, R.L., How do you eliminate noise in air conditioning installations, Refrigerating Engineering V.61, no.7, July 1953, p.738-41.

78. Vance, D.K., The use of vibration and shock control in reducing noise levels, Noise Control, V.5, no.2, Jan. 1959, p. 64- 72.

79. Plunkett, R., Interaction between a vibrating machine and its foundation. Noise control V 4, no.1, Jan., 1958, p. 18 - 22.

80. Chapman, R.A., The use of coating in noise control, Noise control, V 2, no.2, March, 1956, p. 42-45.

81. Finn, T.C., How to control vibration in office and apartment buildings. HPAC V 29, no.5, May, 1957, p.122-26.