A STUDY ON EFFECT OF DIFFERENT OPERATING PARAMETERS ON FRANCIS TURBINE NOISE

A Dissertation

Submitted in partial fulfillment of the requirement for the award of

Degree of

MASTER OF ENGINEERING

IN

CAD/CAM & ROBOTICS

Submitted By

RAJBIR SINGH

Roll No. 800881018

UNDER THE GUIDANCE OF

Mr. PARAS KUMAR

Assistant Professor

Mechanical Engineering Department

Thapar University, Patiala

MECHANICAL ENGINEERING DEPARTMENT

THAPAR UNIVERSITY

PATIALA-147004, INDIA

CERTIFICATE

I hereby declare that the work which is being presented in the dissertation work entitled, "**A Study on Effect of Different Operating Parameters on Francis Turbine Noise**", in partial fulfillment of the requirements for the award of degree of **Master of Engineering in Mechanical Engineering in CAD/CAM & ROBOTICS** submitted in Mechanical Engineering Department, Thapar University, Patiala, is an authentic record of my own work carried out under the supervision of **Mr. Paras Kumar.**

The matter presented in this thesis has not been submitted for the award of any other degree of this or any other university.

Rajbir Singh
15/7/10

Date: - July'2010
Place: - Patiala **(RAJBIR SINGH)**

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This is to certify that the above statement made by the candidate is correct and true to the best of MY knowledge.

15/7/2010

Mr. PARAS KUMAR

Assistant Professor

Mechanical Engineering Department

Thapar University, Patiala

....

Countersigned by

15/7/10 21.7.10

(Dr. S.K.MOHAPATRA) **(Dr. R.K.SHARMA)**

Professor and Head **Dean**

Mechanical Engineering Department **Academic Affairs**

Thapar University, Patiala **Thapar University, Patiala**

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Dated: 15/7/10

RAJBIR SINGH

Roll No. 800881018

ABSTRACT

In our modern, rapidly expanding environment one of the developing problems is that of noise. Apart from the pure annoyance factor of noise, exposure to an intense sound field over a long period of time presents the risk of permanent damage of hearing.

Sound may be described as the disturbance that propagates through a physical (elastic) medium. Rather noise can be conveniently and concisely defined as unwanted sound. With the rise in population which has lead to increase in traffic and industries one of the developing problems is of "Noise".

In the modern day to day world one of the basic needs is for energy generation in order to cope with rise in demands of electricity. For generation of electric energy it is necessary to run turbines. Running of turbines creates a lot of noise which pollutes our environment and leads to detrimental effects on neuroendocrine, cardiovascular, respiratory and digestive systems. Chronic exposure to noise causes fatigue and interferes with concentration, thus reducing work efficiency.

With ever increasing consumption of electricity, it is a matter of concern for our country to control the noise pollution due to it and act for the savage of environmental hazards which can be caused due to it. Public at large scale is raising concerns over the declining state of environment and health.

Most of the electricity is generated by hydro turbines. In the present study the hydro turbine chosen for work is Francis turbine because it is used in energy generation plants and set up was available in the labortary. Turbine set up can be distributed into two units on the basis of noise generation, active noise and passive noise. Components that generate active noise are turbine unit and for passive noise are pumping system, etc. Passive noise propagates from pump, motor, coupling and bearings, etc.

In the present study an attempt has been made to fabricate a acoustic enclosure to control the passive noise from the pumping unit. In this regard an acoustic enclosure was functionally designed and polyurethane form was used on the surfaces of the enclosure.

An attempt has also been made to measure noise impact at points A, B, C, D, E (at a distance of 0.5 m meter surrounding the turbine). It has been figured out that using acoustic enclosure noise reduces to considerable levels near the A, B, C, D and E points surrounding the turbine in comparison to noise generated by setup without enclosure. The effect of enclosure on 1-1 octave band frequency analysis has also been studied in order to figure under which frequency, sound pressure level is high and enclosure is effective for which frequency range. All these studies have been done by varying vane openings, loads and speeds at the marked locations from the turbine.

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NOMENCLATURE

SYMBOLS	DESCRIPTIONS
C_{centre}	Centre Frequency
dB	Decibel
F_{upper}	Frequency of upper limit
F_{lower}	Frequency of lower limit
Hz	Hertz
I	Sound Intensity
L_{10}	10 percentile exceeded sound level
L_{50}	50 median value of sound level
L_{90}	90 percentile exceeded sound level
Leq	Equivalent continuous sound level
opn	Opening
PR HEAD	Pressure Head
Pa	Pascal
PTS	Permanent threshold shift
$p_A(t)$	Instantaneous acoustic pressure
P_O	Reference acoustic pressure of 20 μPa.
SEL	Sound exposure level
SPL	Sound pressure level

TTS	Temporary threshold shift
VO	Vane opening
W. out	Without Enclosure
W. Enc	With Enclosure
σ	Standard deviation
Δf	Bandwidth
100%	Vane Opening of turbine is Full
75%	Vane Opening of turbine is 3/4
50%	Vane Opening of turbine is 2/4
25%	Vane Opening of turbine is 1/4
0%	Vane Opening of turbine is 0
	(At 0% turbine guide blades have some opening)

CHAPTER 1

INTRODUTION TO NOISE

In present scenario, rapidly expanding environment one of the developing problems is that of noise. Apart from the pure annoyance factor of noise, exposure to an intense sound field over a long period of time presents the risk of permanent damage of hearing. This particular problem is becoming a source of serious concern to industrial corporations, trade unions and companies.

The major noise sources are:

1. Industrial noise
2. Traffic noise
3. Community noise
4. Hydraulic noise

Some of the basic terms used in noise are:

1.1 SOUND[19]

Sound is produced as a result of some mechanical disturbance creating pressure variations in an environment such as air or water, or in fact any medium which can transmit a pressure wave. To be able to hear the sound there must always be air or other medium at the ear. The magnitude of the pressure variations (the amplitude of the pressure oscillation) is proportional to the loudness of the sound.

1.2 NOISE

Noise is sound, while under some circumstances sound is noise. The sound from our music player made for our own pleasure appears on our neighbor's side of the wall as noise. Noise is conveniently and concisely defined as "unwanted sound". Noise is measured in decibel.

1.2.1 Decibel: Decibel is the logarithm of a ratio of two quantities and therefore has no units. Decibel is defined by expression as $10 \log_{10} (P/P_0)^2$

When P is the sound pressure amplitude of the measured sound

P_0 is a reference pressure, 20µPa.

Sound Pressure **(N/m^2)**	**Sound Pressure Level** **(dB)**	**Environmental Conditions**
10^2	134 dB	Threshold of pain
10	114 dB	Loud Automobile horn (distance 1m)
1	94 dB	Inside subway train
10^{-1}	74 dB	Average Traffic on street corner
10^{-2}	54 dB	Living room, Typical business office
10^{-3}	34 dB	Library
10^{-4}	14 dB	Broadcasting Studio
$2x10^{-5}$	0 dB	Threshold of Hearing

Table. 1.1 Environmental conditions at different SPL

1.3 AUDIBLE SOUND

The minimum audible sound is referred to as the threshold of hearing of the individual subject, and this varies considerably for individuals according to age and past exposure to noise. Although it is difficult to define normal hearing, the best realizable definition is provided by the average over a large group of people for whom no defect is expected.

1.4 HARMFUL EFFECTS OF NOISE ON HUMAN BEINGS

- Reduces work efficiency.
- Affects the speech communication.
- May cause temporary threshold shift (TTS)/permanent threshold shift (PTS).
- Induces loss of hearing ability.

- Causes psychological strain and mental fatigue.
- May damage the heart.
- Increases the cholesterol level in the blood.
- Dilates the blood vessels of the brain.
- Upsets the chemical balance of the body.
- Causes headache, nausea and general feeling of uneasiness.
- Induces errors in 'motor' performance, in visual perception and in distance and size evaluations.
- Induces psychosis and acute mental agony.

1.5 ENVIRONMENTAL NOISE POLLUTION

- Serious threat to quality of life.
- Permanent part of human environment.
- Increasing noise levels in industries and urban environment cause severe nuisance.
- Annoyance
- Occupational health hazard to industrial worker.

1.6 CHARACTERISTICS OF SOUND [20]

1.6.1 Wavelength: As the sound propagates through the air it creates pressure variations and the distance between succeeding pressure maxima is called the wavelength.

1.6.2 Frequency: Number of cycles per second is known as frequency.

1.6.3 Background noise: When sound measurement on for instance a machine is carried out, it is important that the background noise level is so low, that it does not have any influence on the result. This can be tested in the following manner. Measure the sound at the position where it should be measured with the source (machine) running. Switch off the machine and measure the sound level without the machine running.

If the difference is less then 3dB measurements should be stopped until the background noise has been reduced. If the difference is between 3 and 10 dB use the curve to correct the measured value. If the difference is more than 10 dB, the background noise may be ignored.

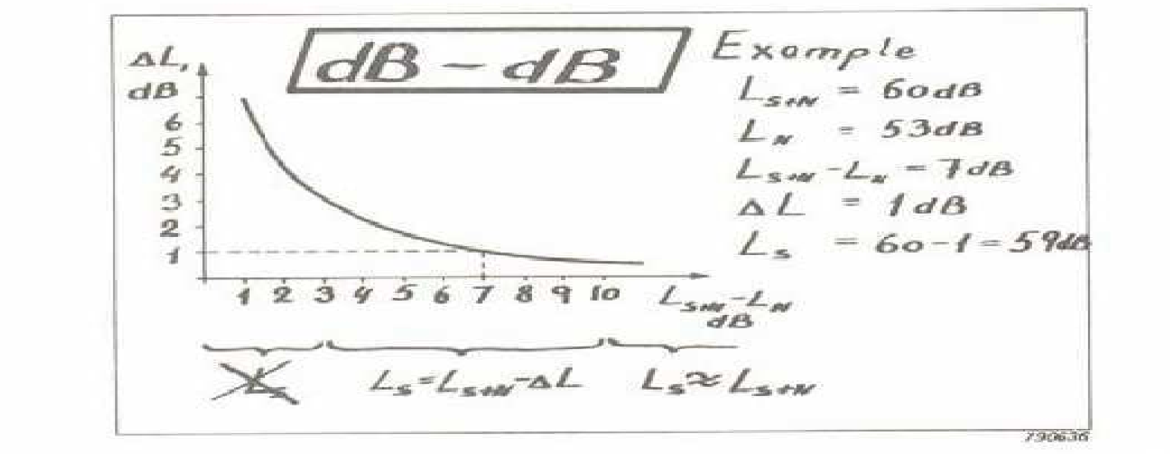


Fig.1.1 Subtraction of background noise in dB

The background noise correction curve is shown in fig 1.1. For example, suppose the measurement without M/C on is 53dB and with M/C on is 60dB. So there is a difference of 7dB and then from background noise curve, 1 dB of correction value is taken to have a corrected value. So the corrected sound pressure level is 59dB.

1.6.4 Loudness: Loudness is a subjectively perceived attribute of sound which enables a listener to order its magnitude on scale from soft to loud. It is defined as subjective intensity of sound. Based on these curves of equal loudness, the "phon" scale was logically conceived as a measure of loudness level. The loudness level of a sound in phons is the sound pressure level in dB re. $2x10^{-5}$ N/m^2 of a pure tone.

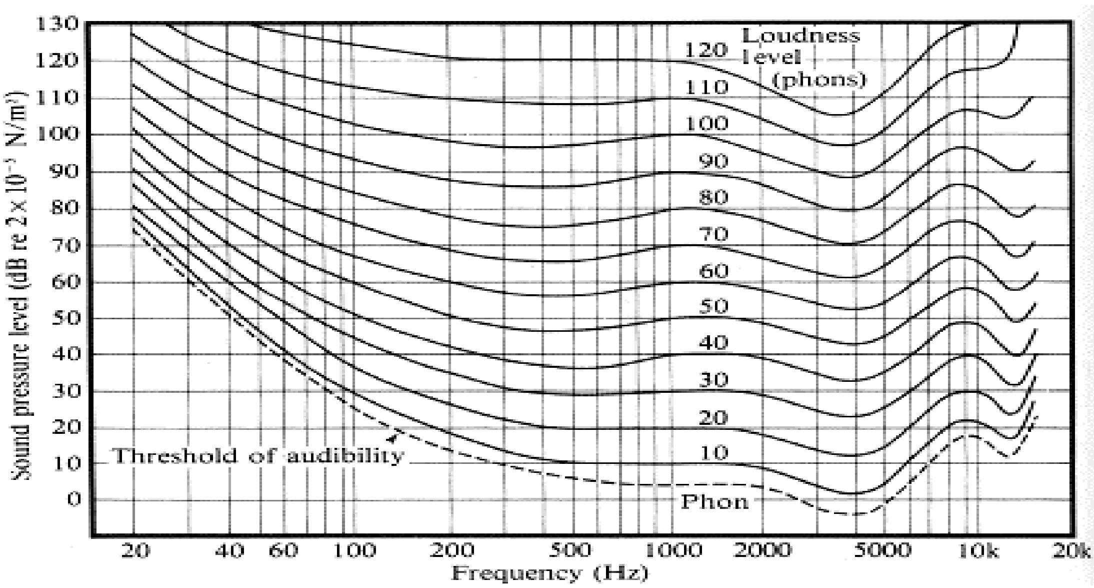


Fig.1.2 Equal loudness contours

Non linear response of the ear:

1. 1000 Hz tone of 40dB (40 phon) is of same loudness as

63 Hz tone of 58dB

Or

4000 Hz tone of 31dB

2. Increase in loudness for a corresponding increase in sound level depends upon frequency and on level also.

1.6.5 Weighting curves: The non-linear response of the ear has lead to the introduction of weighting filters, making it possible to carry out measurements, which correlate well with the response of the ear. The most commonly used of these curves is the A-weighting curve, because it gives the best correlation between the measured values and the annoyance and harmfulness of the sound signal. It follows approximately the 40 phons curve in Fig. 1.3. The B-weighting and C-weighting curve follow more or less the 70 phons and the 100 phons curves. The D-weighting curve follows a contour of perceived noisiness, and is used for aircraft noise measurement. Weighting filters can easily be built into portable Sound Level Meters, and the sound level measured is then given in dB (A) in cases where an A-weighting filter has been used etc.

Some sound level meters also have octave filters built in, or provision for connection of external filters.

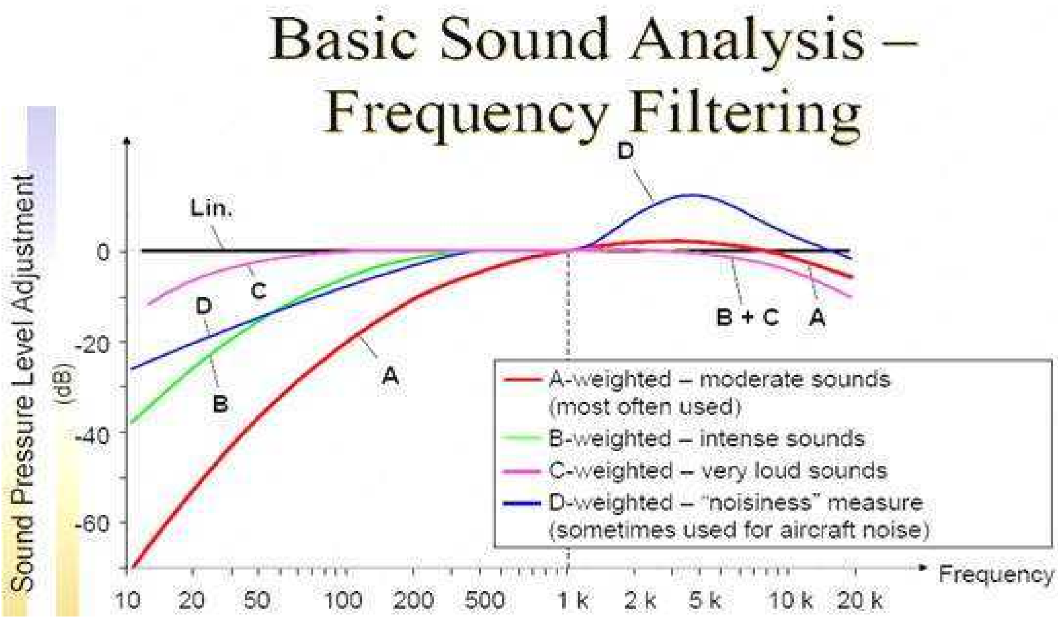


Fig.1.3 Weighting curves

1.6.6 Frequency analyzer: All non-sinusoidal signals are composed of 2 or more sinusoidal signals. The non-sinusoidal signal can be represented in either the time domain as a function of time or in the frequency domain, where the individual frequency components are represented on a frequency scale. A noise signal will contain signals of all frequencies, or at least a broad spectrum of frequencies.

When a sound signal is investigated it is often desirable to investigate a limited part of the frequency spectrum. This can be done with the aid of a filter which will allow passage of only that part of the spectrum which lies inside the bandwidth (Δf) of the filter. A practical filter however will not have such a steep cut-off and the usual filter characteristic is that shown in Fig.1.3 together with the characteristic for an ideal filter.

The bandwidth (Δf) of the filter can be defined as the frequency range between the points where the filter characteristic shows a reduction of 3dB, or as the frequency range of an ideal filter which would allow the same amount of power of a signal containing all frequencies to pass. The difference between the bandwidth found using these two definitions is for most filters very small. For more details see the literature listed at the end of this note.

It is common to classify a filter according to its bandwidth, and there are 2 classes of filters which may be encountered, constant, bandwidth filters and filters with a constant percentage bandwidth.

The constant bandwidth filters have, as the name indicates, a constant bandwidth filters have a constant ratio between bandwidth and center frequency. A special type of constant percentage bandwidth filters is the octave filters, where the upper limiting frequency is twice the lower limiting frequency $f_2 = 2\ f_1$. Where a narrower constant percentage bandwidth is required 1/3 octave filters are used.

By making the octave filter stepwise or continuously variable, it is possible to sweep over a large frequency range and get individual information about each little part of the frequency band.

1.6.7 Equivalent Continues Sound Level (L_{eq}): L_{eq} is the A-weighted energy mean of the noise level averaged over the measurement period. It can be considered as the continuous noise which would have the same total A-weighted acoustic energy as the real fluctuating noise measured over the same period of time and is defined as\

$$L_{eq} = 10\ log_{10}\ \frac{1}{T}\int_0^T \left[\frac{P_A(t)}{Po}\right]^2 dt \qquad ....(1.1)$$

Where T is the total measurement time,

$p_A(t)$ is the A-weighted instantaneous acoustic pressure

And P_O is the reference acoustic pressure of 20 µPa.

1.7 SOUND SOURCES: A distinction is made between 3 different types of sound sources:

1. Point source
2. Line source
3. Plane source

1.7.1 Point source: A sound source can be considered as a point source, if its dimensions are small in relation to the distance to the receiver and it radiates an equal amount of energy in all directions. Typical point sources are industrial plants, aircraft and individual road vehicles. The sound pressure level decreases 6 dB whenever the distance to a point source is doubled.

1.7.2 Line source: A line source may be continuous radiation, such as from a pipe carrying a turbulent fluid, or may be composed of a large number of point sources so closely spaced that their emission may be considered as emanating from a notional line connecting them. The sound pressure level decreases 3 dB, whenever the distance to a line source is doubled.

1.7.3 Plane source: A plane source can be described as follows. If a piston source is constrained by hard walls to radiate all its power into an elemental tube to produce a plane wave, the tube will contain a quantity of energy numerically equal to the power output of the source. In the ideal situation there will be no attenuation along the tube. Plane sources are very rare and only found in e.g. duct systems.

When 2 sources radiate sound energy, they will both contribute to the sound pressure level a distance away from the sources. If they radiate the same amount of energy and the distance from the point of measurement to the sources is the same, the level will increase by 3 dB compared with the level created by one source alone.

1.8 PHYSICAL PROPERTY OF SOUND

1.8.1 Sound power: When sound is produced, a transfer of energy from the source to the surrounding air molecules takes place. The rate of energy transfer is called Sound Power. The unit of Sound Power is W (Watt).

The audible range of sound power extends from 10^{-9} W to more than 1000 W. 10^{-9} W is the lowest level which can be heard by a listener close to the source, and 1000 W will create immediate hearing damage. Lower levels can also create hearing damage, if the listener is exposed for a long period of time.

1.8.2 Sound intensity: When a source produces sound power (P) it will create a certain Sound Intensity (I) at a distance away from the source. The intensity is a measure for the amount of power through a certain area at this distance.

None of these units can be measured directly. Their values can, however, be calculated after measurement of the sound pressure level, knowing the area over which measurements are being made.

The relationship between Sound Pressure (p), Intensity (I) and Sound Power (P) can be written as

$$p^2 \alpha I \alpha P. \qquad \text{......(1.2)}$$

1.8.3 Sound pressure level: Decibel (dB) is logarithmic ratio which defines the sound pressure level L_p as follows:

$$L_p = 20\log_{10} P / P_0 \qquad \text{........(1.3)}$$

Where, P is the sound pressure measured and P_o is the reference sound pressure i.e. 20μPa (the threshold of hearing).

This logarithmic scale has several advantages over a linear scale. The most important advantages are:

1. A linear scale would lead to the use of some enormous and unwieldy numbers.

2. The ear responds not linearly, but logarithmically to stimulus.

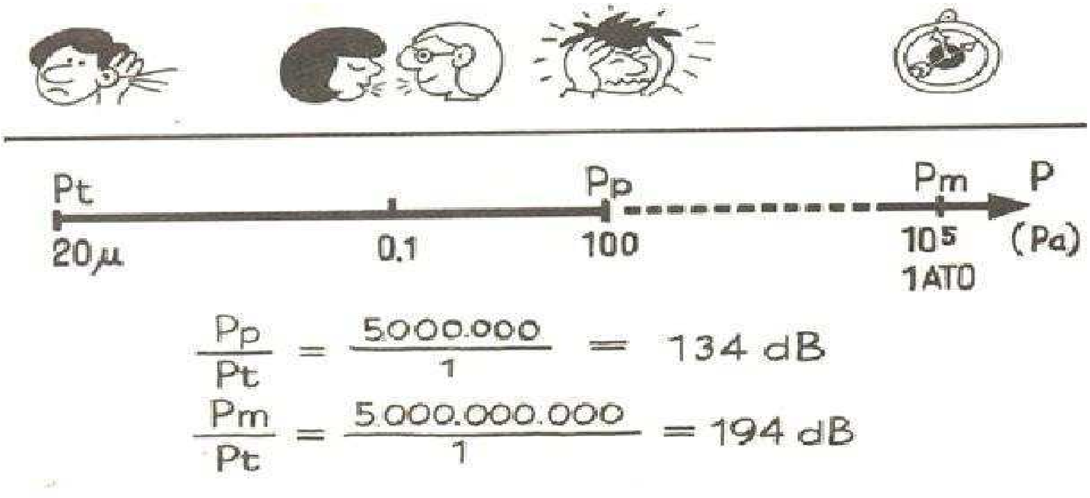


Fig. 1.4 Calculation of SPL

1.9 USEFUL APPLICATIONS OF NOISE:

Noise is not only has harmful affects but sometimes it is very useful. Some of the examples when noise is useful:

1. Study of heart beats: Noise produced by the heart beats is very useful to diagnose the person's health accordingly.

2. Masking effects: Sometimes, it is necessary that nobody should hear the conversation between the two persons. For this, masking effect is used. For e.g., In the doctors chamber, doctor wants that nobody should hear his conversation with the patient so Dr. uses masking effect by putting a more noisy exhaust fan which make noise outside the room.

1.10 NOISE MEASURING INSTRUMENT AND TECHNIQUE

1.10.1 NOISE MEASURING INSTRUMENT

Noise measuring devices typically use a sensor to receive the noise signals emanating from a source. The sensor, however, not only detects the noise from the source, but also any ambient background noise. Thus, measuring the value of the detected noise is inaccurate, as it includes the ambient background noise. For this Sound Level Meter is used.

1.10.1.1 Sound Level Meter has:

1. Microphone

2. Amplifier

3. Rectifier

4. Smoothing circuit

5. Meter

Fig.1.5 Sound Level Meter

1.10.2 PRECAUTIONS IN MEASUREMENT

1. **Records Prior to Measurement:** Record the date and time of measurement, location, weather conditions, personnel names, microphone height, measurement range, frequency compensation of the noise level meter, paper feed speed of the level recorder, and model and manufacturer of equipment.
2. **Wind Effect:** When measuring noise outdoors, attach a wind prevention screen to the microphone of the noise level meter.
3. **Measurement Site:** Select a location that is not effected by reverberated sound or subjected to magnetic fields, vibrations, or extreme temperatures or humidity.
4. **Measurement Period:** Select a time that background noise is stable and there are no other sources possibly effecting measurements. Where the problem source is stable, measurement need last only 2 - 3 min. However, if A-weighted sound pressure level fluctuates greatly, measure for 250 sec or more. If there is background noise from

automobile traffic or other source, measure for the aforementioned duration in a period in which those effects are not noticeable. Especially when recording, the longer the recording, the better**.**

5. **Range Setting:** Get an idea of A-weighted sound pressure level prior to measurement and then set the full scale with some leeway that accounts for the full measurement time. With shock signals, the peak of the waveform can go off the scale even though the needle reading (measured value) may not, therefore it is necessary to keep an eye on the overload warning lamp that lights when a waveform peaks. This same precaution is needed for the audio recorder and not just the measuring equipment.
6. **Keeping Records during Measuring:** Using one's own sense of hearing, distinguish between the target sound and other noise and make a record to that effect on the recording paper during measurement. If the measurement environment changes during measurement, record the change in status, the time it occurred and other related information on the recording paper. For example, if a machine stops or someone passes in front of the noise level meter, make a note of the change in status and the time it happened on the recording paper.
7. **Instructions to Others:** Warn others beforehand not to make sounds while recording noise.
8. **Measurement Point Records:** Differentiate recording points by numbers or other means and mark them on the prepared documents beforehand. Also include the distance from the source, walls, etc. Also, in order to verify the measurement point after measurement, take photographs of the site.
9. **Communications during Measurement:** If the boundary area cannot be seen from the source, station one person at the source to monitor operation and another person at the measurement point, with the two communicating by transceiver. If a large peak or other special event is detected at the measurement point, the person at the measurement should contact the person monitoring the source and record any useful information that can be reported.

1.10.3 STEPS FOR MEASUREMENT OF NOISE

1. Check the Sensitivity (calibration) of the Measurement System: Check the Sensitivity (calibration) of the measuring instrument before and after each measurement.

2. Measure the Acoustical Noise Level

Apply all Necessary Correction to the Observed Measurement.

- Correction for Back Ground Noise.
- Correction for reflection of nearby surfaces.
- Correction for ambient pressure.

3. Out Door Measurement use of Windscreen

Wind can be significant influence on outdoor acoustical measurement.

- Wind effects can be minimized to protect microphone.
- Wind generated noise can be reduced significantly by fitting a wind screen.
- Wind screen is a porous ball of open-cell plastic foam or some other porous material placed over the microphone.

1.11 MEASUREMENT OF SOUND POWER [16]

1.11.1 METHOD TO DETERMINE SOUND POWER LEVEL

1.11.1.1 Sound power level measurement with sound pressure level

The sound power level of noise sources can be measure with the help of sound pressure level with following steps:

1. Surround the source with hypothetical surface of area S (either a hemisphere or a rectangular parallelepiped).

2. Calculate the area of this hypothetical surface if it is hemisphere, S is given by $2\pi r^2$ where r is radius of the hemisphere.

3. If it is rectangular, S is given by ab+2(ac + bc), where a, b, c are its length, width and height.

4. Measure the sound pressure level at designated point on the hypothetical surface.

5. Obtain the average Ls of sound pressure level.

6. Finally calculate the sound power level from the equation.

$$Lw = Ls + 10\log_{10}(S/So) \qquad \ldots\ldots.(1.4)$$

Where, so is reference area

S is hypothetical surface area

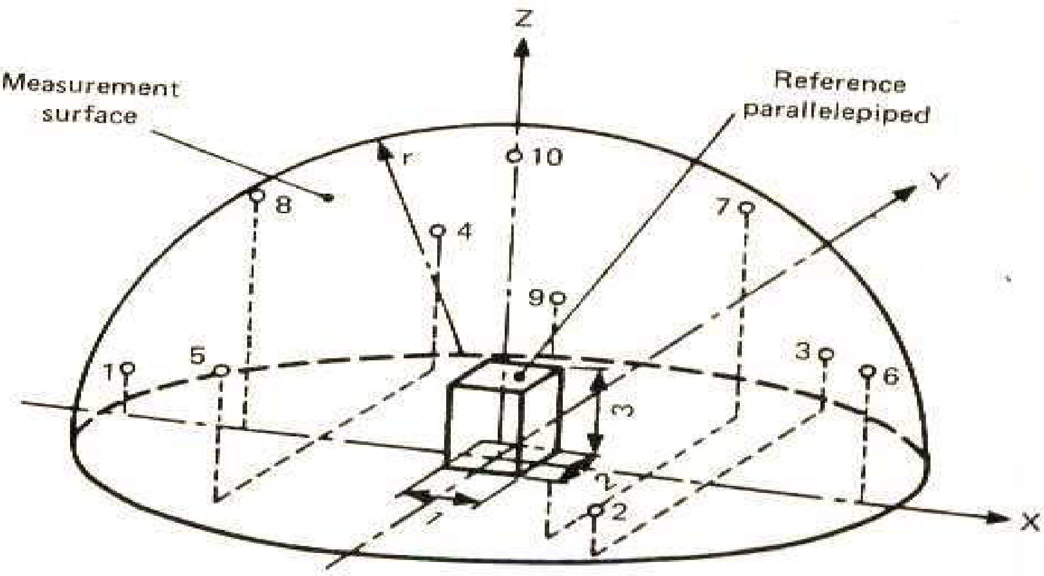


Fig.1.6 Graphical representation of micro phones position on an imaginary hemispherical surface surrounding a source.

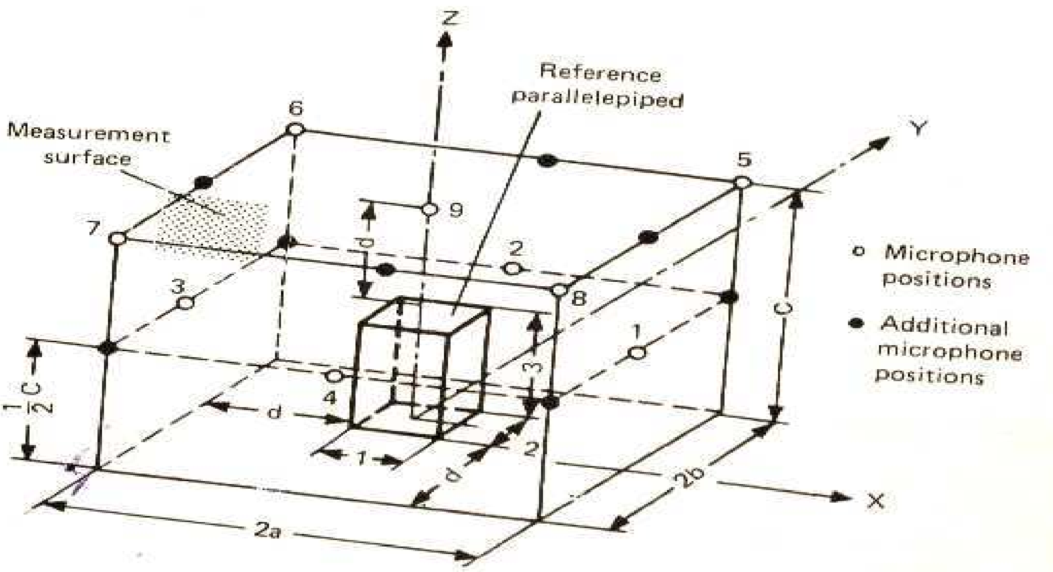


Fig.1.7 Array of microphone positions on an imaginary parallelepiped surface surrounding a source whose sound power is to be measured

1.11.1.2 Additional aspect of measurement which are numbered to correspond to the steps in the above procedure

1. For the small sources those whose largest dimension is significantly less than one meter. It is usually more convenient to se hemisphere the rectangular parallelepiped as a hypothetical measurement surface for large rectangular sources the rectangular parallelepiped surface is usually preferred.

2. The radius of hypothetical hemisphere should be equal to or greater than twice the major source dimension and not less than 1m for the rectangular parallelepiped, the measurement distance “d “, the perpendicular distance between the source and the measurement surface has a preferred value of 1m.

3. For hemisphere the designated point of the microphone locations are shown in Fig. 1.7. The corresponding point for the rectangular parallelepiped is shown in Fig 1.8. The sound pressure level at designated point is measured with A-weighting or in octave or in one-third octave bands.

4. The average sound pressure level over the measurement surface, Ls is calculated from the measured sound pressure level Lsi, after correction for background noise.

1.12 NOISE CONTROL [19]

Noise control is the technology of obtaining an acceptable noise environment, consistent with economic and operational considerations. The acceptable environment may be required for an individual, a group of people, an entire community, or a piece of equipment whose operation is affected by noise. Noise control is not the same as noise reduction. In a specific problem, the amount of noise reduction required to achieve acceptable results sometimes may be obtained simply by applying all the various noise –reduction techniques listed in a following section. The problem should be analyzed systematically to determine.

How acceptable conditions might be achieved in the most economical way. In unusual cases the solution to some noise control problems may even suggest a noise increase, rather than a noise reduction. Consider, for example, the waiting room in a physician’s office that is separated from the consultation room by a partition which provides so little sound insulation that private conversation can be overheard in the waiting room. Acceptable conditions in the waiting room could be achieved by the construction of a partition providing greater airborne sound insulation. A possible alternative solution is to increase the noise level in the waiting room by installing another noise source there (for example, a fan) so as to mask the conversation that would otherwise be overheard. While this latter solution has its disadvantages, it is much more economical and therefore may be more desirable under some circumstances. It illustrates once again that noise control and noise reduction are not always synonymous

HOW MUCH NOISE REDUCTION IS REQUIRED?

The following steps are taken to determine the amount of noise reduction required for a specific problem:

1. **Evaluate the noise environment, under existing or expected conditions**.

Existing conditions may be evaluated from noise measurements which furnish data that are statistically significant. This process requires the appropriate selection and use of measurement equipment, accurate calibration, the taking of data under properly controlled conditions, and the evaluation of any environmental factors which affect the measurements. Under some conditions it is impractical or impossible to evaluate existing conditions. In such cases, or where the noise environment must be estimated for expected or future conditions, an estimate must be made either from empirical engineering formulas or from existing data.

2. **Determine what noise level is acceptable**

This information is provided by an appropriate criterion. A criterion may be defined as a standard or rule for judging, such a standard may be used, for example, for establishing an acceptable limit or restriction that is to be imposed. Noise control criteria provide standard for judging the acceptability of noise levels under various conditions and for various purposes.

3. The difference, between the levels in steps 1 and 2 represents the noise reduction that must be provided to obtain an acceptable environment. This difference usually is determined as a function of frequency.

1.13 NOISES-CONTROL TECHNIQUES [19]

Noise control should always be incorporated at the design stage wherever possible because there are more low cost options and possibilities then to make completed machines and installations quieter. After machines are built or installations completed, noise control approaches can still be achieved through various modifications and add on treatments, but these are frequently more difficult and expensive to implement. Noise control measures may be classified in three categories:

(1) Noise control at the source

(2) Noise control of the transmission path

(3) The use of noise protective measures at the receiver

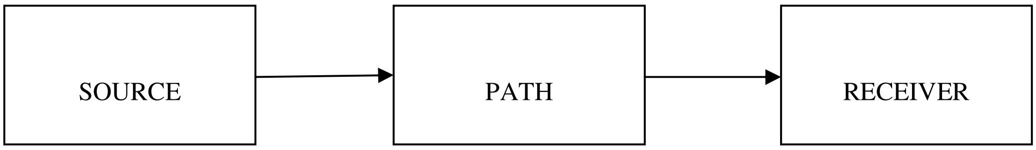


Fig.1.8 Flow of sound from source to receiver

1.13.1 NOISE CONTROL AT THE SOURCE

One important method of controlling noise at the source is to reduce the amplitude of the forces which result in the generation of noise, for example by balancing rotating masses or by isolating vibrating components of the source. Another method is to reduce the motion of the components which are set into vibration; for example, the vibration of panels which are set into vibration maybe reduced by application of vibration-damping materials or by alteration of the resonance frequencies of the panels. Changes in the usual procedure of operation may be an effective noise control technique. Thus some factories, adjacent to residential areas, suspend or reduce noise operations at night, when the normal activity in a community diminishes and the ambient noise level in the community is decreased. Without this noise to mask it, the factory noise becomes more noticeable. Because of this and possible interference with sleep, factories that would otherwise operate on a 24 hour –a-day basis may curtail their operations at night.

1.13.2 CONTROL IN THE TRANSMISSION PATH

Another general technique of noise reduction is that of controlling the transmission path so as to reduce the energy that is communicated to the receiver. This may be done in number of ways:

1. **SITING:** In the open air, maximum attenuation should be provided by increasing insofar as possible –the distance between the source and the receiver. Since many noise sources do not radiate uniformly in all directions, by altering the relative orientation of the source and receiver a considerable reduction in noise level at the receiver may be possible. Thus the orientation of an airport runway may be an important consideration in reducing noise in an adjacent community. Where possible, a site should be chosen that will take advantage of the natural terrain to provide additional shielding of the receiver from the source.

2. **BUILDING LAYOUT:** The careful planning of the location of rooms within a building, with respect to the relative position of the noise sources and those areas in which quit conditions are desired, may result in considerable economy by reducing the extent of the noise control measures that otherwise would be required.
3. **BARRIERS:** Barriers in the open air can be effective when they are large in size compared with wavelength of the sound to be deflected. For example, barriers which make an angle of 45^0 with respect to the horizontal have been used in the noise field of jet aircraft engines to reflect the high frequencies toward the sky.
4. **ENCLOSURES:** Considerable attenuation may be provided by the use of a properly designed enclosure around the noise source or around the receiver.
5. **ABSORPTION** One of the most effective means of attenuating sound in its transmission path is by means if absorption .Suppose a number of machines are in operation in a large room. Most of the noise from these sources that reaches workers on the opposite sides of the room is reflected by the ceiling, walls and floor. Such absorption also reduces the level of sound which reaches the workers after a multiplicity of reflection from the walls, ceiling and floor

1.13.3 PROTECTIVE MEASURES AT THE RECEIVER

The following noise –control techniques may be employed where the noise level at the receiver is excessive:

- **EAR PROTECTION DEVICES**

Ear plugs, car muffs and helmets provide an economical means of reducing noise exposure of industrial workers.

- **BOOTHS**

In many cases it is impractical or uneconomical to reduce noise level to which worker is exposed. It is better to provide a booth or partial or partial enclosure to the worker.

1.14 GENERAL PROPERTIES OF SOUND ABSORPTION MATERIAL

When sound waves strike the surface of a material, a fraction of the incident energy is absorbed by conversion to heat. All materials absorb sound to some extent acoustical materials are those materials whose primary function is to absorb sound. Therefore, they absorb a large function of the acoustical energy which strikes them.

The element which accounts for the dissipation of sound energy in most acoustical materials is a layer of highly porous material in which the pores intercommunicate throughout. The pores may be formed by the felted mineral or fiberglass, by the interstices between small granules, or by a foamed composition in which the solidified bubbles interconnect throughout the material. When a sound wave enters a porous material, the amplitude of vibration of the air molecules is progressively damped out by friction against the surfaces of the fibers or particles forming the porous structure. This friction acts as an acoustical resistance whose value depends on the resistance of the material to direct airflow; such friction depends only slightly on the frequency.

Another factor which affects sound absorption, principally in the low frequency range, is the depth of space between the face of material and a rigid backing surfaces it. The volume of air between these two surfaces includes both the air in the pores of material and any airspace between the material and its backing. The latter may vary from zero, when the material is secured directly to a rigid backing to 1m (3ft) or more in case of suspended acoustical ceilings. When the total depth is less than about one-fourth of the wavelength, low-frequency absorption coefficient of the material decreases with decreasing frequency.

1.15 SOUND ABSORPTION COEFFICIENT

The sound absorption coefficient of a material is a measure of the sound absorptive property of a material. It is the fraction (expressed as a decimal number) of the randomly incident sound power which is absorbed or otherwise not reflected by the material. For example, a sound absorption coefficient of 0.65 indicates that 65 percent of the incident acoustical energy which strikes a material is absorbed. The sound absorption coefficient of every material varies with frequency. It is common practice to list the coefficient of a material at six frequencies: 125, 250, 500, 1000, 2000 and 4000 Hz.

1.16 SELECTION OF ACOUSTIC MATERIAL

A number of other properties or considerations must be considered in the selection of an acoustical material, including:

- Flame spread and fire endurance.
- Mechanical strength, abuse resistance
- Dimensional stability
- Light reflectance

- Sound attenuation
- Maintenance, clean ability, paint ability
- Appearance
- Cost
- Ease of installation, method of mounting
- Space of availability for acoustical installation
- Weight of acoustical installation
- Compatibility with other materials and components

CHAPTER 2

LITERATURE REVIEW

A wealth of literature exists in the area of hydraulic turbines but while going through the literature regarding noise generation by Francis turbine it has been figured out that still a lot of work has to be done regarding it. Usually whenever study of various turbines has been done the focus of researchers has been hydraulic efficiency, mechanical efficiency, work, etc. but noise has been one area that has not been paid much attention. Some important literatures are shown below:

1. **Simpson H. C. et al[1]** studied the a theoretical investigation of hydraulic noise in pumps A theoretical investigation of hydraulic noise produced by volute and diffuser pumps was carried out to determine the effect of design and operating conditions on the noise level in the pumped liquid. Preliminary theoretical work showed that the noise measured by hydrophones in the pump discharge branch was related to the static pressure fluctuations across the exit from the pump volute and a theory based on the potential flow analysis of fluctuating flow through series of blade cascades, in which viscous wake effects were included, produced expressions enabling the effects of pump load, cutwater position and pump speed on the noise level of the pump, to be estimated. This theory showed that although noise generated by the pump casing itself was unimportant, blade circulation and blade wake were of prime importance in determining the pump noise level and the interaction between these two was responsible for the effect of pump loading. The theory also provided an explanation of the observed interaction between the blade frequency noise and the shaft frequency noise if it was also assumed that the impeller flow pattern was asymmetrical. The above theory was compared with some experimental data taken on two pumps, and the effects of pump speed, cutwater clearance and pump load showed satisfactory agreement, particularly since no adjustable parameters were involved. In addition, a simple empirical formula, based on a wider range of pumps, was produced, enabling the general noise level measured in the discharge branch of a pump to be predicted to within ± 2 dB for pumps of a wide range of specific speeds and power consumption groups.
2. **Hubbard H.H. [2]** studied a review of rotating blade noise technology .Rotating blade noise is a topic of wide concern because it is the source of a variety of noise problems.

These concerns range from safety, on the one hand, to community acceptability on the other. The purpose of this paper is to provide a general technical background for the problems of noise due to rotating blades. The topics to be covered are the following: the vehicles and components for which these problems are pertinent, the nature of the noise produced the sources of the noise, concepts of noise generation, identification of the significant parameters in noise generation and reduction, and the methods of noise prediction. Both free rotors and ducted rotors are considered. In addition to the references cited in the text, this paper presents a bibliography of some of the more recent work (since 1968) in the areas of rotor and propeller noise, compressor and fan noise, and duct acoustics.

3. **Ananthapadmanaban T. et al [3]** discussed an investigation of the role of surface irregularities in the noise spectrum of rolling and sliding contacts. The role of surface irregularities in the production of noise in rolling and sliding contacts was investigated. Noise spectra of rolling and sliding contacts were recorded in a small free-field chamber which was specially designed and the results obtained are discussed.
4. **Richards E. J. [4]** studied on the prediction of impact noise, VII: The structural damping of machinery. In earlier parts of this series of papers on the prediction of impact noise, it has been found that in predicting the noise energy radiated from an industrial machine, the only term in the energy accountancy equation which involves the true conversion of vibrational energy into heat is the quantity 10 log η_s; the other terms represent the fraction of impact energy entering the machine and the radiation efficiency change associated with moving this vibrational energy to lower frequencies. Thus the study of the overall damping factor η_s is of crucial importance to the accurate prediction of noise radiated. In spite of the large bibliography available on damping, the practical prediction of this quantity in industrial type machinery is so uncertain that many workers treat the quantity as an unknown “fudge factor” to be obtained from previous similar machines. This forbids the deliberate “designing in” of damping in a new machine, and leads to disappointment if new practices have inadvertently caused a significant loss in η_s, especially when, in fact, the previous versions were relatively highly damped. In this paper a study aimed at improving damping prediction is described. Based upon an investigation of the values of η_s obtained in industrial machinery structures, as opposed to “thin shell” viscoelastically damped structures, a review is presented of the levels of damping which can be obtained by various standard methods. The effects of bolts and

fluid sloshing are included, and specific experiments are described on the effects of adding aggregates in cavities, adding close covers and fitting stick-slip springs on drill rods. There is ample evidence that adequate damping may be obtainable only by the addition of several of these palliatives to different parts of the machinery structure, and accordingly a possible method of summation is proposed, based upon an analogy with room acoustics. The study has led to a realization of the importance of obtaining a simple method of summating damping, and further work is now being done to validate such a method.

5. **Bergman S. et al [5]** studied application of noise analysis for BWR performance monitoring and monitoring. Noise analysis has been used since the mid seventies in order to characterize the operation of the Swedish nuclear power plants. Reactor noise analysis has also been used in performance monitoring of the Boiling Water Reactors Barsebäck 1 and 2, located in southern Sweden. In the beginning reactor noise was addressed as a problem with possible safety implications. Investigations were started in order to characterize and quantify different noise sources. After long experience the safety implications are believed to be small. However, a possible reduction of the noise has indicated a potential for large economical savings and this has motivated the utility, SYDKRAFT AB, to continue the noise analysis research. Application aspects, looking towards early fault detection systems and diagnostic aids for the operators, have also been of a great interest. As a tool for the noise analysis a powerful interactive identification package, IDPAC, running on a VAX 11/780 was used. Both spectral and parametric methods have been used. Almost ten years of reactor noise data describing both normal and abnormal behavior has been compiled and analyzed. Specific noise patterns have been extracted and some examples are presented in this paper. In order to achieve higher reactor controllability, resulting in e.g. shorter start-up times, an extension of the operating area was suggested. By BWR stability monitoring, using the Pressure-Flux transfer function characteristics as a measure, the new operating area was checked and continuous operation permitted. A comparative noise study concerning the in-core neutron flux characteristics of the Barsebäck 1 and 2 was made in 1983. As a result a simple scheme for detection of abnormal boiling was suggested.
6. **M.L. Munjal [6]** presented an overview of the research findings of the author and his students in different aspects of active as well as passive mufflers. Mufflers have been developed over the last seventy years based on electro-acoustic analogies and

experimental trial and error. Passive mufflers based on impedance mismatch, called reflective or reactive mufflers, have been most common in the automobile industry. Mufflers based on the principle of conversion of acoustic energy into heat by means of highly porous fibrous linings, called dissipative mufflers or silencers, are generally used in heating, ventilation and air-conditioning systems. The author has been working in this area for over 16 years.

7. **Maroaa Cuesta. et al [7]** attempted to design and implement a hybrid passive/active system to control the noise radiated by a small generator. Passive control is adored by enclosing the generator with a close, leaky, rectangular box. The measured Insertion Loss is higher than 20 dB above 500 Hz. Special attention is paid to technical aspects such as air refreshing and temperature inside the enclosure. Low frequency noise escapes from the enclosure through air intake and gas exhaust openings. A single-input single-output (SISO) feed forward active noise control (ANC) system, implemented in a commercially available device, is used to reduce the radiated exhaust noise below 500 Hz. The reference signal to the SISO ANC system is supplied by an accelerometer located on the air ®lter case of the generator. The error signal is provided by an electrets microphone along the exhaust pipe. The control source consists of a high temperature loudspeaker, positioned in a side-branch conguration to avoid direct con-tact with the exhaust gas. Some harmonics were attenuated more than 30 dB.
8. **Dina M. [8]** studied detection of cavitation phenomenon in a centrifugal pump using audible sound. One of the sources of instability in a centrifugal pump is cavitation phenomenon. Cavitation within a centrifugal pump can cause more undesirable effects, such as deterioration of the hydraulic performance (drop in head-capacity and efficiency curves), damage of the pump by pitting and erosion and structure vibration and resulting noise. Cavitation can appear within the entire range of operating conditions; therefore it must, by all means, be prevented. To prevent the onset of the cavitation, we have to know the beginning of cavitation phenomenon in the pump. To detect the beginning of the cavitation process, the emitted noise can also be used, among other possibilities. Experiments have shown that there is a discrete frequency tone within the audible noise spectra, at 147 Hz or BPF/2, which is strongly dependent on the cavitation process and its development. Therefore, the discrete frequency tone at 147 Hz was separated from the noise spectra of cavitating pumps and then used to detect the incipient of cavitation and its development. It was also used to determine the net positive suction head required or

the critical value, as well as to prevent cavitation in the pump by means of initiating an alarm, shutdown or control signal via an electrical control system.

9. **Hyeon-Don Ju. et al [9]** discussed the experimental study is made for design of the Acoustic enclosure, which is initially layout by rule of thumb, is evolved systematically through numerical reanalysis procedure, based on indirect boundary element method (IBFM) with a commercial acoustic analysis code. Diesel engine generator sets in heavy industry plants and residential/official buildings can cause serious noise problems. In this paper, a low noise diesel engine generator set is developed through constructing an acoustic enclosure with ventilation duct silencers that electively block the acoustic flow but guarantee good thermal flow. The cooling performance of the acoustically determined enclosing structure is checked and confirmed through numerical heat flow analysis. The acoustic and cooling performances of the developed low noise diesel engine generator set are confirmed by the experiment.
10. **Cudina Mirko. et al [10]** studied the use of audible sound for safe operation of kinetic pumps. Safe operation of kinetic pumps, as liquid movers, can be threatened by Cavitation phenomenon, among others. Cavitation is the Achilles heel of kinetic pumps. It can deteriorate hydraulic performance; damage the pump by pitting and material erosion, and structure vibration and resulting noise. Cavitation can appear within the entire range of operating conditions; therefore it must, by all means, be prevented. To prevent cavitations in a pump, we have to know the beginning and development of cavitations in the pump. For this purpose, the emitted noise in the audible range can be used, among other possibilities. Experiments have shown that there is a discrete frequency tone within the audible noise spectra, which is in strong correlation with the development of the cavitation process in the pump, and so with the net positive suction head (*NPSH*) critical value, which corresponds to the 3% drop in the total delivery head. The discrete frequency tone can thus be used to detect the incipient of cavitation and its development as well as to prevent the onset of the cavitation process in the pump and also in situ operation. Two different measurement methods were used to clarify the noise-generating mechanism, which is responsible for the discrete frequency component: by using a microphone and an accelerometer. Experiments have shown that the characteristic discrete frequency tone, which is in close correlation with the cavitation process, is a result of structural vibrations (modes) or resonances caused by implosion of bubbles and bombardment of the inner surfaces of the pump.

11. **Cudina Mirko. et al [11]** discussed detection of cavitation in operation of kinetic pumps. Use of discrete frequency tone in audible spectra. Safe operation of kinetic pumps, as liquid movers, can be threatened by cavitation phenomenon in, amongst others. Cavitation is the Achilles' heel of kinetic pumps. It can cause deterioration of the hydraulic performance, damage of the pump by pitting and material erosion, and structure vibration and noise. Cavitation can appear within the entire range of operating conditions, therefore it must by all means be prevented. To prevent cavitation in a pump we have to know the beginning and development of the cavitation in the pump. For this purpose, the emitted noise in the audible range can be used, amongst other possibilities. Experiments have shown that there is a discrete frequency tone within the audible noise spectra, which is in strong correlation with development of the cavitation process in the pump. Therefore, the discrete frequency tone can be separated from the noise spectra of a cavitating pump and used to detect the incipient of cavitation and its development as well as to prevent the onset of the cavitation process in the pump, by means of initiating an alarm, shutdown, or control signal via an electrical control system.
12. **Xiang-hong Wang et al [12]** studied Feasibility analysis for monitoring fatigue crack in hydraulic turbine blades using acoustic emission technique. In order to investigate the feasibility of monitoring the fatigue cracks in turbine blades using acoustic emission (AE) technique, the AE characteristics of fatigue crack growth were studied in the laboratory. And the characteristics were compared with those of background noise received from a real hydraulic turbine unit. It is found that the AE parameters such as the energy and duration can qualitatively describe the fatigue state of the blades. The correlations of crack propagation rates and acoustic emission count rates vs stress intensity factor (SIF) range are also obtained. At the same time, for the specimens of 20SiMn under the given testing conditions, it is noted that the rise time and duration of events emitted from the fatigue process are lower than those from the background noise; amplitude range is 49–74 dB, which is lower than that of the noise (90–99 dB); frequency range of main energy of crack signals is higher than 60 kHz while that in the noise is lower than 55 kHz. Thus, it is possible to extract the useful crack signals from the noise through appropriate signal processing methods and to represent the crack status of blade materials by AE parameters. As a result, it is feasible to monitor the safety of runners using AE technique.
13. **Kumar Pardeep [13]** Studied about the cavitation in hydro turbines. Reaction turbines basically Francis turbines and propeller/Kaplan turbines are suitable for

medium and low head hydropower sites. The management of the small hydropower plants is an important factor, for achieving higher efficiency of hydro turbines with time. Turbines show declined performance after few years of operation, as they get severely damaged due to various reasons. One of the important reasons is erosive wear of the turbines due to cavitation. Reaction turbines, however are more prone to cavitation especially Francis turbines where a zone in the operating range is seriously affected by cavitation and considered as forbidden zone. Cavitation is a phenomenon which manifests itself in the pitting of the metallic surfaces of turbine parts because of the formation of cavities. In the present paper, studies undertaken in this field by several investigators have been discussed extensively. Based on literature survey various aspects related to cavitation in hydro turbines, different causes for the declined performance and efficiency of the hydro turbines and suitable remedial measures suggested by various

14. **Ruprecht A. et al [14]** studied the small old hydro power plant equipped with three Francis turbines with one of the turbines replaced by a new one. The new unit produced a low frequency noise. The noise level in the plant is not very high compared to other power plants. The problem in this plant, however, is that the low frequency noise emission carried over to the adjacent apartment of the operator and therefore cannot be tolerated. The disturbing noise depends on the point of operation of the turbine. At low head and consequently also at low discharge the noise level is reduced or even vanishes. Additionally the noise level depends on the guide vane opening. At guide vane opening lower than 40 % the noise appears but is not disturbing. With increasing opening the noise level increases at 70 % it reaches a maximum. Increasing the vane openings further to 100 % the noise level decreases slowly.

CHAPTER 3

NOISE AND ITS CONTROL TECHNIQUES

Noise control should always be incorporated at the design stage wherever possible because there are more low-cost options and possibilities then to make completed machines or installations quieter. After machines are built or installation completed, noise control approaches can still be achieved through various modifications and add-on treatments, but these are frequently more difficult and expensive to implement.

The sources are of two main types:

1. Airborne sound sources caused by gas fluctuations(as in the fluctuating release of gas from an engine exhaust)
2. Structure-borne machinery vibration sources that in turn create sound fields (e.g., engine surface vibrations).

Moreover, these sound pressure and vibration sources are of two types:

1. Steady state
2. Impulsive

Both steady state and impulsive vibrations (caused by impacting parts) are commonly encountered in machines. The paths may also be airborne or structure-borne in nature. Source modifications are the best practice but are sometimes difficult to accomplish. Often changes in the path or at the receiver may be the only real options available. The model is shown in fig. 1.9

In some cases, parts of a machine can be turned off or disconnected to help identify sources. In other cases, parts of a machine can be enclosed, and then sequential exposure of machine parts can be used to identify major sources. Frequency analysis of machines can also be used as a guide to the causes of noise, as with the case of the firing frequency in engines, the pumping frequency of pumps and compressors, and the blade- passing frequency of fans.

3.1 NOISE REDUCTION TECHNIQUES [19]

A study of the literature reveals many successful well-documented methods used to reduce the noise of machines. These can be classified using the source-path-receiver model. Some of the most useful approaches can generally be used only at the source or in the path. Others, such as enclosure can be adapted for use at any location. For instance, a small enclosure can be built inside a machine around a gear or bearing, or a larger enclosure or room can be built around a complete machine. Finally, an enclosure or personnel booth can be built for the use of a machine operator.

Consider the relatively simple system depicted in figure 3.1.which represents a noise source in a typical industrial environment source radiates air borne sound both through the intake and exhaust ventilation openings and through the housing of the machine, the source also radiates structure borne energy through its supports.

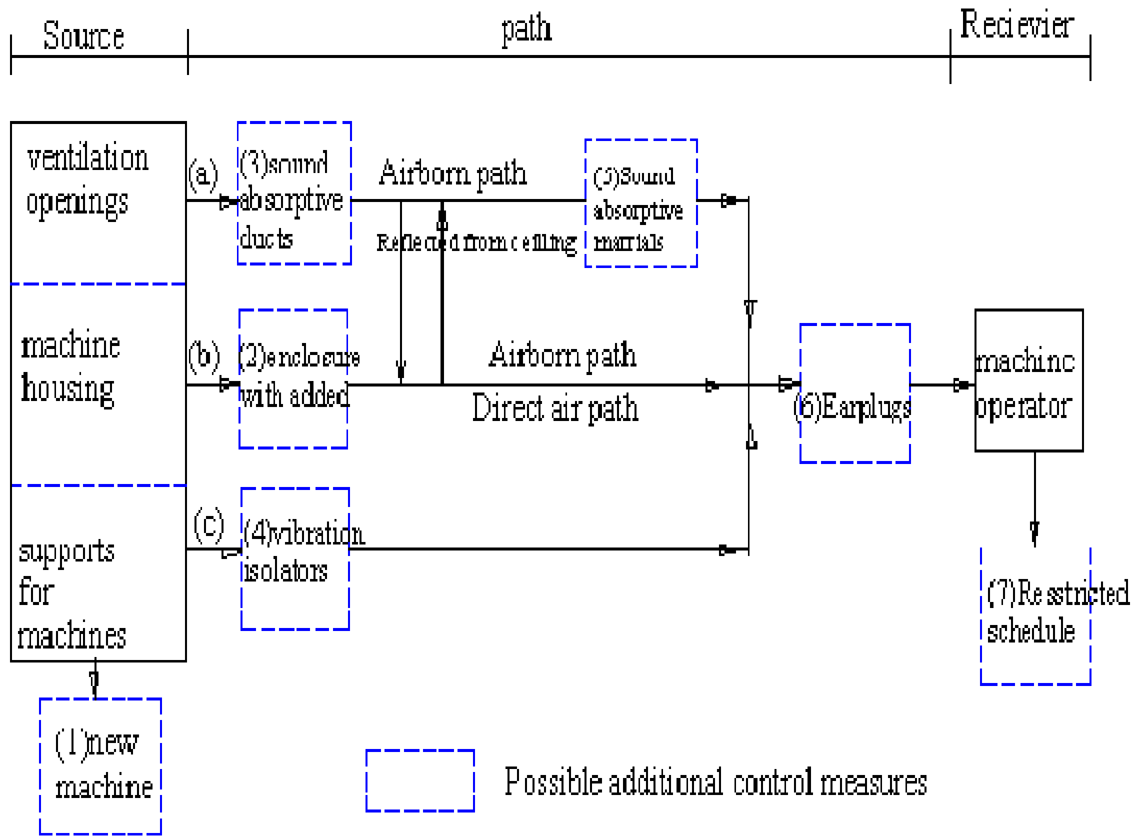


Fig. 3.1 block diagram of noise control system discussed in text, showing sources, paths of energy propagation a, (b) and (c), and possible methods of noise control

3.2 PASSIVE NOISE CONTROL APPROACHES THAT MAY BE CONSIDERED FOR SOURCE, PATH OR RECEIVER [19]

SOURCE

- Choose quietest machine source available
- Reduce force amplitudes
- Apply forces more slowly
- Use softer material for impacting surfaces
- Balance moving parts
- Use better lubrication
- Improve bearing alignment
- Use dynamic absorbers
- Change natural frequencies of machine elements
- Increase damping of machine elements
- Reduce radiating surface areas (by adding holes)
- Stager time of machine operations in plant

PATH

- Install vibration isolator
- Use barriers
- Install enclosures
- Use absorbing materials
- Install reactive or dissipative mufflers
- Use vibration breaks in ductwork
- Mismatch impedances of materials
- Use lined ducts and plenum chambers
- Use flexible ductwork
- Use damping materials

RECEIVER

- Provide earplugs or earmuffs for personnel
- Construct personnel enclosures
- Rotate personnel to reduce exposure time
- Locate personnel remotely from sources

3.3 MAIN APPROACHES TO CONTROL PASSIVE NOISE

The passive noise can be controlled by or reduced by the use of following steps:

1. Vibration isolators
2. Acoustical absorbing material
3. Enclosures
4. Barriers
5. Vibration damping materials

3.4 ACOUSTICAL ENCLOSURES [19]

Acoustical enclosures are used wherever containment or encapsulations of the source or receiver are a good, cost effective, feasible solution. An enclosure corresponds to a nose control measure in the path. Often the main task is to keep the sound energy inside the enclosure and dissipated it by means of sound absorption. In some cases such as with personnel booths or automobile or aircraft cabins, the main task is to keep the noise outside and to absorb as much sound energy as possible that does penetrate the enclosure walls and come inside. They can be classified in four main types:

1. Large loose fitting or room size enclosures in which complete machines or personnel are contained
2. Small enclosures used to enclose small machines or parts of large machines
3. Close fitting enclosures that follow the contours of a machine or a part
4. Wrapping or lagging materials often used to wrap pipes, ducts or other systems.
5. Large booths or room sized enclosures and cabins in which personnel or vehicle passengers are contained.

The performance of such enclosures can be defined in three main ways:

1. Noise reduction (NR), the difference in sound pressure levels between the inside and outside of the enclosures.
2. Transmission loss(TL, or equivalently the sound reduction index), the intensity levels for the enclosure wall
3. Insertion loss (IL), the difference in sound pressure levels at the receiver point without and with the enclosure wall in the place.

The physical behavior and the efficiency of an enclosure depend mainly on:

1. The transmission loss of the walls of the enclosure
2. Its volume and necessary openings (access for passing materials in and out, ventilation and cooling, inspection windows, etc.)
3. The sound energy absorbed inside the enclosure walls that are lined with sound absorbing materials.

3.4 PARTIAL ENCLOSURES

Partial enclosures, illustrated in Fig3.2, represent an intermediate step between the simple one-side barrier and the total enclosure of the equipment noise. Suppose that a source is indoors and that the receiver is far enough from the source that reflected sound predominates. In this environment, the noise reduction provided by a partial enclosure that is completely absorptive depends on the extent to which is surrounds the source. If the enclosure surrounds 50 percent of the source, a reduction of 3dB is achieved, if the enclosure surrounds 75 percent of the source, a 6-dB reduction is achieved.

If the source and the receiver are both outdoors and the receiver lies so close to the source that it is in the direct field of the source, the performance of a partial enclosure usually is similar to that of a barrier. The partial enclosure produces shadow zones within which attenuations in the range 5 to 20 dB can be achieved, depending on the frequency of the source and the construction and configuration of the partial enclosure.

Consider the performance of the partial enclosures shown in Fig. 3.2 Enclosures a, b, and c provides a noise reduction over a wider range of receiver locations. Enclosure c in combination with an absorptive ceiling provides a noise reduction similar to that provided by lined barriers in an open plan office.

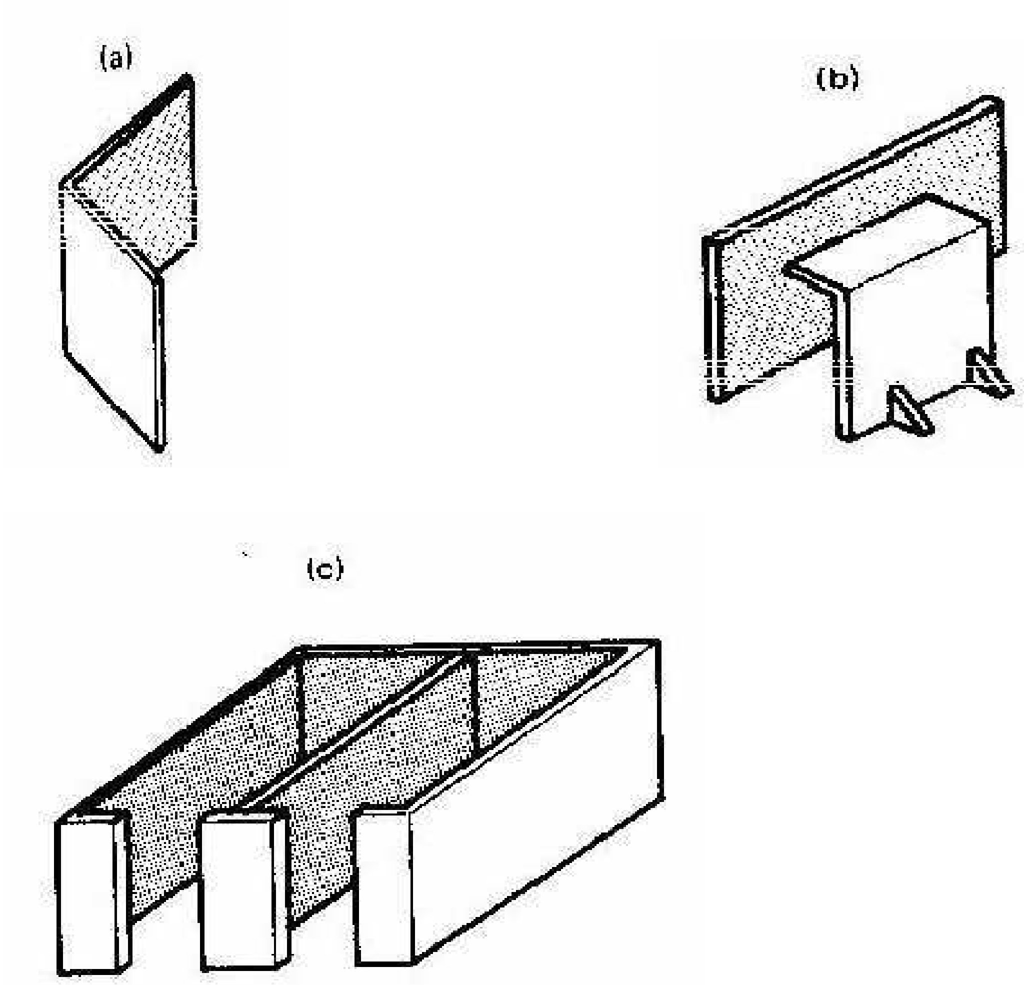
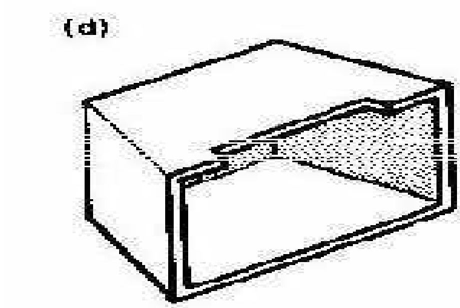


Fig 3.2 Partial enclosures used in machinery noise control [22]

Partial enclosure in fig 3.3 d, e, and f, in general, provide more noise reduction than the simple barrier because the treatment extends over the top of the source. This reduces the extent to which diffractive scattering limits the available attenuation in the shadow zone.



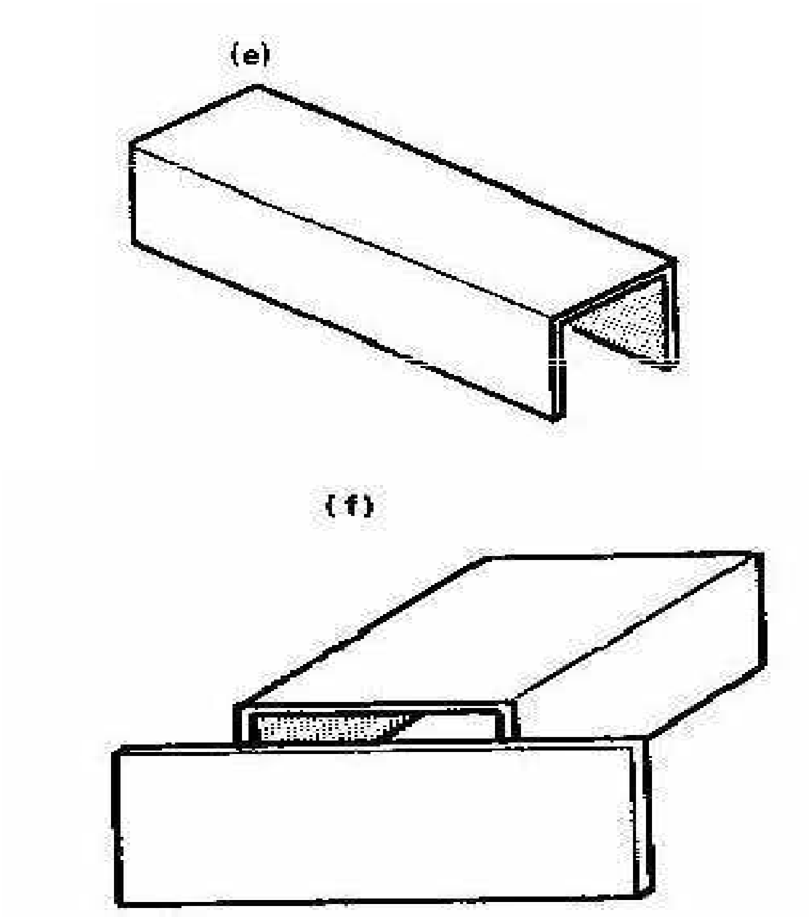


Fig 3.3 Partial enclosures used in machinery noise control

3.5 BARRIERS

An obstacle placed between a noise source and a receiver is termed a barrier or screen. A barrier is a device designed to reflect most of the sound energy incident back towards the source of the sound. When a sound wave approaches the barrier, some of the sound wave is reflected and some is transmitted past.

At high frequency, barriers are quiet effective, and a strong acoustical shadow is cast. At low frequency (when the wavelength can equal or exceed the barrier height) the barrier is less effective and some sound is diffracted into the shadow zone. Indoors, barriers are usually partial walls. Outdoors, the use of walls and even building can protect residential areas from traffic and industrial noise resources.

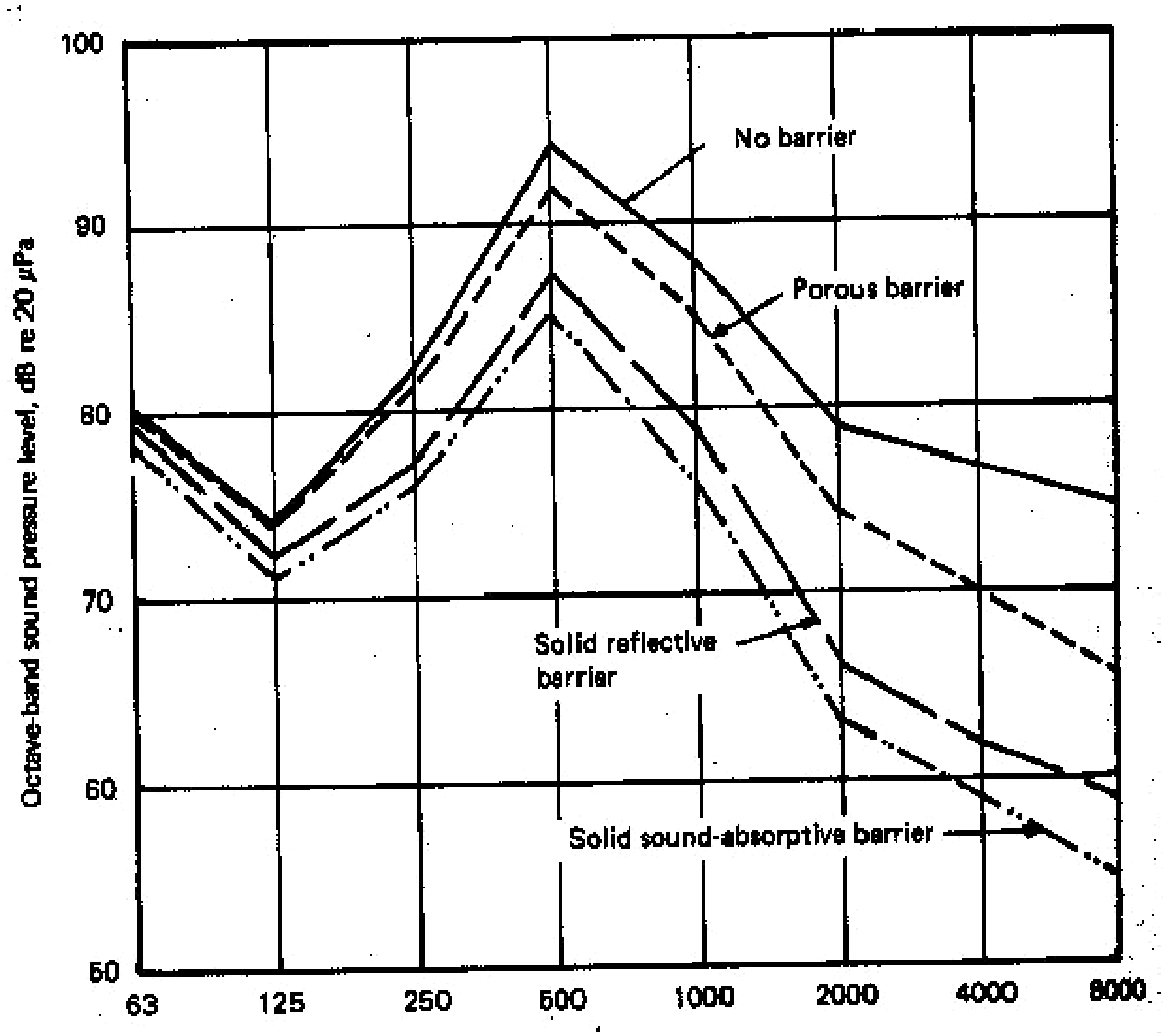


Fig 3.4 Octave-band sound pressure level spectra in which different types of barriers are placed: porous barrier, solid reflected barrier, solid sound absorptive barrier

Figure 3.4 illustrates the effects of barrier material on the barrier's effectiveness in providing noise reduction where only the direct sound is significant. The uppermost curve shows the octave-band spectrum at the receiver in the absence of a barrier. The other three curves show the octave-band spectra for three barrier materials. The difference between the octave band spectrum without the barrier in place and the spectrum with the barrier represents the attenuation provided by the barrier and is a measure of its effectiveness. Three types of barriers are considered here. Porous barrier composed of a sound -absorptive material such as fiberglass Solid reflective barrier composed of nonporous material such as masonry or concrete. Solid sound absorptive barrier composed of a solid reflective barrier covered with a sound absorptive material on the side facing the source.

The effectiveness of the porous barrier in reducing noise level is poor because porous materials have low values of transmission loss. The solid barrier is better, particularly at higher frequencies. Little sound can penetrate the barrier; the sound reaching the receiver must bend over the top of the barrier. Lining the barrier with sound absorptive material on the side facing the source provides some improvement, but only a few decibels

3.5.1 USE OF THE BARRIERS INDOOR

Single screen barriers are widely used in open plan offices (or landscaped offices) to separate individual workplaces to improve acoustical and visual privacy. The basic elements of these barriers are freestanding screens (partial height partitions or panels). However, when placing a sound barrier in a room, the reverberant sound field and reflections from other surfaces cannot be ignored.

3.5.2 USE OF THE BARRIERS OUTDOOR

The use of the barriers outdoors to control the noise from highway is surely the most well known application of barriers

While noise barriers do not eliminate all highway traffic noise, they do reduce it substantially and improve the quality of life for the people who live adjacent to busy highways. Noise barriers include walls, fences, earth berms, dense planting, building, or combinations of them that interrupt the line of sight between source and observer. It appears that construction of the barriers is the main alternative used for reduction of noise, although quite road surfaces, insulation of properties, or use of tunnels have also been used for this purpose.

3.6 NOISE CONTROL PROCEDURE FOR ACHIEVING AN ECONOMICAL SOLUTION [22]

To determine an appropriate and economical noise control treatment for noise problem such as one illustrated in figure 1.8. The following steps should be taken.

STEP 1 - **Determine the contribution of noise that is propagated along each of the paths between the source and receiver**

This may be done in terms of octave –band sound pressure levels as illustrated in figure 4.3 , In this example it is assumed that ambient noise is sufficiently low to be disregarded .The octave-band spectrum labeled “ total noise” represents the spectrum at the receiver when the equipment is in operation before any noise control measure are introduced.

The octave–Band spectrum labeled “air born reflected” represents the result of measurements in which energy transfers along both paths b and c has been eliminated. For example, energy transfer along path b may be eliminated by the temporary insulation of a barrier which effectively shields the source from the receiver. Energy transfer along path c may be eliminated by isolating the equipment from the floor on which it rests.

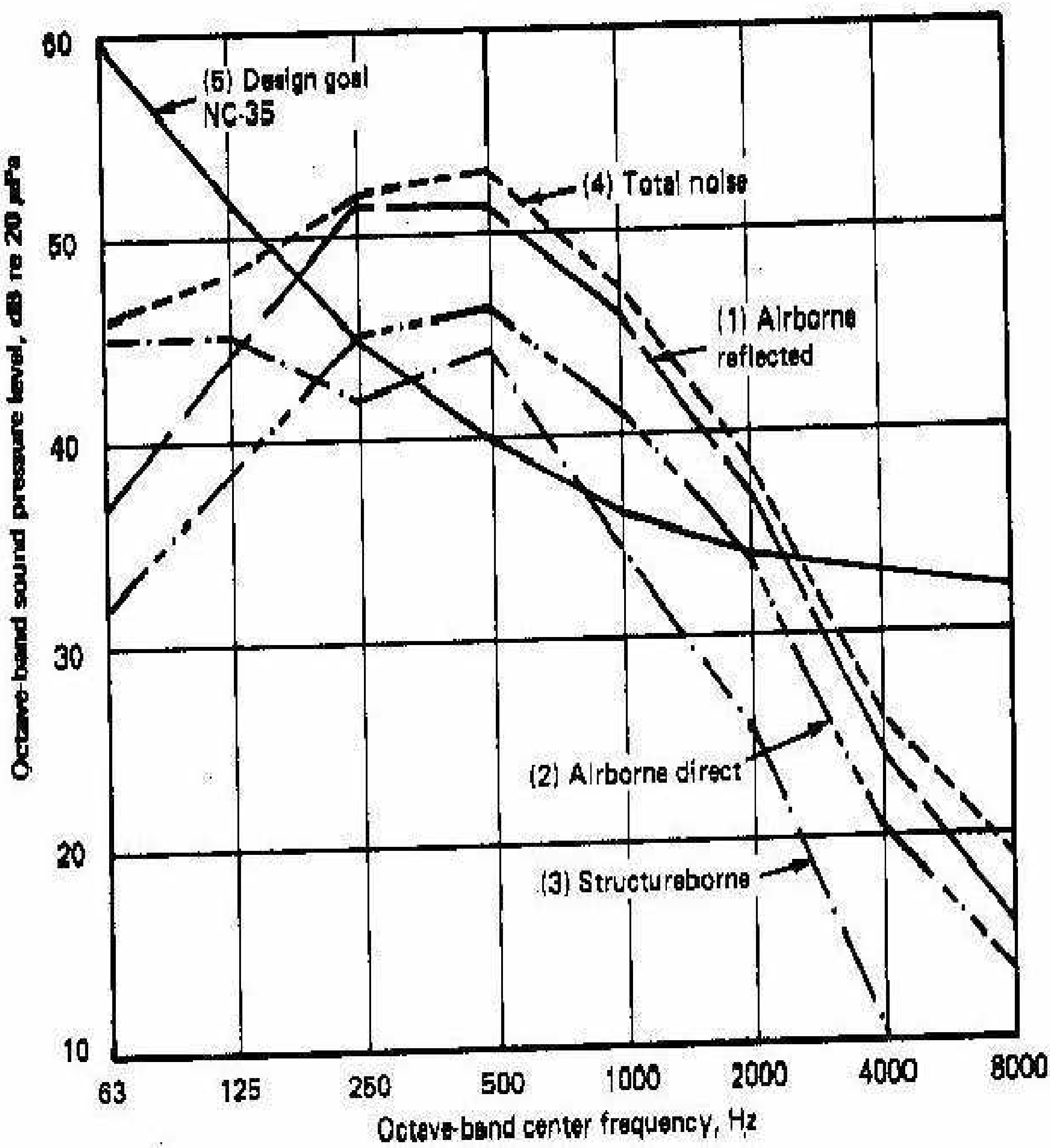


Fig. 3.5 Octave-band spectra for the system of Fig. 3.1 showing:-

(1) The spectrum of the airborne sound that reaches the receiver by reflections path a.

(2) The spectrum of the airborne sound that reaches the receiver directly, with no reflection path b.

(3) The spectrum of the structure borne sound that reaches the receiver by path c.

(4) The spectrum of the total sound that reaches the receiver by both airborne and structure borne paths.

(5) The design goal.

The octave-band spectrum labeled “air borne direct” represents the result of measurements in which energy transfer along paths a and c has been eliminated, so that the only sound that reaches the receiver is propagated along a direct air path between the source and receiver. Such a spectrum is obtained by carrying out measurements on the machine in free field. It can be approximated by covering all surfaces of the room with a highly absorptive materials, thereby eliminating propagation along path a. Energy transfer along path c can be eliminated, as above, by isolating the equipment from the floor.

The octave band spectrum labeled structure borne represents the result of measurement taken in which the source of noise is completely surrounded by highly affective enclosure that eliminates the air borne contributions transmitted along with both paths a and b .

STEP 2- **Select a design goal**

The design goal (example the octave–band spectrum to be achieved) depends on the nature of the equipment, the environment in which it is located and the nature of the requirements. For example, in an industrial environment , the designed goal may be to achieve a noise spectrum that compiles with government requirements ; in an outdoor environment, the design goal may be a noise spectrum that satisfies a local noise ordinance ; in an office environment, the design goal usually a noise spectrum that provides accepted standards for comfort and speech communication . Typical design goals for various occupancies are given in table 3.1 which may be used with the noise control curves given in figure 1.1. In this example, a design goal of NC 35 was selected, whose octave–band spectrum is plotted in figure 3.5.

Type of room	Recommended		Approximate A-weighted sound level dB
Preferred		Alternative	
Recording studios	RC 10-20(N)	NC 10-20	18-28
Concert and recital halls	RC 15-20(N)	NC 15-20	23-28
TV studios, music rooms	RC 20-25(N)	NC 20-25	28-33
Legitimate theaters	RC 20-25(N)	NC 20-25	28-33
Private residences	RC 25-30(N)	NC 25-30	33-38
Conference rooms	RC 25-30(N)	NC 25-30	33-38
Lecture rooms, classrooms	RC 25-30(N)	NC 25-30	33-38
Executive offices	RC 25-30(N)	NC 25-30	33-38
Private offices	RC 30-35(N)	NC 30-35	38-43
Churches	RC 30-35(N)	NC 30-35	38-43
Cinemas	RC 30-35(N)	NC 30-35	38-43
Apartments, hotel bedrooms	RC 30-35 (N)	NC 30-35	38-43
Courtrooms	RC 35-40(N)	NC 35-40	43-48
Open-plan offices and	RC 35-40(N)	NC 35-40	43-48
Libraries	RC 35-40(N)	NC 35-40	43-48
Lobbies, public areas	RC 35-40(N)	NC 35-10	43-48
Restaurants	RC 40-45(N)	NC 40-45	48-53
Public offices (large)	RC 40-45(N)	NC 40-45	48-53

Table 3.1 Acceptability criteria for steady background noise in unoccupied rooms

STEP 3 **- Determine the noise required in each octave-band in order to achieve the design goal**

The required noise reduction is given by the difference between (1) The octave band spectrums for the total noise at the receiver and no (2) the octave band spectrum for the design goal.

STEP 4 - **Evaluate the various options that are available for achieving the required noise reduction determined in step 3**

The design goal should be achieved as economically as possible, both the direct costs and the indirect costs should be considered in evaluating each possible solutions. In addition to the cost of a solution, it's possible adverse effects of in terms of operational restrictions must be considered

CHAPTER 4

SOURCES OF NOISE IN HYDRAULIC TURBINES

4.1ELECTRIC MOTOR NOISE [19]

Electric motors or machines are used to convert electrical energy into mechanical energy and may be subdivided into asynchronous, synchronous, and direct current machines (dc) machines. The sources of noise and vibration in electric machines are the following:

- Electromagnetic forces
- Bearings
- Aerodynamic forces
- Imbalance and rubbing

Electromagnetic forces are active in air gap between the stator and rotor and are characterized by rotating or pulsating power waves. In most types of electric machines, the magnetic vibration is in the range of 100 Hz to 4000 Hz. The intensity of noise and vibration in bearing is determined by the quality of manufacture, the accuracy of the machining of the bearing seats, and the vibroacoustic properties of the end brackets.

Noise based on aerodynamic forces depends upon the construction of fan and ventilation channels of the machine. Mechanical imbalance of rotors may excite an appreciable amount of vibration, particularly in high speed machines. The rubbing of the brushes against the commutator or contact rings is predominantly a high frequency source of noise. The vibration and noise prediction and control methods of electromagnetic origin are based on the diagnostics of electrical motor defects using spectrum analysis of the vibration signal.

4.2 PUMP AND PUMPING SYSTEM NOISE AND CONTROL

Hydraulic pumps are used for transporting chilled, hot, and condenser water or any other liquid in hydraulic systems. They are also used for transporting suspensions of liquid and solid particles from one location to another through pipes or closed/opened channels and passage. Common applications include use in processing plants, hydraulic fluid power, clean water supply, ships and the like. Operation of a pump creates pressure pulsations, vibration, and noise, which can be spread by pipes (structure borne noise) and by liquid (fluid borne noise) far away from the pump itself, emitting noise (from pipes) around the whole pumping system.

Noise generated by the pump depends upon its:

- Size and type
- Power
- Operating conditions (speed and load)

Noise of pumping system depends on its geometry and type and number of the built in fittings and armatures. Noise and vibration of hydraulic system can be controlled by proper selection or redesigning of the pump and/or pumping system and by optimizing of the operating conditions. Tuning of the operating frequency with regard to the resonance frequencies of the pumping system and by proper balancing of all rotating masses, and the like is important.

4.2.1 TYPES OF PUMPS:

Pumps are classified according to principle of adding energy to fluid into three main groups

- Kinetic pumps: those pumps in which energy is continuously added to the pumped liquid within the pump first by increasing the fluid velocity and then toward the end of the pump by increasing the delivery pressure. These are subdivided into centrifugal or radial flow, diagonal or mixed flow, and axial flow or propeller pumps.
- Reciprocating pumps: they are self priming pumps and are used for high pressure and low capacity. The piston or plunger or diaphragm are driven through slider crank mechanisms and crank shaft from an external source. Therefore they have a pulsating discharge. The flow rate is varied by changing either the rotating speed or the stroke length.
- Rotary pumps- They are divided into pumps with single rotor and a multiple rotor. They are also self priming, but they do not need valves or a crankshaft. Therefore they are relatively small and due to relative high efficiency are appropriate for pumping of liquids with high viscosity at medium pressures ad capacities.

4.3 HYDRAULIC AND MECHANICAL SOURCES OF NOISE [19]

The noise generated by hydraulic pumping system consists of the noise generated by pump itself, by the driving motor, usually a fan cooled electric motor and that generated by the pumping system and is mainly hydraulic and partially mechanical in origin. Both types of noise origins are transmitted through the system over the fluid and / or structure as structure borne sound before exciting the surrounding air as reaching us a noise. A pump as a prime mover is only part of pumping system; therefore, it is necessary to distinguish between by the noise generated by the pump and pumping system. Noise generated by a pump depends on the pump type, on its geometry and on the operating conditions.

4.3.1 Noise in kinetic pumps: The main sources of noise in kinematic pumps are interaction between the impeller blade and the volute tongue or the diffuser vanes in centrifugal pumps, and between the impeller blades and the guides' vanes in axial flow pumps so called blade passage frequency (BPF). Followed by pressure fluctuations caused by turbulence, flow friction, flow separation and vortices in the radial and axial clearances, specially between the open and semi-open rotor and the adjacent stationary part of casing.

4.3.2 Noise in reciprocating pumps: Major sources of noise in reciprocating pumps are piston induced pulsation, piston mechanical reaction and impacts in inlet and discharge valves, which are exerted by the up and down motion of a piston or plunger. Intermittent liquid flow and fluctuating dynamic forces and machinery imbalance of the reciprocating and rotating parts (piston, crank shaft, etc) are then sources of intensive noise. Additional sources of noise in reciprocating pumps are turbulence, flow friction, vortex formation from separated flow around obstruction, piston slap in a form of impact excitation, which is a result of clearance between the piston and cylinder valve and the high pressure that pusses on the top of the piston during compression strokes.

4.3.3 Noise in rotary pumps: Major sources of noise in rotary pumps are impact between teeth in gear pumps, friction between screws at the screws pumps and the sliding of vanes on the casing in vanes rotary pump. When the cavities between the gear teeth are not filled completely by the incoming liquid, the resulting vortices in voids generate a distinct noise, similar in nature to flow separation in centrifugal pumps during recirculation. Additional sources of noise are turbulence and flow friction in the compression spaces. The magnitude and frequency of the noise vary from pump to pump and are dependent on the magnitude of the pump head being generated and the amount by which pump flow at off design operation deviates from ideal flow at the design or best efficiency point (BEP).

4.3.4 Cavitation: Cavitation in pumps is also the major source of noise. It occurs when the absolute static pressure at some point with in a pump or pumping system false below the saturated vapour pressure of the fluid at the prevailing temperature condition; the fluid starts to flash and vaporization occurs. Vaporization of the flowing fluid is connected with the onset of bubbles. The bubbles are caught up by the flowing liquid and collapse within the pump (valve or piping system) when they reach a region of higher pressure, where they condense. This process is accompanied by a violent collapse or implosion of the bubbles and a tremendous increase in pressure, which has a character of water hammer blows. The process of cavitations and bombardment of the pump surface by the bursting bubbles causes three different undesired effects.

- Deterioration of the hydraulic performance of the pump(total delivery head, capacity and efficiency)
- Possible pitting and material erosion in the vicinity of the collapsing bubbles.
- Vibration of the pump walls excited by the pressure and flow pulsations and resultant undesirable noise.

4.4 SOURCES OF NOISE IN PUMP COUPLING AND SUPPORTING ELEMENTS:-

A major source of externally induced vibration and noise is miss alignment between the pump and its driver, which depend on the coupling used. There are two primary groups of coupling: rigid and flexible. Rigid coupling are used for direct coupling and for precise alignment whereas the flexible coupling are used for accommodating a set of misalignment (angular, axial and radial or parallel) between the driving and driven shaft, and also radial and axial loads on the motor bearings. Rigid couplings are in sleeve type of flange type and are used when the pump impeller are mounted directly on the motor shaft. Flexible couplings are may be mechanically flexible and material flexible. The mechanically flexible couplings are in gear or grid form which needs lubrication. The material flexible couplings have a flexible element in steel, rubber, elastomer, or plastic material and do not need lubrication. They are in the form of metal disk or disk pack or in the form of contoured diaphragm or as an elastomer in compression or sheers type with rubber jaws on driving and driven shaft corresponding to the number of cogs belt, or a central spider on the flexible element with several radial segments.

There are also special types of coupling to transmit power with independent input and output shaft that allow desired adjustment of load speeds. Such coupling are the eddy current slip couplings and fluid couplings (in hydrokinetic, hydrodynamic, hydro viscous, or hydrostatic form) using fluid (natural or synthetic oil).

A second source of externally induced vibration noise at a pump is supporting bearings, with which it is equipped. The hydraulically and mechanically produced noise and other failings of the pump rotating and moving elements are transmitted over the bearing to the pump housing and as structural noise throughout the pumping system. Most pumps are equipped with external bearings in a classical three-bearing arrangements, journal or sleeve type (with hydrodynamic fluid film), antifriction (rollers, tapered rollers, or bearings in single or multi rows), and magnetic bearings. At normal operation of a general bearing the shaft form a liquid film of the lubrication oil completely separates it from the bearing. The anti friction bearing operate with a very low coefficient of friction associated with rolling motion as distinct from sliding motion. To minimize the heat generated by sliding friction, anti friction bearing require oil or grease lubrication. The magnetic bearing are free from contact within the bearings, therefore they produce the lowest mechanical noise among all type of bearings follow the general or sleeve antifriction bearing, which are the nosiest.

Vibration problem and noise in pump increase when a failure in the bearings or couplings misalignment occurs. Bearings failure may be caused by water or product contamination in the bearing housing or by highly unbalance radial load, which can result from operating at or near shut off. Nonconstant friction forces usually are caused by poor lubrication, at low shaft speed or low fluid viscosity in general bearings, when the strength of liquid film is insufficient or support the load on the bearings and the shaft rubbing the bearing, or by unfavorable combinations of sliding interface material, geometric arrangements, and any sliding velocities. Damage of the rolling bearing causes vibration at the integer number of rotational frequency, whereas the oil whirl in hydrodynamic bearing causes vibration at a friction of rotational speed. Poor lubrication and defect in the rolling element or races in antifriction bearings exhibit very high frequency response and noise of about 20 kHz.

CHAPTER 5

HYDRAULIC TURBINES

A turbine is a rotary engine that extracts energy from a fluid or air flow and converts it into useful work. The simplest turbines have one, moving part, a rotor assembly, which is a shaft or drum, with blades attached. Moving fluid acts on the blades, or the blades react to the flow, so that they move and impart rotational energy to the rotor. Early turbine examples are windmills and water wheels.

5.1 TYPES OF TURBINES [29]

- Steam turbines are used for the generation of electricity in thermal power plants, such as plants using coal or fuel oil or nuclear power. They were once used to directly drive mechanical devices such as ships' propellers, but most such applications now use reduction gears or an intermediate electrical step, where the turbine is used to generate electricity, which then powers an electric motor connected to the mechanical load.
- Gas turbines are sometimes referred to as turbine engines. Such engines usually feature an inlet, fan, compressor, combustor and nozzle (possibly other assemblies) in addition to one or more turbines.
- Transonic turbine. The gas flow in most turbines employed in gas turbine engines remains subsonic throughout the expansion process. In a transonic turbine the gas flow becomes supersonic as it exits the nozzle guide vanes, although the downstream velocities normally become subsonic. Transonic turbines operate at a higher pressure ratio than normal but are usually less efficient and uncommon.
- Stator less turbine. Multi-stage turbines have a set of static (meaning stationary) inlet guide vanes that direct the gas flow onto the rotating rotor blades. In a stator less turbine the gas flow exiting an upstream rotor impinges onto a downstream rotor without an intermediate set of stator vanes (that rearrange the pressure/velocity energy levels of the flow) being encountered.
- Ceramic turbine. Conventional high-pressure turbine blades (and vanes) are made from nickel based alloys and often utilizes intricate internal air-cooling passages to prevent the metal from overheating. In recent years, experimental ceramic blades have been manufactured and tested in gas turbines, with a view to increasing Rotor Inlet

Temperatures and/or, possibly, eliminating air cooling. Ceramic blades are more brittle than their metallic counterparts, and carry a greater risk of catastrophic blade failure. This has tended to limit their use in jet engines and gas turbines, to the stator (stationary) blades.

- Shrouded turbine. Many turbine rotor blades have shrouding at the top, which interlocks with that of adjacent blades, to increase damping and thereby reduce blade flutter. In large land-based electricity generation steam turbines, the shrouding is often complemented, especially in the long blades of a low-pressure turbine, with lacing wires. These are wires which pass through holes drilled in the blades at suitable distances from the blade root and the wires are usually brazed to the blades at the point where they pass through. The lacing wires are designed to reduce blade flutter in the central part of the blades. The introduction of lacing wires substantially reduces the instances of blade failure in large or low-pressure turbines.
- Shroudless turbine. Modern practice is, wherever possible, to eliminate the rotor shrouding, thus reducing the centrifugal load on the blade and the cooling requirements.
- Bladeless turbine uses the boundary layer effect and not a fluid impinging upon the blades as in a conventional turbine.
- Water turbines
    - Pelton turbine, a type of impulse water turbine.
    - Francis turbine, a type of widely used water turbine.
    - Kaplan turbine, a variation of the Francis Turbine.
- Wind turbine. These normally operate as a single stage without nozzle and interstage guide vanes. An exception is the Éolienne Bollée, which has a stator and a rotor, thus being a true turbine.

5.2 USES OF TURBINES [27]

Almost all electrical power on Earth is produced with a turbine of some type. Very high efficiency steam turbines harness about 40% of the thermal energy, with the rest exhausted as waste heat.

- Most jet engines rely on turbines to supply mechanical work from their working fluid and fuel as do all nuclear ships and power plants.

- Turbines are often part of a larger machine. A gas turbine, for example, may refer to an internal combustion machine that contains a turbine, ducts, compressor, combustor, heat-exchanger, fan and (in the case of one designed to produce electricity) an alternator. However, it must be noted that the collective machine referred to as the turbine in these cases is designed to transfer energy from a fuel to the fluid passing through such an internal combustion device as a means of propulsion, and not to transfer energy from the fluid passing through the turbine to the turbine as is the case in turbines used for electricity provision etc.
- Reciprocating piston engines such as aircraft engines can use a turbine powered by their exhaust to drive an intake-air compressor, a configuration known as a turbocharger (turbine supercharger) or, colloquially, a "turbo".
- Turbines can have very high power density (ie the ratio of power to weight, or power to volume). This is because of their ability to operate at very high speeds. The Space Shuttle's main engines use turbo pumps (machines consisting of a pump driven by a turbine engine) to feed the propellants (liquid oxygen and liquid hydrogen) into the engine's combustion chamber. The liquid hydrogen turbo pump is slightly larger than an automobile engine (weighing approximately 700 lb) and produces nearly 70,000 hp (52.2 MW).
- Turbo expanders are widely used as sources of refrigeration in industrial processes.

5.3 HYDRAULIC OR WATER TURBINES [23]

Hydraulic turbines are the machines which use the energy of water and than convert it into mechanical energy. Hydraulic Turbines transfer the energy from a flowing fluid to a rotating shaft. Turbine itself means a thing which rotates or spins. Hydraulic Turbines can transfer the energy from flowing water to the shafts of dynamos producing electrical power.

Hydraulic Turbines have a row of blades fitted to the rotating shaft or a rotating plate. Flowing liquid, mostly water, when pass through the Hydraulic Turbine it strikes the blades of the turbine and makes the shaft rotate. While flowing through the Hydraulic Turbine the velocity and pressure of the liquid reduce, these result in the development of torque and rotation of the turbine shaft. There are different forms of Hydraulic Turbines in use depending on the operational requirements. For every specific use a particular type of Hydraulic Turbine provides the optimum output.

5.4 CLASSIFICATION OF HYDRAULIC TURBINES: BASED ON FLOW PATH

Water can pass through the Hydraulic Turbines in different flow paths. Based on the flow path of the liquid Hydraulic Turbines can be categorized into three types.

1. **Axial Flow Hydraulic Turbines:** This category of Hydraulic Turbines has the flow path of the liquid mainly parallel to the axis of rotation. Kaplan Turbines has liquid flow mainly in axial direction.
2. **Radial Flow Hydraulic Turbines:** Such Hydraulic Turbines has the liquid flowing mainly in a plane perpendicular to the axis of rotation.
3. **Mixed Flow Hydraulic Turbines:** For most of the Hydraulic Turbines used there is a significant component of both axial and radial flows. Such types of Hydraulic Turbines are called as Mixed Flow Turbines. Francis Turbine is an example of mixed flow type, in Francis Turbine water enters in radial direction and exits in axial direction.

None of the Hydraulic Turbines are purely axial flow or purely radial flow. There is always a component of radial flow in axial flow turbines and of axial flow in radial flow turbines.

5.5 CLASSIFICATION OF HYDRAULIC TURBINES: BASED ON PRESSURE CHANGE

One more important criterion for classification of Hydraulic Turbines is whether the pressure of liquid changes or not while it flows through the rotor of the Hydraulic Turbines. Based on the pressure change Hydraulic Turbines can be classified as of two types.

1. **Impulse Turbine:** The pressure of liquid does not change while flowing through the rotor of the machine. In Impulse Turbines pressure change occur only in the nozzles of the machine. Example of impulse turbine is:

- Pelton

2. **Reaction Turbine:** The pressure of liquid changes while it flows through the rotor of the machine. The change in fluid velocity and reduction in its pressure causes a reaction on the turbine blades; this is where from the name Reaction Turbine may have been derived. Examples of reaction turbines are:

- Francis
- Kaplan
- Water wheel

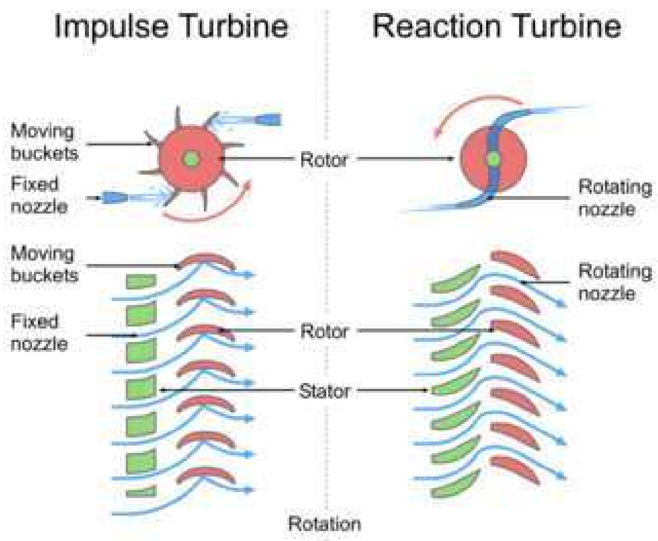


Fig 5.1 Comparison of impulse and reaction turbine [27]

5.6 FRANCIS TURBINE

Francis Turbine is the first hydraulic turbine with radial inflow. It was designed by American scientist James Francis. Francis Turbine is a reaction turbine. Reaction Turbines have some primary features which differentiate them from Impulse Turbines. The major part of pressure drop occurs in the turbine itself, unlike the impulse turbine where complete pressure drop takes place up to the entry point and the turbine passage is completely filled by the water flow during the operation.

5.7 Design and working of Francis Turbine [23]

The Francis turbine is a mixed flow type in which water enters the runner radialy at its outer periphery and leaves axially at its center. The water from the penstock enters a scroll casing (also called spiral casing) which completely surrounds the runner. The purpose of the casing is to provide an even distribution of water around the circumference of the turbine runner, maintaining and approximately constant velocity for the water so distributed. In order to keep the velocity of water constant throughout its path around the runner, the cross sectional area of the casing is gradually decreased.

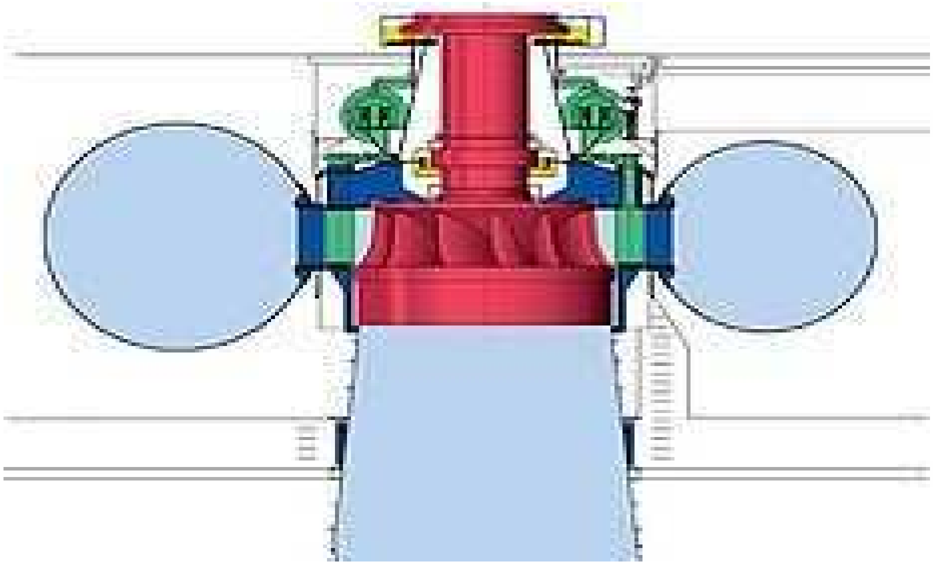


Fig 5.2 (a) Francis turbine [28]

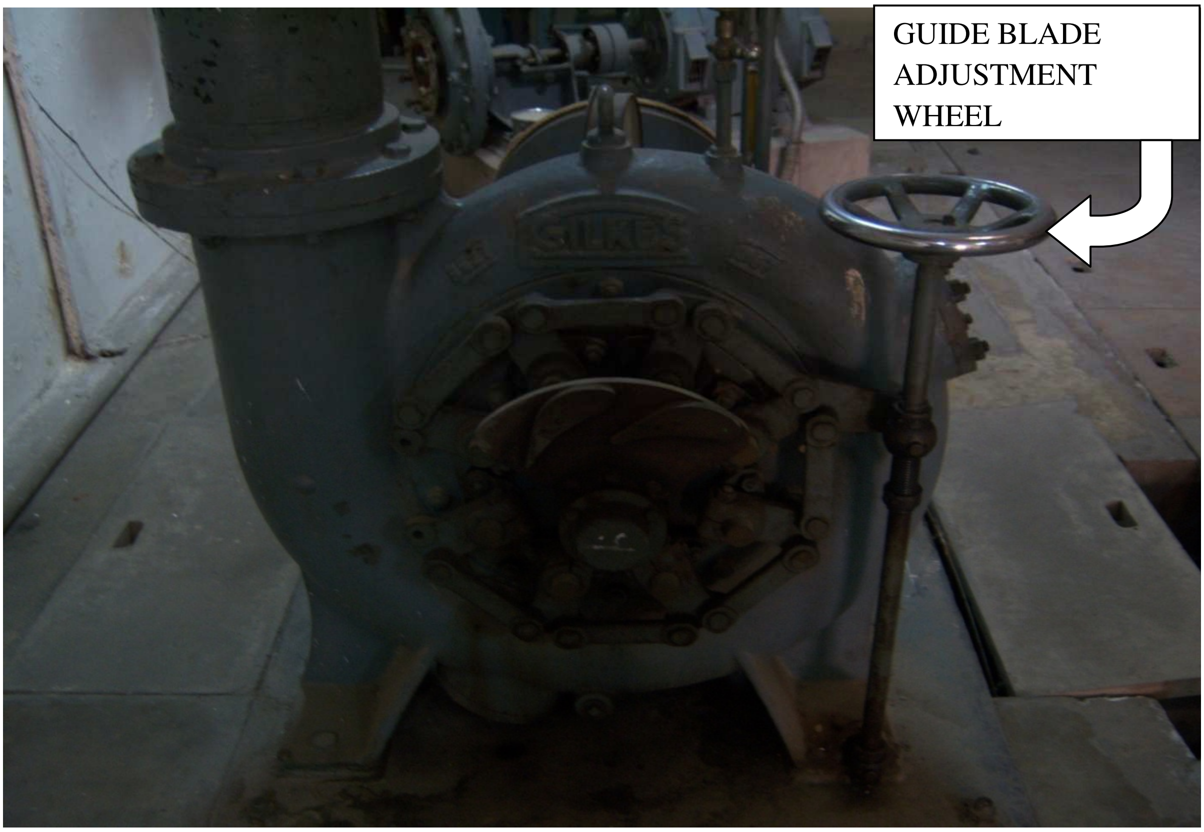


Fig 5.2 (b) Francis turbine

The casing is made up of cast steel, plate steel, concrete or concrete and steel depending upon the pressure to which it is subjected. Out of these a plate steel scroll casing is commonly provided for turbines operating under 30 m or higher heads. From the scroll casing the water passes through a speed ring or stay ring (see fig.5.1(a)(b)) the speed ring is consists of an upper and lower ring held together by series of fixed vanes called stay vanes. The number of stay vanes is usually taken as half the number of the guide vanes. The speed ring has two functions to perform. It directs the water from scroll casing to the guides vanes or wicket gate. Further it resists the load imposed upon it by the internal pressure of water and the weight of the turbine and the electrical generator and transmits the same to the foundations. The speed ring may be either of cast iron or cast steel or fabricated steel.

From the speed ring water passes through a series of guide vanes or wicket gates provided all around the periphery of the turbine runner. The function of the guide vanes is to regulate the quantity of water supplied to the runner and to direct water on to the runner at the angle appropriate to the design. The guide vanes are air foiled shape and may be of cast steel, stainless steel or plate steel. Each guide vane is provided with two stems, upper stem is passes through the head cover and the lower stem seats in a bottom ring. By the system of lever and links, all the guide vanes may be turned about their stems, so as to alter the weight of the passes between the adjacent guide vanes, thereby allowing a variable quantity of water to strike the runner. The guide vanes are operated either by a means of a wheel (for very small units) or automatic by a governor.

The main purpose of the various components so far describe is to lead the water to the runner with a minimum loss of energy. The runner of a Francis turbine consists of a series of a curved vanes (16-24 in number) evenly arrange around the circumference in the annular space b/w the two plates. The vanes are so shaped that water enters the runner radially at the outer periphery and leaves it axially at the inner periphery. The change in the direction of flow of water , from radial to axial, as it passes through the runner, produces a circumferential force on the runner which makes the runner to rotate and thus contributes to the useful output of the runner. The runner is usually made up of cast iron, cast steel or mild steel or stainless steel. Often instead of making the complete runner of stainless steel, only those portions of the runner blades which may be subjected to the cavitation erosion are made of stainless steel. This reduces the cost of runner and at the same time ensures the operations of the runner with a minimum amount of maintenances. The runner is kept to a shaft which is usually forged steel. The torque produced by the runner is transmitted to the generator through the shaft which is usually connected to the generator shaft by a bolted flange connections.

The water after passing through the runner flows to the tail races through a draft tube. A draft tube is a pipe or a passes of gradually increase in cross sectional area which connects the runner exit to the tail race. It may be made up of cast iron or plate steel or concrete. It must be air tight and under all conditions of operations its lover end must be submerged below the level of water in the tail race.

CHAPTER 6

DESIGNING OF ACOUSTIC ENCLOSURE

ACOUSTIC ENCLOSURE:

As discussed in previous chapter to control the noise of pumping system completely it is needed to design an acoustic enclosure. There is need to provide ventilation to prevent the overheating of the motor. Before to fabricate a functional design is to be made on the bases of following fundamentals

6.1 THE USE OF SOUND BARRIERS IN ACOUSTIC ENCLOSURE [22]

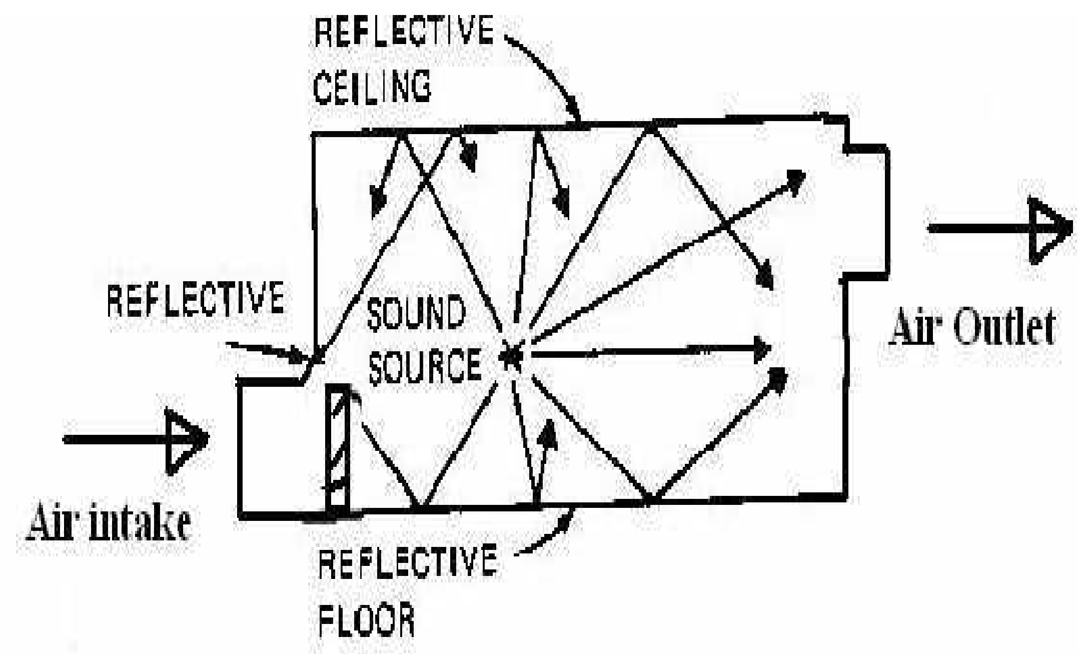
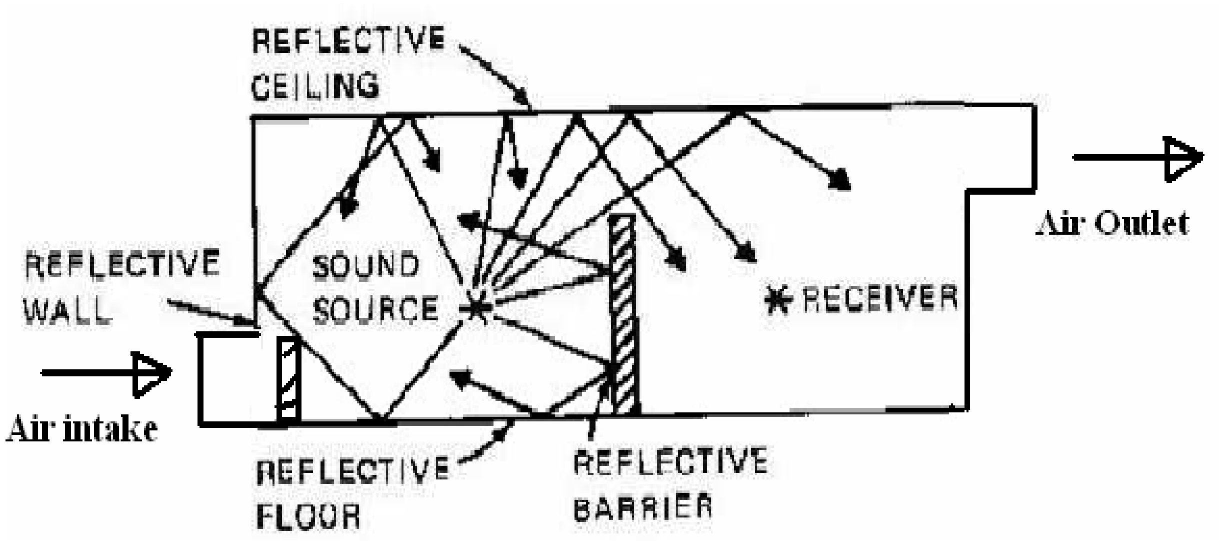


Fig 6.1(a).Inside an acoustic enclosure sound will travel through air ventilation

Which sound from a noise sources travels to the receiver by a direct path and also by a multiplicity of reflections from the sides of enclosure, ceiling, and floor.



In Fig 6.1 (b) a solid reflective barrier is installed, which is effective in eliminating the direct sound; however, it increases (by reflections) the sound reaching the receiver that is not shielded by the barrier. The effectiveness of the barrier thus is greatly compromised by the reflected sound.

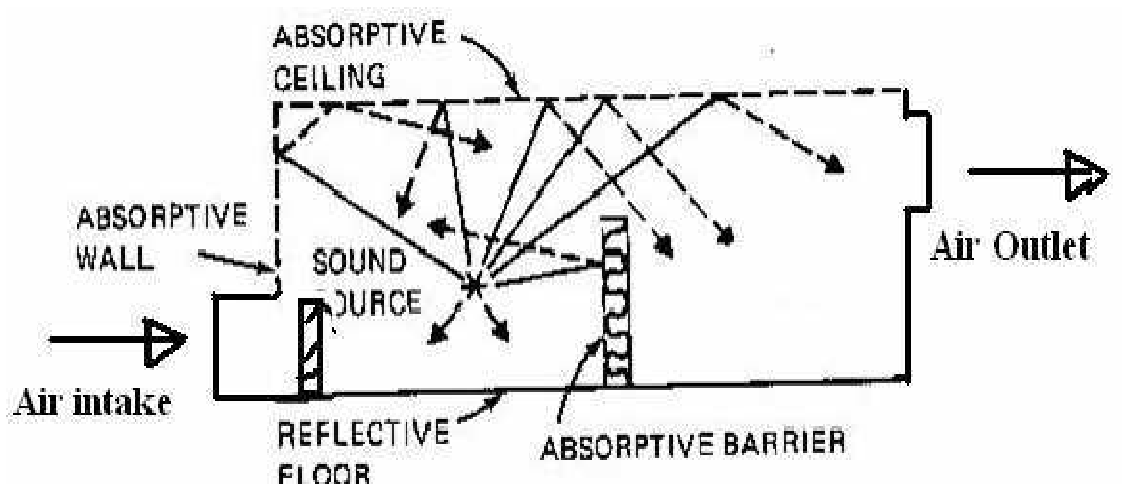


Fig 6.1(c) with absorptive material

In Fig 6.1 (c), sound absorptive material has been installed on the barrier. Under these conditions, the reflected paths are greatly attenuated and the maximum noise reduction is achieved.

6.2 RULES FOR USE OF BARRIERS INSIDE ACOUSTIC ENCLOSURE

- Place the barriers as close to the source as possible - but not in contact with the source.
- Extend the barrier beyond the line of sight of the source (both vertically and horizontally) by one-quarter wavelength of the lowest frequency for which significant attenuation is required.
- Select a solid barrier material (with no openings or holes) having a sound transmission loss at least 10dB higher than the required attenuation.

6.3 COMPLETE ENCLOSURES AROUND A SOURCE

The purpose of a complete enclosure is to contain and absorb the acoustic energy radiated by the source as shown in fig 6.2. This reduces the sound pressure levels at all distances from the source. The amount of noise reduction may approach the transmission loss of the material from which the enclosure is fabricated.

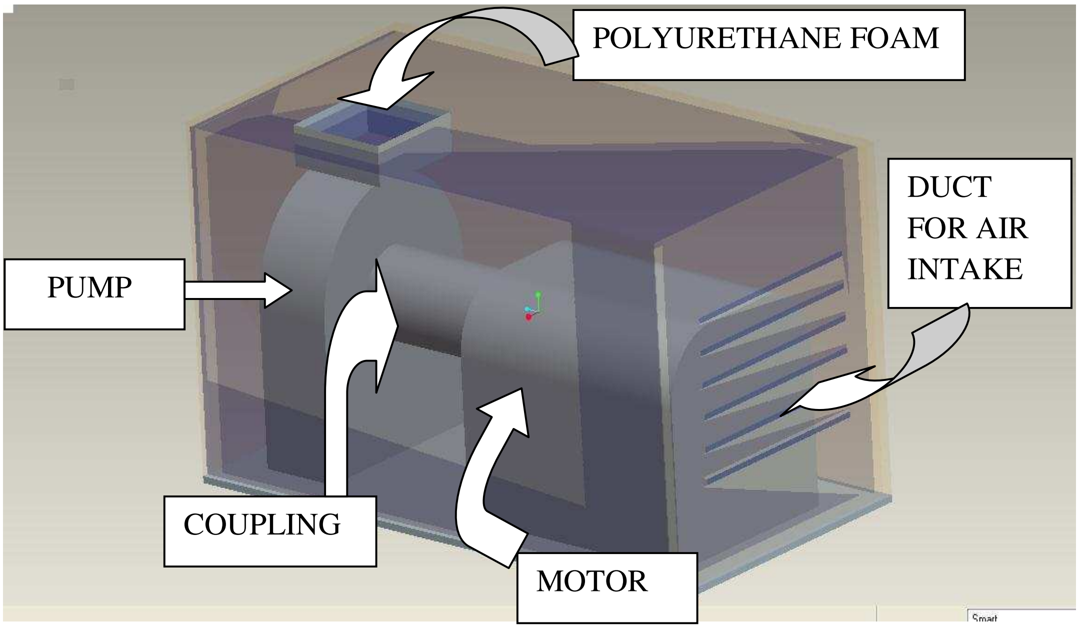


Fig 6.2(a) Designed view of enclosure

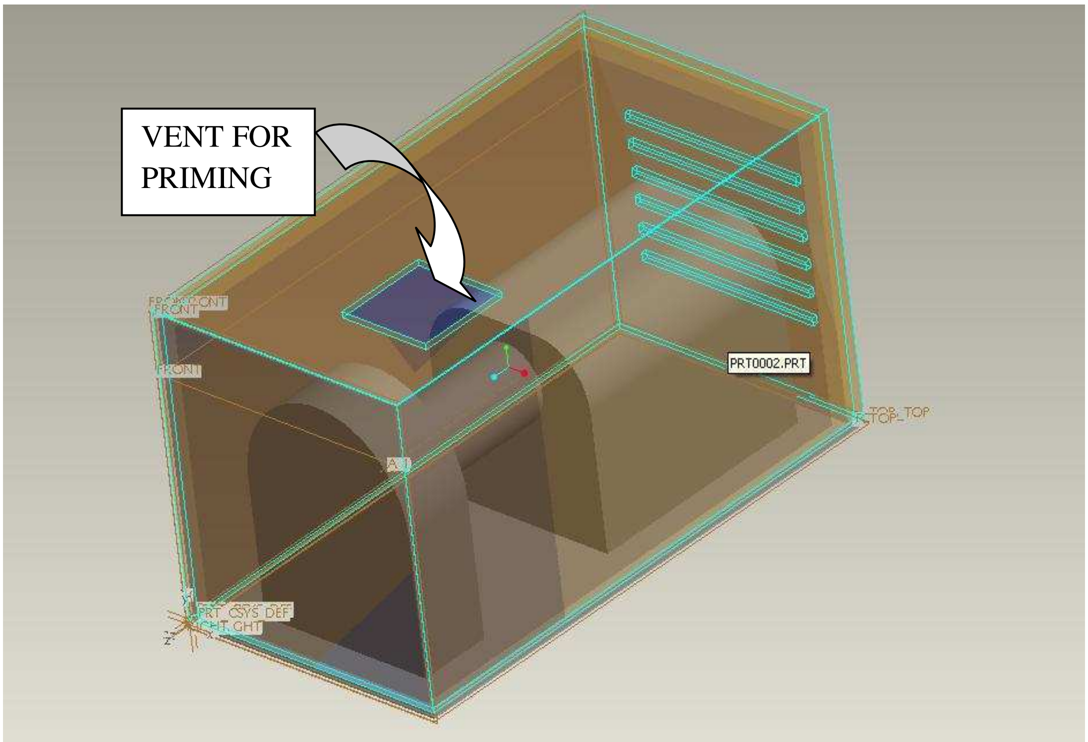


Fig 6.2(b) Designed view of enclosure(standard view)

The importance of using a solid rather than a porous material for the walls of the enclosure is illustrated in Fig. 6.3 although the performance of the solid enclosure is much better than that of the porous enclosure; it is limited by multiple reflections within the enclosure. This process causes the internal sound pressure levels to build up until a steady state is achieved within the enclosure. The amount of buildup depends on the dimensions of the enclosure and the sound absorptive characteristics of the enclosure, but a buildup of 10 to 20dB is not unusual. This greatly reduces the effectiveness of the enclosure. For this reason an enclosure should be lined to the maximum extent possible with sound absorptive material. Because of this buildup, the transmission loss (TL) of the wall material must be greater than the noise reduction (NR) required at each frequency of interest..

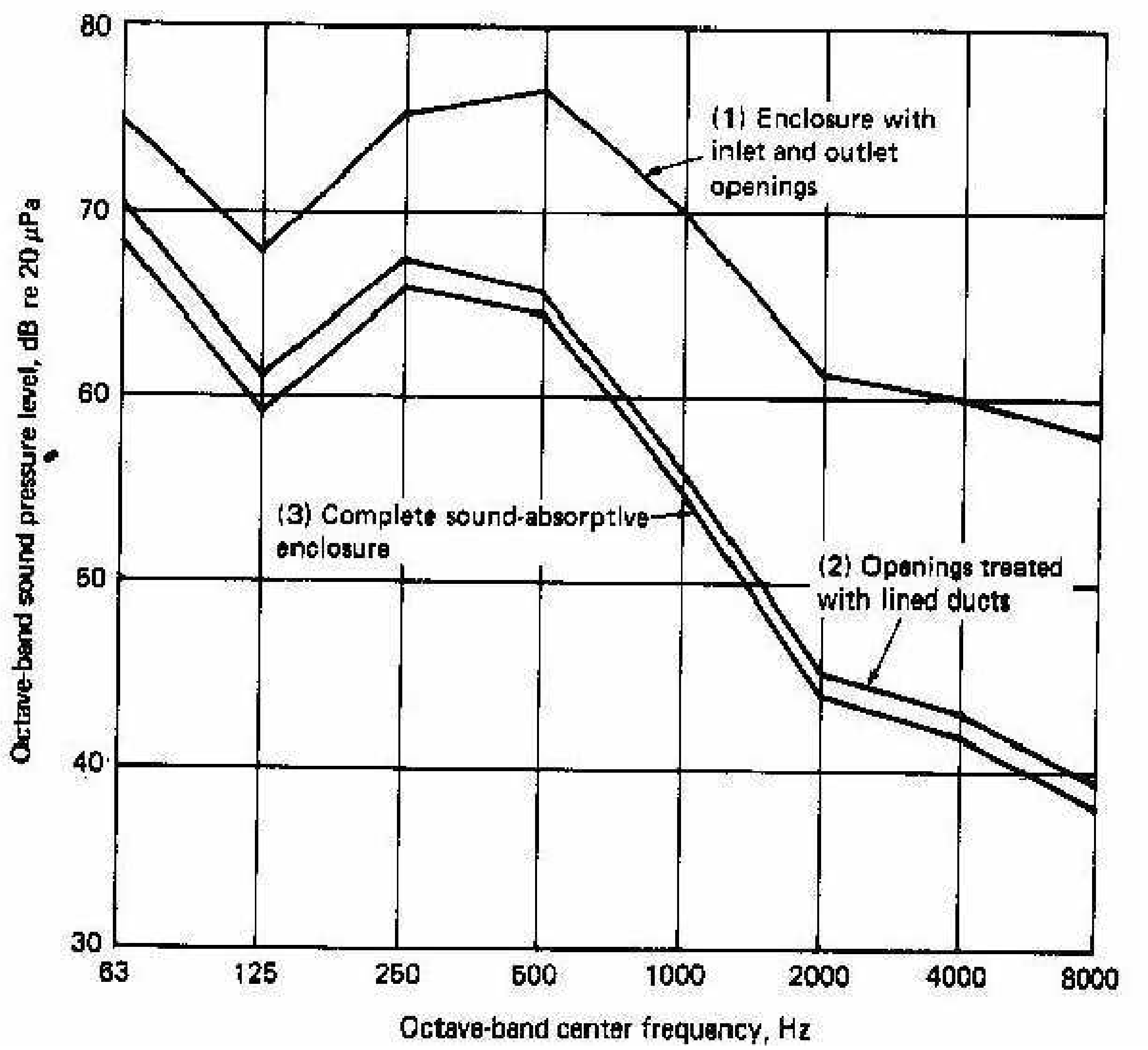


Fig 6.3 octave band 1/1 frequency spectrum

In some cases, the housing (or cabinet) of a machine. May serve as an effective enclosure for noise sources that are contained within the equipment. In this case the addition of a sound absorptive lining (and the sealing of leakage paths) may provide significant noise reduction.

Rules of Thumb

If enclosure has no internal absorption, then

$$TL > NR + 20$$

If enclosure has partial internal absorption, then

$$TL > NR + 15$$

If enclosure has complete internal absorption, then

$$TL > NR + 10$$

6.4 USE OF LINED DUCTS FOR VENTILATION OPENINGS

Complete enclosures around noisy equipment may require ventilation to prevent overheating of the equipment. The effect of inlet and outlet openings (for ventilation of the cabinet shown in Fig. 6.4 (a) is illustrated in Fig. 6.4 (b). The openings substantially reduce the attenuation provided by the complete enclosure with absorption.

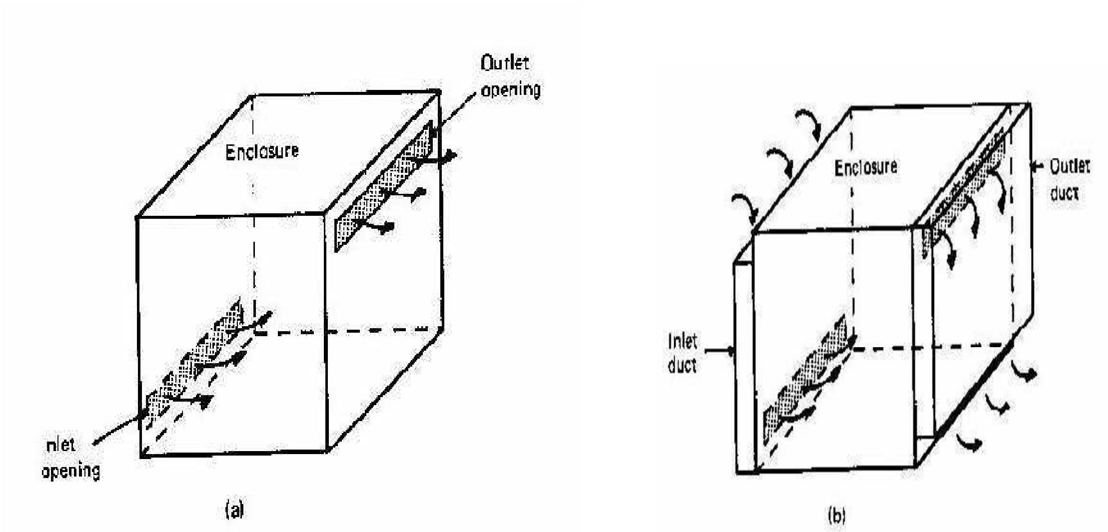


Fig 6.4 (a) Untreated inlet out let opening, (b) opening with ducts lined with sound absorptive material

6.5 SPECIFIC DATA, LOCATION OF COMPLETE SET UP USED IN EXPERIMENT

6.5.1 SPECIFIC DATA, LOCATION OF PUMPING SYSTEM USED IN EXPERIMENT

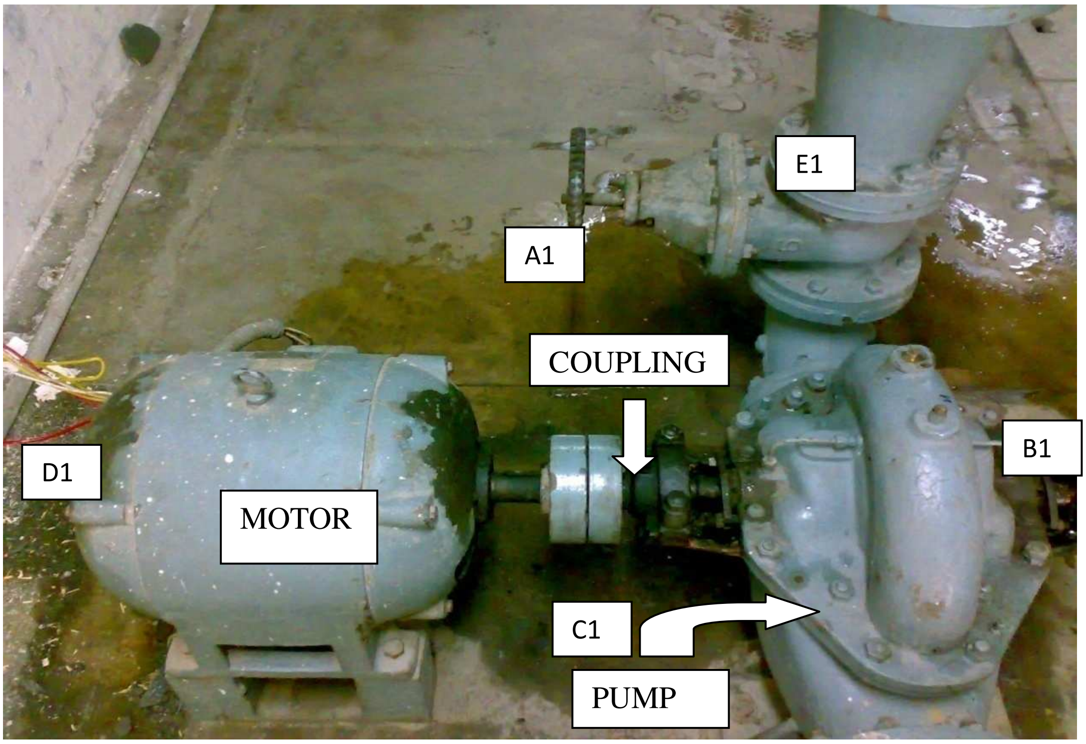


Fig 6.5 An overview of pumping system

1.Motor Detail	Location	Manufactu-rer	Type	H.P	R.P.M	Voltage (Volts)	Current (Ampere)
	Thapar university Patiala. (Fluid machines Lab)	Kirloskar Electric Co. Ltd. Banglore	Induction Motor	15	1440	400/440	19

Table 6.1 specific data, location of pumping system used in experiment

6.5.2 SPECIFIC DATA, LOCATION OF FRANCIS TURBINE USED IN EXPERIMENT

Fig 6.6 An overview of turbine set up used in experiment

1.Turbine Detail	Location	Manufacturer	Type of Turbine	Power (KW)	Size of impeller of pump	Diameter of penstock	Vane opening
	Thapar university Patiala. (Fluid machines Lab)	GILKES, Kendal, England	Francis	3.75	15.24 cm	20.32 cm	0, 1/4, 1/2, 3/4, Full.

Table 6.2 specific data, location of turbine set up used in experiment

6.7 MATERIAL USED FOR FABRICATION OF ACOUSTIC ENCLOSURE:

6.1 Wood in the form of ply board

6.2 Steel nails of different length and thickness

6.3 Polyurethane foam.

6.4 Fevicol

6.5 Highly adhesive material for pasting polyurethane foam with ply board

Major sound absorption is done by polyurethane foam.

6.7.1 SPECIFICATION OF POLYURETHANE FOAM [27]

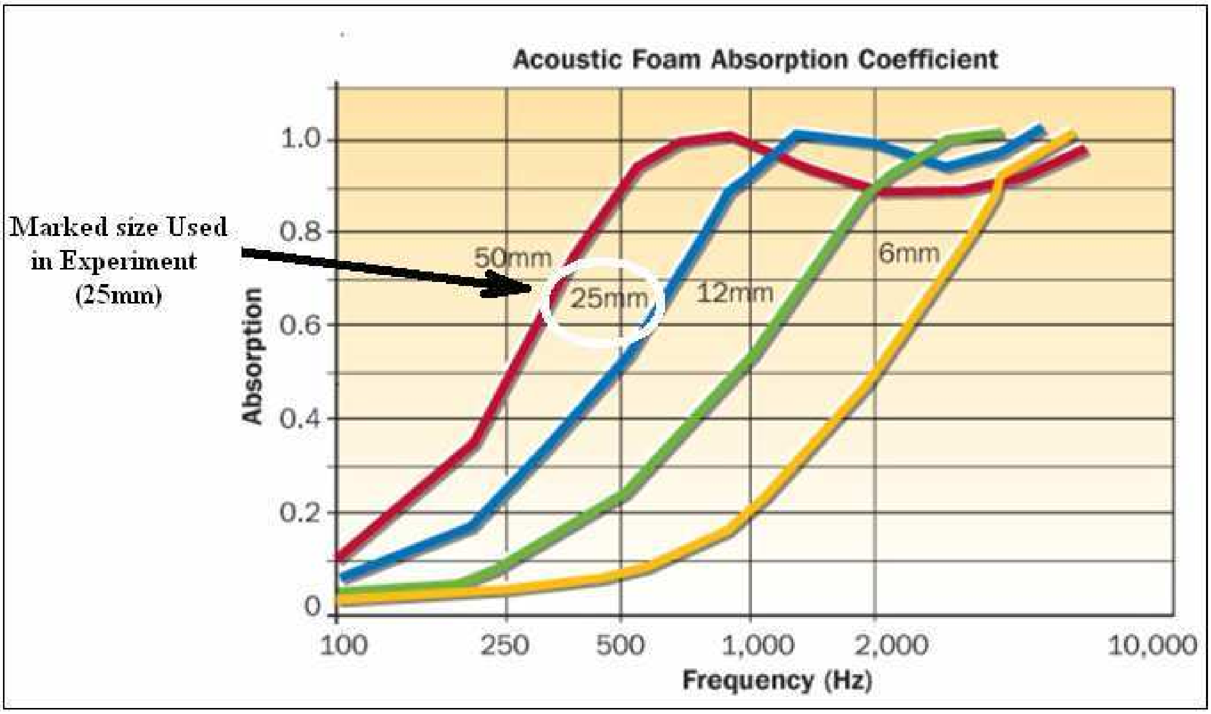


Fig 6.7

Specification's of poly-urethane foam	Sheet Size	Piece Density	Tensile Strength	Flammability
	1500 x 1000mm	27-30 kg/m^3	100 kN/m^2	FMVSS 302 Self-Extinguishing

Table 6.3

6.8 AN OVERVIEW AND DIMENSIONS OF ENCLOSURE

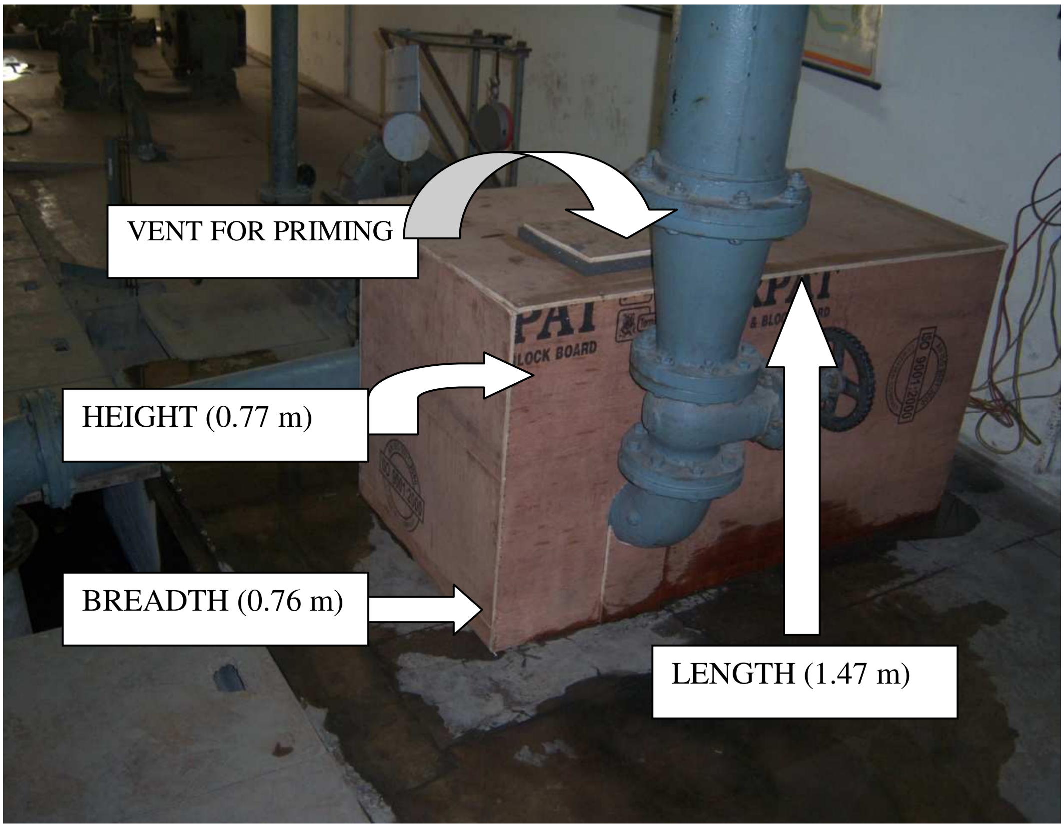


Fig 6.8 An overview of enclosure

Dimensions ⟶	Length	Breadth	Height
Enclosure	1.47 m	0.76 m	0.77 m
Vent for priming	0.26 m	0.26 m	-

Table 6.4 Dimension's of enclosure

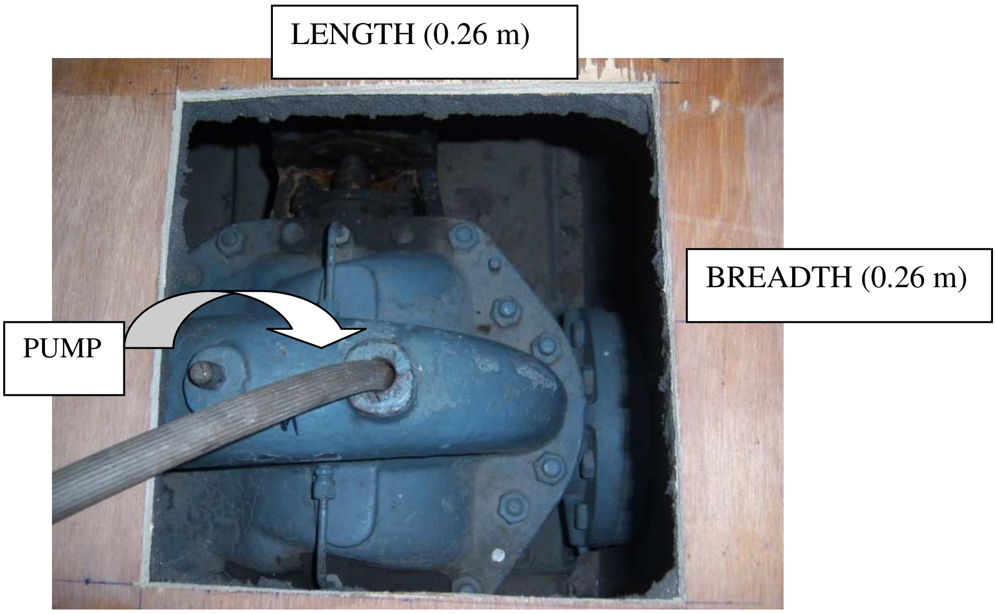


Fig 6.9 An overview of vent showing priming

CHAPTER 7

GENERAL PROCEDURE AND EXPERIMENTAL SETUP

7.1. TURBINE DETAILS

The turbine used in the study was Francis turbine connected to rope type dynamometer for loading. For the operation of turbine, hydraulic pumping system was used which consisted of centrifugal pump, induction motor (15 H.P), etc. The artificial head to run the turbine was created by a hydraulic pump. A single stage pump was supplying water to the Francis turbine through a penstock. The water entered the guide blade (vane) section of the turbine where the static pressure of the water was partially converted into kinetic energy. The water then entered moving blades where further fall of pressure occurred which resulted into the motion of the turbine. When the pressure used to fall below atmospheric value in the draft tube, water flew into tale race. The rope break dynamometer was used for loading and speed was measured by tachometer. Provision was also made available for different loads, speed, vane openings and pressure head. One can vary pressure head from 12 to 18 feet head of water and the load can be varied from 0 Kg to 8 Kg and speed can be varied from 700 to 1000 rpm at different vane openings (0, 25, 50, 75 and full).

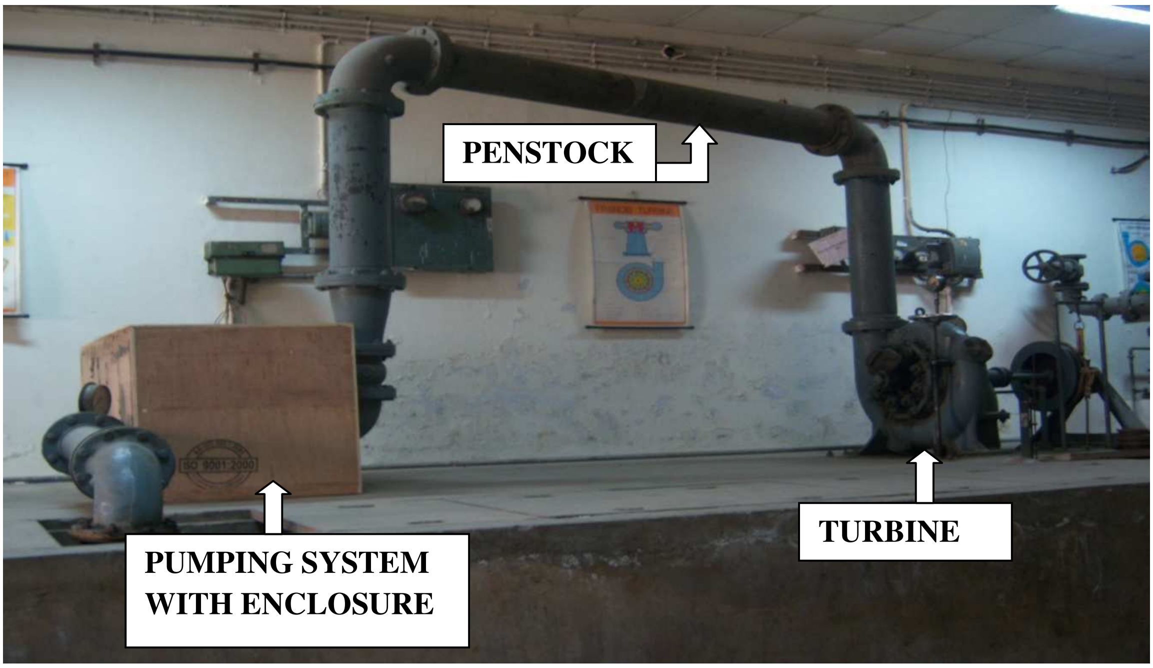


Fig.7.1 Francis turbine set up

5.2. EXPERIMENTAL PROCEDURE AND METHODOLOGY

1. First of all the length, breadth and height of pumping system were measured to fabricate the enclosure.

2. Points A, B, C, D and E were marked at 0.5 m distance from the turbine boundary. To study the noise at different operating conditions it was required to take the SPL at various points as close as possible to the turbine as per the ISO standards. The four points A, B, C and D were taken at a distance of 0.5 m around the turbine and 0.5 m above the ground plane and fifth point E was taken at the 0.5 m above the C.G of the system.

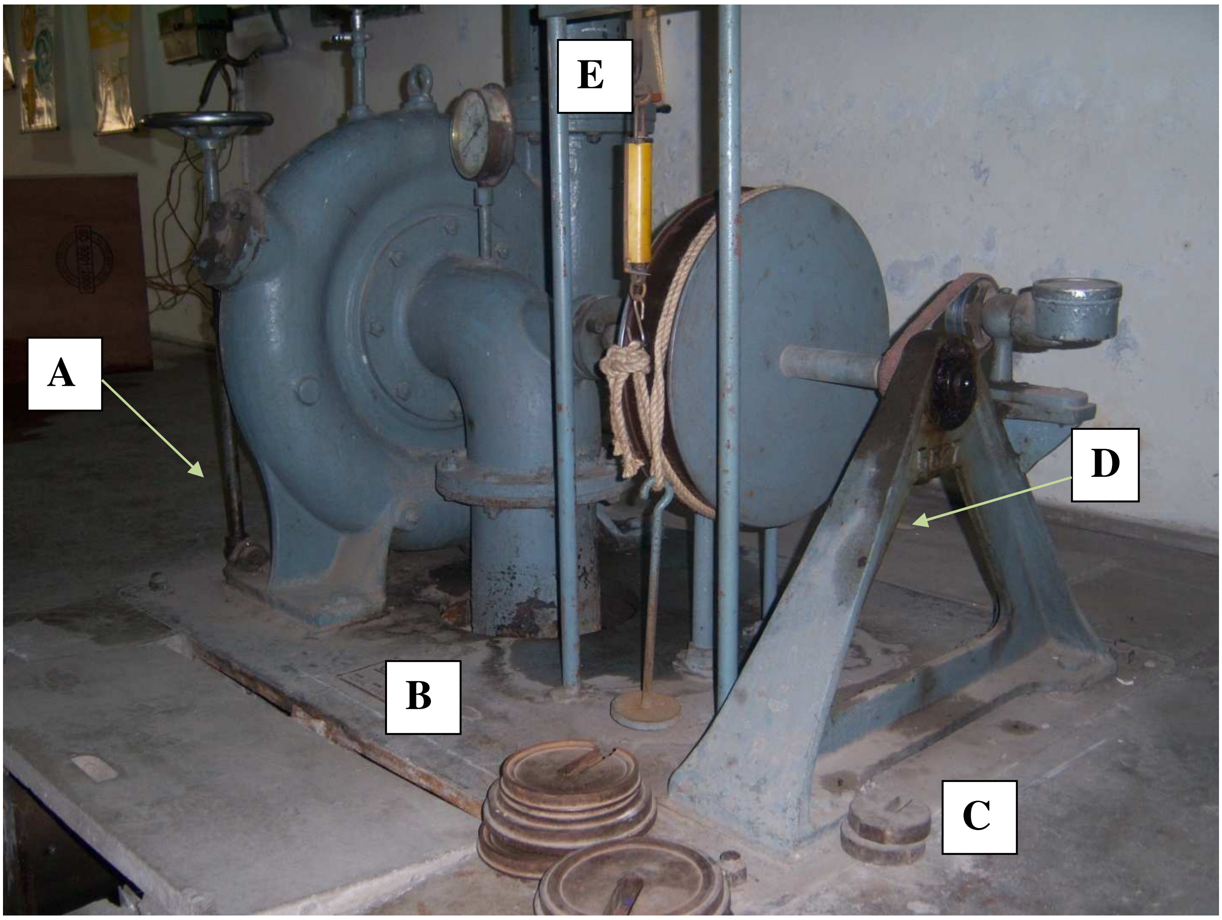


Fig.7.2 Five points at 0.5 m away from turbine boundary

7.2.1. PROCEDURE OF SOUND PRESSURE LEVEL MEASUREMENT: It was measured by using SLM (Sound level meter), SLM used in the experiment is manufactured by CESVA (a Spanish company) model no. SC310:

Fig.7.3 sound level meter with windscreen

7.2.1.1 STEPS FOR MEASUREMENT OF SOUND PRESSURE LEVEL:

- First of all the SLM (Sound level meter) was calibrated and the calculating error was removed. It was calibrated to 94 dB (A) on the lower side and 104 dB (A) on the upper side.
- Passive noise was measured by SLM with and without enclosure to check the reduction in SPL.
- Sound Pressure Level (SPL) was measured at different points of turbine with the help of SLM by keeping pressure head constant without the enclosure.
- Sound Pressure Level (SPL) was measured at different points of turbine with the help of SLM by keeping speed constant without the enclosure.
- Same procedure was repeated with the enclosure.

7.2.1.2 PRECAUTIONS TO BE KEPT IN MIND BEFORE RUNNING THE TURBINE SET UP.

- Always close the discharge valve of centrifugal pump at the start of the experimentation
- Prime the pump before starting the motor

- The tap supplying the cooling water to break drum should be opened before starting the turbine
- Open the discharge valve of the pump gently.
- Add and remove weights gently.

7.3 MEASUREMENT OF SPL IN FREQUENCY SPECTRUM

The value of SPL at 1-1 octave band frequency gives one an idea of the maximum and minimum value at available frequencies. In this case the frequency spectrum was formed at that point where SPL was maximum among A, B, C, D and E. The readings were taken by changing the parameters of pressure head, speed, vane openings and loads. This instrument gave the data of all frequencies in software “CESVA CAPTURE STUDIO”.

CHAPTER 8

RESULTS AND DISCUSSIONS

After all the observations were taken, it was required to analyze the sound pressure level by varying pressure heads, speeds and vane openings (0%,25%,50%,75% and 100%) at different loads i.e.,0, 2, 4, 6, 8(Kg). Data for Sound Pressure Level was taken for five locations i.e. A, B, C, D and E. A, B, C and D are 0.5 m away from the turbine boundary and point E is 0.5 m above the C.G of the turbine set up. The graphs were plotted between sound pressure level and Loads keeping pressure head and speed constant. All the data used in these graphs has been given in Appendix A.

8.1 GRAPHS FOR POINT-"A" W.R.T CONSTANT PRESSURE HEADS

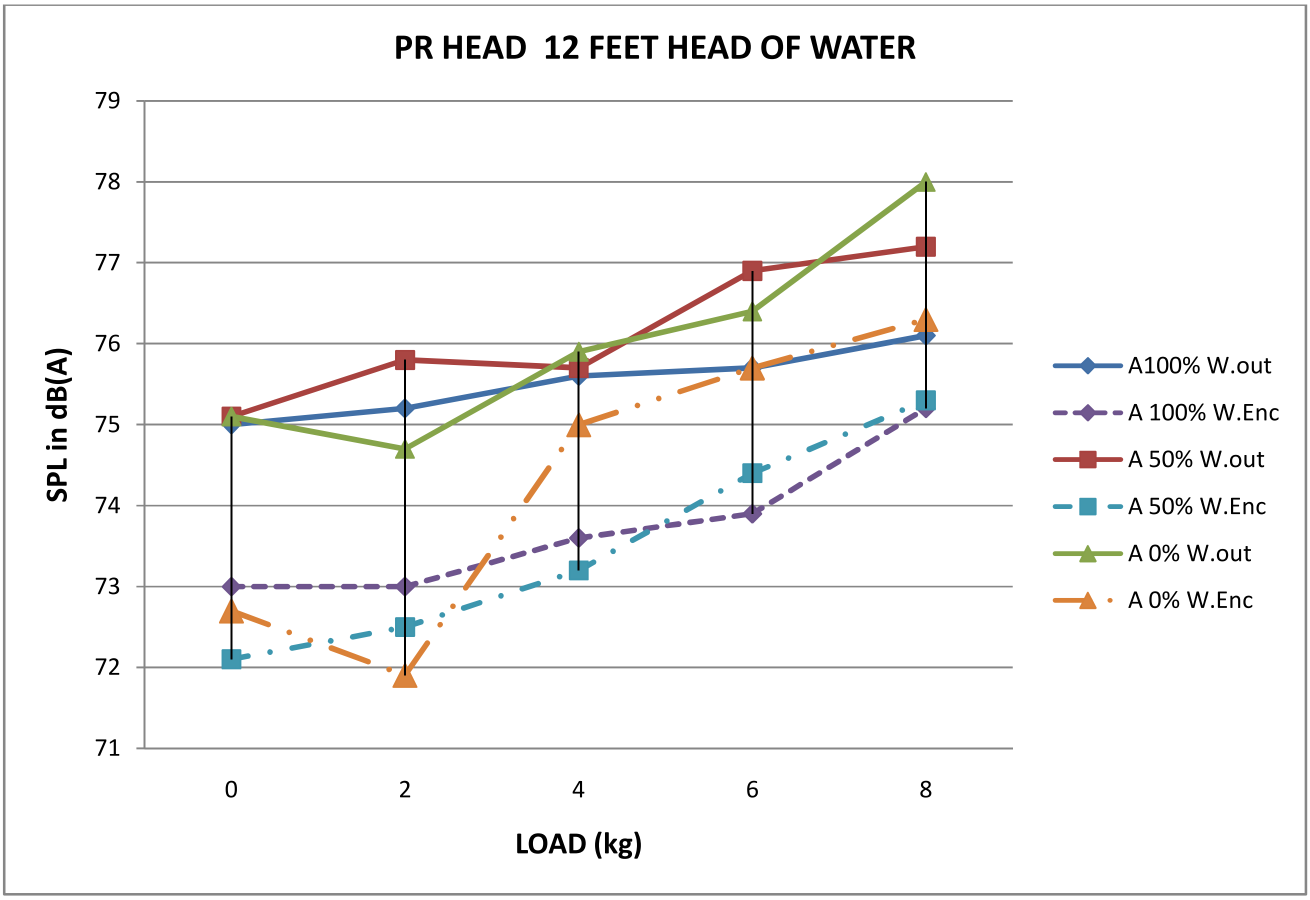


Figure No 8.1 SPL Vs Load (kg) at point A

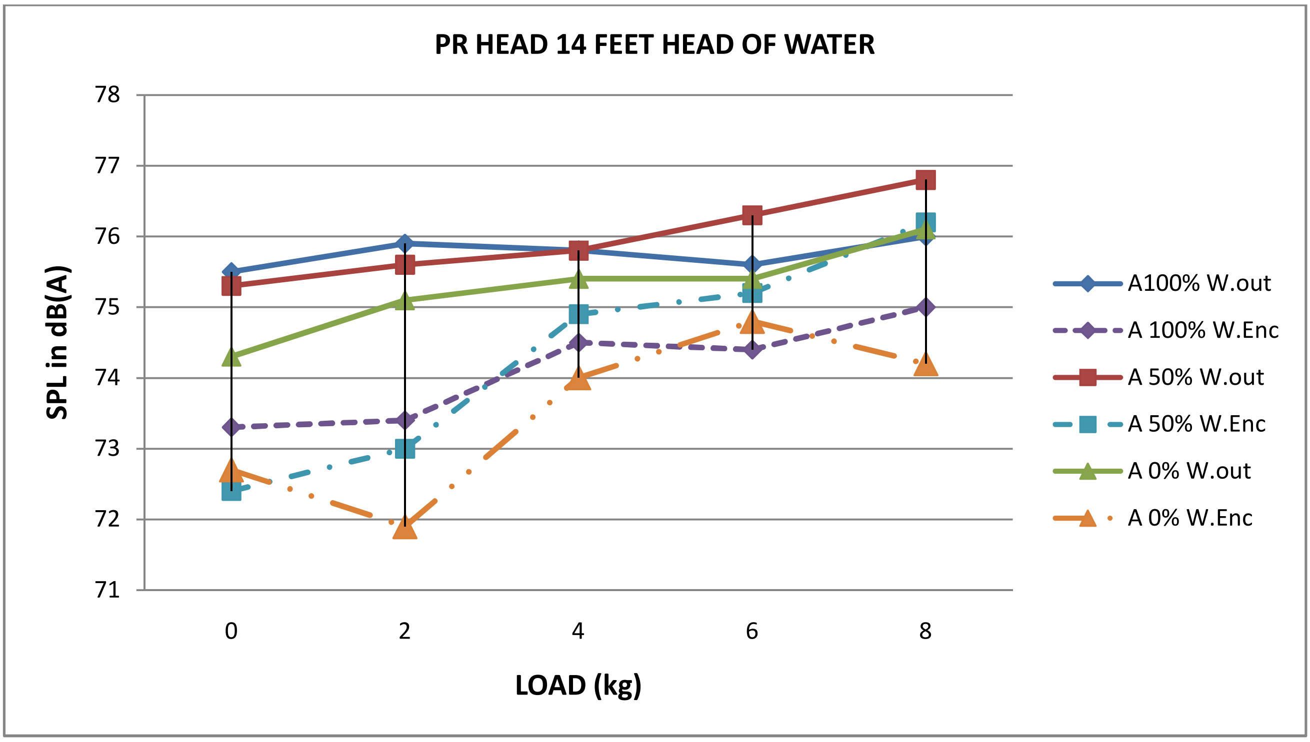


Figure No 8.2 SPL Vs Load (kg) at point A

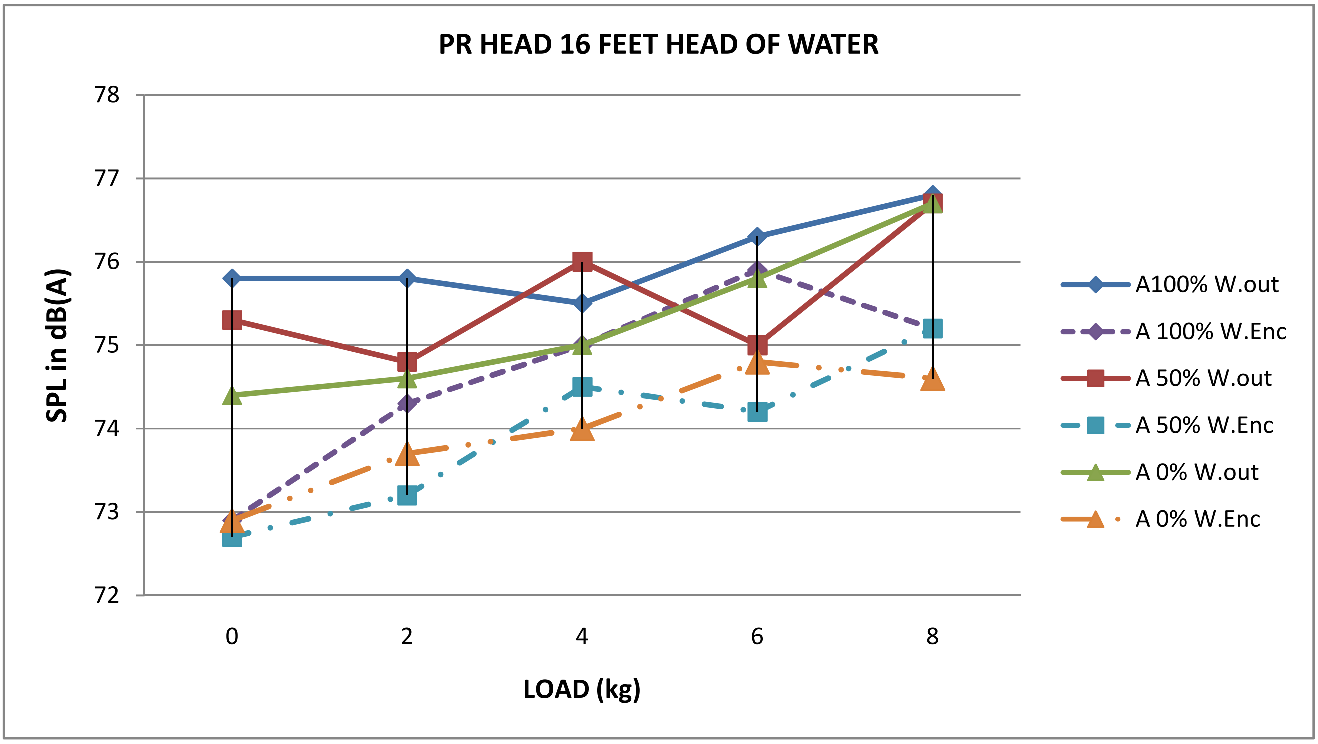


Figure No 8.3 SPL Vs Load (kg) at point A

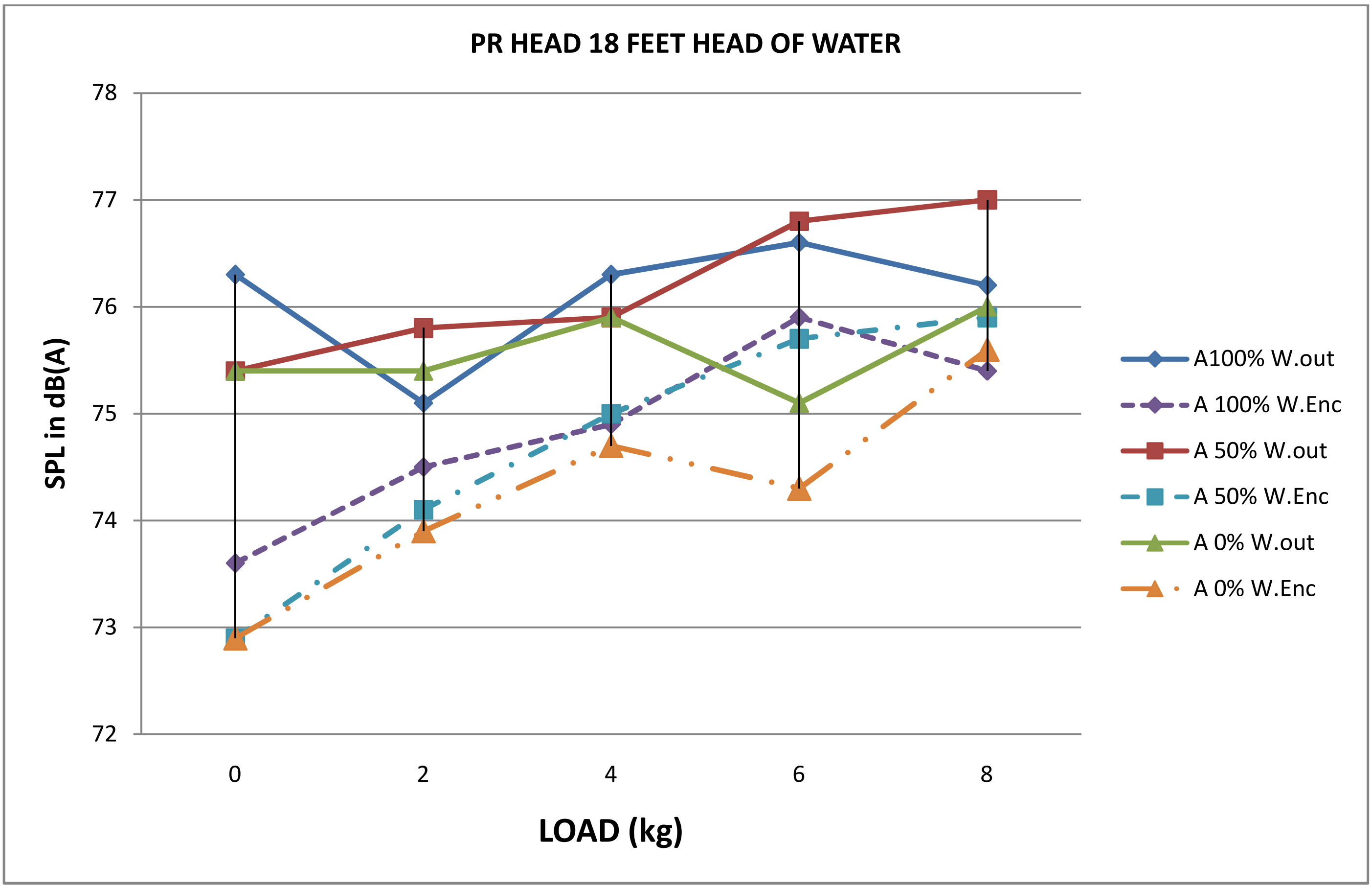


Figure No 8.4 SPL Vs Load (kg) at point A

8.2 ANALYSIS FOR POINT-"A" W.R.T CONSTANT PRESSURE HEADS

The figures 8.1 to 8.4 show the variation of sound pressure level at point "A" with and without enclosure to check the effectiveness of enclosure. The bold lines show the SPL in dB (A) without the enclosure and dotted lines show the SPL with enclosure. The enclosure is used to reduce the background noise. The SPL is measured by keeping the pressure heads constant, using different vane openings (100%, 50% and 0%) and by varying the loads.

- As observed from Fig.8.1 the behavior of reduction curve in SPL for 100% and 50% cane opening is almost similar but for 0% vane opening one can observe high reduction at 0 and 2 Kg but suddenly the behavior of reduction curve changes in comparison to expected behavior for higher loads. For pressure head 12 the maximum reduction in SPL observed with enclosure is for load 2 kg at 0% vane opening and minimum reduction in SPL is at 0% vane opening at load 6.
- As shown in Fig.8.2 for constant pressure head 14 the maximum reduction in SPL observed with enclosure is for load 2 kg at 0% vane opening and minimum reduction in SPL is at 0% vane opening at load 6.

- As shown in Fig.8.3 for constant pressure head 16 the maximum reduction in SPL observed with enclosure is for load 0 kg at 100% vane opening and minimum reduction in SPL is at 100% vane opening at load 6 and almost equivalent reduction in SPL is observed at 50 % vane opening at load 6.
- As shown in Fig.8.4 for constant pressure head 18 the maximum reduction in SPL observed with enclosure is for load 0 kg at 100% vane opening and minimum reduction in SPL is at 0% vane opening at load 8.

8.3 GRAPHS FOR POINT-"B" W.R.T CONSTANT PRESSURE HEAD

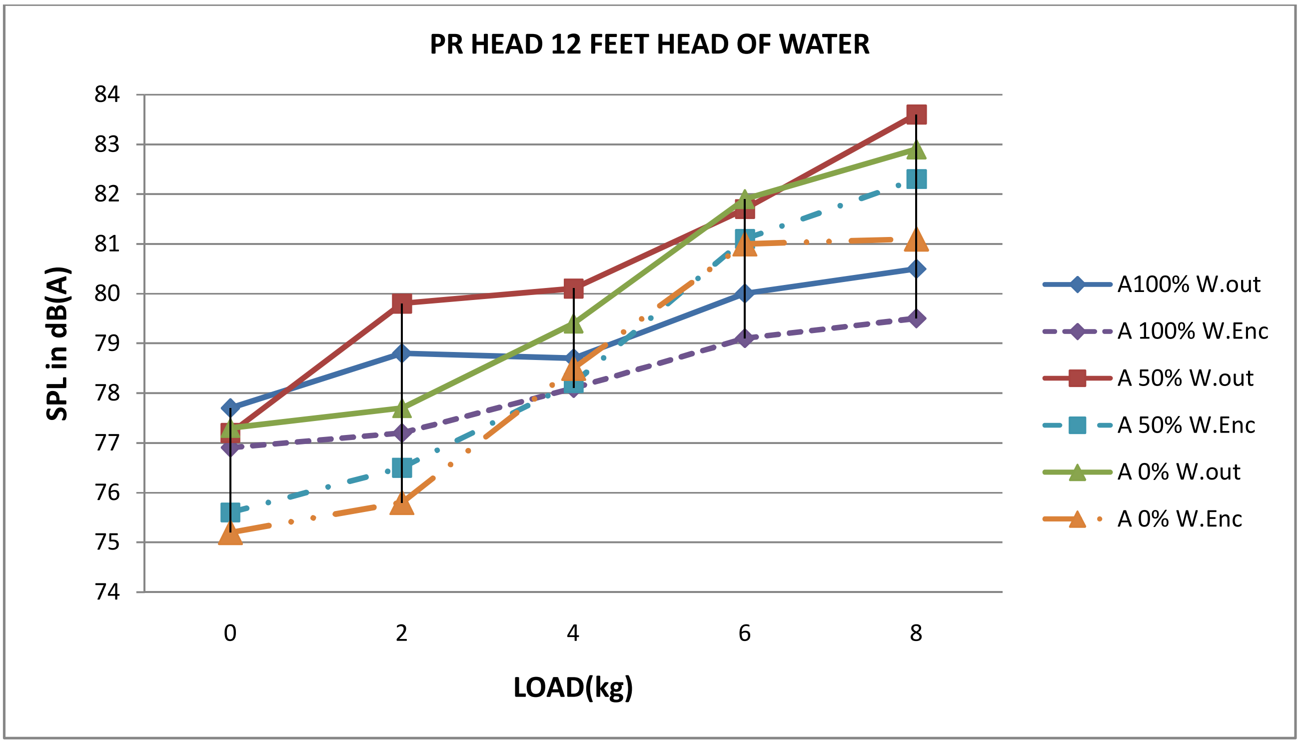


Figure No 8.5 SPL Vs Load (kg) at point B

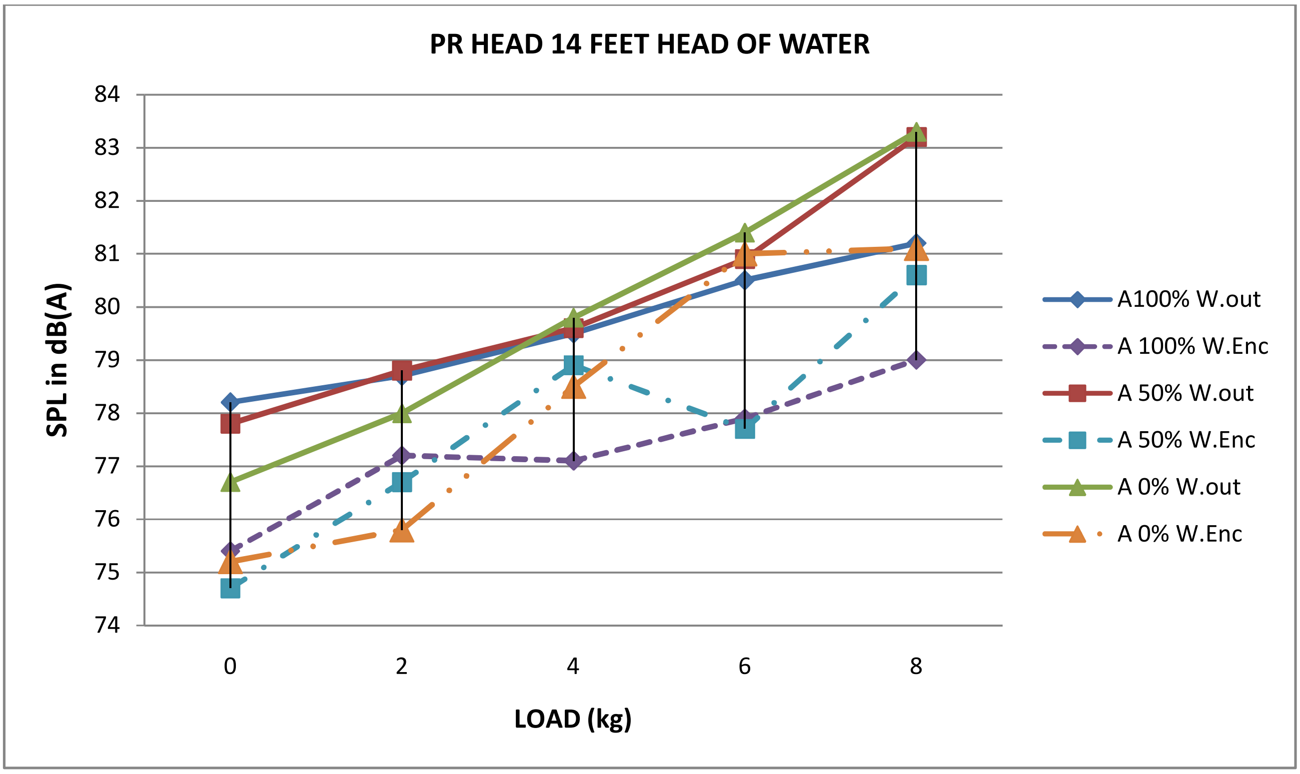


Figure No 8.6 SPL Vs Load (kg) at point B

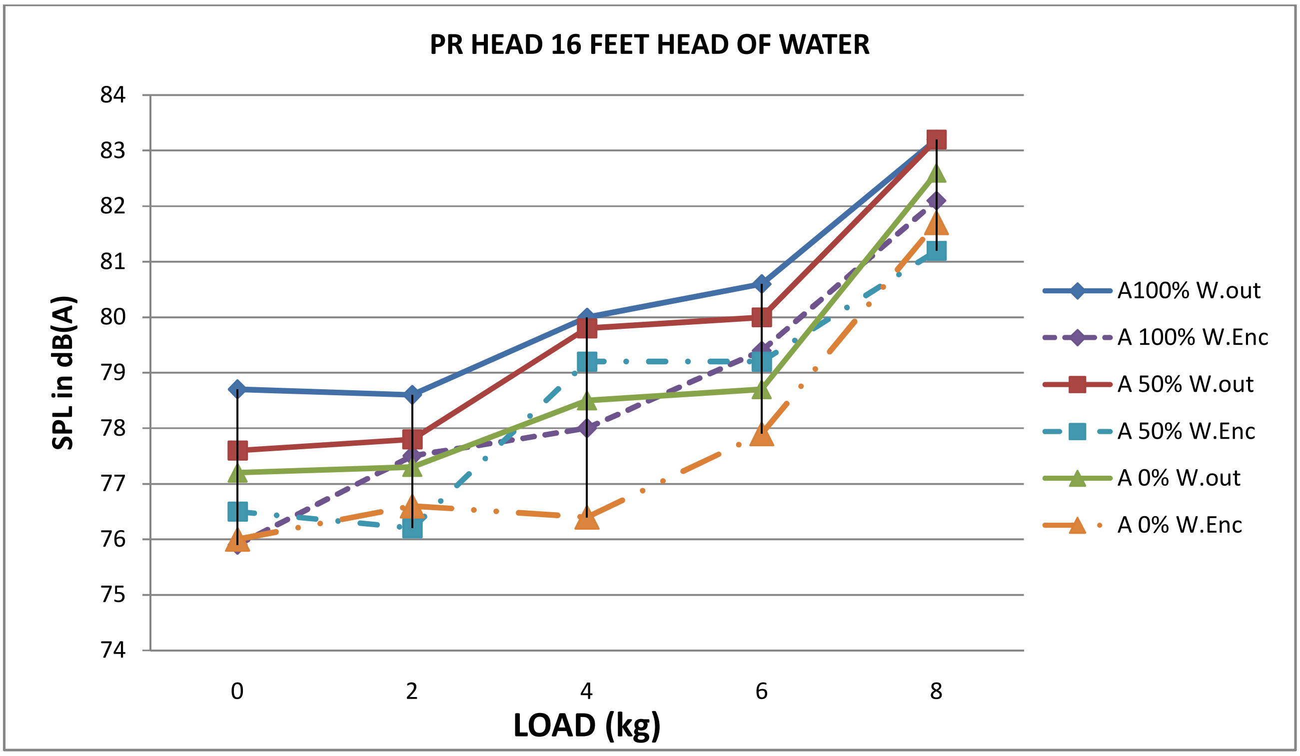


Figure No 8.7 SPL Vs Load (kg) at point B

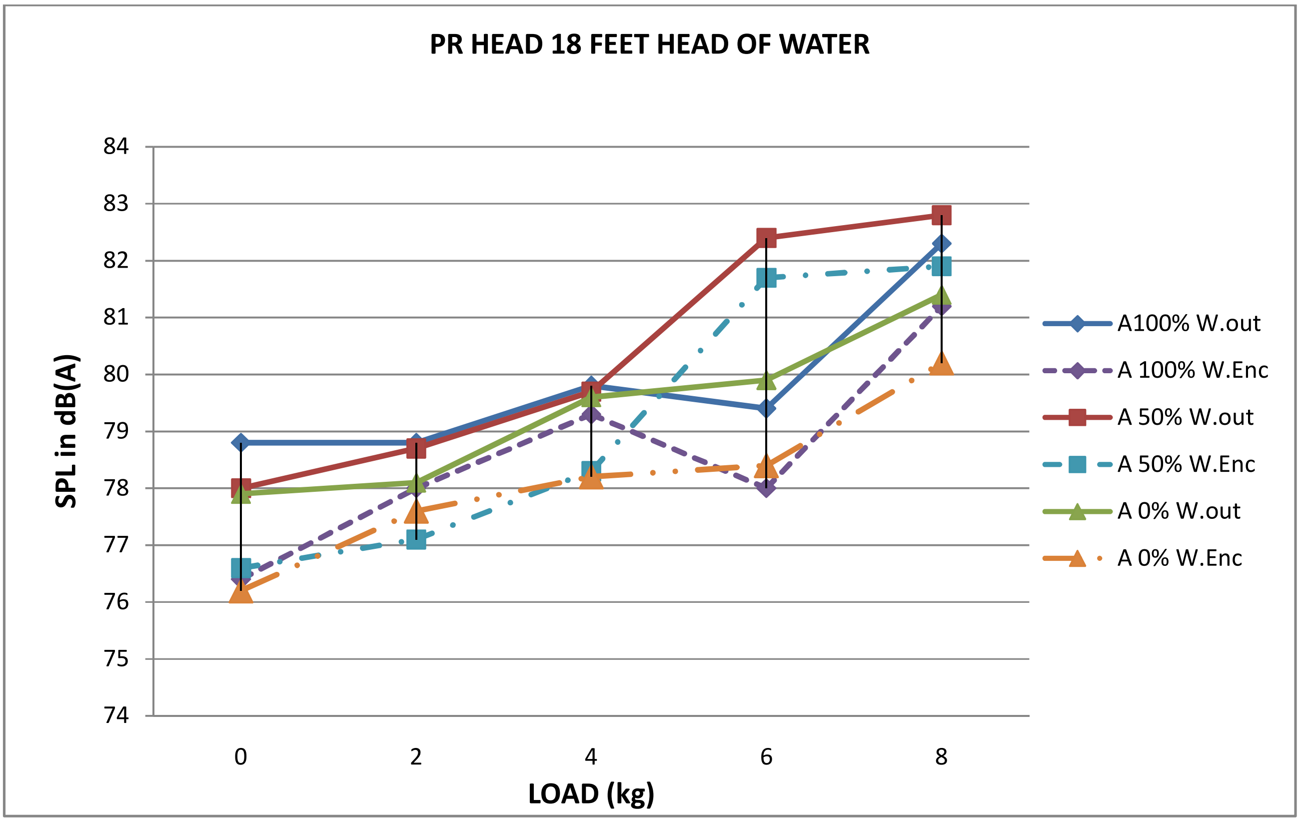


Figure No 8.8 SPL Vs Load (Kg) at point B

8.4 ANALYSIS FOR POINT-"B" W.R.T CONSTANT PRESSURE HEAD

- In fig 8.5 For pressure head 12 the maximum reduction in SPL observed with enclosure is for load 2 kg at 50% vane opening and minimum reduction in SPL is at 0% vane opening at load 6.
- As shown in Fig.8.6 for constant pressure head 14 the maximum reduction in SPL observed with enclosure is for load 0 and at 6 kg at 50% vane opening and minimum reduction in SPL is at 0% vane opening at load 6 kg.
- As shown in Fig.8.7 for constant pressure head 16 has same reduction as in fig 8.1 for 0% vane opening.
- In Fig.8.8 for constant pressure head 18 the SPL observed for 50% vane opening and at load 6 kg is more than other vane openings and loads.

8.5 GRAPHS FOR POINT-"C" W.R.T CONSTANT PRESSURE HEADS

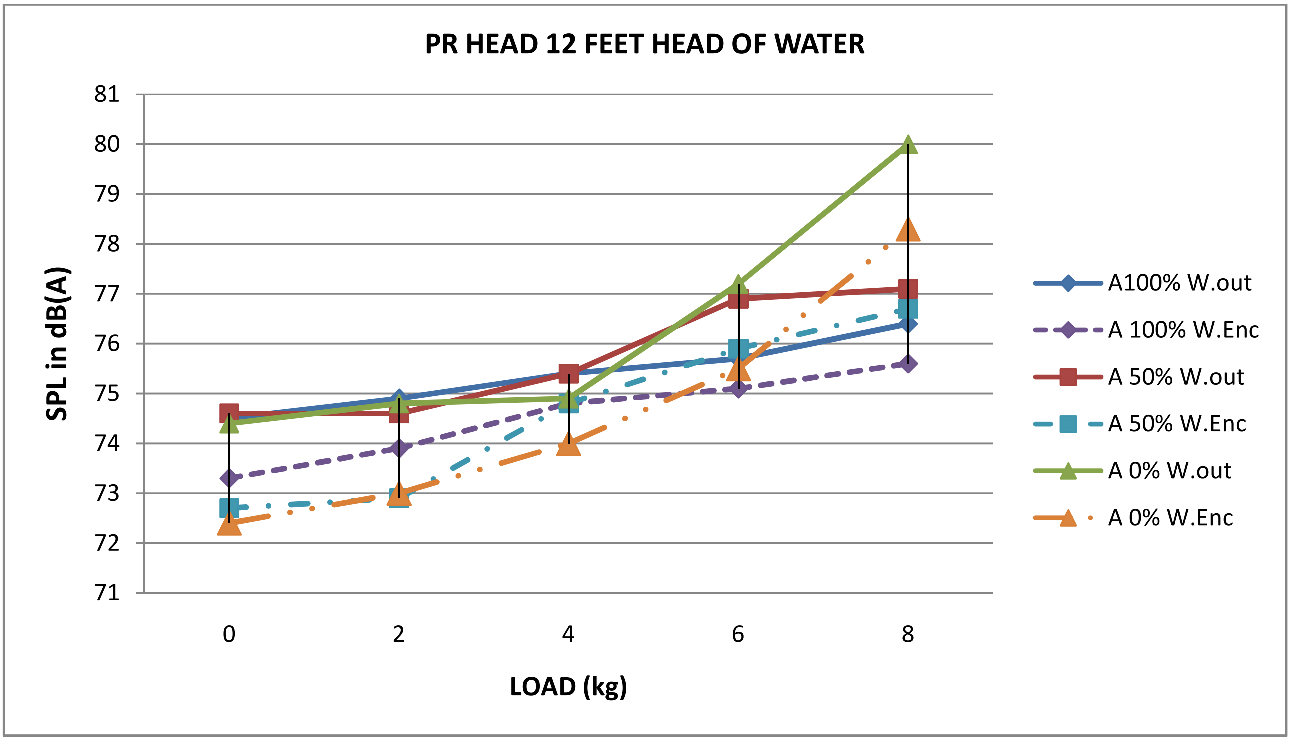


Figure No 8.9 SPL Vs Load (kg) at point C

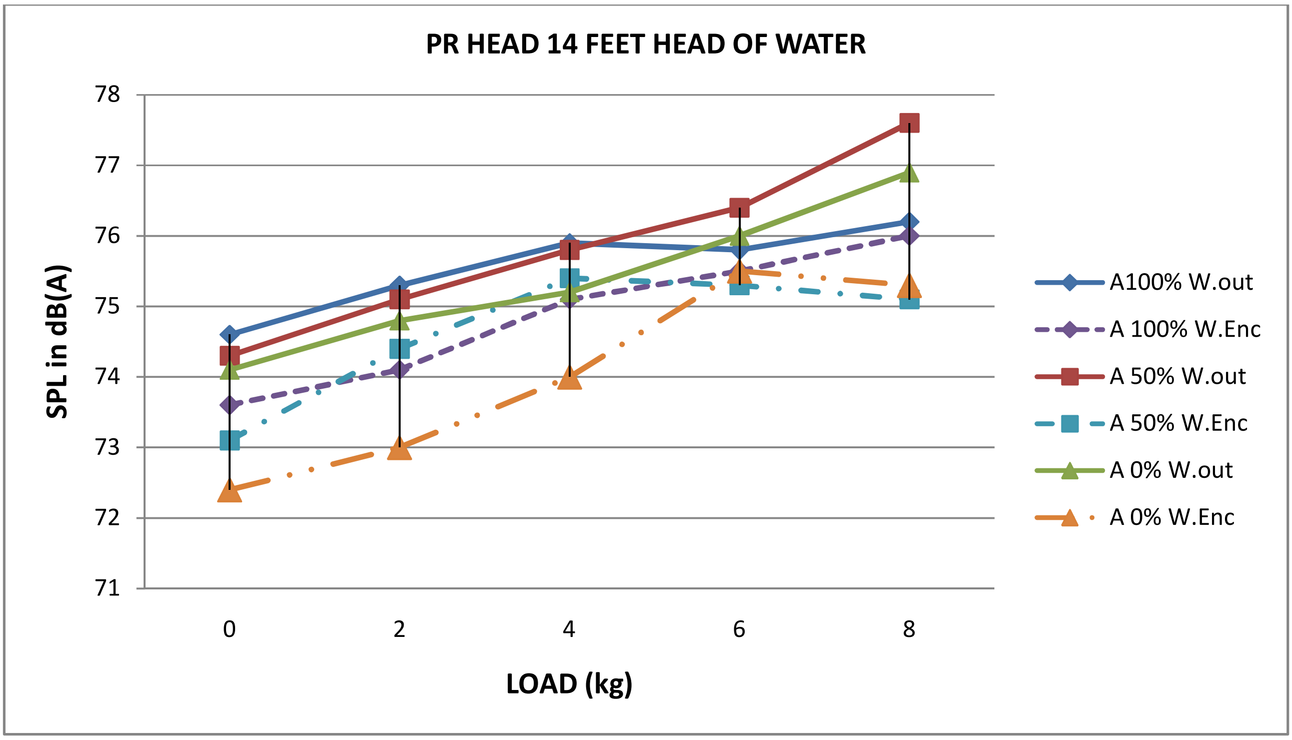


Figure No 8.10 SPL Vs Load (kg) at point C

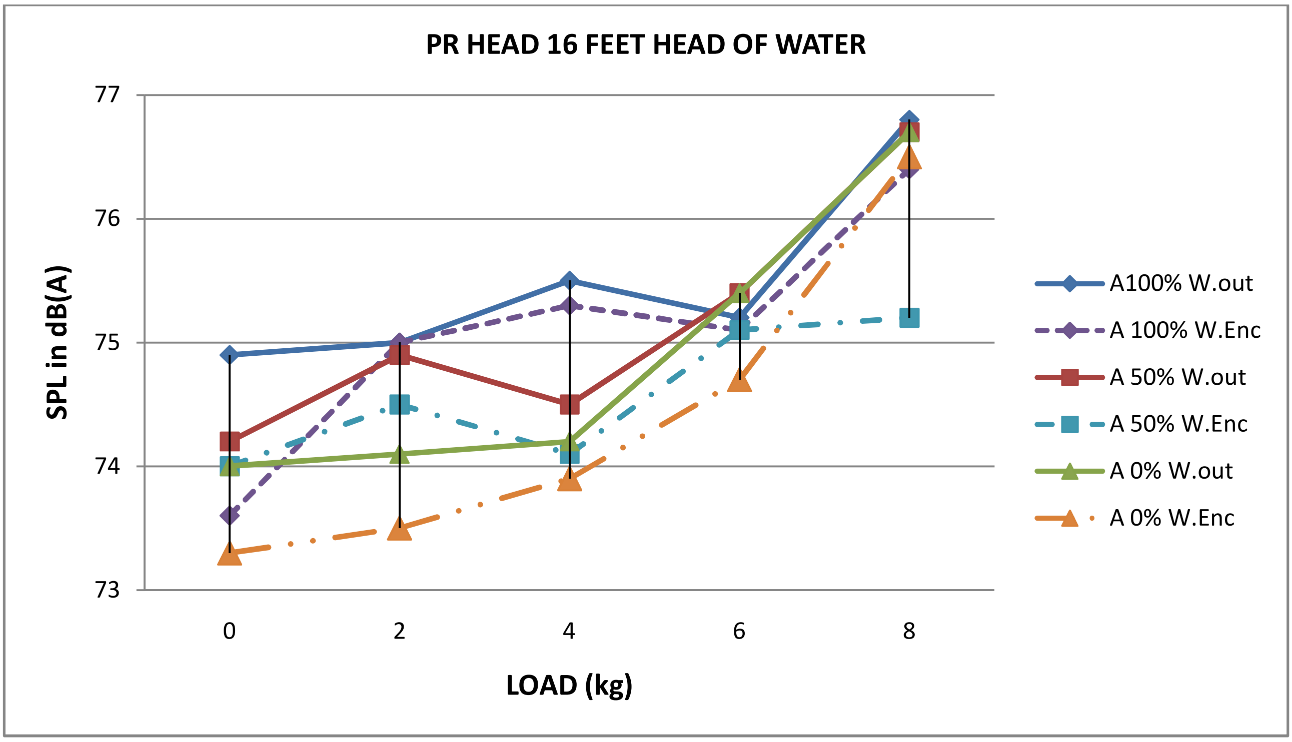


Figure No 8.11 SPL Vs Load (kg) at point C

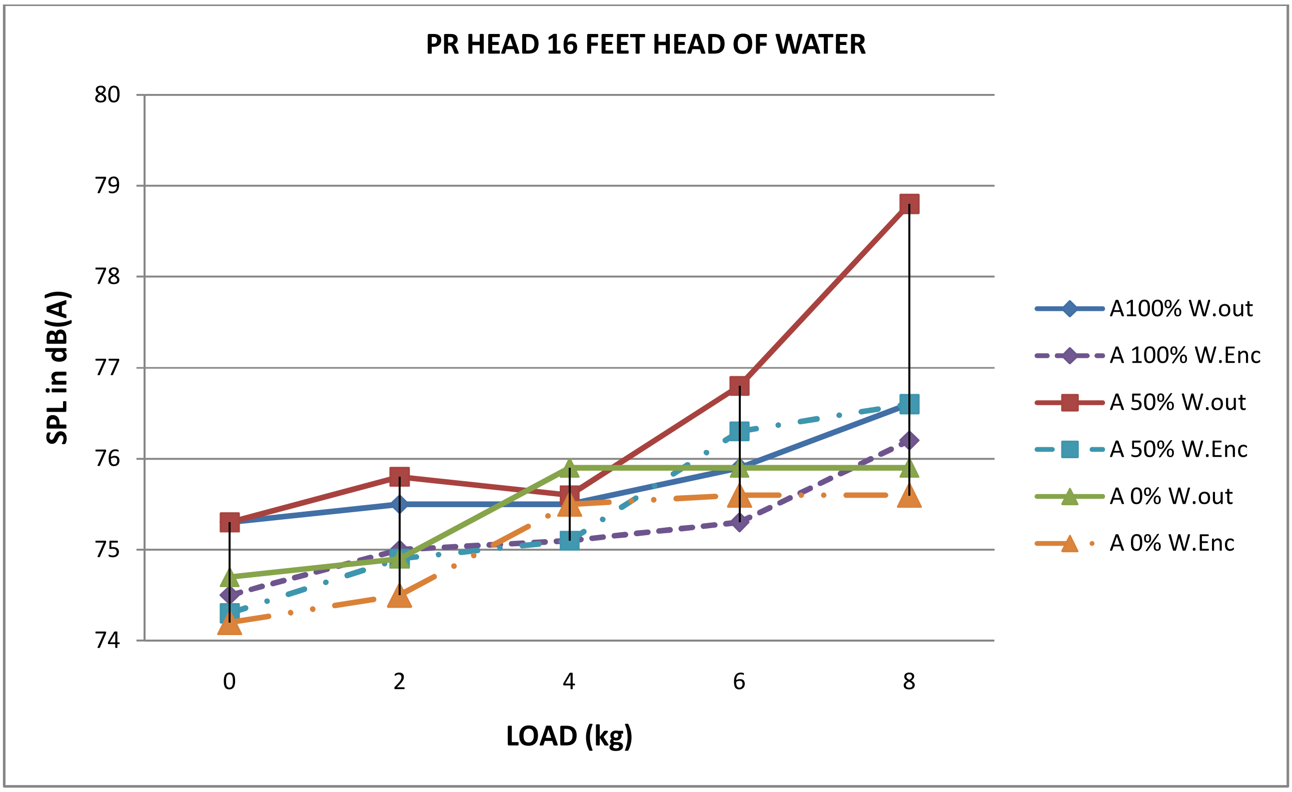


Figure No 8.12 SPL Vs Load (kg) at point C

8.6 ANALYSIS FOR POINT-“C” W.R.T CONSTANT PRESSURE HEADS

- In fig 8.9 For pressure head 12 one can observe linear rise in SPL at 0%,50% and 100% vane openings but then one can observe sudden rise in SPL after load 4 kg for 0% vane openings.
- As per the observations Fig.8.10 maximum reduction in SPL observed with 0% vane opening is higher than other vane openings.
- As shown in Fig.8.11 for constant pressure head 16 the curves show the zig-zag behavior and one can observe that small reduction in 100% vane opening for loads 2, 4, 6, 8 kg loads.
- In the Fig.8.12 the maximum reduction in SPL observed with enclosure is for load 8 kg at 50% vane opening and minimum reduction in SPL is at 0% vane opening at load 6.

8.7 GRAPHS FOR POINT-“D” W.R.T CONSTANT PRESSURE HEADS

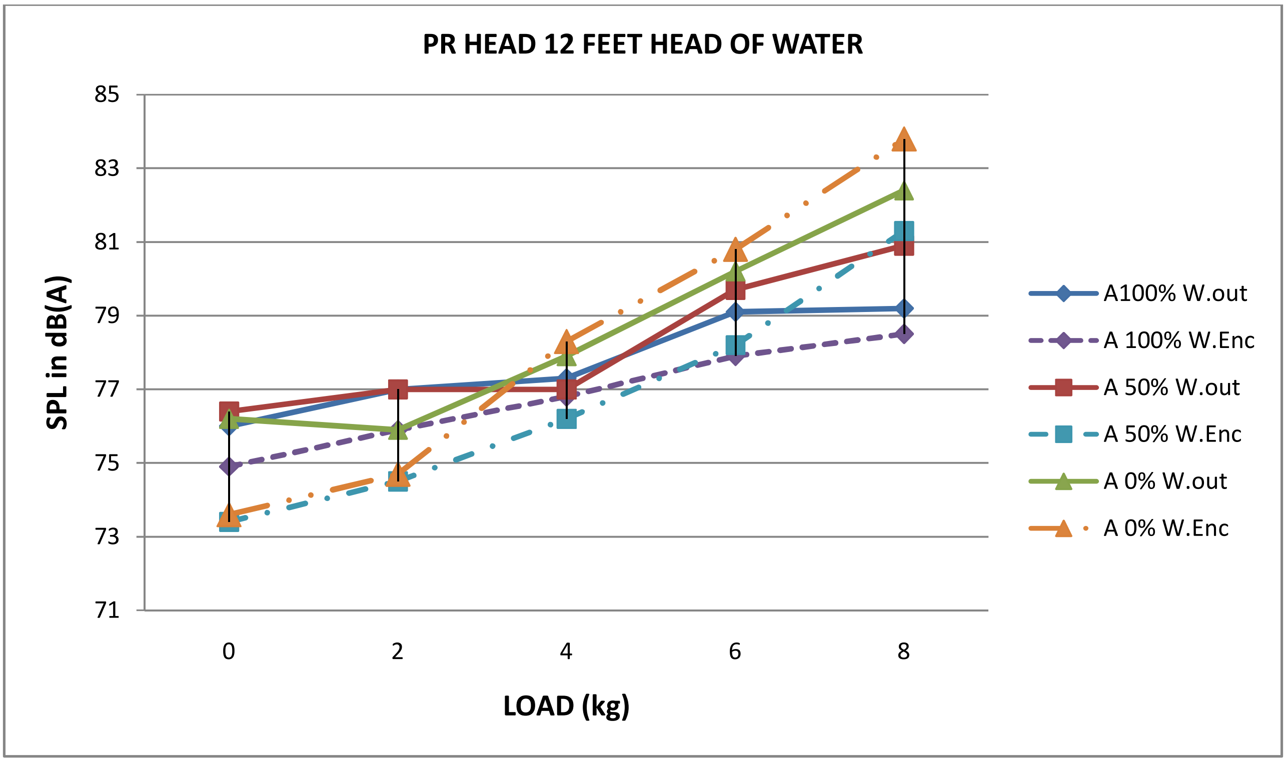


Figure No 8.13 SPL Vs Load (kg) at point D

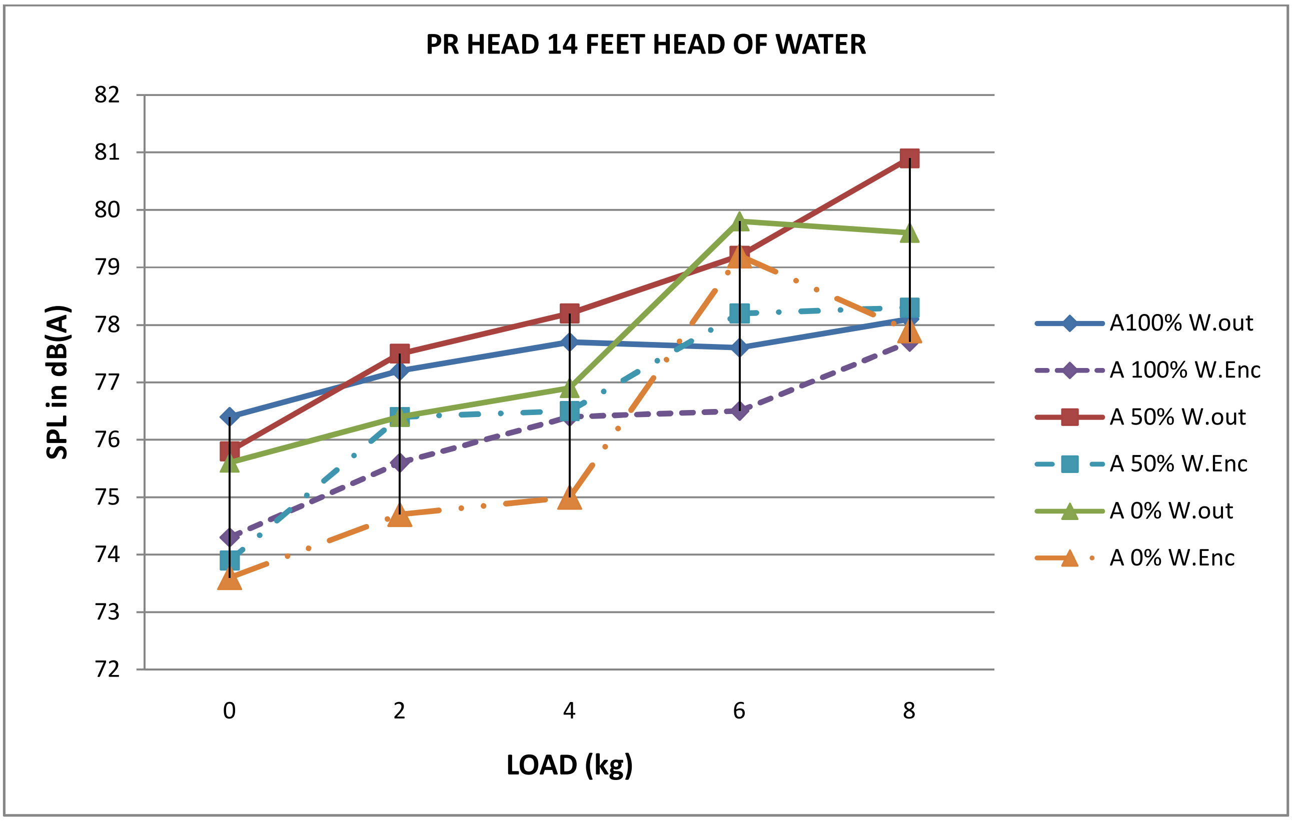


Figure No 8.14 SPL Vs Load (kg) at point D

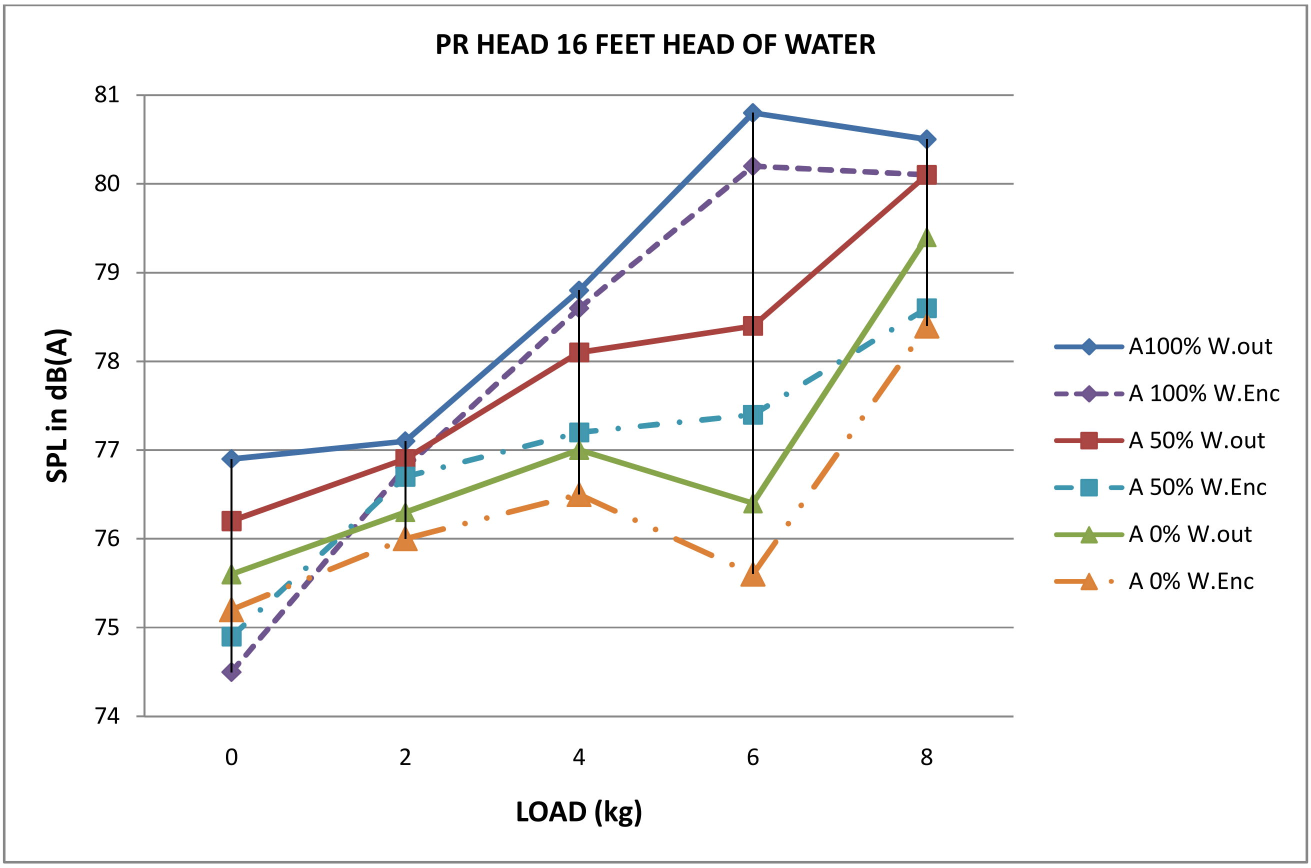


Figure No 8.15 SPL Vs Load (kg) at point D

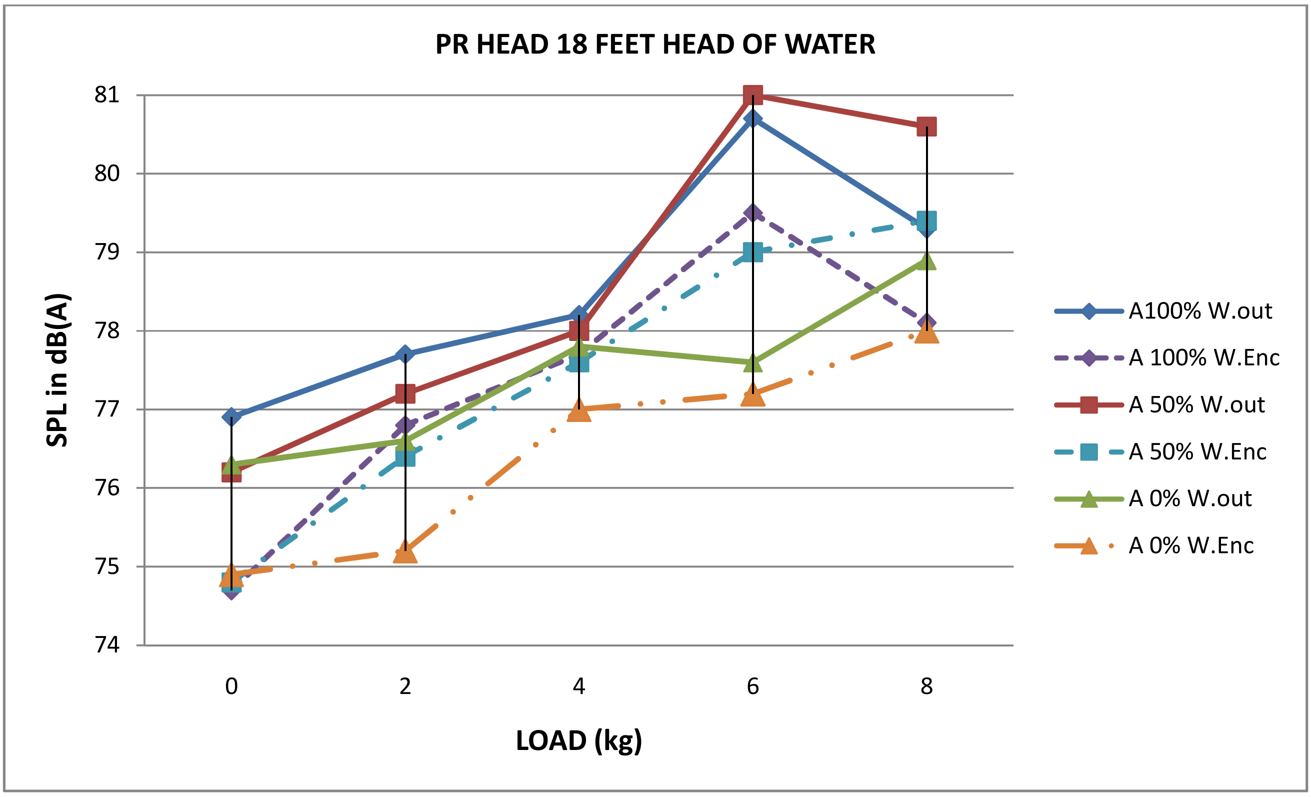


Figure No 8.16 SPL Vs Load (kg) at point D

8.8 ANALYSIS FOR POINT-“D” W.R.T CONSTANT PRESSURE HEADS

- In fig 8.13 for pressure head 12 one can observe very different behavior of SPL 0%vane opening and after 4 kg load in comparison to other vane openings.
- In the Fig.8.14 the maximum reduction in SPL observed with enclosure is for load 8 kg at 0% vane opening and minimum reduction in SPL is at 0% vane opening at load 6.
- As shown in Fig.8.15 it is observed at there is large difference in values of vane openings at 6 kg load.
- In the Fig.8.16 shows similar type of behavior as fig 8.15.

8.9 GRAPHS FOR POINT-"E" W.R.T CONSTANT PRESSURE HEADS

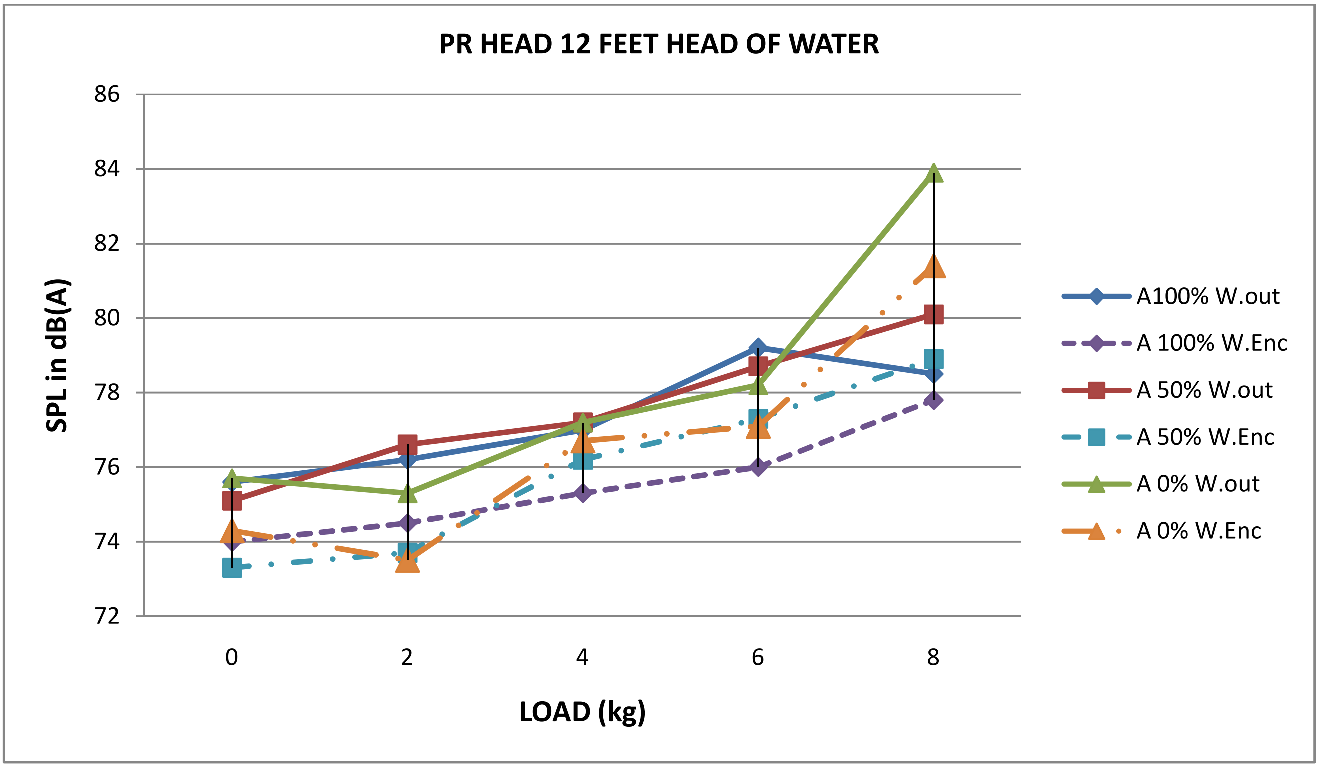


Figure No 8.17 SPL Vs Load (kg) at point E

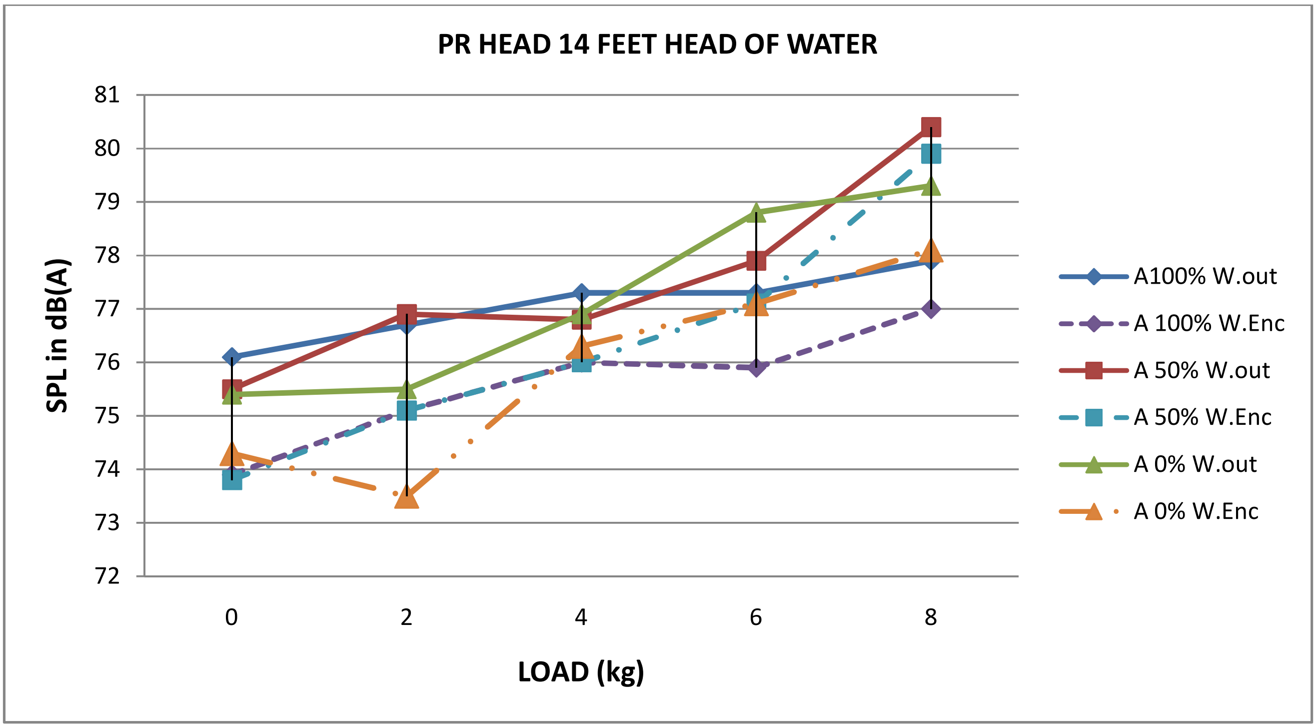


Figure No 8.18 SPL Vs Load (kg) at point E

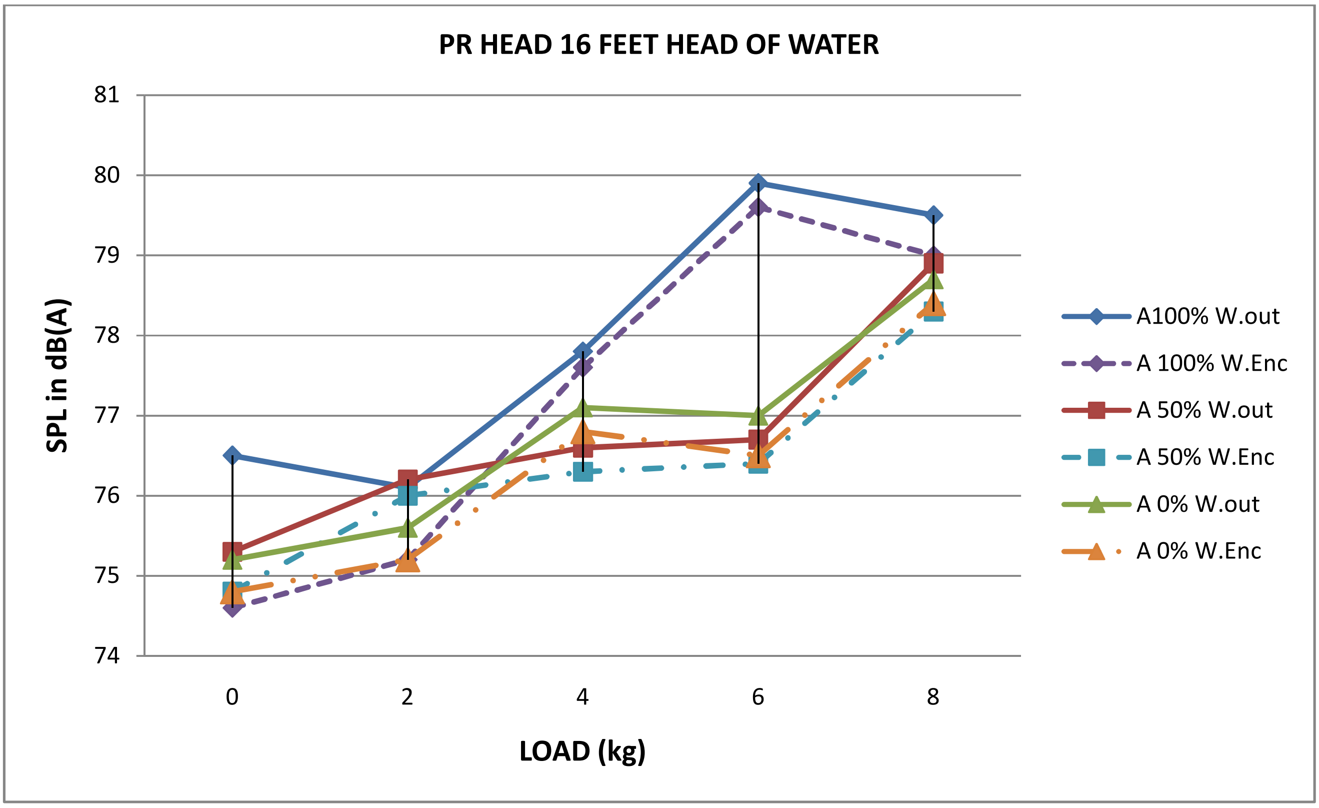


Figure No 8.19 SPL Vs Load (kg) at point E

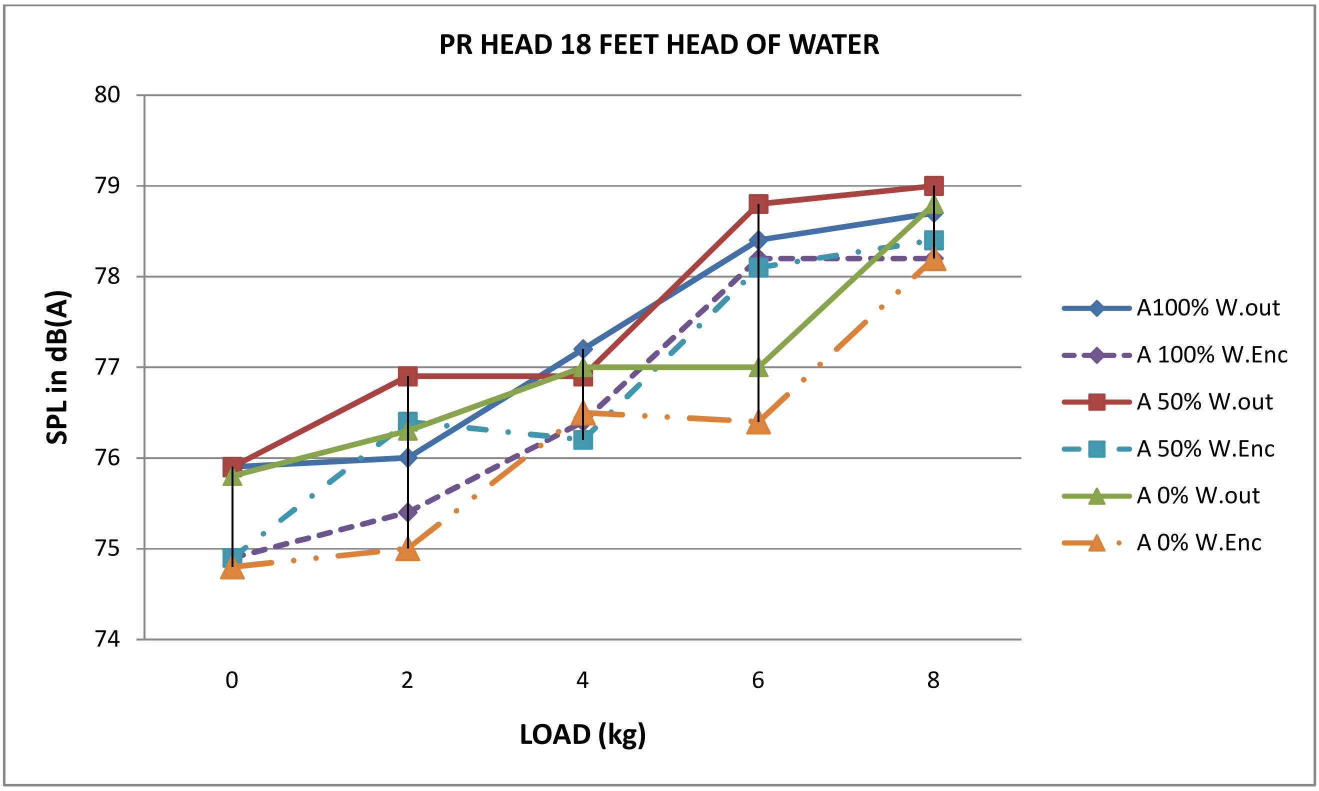


Figure No 8.20 SPL Vs Load (kg) at point E

8.10 ANALYSIS FOR POINT-"E" W.R.T CONSTANT PRESSURE HEADS

- In fig 8.17 maximum SPL observed is at 0% vane opening at load 8 minimum SPL observed is at 0% vane opening at load 2.
- As per the observations Fig.8.18 curves show zig zag pattern with maximum reduction at 0% vane opening with load 2 kg.
- As shown in Fig.8.19 curves show the similar behavior of fig 8.15 of point D
- In the Fig.8.20 the curves for 0% and 50% follow the same pattern till load 4 and again till load 8.

8.11 GRAPHS FOR POINT-"A" W.R.T CONSTANT SPEED

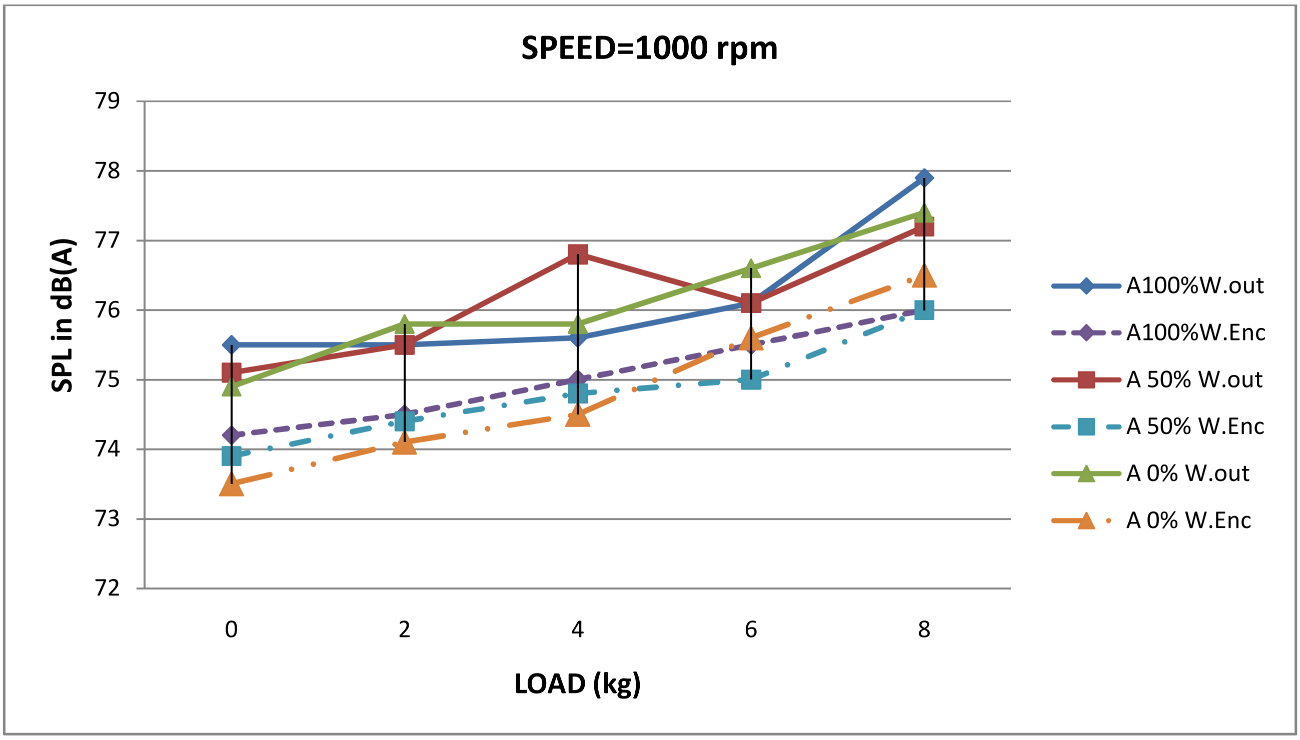


FIGURE NO 8.21 SPL VS LOAD (kg) AT POINT A

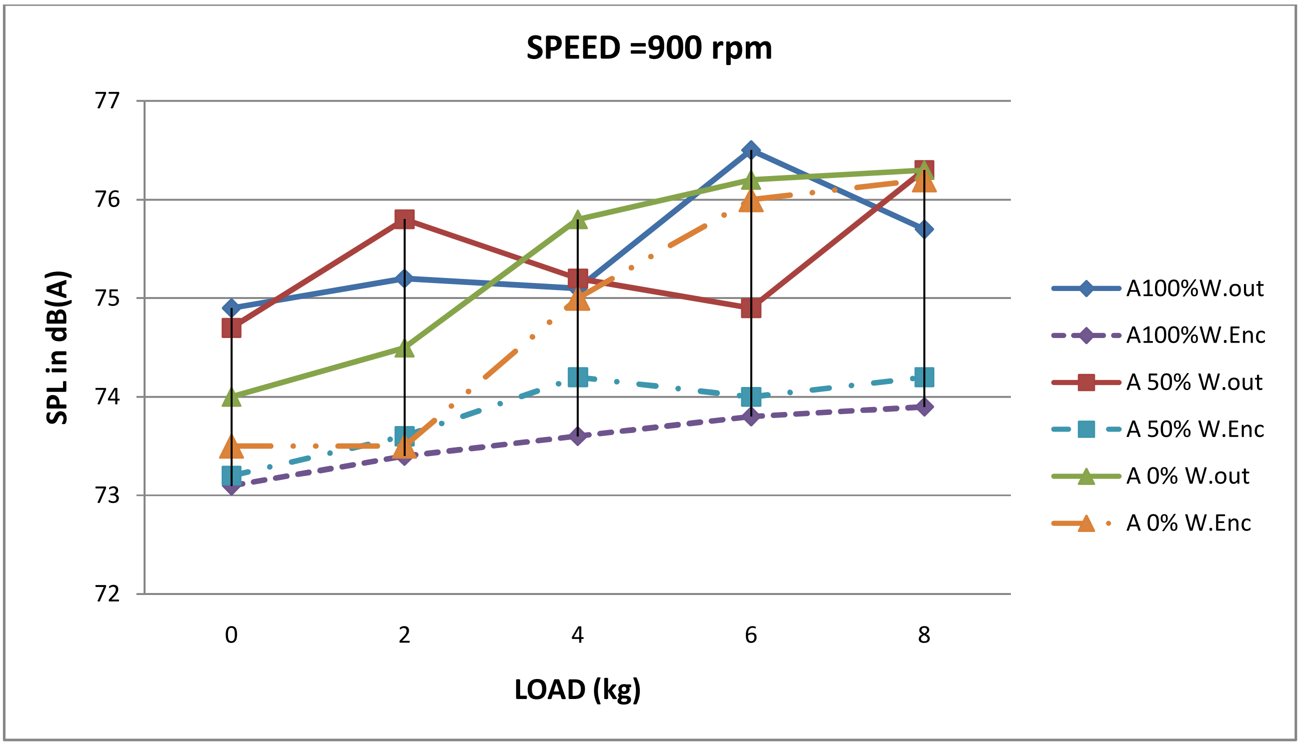


FIGURE NO 8.22 SPL VS LOAD (kg) AT POINT A

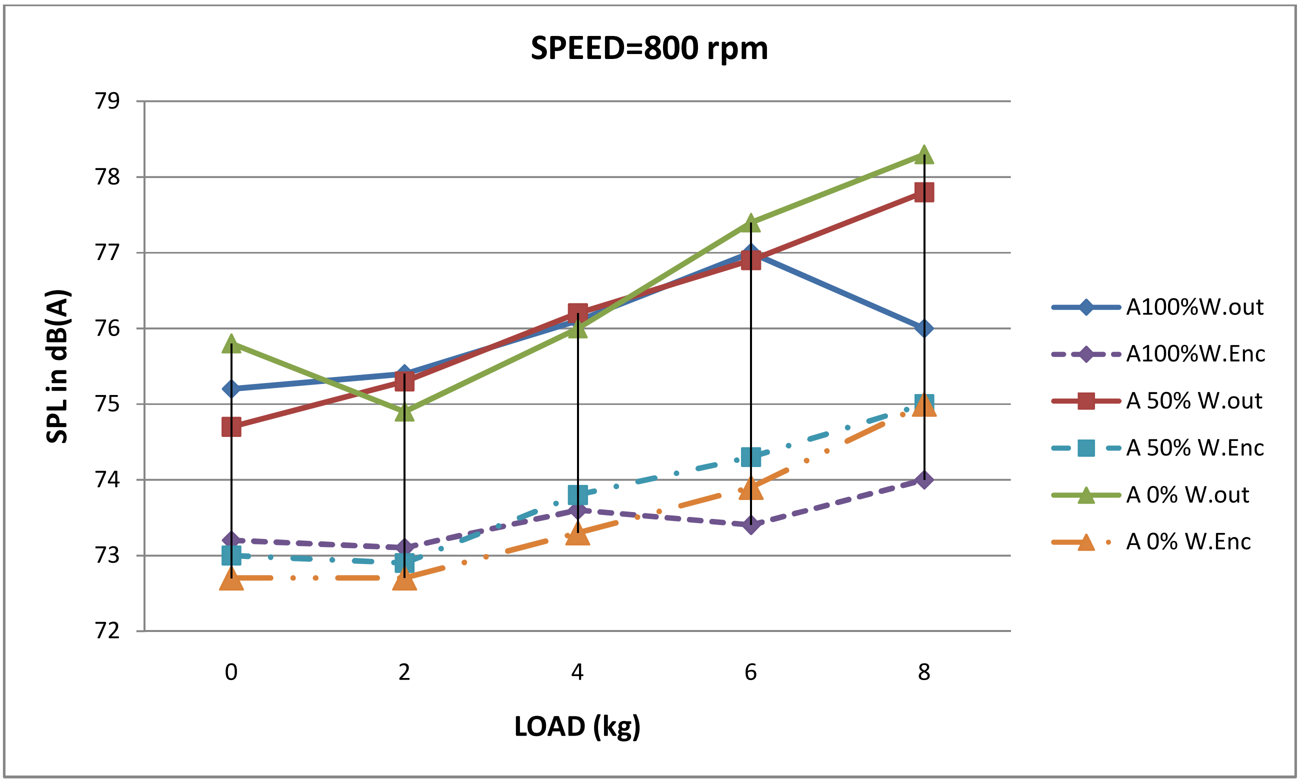


FIGURE NO 8.23 SPL VS LOAD (kg) AT POINT A

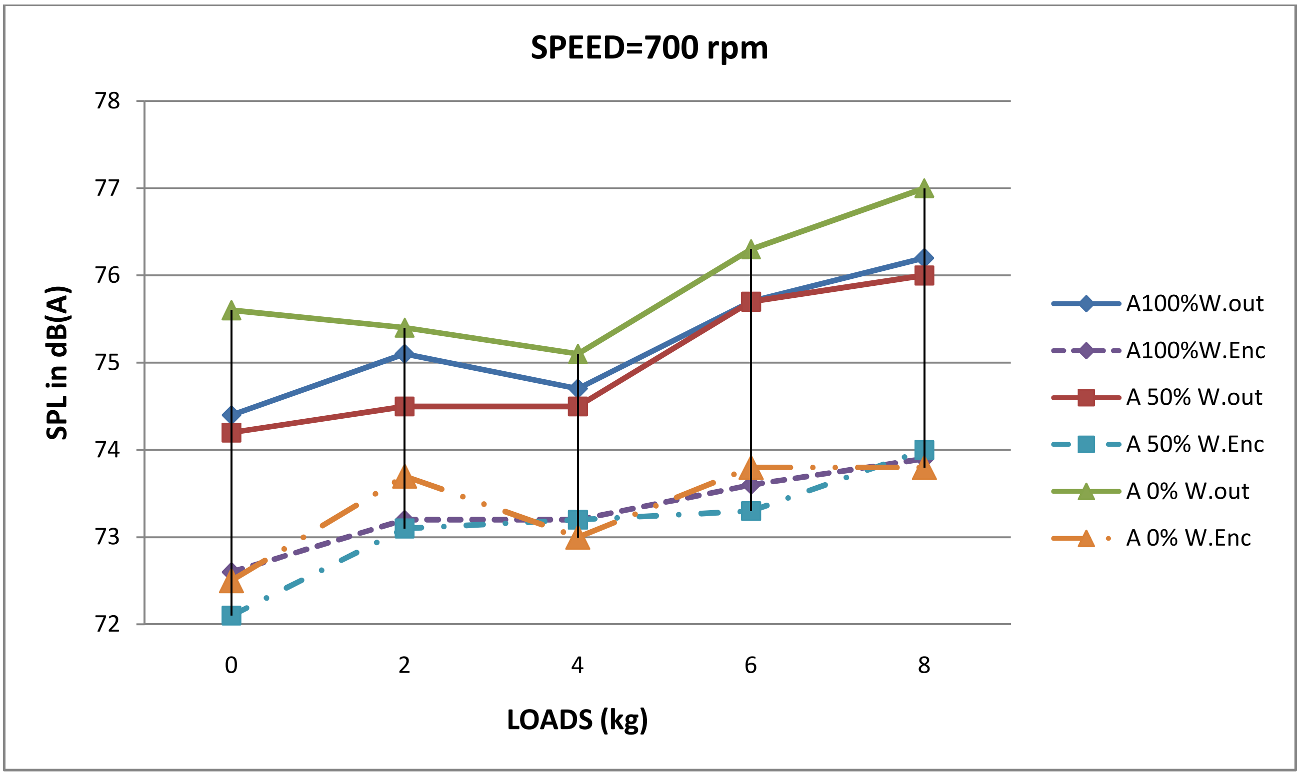


FIGURE NO 8.24 SPL VS LOAD (kg) AT POINT A

8.12 ANALYSIS OF POINT "A" W.R.T SPEED

- As observed from Fig.8.21 for speed 1000 the maximum reduction in SPL observed with enclosure is for load 4 kg at 50% vane opening and minimum reduction in SPL is at 100% vane opening at load 6.
- As shown in Fig.8.22 for constant speed 900 the maximum reduction in SPL observed with enclosure is for load 2 kg at 50% vane opening and minimum reduction in SPL is at 0% vane opening at load 6.
- As shown in Fig.8.23 for speed 800 almost all curves experience similar reduction with maximum reduction in SPL observed with enclosure is for load 8 kg at 0%.
- As shown in Fig.8.24 for speed 700 the maximum reduction in SPL observed with enclosure is for load 0 kg at 0% vane opening and minimum reduction in SPL is at 50% vane opening at load 4.

8.13 GRAPHS FOR POINT-“B” W.R.T CONSTANT SPEED

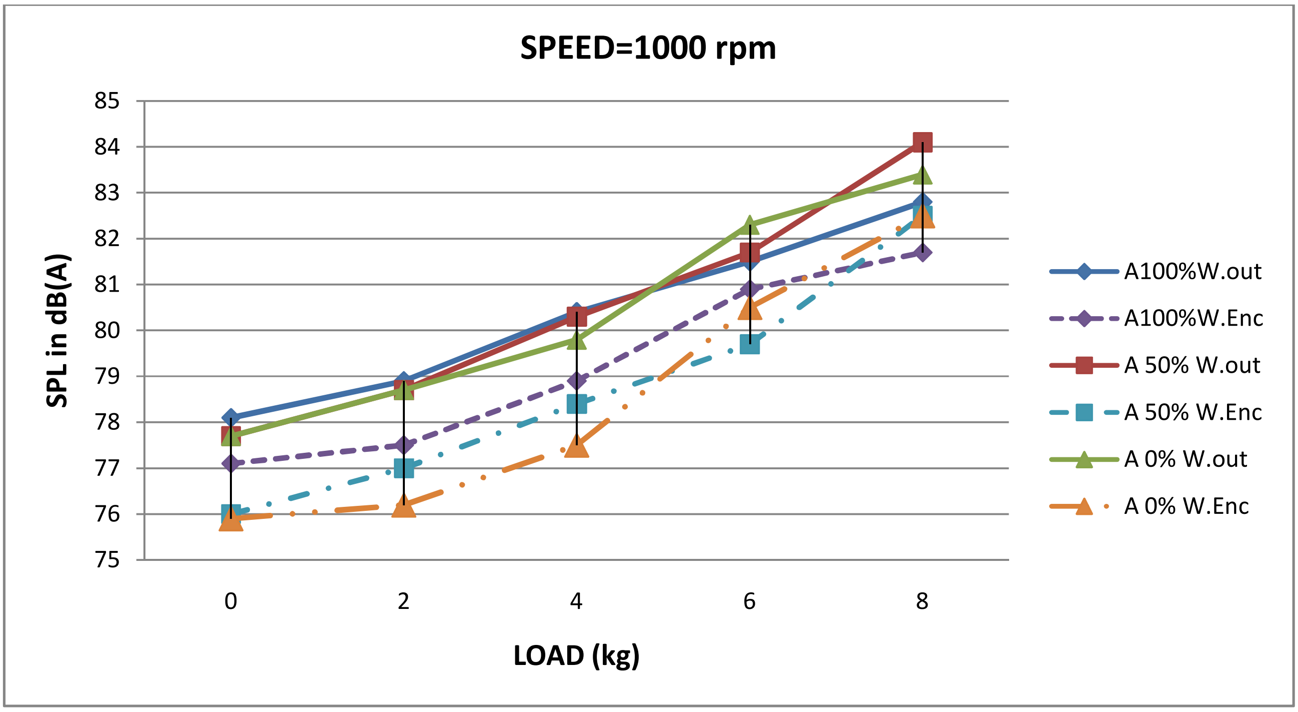


FIGURE NO 8.25 SPL VS LOAD (kg) AT POINT B

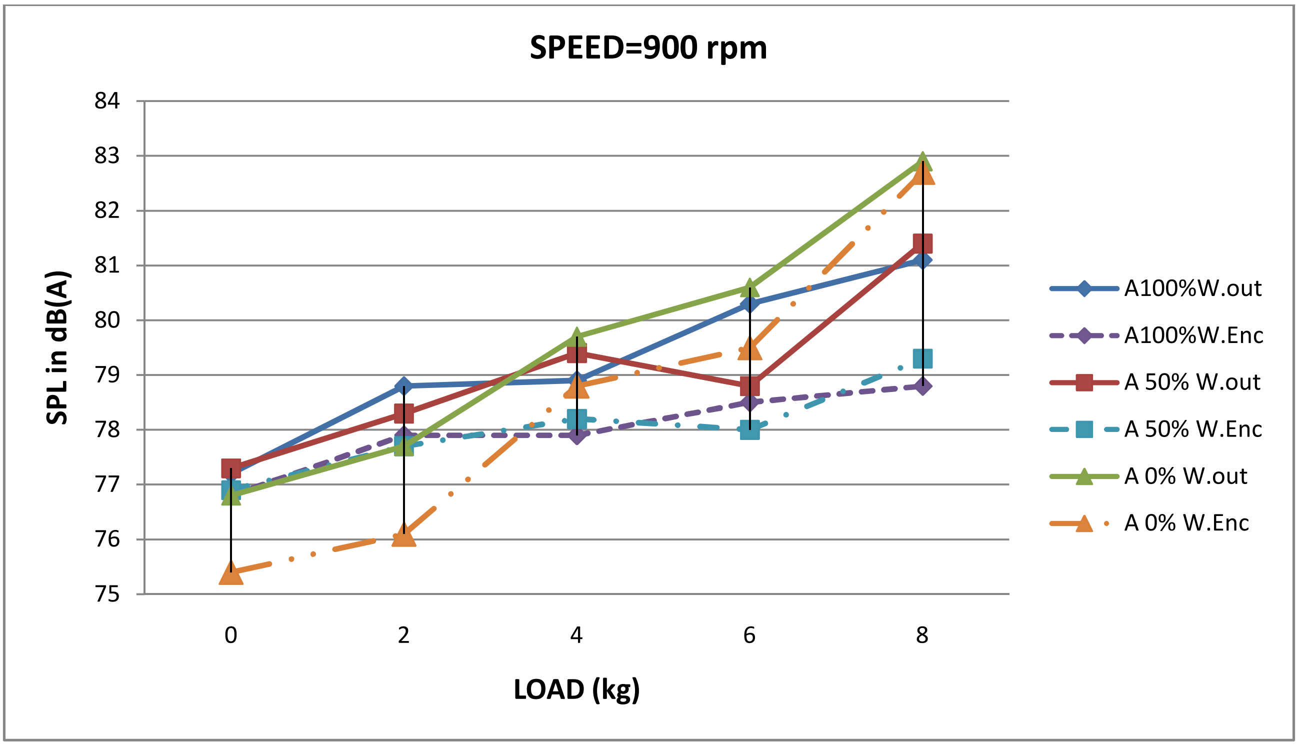


FIGURE NO 8.26 SPL VS LOAD (kg) AT POINT B

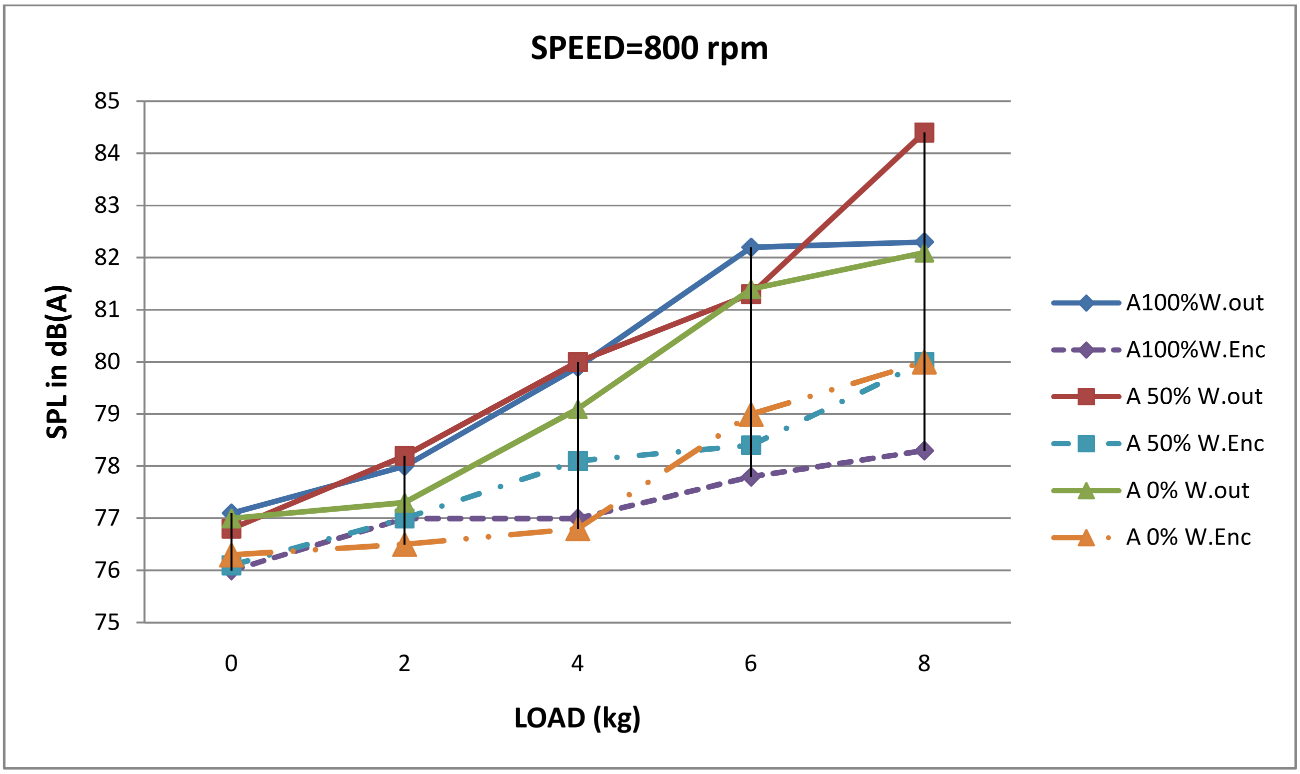


FIGURE NO 8.27 SPL VS LOAD (kg) AT POINT B

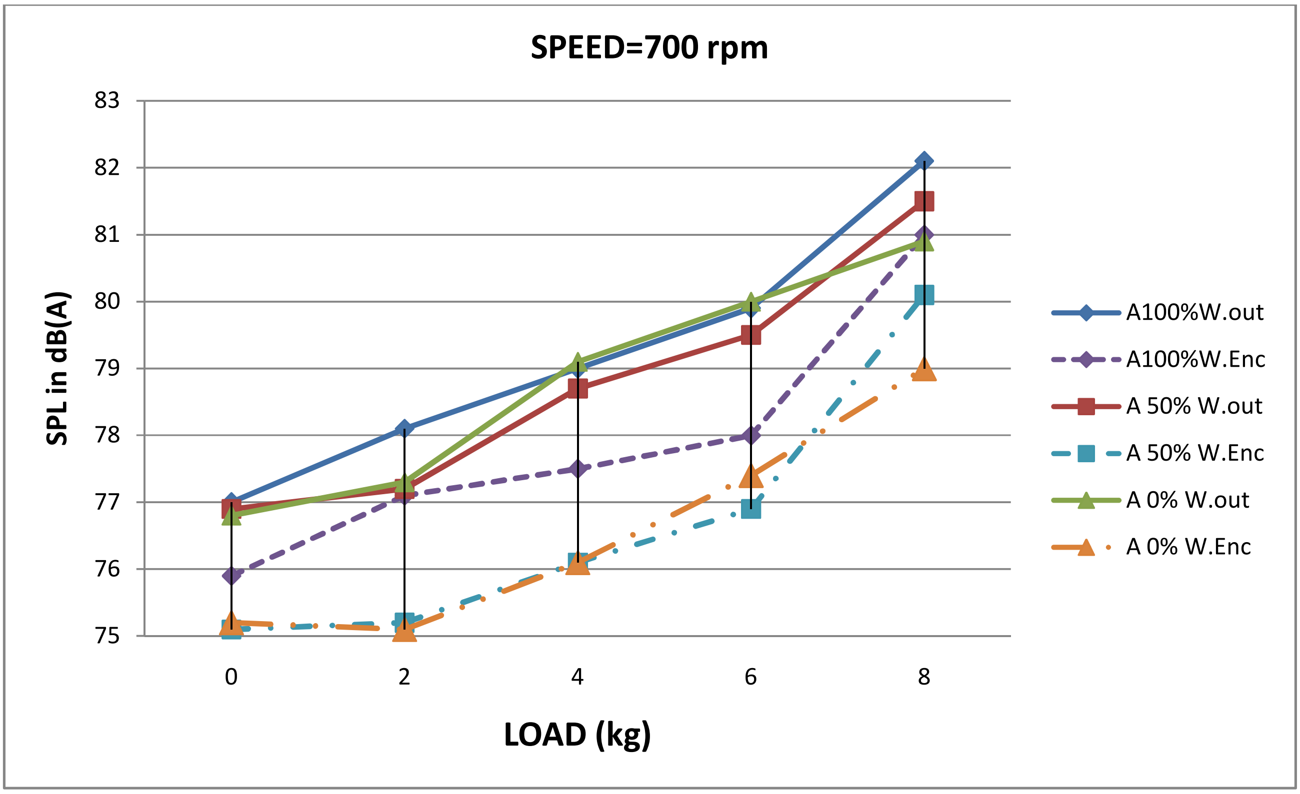


FIGURE NO 8.28 SPL VS LOAD (kg) AT POINT B

8.14 ANALYSIS OF POINT "B" W.R.T SPEED

- In fig 8.25 one can observe small reduction in all the curves
- As shown in Fig.8.26 for constant speed 900 the maximum reduction in SPL observed with enclosure is for load 8 at 100% vane opening and minimum reduction in SPL is at 0% vane opening at load 8 kg.
- As shown in Fig.8.27 shows linear type of behavior till load 6 an reduction in SPL increases with increase in load
- In Fig.8.28 for the maximum reduction in SPL observed with enclosure is for load 4 at 0% vane opening and minimum reduction in SPL is at 100% vane opening at load 8 kg.

8.15 GRAPHS FOR POINT-"C" W.R.T CONSTANT SPEED

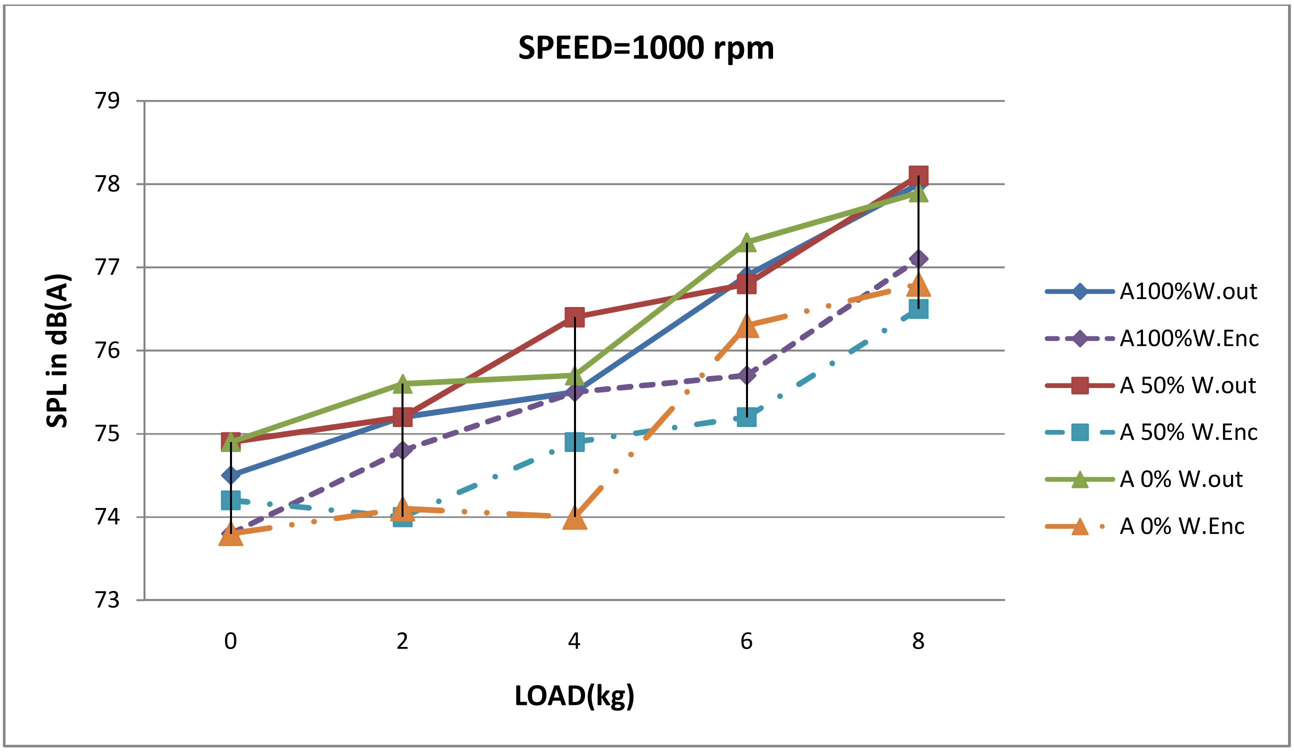


FIGURE NO 8.29 SPL VS LOAD (kg) AT POINT C

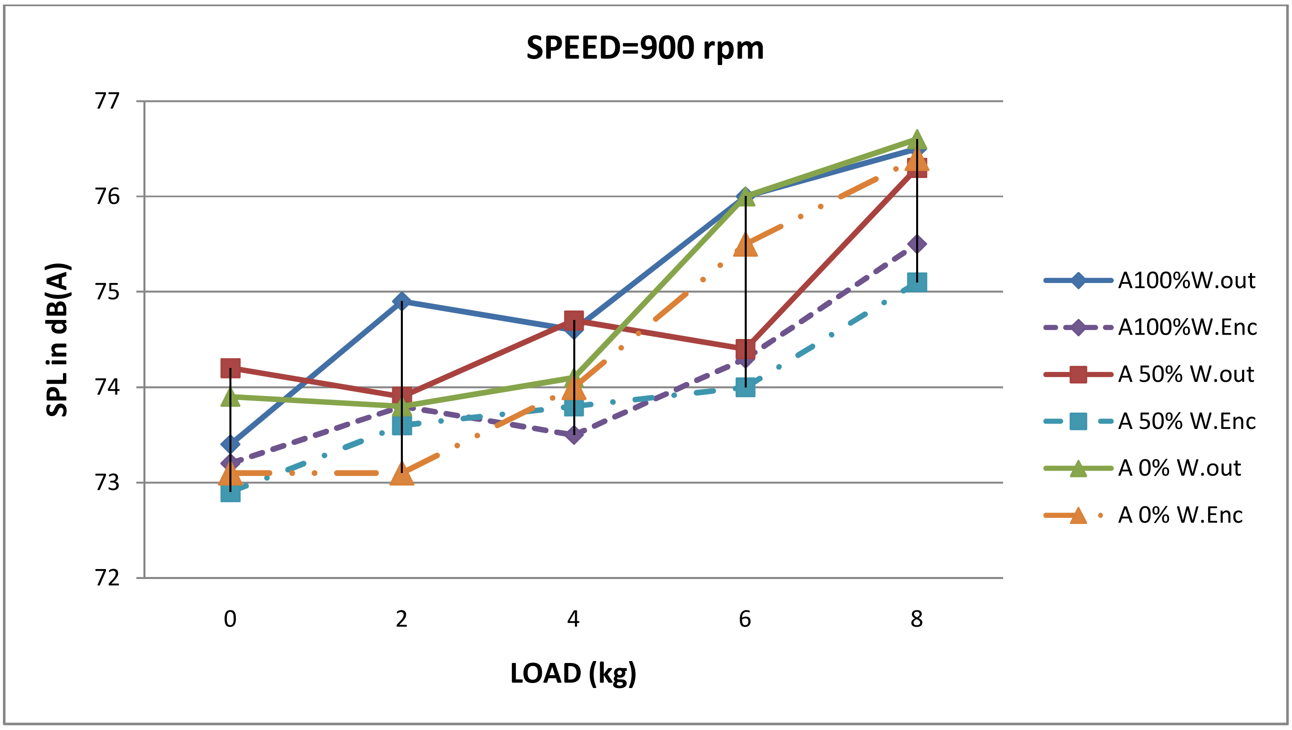


FIGURE NO 8.30 SPL VS LOAD (kg) AT POINT C

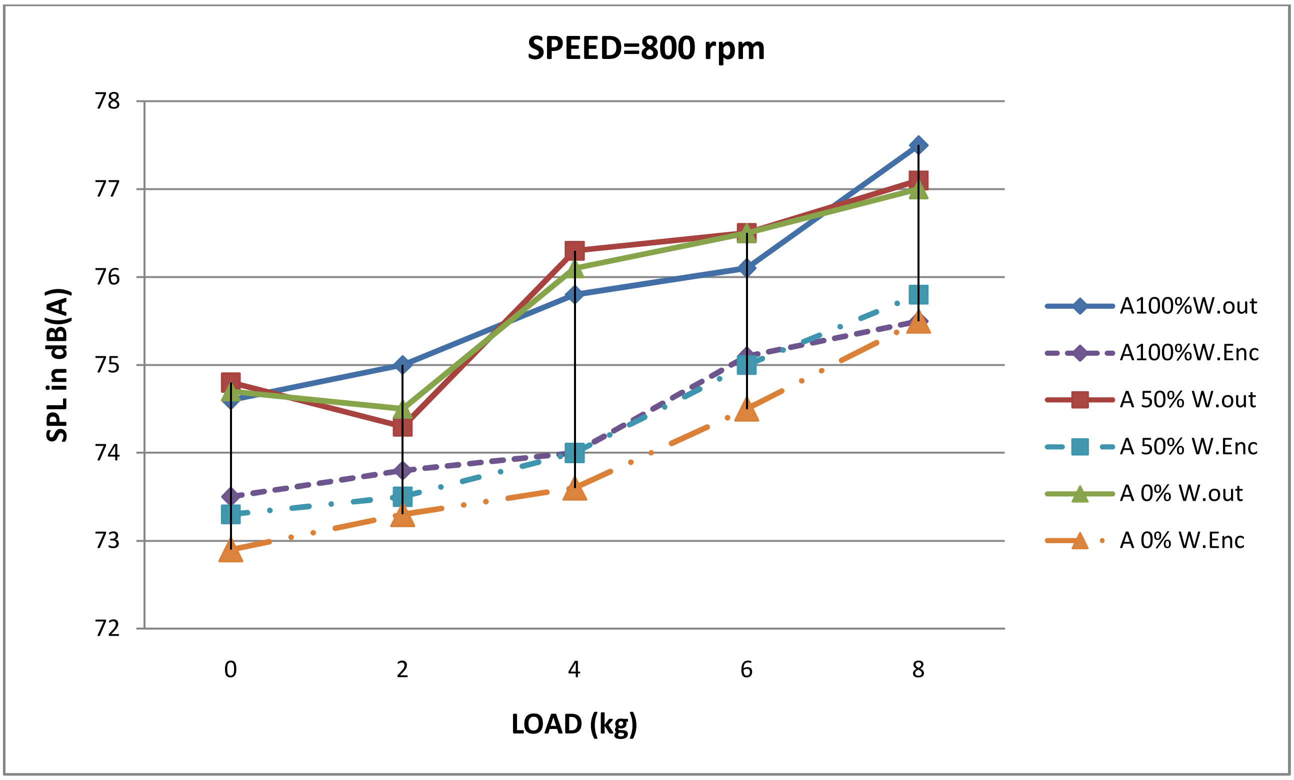


FIGURE NO 8.31 SPL VS LOAD (KG) AT POINT C

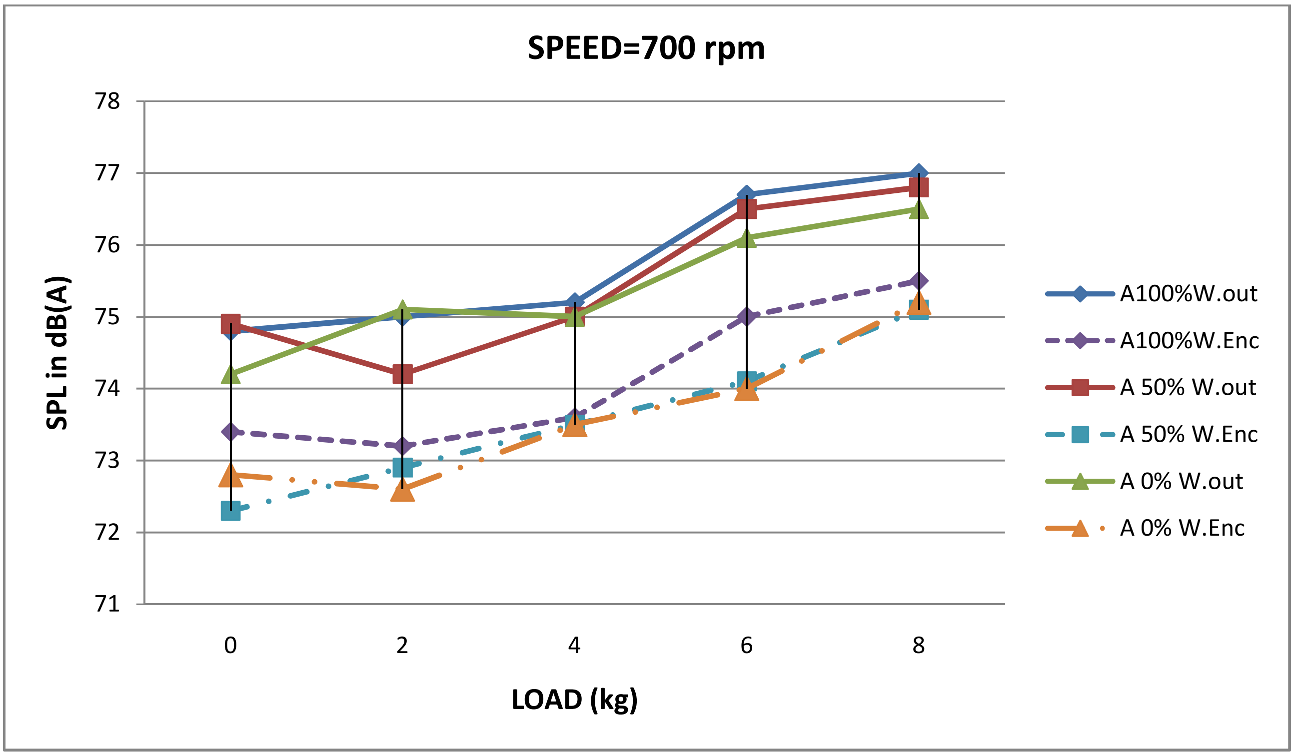


FIGURE NO 8.32 SPL VS LOAD (kg) AT POINT C

8.16 ANALYSIS OF POINT "C" W.R.T SPEED

- In fig 8.29 shows zig zag behavior of curves with minimum reduction of 100% vane opening at load 4 kg.
- As per the observations Fig.8.30 follows the same zig zag behavior as 6.29.
- As shown in Fig.8.31 the maximum reduction in SPL observed with enclosure is for load 4 kg at 0% vane opening and minimum reduction in SPL is at 50% vane opening at load 2.
- In the Fig.8.32 the maximum reduction in SPL observed with enclosure is for load 2 kg at 0% vane opening.

8.17 GRAPHS FOR POINT-"D" W.R.T CONSTANT SPEED

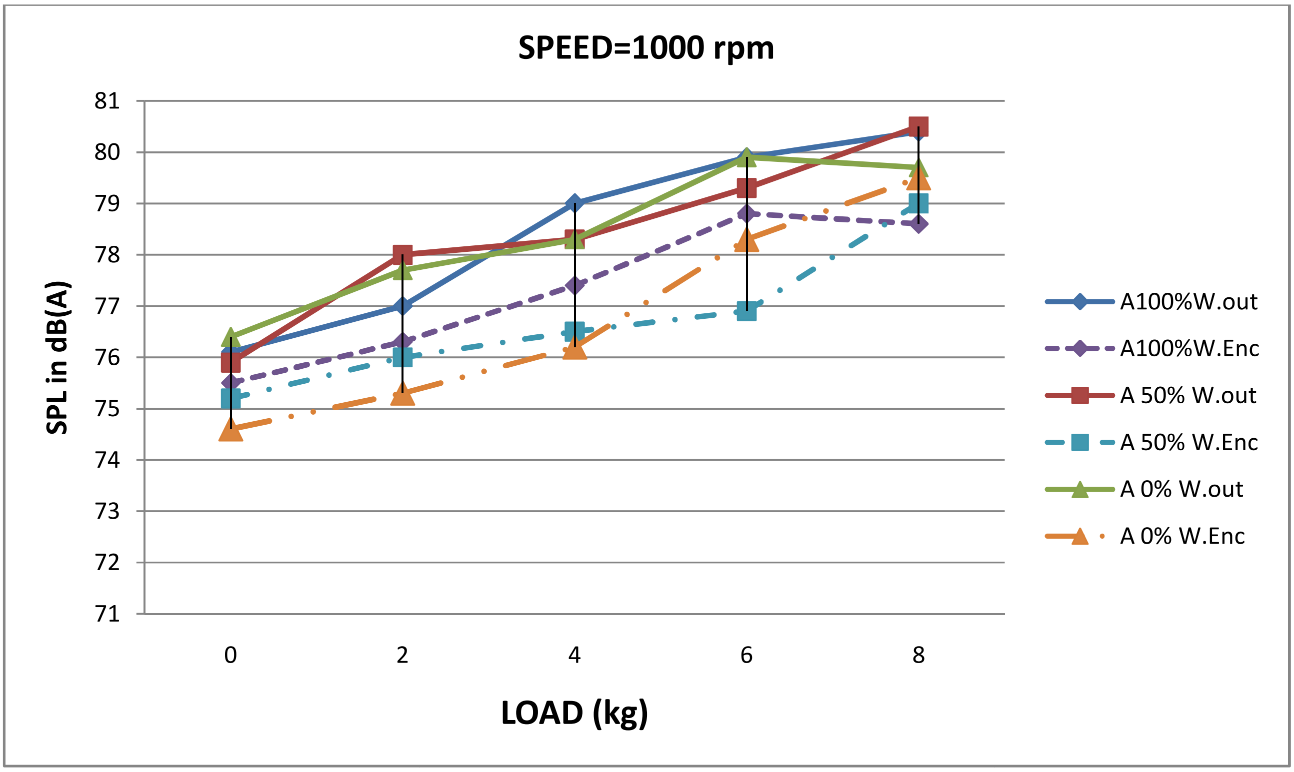


FIGURE NO 8.33 SPL VS LOAD (kg) AT POINT D

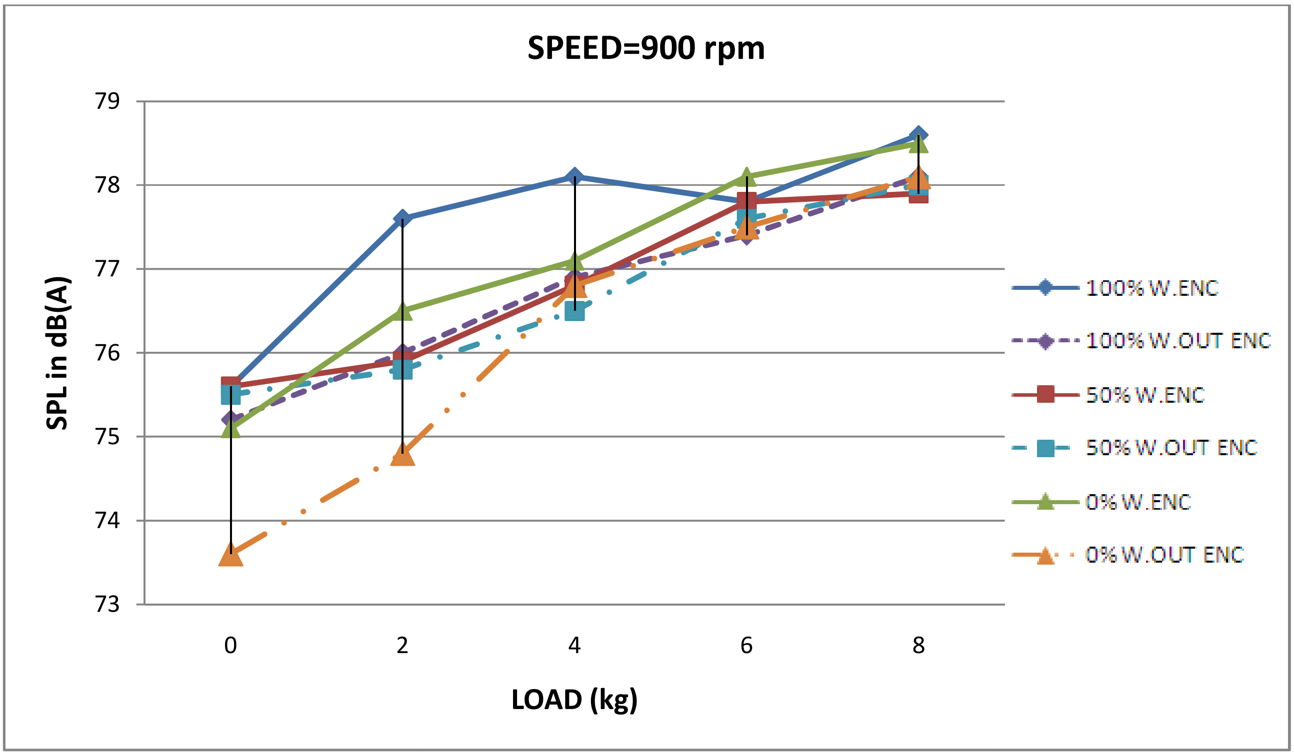


FIGURE NO 8.34 SPL VS LOAD (kg) AT POINT D

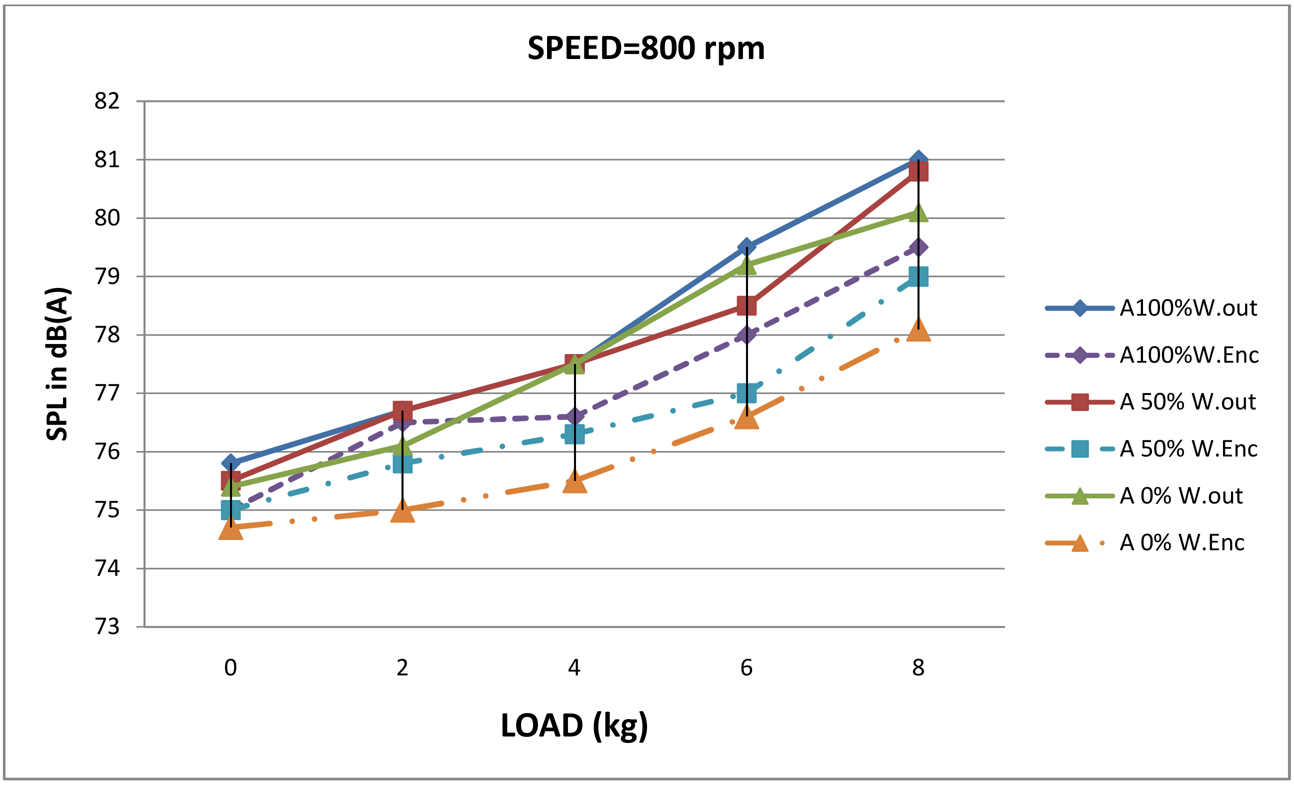


FIGURE NO 8.35 SPL VS LOAD (kg) AT POINT D

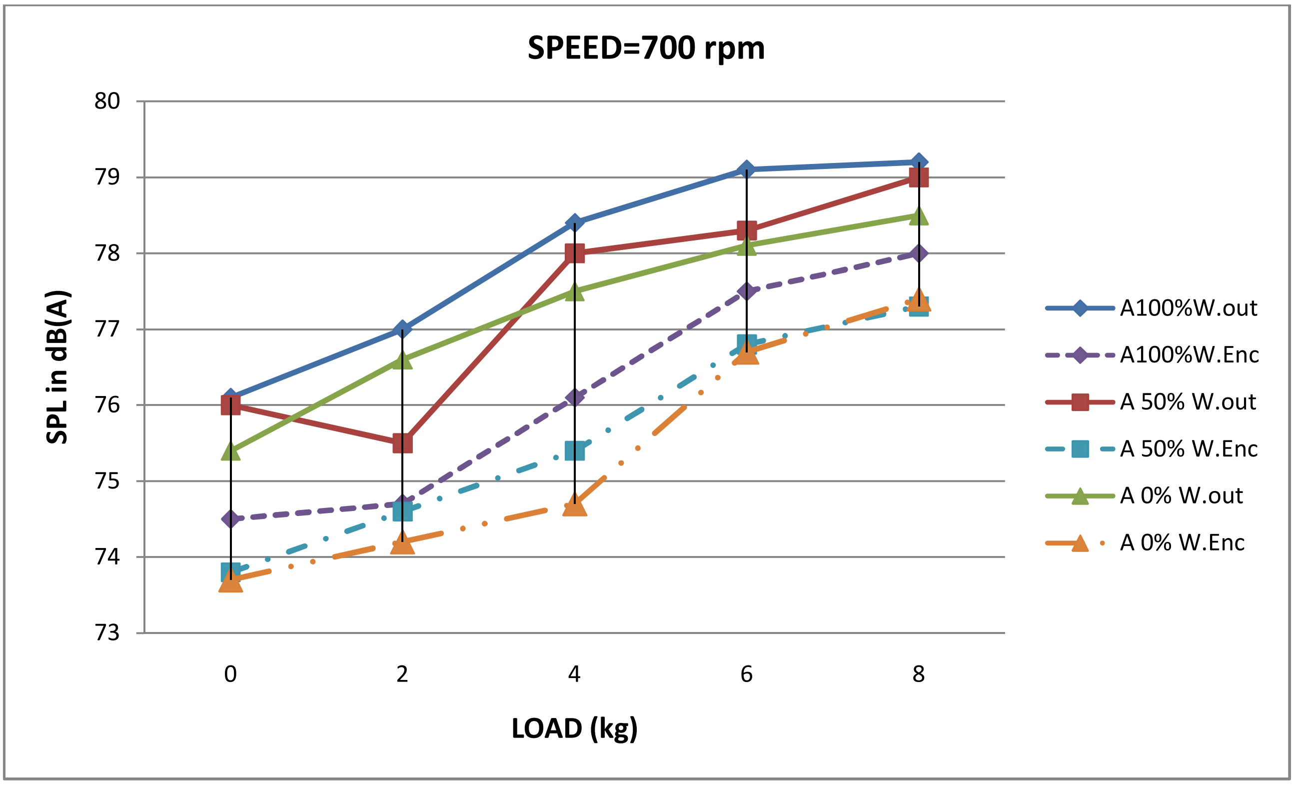


FIGURE NO 8.36 SPL VS LOAD (kg) AT POINT D

8.18 ANALYSIS OF POINT “D” W.R.T SPEED

ANALYSIS FOR POINT-“D” W.R.T CONSTANT PRESSURE HEADS

- In fig 8.33 follows the similar behavior as 8.25.
- In the Fig.8.34 one can observe small reduction in 50% vane opening as compared to others.
- As shown in Fig.8.35 reduction increases with increase in load
- In the Fig.8.36 maximum reduction of 0% vane opening curve at load 4 and minimum of 50% at load 2.

8.19 GRAPHS FOR POINT-“E” W.R.T CONSTANT SPEED

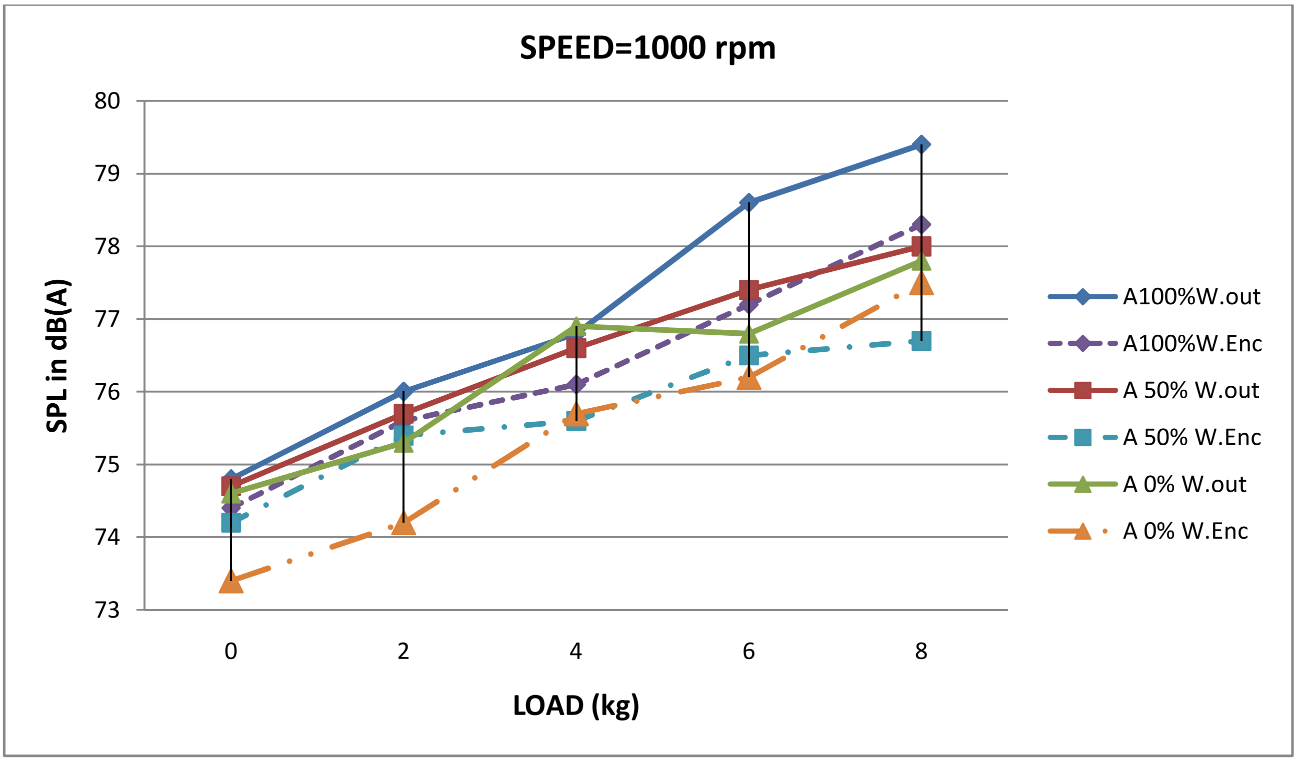


FIGURE NO 8.37 SPL VS LOAD (kg) AT POINT E

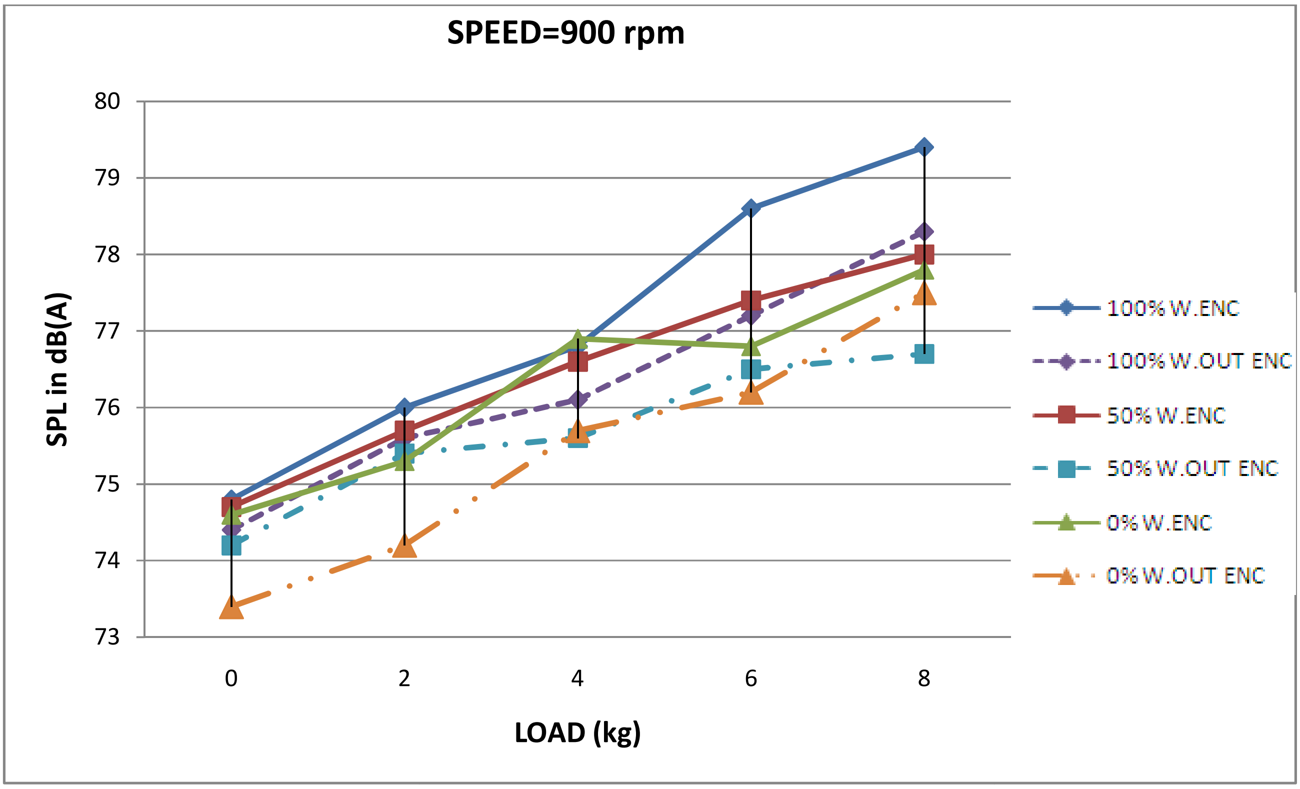


FIGURE NO 8.38 SPL VS LOAD (kg) AT POINT E

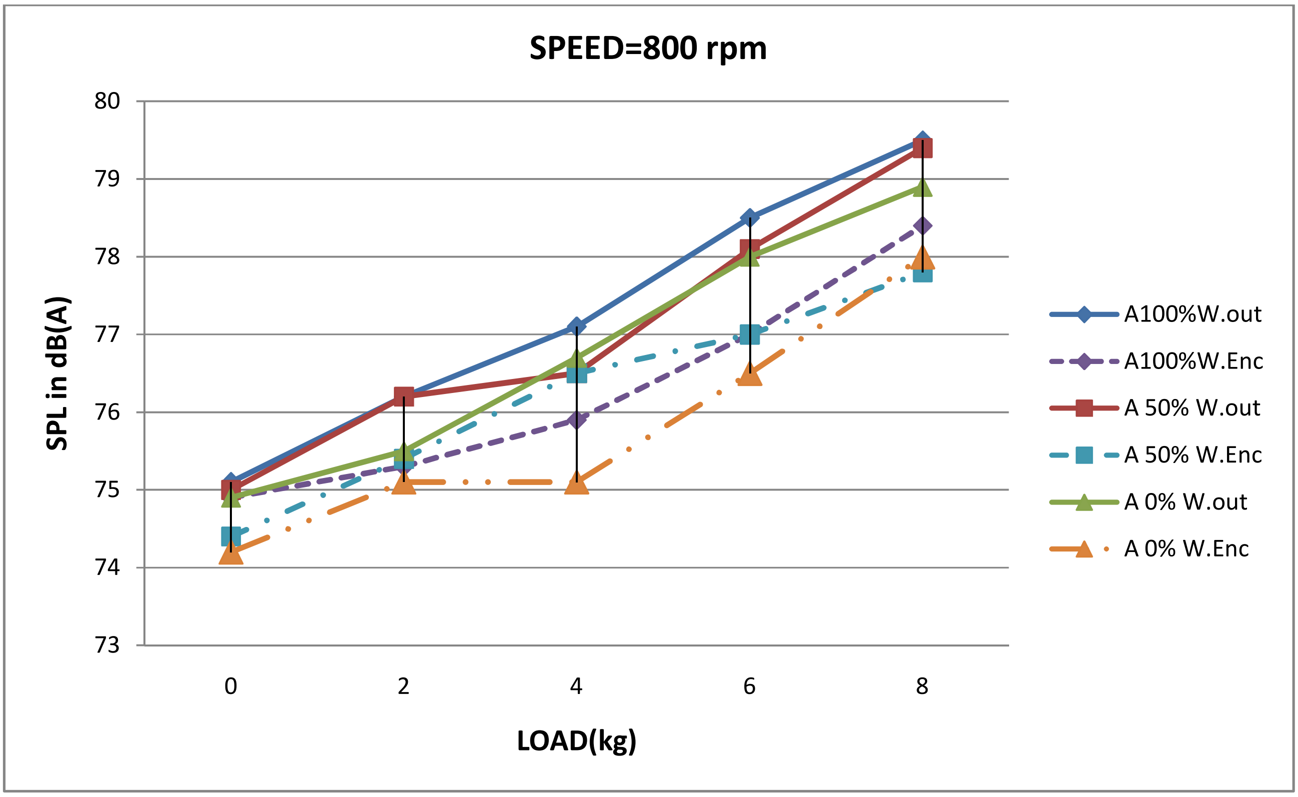


FIGURE NO 8.39 SPL VS LOAD (kg) AT POINT E

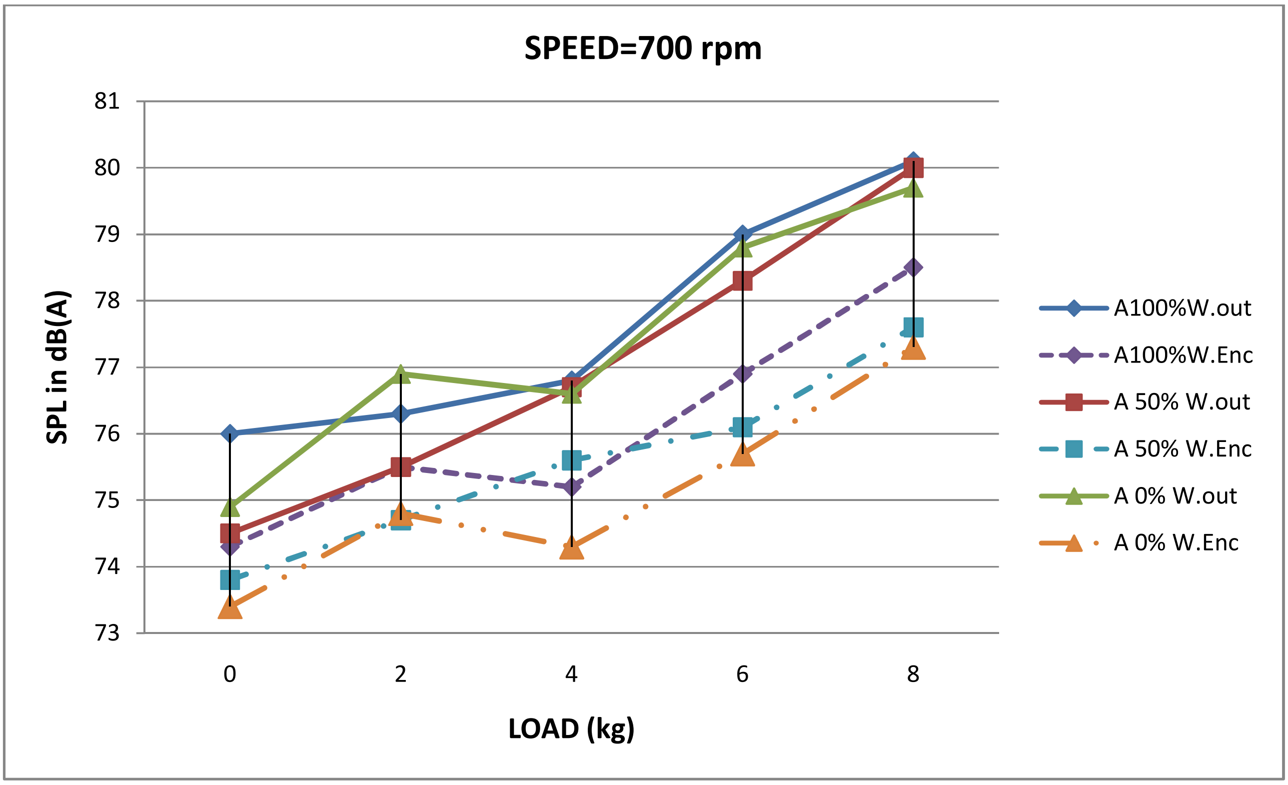


FIGURE NO 8.40 SPL VS LOAD (kg) AT POINT E

8.20ANALYSIS FOR POINT-“E” W.R.T CONSTANT SPEED

- In fig 8.37 one can observe 0% curves show greater reduction at smaller loads.
- In Fig.8.38 100% and 50% curves shows small reduction till lower loads but increases at higher loads.
- As shown in Fig.8.39 0% and 100% curves show almost linear behavior
- In the Fig.8.40 maximum reduction of 0% vane opening curve at load 4 and minimum of 50% at load 0.

8.21 FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR MOTOR AND PUMP FOR LOAD 0, 4, 8 kg FOR CONSTANT PRESSURE HEAD AND VANE OPENING

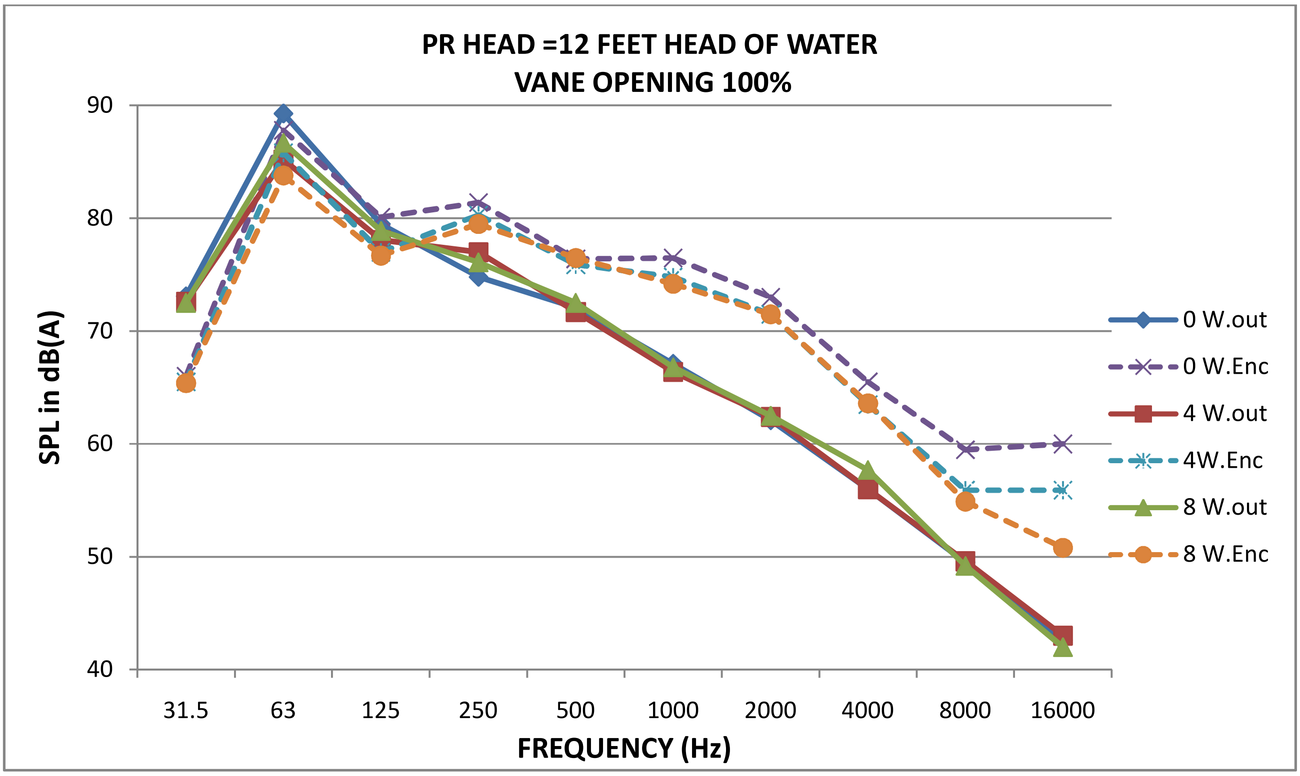


Fig 8.41 SPL Vs FREQUENCY (Hz)

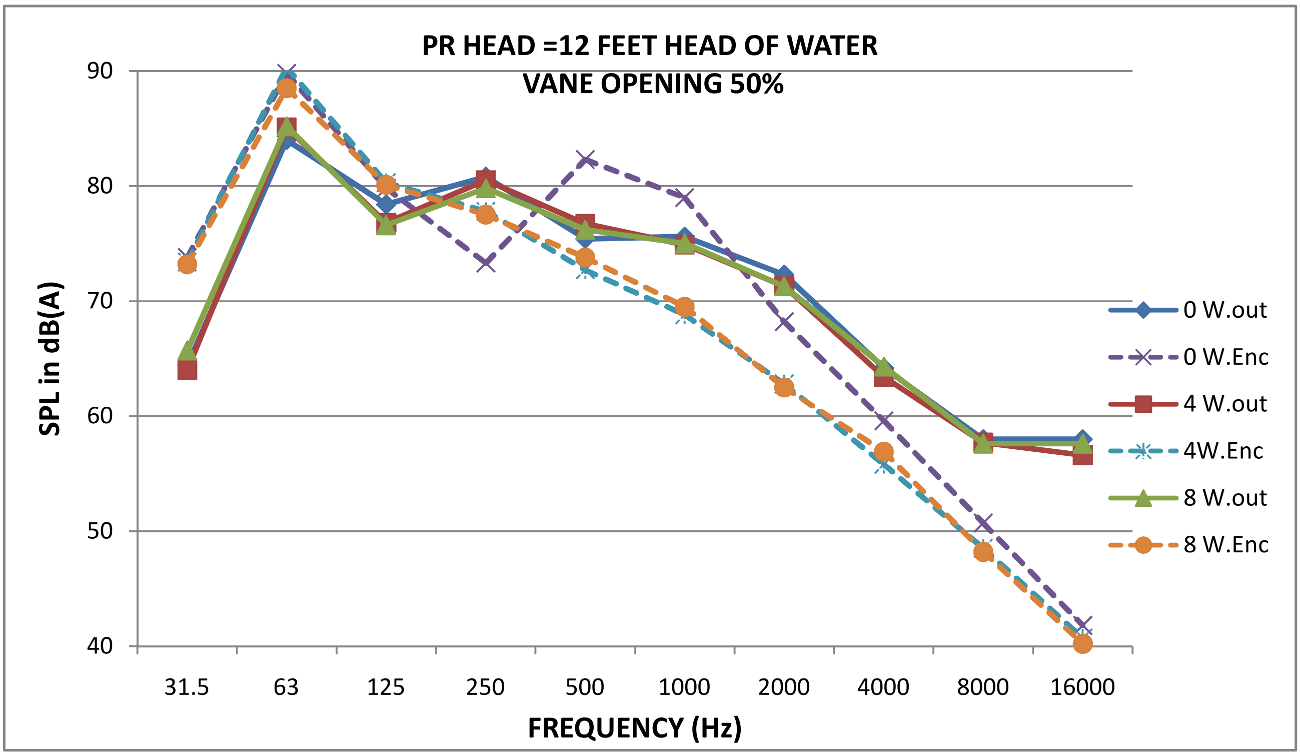


Fig 8.42 SPL Vs FREQUENCY (Hz)

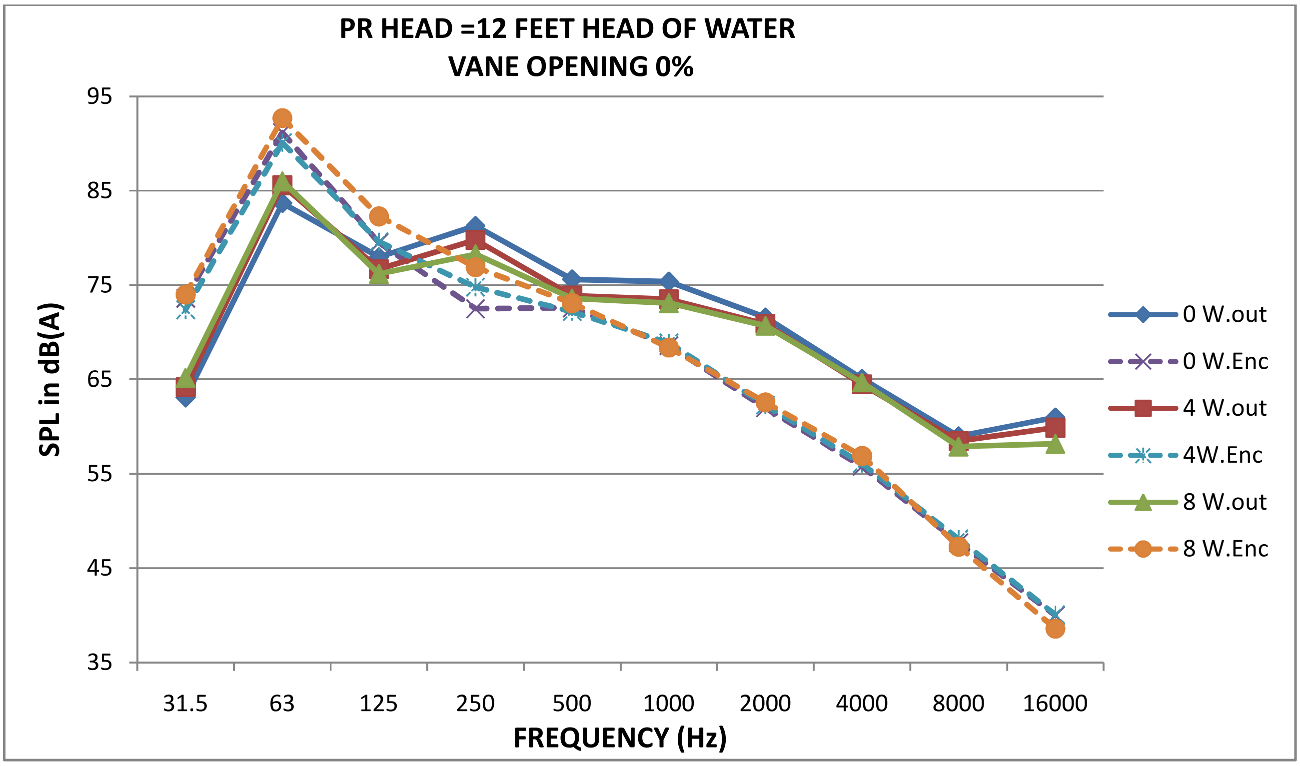


Fig 8.43 SPL Vs FREQUENCY (Hz)

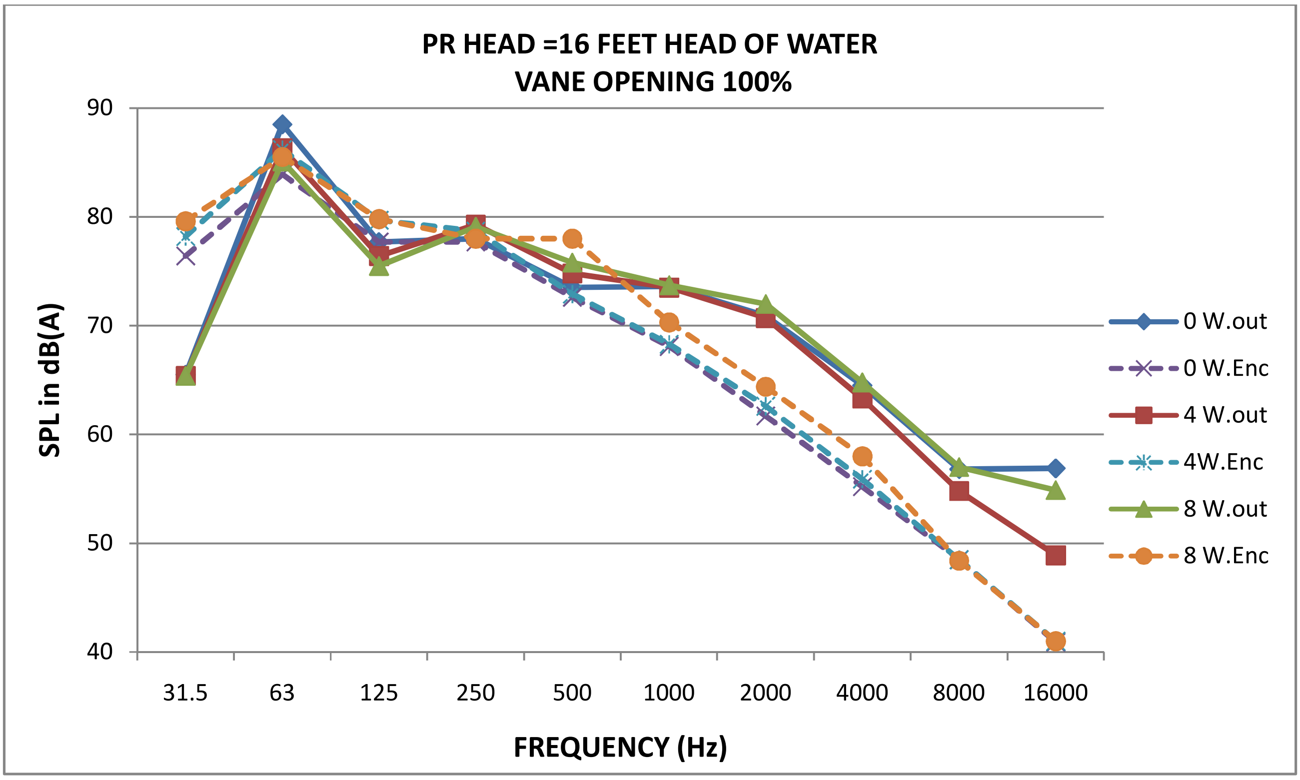


Fig 8.44 SPL Vs FREQUENCY (Hz)

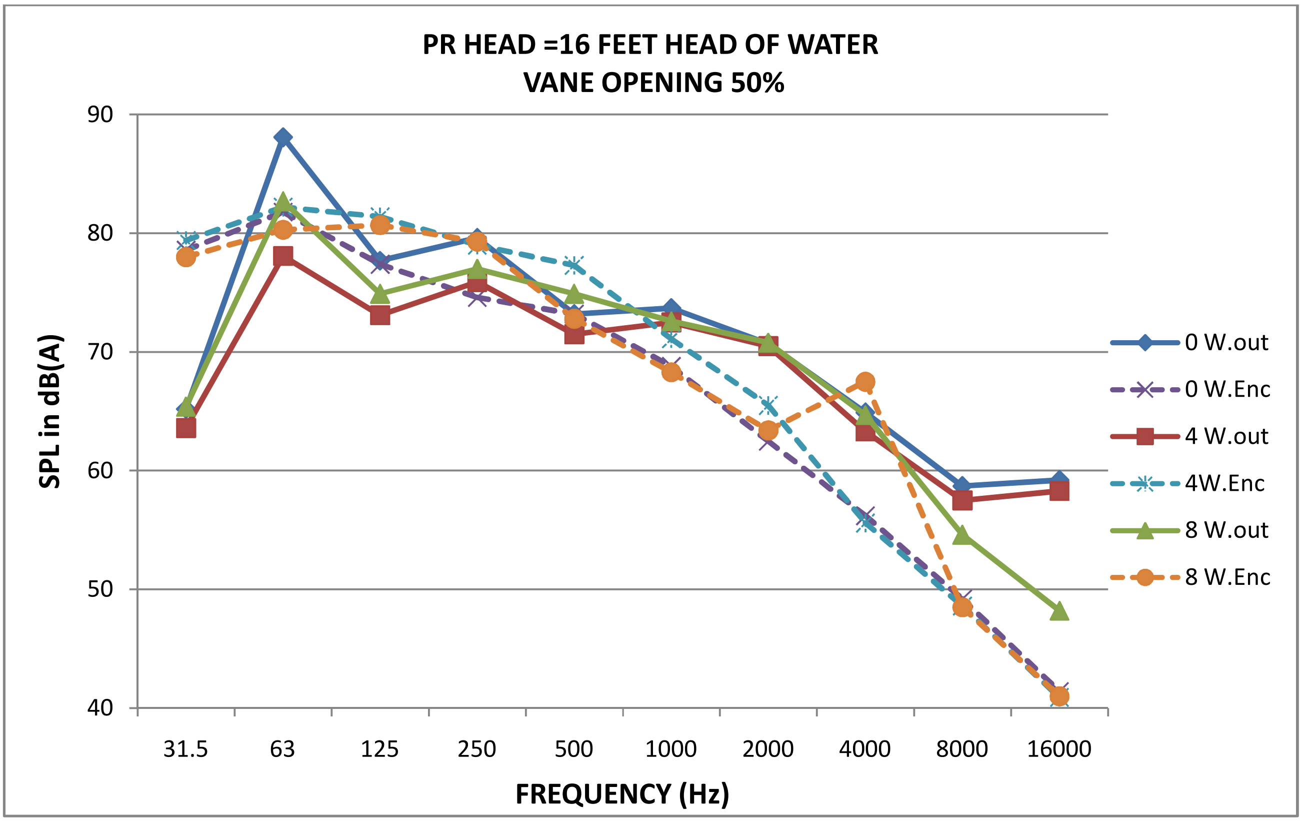


Fig 8.45 SPL Vs FREQUENCY (Hz)

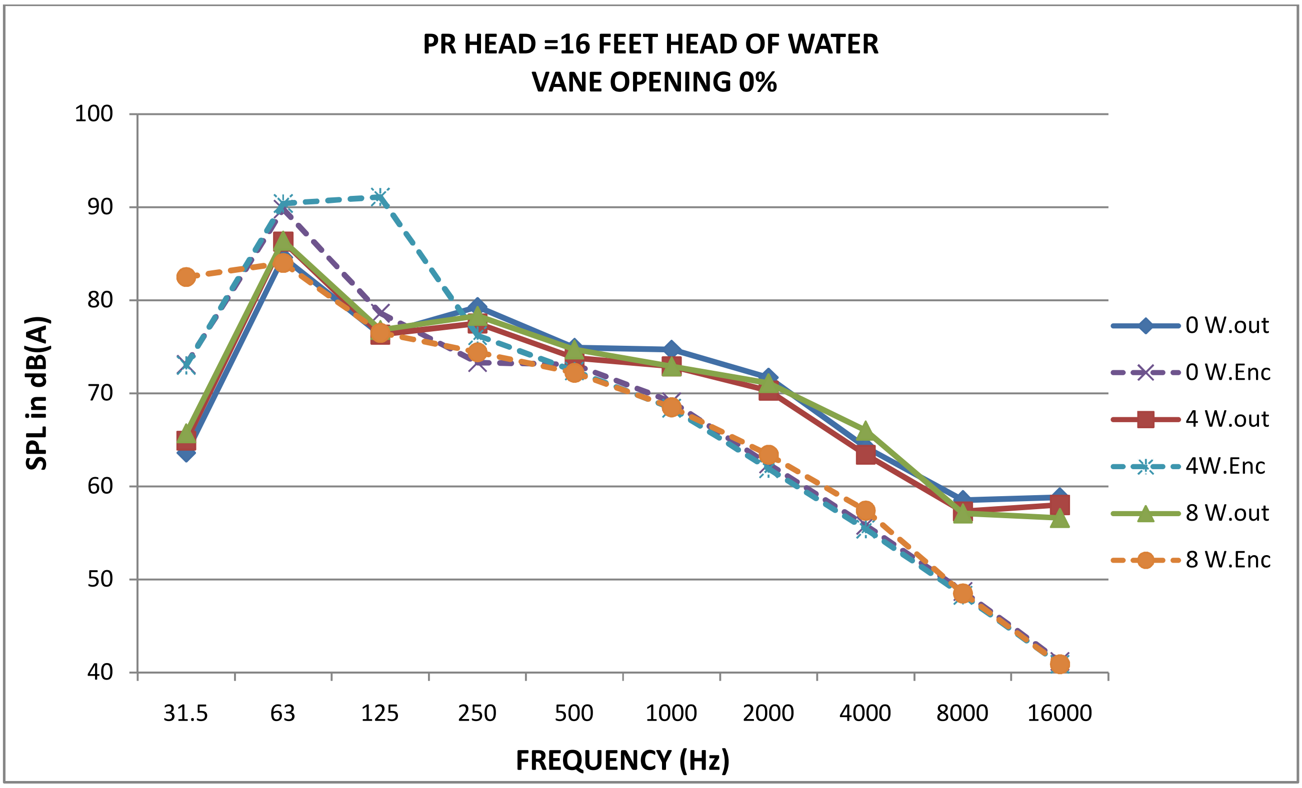


Fig 8.46 SPL Vs FREQUENCY (Hz)

8.22 ANALYSIS OF FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR MOTOR AND PUMP FOR LOAD 0, 4, 8 kg FOR CONSTANT PRESSURE HEAD AND VANE OPENING

8.22.1 FOR PRESSURE HEAD 12

- In fig 8.41 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency range of 31.5 -125 Hz.
- In Fig 8.42 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency 250 Hz and above
- In Fig 8.43 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency 250 Hz and above

8.22.2 FOR PRESSURE HEAD 16

- In fig 8.44 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency 63 Hz. It can also be observed that enclosure is effective for frequency range 250 Hz and above.
- In Fig 8.45 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency 63 Hz and higher frequencies.
- In Fig 8.46 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency 250 Hz and above

8.23 FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR FRANCIS TURBINE FOR LOAD 0, 4, 8 kg AT CONSTANT PRESSURE HEAD AND VANE OPENING

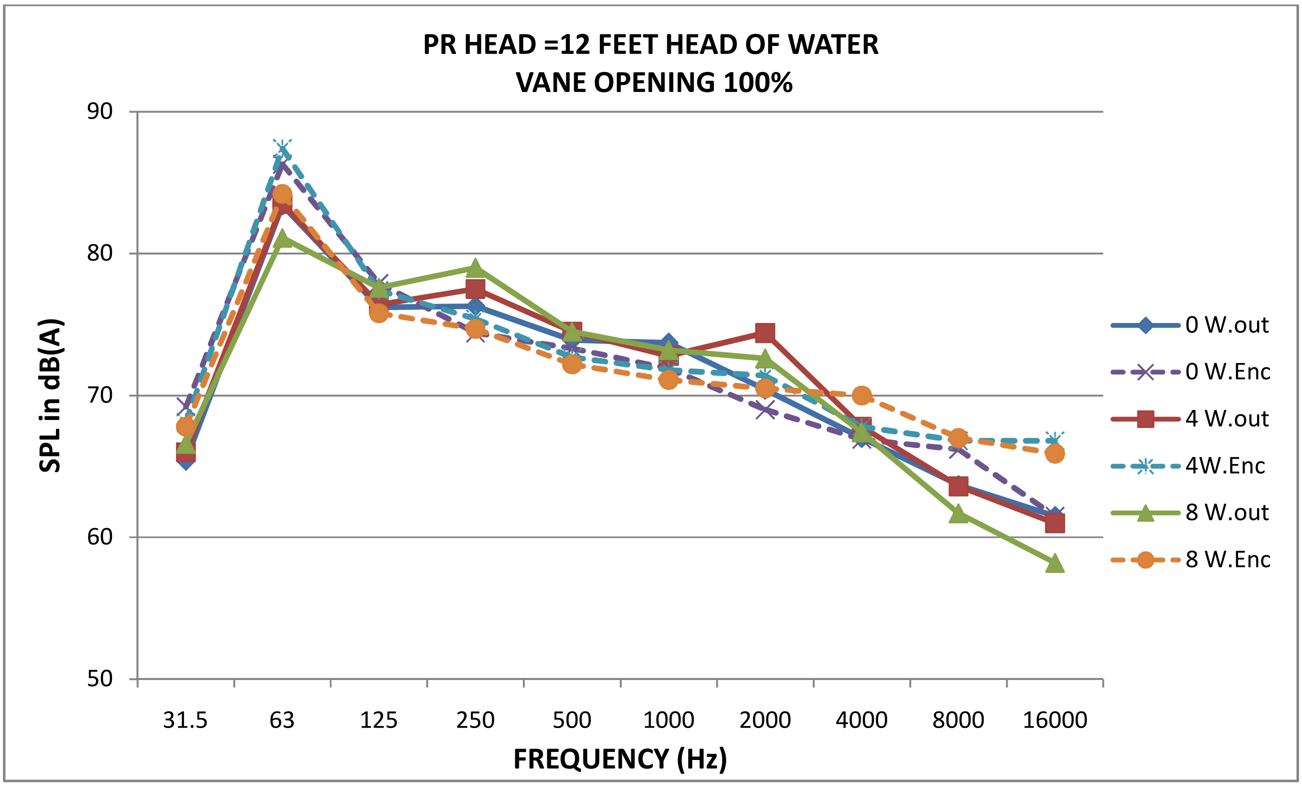


Fig 8.47 SPL Vs FREQUENCY (Hz)

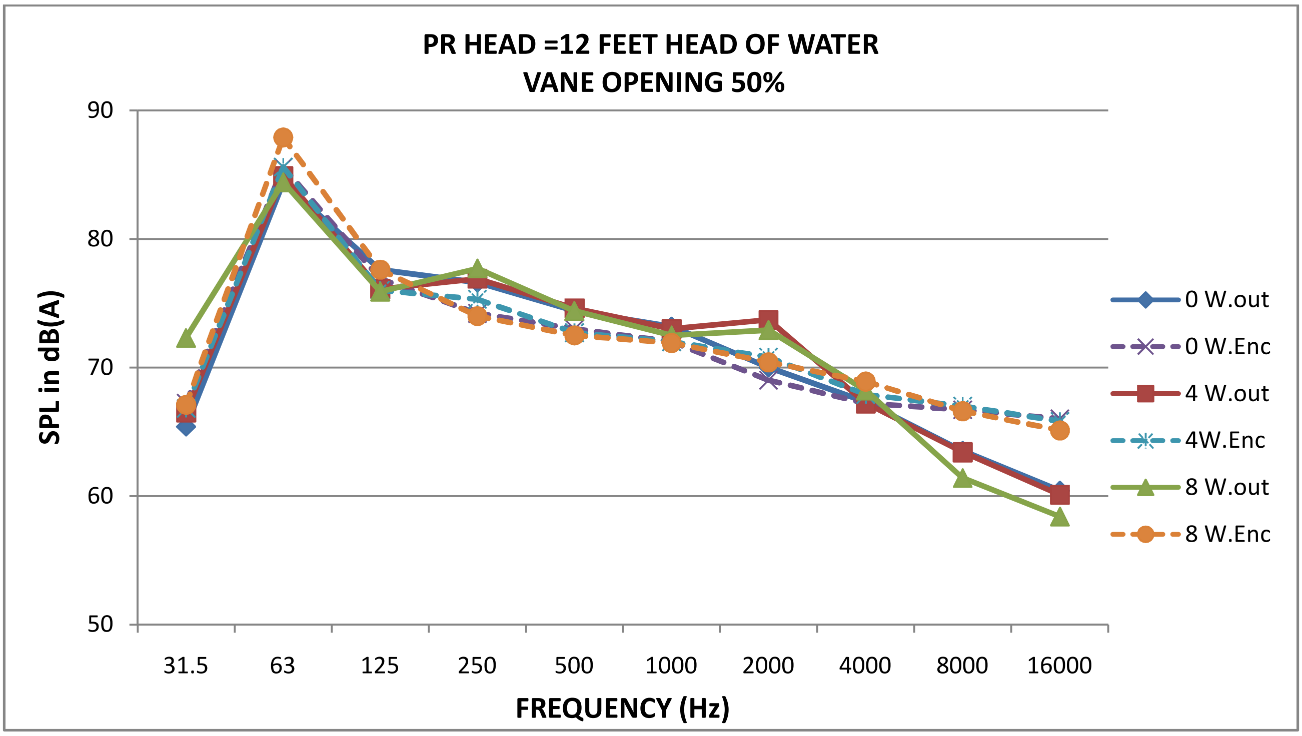


Fig 8.48 SPL Vs FREQUENCY (Hz)

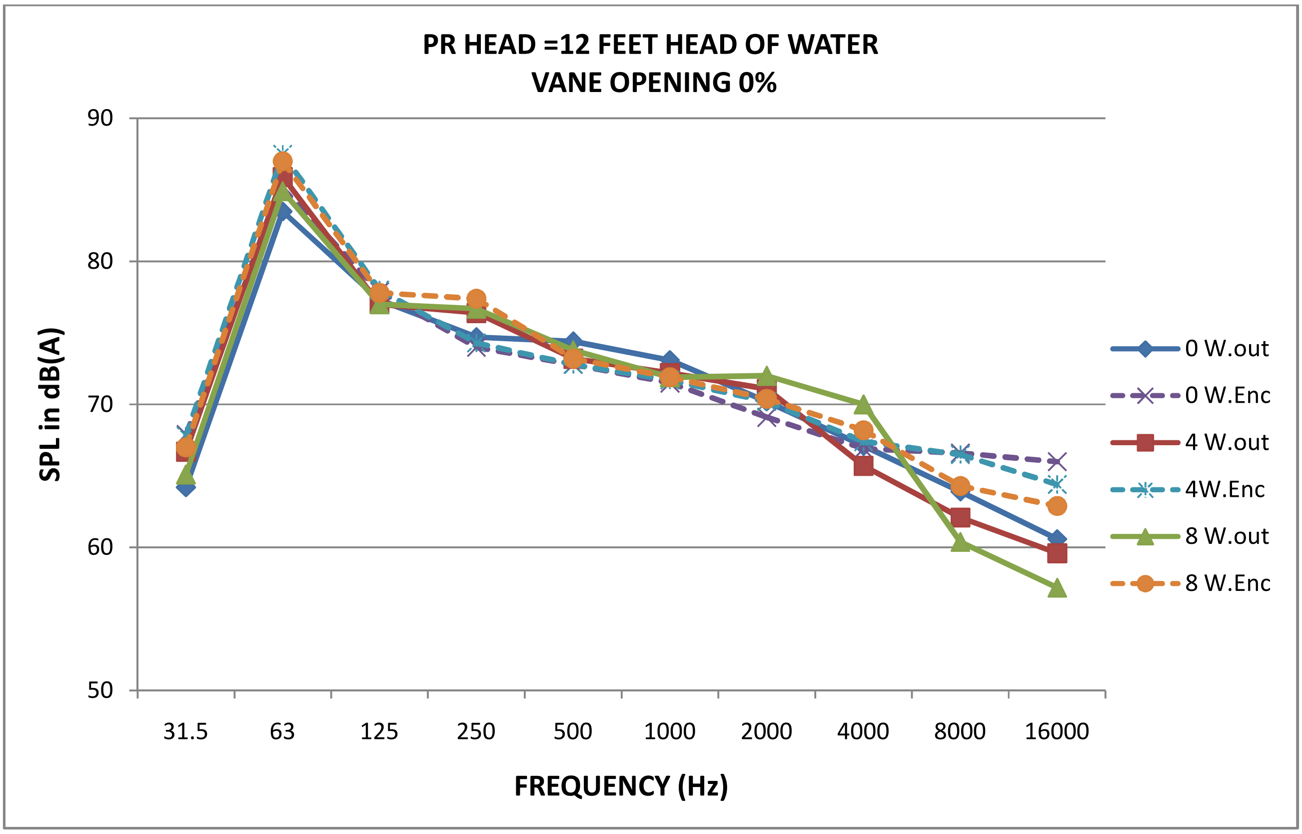


Fig 8.49 SPL Vs FREQUENCY (Hz)

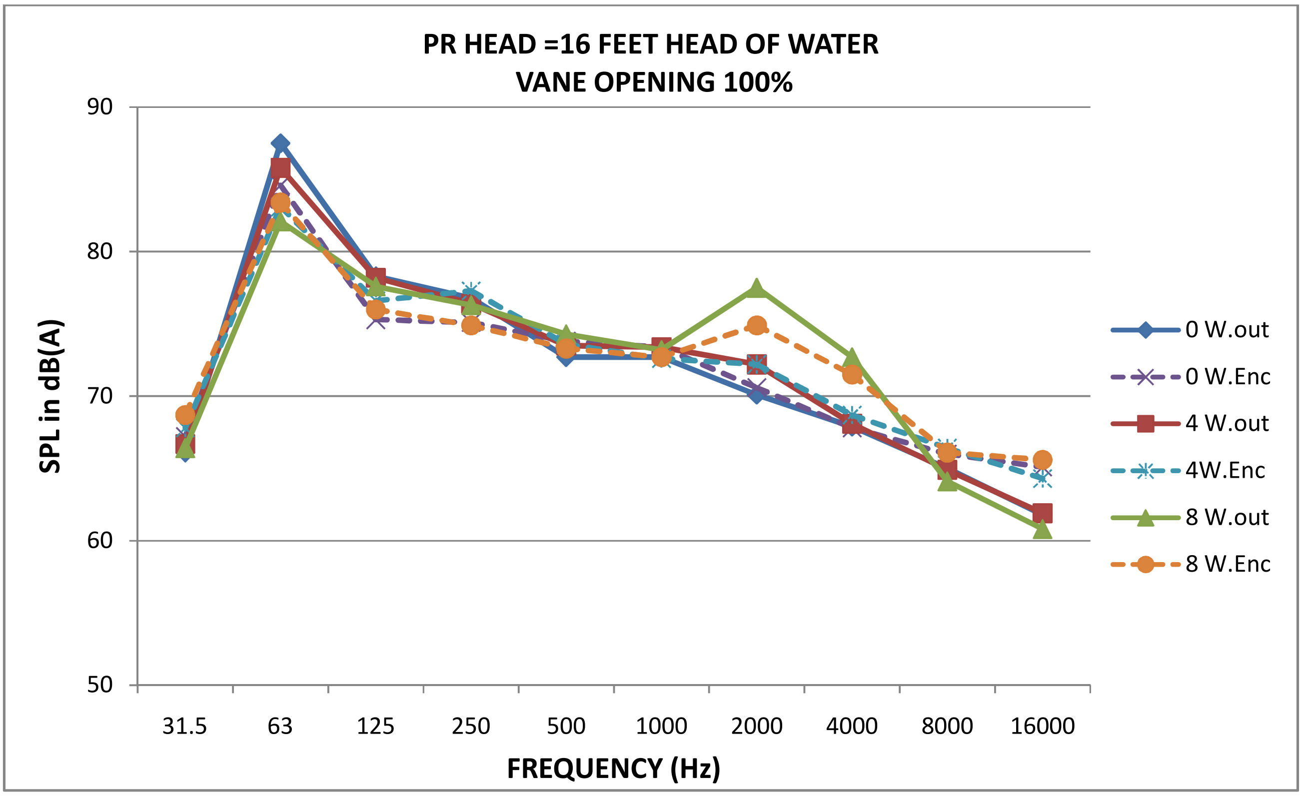


Fig 8.50 SPL Vs FREQUENCY (Hz)

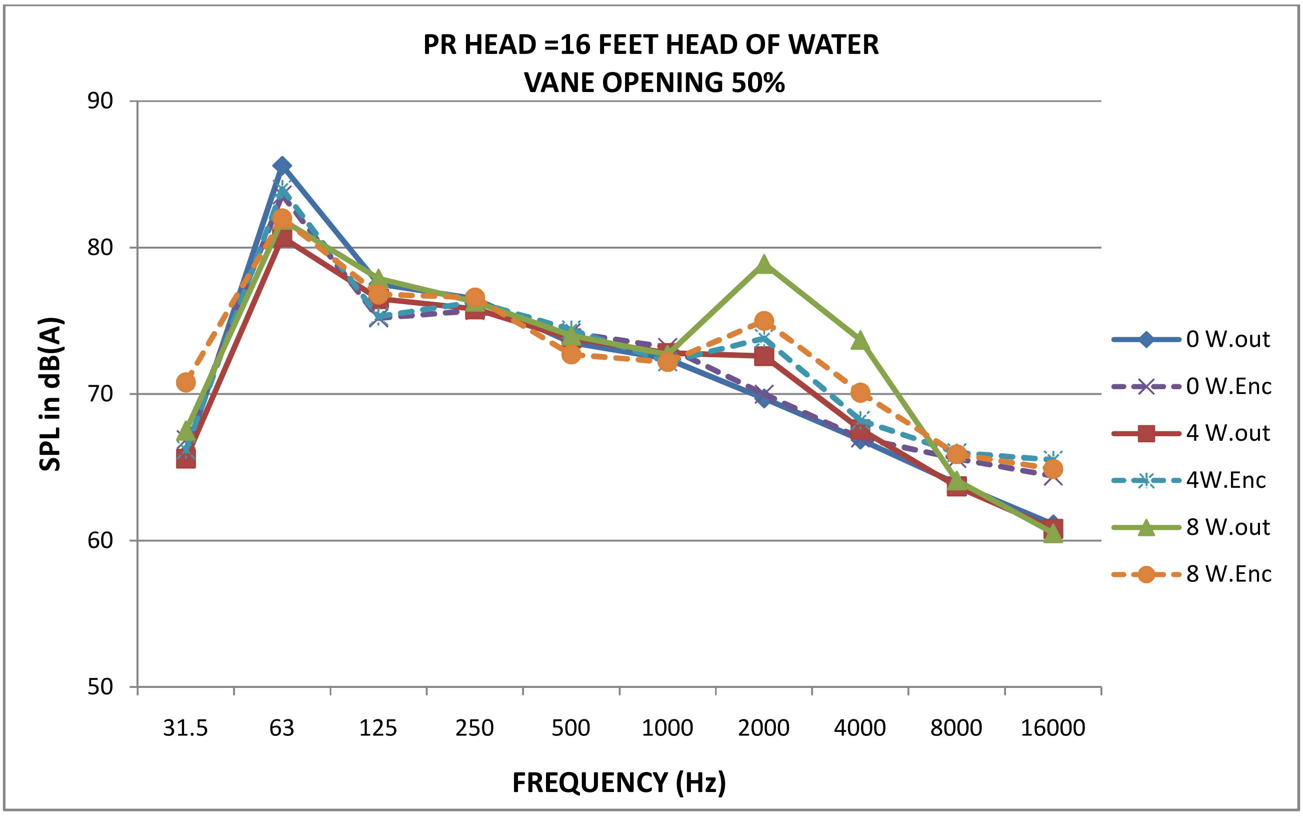


Fig 8.51 SPL Vs FREQUENCY (Hz)

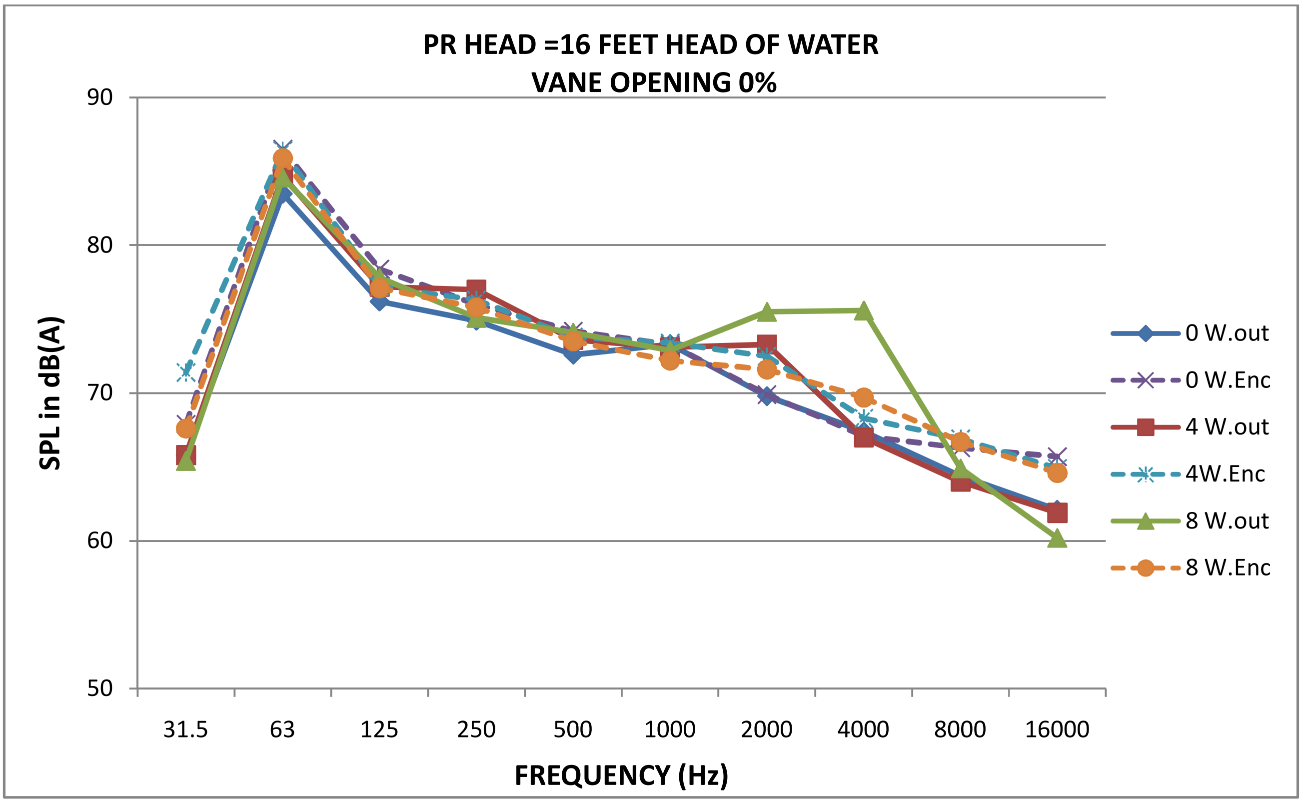


Fig 8.52 SPL Vs FREQUENCY (Hz)

8.24 ANALYSIS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR FRANCIS TURBINE FOR LOAD 0, 4, 8 kg AT CONSTANT PRESSURE HEAD AND VANE OPENING

8.24.1 FOR PRESSURE HEAD 12

- In fig 8.47 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency range of 125-4000 Hz.
- In Fig 8.48 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency range of 250-4000 Hz.
- In Fig 8.49 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that for load 0 kg and 4 kg the enclosure is effective for frequency 125-2000 Hz but for load 8 kg the enclosure is effective for frequency 500-2000 Hz.

8.24.2 FOR PRESSURE HEAD 16

- In fig 8.50 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that the enclosure is effective for frequency range 63-1000 Hz and above.
- In Fig 8.51 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz. One can observe that for load 0 kg and 8 kg the enclosure is effective for frequency 63 Hz but for load 4 kg the enclosure is effective for frequency 125 Hz.
- In Fig 8.52 it is observed that Peak sound pressure level occurs for lower frequencies at 63 Hz.

8.25 SUMMARIZED RESULTS AND DISCUSSIONS

An Analysis of collected data indicates the following results:

8.25.1 EFFECT OF ACOUSTIC ENCLOSURE ON PUMPING SYSTEM

- **Effect of enclosure on sound pressure level at point A1:** It is observed that in pumping system value of sound pressure level at point A1 varies from 75.5 dB(A) to 76.1 dB(A). With the use of acoustic enclosure the value of SPL observed from 65.7 dB(A) to 76.1 dB(A). Maximum attenuation in SPL at point A1 is 10.4 dB(A).
- **Effect of enclosure on sound pressure level at point B1:** It is observed that in pumping system value of sound pressure level at point B1 varies from 72.6 dB(A) to 74 dB(A). With the use of acoustic enclosure the value of SPL observed from 65.3 dB(A) to 67.8 dB(A). Maximum attenuation in SPL at point B1 is 8.7 dB(A).
- **Effect of enclosure on sound pressure level at point C1:** It is observed that in pumping system value of sound pressure level at point C1 varies from 75.4 dB(A) to 76.5 dB(A). With the use of acoustic enclosure the value of SPL observed from 66.5 dB(A) to 66.7 dB(A). Maximum attenuation in SPL at point C1 is 10 dB(A).
- **Effect of enclosure on sound pressure level at point D1:** It is observed that in pumping system value of sound pressure level at point D1 varies from 75.8 dB(A) to 76.7 dB(A). With the use of acoustic enclosure the value of SPL observed from 69.1 dB(A) to 70.5 dB(A). Maximum attenuation in SPL at point D1 is 7.6 dB(A).
- **Effect of enclosure on sound pressure level at point E1:** It is observed that in pumping system value of sound pressure level at point E1 varies from 74.4 dB(A) to 75.2 dB(A). With the use of acoustic enclosure the value of SPL observed from 64.7 dB(A) to 10.5 dB(A). Maximum attenuation in SPL at point E1 is 7.6 dB(A).
- All the points A1, B1, C1, D1 and E1 as shown in fig 6.5 are at a distance of 0.5 m each surrounding the pumping system as was taken for the case of turbine.

8.25.2 EFFECT ON FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR LOAD 0, 4, 8 KG FOR CONSTANT PRESSURE HEAD AND VANE OPENING

8.25.2.1 Effect of vane opening: Maximum SPL in dB(A) is 89.3 at 63 Hz frequency for 100% vane opening. It is observed that with decrease in vane opening the value for SPL decreases.

At 100% vane opening the enclosure is effective for lower frequencies 31.5Hz-63Hz but with decrease in vane opening the effectiveness changes towards higher frequencies in range of 500 Hz-16000Hz

8.25.2.2 Effect of pressure head

Maximum SPL in dB(A) is 89.3 at 63 Hz frequency for PR head 12 feet head of water. It is observed that with increase in PR head the value for SPL peak decreases as it is 88.5 dB(A) at PR head 16 feet head of water .

The effectiveness of enclosure decreases with increase in PR head for lower frequencies 31.5Hz-125Hz but it is same for higher frequencies in range of 500 Hz-16000Hz.

8.25.2.3 Effect of load

The value of SPL shows some variation between the frequency range of 31.5 Hz to 500 Hz with increase in load but it shows almost same values in frequency range of 500Hz-8000 Hz.

8.25.3 EFFECT OF ACOUSTIC ENCLOSURE ON FRANCIS TURBINE

- **Effect of enclosure on sound pressure level at point A:** Value of sound pressure level at point A varies from 74.3 dB(A) to 78 dB(A) for 0 to 8 kg load and for PR head 12 to 18 feet head of water for all vane opening, 100%, 50% and 0%. It is maximum 78 dB(A) at 0% vane opening for PR head 12 feet head of water at 8 kg load. Maximum attenuation in SPL is 4.1 dB(A) using enclosure at 14 PR head at 2 kg load for 0% vane opening. Value of sound pressure level at point A varies from 74 dB(A) to 78.3 dB(A) for 0 to 8 kg load and for speed 700 to 1000 rpm for all vane opening, 100%, 50% and 0%. It is maximum 78.3 dB(A) at 0% vane opening for speed 800 rpm at 8 kg load. Maximum attenuation in SPL is 3.5 dB(A) using enclosure at speed 800 rpm at 6 kg load for 100% vane opening.

- **Effect of enclosure on sound pressure level at point B:** Value of sound pressure level at point B varies from 76.7dB(A) to 83.6 dB(A) for 0 to 8 kg load and for PR head 12 to 18 feet head of water for all vane opening, 100%, 50% and 0%. It is maximum 83.6 dB(A) at 50% vane opening for PR head 12 feet head of water at 8 kg load. Maximum attenuation in SPL is 3.4 dB(A) using enclosure at 12 PR head at 2 kg load for 50% vane opening. Value of sound pressure level at point B varies from 76.8 dB(A) to 84.4 dB(A) for 0 to 8 kg load and for speed 700 to 1000 rpm for all vane opening, 100%, 50% and 0%. It is maximum 84.4 dB(A) at 50% vane opening for speed 800 rpm at 8 kg load. Maximum attenuation in SPL is 4.1 dB(A) using enclosure at speed 800 rpm at 8 kg load for 50% vane opening.
- **Effect of enclosure on sound pressure level at point C:** Value of sound pressure level at point C varies from 74 dB(A) to 80dB(A) for 0 to 8 kg load and for PR head 12 to 18 feet head of water for all vane opening, 100%, 50% and 0%. It is maximum 80 dB(A) at 0% vane opening for PR head 12 feet head of water at 8 kg load. Maximum attenuation in SPL is 2.5 dB(A) using enclosure at 14 PR head at 8 kg load for 0% vane opening. Value of sound pressure level at point C varies from 73.4 dB(A) to 78 dB(A) for 0 to 8 kg load and for speed 700 to 1000 rpm for all vane opening, 100%, 50% and 0%. It is maximum 78 dB(A) at 100% vane opening for speed 1000 rpm at 8 kg load. Maximum attenuation in SPL is 2.5 dB(A) using enclosure at speed 700 rpm at 0 kg load for 50% vane opening.

- **Effect of enclosure on sound pressure level at point D:** Value of sound pressure level at point D varies from 75.6 dB(A) to 82.4 dB(A) for 0 to 8 kg load and for PR head 12 to 18 feet head of water for all vane opening, 100%, 50% and 0%. It is maximum 82.4 dB(A) at 0% vane opening for PR head 12 feet head of water at 8 kg load. Maximum attenuation in SPL is 2.3 dB(A) using enclosure at 14 PR head at 8 kg load for 50% vane opening. Value of sound pressure level at point D varies from 75.4 dB(A) to 81 dB(A) for 0 to 8 kg load and for speed 700 to 1000 rpm for all vane opening, 100%, 50% and 0%. It is maximum 81 dB(A) at 100% vane opening for speed 800 rpm at 8 kg load. Maximum attenuation in SPL is 2.5 dB(A) using enclosure at speed 800 rpm at 6 kg load for 0% vane opening.

- **Effect of enclosure on sound pressure level at point E:** Value of sound pressure level at point E varies from 75.1 dB(A) to 83.9 dB(A) for 0 to 8 kg load and for PR head 12 to 18 feet head of water for all vane opening, 100%, 50% and 0%. It is maximum 83.9 dB(A) at 0% vane opening for PR head 12 feet head of water at 8 kg load. Maximum attenuation in SPL is 2.2 dB(A) using enclosure at 14 PR head at 0 kg load for 100% vane opening. Value of sound pressure level at point E varies from 74.6 dB(A) to 80.1 dB(A) for 0 to 8 kg load and for speed 700 to 1000 rpm for all vane opening, 100%, 50% and 0%. It is maximum 80.1 dB(A) at 100% vane opening for speed 700 rpm at 8 kg load. Maximum attenuation in SPL is 3.1 dB(A) using enclosure at speed 700 rpm at 6 kg load for 0% vane opening.

8.25.3.1 Effect of pressure head on SPL

It is observed that maximum SPL in dB(A) for A, B, C, D, E points is at PR head 12 feet head of water.

Among all the A, B, C, D, E points, point B has maximum SPL in dB(A) because point B is near to outlet of water used to run the Francis turbine.

Maximum attenuation of SPL is 4.1 dB(A) among all the A, B, C, D, E points using enclosure is at point A because this point is near to the source of passive noise.

8.25.3.2 Effect of speed on SPL

It is observed that maximum SPL in dB(A) for A, B, C, D, E points is at speed 800 rpm.

Among all the A, B, C, D, E points, point B has maximum SPL in dB(A) because point B is near to outlet of water used to run the Francis turbine.

Maximum attenuation of SPL is 4.1 dB(A) among all the A, B, C, D, E points using enclosure is at point B.

8.25.3.3 Effect of load on SPL

It is observed that maximum SPL in dB(A) for A, B, C, D, E points is at higher load.

Among all the A, B, C, D, E points, the value of SPL increases with increase in the load and point B has maximum SPL 84.4 dB(A) at 8 kg load.

Maximum attenuation of SPL is 4.1 dB(A) among all the A, B, C, D, E points using enclosure is at point A because this point is near to the source of passive noise.

8.25.4FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR FRANCIS TURBINE FOR LOAD 0, 4, 8 kg FOR CONSTANT PRESSURE HEAD AND VANE OPENING

8.25.4.1 Effect of vane opening

Maximum SPL in dB(A) is 87.5 at 63 Hz frequency for 100% vane opening. It is observed that with decrease in vane opening the value for SPL decreases.

At 100% vane opening the enclosure is effective for lower frequencies 31.5Hz-63Hz but with decrease in vane opening the effectiveness changes towards higher frequencies in range of 250Hz-3000Hz

8.25.4.2 Effect of pressure head

Maximum SPL in dB(A) is 87.5 at 63 Hz frequency for PR head 16 feet head of water. It is observed that with increase in PR head the value for SPL peak increases.

The effectiveness of enclosure increases with increase in PR head for lower frequencies 63Hz-125Hz and it decreases for frequencies in range of 125 Hz-4000Hz but effectiveness remains same at higher frequencies above 4000Hz.

8.25.4.3 Effect of load

The value of SPL shows some variation between the frequency range of 31.5 Hz to 500 Hz and 1000Hz-16000Hz with increase in load but it shows almost same values in frequency range of 250Hz-500 Hz.

CHAPTER 9

CONCLUSION AND SCOPE FOR FUTURE

9.1 CONCLUSION: following points are concluded from this thesis work

- Acoustic enclosure fabricated with consideration as functional design can reduce the SPL of pumping system by 10.5 dB (A).
- For noise reduction, Partial barriers are most economical and effective.
- Peak values of sound pressure level are on lower frequencies in range of 31.5Hz-63Hz.
- Complete enclosure is more effective for lower frequencies in range of 31.5 Hz-63 Hz but also effective at higher frequencies in range of 500 Hz-16000 Hz.
- With increase in load the SPL increases
- For 0% vane opening the value of SPL is less and for 100% and 50 % vane openings values of SPL is more.
- The value of SPL increases with increase in Speed but with increase in pressure head the value of SPL doesn’t show that much rise as in case of speed.
- Among the 5 points A, B, C, D and E surrounding the turine it is found that the maximum sound pressure level is at point B because B is nearest to the outlet of the water and minimum SPL is for point C because C is farthest from noise creating source.

9.2 SCOPE FOR FUTURE WORK

Lot of future scope is there in this experiment. Following work can be further carried out in future.

1. There are other sound absorbing materials such as glass wool and lead sheets. One can try those and observe the reduction in noise.
2. There are other sound absorbing methods like use of sound barriers between the source that produces active noise and passive noise and observe the difference in reduction.

3. Turbine is also source of noise in this case. One can design an enclosure for turbine as well and compare the difference in noise.

4. From the results it is clear that maximum sound pressure level is on 31.5, 63,126 Hz frequencies, so there is need to find out the particular parts of the pumping system and turbine which are producing these frequencies.

5. One can try the results on different type of turbines like Pelton and Kaplan and verify the results.

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27 **www.google.co.in**

28 **www.osun.org**

29 **www.wikipedia.com**

OBSERVATIONS IN SPL (dB(A)) FOR CONSTANT PRESSURE HEAD AT DIFFERENT LOADS AND VANE OPENINGS WITH AND WITHOUT ENCLOSURE

TABLE NO 9.1 PRESSURE HEAD IN FEET HEAD OF WATER WITH OUT ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD= 12					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	75	75.2	75.6	75.7	76.1
B	77.7	78.8	78.7	80	80.5
C	74.5	74.9	75.4	75.7	76.4
D	76	77	77.3	79.1	79.2
E	75.6	76.2	77	79.2	78.5
VANE OPENING=75%					
A	75.4	75.7	76	76.4	76.7
B	77.6	78.6	79.8	80.7	84.4
C	74.5	75.6	76	76.1	78
D	76.1	77.4	77.7	78	81.1
E	75.5	76.6	77.2	77.3	80.7
VANE OPENING=50%					
A	75.1	75.8	75.7	76.9	77.2
B	77.2	79.8	80.1	81.7	83.6
C	74.6	74.6	75.4	76.9	77.1
D	76.4	77	77	79.7	80.9
E	75.1	76.6	77.2	78.7	80.1
VANE OPENING=25%					
A	74.8	75.3	75.5	75.9	75.4
B	77.5	78.4	79	82.1	85.4
C	73.6	74.1	75.2	77	79.1
D	75.5	76	77	79.7	84.5
E	76.5	75.9	76	79.6	83.9
VANE OPENING=0%					
A	75.1	74.7	75.9	76.4	78
B	77.3	77.7	79.4	81.9	82.9
C	74.4	74.8	74.9	77.2	80
D	76.2	75.9	77.9	80.2	82.4
E	75.7	75.3	77.2	78.2	83.9

TABLE NO 9.2

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD = 14					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	75.5	75.9	75.8	75.6	76
B	78.2	78.7	79.5	80.5	81.2
C	74.6	75.3	75.9	75.8	76.2
D	76.4	77.2	77.7	77.6	78.1
E	76.1	76.7	77.3	77.3	77.9
VANE OPENING=75%					
A	75.1	75.7	75.5	76.3	76.7
B	78.2	78.8	79.5	80.8	82.8
C	75.1	74.8	75.9	76.6	77
D	76.1	77	79.5	79.1	80.4
E	75.7	76.5	77.9	79.5	78.7
VANE OPENING=50%					
A	75.3	75.6	75.8	76.3	76.8
B	77.8	78.8	79.6	80.9	83.2
C	74.3	75.1	75.8	76.4	77.6
D	75.8	77.5	78.2	79.2	80.9
E	75.5	76.9	76.8	77.9	80.4
VANE OPENING=25%					
A	75	75.5	75.8	76.1	77.4
B	77.4	78	80.7	80.7	82.8
C	74.1	74.6	76.4	76.1	77.1
D	75.3	76.9	77.9	78.4	79.7
E	74.8	75.8	77.4	77.7	78.9
VANE OPENING=0%					
A	74.3	75.1	75.4	75.4	76.1
B	76.7	78	79.8	81.4	83.3
C	74.1	74.8	75.2	76	76.9
D	75.6	76.4	76.9	79.8	79.6
E	75.4	75.5	76.9	78.8	79.3

TABLE NO 9.3

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD=16					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	75.8	75.8	75.5	76.3	76.8
B	78.7	78.6	80	80.6	83.2
C	74.9	75	75.5	75.2	76.8
D	76.9	77.1	78.8	80.8	80.5
E	76.5	76.1	77.8	79.9	79.5
VANE OPENING=75%					
A	74.5	74.7	75.4	75.8	76
B	78.4	78.6	79.5	81	81.6
C	73.9	74.5	75.2	75.7	76.6
D	76.3	77	78	78.3	79.5
E	75.5	76.2	78.2	77.8	78.3
VANE OPENING=50%					
A	75.3	74.8	76	75	76.7
B	77.6	77.8	79.8	80	83.2
C	74.2	74.9	74.5	75.4	76.7
D	76.2	76.9	78.1	78.4	80.1
E	75.3	76.2	76.6	76.7	78.9
VANE OPENING=25%					
A	75	75.7	74.9	75.4	76.4
B	77.3	78.4	79.8	79.6	79.3
C	74.4	74.5	75	74.7	75
D	76.4	76.8	77.6	77.8	77.9
E	75.6	76.1	76.6	76.3	77.5
VANE OPENING=0%					
A	74.4	74.6	75	75.8	76.7
B	77.2	77.3	78.5	78.7	82.6
C	74	74.1	74.2	75.4	76.7
D	75.6	76.3	77	76.4	79.4
E	75.2	75.6	77.1	77	78.7

TABLE NO 9.4

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD=18					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	76.3	75.1	76.3	76.6	76.2
B	78.8	78.8	79.8	79.4	82.3
C	75.3	75.5	75.5	75.9	76.6
D	76.9	77.7	78.2	80.7	79.3
E	75.9	76	77.2	78.4	78.7
VANE OPENING=75%					
A	75.8	75.2	76.3	76	77.2
B	78.5	78.8	79.7	80.7	83.1
C	75.7	75.2	75.7	76.2	77.1
D	76.5	77.3	77.6	78.7	82.2
E	75.5	76.7	77.9	79.6	80.2
VANE OPENING=50%					
A	75.4	75.8	75.9	76.8	77
B	78	78.7	79.7	82.4	82.8
C	75.3	75.8	75.6	76.8	78.8
D	76.2	77.2	78	81	80.6
E	75.9	76.9	76.9	78.8	79
VANE OPENING=25%					
A	75.4	75.4	75.9	76.8	77
B	77.9	78.1	79.7	82.4	82.8
C	74.7	74.9	75.6	76.8	78.8
D	76.3	76.6	78	81	80.6
E	75.8	76.3	76.9	78.8	79
VANE OPENING=0%					
A	75.4	75.4	75.9	75.1	76
B	77.9	78.1	79.6	79.9	81.4
C	74.7	74.9	75.9	75.9	75.9
D	76.3	76.6	77.8	77.6	78.9
E	75.8	76.3	77	77	78.8

OBSERVATIONS IN SPL (dB(A)) FOR CONSTANT PRESSURE HEAD AT DIFFERENT LOADS AND VANE OPENINGS WITH ENCLOSURE

TABLE NO 9.5 PRESSURE HEAD DATA WITH ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD=12					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	73	73	73.6	73.9	75.2
B	76.9	77.2	78.1	79.1	79.5
C	73.3	73.9	74.8	75.1	75.6
D	74.9	75.9	76.8	77.9	78.5
E	74	74.5	75.3	76	77.8
VANE OPENING=75%					
A	72	72.6	73.4	72.5	74.8
B	76.4	77.2	78.5	80.2	83.8
C	72.7	73.7	73.9	75	77.1
D	73.9	72.9	76.1	76.8	77.9
E	73.7	72.7	75.5	75.9	78.2
VANE OPENING=50%					
A	72.1	72.5	73.2	74.4	75.3
B	75.6	76.5	78.2	81.1	82.3
C	72.7	72.9	74.8	75.9	76.7
D	73.4	74.5	76.2	78.2	81.3
E	73.3	73.7	76.2	77.3	78.9
VANE OPENING=25%					
A	72.4	72.7	73.1	74.9	74.5
B	75.7	76.4	78.5	81	84.5
C	73.4	73.3	75	75.5	77.8
D	73.8	74	76.3	78	83
E	73.5	73.8	74.1	77.6	80.8
VANE OPENING=0%					
A	72.7	71.9	75	75.7	76.3
B	75.2	75.8	78.5	81	81.1
C	72.4	73	74	75.5	78.3
D	73.6	74.7	78.3	80.8	83.8
E	74.3	73.5	76.7	77.1	81.4

TABLE NO 9.6

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD=14					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	73.3	73.4	74.5	74.4	75
B	75.4	77.2	77.1	77.9	79
C	73.6	74.1	75.1	75.5	76
D	74.3	75.6	76.4	76.5	77.7
E	73.9	75.1	76	75.9	77
VANE OPENING=75%					
A	72.9	73.4	74	75.2	75.1
B	74.6	77.6	78	80	81
C	73.4	74.1	75.2	75.3	76.5
D	73.8	75	79	78.2	79.2
E	73.7	74.8	77.3	78.1	77.1
VANE OPENING=50%					
A	72.4	73	74.9	75.2	76.2
B	74.7	76.7	78.9	77.7	80.6
C	73.1	74.4	75.4	75.3	75.1
D	73.9	76.4	76.5	78.2	78.3
E	73.8	75.1	76	77.1	79.9
VANE OPENING=25%					
A	72	72.8	74	74.3	76.1
B	74.9	76.8	79.8	79	80.1
C	73	74	75.2	75.3	77
D	73.5	75.5	77.6	77.9	78.2
E	74.1	75	77.1	77.1	78.2
VANE OPENING=0%					
A	72.7	71.9	74	74.8	74.2
B	75.2	75.8	78.5	81	81.1
C	72.4	73	74	75.5	75.3
D	73.6	74.7	75	79.2	77.9
E	74.3	73.5	76.3	77.1	78.1

TABLE NO 9.7

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD=16					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	72.9	74.3	75	75.9	75.2
B	75.9	77.5	78	79.4	82.1
C	73.6	75	75.3	75.1	76.4
D	74.5	76.8	78.6	80.2	80.1
E	74.6	75.2	77.6	79.6	79
VANE OPENING=75%					
A	72.8	73.3	74.8	75.1	74.9
B	76	77.3	78.1	80.4	81.2
C	73.6	74.2	74.7	75.2	76.4
D	74.5	76.6	77.6	77.5	79.2
E	74.4	75.6	77.5	77.3	77.8
VANE OPENING=50%					
A	72.7	73.2	74.5	74.2	75.2
B	76.5	76.2	79.2	79.2	81.2
C	74	74.5	74.1	75.1	75.2
D	74.9	76.7	77.2	77.4	78.6
E	74.8	76	76.3	76.4	78.3
VANE OPENING=25%					
A	72.9	74.1	74.5	74.6	75.3
B	76.1	77.3	78.4	78.3	78.7
C	74.1	74.3	74.8	74.5	74.9
D	75.2	76.4	76.5	76.8	77.6
E	74.8	75.8	76.1	76	77.1
VANE OPENING=0%					
A	72.9	73.7	74	74.8	74.6
B	76	76.6	76.4	77.9	81.7
C	73.3	73.5	73.9	74.7	76.5
D	75.2	76	76.5	75.6	78.4
E	74.8	75.2	76.8	76.5	78.4

TABLE NO 9.8

OBSERVATIONS IN SPL (dB(A)) FOR PR HEAD=18					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	73.6	74.5	74.9	75.9	75.4
B	76.4	78	79.3	78	81.2
C	74.5	75	75.1	75.3	76.2
D	74.7	76.8	77.7	79.5	78.1
E	74.9	75.4	76.4	78.2	78.2
VANE OPENING=75%					
A	72.5	73.4	75.4	74.9	76.4
B	74.3	76.4	78.2	80.7	82.1
C	73.2	74.6	75.7	76.2	77.1
D	72.3	75.4	78	78.7	79.8
E	73.6	75.1	77.9	77.9	78.2
VANE OPENING=50%					
A	72.9	74.1	75	75.7	75.9
B	76.6	77.1	78.3	81.7	81.9
C	74.3	74.9	75.1	76.3	76.6
D	74.8	76.4	77.6	79	79.4
E	74.9	76.4	76.2	78.1	78.4
VANE OPENING=25%					
A	72.9	73.8	74.8	76.2	76.2
B	75.9	76.3	78.1	81.5	82.3
C	74.1	74.8	75.2	76	77.5
D	74.9	75.1	77.5	80.7	79.8
E	74.7	75.8	76.3	78.3	78.3
VANE OPENING=0%					
A	72.9	73.9	74.7	74.3	75.6
B	76.2	77.6	78.2	78.4	80.2
C	74.2	74.5	75.5	75.6	75.6
D	74.9	75.2	77	77.2	78
E	74.8	75	76.5	76.4	78.2

OBSERVATIONS IN SPL (dB(A)) FOR CONSTANT SPEED AT DIFFERENT LOADS AND VANE OPENINGS WITHOUT ENCLOSURE

TABLE NO 9.9 SPEED DATA WITHOUT ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=700					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	74.4	75.1	74.7	75.7	76.2
B	77	78.1	79	79.9	82.1
C	74.8	75	75.2	76.7	77
D	76.1	77	78.4	79.1	79.2
E	76	76.3	76.8	79	80.1
VANE OPENING=75%					
A	74.9	75	74.7	75.7	76.3
B	76.9	77.2	78.7	80	81.9
C	74.9	74.2	75	77	78.1
D	76.1	76.5	78.1	78.9	80.7
E	76	76.3	76.8	78	81
VANE OPENING=50%					
A	74.2	74.5	74.5	75.7	76
B	76.9	77.2	78.7	79.5	81.5
C	74.9	74.2	75	76.5	76.8
D	76	75.5	78	78.3	79
E	74.5	75.5	76.7	78.3	80
VANE OPENING=25%					
A	75.1	74.8	75.5	75.7	76.8
B	76.2	77.2	78.7	79.1	81.1
C	74.8	74.4	75.7	77	78.2
D	75.5	75.5	78.7	79.1	79.5
E	75.1	75.5	76.7	78	79.6
VANE OPENING=0%					
A	75.6	75.4	75.1	76.3	77
B	76.8	77.3	79.1	80	80.9
C	74.2	75.1	75	76.1	76.5
D	75.4	76.6	77.5	78.1	78.5
E	74.9	76.9	76.6	78.8	79.7

TABLE NO 9.10 SPEED DATA WITHOUT ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=800					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	75.2	75.4	76.1	77	76
B	77.1	78	79.9	82.2	82.3
C	74.6	75	75.8	76.1	77.5
D	75.8	76.7	77.5	79.5	81
E	75.1	76.2	77.1	78.5	79.5
VANE OPENING=75%					
A	75.2	75.6	75.5	76.9	78.1
B	76.7	78.9	79.8	83.6	85
C	74.5	74.9	75.9	77	80
D	76.2	76.5	77.4	80	82
E	75.4	76.1	77.3	78.5	80.7
VANE OPENING=50%					
A	74.7	75.3	76.2	76.9	77.8
B	76.8	78.2	80	81.3	84.4
C	74.8	74.3	76.3	76.5	77.1
D	75.5	76.7	77.5	78.5	80.8
E	75	76.2	76.5	78.1	79.4
VANE OPENING=25%					
A	75.2	74.9	75.8	76.3	77.6
B	76.5	77.2	79.5	81.5	84
C	73.5	74.1	76	76.9	79
D	75.2	76.5	77.8	79	80.5
E	75.3	75.5	77.4	77.5	79.5
VANE OPENING=0%					
A	75.8	74.9	76	77.4	78.3
B	77	77.3	79.1	81.4	82.1
C	74.7	74.5	76.1	76.5	77
D	75.4	76.1	77.5	79.2	80.1
E	74.9	75.5	76.7	78	78.9

TABLE NO 9.11 SPEED DATA WITHOUT ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=900					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	74.9	75.2	75.1	76.5	75.7
B	77.2	78.8	78.9	80.3	81.1
C	73.4	74.9	74.6	76	76.5
D	75.6	77.6	78.1	77.8	78.6
E	74.8	76	76.8	78.6	79.4
VANE OPENING=75%					
A	75.1	74.8	76	76.6	76.1
B	76.8	77.7	79.6	79.7	82.2
C	73.7	74.5	74.7	74.9	76.5
D	74.9	76.1	77.5	77.1	79.6
E	74.5	75	76.9	77.5	78.6
VANE OPENING=50%					
A	74.7	75.8	75.2	74.9	76.3
B	77.3	78.3	79.4	78.8	81.4
C	74.2	73.9	74.7	74.4	76.3
D	75.6	75.9	76.8	77.8	77.9
E	74.7	75.7	76.6	77.4	78
VANE OPENING=25%					
A	75.5	75.1	75.9	75.8	75.7
B	77.1	77.8	79.2	80.2	81.6
C	73.3	75	75.6	75.2	76.2
D	74.9	77.5	77.9	80.6	80.1
E	75	75.9	77.2	78.1	79.5
VANE OPENING=0%					
A	74	74.5	75.8	76.2	76.3
B	76.8	77.7	79.7	80.6	82.9
C	73.9	73.8	74.1	76	76.6
D	75.1	76.5	77.1	78.1	78.5
E	74.6	75.3	76.9	76.8	77.8

TABLE NO 9.12 SPEED DATA WITHOUT ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=1000					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	75.5	75.5	75.6	76.1	77.9
B	78.1	78.9	80.4	81.5	82.8
C	74.5	75.2	75.5	76.9	78
D	76.1	77	79	79.9	80.4
E	76.5	75.9	77.1	78.9	79.6
VANE OPENING=75%					
A	75.4	75.7	75.8	77.3	76.8
B	78.1	78.3	80.7	81.5	84.4
C	74.6	74	75.6	76.6	78.6
D	75.8	76.8	79.2	79.4	80.9
E	75.5	77	77.3	78.7	80
VANE OPENING=50%					
A	75.1	75.5	76.8	76.1	77.2
B	77.7	78.7	80.3	81.7	84.1
C	74.9	75.2	76.4	76.8	78.1
D	75.9	78	78.3	79.3	80.5
E	75.4	76.8	77.2	78.8	79.9
VANE OPENING=25%					
A	74.9	75.1	76	76.8	76.9
B	77.5	78.8	79.8	81.7	83.3
C	76.5	75.5	76.1	76.4	77.3
D	76.6	77.4	78.1	79.4	79.9
E	76.7	76.8	77.9	78.9	79.6
VANE OPENING=0%					
A	74.9	75.8	75.8	76.6	77.4
B	77.7	78.7	79.8	82.3	83.4
C	74.9	75.6	75.7	77.3	77.9
D	76.4	77.7	78.3	79.9	79.7
E	75.7	76.9	78.4	78.3	79.8

OBSERVATIONS IN SPL (dB(A)) FOR CONSTANT SPEED AT DIFFERENT LOADS AND VANE OPENINGS WITH ENCLOSURE

TABLE NO 9.13 SPEED DATA WITH ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=700					
VANE OPENING=100%					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	72.6	73.2	73.2	73.6	73.9
B	75.9	77.1	77.5	78	81
C	73.4	73.2	73.6	74.5	75.5
D	74.5	74.7	76.1	77.5	78
E	74.3	75.5	75.2	76.9	78.5
VANE OPENING=75%					
A	72.6	72.8	73.5	73.4	74
B	75.3	75.8	76.3	77.2	78.6
C	72.8	73.1	73.3	74.1	76.1
D	74.3	75.4	75.3	76.2	78.5
E	74.6	75.7	75.2	76.4	78
VANE OPENING=50%					
A	72.1	73.1	73.2	73.3	74
B	75.1	75.2	76.1	76.9	80.1
C	72.3	72.9	73.5	74.1	75.1
D	73.8	74.6	75.4	76.8	77.3
E	73.8	74.7	75.6	76.1	77.6
VANE OPENING=25%					
A	72.5	72.9	73.1	73.3	74.3
B	75	75.8	76.6	77	79
C	72	73.2	73.4	73.7	74.6
D	74.3	74.6	75.2	76.1	77.9
E	73.9	74.7	75.1	75.5	76.7
VANE OPENING=0%					
A	72.5	73.7	73	73.8	73.8
B	75.2	75.1	76.1	77.4	78.1
C	72.8	72.6	73.5	74	75.2
D	73.7	74.2	74.7	76.7	77.4
E	73.4	74.8	74.3	75.7	77.3

TABLE NO 9.14 SPEED DATA WITH ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=800					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	73.2	73.1	73.6	73.4	74
B	76	77	77	77.8	78.3
C	73.5	73.8	74	75.1	75.5
D	75	76.5	76.6	78	79.5
E	74.3	75.5	75.2	76.9	78.5
VANE OPENING=75%					
A	72.8	73.1	73.4	73.8	74.7
B	76	77	77.3	77.9	80.5
C	73.4	73.3	73.7	74.5	76
D	74.9	75.6	76	76.5	78.5
E	74.4	74.3	75.2	76.2	76.9
VANE OPENING=50%					
A	73	72.9	73.8	74.3	75
B	76.1	77	78.1	78.4	80
C	73.3	73.5	74	75	75.8
D	75	75.8	76.3	77	79
E	73.8	74.7	75.6	76.1	77.6
VANE OPENING=25%					
A	72.4	72.6	73.2	73.4	73.8
B	75	76.7	77.1	78	80
C	73	73.3	73.6	74.7	77
D	74.6	75.5	75.9	76.9	78
E	74	74.9	75.8	76.1	77
VANE OPENING=0%					
A	72.7	72.7	73.3	73.9	75
B	76.3	76.5	76.8	79	80
C	72.9	73.3	73.6	74.5	75.5
D	74.7	75	75.5	76.6	78.1
E	73.4	74.8	74.3	75.7	77.3

TABLE NO 9.15 SPEED DATA WITH ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=900					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	73.1	73.4	73.6	73.8	73.9
B	76.8	77.9	77.9	78.5	78.8
C	73.2	73.8	73.5	74.3	75.5
D	75.2	76	76.9	77.4	78.1
E	74.4	75.6	75.7	75.9	76.5
VANE OPENING=75%					
A	73	73	74.3	74.1	74.9
B	76.6	77.4	78.4	79.4	79.8
C	73	73.4	74.3	74.8	74.8
D	74.3	75.8	77.5	77	79.2
E	74.6	75	75.7	76.4	76.7
VANE OPENING=50%					
A	73.2	73.6	74.2	74	74.2
B	76.9	77.7	78.2	78	79.3
C	72.9	73.6	73.8	74	75.1
D	75.5	75.8	76.5	77.6	78
E	74.2	75.4	75.6	75.8	76.4
VANE OPENING=25%					
A	73.1	73	73.6	74.3	75
B	75.6	77.1	77.1	78.7	79.5
C	72.9	73.4	73.5	74	74.7
D	74	75.6	75.1	77.5	78.2
E	74.7	74.5	74.7	75.8	77.4
VANE OPENING=0%					
A	73.5	73.5	75	76	76.2
B	75.4	76.1	78.8	79.5	82.7
C	73.1	73.1	74	75.5	76.4
D	73.6	74.8	76.8	77.5	78.1
E	73.4	74.2	75.7	76.2	77.5

TABLE NO 9.16 SPEED DATA WITH ENCLOSURE

OBSERVATIONS IN SPL (dB(A)) FOR speed=1000					
VANE OPENING=100%					
LOADS (Kg)	0	2	4	6	8
POINTS					
A	74.2	74.5	75	75.5	76
B	77.1	77.5	78.9	80.9	81.7
C	73.8	74.8	75.5	75.7	77.1
D	75.5	76.3	77.4	78.8	78.6
E	74.3	75.4	75.9	77.6	78.8
VANE OPENING=75%					
A	73.8	74.4	74.5	74.9	76.2
B	76	76.1	78	79.9	81.3
C	73.4	73.6	74.1	75.2	76.5
D	74.6	75	76.2	77.4	78.6
E	74.1	74.4	76	76.5	78.4
VANE OPENING=50%					
A	73.9	74.4	74.8	75	76
B	76	77	78.4	79.7	82.5
C	74.2	74	74.9	75.2	76.5
D	75.2	76	76.5	76.9	79
E	74.3	74.7	75.2	76.6	78.2
VANE OPENING=25%					
A	73.8	74.2	74.9	75.7	76.7
B	76.5	76.9	78.2	80.2	82
C	73.6	73.9	74.9	75.9	77.1
D	74.1	74.9	76.5	78.9	80
E	73.7	74.1	75.2	77.4	78.5
VANE OPENING=0%					
A	73.5	74.1	74.5	75.6	76.5
B	75.9	76.2	77.5	80.5	82.5
C	73.8	74.1	74	76.3	76.8
D	74.6	75.3	76.2	78.3	79.5
E	74	74.6	76.1	77.4	79.2

OBSERVATIONS IN SPL (dB(A)) FOR NOISE GENERATED BY PUMPING SYSTEM WITH AND WITHOUT ENCLOSURE

TABLE NO 9.17

NOISE OF PUMPING SYSTEM		
TURBINE CONDITION=OFF		
WITHOUT ENCLOSURE		
A1	75.5	76.1
B1	72.6	74
C1	75.4	76.5
D1	76.7	75.8
E1	74.4	75.2
WITHENCLOSURE		
A1	65.7	66.4
B1	65.3	67.8
C1	66.5	66.7
D1	70.5	69.1
E1	65.4	64.7

1. OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT DIFFERENT SPEEDS, VANE OPENINGS AND LOADS FOR PUMPING SYSTEM

TABLE NO. 9.18 FOR PRESSURE HEAD 12 WITHOUT ENCLOSURE

VANE OPENING = 100%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 12										
FREQ (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	66	87.8	80.1	81.4	76.4	76.5	73	65.5	59.5	60
4	65.5	85.8	77	80.3	75.9	74.8	71.5	63.5	55.9	55.9
8	65.4	83.8	76.7	79.5	76.5	74.2	71.5	63.6	54.9	50.8

TABLE NO. 9.19 FOR PRESSURE HEAD 12 WITHOUT ENCLOSURE

VANE OPENING = 50%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 12										
FREQ (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	64.4	84	78.4	80.8	75.4	75.6	72.3	64.2	58	58
4	64	85.1	76.8	80.5	76.7	74.9	71.3	63.4	57.7	56.6
8	65.7	85.2	76.6	79.8	76.2	75	71.3	64.3	57.6	57.6

TABLE NO. 9.20 FOR PRESSURE HEAD 12 WITHOUT ENCLOSURE

VANE OPENING = 0%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 12										
FREQ (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	63.1	83.7	78	81.3	75.6	75.4	71.6	65.1	59	61
4	64.2	85.6	76.7	79.8	73.9	73.5	70.9	64.5	58.5	59.9
8	65.2	86	76.2	78.3	73.6	73.1	70.7	64.7	57.9	58.2

TABLE NO. 9.21 FOR PRESSURE HEAD 12 WITHOUT ENCLOSURE

VANE OPENING = 100%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 16										
FREQ (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	65.5	88.5	77.7	78	73.5	73.6	70.9	64.5	56.8	56.9
4	65.4	86.3	76.4	79.3	74.8	73.5	70.7	63.3	54.8	48.9
8	65.4	85.1	75.5	79.1	75.8	73.7	72	64.8	57	54.9

TABLE NO. 9.22 FOR PRESSURE HEAD 16 WITHOUT ENCLOSURE

VANE OPENING = 50%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 16										
FREQ (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	65.2	88.1	77.7	79.6	73.2	73.7	70.7	64.9	58.7	59.2
4	63.6	78.1	73.1	75.9	71.5	72.5	70.5	63.3	57.5	58.3
8	65.4	82.7	74.9	77	74.9	72.6	70.8	64.7	54.6	48.2

TABLE NO. 9.23 FOR PRESSURE HEAD 16 WITHOUT ENCLOSURE

VANE OPENING = 0%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 16										
FREQ (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	63.6	84.6	76.3	79.3	74.9	74.7	71.7	64.2	58.5	58.8
4	64.9	86.3	76.3	77.5	73.8	72.9	70.3	63.4	57.3	58
8	65.7	86.4	76.8	78.3	74.7	72.9	71.1	66	57.1	56.6

2. OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT DIFFERENT SPEEDS, VANE OPENINGS AND LOADS FOR FRANCIS TURBINE

TABLE NO. 9.24 FOR PRESSURE HEAD 12 WITHOUT ENCLOSURE

VANE OPENING = 100%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 12										
FREQ (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	65.4	83.4	76.2	76.3	73.9	73.7	70.4	67	63.7	61.5
4	66	83.5	76.4	77.5	74.5	72.8	74.4	67.8	63.6	61
8	66.6	81.1	77.6	79	74.5	73.2	72.6	67.4	61.7	58.2

TABLE NO. 9.25 FOR PRESSURE HEAD 16 WITHOUT ENCLOSURE

VANE OPENING = 50%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 12										
FREQ (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	65.4	84.4	77.6	76.6	74.4	73.2	70	67.3	63.5	60.4
4	66.5	84.9	76.1	76.9	74.6	73	73.7	67.2	63.4	60.1
8	72.3	84.4	75.9	77.7	74.4	72.5	72.9	68.2	61.4	58.4

TABLE NO. 9.26 FOR PRESSURE HEAD 12 WITHOUT ENCLOSURE

VANE OPENING = 0%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 12										
FREQ (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	64.2	83.5	77.3	74.7	74.4	73.1	70.2	67.1	63.9	60.6
4	66.7	85.9	77	76.4	73.2	72.2	71.1	65.7	62.1	59.6
8	65.1	84.9	77	76.7	73.8	71.9	72	70	60.4	57.2

TABLE NO. 9.27 FOR PRESSURE HEAD 16 WITHOUT ENCLOSURE

VANE OPENING = 100%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 16										
FREQ (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	66.1	87.5	78.3	76.8	72.7	72.7	70.1	67.9	65	61.8
4	66.7	85.8	78.2	76.4	73.5	73.4	72.2	68.1	64.9	61.9
8	66.4	82.1	77.6	76.3	74.3	73.2	77.5	72.7	64.1	60.8

TABLE NO. 9.28 FOR PRESSURE HEAD 16 WITHOUT ENCLOSURE

VANE OPENING = 50%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 16										
FREQ (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	65.9	85.6	77.5	76.5	73.5	72.4	69.7	66.9	63.9	61.1
4	65.6	80.7	76.5	75.8	73.8	72.8	72.6	67.6	63.7	60.8
8	67.5	81.9	77.9	76.3	74	72.7	78.9	73.7	64.1	60.5

TABLE NO. 9.29 FOR PRESSURE HEAD 16 WITHOUT ENCLOSURE

VANE OPENING = 0%

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT PRESSURE HEAD 16										
FREQ (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	65.8	83.5	76.2	74.9	72.6	73.3	69.8	67.4	64.4	62.1
4	65.8	84.7	77.2	77	73.6	73.1	73.3	67	64	61.9
8	65.4	84.6	77.8	75.1	74.1	72.9	75.5	75.6	64.9	60.2