A STUDY ON 4 STROKE DIESEL ENGINE NOISE USING DIFFERENT BLENDS OF PONGAMIA OIL

A dissertation

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Submitted By

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ABSTRACT

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”.

Internal combustion engines are designed to operate on gasoline or diesel fuels (or their substitutes) have different combustion systems that lead to significantly different noise characteristics. Gasoline engines use spark ignition to control combustion process, while diesel cycle engines use compression ignition. The forces generated inside internal combustion engines cause vibration of engine structure, leading to radiated noise.

With ever increasing consumption of fossil fuels and petroleum products, It is a matter of concern for our country for huge out-go of foreign exchange on the one hand and increasing emission causing environmental hazards on the other. Public at large are raising their concerns over the declining state of environment and health.

With domestic crude oil output stagnating, the momentum of growth experienced a quantum jump since 1990s when the economic reforms were introduced paving the way for a much higher rate of development leading the demand for oil to continue to rise at an ever increasing pace. The situation offers us a challenge as well as an opportunity to look for substitutes of fossil fuels for both economic and environmental benefits to the country. Pongamia oil is one such substitute that can be produced from seeds of karanj and used in blends with diesel for automobiles and used for transportation of vehicles, in generators, and irrigation pumps, etc.

As it is well known that with crude oil depleting every day due to its ever increasing demands future belongs to alternative fuels. Using biodiesel blends a lot of work has been done by various researchers regarding the study of engine characteristics like brake horse power, thermal efficiency, etc. but noise has been one area that requires immediate attention where still a lot of work has to be done.

In the present work of engine noise using pure diesel, B10 (10% biodiesel and 90% diesel), B20 (20% biodiesel and 80% diesel) and B30 (30% biodiesel and 70% diesel) blends of biofuels has been done on points A, B, C, D (at a distance of 1 meter surrounding the engine)and at the exhaust point. It has been figured out that using B20 or B10 fuels noise reduces to considerable levels near the A, B, C, D points surrounding the engine in comparison to noise generated by pure diesel. Along with that effect of blends on 1-1 octave band frequency analysis has also been studied in order to figure under which frequency range sound pressure level is high. Attempt has also been made to study the effect of blends on acoustic power. All these studies have been done by varying compression ratios and loads at the marked locations from the engine.

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NOMENCLATURE

SYMBOLS	DESCRIPTIONS
ASTM	American society for testing and materials
B10-B10%	10% biodiesel and 90% diesel
B20-B20%	20% biodiesel and 80% diesel
B30-B30%	30% biodiesel and 70% diesel
BHP / bhp	Brake Horse power
BIS	Bureau of Indian Standards
BOD	Bio-Chemical Oxygen Demand (A measure of Effluent pollution)
CI	Compression Ignition
CNG	Compressed Natural Gas
COD	Chemical Oxygen Demand (A measure of effluent Pollution)
C_{centre}	Centre Frequency
C.R.	Compression ratio
cSt	centistokes
dB	Decibel
dB(A)	Equivalent SPL for A- weighting
DF	Diesel Fuel
EPA	Environment Protection Agency
F_{upper}	Frequency of upper limit
F_{lower}	Frequency of lower limit
HSD	High Speed Diesel
Hz	Hertz
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
L_w	Acoustic power in dB(A) (ref. 10^{-12} W)
Mha or M ha	Million Hectare
N_{OX}	Nitrogen Oxides
NPL	Noise pollution level
Pa	Pascal
PM	Particulate matter
RSME	Rape Seed Methyl Ester or Biodiesel made from Rapeseed oil
S	Hypothetical surface area
S_o	Reference area
SPL	Sound pressure level
VCR	Variable compression ratio
σ	Standard deviation

CHAPTER 1

INTRODUTION

1.1 INTRODUCTION TO NOISE: In our modern, rapidly expanding environment one of the major problems which required immediate efforts is that of noise. [21]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 object of this part is to discuss the concept of noise, problems of noise and its effect on man and environment both as annoyance and as a danger to health.

The noise sources major parts are from:

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

Out of above three parameters, the source that affects the most is Traffic noise though nowadays industrial contribution to noise is increasing every day with rise in machinery and demands due to growing needs of population. In traffic noise, almost 70% of noise is contributing by vehicle noise. Vehicle noise, mainly, arises from two parameters i.e. Engine noise and Tire noise. The major concern is to study the engine noise and the sources from where the noise is generated by the engine by changing the different parameters.

SOME BASIC TERMS USED IN NOISE:

NOISE: Noise is conveniently and concisely defined as “unwanted sound”.

SOUND: Sound waves are pressure variations produced as a result of mechanical disturbance in a material medium.

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

FREQUENCY: Number of cycles per second is known as frequency.

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

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
$2*10^{-5}$	0 dB	Threshold of Hearing

Table 1.1 ENVIRONMENTAL CONDITIONS AT DIFFERENT SPL [25]

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

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

1.2.1 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.

1.2.2. 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.

1.2.3. 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.

1.3. PHYSICAL PROPERTIES OF SOUND

1.3.1 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.

1.3.2. 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

$$p^2 \propto I \propto P \qquad \text{……(1.1)}$$

1.3.3 SOUND PRESSURE LEVEL: Decibel (dB) is logarithmic ratio which defines the sound pressure level L_p as follows:

$$L_p = 20 \log_{10} P / P_0 \qquad \text{……(1.2)}$$

P is the sound pressure measured

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.

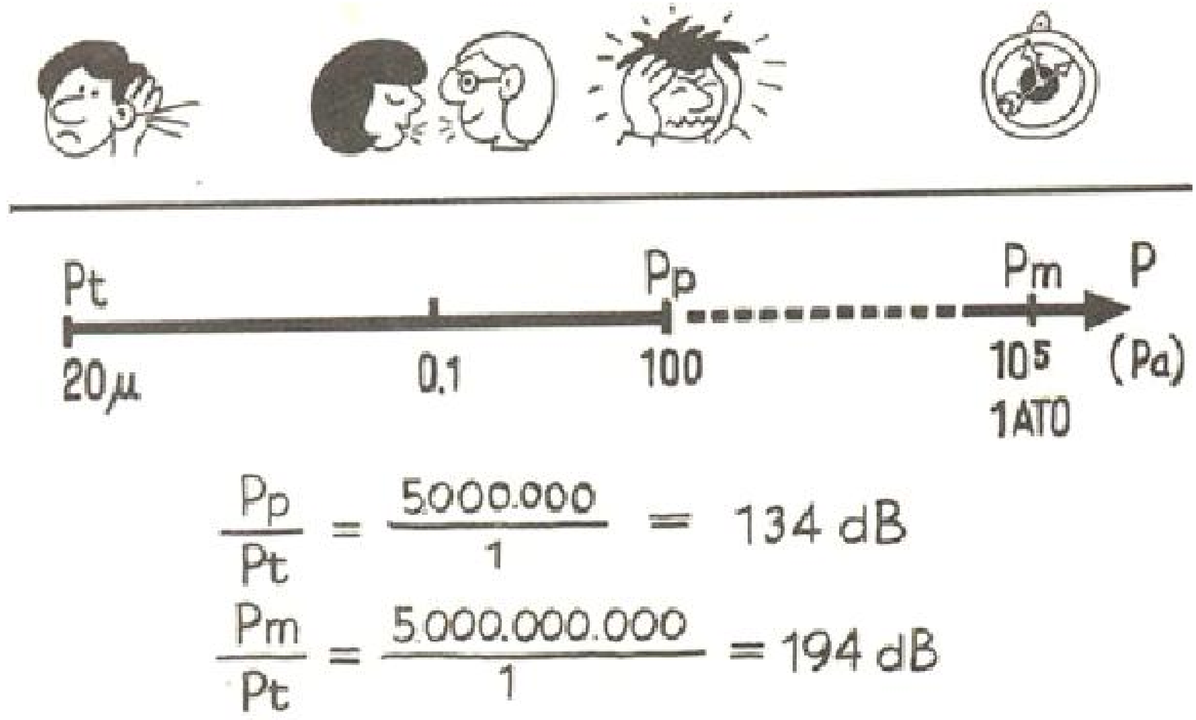

Fig. 1.1 Calculation of SPL [22]

1.4. CHARACTERISTICS OF SOUND

1.4.1. 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.

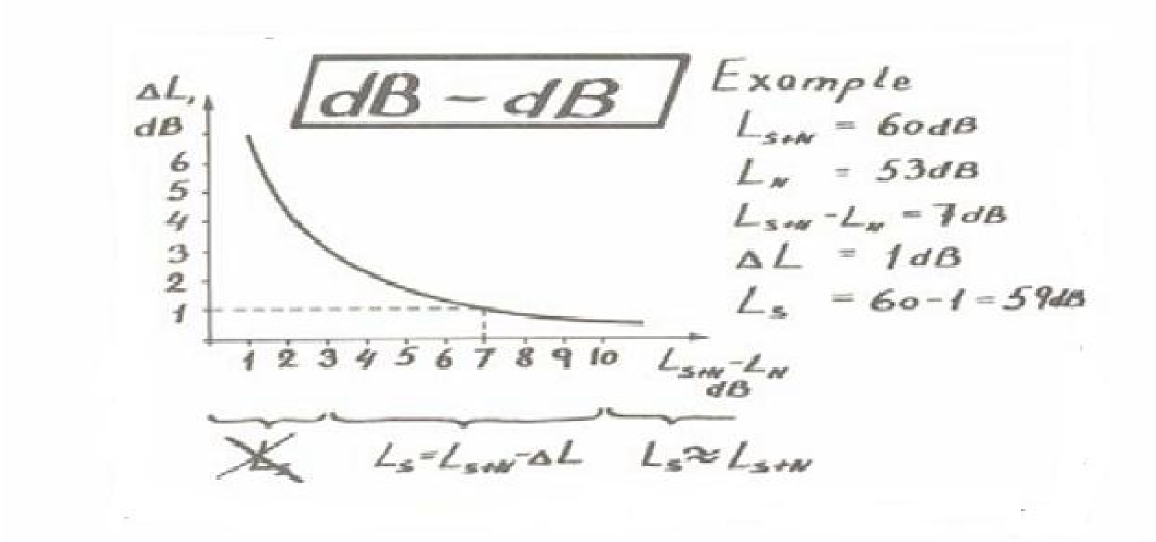

Fig.1.2 Subtraction of background noise in dB [22]

This is the curve for reducing the background noise. In this example, 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 59 dB.

1.4.2. 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.

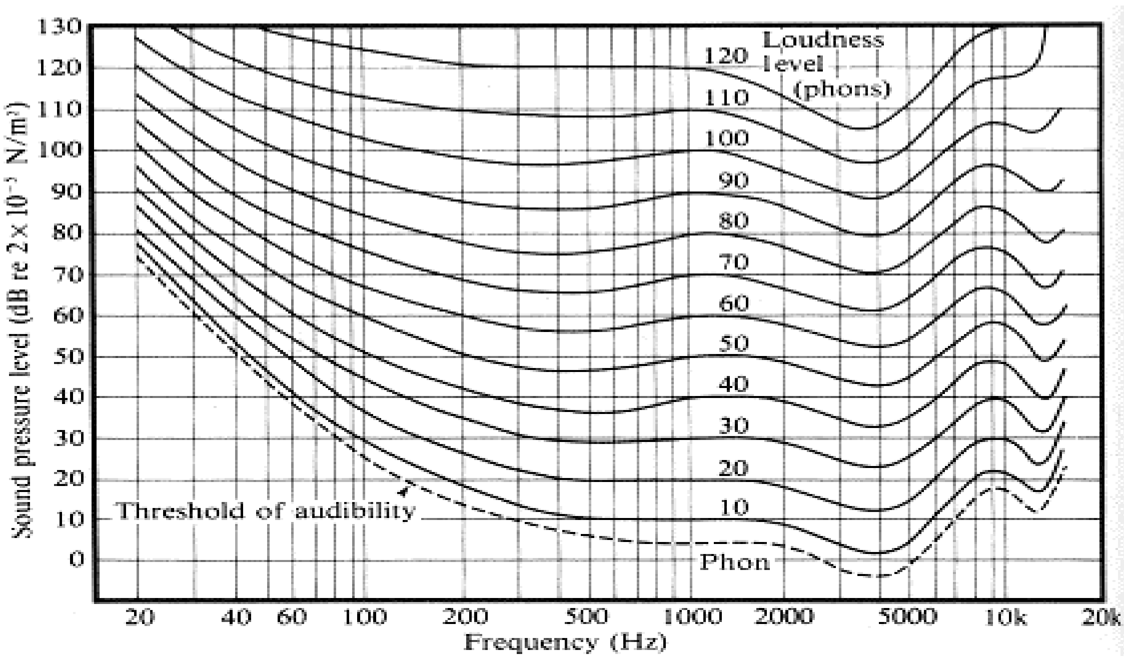

Fig.1.3 EQUAL LOUDNESS CONTOURS [26]

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.

SUBJECTIVE EFFECT OF CHANGES IN NOISE LEVEL	
Change in levels in dB	**Subjective Effects**
3	**Just Perceptible**
5	**Clearly Perceptible**
10	**Twice as Loud**

Table 1.2

1.4.3. 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. 4. 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.

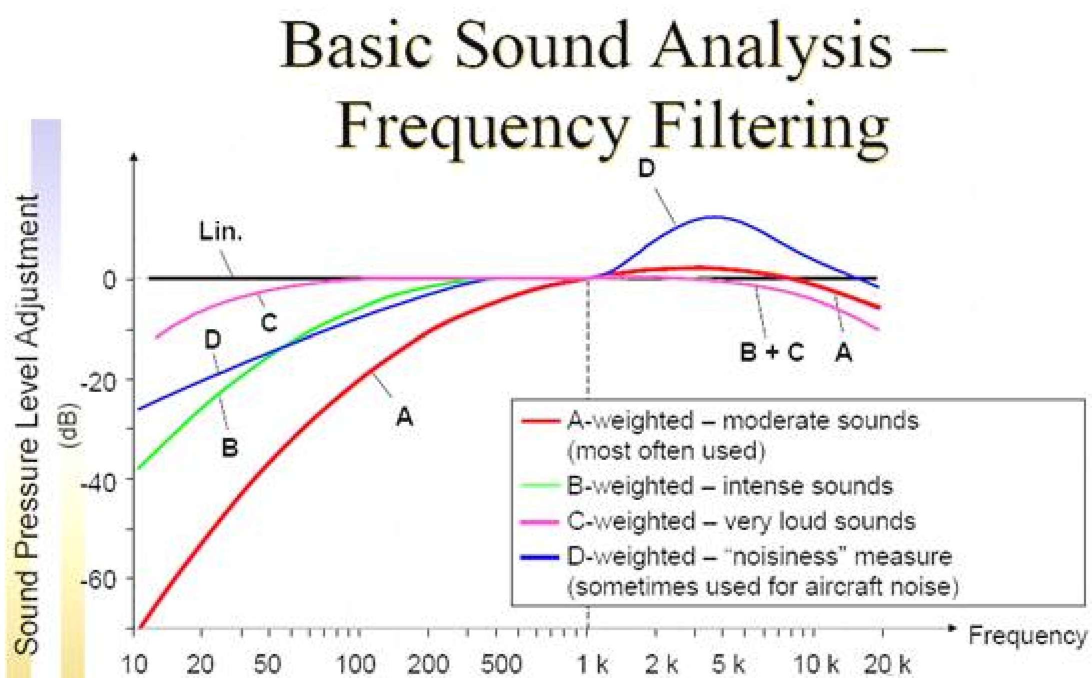

Fig.1.4 weighting curves [25]

1.4.3.1 PERCENTILE EXCEEDED SOUND LEVELS

Percentile Exceeded Sound Levels:

L10 = 10 percentile exceeded Sound level (av. Peak level)

L90 = 90 percentile exceeded Sound level (av. Background level)

L50 = Median value of Sound level

L10 – L90 = Noise climate

Equivalent continuous Sound level Leq

1.4.4 FREQUENCY SPECTRUM: If a sound has components at one frequency only, it is said to be a pure tone. Such sounds are not very common in nature, however, and the only common example of a pure tone is the sound of a tuning fork. Most usually, sounds have components at several frequencies and the character or timbre of a steady sound is determined by the pressure amplitudes at the different component frequencies. We can therefore describe a steady sound by a graph of frequency against amplitude, and such a graph is referred to as the frequency spectrum of the sound. Sound measuring instruments are usually constructed to measure the frequency spectrum, but for measurement convenience and simplicity of the instrument in practice we measure the energy content in a particular range of frequencies. Some examples of the frequency spectra of particular sounds are shown in fig 1.5.

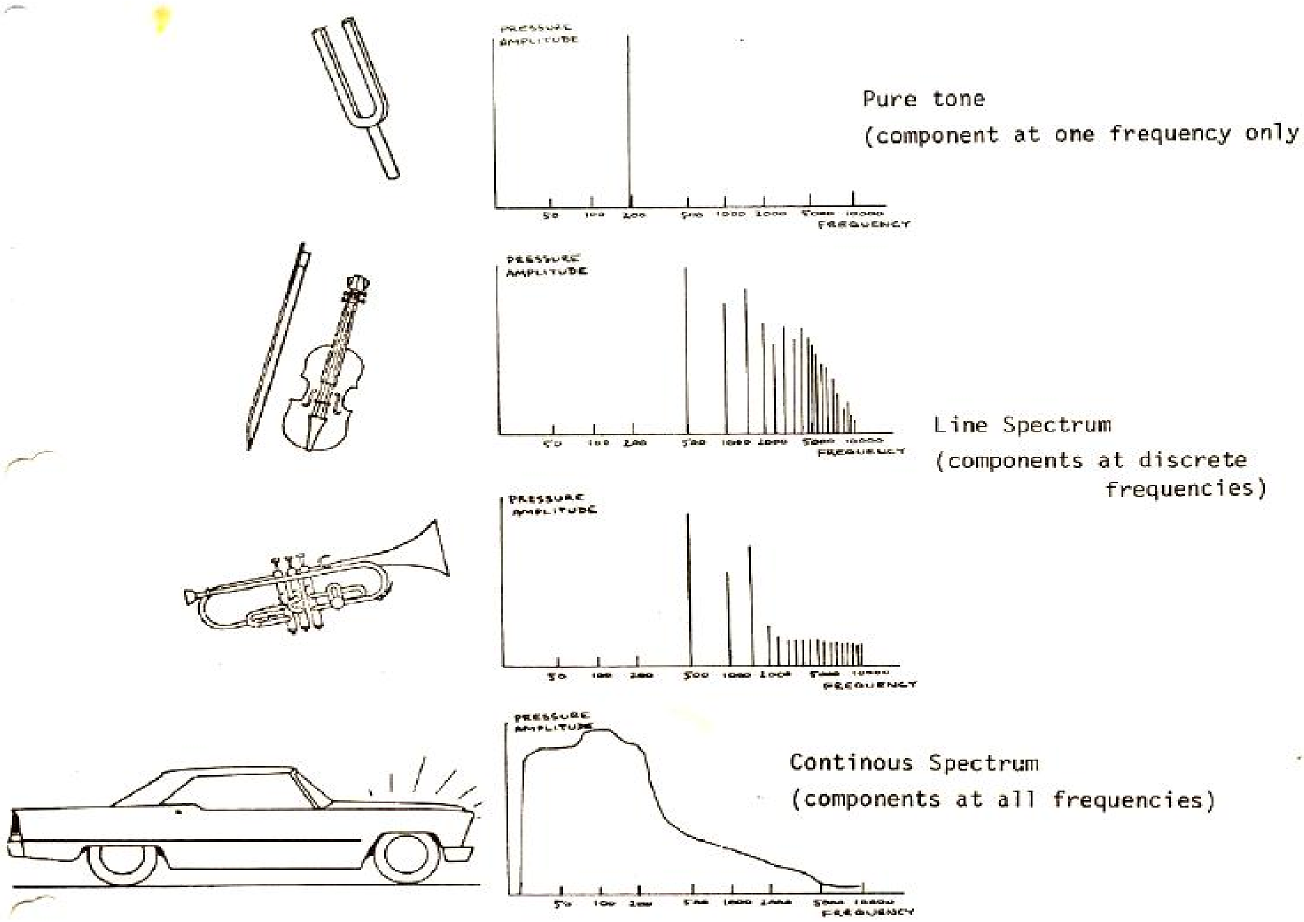

Fig 1.5 EXAMPLES OF FREQUENCY SPECTRA

1.4.5. FREQUENCY ANALYZER: All non-sinusoidal signals are composed of two 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 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 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. It is common to classify a filter according to its bandwidth, and there are two classes of filters which may be encountered i.e. constant bandwidth filters and filters with a constant percentage bandwidth.

(1) OCTAVE-BAND LEVEL

An important measure of noise is its frequency distribution. Instruments used to measure the distribution of sound over the audible frequency range, called spectrum analyzers. The spectrum analyzer which is in most common use divides the audible frequency range into bands one octave wide (An octave is a frequency interval between two sounds whose frequency ratio is 2, e.g., from 707 to 1414 Hz.) Such an analyzer is called an octave-band analyzer. The plot of the different octave-band levels vs. frequency is called an octave-band spectrum.

(2) ONE-THIRD-OCTAVE-BAND LEVEL

When more detailed information is required than that provided by an octave-band analysis, a one-third-octave-band analysis may be employed.

1.4.5.1 COMBINING LEVELS

Often it is necessary to combine levels, for example:

1. To calculate the sound level that results from a combination of sources of noise.
2. To determine the combined sound level of a source plus background noise.
3. To calculate the overall sound pressure level from octave-band levels or one-third-octave-band levels.
4. To combine the A-weighted sound level for a given octave-band spectrum.
5. To combine the sound power level of two or more sound sources

1.4.6. 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

$$L_{eq} = 10\, log_{10}\ \frac{1}{T}\int_0^T \left[\frac{P_A(t)}{Po}\right]^2 dt \qquad \textbf{……} (1.3)$$

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.

1.5. 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.

1.6. 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:

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

- **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.

1.7. NOISE MEASURING INSTRUMENTS AND TECHNIQUE

1.7.1. 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 we use Sound Level Meter ref to fig 1.6.

1.7.1.1. COMPONENTS OF SOUND LEVEL METER:

1. Microphone
2. Amplifier
3. Rectifier
4. Smoothing circuit
5. Meter

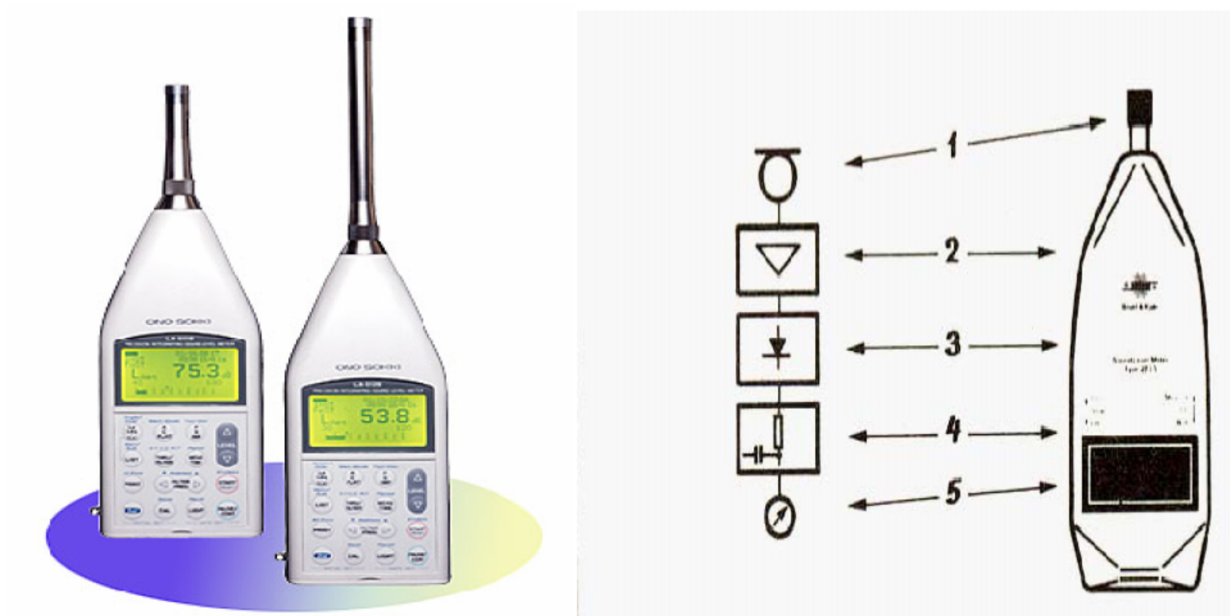

Fig.1.6 [23] Sound Level Meter

1.7.2. PRECAUTIONS IN MEASUREMENT

1. **POINTS TO BE KEPT IN MIND 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.

1.7.3. NOISE MEASUREMENT PROCEDURE

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 have 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.

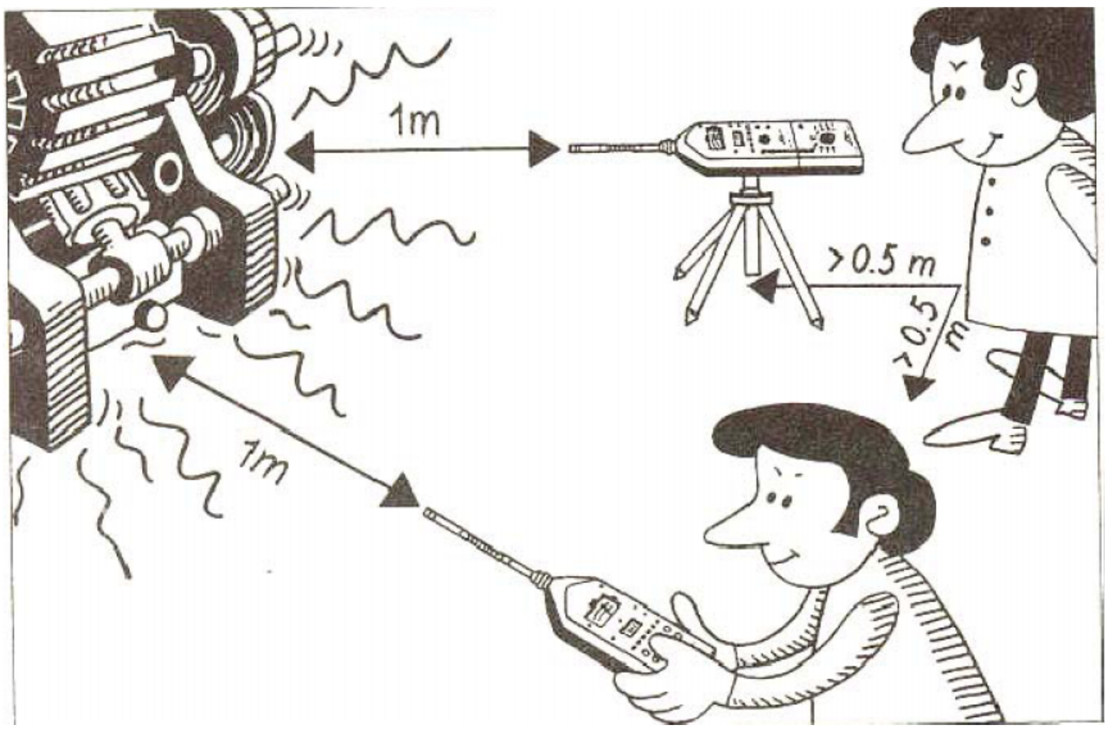

FIG 1.7 REPRESENTING ACCURATE METHOD OF MEASUREMENT THE SOUND LEVEL

- SLM should be at least at a distance of 0.5 m from the body of the observer.
- Reflections from the body of the observer can cause an error of up to 6 dB at frequencies around 400 Hz.
- SLM should be at a height of 1.2 –1.5 m from the floor level.
- Position from near buildings and windows is 1 –2 m away.
- Outdoor measurements to be made at least 3.5 m away from other reflecting structure.
- Within the room measurement should be made in the Free Field zone.

1.8. MEASUREMENT OF SOUND POWER

1.8.1. METHOD TO DETERMINE SOUND POWER LEVEL

1. Sound power level measurement with sound pressure level.

1.8.1.1. 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 (fig1.8) or a rectangular parallelepiped (fig 1.9).

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.

$$L_w = L_s + 10\log_{10}(S/S_o)$$

Where, So is reference area

S is hypothetical surface area

Hemisphere Positions

Sound Power Standards

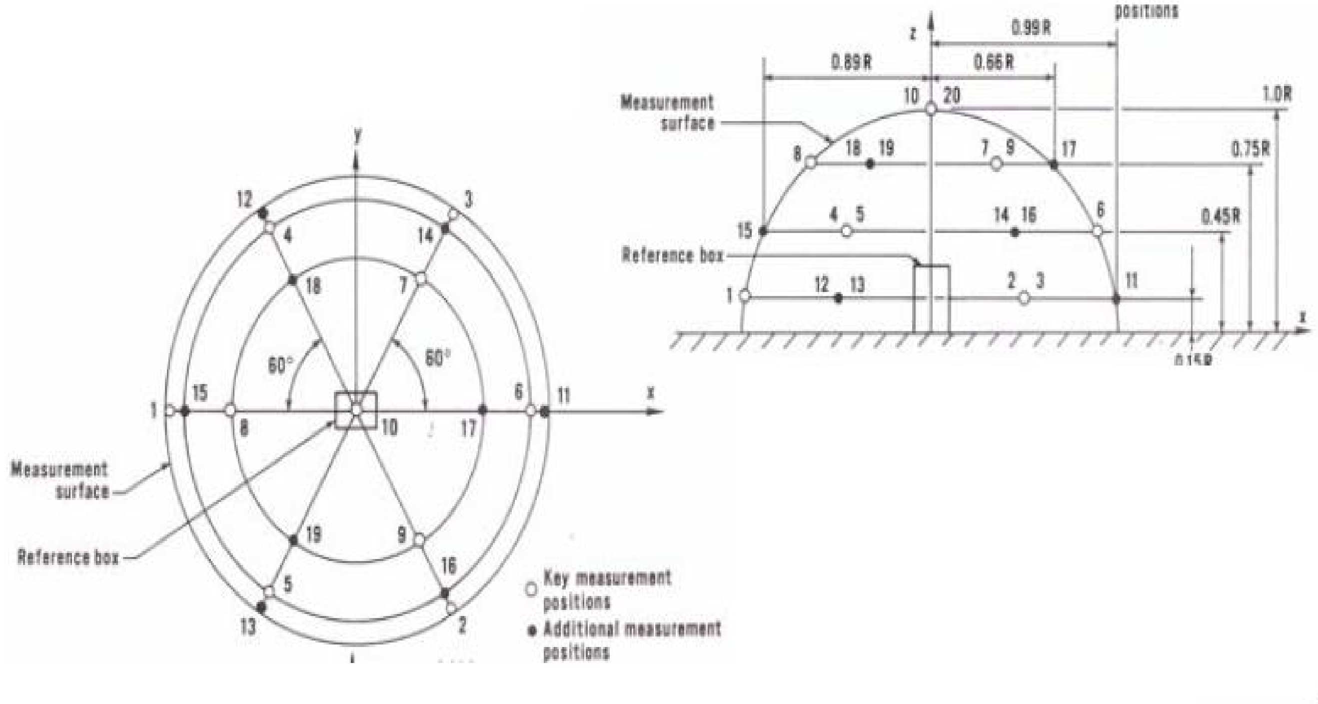

Fig.1.8 REPRESENTATION OF ARRAY OF MICRO PHONE POSITIONS ON AN IMAGINARY HEMISPHERICAL SURFACE SURROUNDING THE SOURCE WHOSE SOUND POWER IS TO BE MEASURED [6]

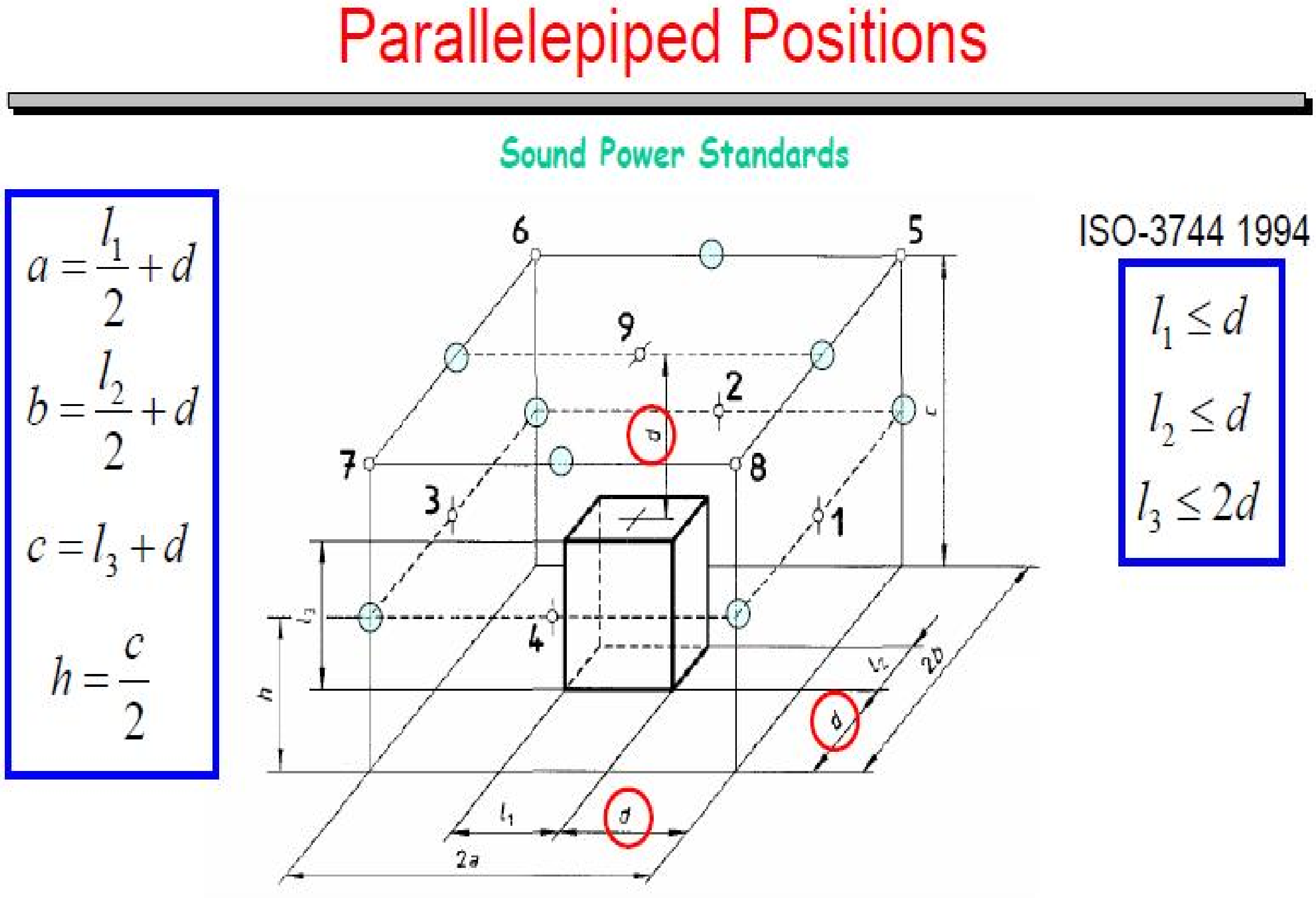

Fig.1.9 ARRAY OF MICRO PHONE POSITIONS ON AN IMAGINARY PARALLELEPIPED SURFACE SURROUNDING A SOURCE WHOSE SOUND POWER IS TO BE MEASURED [6]

1.8.1.2. ADDITIONAL ASPECT OF MEASUREMENT WHICH ARE NUMBERED TO CORRESPOND TO THE STEPS IN THE ABOVE PROCEDURE

1. For the small sources whose largest dimension is significantly less than one meter. It is usually more convenient to use 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. 1.8. The corresponding points for the rectangular parallelepiped are shown in Fig 1.9. 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.

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CHAPTER 2

INTRODUCTION TO C.I. ENGINE NOISE

An internal combustion engine operating on a thermodynamic cycle in which the ratio of compression of the air charge is sufficiently high to ignite the fuel subsequently injected into the combustion chamber. Since the combustion of fuel takes place inside the engine cylinder, so these engines are very noisy. C.I. Engine is also known as diesel engine.

According to the number of strokes per cycle, it is divided into two types:

1. Two stroke cycle engines
2. Four stroke cycle engines

   In the two-stroke, or two-cycle, type there is a complete cycle of operation in every two strokes of a piston. This type of engine requires a supply of compressed air for operating and for starting.

   In the four-stroke, or four-cycle, type the first down stroke of the piston draws in air, which is compressed on the upstroke to about 500 lb per sq in. (35 kg per sq cm). At the top of the stroke a jet of oil is sprayed in through a fuel injector. The oil is ignited and the rapid expansion of the gas created by the explosion forces the piston down in the working, or firing, stroke. The next upstroke drives the waste gases out through the exhaust valve, and the cycle is complete.

2.1. FACTORS CONTRIBUTING TO NOISE IN ENGINES

2.1.1. ENGINE NOISE

An engine is a mechanical device that produces some form of output from a given input.

THE VARIOUS FACTORS THAT CONTRIBUTE TO THE NOISE IN ENGINE ARE:

2.1.1.1. COMBUSTION NOISE: Combustion noise is caused by the rapid pressure rise and high peak pressure in the combustion chamber during combustion. This occurs once per revolution per cylinder for two stroke engines and once per two revolutions per cylinder for four stroke engines.

Combustion noise is produced because of unsteady combustion of fluid and is of two types: Turbulent combustion noise and periodic combustion oscillation. The turbulent combustion noise or combustion roar has no specific frequency but is composed of broad-band frequency spectrum. This noise is amplified if the flame is enclosed with the system resonance frequencies dominating. The requirements for reduction of this noise tend to be opposition to those for efficient combustion.

Combustion oscillations involve a feedback cycle that converts chemical energy into oscillatory energy in the gas flow to the combustion region. The mechanism is such that the pressure waves generated are so phased to the velocity fluctuations. The noise spectrum involves one specific frequency and its harmonics and that frequency is related to the resonant modes of the combustion chamber. Some of the possible cures are:

- **Modification of combustion chamber geometry.**
- **Change of air-fuel ratio, burner type, etc.**
- **Change of burning rate.**

It should be noted that Combustion roar in reciprocating engines which has frequency of the firing rate is not related to the combustion noise, but is due to the gross fluctuation in the flow rate produced by periodic action.

2.1.1.2. EXHAUST NOISE: The engine exhaust noise originates at the exhaust tailpipe openings and is transmitted through the exhaust cabin walls, exhaust firewall, and nose gear bay. Exhaust noise is the loudest noise source from outlet of the engine.

2.1.1.3. MECHANICAL NOISE: Mechanical noise is the noise which is generated by various impacts between the engine parts. This noise source is more important in the higher frequency range rather than in lower frequency range where combustion noise is important. There are lots of moving parts, for example, gear, valves, and rocker arms, piston and cylinder liner.

SOME ARE AS FOLLOWS:

1. **ENGINE CLICKING NOISE:** A clicking or tapping noise that gets louder when the engine is probably "tappet" or upper valve train noise caused by one of several things: low oil pressure, excessive valve lash, or worn or damaged parts.

2. **COLLAPSED LIFTER NOISE:** Worn, leaky or dirty lifters can also cause valve train noise. If oil delivery is restricted to the lifters (plugged oil galley or low oil pressure), the lifters won't "pump up" to take up the normal slack in the valve train. A "collapsed" lifter will then allow excessive valve lash and noise.

3. **VALVE LASH NOISE:** Too much space between the tips of the rocker arms and valve stems can make the valve train noisy and possibly cause accelerated wear of both parts.

4. **DAMAGED ENGINE PARTS NOISE:** Excessive wear on the ends of the rocker arms, cam followers (overhead cam engines) and/or valve stems can open up the valve lash and cause noise.

5. **RAPPING OR DEEP KNOCKING ENGINE NOISE:** A deep rapping noise from the engine is usually "rod knock," a condition brought on by extreme bearing wear or damage. If the rod bearings are worn or loose enough to make a dull, hammering noise.

2.1.1.4. PISTON SLAP: Piston slap noise is generated by the sudden impact of the piston to the cylinder wall is considered to be predominant due to the higher amount of energy released.

In the compression stroke, the connecting rod pushes the piston upwards overcoming the gas force. The force acting on the piston has a lateral component and the piston slides upwards on the minor thrust side of the cylinder wall. As the piston moves through T.D.C. the gas forces dominate the internal forces and keep the connecting rod in compression. Thus, as the crank pin passes through the cylinder center line, the lateral component of force on the piston pin changes direction, causing the piston to accelerate through the clearance and slap against the major thrust side of the cylinder wall. There are at least two piston slaps per revolution, but the major impact occurs at T.D.C. before the power stroke.

These simple models do not take into account others factors which may affect the piston motion such as:

1. Piston pin offset

2. Rocking motion of piston

3. Frictions at piston pin as well as piston's outer surface

4. Piston configuration, especially under operation

5. Pressure distribution around piston due to the squeezing motion of oil film

6. Compliance of cylinder liner wall

7. Cylinder liner deformation.

2.1.1.5. BEARING NOISE: Crankshaft bearings are always replaced when rebuilding an engine because they are a wear component. Heat, pressure, chemical attack, abrasion and loss of lubrication can all contribute to deterioration of the bearings. The above features give rise to the noise. Some of the factors that cause bearing noise are as follows:

1. **DIRT:** Dirt contamination often causes premature bearing failure. When dirt or other abrasives find their way between the crankshaft journal and bearing, it can become embedded in the soft bearing material. The softer the bearing material, the greater the embed ability, which may or may not be a good thing depending on the size of the abrasive particles and the thickness of the bearing material.

2. **HEAT:** Heat is another factor that accelerates bearing wear and may lead to failure if the bearings get hot enough. Bearings are primarily cooled by oil flow between the bearing and journal. Anything that disrupts or reduces the flow of oil not only raises bearing temperatures but also increases the risk of scoring or wiping the bearing.

3. **MISALIGNMENT:** Misalignment is another condition that can accelerate bearing wear. If the center main bearings are worn more than the ones towards either end of the crankshaft, the crankshaft may be bent or the main bores may be out of alignment.

4. **DISASSEMBLY:** Disassembly can be another cause of premature bearing failure. Common mistakes include installing the wrong sized bearings, installing the wrong half of a split bearing as an upper, getting too much or not enough crush because main and/or rod caps are too tight or loose, forgetting to tighten a main cap or rod bolt to specs, failing to clean parts thoroughly and getting dirt behind the bearing shell when the bearing is installed.

5. **CORROSION:** Corrosion can also play a role in bearing failure. Corrosion results when acids accumulate in the crankcase and attack the bearings causing pitting in the bearing surface. This is more of a problem with heavy-duty diesel engines that use high sulfur fuel rather than gasoline engines, but it can also happen in gasoline engines if the oil is not changed often enough and acids are allowed to accumulate in the crankcase.

2.1.1.6 GEAR NOISE: Engines with timing gears have backlash rattle, and most timing drives experience meshing frequency forces that often contribute to overall engine noise.

2.1.1.7. KNOCKING: The injection of fuel in C.I. engines takes place for a certain interval of time. This means, that as the first droplets to be injected are passing through the ignition delay period, additional droplets are being injected into the combustion chamber. In case of shorter delay period, the first droplets of fuel being injected will commence actual burning phase in a relatively short time after injection, and small amount of fuel will be accumulated in the chamber when the actual burning commences. Consequently the mass burning of the mixture will produce a smooth pressure rise and the combustion will be normal. If, the ignition delay is longer, the actual burning of first droplets is delayed and greater quantities of fuel droplet are accumulated in the combustion chamber. When the actual burning starts, the ignition of the large amount of accumulated fuel causes a violent and instantaneous rise of pressure. Under such conditions extreme pressure differentials are produced and violent gas vibrations known as detonation or knock occurs.

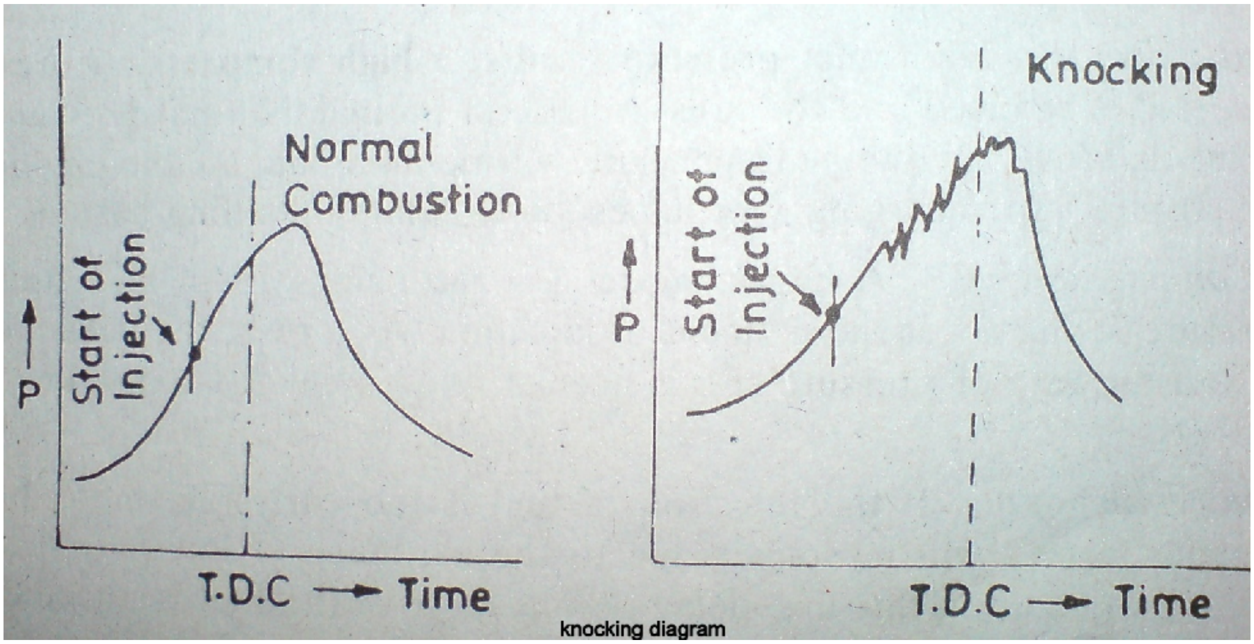

Fig 2.1 Knocking [25]

THE FOLLOWING OPERATING VARIABLES AFFECT THE DELAY PERIOD AND CAUSE DIESEL KNOCK:

1. **INLET TEMPERATURE:** A higher inlet air temperature increase the cylinder air temperature at the moment of injection. This decrease the delay period.

2. **INLET PRESSURE:** High inlet pressure results in high compression pressure. The delay period is reduced and the pressure rise is normal. Normal pressure rise eliminates knock.

3. **COMPRESSION RATIO:** A high compression ratio brings about high pressure and temperature in the air at the moment of injection. As a result, the delay period is reduced.

4. **INJECTION TIMING:** If the injection of the fuel is too early. The delay period will be longer due to the existence of low pressure and temperature of air. During this long delay period most of the fuel is injected and the mixing is relatively through. This results in an abnormal pressure rise.

5. **ENGINE SPEED:** The amount of fuel injected in the engine depends upon the engine speed because the fuel pump is driven by the engine crankshaft. At high speed more fuel will be injected during the delay period, and the mixing of air and fuel will also be better, both of which cause a high rate of pressure rise giving rise to knock.

6. **ENGINE SIZE:** A large engine runs at low speed and hence the quantity of fuel injected during delay period is less. Moreover in large engines, the compression temperature is higher. Both these effects have the tendency to reduce knocking.

7. **JACKET WATER TEMPERATURE:** Increase in jacket water temperature decreases the delay period.

8. **INJECTION PRESSURE:** Increase in injection pressure gradually decreases the delay period.

9. **TYPE OF COMBUSTION CHAMBER:** In general, a precombustion gives a shorter delay period compared to an open type combustion chamber.

10. **TURBULENCE:** The effect of turbulence is to strip the fuel from the injected spray and, therefore, promote a homogeneous mixture. A high turbulence thus decreases the delay period.

11. **FUEL VOLATILITY:** Highly volatile fuels with low viscosity are usually desirable in order to promote a homogeneous mixture quickly and so reduce physical delay. On the other hand, a highly volatile fuel may knock badly because a longer supply of combustion

mixture is furnished during delay period. In general, however, a low volatile fuel increases the probability of knock.

12. CHEMICAL COMPOSITION OF FUEL: Fuels having high self-ignition temperatures give longer delay periods.

13. ADDITIVES: some additives such as amylnitrate, reduce the ignition delay and hence the tendency of knock.

2.2. GENERAL CONSIDERATIONS OF THE MECHANISM OF ENGINE NOISE

The engine can be considered to consist of two basic structural elements.

1. The internal load-carrying structure, i.e. piston-connecting rod-crank-shaft system.

2. Outer load carrying cylinder block structure.

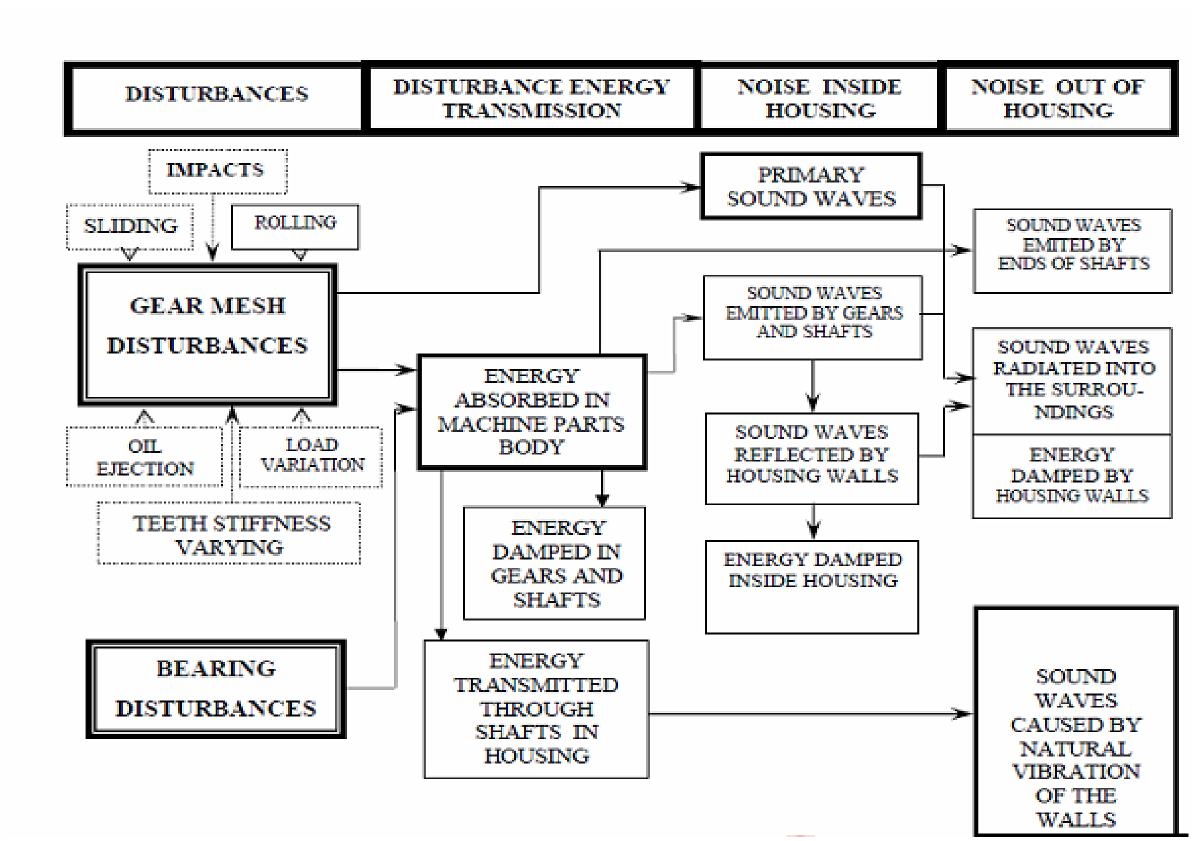

Fig.2.2 DIAGRAM SHOWING PROCESS OF DISTURBANCE ENERGY TRANSFORMATION WHICH CAUSES NOISE [23]

MECHANISM OF ENGINE NOISE: The internal load-carrying structure is mechanically separated from the main outer load –carrying structure by running clearance. A simple equivalent system of the engine thus can be developed. The principle existing forces are responsible of vibration and noise. The outer elastic load-carrying structure provides location for the piston and crankshaft represented as two masses joined together by a spring.

There are two major forces which are responsible for the engine structure vibration and the emitted noise, namely unidirectional forces and reversible forces.

2.2.1. COMBUSTION-INDUCED NOISE – UNIDIRECTIONAL FORCE EXCITATION

Unidirectional forces are only important in the vicinity of TDC on the compression stroke and are produced from compression and subsequent pressure rise resulting from combustion. The clearances in the vertical direction for the equivalent mass of piston, connecting rod, and crankshaft are taken up by these forces, and a linear vibratory system results. Since during this period the force does not change its direction any appreciable vibration can only be produced if there is a rapid change in the magnitude of the force. The rapid change in magnitude is produced by the onset of combustion in the engine cylinder and thus can be defined as combustion-induced noise. The gas forces excites the top part of the engine structure (i.e. cylinder head) while the lower part of the structure is excited by the combined gas forces and the inertia force.

2.2.2. MECHANICALLY-INDUCED NOISE - REVERSIBLE FORCE EXCITATION

Reversible forces which change direction are produced by the engine crank mechanism and associated inertia forces. Although these forces change in magnitude, the rate of change is too low to induce any appreciable vibration amplitudes in the comparatively stiff (high natural frequency) engine structure. These forces, however, accelerate the various elements of the internal load-carrying structure across the clearances and thus cause impact which effectively induces structure vibration and noise.

2.2.3. RELATION BETWEEN NOISE, ENGINE DESIGN AND OPERATING PARAMETERS [32]

Despite the numerous exciting forces which almost simultaneously excite the engine structure there is some justification to look at the problem in a simpler way. Since the gas force resulting from combustion tends to be the predominant force in most of the engines, the relationship between the gas force characteristics and emitted noise can be used to establish a basic model to identify the effects of fundamental engine design and operating parameters.

2.2.3.1 THE THREE BASIC PARAMETERS OF AN ENGINE ARE

1. **SPEED**
2. **SIZE**
3. **LOAD**

1. ENGINE SPEED

Engine speed can be considered as one of the basic parameters of engine. One can take an example of electro-dynamic vibration generators, and the broad response readily established by them. If the engine speed is doubled, the engine structure gets excited with lower order harmonics which have higher amplitudes. Since the general slope of the force spectrum is about 30 dB/ decade, an increase of excitation by 9 dB will be obtained with further speed the same pattern is followed.

It can be concluded that the characteristics of force determine the rate of increase with engine speed which in this instance, for a naturally aspirated diesel engine, is 30 dB per tenfold increase of speed.

EFFECT OF COMBUSTION SYSTEM ON NOISE: It has been seen that the combustion system determines the rate of increase of noise with engine speed and its exponent for petrol engine is 5, while for the most viscous diesel engine about 2.5. It can be seen that at low speeds there is a possibility of reducing noise by as much as 25 dB(A) by smoothing the development of the gas force.

2. **ENGINE SIZE**

   Measurement carried out on a large number of engines with engine size is considerably less. An increase of size to ten times gives an increase of noise of 17.5 dB(A). The detailed investigations now indicate that vibration levels of the engine surfaces are about the same irrespective of their size, thus the increase of noise with size is simply due to larger radiating surface area.

3. **ENGINE LOAD**

   Engine load has no effect on noise, which is in agreement with the findings that noise is simply due to the initial ignition of the fuel. This occurs at the same intensity whether the engine is running at no load at all or full load. It can be concluded that:

- The form of the exciting gas force determines the rate of increase of noise with engine speed.
- At high engine speeds the form of the gas force has a less significant effect on noise.
- Engine noise is independent of the horsepower produced

Fig 2.3 DYNAMOMETER USED IN VARIABLE COMPRESSION RATIO TEST RIG USED TO VARY LOAD [35]

CHAPTER 3
BIO-DIESEL

3.1 INTRODUCTION

Bio-diesel is fatty acid ethyl or methyl ester made from virgin or used vegetable oils (both edible & non-edible) and animal fats. The main commodity sources for bio-diesel in India can be non-edible oils obtained from plant species such as Jatropha Curcas (Ratanjyo t), Pongamia Pinnata (Karanj) , Calophyllum inophyllum (Nagchampa), Hevca brasiliensis (Rubber) etc. Bio-diesel contains no petroleum, but it can be blended at any level with petroleum diesel to create a bio-diesel blend or can be used in its pure form. Just like petroleum diesel, bio-diesel operates in compression ignition engine; which essentially require very little or no engine modifications because bio-diesel has properties similar to petroleum diesel fuels. It can be stored just like the petroleum diesel fuel and hence does not require separate infrastructure. The use of bio-diesel in conventional diesel engines results in substantial reduction of un-burnt hydrocarbons, carbon monoxide and particulate matters. Bio-diesel is considered clean fuel since it has almost no sulphur, no aromatics and has about 10 % built- in oxygen, which helps it to burn fully. Its higher cetane number improves the ignition quality even when blended in the petroleum diesel.

3.2 BIO-DIESEL AS AN OPTION FOR ENERGY SECURITY

India ranks sixth in the world in terms of energy demand accounting for 3.5% of world commercial energy demand in 2001.The energy demand is expected to grow at 4.8%. A large part of India's population, mostly in the rural areas, does not have access to it. At 479 kg of oil equivalent the per capita energy consumption is very low. Hence a programme for the development of energy from raw material which grows in the rural areas will go a long way in providing energy security to the rural people. The growth in energy demand in all forms is expected to continue unabated owing to increasing urbanization, standard of living and expanding population with stabilization not before mid of the current century. The demand of Diesel (H.S.D) is projected to grow from 39.81 MMT in 2001-02 to 52.32 MMT in 2006-07 @ 5.6% per annum. Our crude oil production as per the Tenth Plan Working Group is estimated to hover around 33-34 MMT per annum even though there will be increase in gas production from 86 MMSCMD (2002-03) to 103 MMSCMD in (2006-07). Only with joint venture abroad there is a hope of oil production to increase to 41 MMT by (2016-17). The gas production would decline by this period to 73 MMSCMD. The increasing gap between demand and domestically produced petroleum is a matter of serious concern. In other words, our dependence on import of oil will increase in the foreseeable future.

The Working Group has estimated import of crude oil to go up from 85 MMT per annum 45 to 147 MMT per annum by the end of 2006-07 correspondingly increasing the import bill from $ 13.3 billion to $ 15.7 billion. Transport sector remains the most problematic sector as no alternative to petroleum based fuel has been successful so far. Hence petro based fuels especially petroleum diesel (HSD) will continue to dominate the transport sector in the foreseeable future but their consumption can be minimized by implementation of Bio-diesel programme expeditiously.

Targets need to be set up for bio-diesel production to achieve blending ratios of 5, 10 and 20 percent in phased manner. The estimated bio-diesel requirements for blending with petro-diesel over the period of next 5 years are given in Table 3.1:

The Biodiesel requirement for 5%, 10% and 20% blends has been discussed below in table number 3.1:

Year	Diesel Demand Million Ton	Bio-diesel requirement for blending Million Ton		
		@ 5 %	@ 10 %	@ 20 %
2001-02	39.81	1.99	3.98	7.96
2002-03	42.15	2.16	4.32	8.64
2003-04	44.51	2.28	4.56	9.12
2004-05	46.97	2.35	4.70	9.40
2005-06	49.56	2.48	4.96	9.92
2006-07	52.33	2.62	5.24	10.48

TABLE 3.1 [14]

3.3 FEASIBILITY OF PRODUCING BIO-DIESEL AS DIESEL SUBSTITUTE

While the country is short of petroleum reserve, it has large arable land as well as good climatic conditions (tropical) with adequate rainfall in large parts of the area to account for large biomass production each year. For the reason of edible oil demand being higher than its domestic production, there is no possibility of diverting this oil for production of bio-diesel. Fortunately there is a large junk of degraded forest land and un-utilised public land, field boundaries and fallow lands of farmers where non-edible oil-seeds can be grown. There are many tree species which bear seeds rich in oil. Of these some promising tree species have been evaluated and it has been found that there are a number of them such as *Jatropha curcas* and ***Pongamia Pinnata*** **('Honge' or 'Karanja')** which would bevery suitable in our conditions. It will use lands which are largely unproductive for the time being andare located in poverty stricken areas and in degraded forests. It will also be planted on farmers' field boundaries and fallow lands. They will also be planted in public lands such as along the railways, roads and irrigation canals.

Economics of *Jatropha bio-diesel* US produces bio-diesel from edible oil (mainly soya oil), the 100% bio-diesel costsaround $ 1.25 to $2.25 per gallon depending upon purchase volume and the delivery costsand competes with low sulfur diesel oil. However, it is costlier to normal diesel and the B20 blend costs 13 to 22 cents more per gallon than normal diesel. It takes about 7.3 pounds of soybean oil which costs about 20 cents/pound, to produce a gallon of biodiesel. Feedstock costs alone therefore are at least $ 1.5 per gallon of soya-diesel. Under the mustard seed program, oil can be produced today for approximately 10 cents/pound and the total cost of producing mustard bio-diesel is around $ 1 per gallon. The mustard oil, a low value product contains as much as 90% mono-saturated fatty acids which makes it perfect for bio-diesel, balancing cold flow issues with N_{OX} emission issues. US is planning to add 5-10 billion gallons of bio-diesel through mustard seeds having mustard meal a high value pesticide that helps keep the price of mustard oil low.

In India, it is estimated that cost of Bio-Diesel produced by trans-esterification of oil obtained from Jaropha Curcas oil-seeds shall be approximately same as that of petrodiesel. The bye products of Bio-diesel from Jatropha seed are the seed oil cake and glycerol which have good commercial value. The seed oil cake is very good compost being rich in plant nutrients. It can also yield biogas which can be used for cooking and the residue will be used as a compost. Hence oil cake will fetch good price. Glycerol is produced as a bye product in the trans-esterification of oil. These bye-products shall reduce the cost of Bio-diesel to make it at par with petro-diesel.

The cost components of Bio-diesel are the price of seed, seed collection and oil extraction, oil trans-esterification, transport of seed and oil. As mentioned earlier, cost recovery will be through sale of oil-cake and of glycerol. Taking these elements into account, the price of Bio-diesel has been worked out assuming raw material cost of Rs. 3 per kg and varying prices of by-products. The cost of Bio-diesel varies between Rs. 9.37 per litre to Rs. 16.02 per litre depending upon the price assumed for the oil- cake and glycerol. The use of Bio-diesel is thus economically feasible.

3.4 BLENDING OF ESTERS & DIESEL

Blending conventional Diesel Fuel (DF) with esters (usually methyl esters) of vegetable oils is presently the most common form of bio-diesel. The most common ratio is 80% conventional diesel fuel and 20% vegetable oil ester, also termed "B20," indicating the 20% level of bio-diesel; There have been numerous reports that significant emission reductions are achieved with these blends.

No engine problems were reported in larger-scale tests with, for example, urban bus fleets running on B20. Fuel economy was comparable to DF2, with the consumption of bio-diesel blend being only 2-5% higher than that of conventional DF. Another advantage of bio-diesel blends is the simplicity of fuel preparation, which only requires mixing of the components.\

3.5 EFFECTS ON ENVIRONMENT AND HUMAN HEALTH - BIODIESEL

Biodiesel (mono alkyl esters) is a cleaner-burning diesel fuel made from renewable sources such as vegetable oils. Just like petroleum diesel, biodiesel operates in combustion-ignition engines. The use of biodiesel in a conventional diesel engine results in substantial reduction of unburned hydrocarbons, carbon monoxide and particulate matter. However, Emissions of nitrogen dioxides are either slightly reduced or slightly increased depending on the duty cycle and testing methods.

The use of biodiesel decreases the solid carbon fraction of particulate matter (since the oxygen in biodiesel enables more complete combustion to CO2), eliminates the sulphur fraction (as there is no sulphur in the fuel), while the soluble or hydrogen fraction stays the same or is increased. Therefore, biodiesel works well with new technologies such as oxidation catalysts.

Parameters	Pongamia (karanja) oil	Biodiesel	Diesel
Density (gm/ m^3)	**0.9048**	**-**	**0.85**
Viscosity (cSt)	**29.65**	**8.73**	**8.6**
Flash point (^oC)	**241**	**217**	**52**
Fire point (^oC)	**253**	**223**	**63**
Cloud point ($_0C$)	**7**	**6**	**5**

TABLE 3.2 [14] PROPERTIES OF PONGAMIA OIL

3.6 ECONOMICS REGARDING COST OF PONGAMIA (KARANJA) OIL

Local price of Pongamia seeds per kg = Rs. 5.00

Oil Extraction and transportation charge per kg = Rs. 1.00

Cost of 4 kg of seeds = 4 x 5 = Rs. 20.00

Oil Extraction cost = 4 x 1 = Rs. 4.00

Total Cost = Rs. 24.00

Income from sale of 3 kg of Cake = 3 x 4.5 = Rs. 13.50

Net Price of 1 Kg Oil = 24.00 – 13.50 = Rs. 10.50

CHAPTER 4

LITERATURE REVIEW

A wealth of literature exists in the area of engine noise using diesel as a fuel, but while finding out literatures regarding engine noise with biodiesel as alternative fuel it has been figured out that still a lot of work can be done regarding it. Using biofuels various researchers have worked extensively to study the other engine characteristics but noise has been one area which still requires attention. With crude reserves depleting every day future belongs to alternate fuels. In this literature survey lot of time and effort has been devoted to find out available literature using biofuels to observe their impact on engine noise. Some important literature are mentioned below:

1. **Mills et al [1] (1968)** discussed the various sources of the noise in I.C. engine commercial vehicles. They suggested some methods of noise reduction by the use of high transmission-loss enclosures, sound-absorbent and panel-damping materials were described.

   Examples were given on the practical applications of acoustical treatments to the reduction of the noise within and emitted by typical road haulage vehicles. Useful reductions can achieve by palliative treatments but attention was drawn to the practical and economic difficulties associated with incorporation of sound reducing treatments in production vehicles.
2. **Jha S.K**. **[2] (1976)** studied the characteristics of noise and vibration in a motor car. The predominant frequency regions in which noise levels are high were established. It was shown that the major part of the sound energy lies within the frequency region below 20 Hz and was caused mainly by road excitation being transmitted through the wheel and suspension system. The predominant noise in the audible range lies within 30-300 Hz frequency band and was produced primarily by body resonances excited by various engine harmonics. The vibrational and acoustical behavior of the car body at some of these critical frequencies was also discussed. Finally it was shown that by structural modification a substantial noise reduction can be obtained.

3. **Strahle Warren [3] (1977)** studied and practically performed the combustion randomness and diesel engine noise: Theory and initial experiments. During the process theoretical reasoning and preliminary experimental results show that the cycle to cycle variations in the turbulent combustion process in a diesel engine were responsible for a substantial amount of noise radiated from the engine. These results show that a new interpretation was required of past results which have compared the cylinder pressure spectrum and the radiated noise spectrum. A two-time variable procedure was used to construct a theory of the three dimensional, unsteady diesel engine combustion chamber pressure development. This theory was used to extract the spectral behavior of the chamber pressure when the heat release rate contains both periodic and random components. It was then theoretically shown, by use of the theory and past experience on combustion noise, that the randomness can be expected to dominate the periodic part of the pressure-time trace above some lower frequency bound of the order of 1000 Hz. Initial experiments to confirm the theoretical findings were performed under no-load conditions for a single cylinder, air-cooled Diesel engine.
4. **Kantarelis C. et al [4] (1988)** worked on the identification and subjective effect of amplitude modulation in diesel engine exhaust noise. The study reported in this paper examines the sources of diesel locomotive noise that give rise to annoyance. It was shown that the amplitude modulation present in diesel engine noise has importance in determining annoyance. The frequency and depth of the modulation were related to the fundamental firing frequency, the firing frequency of the engine and exhaust manifold design. Changes in the manifold design could reduce the adverse subjective response.
5. **Johansson Orjan et al [5] (1996)** studied that the lower front end of a diesel engine is a major noise source. Describing the source mechanisms of this area as problematic as it consists of a rotating torsional vibration damper in front of the timing transmission cover and the oil sump. This experimental study focused on the acoustic interaction phenomena between the damper and the structure behind it. To describe the source mechanisms a test series of different modifications by conventional lead wrapping technique was performed. The vibration behavior of each substructure was determined by operational deflection shape measurements and the source strength for each modification was determined by near-field sound intensity measurements. The results show the

contributions from different substructures and describe the interference effects due to coherent radiation. It was concluded that the radiation was dominated by the timing transmission cover structure behind the damper. At some frequencies though, the torsional vibration damper in combination with the timing transmission cover behind it, causes the high radiation. This effect was mainly due to coherent vibrations and a resonance phenomenon in the cavity between the structures.

6. **Bao Y.D. and He Y. [8] (2005)** studied the noise of rapeseed oil blends in a single-cylinder diesel engine. This study was undertaken to obtain the knowledge necessary for reducing noise of mixed oil composed of rapeseed oil and conventional diesel oil and for improving the performance of engine fuelled by the mixture. A S195 (8.8 kW) type single-cylinder diesel engine was used to determine the effect of four adjustable working parameters, i.e. intake-valve-closing angle (α), exhaust-valve-opening angle (β), fuel delivery angle (θ) and injection pressure (P, in 10^4 Pa) on noise when an oil mixture of 30% rapeseed oil and 70% diesel oil was used. Single-factor and multi-factor quadratic regressive orthogonal design test method were adopted in the experiments to find the relationship between noise and four adjustable working parameters. Relationship between these parameters and noise was analyzed under two typical operating conditions and mathematical equations characterizing the relationship were formulated. The equation of noise from the regressive test under each operating condition was set as the objective function and the ranges for the four adjustable working parameters were the given constraint condition. Models of nonlinear programming were then constructed. Computer-aided optimization of the working parameters for 30:70 rapeseed oil/diesel oil mixed fuel was achieved. Field test verified that the engine (in use) working condition was found to be bad at maladjustment. The optimum working parameters for two working conditions of the engine were used to adjust the four working parameters. Test results showed that optimum adjustment could achieve noise reduction between 2 and 4 dB and that the power could be increased by 0.6–1.8 kW. The experimental results also provided useful reference material for selection of the most preferable combination of working parameters.
7. **Bhattacharya M et al [9] (2005)** studied the operating parameters and the injection parameter (injection timing) on engine performance and environmental pollution factors

was studied. As an optimization objective, a 3.5 kW small direct injection diesel engine was used as the test engine, and its speed, load, and static injection timing were varied as per 4 x 4 x 3 full factorial design array. Radiated engine noise, smoke level, brake specific fuel consumption, and emissions of unburned hydrocarbons and nitrogen oxides were captured for all test runs. Objective functions relating input and output parameters were obtained using response surface methodology (RSM). Parameter optimization was carried out to control output responses under their mean limit using multi-objective goal programming and minimax programming optimization techniques.

8. **Nicklas Frenne et al [10] (2006)** performed an experiment on a diesel engine for validation of a method that retrieves source strength spectra, source strength time histories and sound pressure time histories of the engines complex partial sources. The method was based on empirical transfer function measurements and inverse matrix calculations briefly described in this article. Different simplifying source models were selected by comparison of calculated and measured auto spectra. The results represented: (1) indication of time efficient measurements of source strength spectra, (2) the importance of correct source models in the case of separated source strength time histories, and (3) spectra of separated sound pressure time histories. Listening tests reported that it was possible to detect well differentiated sounds of the partial sources.
9. **Kushwah Y S et al[11] (2006)** present work deals with testing and performance of a CI engine with different blends of non edible oil as well as their methyl esters with petro diesel. Various blends of non edible vegetable oil, commonly known as honge (Pongamia pinnata L) in india were prepared and tested over a wide range of engine load. Results obtained from the study show that 15 to 20% methyl ester diesel blend B15 and B20 could be a better fuel in terms of fuel efficiency and power developed. Further a blend can be used in any existing CI engine without modification and preheating. The Pongamia oil is transesterified by using base catalyzed transesterification process in a batch type transesterification reactor. Transesterification shows improvement in fuel properties of pongamia oil. Results obtained with B15 and B20 show imp in break thermal efficiency and reduction in break specific fuel consumption especially at higher load. Emission results show significant reduction in % of CO and HC for B15 and B20 and medium and higher power output.

10. **Gequn Shu et al [12] (2007)** studied the identification of complex diesel engine noise sources based on coherent power spectrum analysis. Their concept presented an identification of complex diesel engine noise sources based on coherent power spectrum analysis. Noise sources identification was essential for making noise reduction strategies. The author adopted the methods of hierarchy diagnosis together with coherent power spectrum analysis to identify complex noise sources of diesel engine. In investigation of the noise sources, the hierarchy tree and judgment matrix were given. Through noise sources identification, the main part of radiating surface noise were found. The result shows that the noise of low-frequency belt was mainly machinery noise from oil pump, gear and valve mechanism, etc., and it was radiated mostly from thin-walled part like gear cover and valve cover etc., while the noise of high-frequency belt was mainly combustion noise and oil pan was identified as the main part of noise reduction.

11. **Nivesrangsan. P et al [13] (2007)** identified the Source location of acoustic emission in diesel engines. They worked on third in a series developing methods of mapping acoustic emission (AE) signals and wave propagation in engines and focuses on source location techniques for the multi-source signals on relatively complex structures typical of machinery applications. Two source location techniques, a traditional wave velocity-based and an AE energy-based technique, using triangular sensor arrays, were used to locate source positions on the cylinder head of a 74 kW diesel engine using simulated sources (pencil lead break) and real sources (e.g. injectors (INJs) and exhaust valves during engine running).Source location using both techniques were demonstrated on the cylinder head of a 74 kW four-stroke diesel engine. The velocity-based technique uses AE wave speeds and time-of-flight (wave arrival time) to locate source position and was found to be most effective for single source signals with a sharp rising edge and good signal to noise ratios. The energy-based technique was based on a simple absorption attenuation model and was found to be useful for multiple source signals such as INJ signals, although structure-specific attenuation coefficients need to be measured for accurate source location.

12. **Rao Venkateswara et al [15] (2008)** after their study suggested that methyl esters of vegetable oils, known as biodiesel are becoming increasingly popular because of their

low environmental impact and potential as a green alternative fuel for diesel engine and they would not require significant modification of existing engine hardware. Methyl ester of Pongamia (PME), Jatropha (JME) and Neem (NME) were derived through transesterification process. Experimental investigations were carried out by them to examine properties, performance and emissions of different blends (B10, B20, and B40) of PME, JME and NME in comparison to diesel. Results indicated that B20 have closer performance to diesel and B100 had lower brake thermal efficiency mainly due to its high viscosity compared to diesel. However, its diesel blends showed reasonable efficiencies, lower smoke, CO and HC. Pongamia methyl ester gives better performance compared to Jatropha and Neem methyl esters.

13. **Rakopoulos D.C et al [16] (2008)** studied the experimental-stochastic investigation of the combustion cyclic variability in HSDI diesel engine using ethanol–diesel fuel blends. An experimental investigation was conducted to evaluate the combustion characteristics of a fully instrumented, high-speed, direct injection (HSDI), standard 'Hydra' diesel engine, at various loads when using ethanol–diesel fuel blends up to 15% by vol. ethanol. In each test, combustion chamber and fuel injection pressure diagrams of many consecutive cycles were obtained using a specially developed, high-speed, data acquisition and processing system. Following a performance and exhaust emissions investigation and a heat release analysis of the measured cylinder pressure diagrams reported by the authors, the present work focuses on the cycle-by-cycle combustion variation (cyclic variability) as reflected in the pressure indicator diagrams, by analyzing for the maximum pressure, maximum pressure rate, (gross) indicated mean effective pressure, and dynamic injection timing and ignition delay. These parameters were analyzed using stochastic analysis techniques for averages, standard deviations, coefficients of variation, probability density functions, auto-correlations, power spectra and cross-correlation coefficients. Thus, any cause and effect relationship between cyclic pressure variations and the injection system or the kind of fuel used can be revealed, given the concern for the low cetane number of ethanol blends promoting cyclic variability that can lead to degraded performance and emissions characteristics.
14. **Paswan M.K. et al [17] (2008)** suggested a new concept of Bio-Diesel in six stroke engines as conventional fuels are depleting day by day as they are limited reserves. These

finite reserves are highly concentrated in certain regions of the world. The increasing industrialization and motorization in the world has led to a steep rise on the demand of petroleum-based fuels. Therefore, countries not having these resources are facing energy/foreign exchange crisis, mainly due to the import of crude petroleum. Hence, it is necessary to look for alternative fuels which can be produced from resources available locally within the country such as jatropha alcohol, biodiesel, vegetable oils, etc. Their work reviews the production, characterization and current status of jatropha oil and biodiesel as well as the experimental research work carried out in various countries. Internal combustion engines are basically four stroke engines and generate a considerable amount of power. But the problem with these engines is that they generate only 40-50% of the energy and the rest is lost in various forms. Since these engines use non renewable fuels such as petrol or diesel, and since the depletion of these resources is a growing concern in the modern day, it has become mandatory to find alternative sources or fuel, or at least, change the design of the engine in such a way so as to get maximum possible efficiency. Their work dealt with tackling this problem.

15. **Ilkilic. C [18] (2009)** worked to evaluate exhaust emissions of a diesel engine operating by biodiesel fuel. In his study, an alternative diesel fuel, of which chemical modification was made by transesterification with short chain methyl alcohols, was produced from sunflower oil. The modified products were then evaluated according to their fuel properties as compared to diesel fuel. The fuel properties considered were viscosity, pour point, calorific value, flash point, and cetane number in addition to some other properties. The effects of biodiesel and diesel fuel on a direct injected, four stroke, single cylinder diesel engine exhaust emissions were studied. The results showed on the emissions of the engine that there were about 30% reduction in CO, about 26% reduction in CO_2, and about 25% reduction in NOx. The reduction exhaust emissions made the biodiesel a suitable alternative fuel for diesel engine and could help in controlling air pollution.
16. **Agarwal A. K. et al [19] (2010)** studied the aspects of Karanja oil utilization in a direct-injection engine by preheating and attempted the experimental investigations of engine durability and lubricating oil properties. While performing initial investigations they figured out that straight vegetable oil utilization as a diesel engine fuel has the advantages of eliminating the energy, time, and cost involved in biodiesel production. Since straight

vegetable oils have quite high viscosity compared with mineral diesel, they have to be modified to bring their combustion-related properties closer to mineral diesel. In there experimental study, a heat exchanger was designed and used to utilize the waste heat of exhaust gases for reducing the viscosity of Karanja oil by preheating. Carbon deposits, wear of vital engine components, and various effects of utilizing preheated Karanja oil on lubricating oil were analyzed during a long-term endurance test executed for 512h. Carbon deposits on various critical parts of the power cell were analyzed by visual inspection. For pistons, the piston merit rating was determined. The effect on a lubricating oil because of heated Karanja oil *vis-à-vis* mineral diesel was evaluated by comparing the densities, viscosities, flash points, carbon residues, ash contents, copper corrosion effect and pentane, benzene insolubility measurements at intervals of every 128h for 512h. The wear of the cylinder liner, piston rings, gudgeon pin, and small- and big-end bearings for a Karanja-oil-fuelled engine were also measured and compared *vis-à-vis* mineral diesel-fuelled engine.

17. **Haribabu. N et al [20] (2010)** worked on performance and emission studies on di –diesel engine fueled with pongamia methyl ester injection and methanol carburetion. The target of their study was to clarify ignition characteristics, combustion process and knock limit of methanol premixture in a dual fuel diesel engine, and also to improve the trade-off between NOx and smoke markedly without deteriorating the high engine performance. Experiment was conducted to evaluate the performance and emission characteristics of direct injection diesel engine operating in duel fuel mode using Pongamia methyl ester injection and methanol carburetion. Methanol was introduced into the engine at different throttle openings along with intake air stream by a carburetor which was arranged at bifurcated air inlet. Pongamia methyl ester fuel was supplied to the engine by conventional fuel injection. The experimental results represented that exhaust gas temperatures were moderate and there was better reduction of N_{OX}, HC, CO and CO_2 at methanol mass flow rate of 16.2 mg/s. Smoke level was observed to be low and comparable. Improved thermal efficiency of the engine was observed by them.

CHAPTER 5

EXPERIMENTAL SETUP

5.1. ENGINE DETAIL

The engine used in the present study was a Kirloskar four stroke, single cylinder, VCR (Variable Compression Ratio) diesel engine connected to eddy current type dynamometer for loading. The compression ratio could be changed without stopping the engine and without altering the combustion chamber geometry by specially designed tilting cylinder block arrangement. These signals were interfaced to computer. Provision was also made available for interfacing airflow, fuel flow, temperatures and load measurement. The compression ratio could be varied from 12 to 18 and the load could be varied from 0 kg to 8 kg. The set up had a stand-alone panel box consisting of air box, two fuel tanks for duel fuel test, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and engine indicator.

Fig.5.1 VARIABLE COMPRESSION RATIO ENGINE SET UP [36]

Model	Mahabli-6
Type	Kirloskar
No. Of cylinders	1
No. Of strokes	4
Bore	87.5mm
Stroke length	110mm
Rated power	5.2 kw
Speed	1500 rpm
Cooling system	Water cooled
Compression ratio range	12 to 18
Load	0 kg to 8 kg

Table 5.1 SPECIFICATIONS OF ENGINE

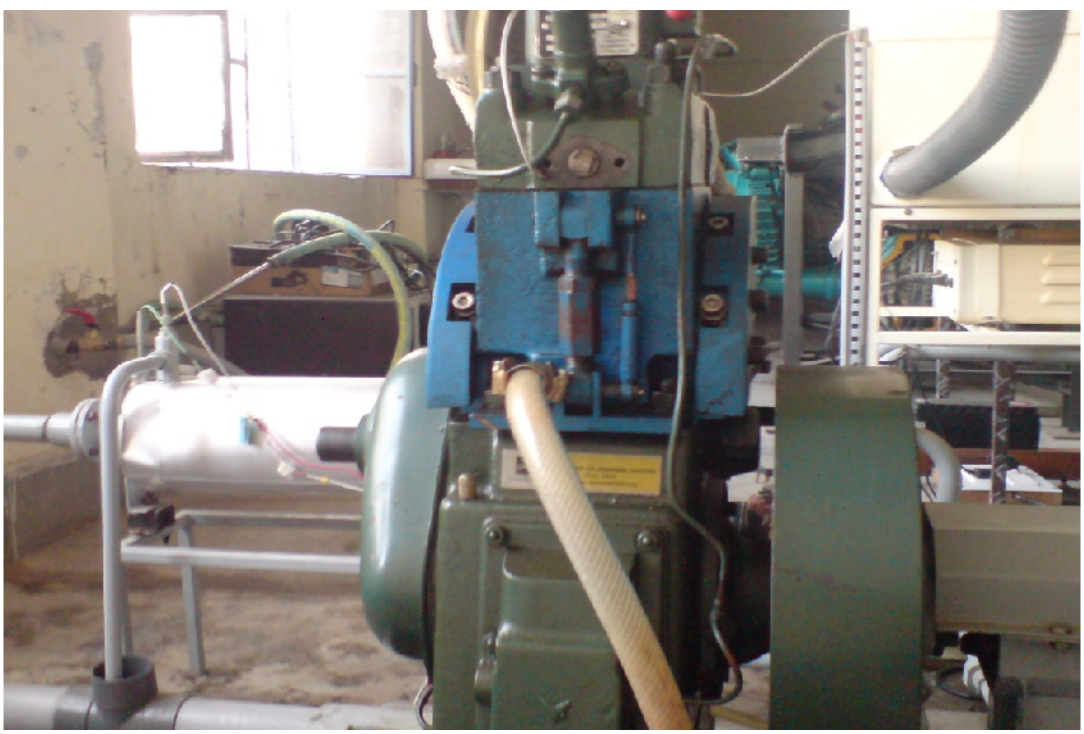

Fig.5.2 KIRLOSKAR ENGINE

5.2. EXPERIMENTAL PROCEDURE AND METHODOLOGY

1. First of all the engine size i.e. length, breadth, height were measured and then its area was calculated for the acoustic power measurement.

2. Points A, B, C, D were marked at 1 m distance from the engine boundary. To study the noise at different blends it was required to take the SPL at various points as close as possible to the engine as per the ISO standards. The four points A, B, C and D were taken at a distance of 1 m from each side of the engine and fifth point was taken at the Exhaust.

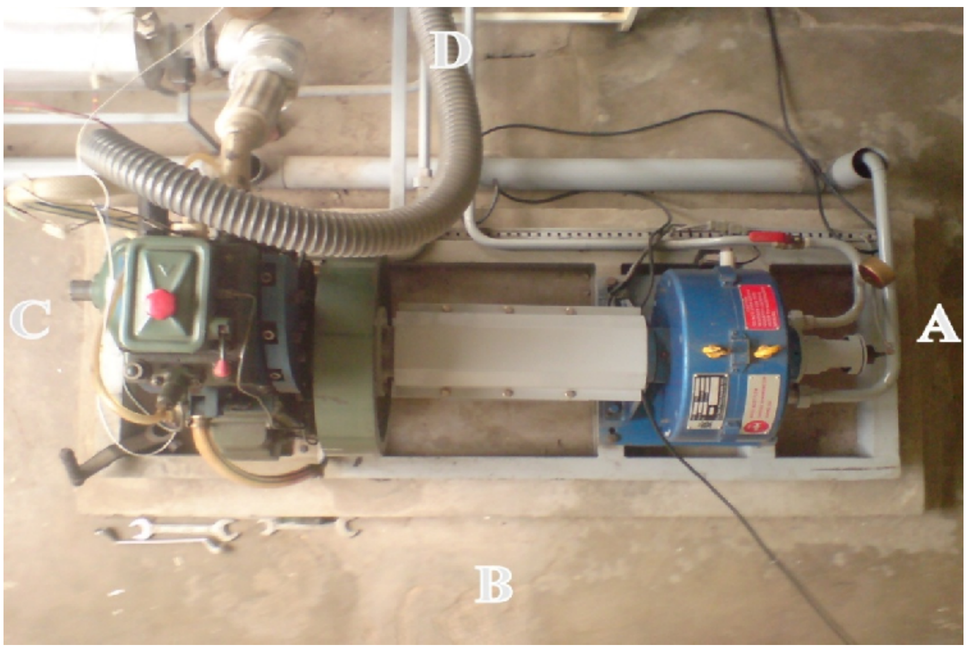

Fig.5.3 FOUR POINTS AT 1 M AWAY FROM ENGINE BOUNDARY [37]

3. Also mark the exhaust point of the engine which is going outside the wall to open environment as shown in figure 5.4 on next page.

Fig.5.4 EXHAUST GOING OUTSIDE

5.2.1. 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.5.5 SOUND LEVEL METER WITH WINDSCREEN

5.2.1.1 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.
- Sound Pressure Level (SPL) was measured with the help of SLM.
- Apply necessary corrections for background noise if any.

5.2.1.2 STEPS FOR CALCULATION OF SOUND POWER LEVEL:

- Hypothetical surface area according to reference area was determined with reference to engine.
- According to ISO-3744:1994 [6] Rectangular parallelepiped method was used as described in article 1.7.
- Creation of grid points was done with the help of threads as shown in fig 5.6 (a) and 5.6 (b) showing 17 grid points below

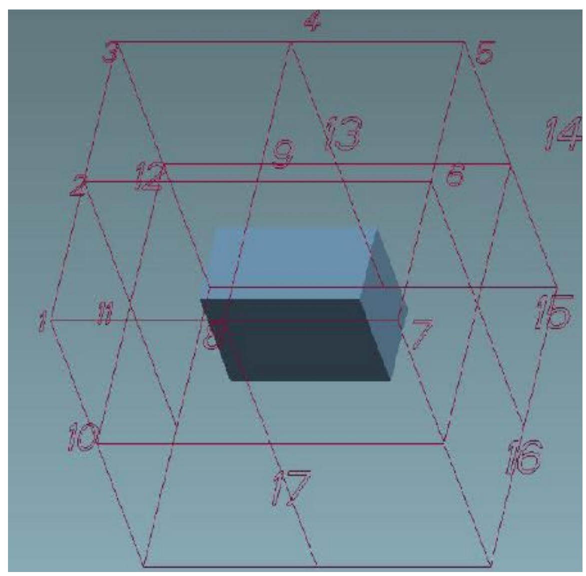

Fig.5.6 (a) GRID FORMATION, (b) REPRESENTING 17 GRID POINTS

- Microphone was arranged on the first specified point and corresponding SPL was measured the procedure was repeated for all the other specified points.

- Above step was repeated for different loads, different compression ratios and at different blends of Pongamia oil.
- Ls are calculated for all the points by taking mean of all seventeen grid points.
- The sound power level was calculated from

$$Lw = Ls + 10\log_{10}(S/S_o) \qquad \ldots\ldots (5.3)$$

- Where, S_o is reference area and S is hypothetical surface area
- The data above was used for calculating sound power level and for graphical representation different blends.

5.3 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 obtained at that point where SPL was maximum among A, B, C, and D. The readings were taken by changing the parameters of compression ratio (12 to 18) and loads (0 kg to 8 kg) and their effect on CI engine noise was studied on pure DF, B10, B20 and B30 blends of Pongamia oil. This was done using the SLM. This instrument gave the data of all frequencies in software "CESVA CAPTURE STUDIO".

The measurements were taken at point "B" by changing various compression ratios and loads.

CHAPTER 6

RESULTS AND DISCUSSIONS

After all the measurements were taken, it was required to analyze the sound pressure level by varying compression ratio and loads at different blends i.e., B10, B20, B30 and at pure DF [28]. The data for Sound Pressure Level was taken for five locations i.e. A, B, C, D and at exhaust. The points A, B, C and D were 4 typical points around the engine with each point being 1 m away surrounding the engine boundary as shown in fig 5.3 and exhaust went outside (fig 5.4). The value for exhaust has been taken at its outlet by keeping the SLM at 45^{o} and at the appropriate distance as per standards to measure the sound pressure level. The graphs are plotted between sound pressure level and loads keeping compression ratio constant as they clearly distinguish the reduction in SPL at different blends. All the data used in these graphs has been shown in Appendix A.

6.1. EFFECT OF PONGAMIA OIL BLENDS ON SPL AT DIFFERENT LOADS AND COMPRESSION RATIO'S

6.1.1 GRAPHS FOR POINT- A AT DIFFERENT BLENDS

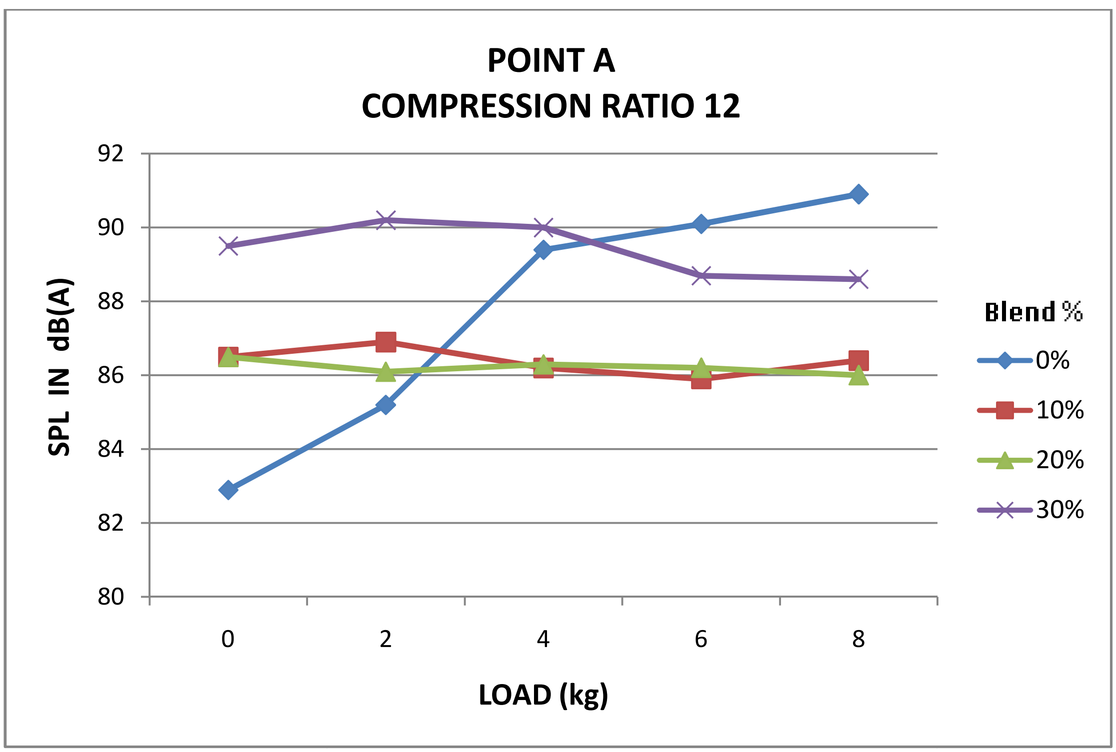

Fig 6.1 at POINT A

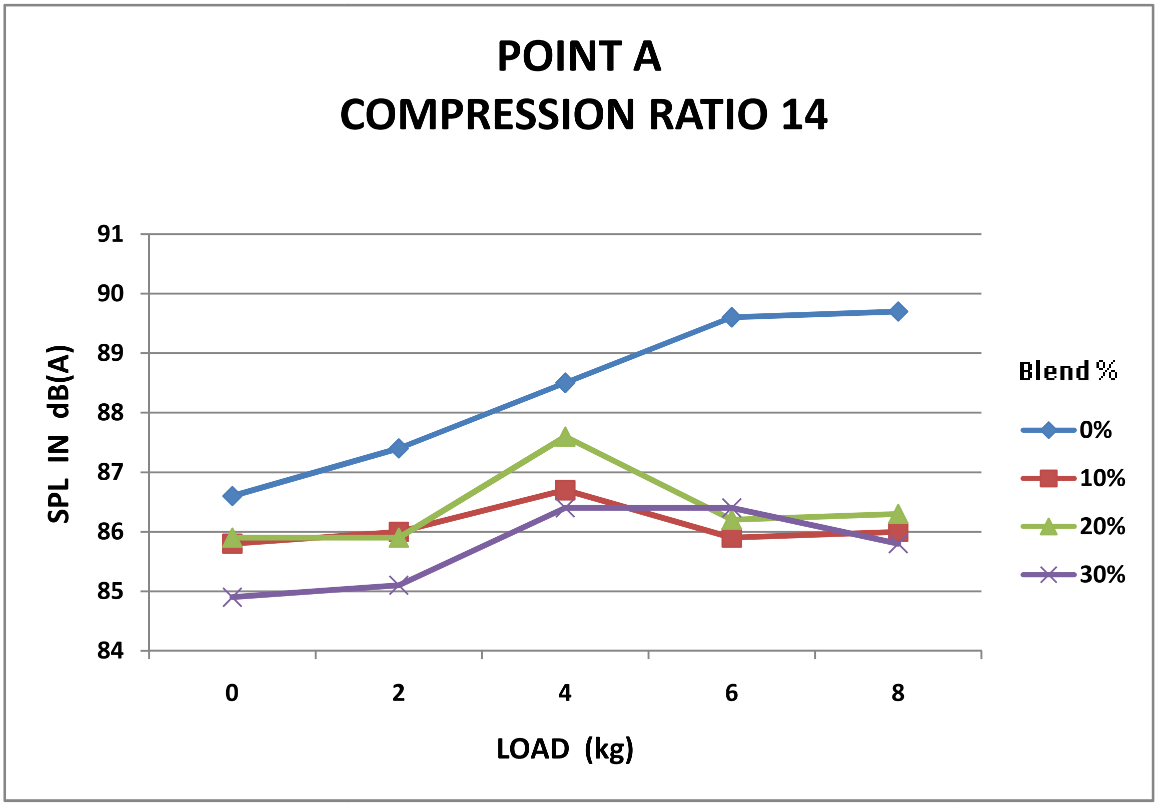

Fig 6.2 at POINT A

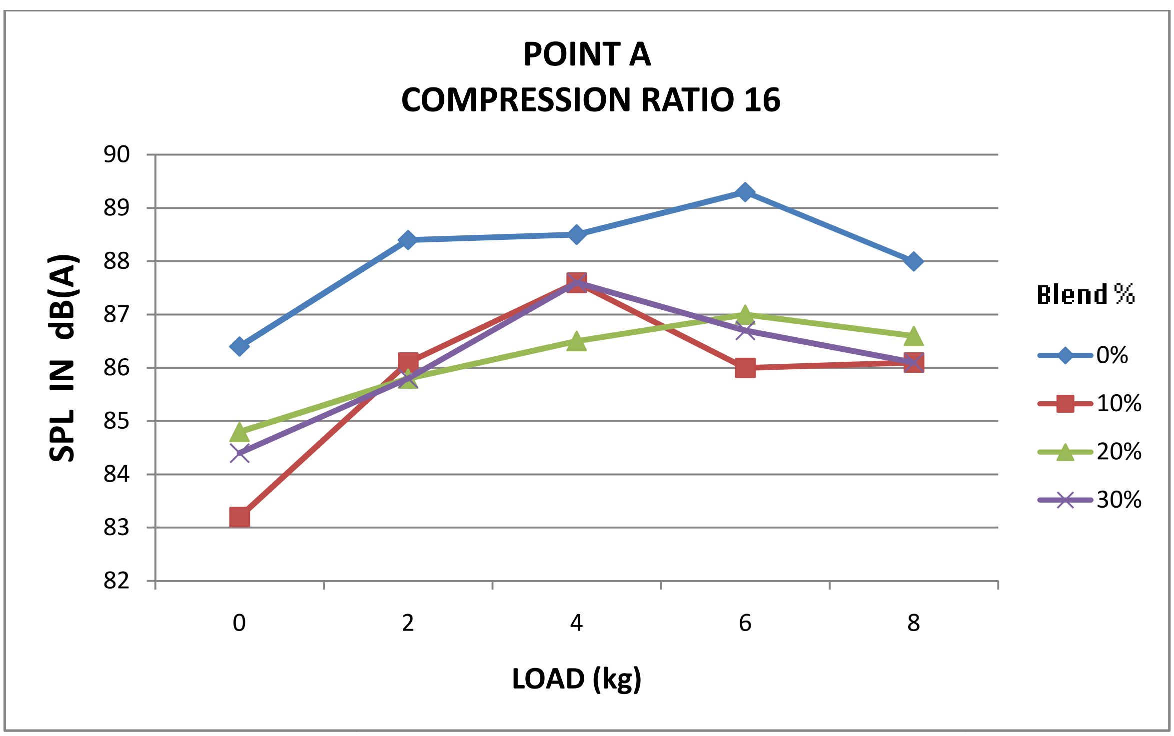

Fig 6.3 at POINT A

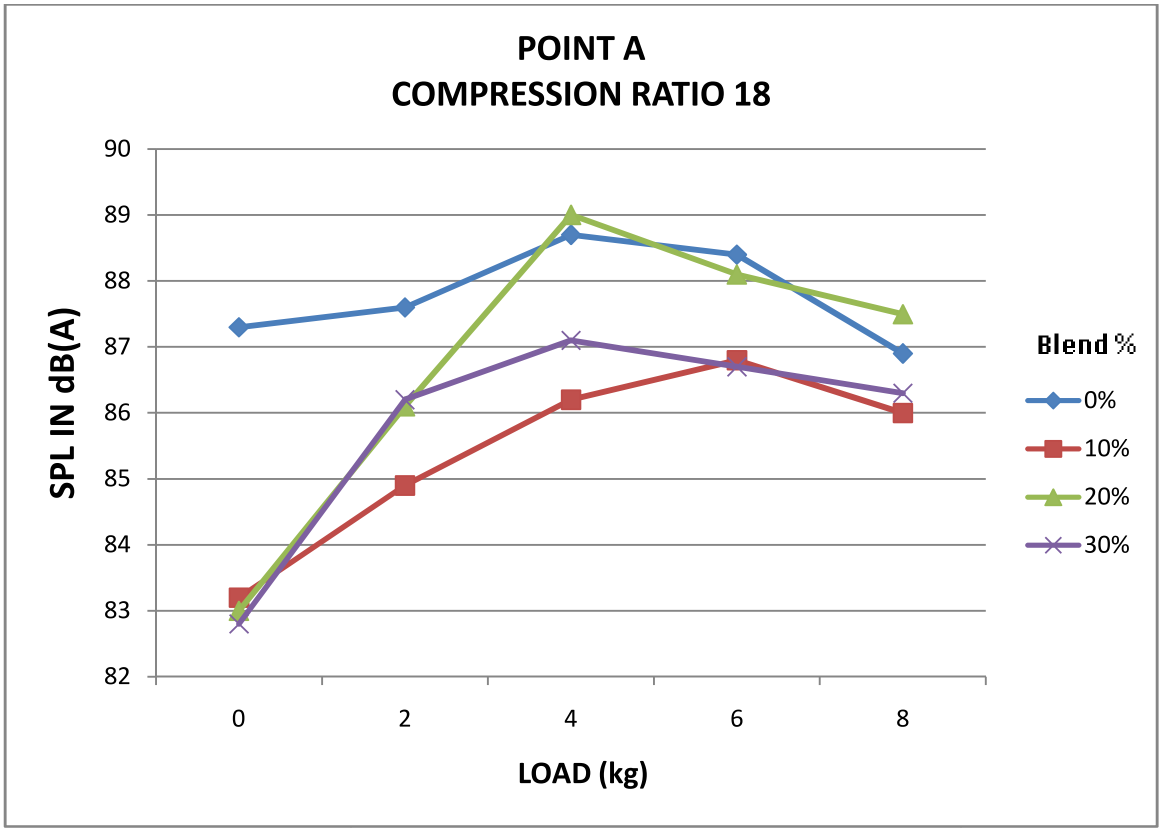

Fig 6.4 at POINT A

6.1.1.1 ANALYSIS FOR SOUND PRESSURE LEVEL AT POINT A:

The figures 6.1 to 6.5 shows the variation of sound pressure level at point "A" using different blends of Pongamia oil and by using pure diesel. Sound pressure level is measured by keeping the compression ratio constant and varying the loads at different blends B10, B20, B30 and pure DF [28].

- As shown in Fig.6.1 for compression ratio 12 the noise levels for B20 and B10 are considerably less than noise levels of pure DF and B30. For C.R.12 the B20 and B10 are almost similar.

- In Fig.6.2 for compression ratio 14 the pure DF produces a lot more noise than blended B20, B10 or B30 fuel. Addition of Pongamia oil in diesel reduces the noise level at point A 1 m away from the engine for C.R 14 than produced by pure DF.

- Figure 6.3 clearly indicates that minimum noise levels are at B20 for C.R. 16 in comparison to B10 and B30. The noise generated by using pure DF as fuel overshoots the noise produced by different blends at point A for C.R.16.

- Figure 6.4 represents that even at higher C.R. and higher loads there is some noise reduction except at loads 4 kg and 6 kg. Here noise levels are less at B10 rather than B20.

6.1.2 GRAPHS FOR POINT-B AT DIFFERENT BLENDS

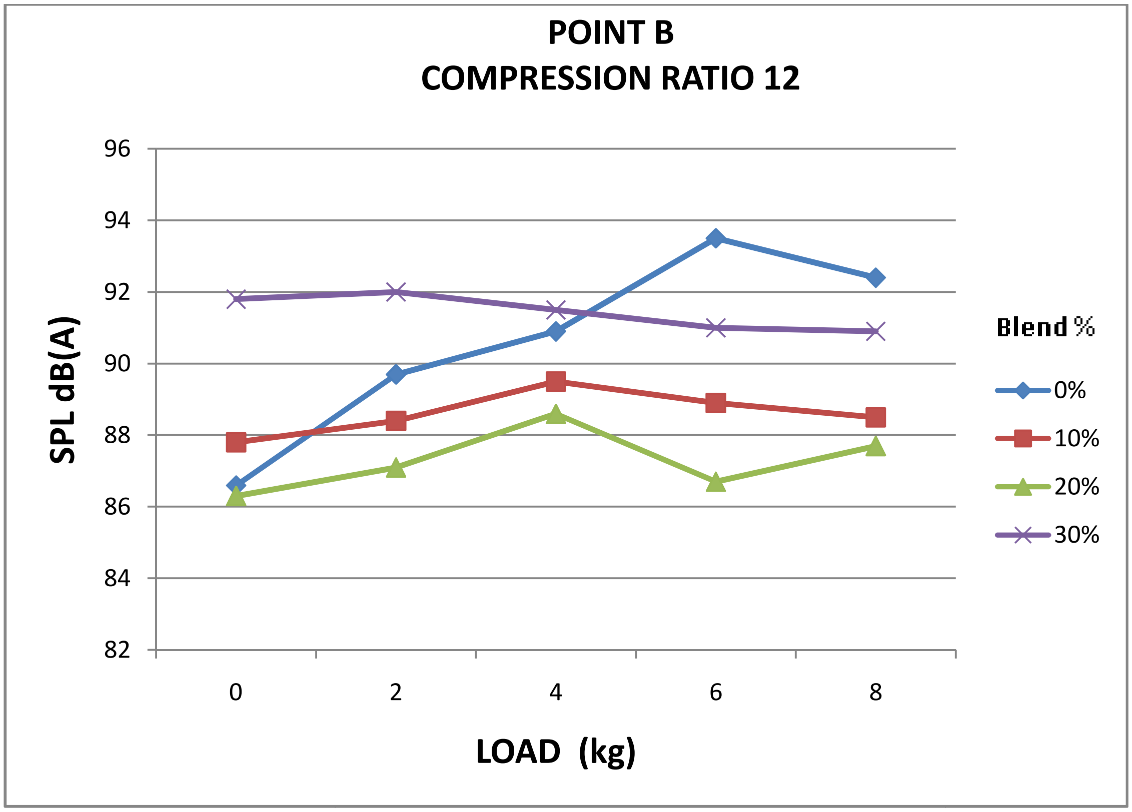

Fig 6.5 at POINT B

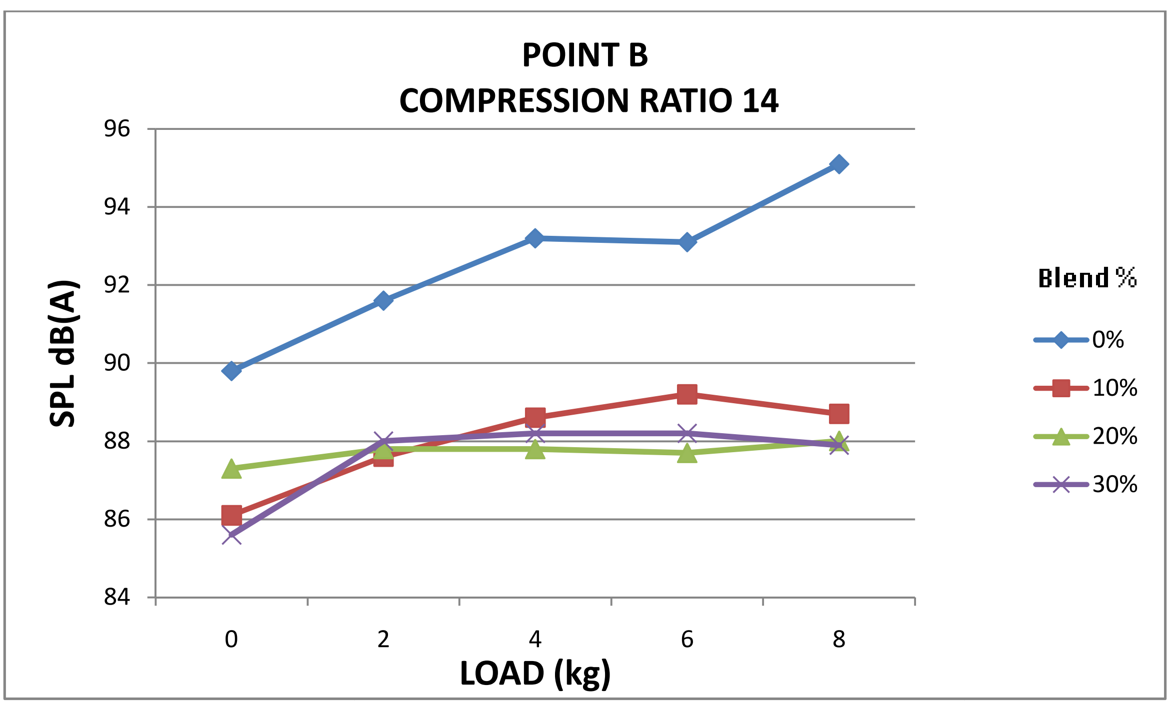

Fig 6.6 at POINT B

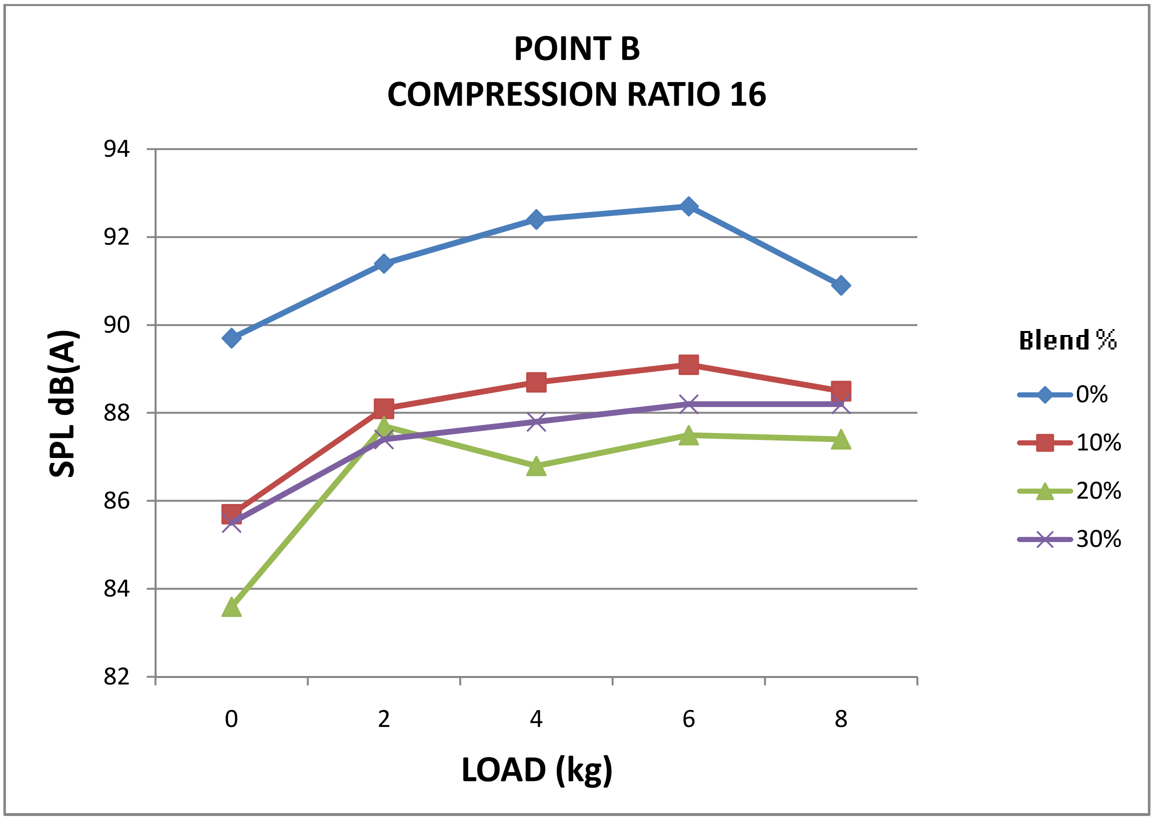

Fig 6.7 at POINT B

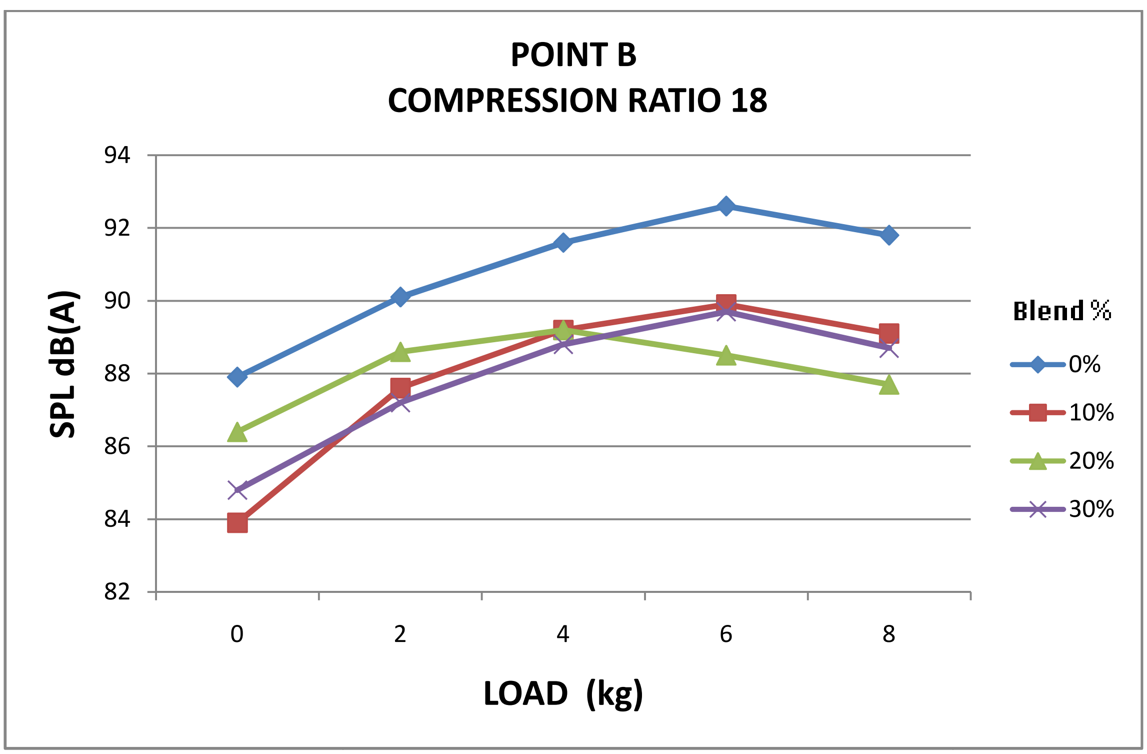

Fig 6.8 at POINT B

6.1.2.1 ANALYSIS FOR SOUND PRESSURE LEVEL AT POINT B:

The figures 6.5 to 6.8 shows the variation of sound pressure level at point "B" using different blends of Pongamia oil and by using pure diesel. Sound pressure level is measured by keeping the compression ratio constant and varying the loads at different blends B10, B20, B30 and pure DF [28].

- Figure 6.5 shows for compression ratio 12 the noise levels produced using B20 and B10 as fuel are considerably less than noise levels of pure DF and B30. The maximum SPL is 93.5 (dB (A)) at pure DF at 6 load and minimum SPL is 86.3 (dB (A)) at 0 load for B20.

- Figure 6.6 represents the data for C.R.14 at point B. Noise generation for B20, B10 or B30 is almost linear and is quite less than pure DF.

- In Figure 6.7 the noise produced by blended B20, B10 or B30 is less than pure DF for C.R.16.

- Figure 6.8 clearly indicates that overall minimum noise levels are at B20 for C.R. 18 in comparison to other combination of fuels or pure DF. The SPL for B10 and B30 almost varies linearly with changing loads.

6.1.3 GRAPHS FOR POINT-C AT DIFFERENT BLENDS

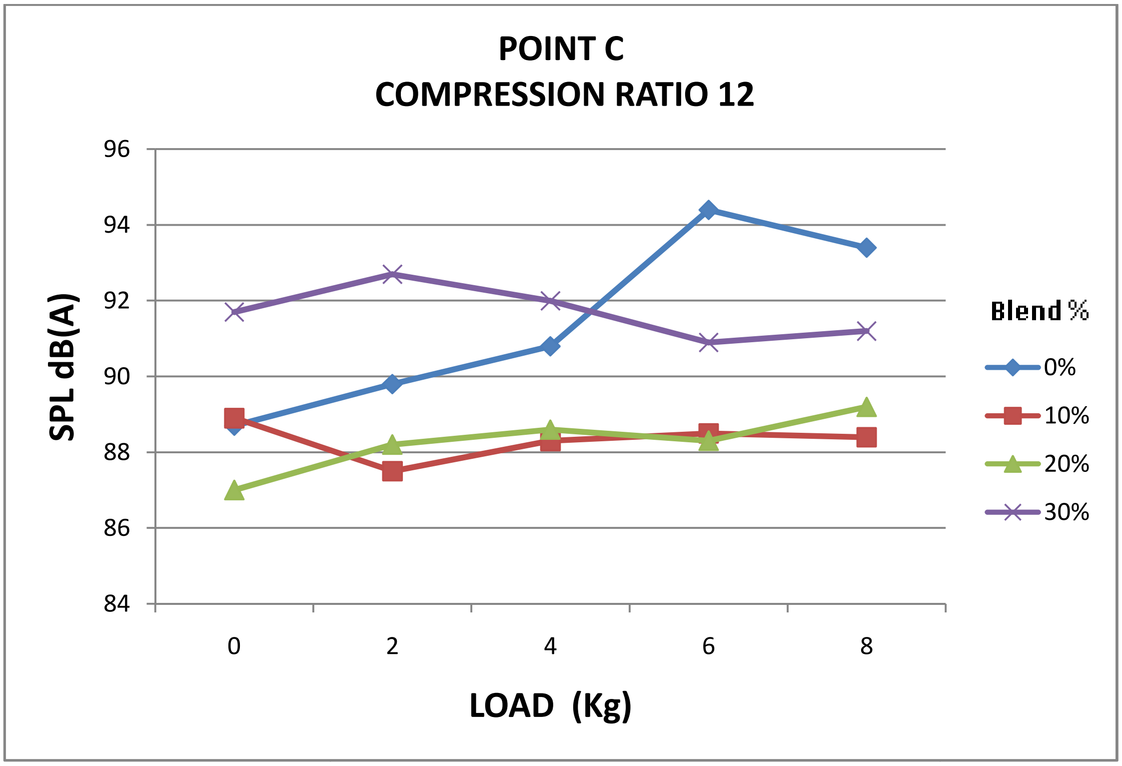

Fig 6.9 at POINT C

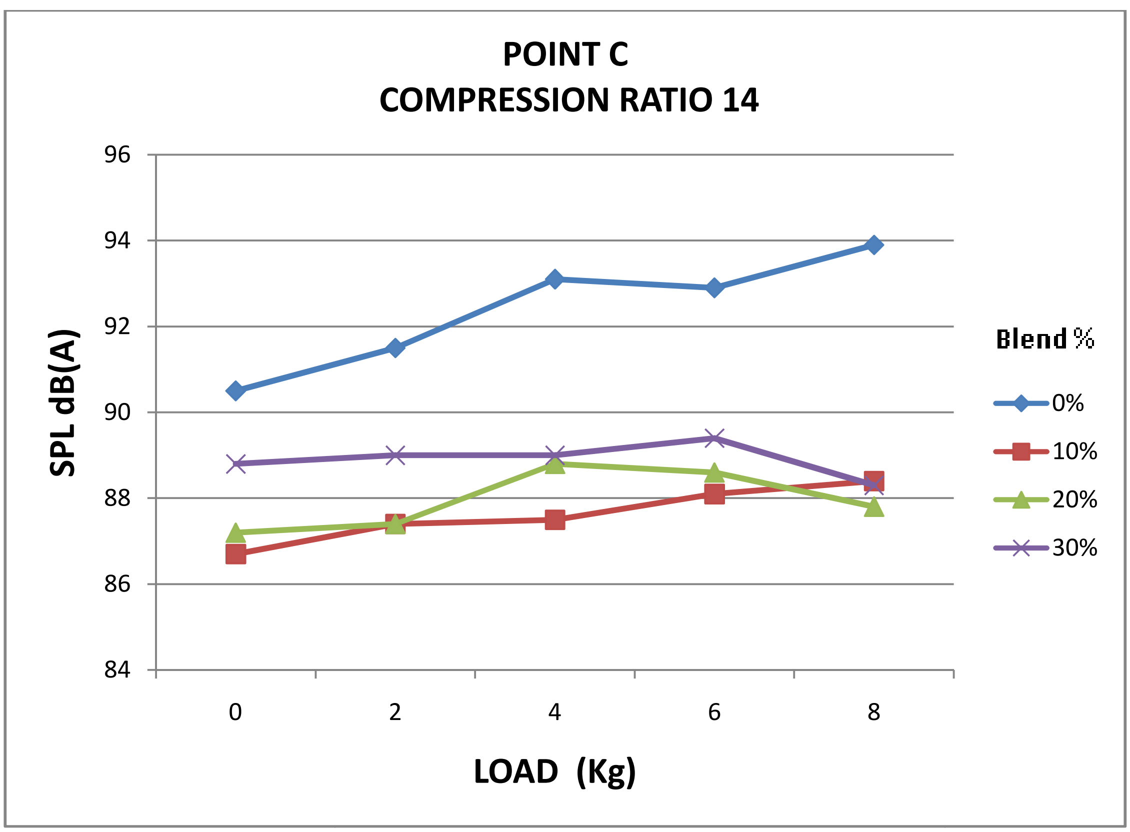

Fig 6.10 at POINT C

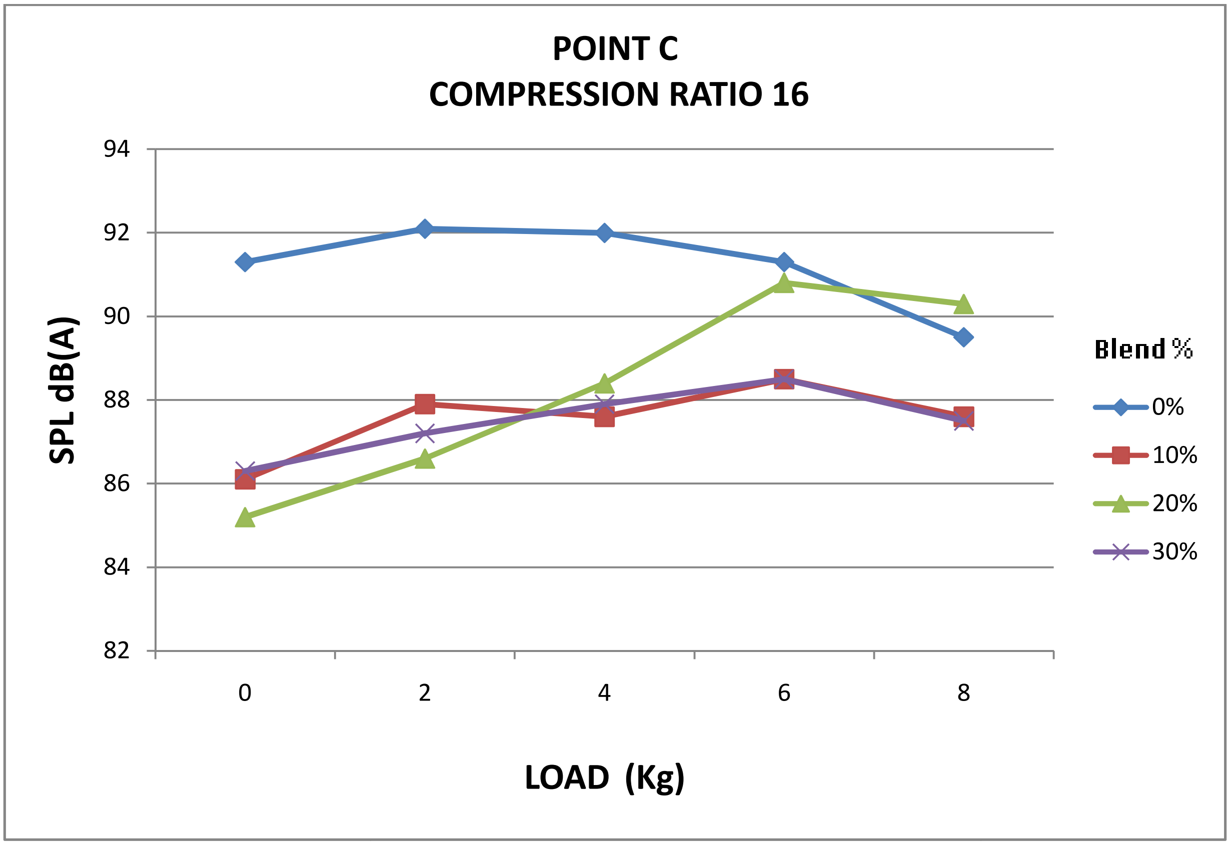

Fig 6.11 at POINT C

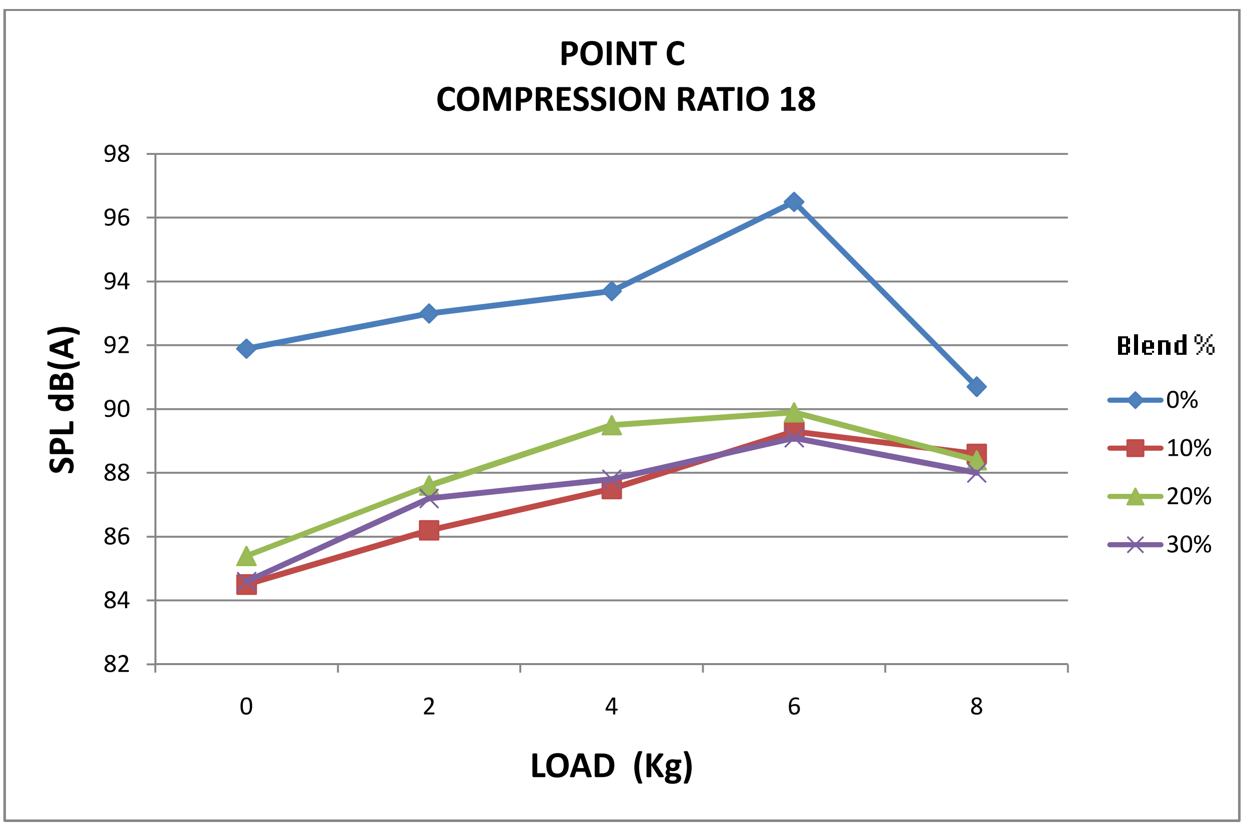

Fig 6.12 at POINT C

6.1.3.1 ANALYSIS FOR SOUND PRESSURE LEVEL AT POINT C:

The figures 6.9 to 6.12 show the variation of sound pressure level at point "C" using different blends of Pongamia oil and by using pure diesel. Sound pressure level is measured by keeping the compression ratio constant and varying the loads at different blends B10, B20, B30 and pure DF [28].

- Figure 6.9 shows for compression ratio 12 the noise levels for B20 and B10 as fuel are considerably less than noise levels of pure DF and B30. The maximum SPL is 93.4 (dB (A)) at pure DF at 8 load and minimum SPL is 87 (dB (A)) at 0 load for B20 fuel.

- Figure 6.10 represents the data for C.R.14. Noise variation for B20, B10 or B30 is almost linear as was for point B in fig 6.7 and is quite less than pure DF.

- In Figure 6.11 the noise produced by blended B10 or B30 is less than pure DF and B20. Initially noise levels are similar for B10, B20 and B30 till load 4 but afterwards the SPL for B20 is quite similar to pure DF for C.R.16.

- Figure 6.12 clearly indicates that Sound pressure levels for B10, B20 and B30 for C.R. 18 are quite less in comparison pure DF.

6.1.4 GRAPHS FOR POINT-D AT DIFFERENT BLENDS

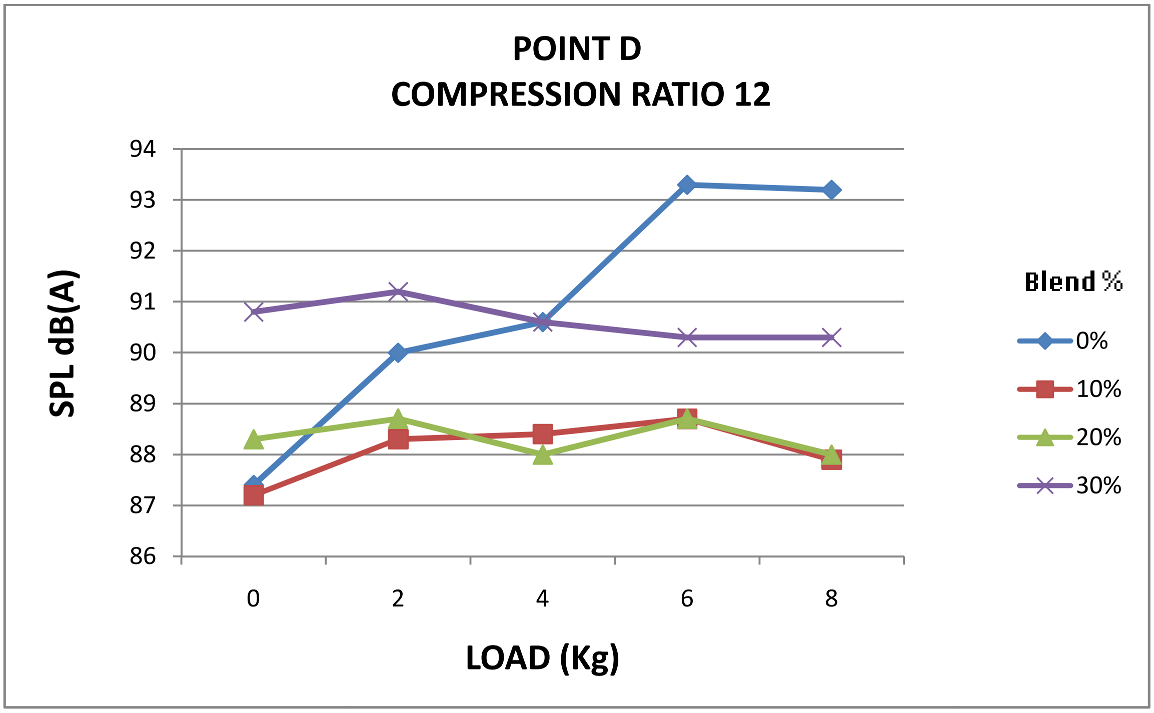

Fig 6.13 at POINT D

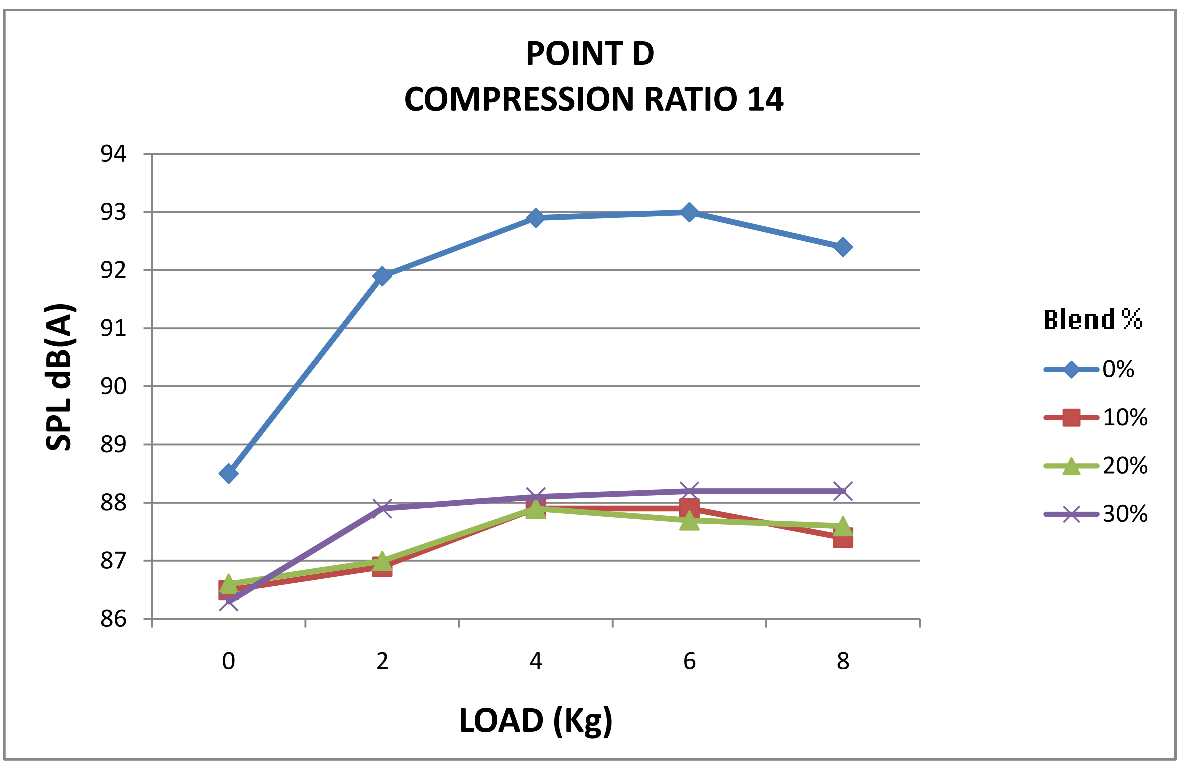

Fig 6.14 at POINT D

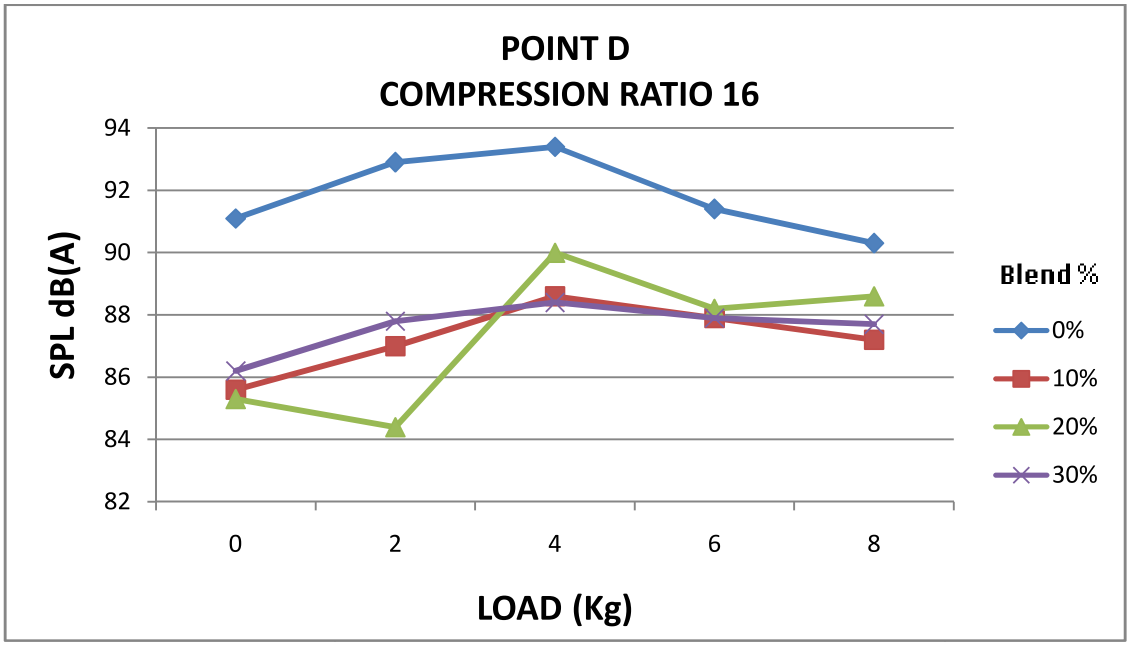

Fig 6.15 at POINT D

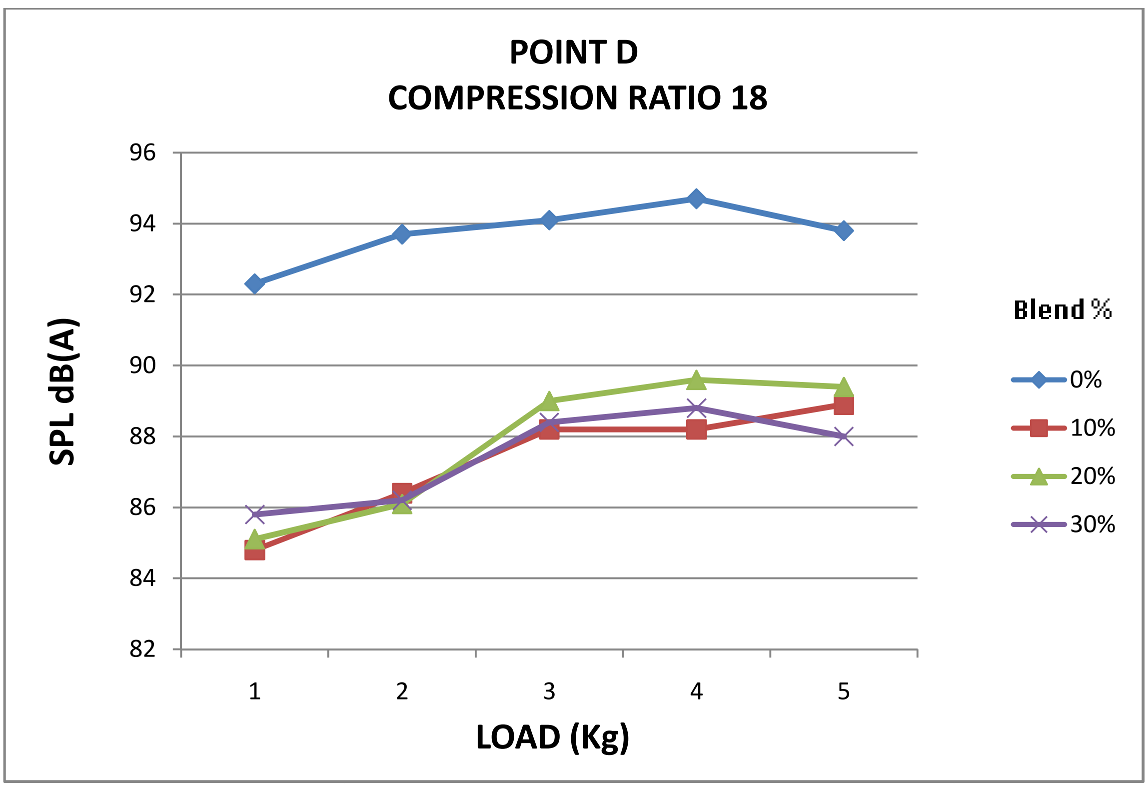

Fig 6.16 at POINT D

6.1.4.1 ANALYSIS FOR SOUND PRESSURE LEVEL AT POINT D:

The figures 6.13 to 6.16 show the variation of sound pressure level at point "D" using different blends of Pongamia oil and by using pure diesel. Sound pressure level is measured by keeping the compression ratio constant and varying the loads at different blends B10, B20, B30 and pure DF [28].

- Figure 6.13 shows that for compression ratio 12 the noise levels for B20, B10 and B10 as fuel for used VCR engine are considerably less than noise levels of pure DF. The behavior of noise variation of B10 and B20 was quite similar rather than B30 and pure DF. The maximum SPL is 93 (dB (A)) at pure DF at 6 load and minimum SPL is 86.5 (dB (A)) at 0 load for B10 fuel.

- Figure 6.14 represents the data for C.R.14. Noise variation for B20, B10 or B30 is almost linear as was for point C in fig 6.11 and is quite less than pure DF.

- In Figure 6.15 the noise produced by blended B10, B20 and B30 is less than pure DF. The maximum SPL is 93.4 (dB (A)) at pure DF at 4 load and minimum SPL is 85.3 (dB (A)) at 0 load for B20 fuel.

- Figure 6.16 clearly indicates that Sound pressure levels for B10, B20 and B30 for C.R. 18 are quite less in comparison pure DF.

6.1.5 GRAPHS FOR EXHAUST AT DIFFERENT BLENDS

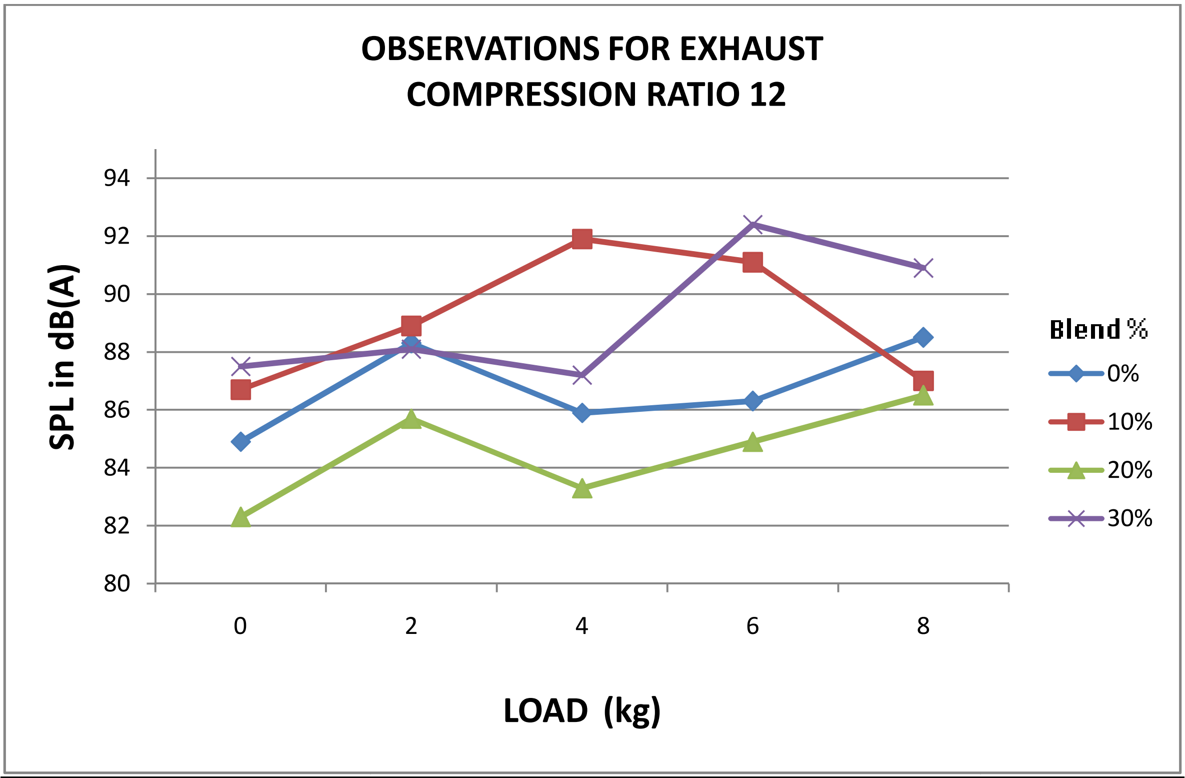

Fig 6.17 at EXHAUST

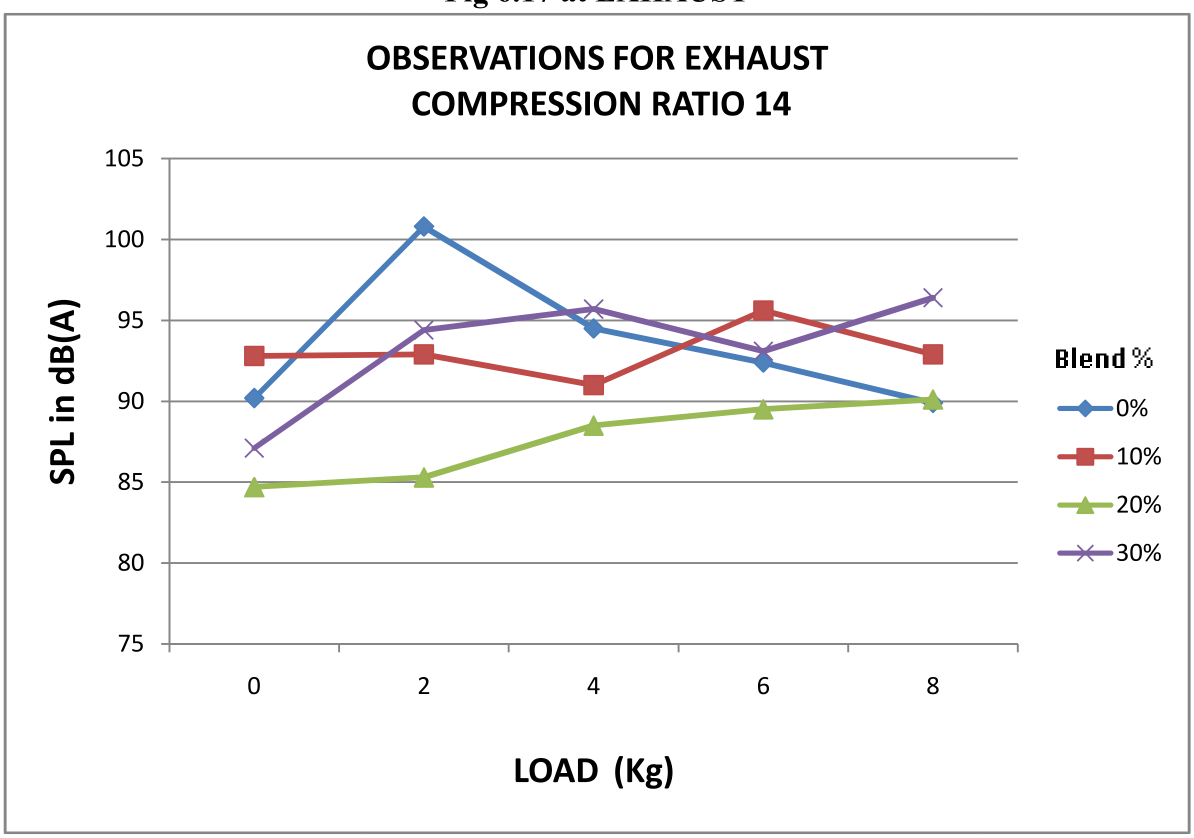

Fig 6.18 at EXHAUST

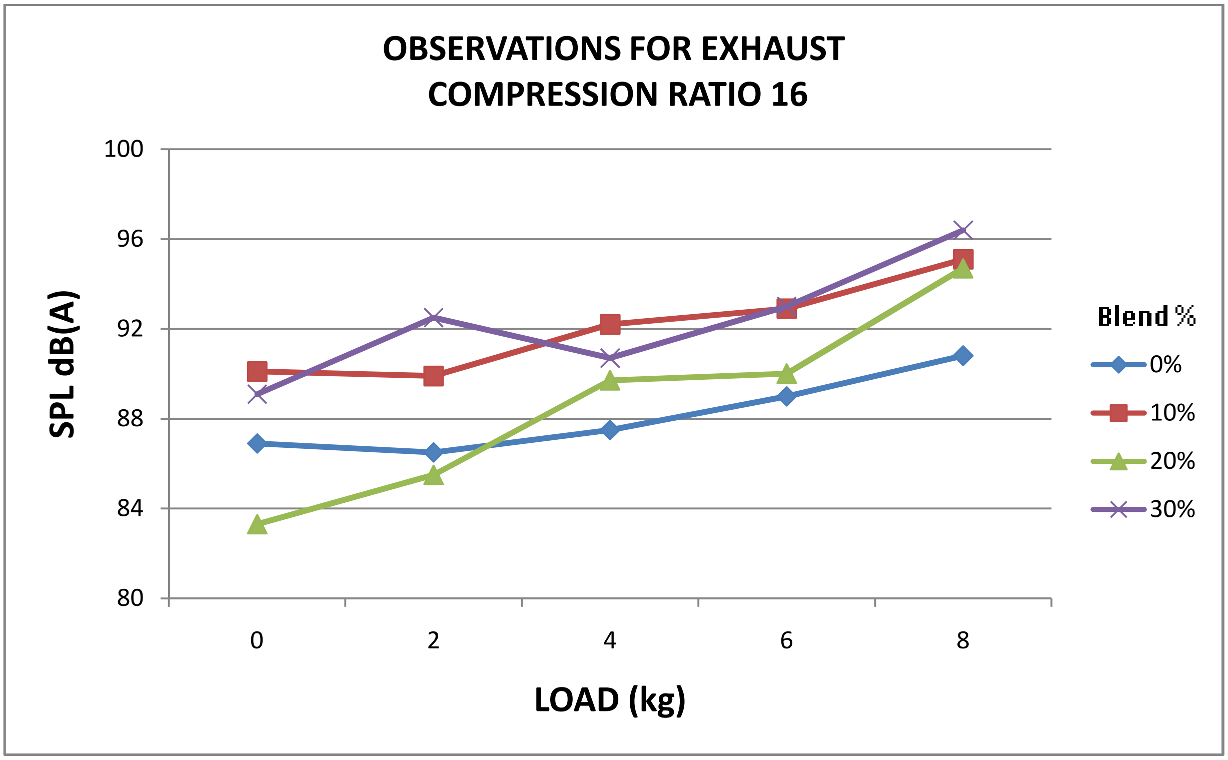

Fig 6.19 at EXHAUST

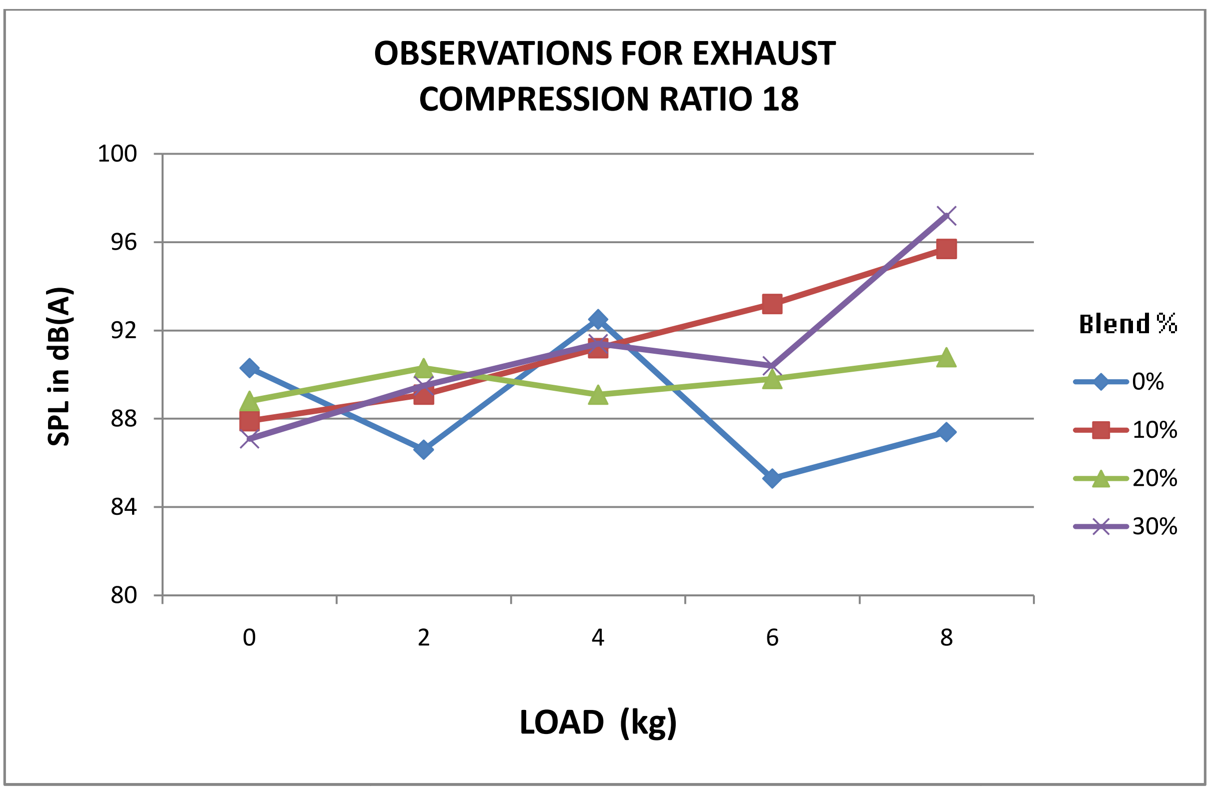

Fig 6.20 at EXHAUST

6.1.5.1 ANALYSIS FOR SOUND PRESSURE LEVEL AT EXHAUST:

The figures 6.17 to 6.20 show the variation of sound pressure level at "exhaust" point using different blends of Pongamia oil and by using pure diesel. Sound pressure level is measured by keeping the compression ratio constant and varying the loads at different blends B10, B20, B30 and pure DF [28].

- Figure 6.17 shows that for compression ratio 12 the noise levels for B20 as fuel for used VCR engine are considerably less than noise levels of pure DF and for other blends at B10 and B30. The behavior of noise variation of B10 and B30 was almost similar except at load 4 where SPL obtained using B30 was less than SPL produced by B10. The maximum SPL obtained was at B30 i.e., 92.4 (dB (A)) at pure DF at load 6 kg and minimum SPL was at 82.3 (dB (A)) at 0 load for B20 fuel.

- Figure 6.18 represents the data for C.R.14. Sound pressure level for B20, B10 or B30 is less than pure DF till load 4. Afterwards SPL for B30 increases and SPL for B10 and pure DF decreases. But minimum SPL was produced by B20 at C.R.14 among B10, B20, B30 and pure DF at Exhaust point.

- Figure 6.19 shows the data for C.R.16. The SPL produced by blended B20 and pure DF is less than B10 and B30. The maximum SPL is 96.4 (dB (A)) at B30 at 8 load and minimum SPL is 83.3 (dB (A)) at 0 load for B20 fuel.

- Figure 6.20 indicates the behavior for SPL at exhaust point for C.R.18. It represents Sound pressure levels for B10, B20 and B30 are almost linear. SPL for pure DF is lesser at load 6 and 8 in comparison to other fuel used and shows zig- zag behavior rather than showing linear behavior.

6.2 EFFECT OF PONGAMIA OIL BLENDS ON ACOUSTIC POWER KEEPING LOADS FIXED AT DIFFERENT BLENDS

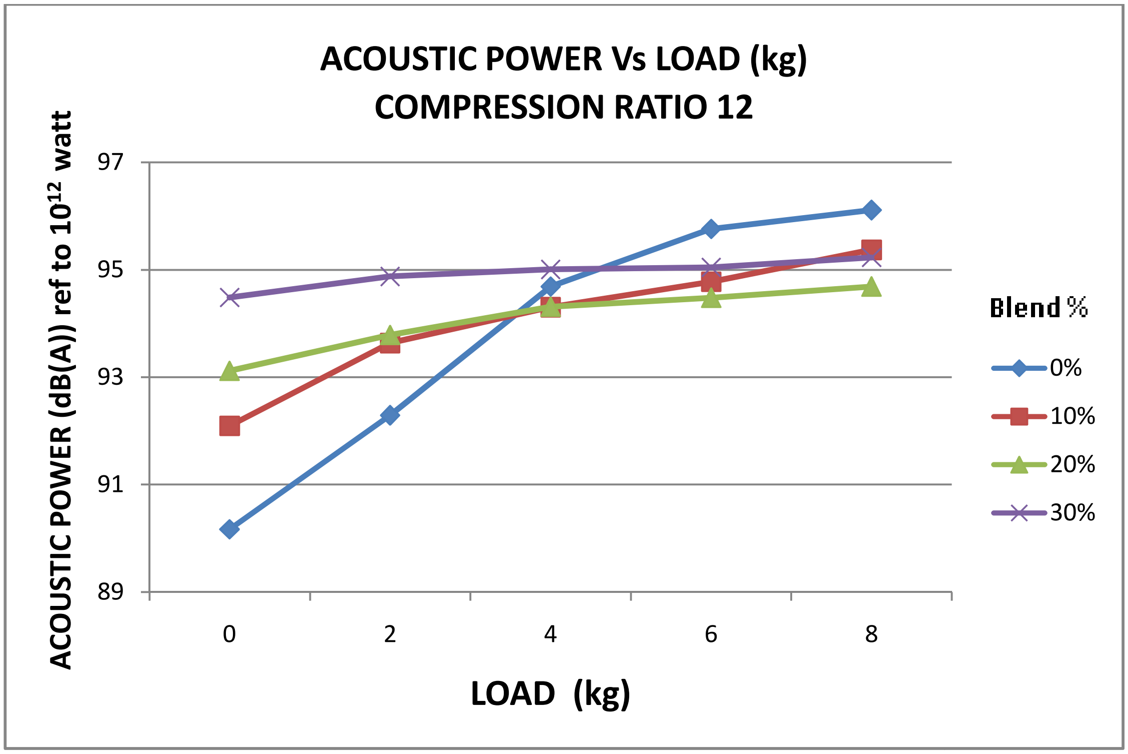

Fig 6.21 ACOUSTIC POWER AT COMPRESSION RATIO 12

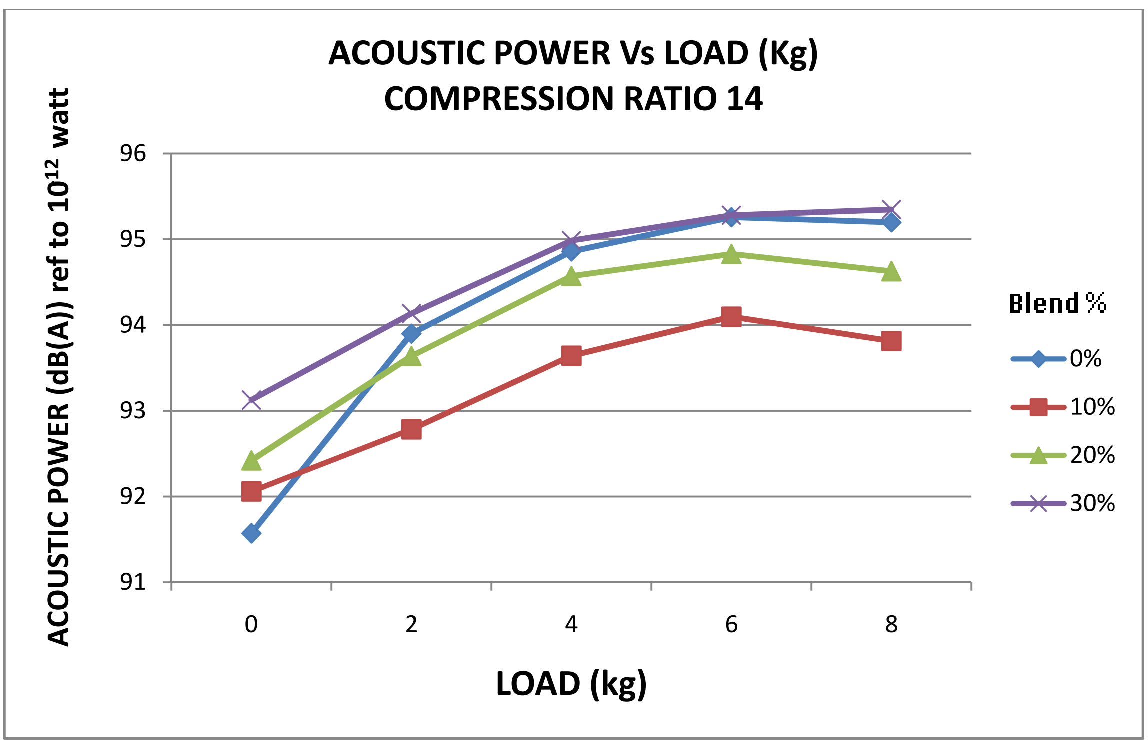

Fig 6.22 ACOUSTIC POWER AT COMPRESSION RATIO 14

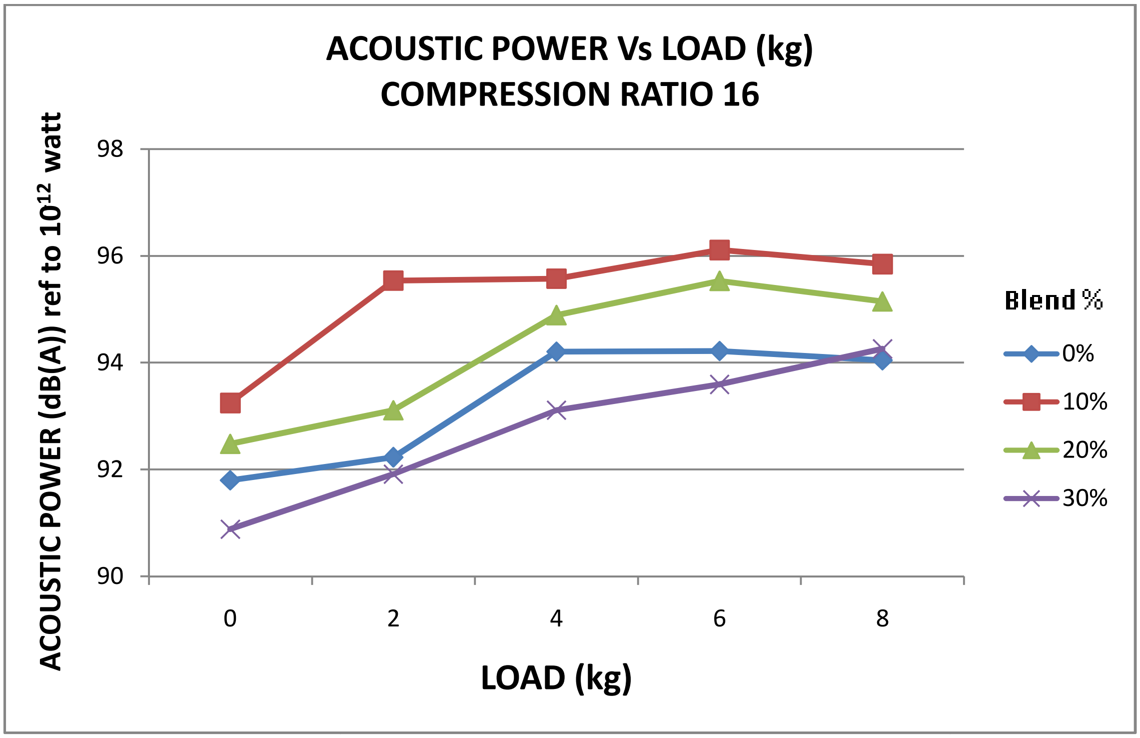

Fig 6.23 ACOUSTIC POWER AT COMPRESSION RATIO 16

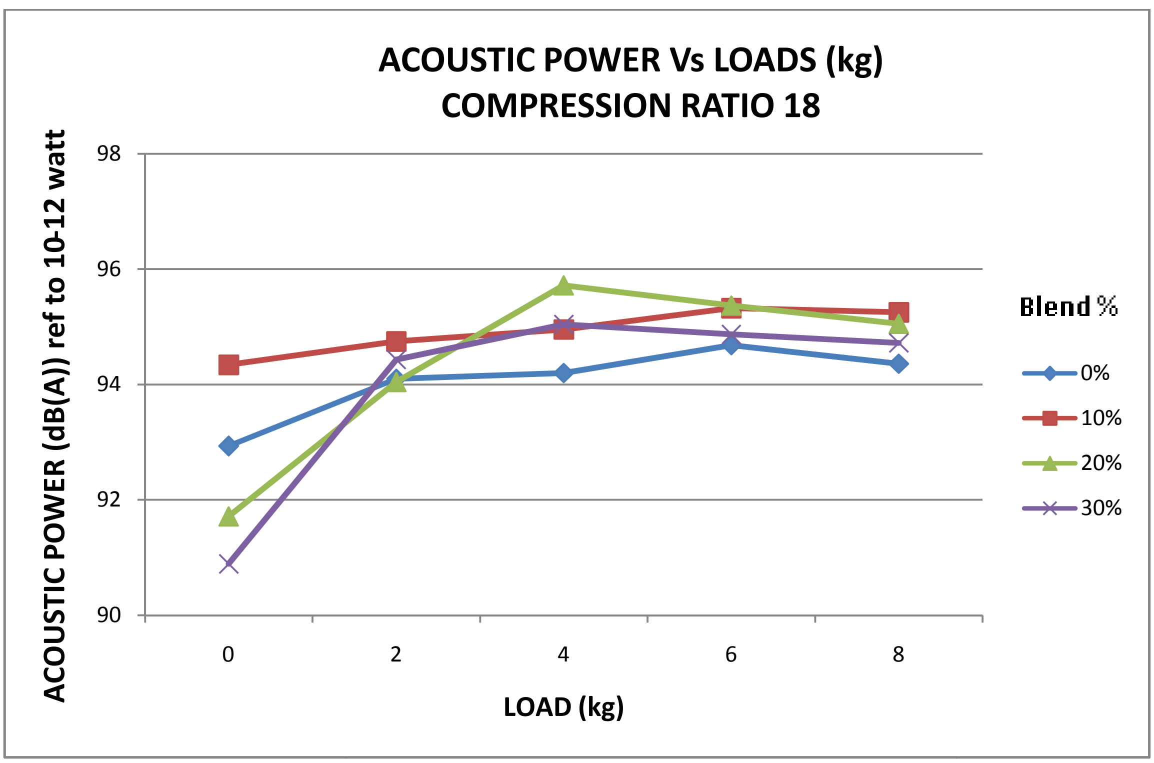

Fig 6.24 ACOUSTIC POWER AT COMPRESSION RATIO 18

6.2.1 ANALYSIS FOR ACOUSTIC POWER (dB (A)) ref to 10^{-12} WATT KEEPING LOADS FIXED AT DIFFERENT BLENDS

Fig.6.21 to 6.24 shows the Acoustic power ((dB (A)) ref to 10^{-12} watt) vs. Load (kg) graphs at different blends of Pongamia oil. The acoustic power is represented at loads 0 kg, 2 kg, 4 kg, 6 kg and 8 kg and Compression ratios 12, 14, 16 and 18. Their variation among different blends is shown in different graphs.

The value for acoustic power at different blends is calculated in Appendix-B. These are obtained by substituting values of Ls for different compression ratios and different loads in equation 1 shown in Appendix-B. All observations for acoustic power are shown in graphs above. Detailed Analysis for acoustic power is written below

- Figure 6.21 shows that for compression ratio 12 the maximum acoustic power is 96.11(dB (A)) ref to 10^{-12} watt) for pure DF at 8 kg and its comparative power for B20 as fuel is 94.69 (dB (A)) ref to 10^{-12} watt). Though the minimum acoustic power at 0 kg load for pure DF is 90.17 (dB (A)) ref to 10^{-12} watt). The curve for B20 follows linear behavior.

- Figure 6.22 represents the data for C.R.14 the maximum acoustic power is 95.35(dB (A)) ref to 10^{-12} watt) for B30 at 8 kg and its comparative power for B20 as fuel is 94.69 (dB (A)) ref to 10^{-12} watt) and minimum power is approximate 92 dB (A)) for B10, B20 at pure DF.

- Figure 6.23 for C.R. 16 overall power shows very different behavior with overall maximum power for B10 biodiesel blend.

- Figure 6.24 shows that for higher C.R like 18 the acoustic power follows almost similar pattern for B10, B20, B30 and pure DF with maximum acoustic power being 95.72(dB (A)) ref to 10^{-12} watt) for B20 at 4 kg load and minimum power is approximate 92 dB (A)) for B10, B20 at pure DF.

6.3 EFFECT OF PONGAMIA OIL BLENDS ON FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND

The value of SPL at 1-1 octave band frequency gives an idea of the maximum and minimum value at particular frequency. Frequency spectrum is formed at that point where SPL is maximum. In this case it is taken at point "B". Readings are taken at that point by changing the different compression ratios and load at different blends. This instrument gives the data for all frequencies of 1-1 octave band in software "Cesva Capture Studio". The observations for frequency spectrum are shown in APPENDIX C.

There are two major aspects to carried out in this experiment

1. To determine those frequencies at which maximum sound pressure level is achieving at different loads and at different points.
2. Second aspect of this experiment is to find out the effect on those peak frequencies in following conditions.

- B10
- B20
- B30

To find the peak frequencies experiment is carried out as given in previous chapter. Results are displayed in the form of graphs. Related data collected and written in appendix –C.

There is a wide range of frequencies in 1/1 Octave band from 32 to 16000 Hz. It is decided that data should be represent in different graphs by varying the Compression ratios and loads so that one can observe the difference in peaks at different blends on variable C.R and loads.

6.3.1 GRAPHS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 12 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

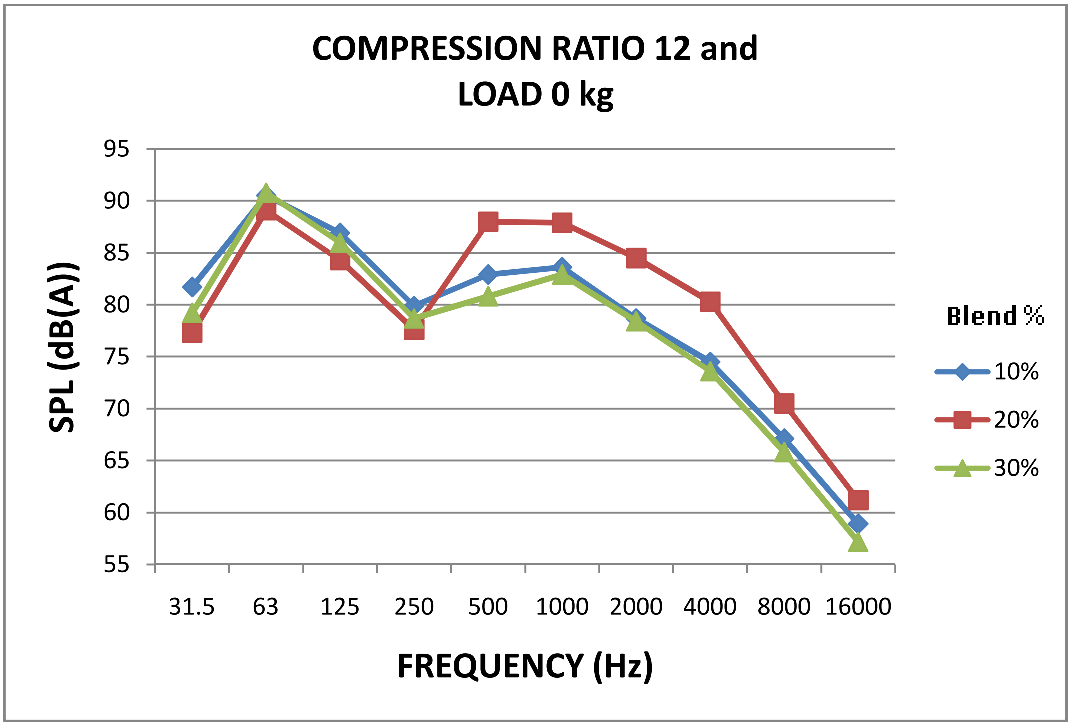

Fig 6.25 SPL VS FREQUENCY AT COMPRESSION RATIO 12 AND LOAD 0 kg

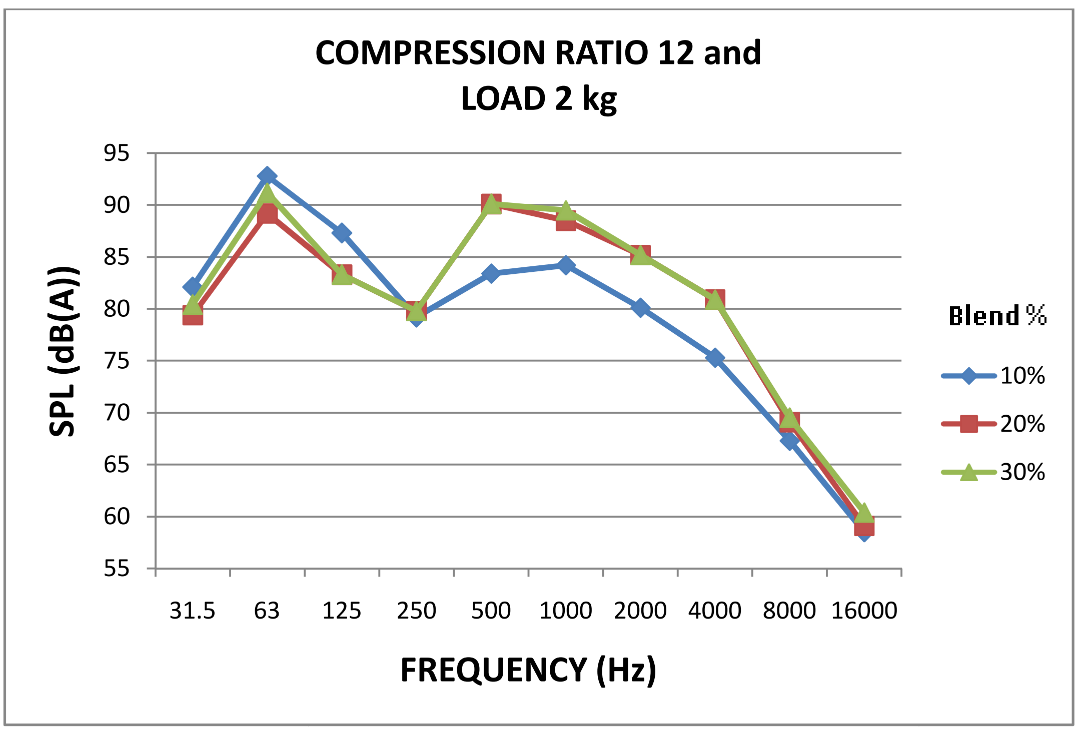

Fig 6.26 SPL VS FREQUENCY AT COMPRESSION RATIO 12 AND LOAD 2 kg

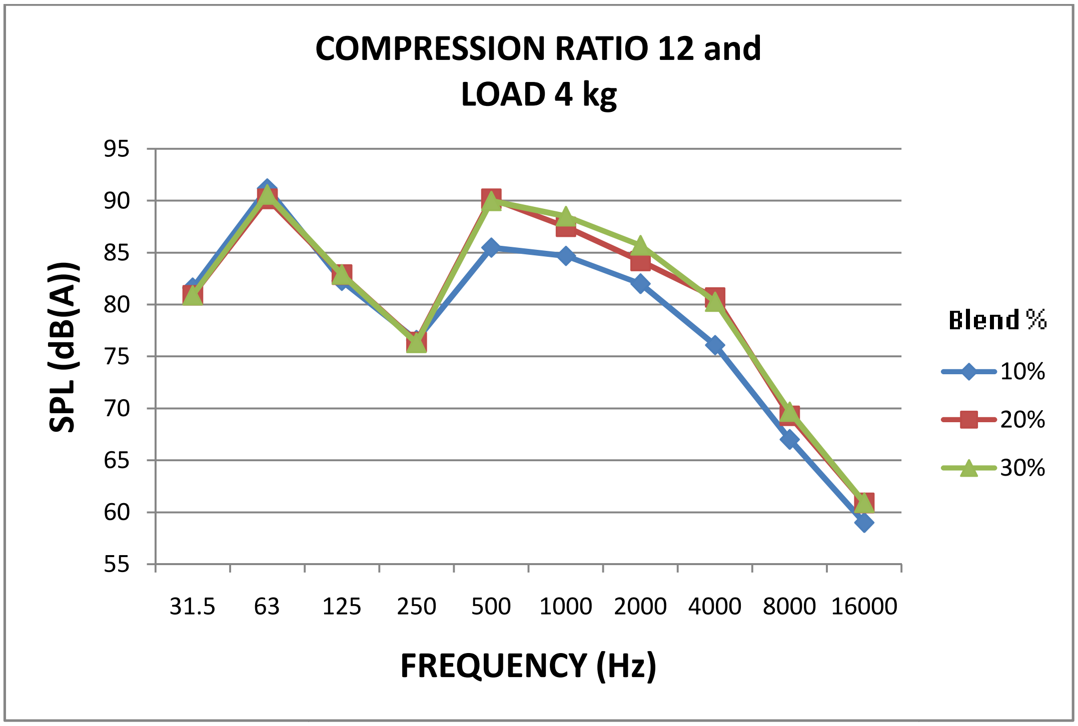

Fig 6.27 SPL VS FREQUENCY AT COMPRESSION RATIO 12 AND LOAD 4 kg

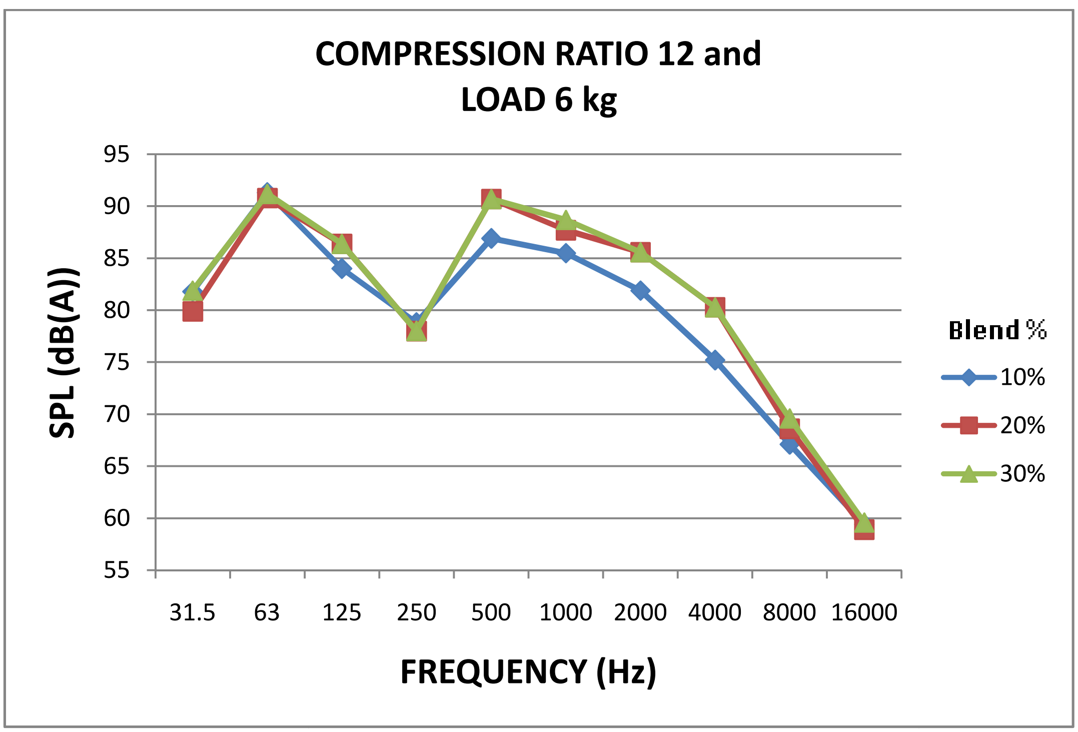

Fig 6.28 SPL VS FREQUENCY AT COMPRESSION RATIO 12 AND LOAD 6 kg

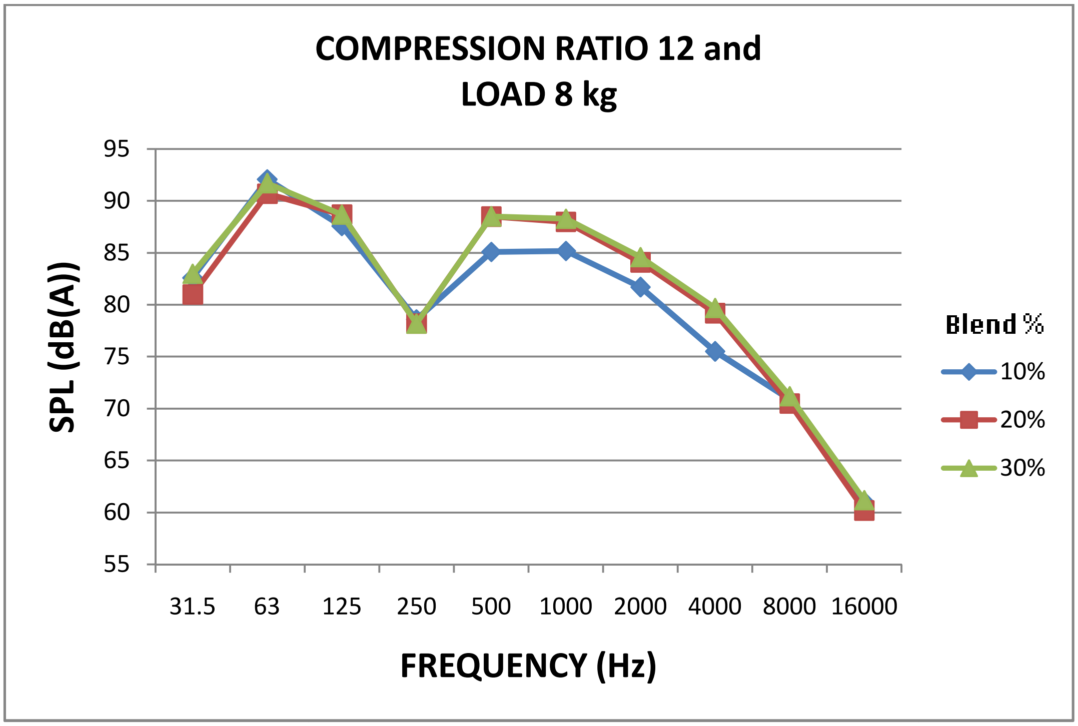

Fig 6.29 SPL VS FREQUENCY AT COMPRESSION RATIO 12 AND LOAD 8 kg

6.3.2 ANALYSIS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 12 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

The figures from 6.25 to 6.29 represent the comparison between Sound pressure level and frequency at a particular load and compression ratio for B10, B20 and B30 blends of Pongamia Oil. In these graphs Load varies from 0 kg to 8 kg for Compression ratio 12.

- In fig 6.25 it is observed that for C.R 12 and 0 kg load at low frequency range of 63 to 250 Hz SPL for B20 is less SPL for B30 and B10 with peak at 63 Hz for all the three blends. One can also observe that for higher frequency range of 500 Hz and above SPL for B20 is more than B10 and B30.

- In Fig 6.26 it is observed that for C.R 12 and 2 kg load at low frequency range of 63 to 250 Hz SPL for B20 is less than SPL for B30 and B10 with peak frequency at 63 Hz for B10, B20 and B30. But for higher frequency range of 500 Hz and above SPL for B10 is less than B20 and B30.

- In Fig 6.27 it is observed that for C.R 12 and 4 kg load for low frequency range of 63 to 250 Hz SPL for B20, B30 and B10 follow similar pattern. Peak frequency occurs at 63 Hz.

- In Fig 6.28 it is observed that for C.R 12 and 6 kg load at low frequency range of 63 to 250 Hz SPL for B20, B30 and B10 follow similar pattern with peak frequency occurring at 63 Hz.

- In Fig 6.29 it is observed that for C.R 12 and 8 kg load behavior for frequency range of 63 to 250 Hz SPL for B20, B30 and B10 follow similar pattern as in fig 6.28 with peak frequency occurring at 63 Hz for all three blends.

6.3.3 GRAPHS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 14 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

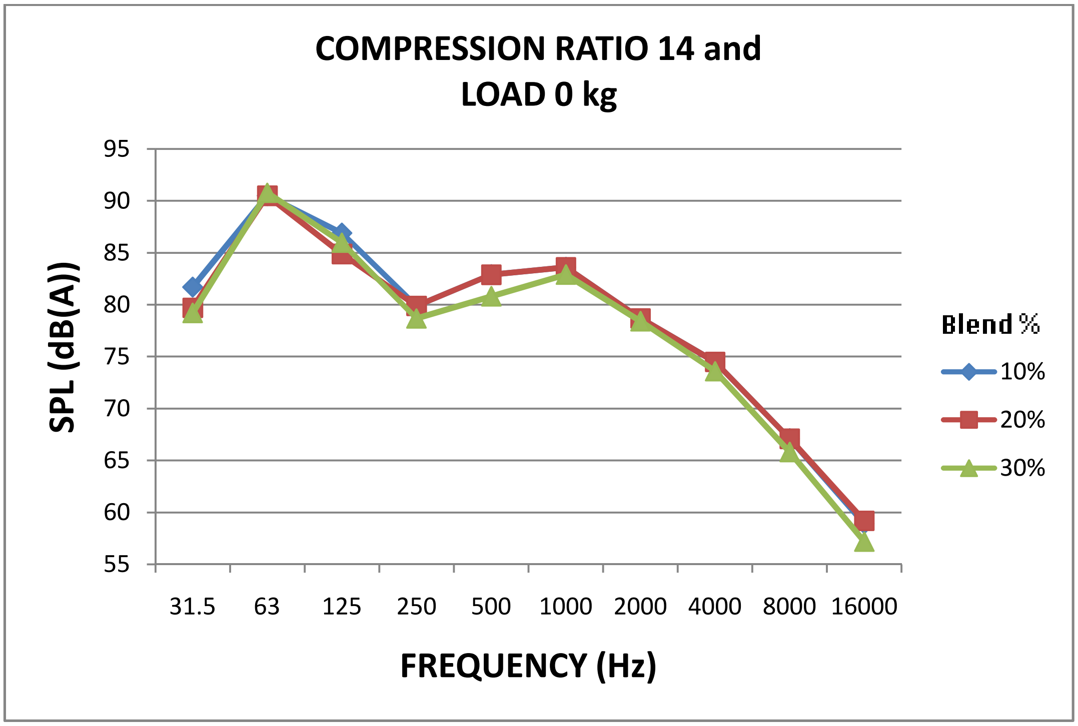

Fig 6.30 SPL VS FREQUENCY AT COMPRESSION RATIO 14 AND LOAD 0 kg

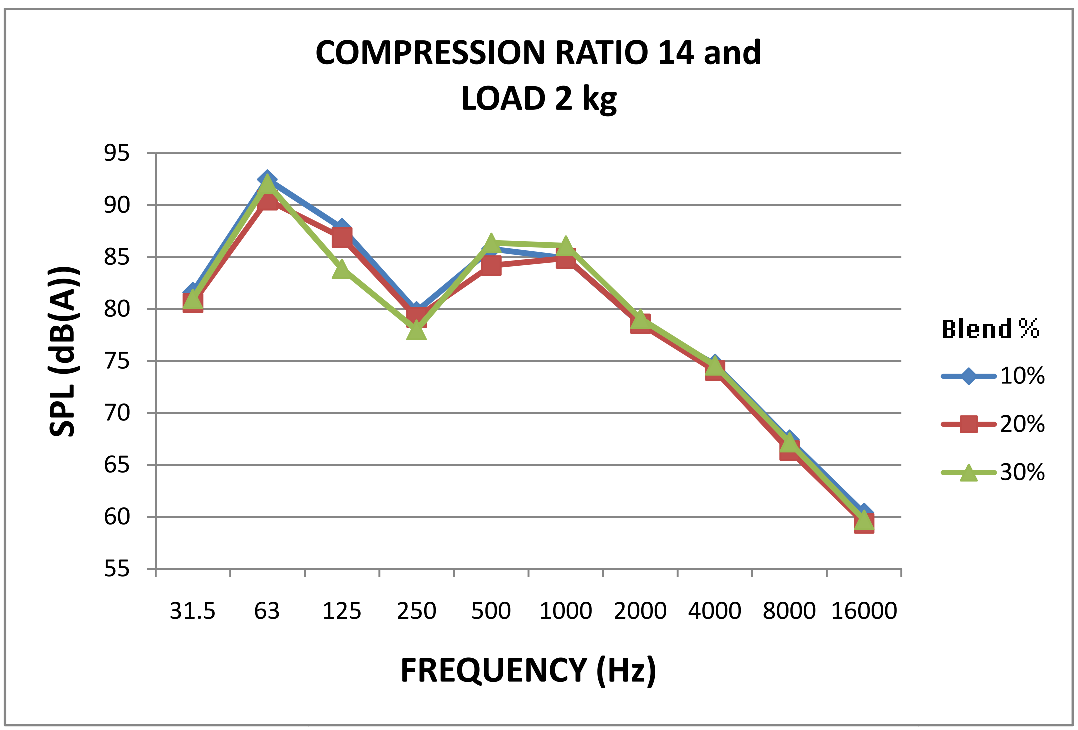

Fig 6.31 SPL VS FREQUENCY AT COMPRESSION RATIO 14 AND LOAD 2 kg

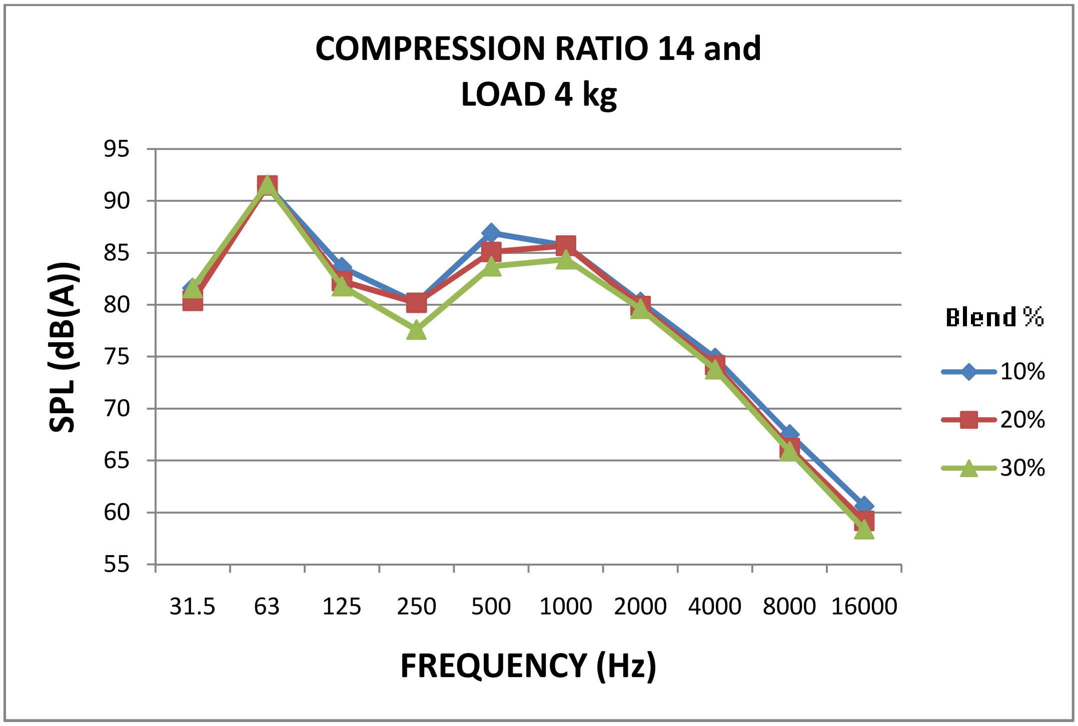

Fig 6.32 SPL VS FREQUENCY AT COMPRESSION RATIO 14 AND LOAD 4 kg

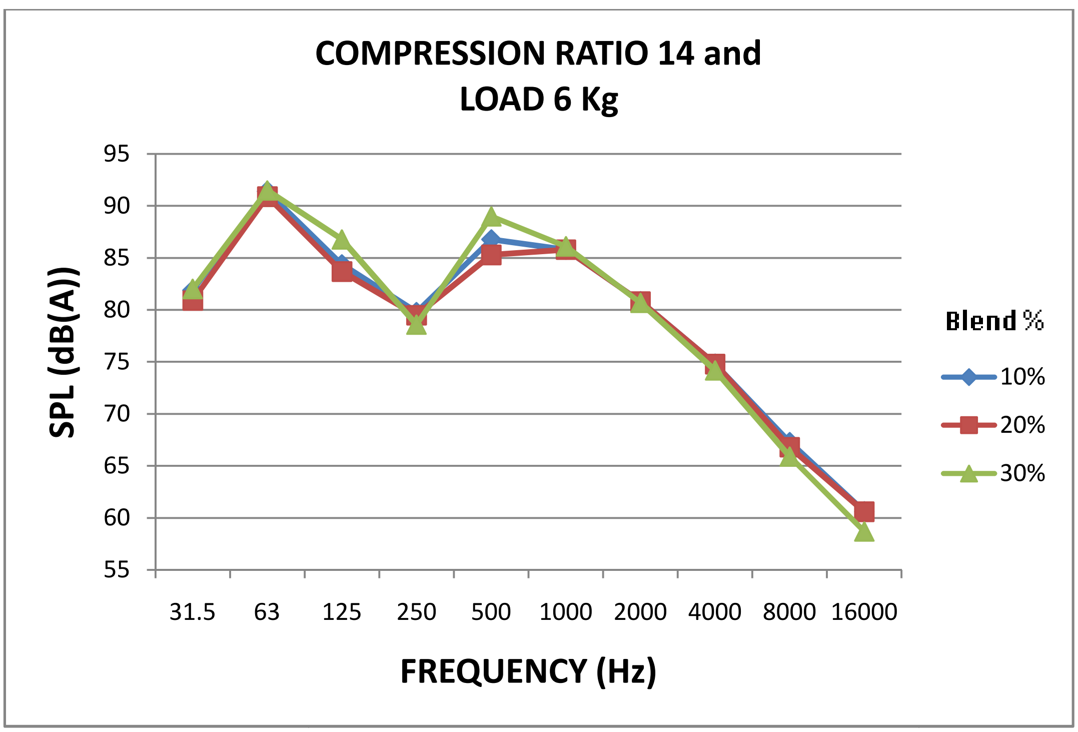

Fig 6.33 SPL VS FREQUENCY AT COMPRESSION RATIO 14 AND LOAD 6 kg

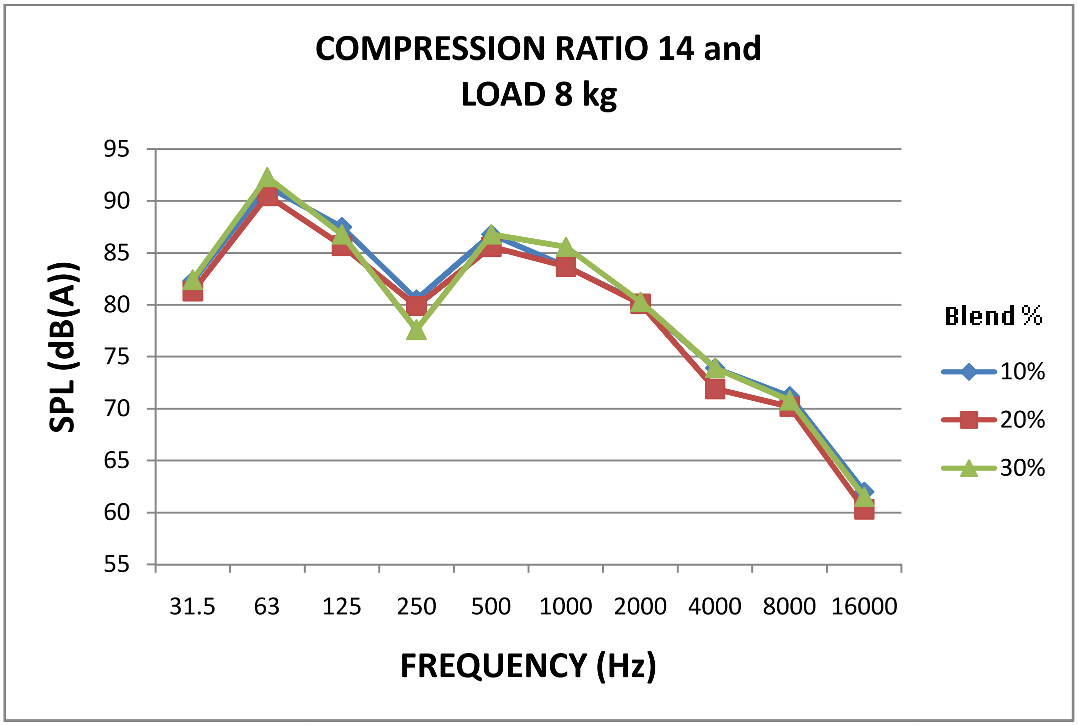

Fig 6.34 SPL VS FREQUENCY AT COMPRESSION RATIO 14 AND LOAD 8 kg

6.3.4 ANALYSIS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 14 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

The figures from 6.30 to 6.34 represent the comparison between Sound pressure level and frequency at a particular load and compression ratio for B10, B20 and B30 blends of Pongamia Oil. In these graphs Load varies from 0 kg to 8 kg for Compression ratio 14.

- In fig 6.30 it is observed that for C.R 14 and 0 kg load at low frequency range of 63 to 250 Hz SPL for B20, B30 and B10 is almost equal with peak at 63 Hz for all the three blends. After obtaining a decline from 63 Hz to 250 Hz, SPL again inclines upto 1000 Hz and further it declines for higher frequency range till 16000 Hz.

- In Fig 6.31 one can observe that for C.R 14 and 2 kg load for peak frequency of 63 Hz SPL for B20 is less than SPL for B30 and B10 with peak frequency at 63 Hz for B10, B20 and B30 blends of Pongamia oil.

- In Fig 6.32 it is observed that for C.R 14 and 4 kg load for low frequency range of 63 to 250 Hz SPL for B20, B30 and B10 follow similar pattern. Peak frequency occurs at 63 Hz.

- In Fig 6.33 it is observed that for C.R 14 and 6 kg load at low frequency range of 31.5 to 63 Hz SPL for B20, B30 and B10 follow similar pattern with peak frequency occurring at 63 Hz. For 125 Hz SPL for B20 and B10 is less than SPL for B30.

- In Fig 6.34 it is observed that for C.R 14 and 8 kg load behavior for low frequencies of 31.5 Hz and 63 Hz SPL for B20 is less than SPL for B30 and B10.Peak frequency occurs at 63 Hz.

6.3.5 GRAPHS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 16 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

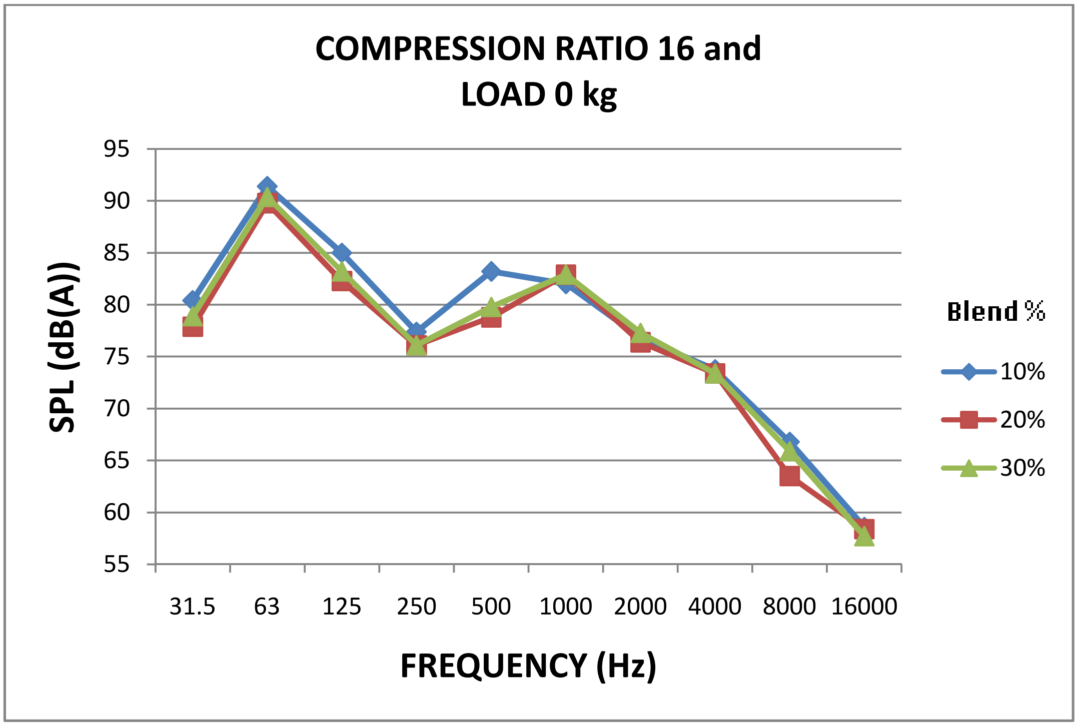

Fig 6.35 SPL VS FREQUENCY AT COMPRESSION RATIO 16 AND LOAD 0 kg

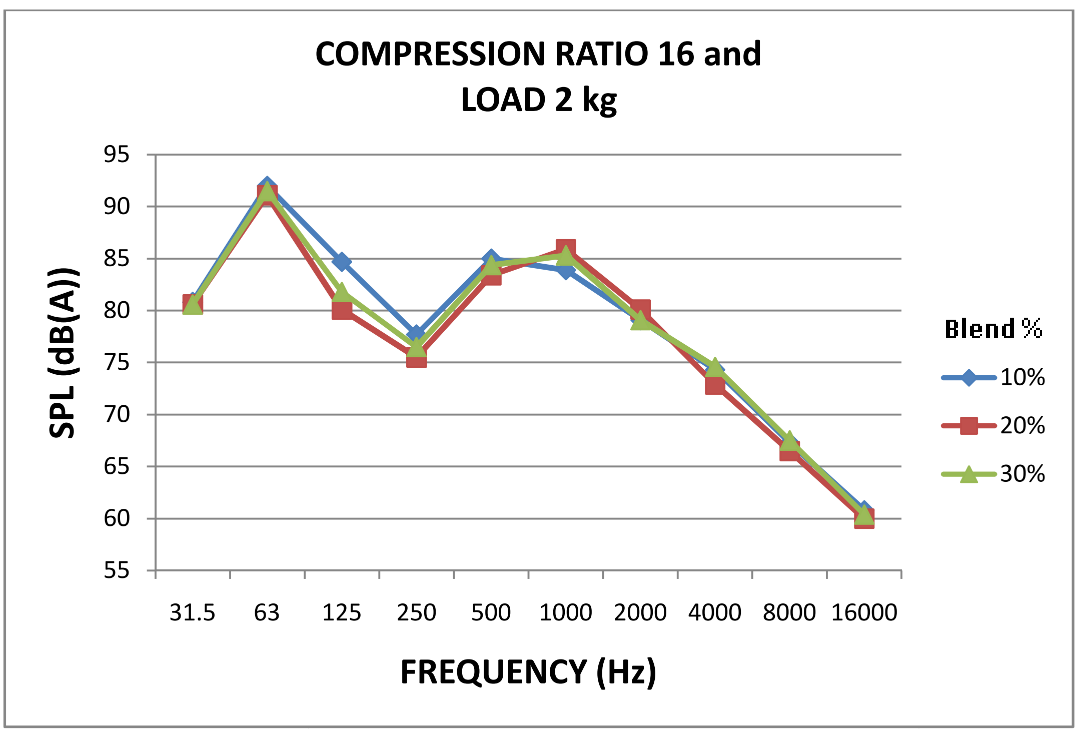

Fig 6.36 SPL VS FREQUENCY AT COMPRESSION RATIO 16 AND LOAD 2 kg

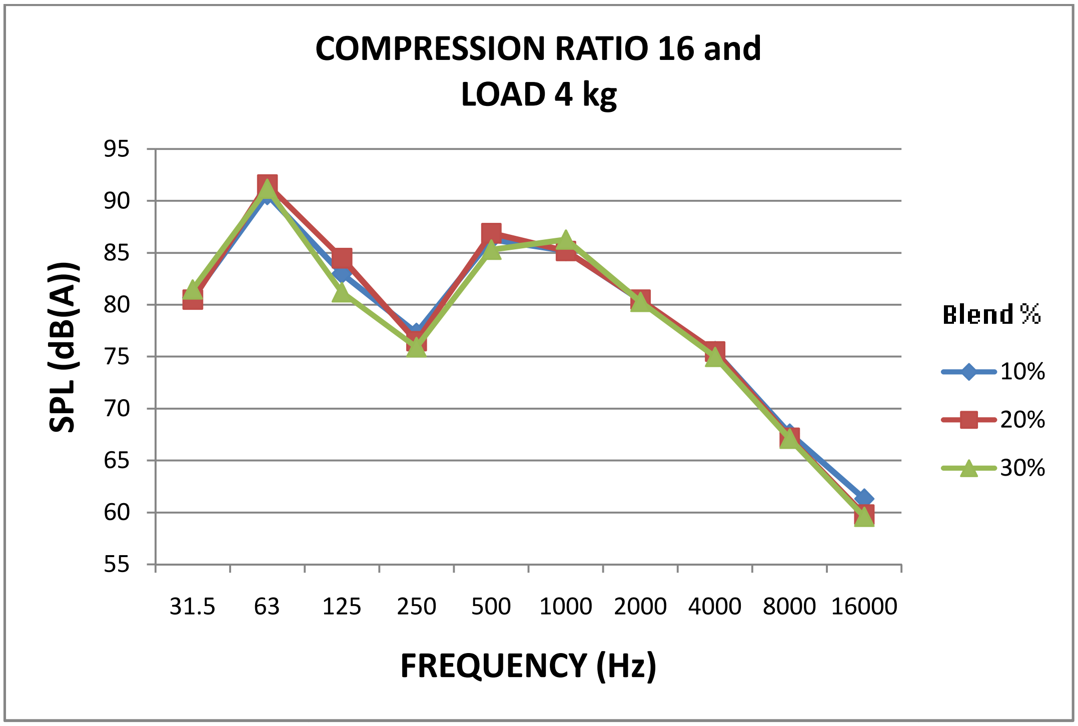

Fig 6.37 SPL VS FREQUENCY AT COMPRESSION RATIO 16 AND LOAD 4 kg

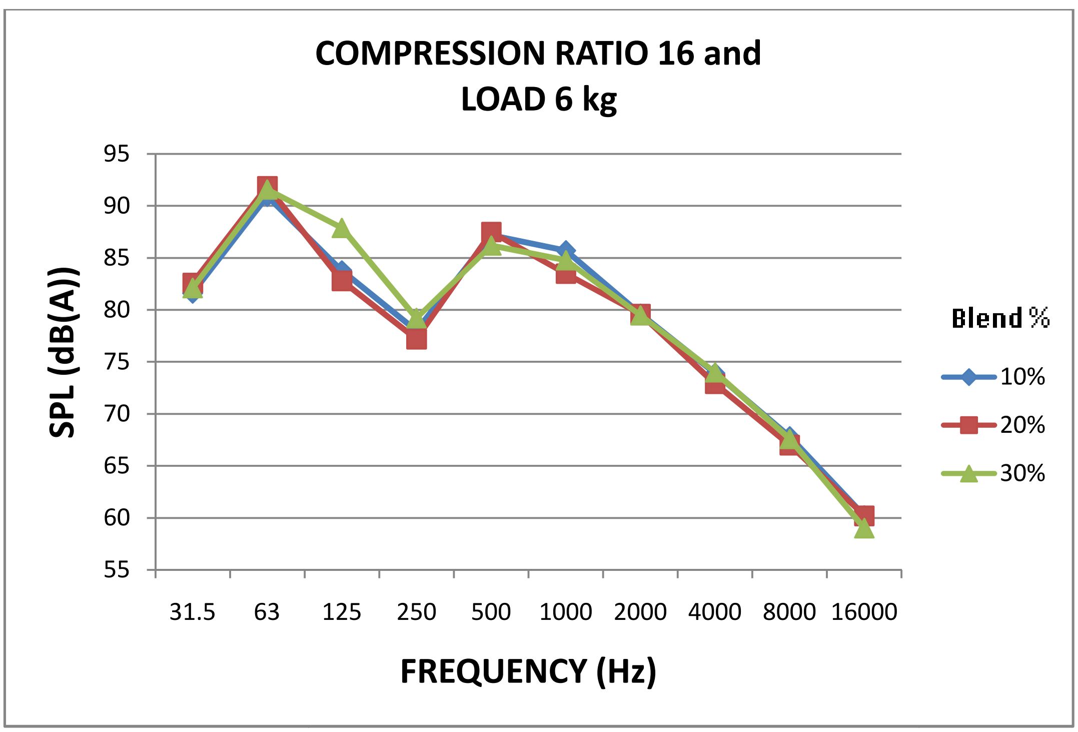

Fig 6.38 SPL VS FREQUENCY AT COMPRESSION RATIO 16 AND LOAD 6 kg

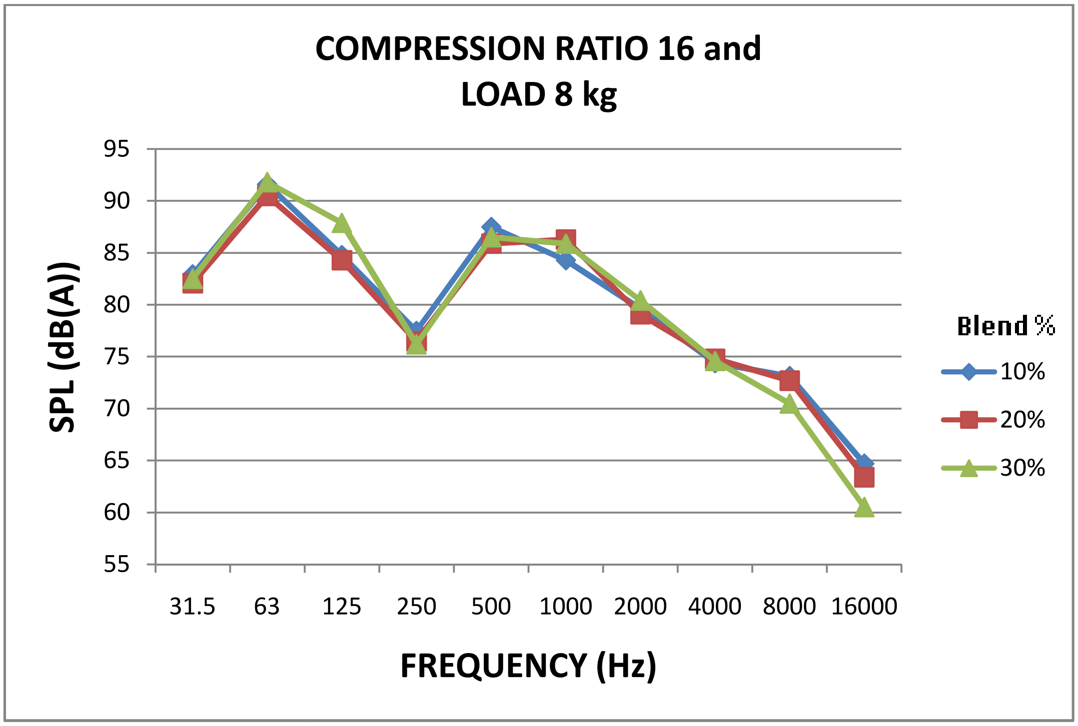

Fig 6.39 SPL VS FREQUENCY AT COMPRESSION RATIO 16 AND LOAD 8 kg

6.3.6 ANALYSIS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 16 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

The figures from 6.35 to 6.39 represent the comparison between Sound pressure level and frequency at a particular load and compression ratio for B10, B20 and B30 blends of Pongamia Oil. In these graphs Load varies from 0 kg to 8 kg for Compression ratio 16.

- In fig 6.35 one can observe that for C.R 16 and 0 kg load peak frequency occurs at 63 Hz for all the three blends. Maximum SPL occurs for B10 approximate 92 dB(A).

- In Fig 6.36 it is observed that for C.R 16 and 2 kg load at frequency of 63 Hz SPL for B20 is almost equal for SPL of B30 and B10 fuel with peak frequency also occurring at 63 Hz. But for higher frequency range of 63-250 Hz and above SPL for B20 is less than B10 and B30.

- In Fig 6.37 for C.R 16 and 4 kg load at frequency of 63 Hz SPL for B20 is almost equal for SPL of B30 and B10 fuel with peak frequency also occurring at 63 Hz. But for higher frequency range of 63-250 Hz and above SPL for B20 is less than B10 and B30.

- In Fig 6.38 it is observed that for C.R 16 and 6 kg load at frequency range of 63-250 Hz SPL for B20 is almost equal for SPL of B10 but SPL for B30 is more than SPL for B20 and B10 for frequency range of 63-250 Hz.

- In Fig 6.39 it is observed that for C.R 16 and 8 kg load behavior for frequency range of 63 to 250 Hz SPL for B20 is almost equal for SPL of B10 but SPL for B30 is more than SPL for B20 and B10 for frequency range of 63-250 Hz with peak frequency occurring at 63 Hz for all three blends. For higher very frequency range i.e. 4000 Hz and above, SPL for B30 is less than B20 and B10.

6.3.7 GRAPHS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 18 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

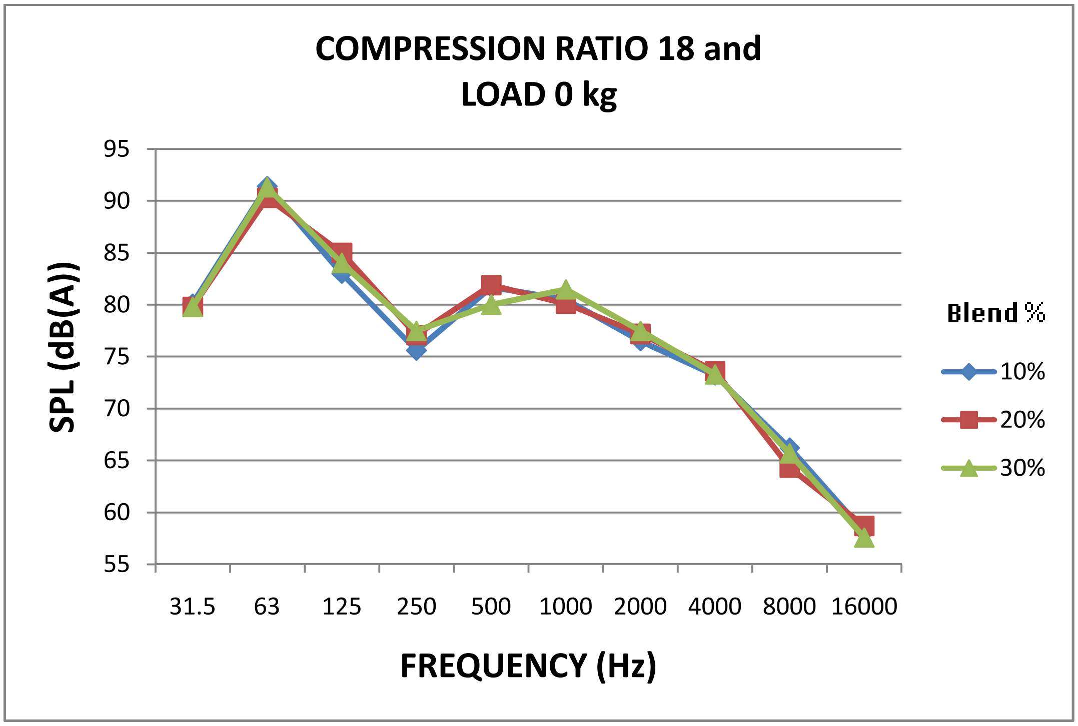

Fig 6.40 SPL VS FREQUENCY AT COMPRESSION RATIO 18 AND LOAD 0 kg

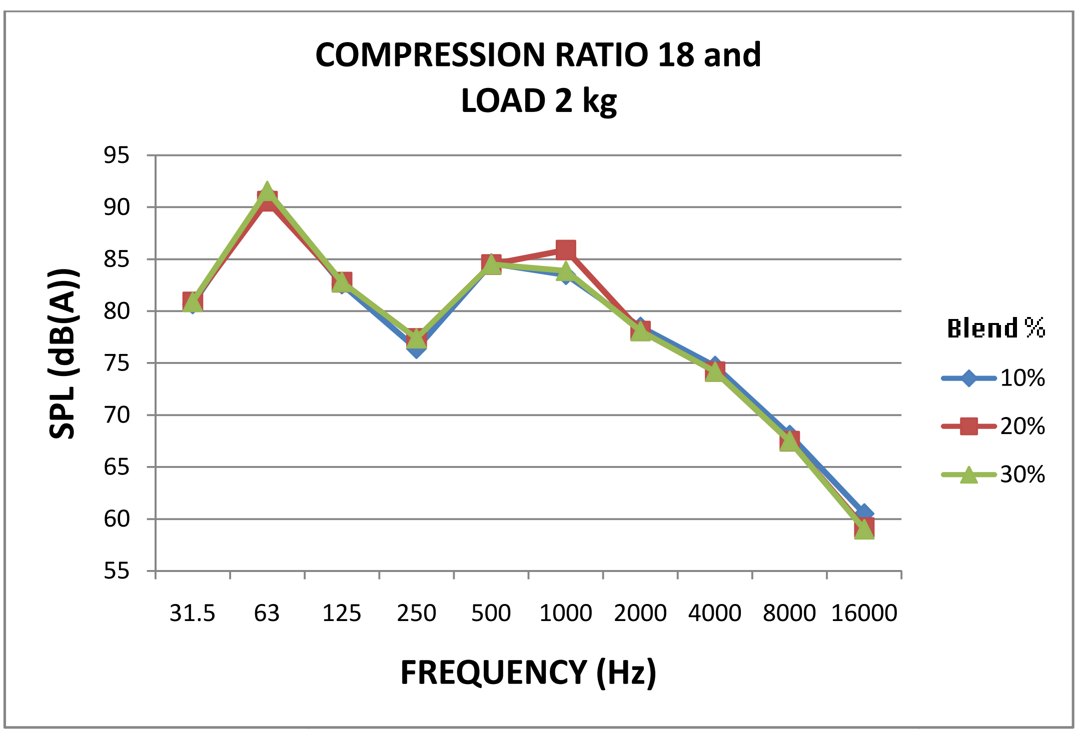

Fig 6.41 SPL VS FREQUENCY AT COMPRESSION RATIO 18 AND LOAD 2 kg

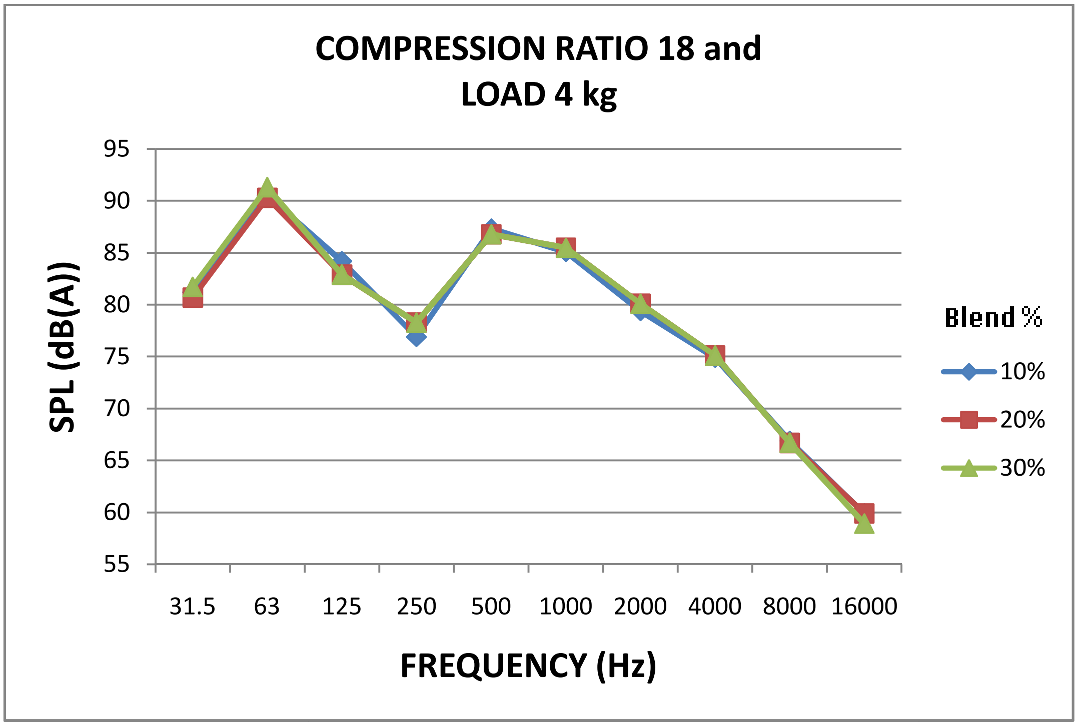

Fig 6.42 SPL VS FREQUENCY AT COMPRESSION RATIO 18 AND LOAD 4 kg

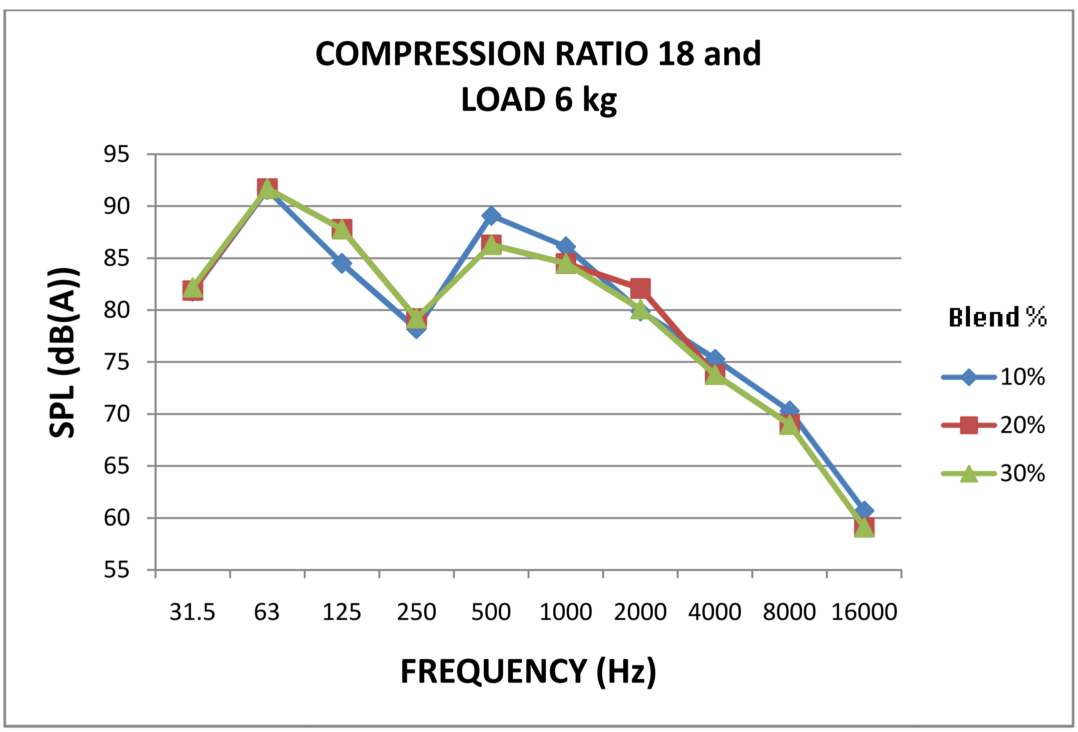

Fig 6.43 SPL VS FREQUENCY AT COMPRESSION RATIO 18 AND LOAD 6 kg

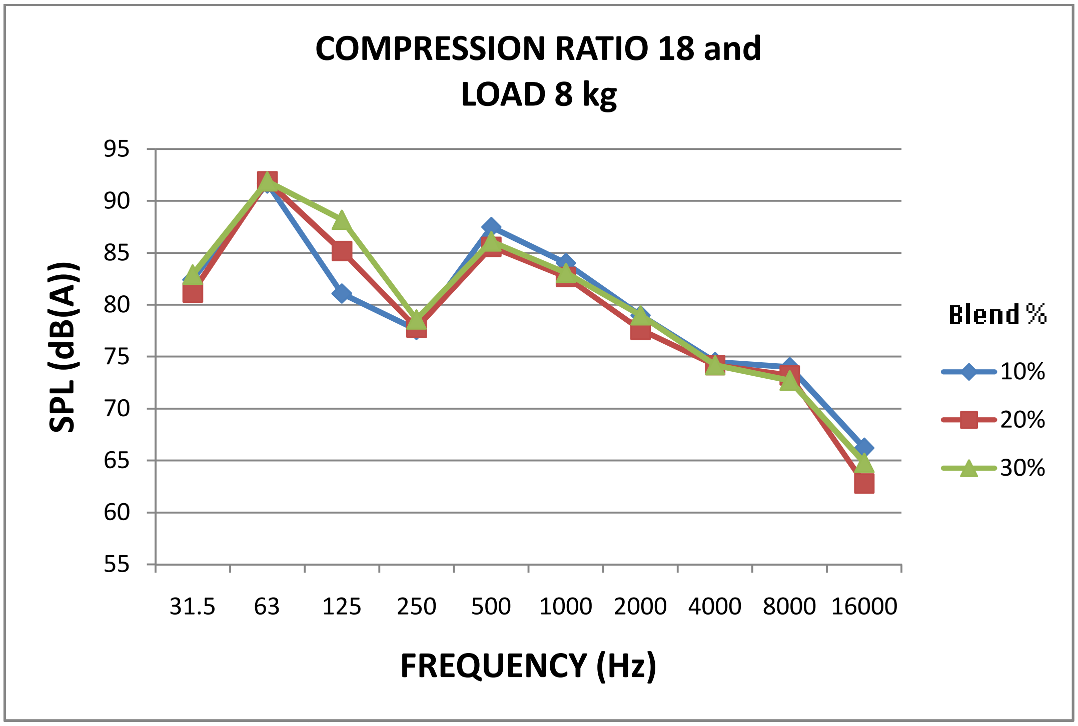

Fig 6.44 SPL VS FREQUENCY AT COMPRESSION RATIO 18 AND LOAD 8 kg

6.3.8 ANALYSIS FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND FOR C.R. 16 AND DIFFERENT LOADS (0, 2, 4, 6, 8) kg

The figures from 6.40 to 6.44 represent the comparison between Sound pressure level and frequency at a particular load and compression ratio for B10, B20 and B30 blends of Pongamia Oil. In these graphs Load varies from 0 kg to 8 kg for Compression ratio 18.

- In fig 6.40 it is observed that for C.R 18 and 0 kg load at low frequency of 63 Hz SPL for B20 is less than SPL for B30 and B10 with peak occurring at 63 Hz for all the three blends. One can also observe that further on for frequency range of 63 to 250 Hz frequency SPL decreases, then it increases till 500 hz, and further it decreases till 16000 Hz.

- In Fig 6.41 one can observe that for C.R 18 and 2 kg load all the B20, B30 and B10 follow similar pattern expect at 1000 Hz where SPL for B20 is more than SPL of other 2 blends. Peak frequency occurs at 63 Hz for all the three B10, B20 and B30 blends.

- In Fig 6.42 it is observed that for C.R 18 and 4 kg load frequency spectrum Obtained for SPL is almost similar. Peak frequency occurs at 63 Hz for all the three fuels.

- In Fig 6.43 it is observed that for C.R 18 and 6 kg load at low frequency of 63-250 Hz SPL for B10 is less than SPL for B30 and B20 with peak occurring at 63 Hz for all the three blends. But one can observe that for frequency range of 250 to 4000 Hz for B10 is more than SPL for B30 and B20 blends.

- In Fig 6.44 one can observe that for C.R 18 and 8 kg load at low frequency of 63-250 Hz SPL for B20 is less than SPL for B30 but more than SPL for B10 blend, with peak frequency occurring at 63 Hz for all the three blends. But one can observe that for frequency range of 250 Hz and above SPL for B20 is less than SPL for B10 and B30 Pongamia oil blends.

6.4 SUMMARIZED RESULTS AND DISCUSSIONS:

An Analysis of collected data indicates the following results:

1. **Effect of blends on sound pressure level at point A:** The value of sound pressure level at point A varies from 82.9 dB(A) to 90.9 dB(A) for 0 to 8 kg load and for C.R 12 to 18 for B10, B20, B30 and pure diesel fuel. At point A the worst point obtained is for pure diesel having SPL of 90.9 dB(A) at C.R.12 and load 8 kg and best point obtained is also for pure diesel having SPL of 82.9 dB(A) at C.R.12 and load 0 kg. For B10 it varies from 83.2 dB(A) to 87.6 dB(A) for all the combinations. For B20 biodiesel blend it varies from 83 dB (A) to 87.6 dB(A). For B30 pongamia oil blend it varies from 82.8 dB(A) to 90.2 dB(A). For pure diesel fuel it varies from 82.9 dB (A) to 90.9 dB (A). In case of point A one can observe that the value of SPL increases till load 6 kg then it falls down till 8 load kg for almost all the combinations. The value of SPL for all the three blends tested for point A is less than value obtained for pure diesel for almost all the attempted combinations of loads and compression ratios.

2. **Effect of blends on sound pressure level at point B :** The value of sound pressure level at point B varies from 83.6 dB(A) to 95.1 dB(A) 0 to 8 kg load and for C.R 12 to 18 for B10, B20, B30 and pure diesel fuel. For point B the worst point obtained is for pure diesel having SPL of 95.1 dB(A) at C.R.14 and load 8 kg but the best point obtained in this case for B20 having SPL of 83.6 dB(A) at C.R.16 and load 0 kg. For B10 it varies from 83.9 dB(A) to 89.5 dB(A) for all the combinations. For B20 biodiesel blend it varies from 83.6 dB (A) to 89.2 dB(A). For B30 pongamia oil blend it varies from 84.8 dB (A) to 92.8 dB(A). For pure diesel fuel it varies from 86.6 dB(A) to 95.1 dB(A). One can evaluate that value of SPL for all the three blends tested for point B is less than value obtained for pure diesel for almost all the attempted combinations.

3. **Effect of blends on sound pressure level at point C :** The value of sound pressure level at point C varies from 84.5 dB(A) to 96.5 dB(A) for 0 to 8 kg load and for C.R 12 to 18 for B10, B20, B30 and pure diesel fuel. At point C the worst point obtained is for pure diesel having SPL of 95.1 dB(A) at C.R.18 and load 6 kg and the best point obtained in this case for B10 having SPL of 84.5 dB(A) at C.R.18 and load 0

kg. For B10 in dB(A) varies from 84.5 dB(A) to 88.9 dB(A) for all the combinations of loads and compression ratios. For B20 biodiesel blend it varies from 85.2 dB(A) to 90.8 dB(A). For B30 pongamia oil blend it varies from 84.6 dB (A) to 92.7 dB(A). For pure diesel fuel it varies from 88.7 dB (A) to 96.5 dB (A). The value of SPL at different blends for point C is considerably less than value obtained for pure diesel for almost all the attempted combinations of loads and compression ratios.

4. **Effect of blends on sound pressure level at point D :** The value of sound pressure level at point D varies from 84.8 dB(A) to 94.7 dB(A) for 0 to 8 kg load and for C.R 12 to 18 for B10, B20, B30 blends of pongamia oil and for pure diesel fuel. For point D the worst point obtained is for pure diesel having SPL of 94.7 dB(A) at C.R.18 and load 6 kg and the best point obtained in this case is for B10 having SPL of 84.8 dB(A) at C.R.18 and load 0 kg. For B10 SPL in dB(A) varies from 84.8 dB(A) to 88.9 dB(A) for all the attempted combinations of loads and compression ratios. For B20 biodiesel blend it varies from 85.1 dB(A) to 90.0 dB(A). For B30 pongamia oil blend it varies from 85.8 dB (A) to 91.2 dB(A). For pure diesel fuel it varies from 87.4 dB (A) to 94.7 dB (A). The value of SPL obtained at different blends for point D is considerably less than value obtained for pure diesel for almost all the attempted combinations of loads and compression ratios.

5. Among the 4 points surrounding the engine it is found out that the maximum sound pressure level is at point B as compared to points A, C and D and minimum SPL is for point A in comparison to points B, C and D for almost all the combinations of loads and compression ratios.

6. **Effect of blends on sound pressure level at Exhaust point :** The value of sound pressure level at exhaust varies from 82.3 dB(A) to 100.8 dB(A) for 0 to 8 kg load and for C.R 12 to 18 for B10, B20, B30 blends of pongamia oil and for pure diesel fuel. In case of exhaust the worst point obtained is for pure diesel having SPL of 100.8 dB(A) at C.R.14 and load 2 kg and the best point obtained in this case for B20 having SPL of 82.3 dB(A) at C.R.12 and load 0 kg. For B10 it varies from 86.7 dB(A) to 95.7 dB(A). For B20 pongamia oil blend it varies from 82.3 dB(A) to 94.7 dB(A). For B30 pongamia oil blend it varies from 85.8 dB (A) to 97.2 dB(A) for all

the attempted combinations. For pure diesel fuel it varies from 84.9 dB (A) to 100.8 dB(A). The value of SPL obtained for exhaust is maximum among all the attempted points where SPL was measured. The SPL obtained in case of exhaust for different blends is similar to the SPL obtained in case of pure diesel. Whereas in case of other 4 points surrounding the engine one can observe certain attenuation in SPL levels while the certain combinations of loads and C.R. were attempted using different blends.

7. **Effect of blends on Acoustic power :** The value of acoustic power varies from 90.17 dB(A) to 96.11 dB(A) for 0 to 8 kg load and for C.R 12 to 18 for B10, B20, B30 blends of pongamia oil and for pure diesel fuel. The maximum acoustic power calculated is for pure diesel having acoustic power of 96.11 dB(A) ref to(10^{-12} watt) for C.R.14 and load 8 kg as well as for B10 at for C.R. 16 and load 6 kg and the minimum acoustic power calculated is also for pure diesel having acoustic power of 90.17 dB(A) ref to(10^{-12} watt) for C.R.12 and load 0 kg. For B10 acoustic power varies from 91.71 dB(A) to 95.15 dB(A. For B20 acoustic power varies from 92.42 dB(A) to 95.15 dB(A). For B30 pongamia oil blend it varies from 90.88 dB (A) to 95.35 dB(A). For pure diesel fuel used it varies from 90.17 dB (A) to 96.11 dB (A). Among all the blends used maximum acoustic power is for B30. One can observe that acoustic power increases with increase in compression ratio as well as with increase in load in case of different blends used.

8. **Effect of blends on frequency spectrum in 1-1 octave band:** The value of SPL in frequency spectrum lies between 57.2 dB(A) to 92.8 dB(A) for all the B10, B20 and B30 blends tested on the engine with peak frequency occurring at 63 Hz for all the different combinations used.

   - B10: For B10 value of SPL in frequency spectrum varies from 57.3 dB(A) to 92.8 dB(A). Its maximum value 92.8 dB(A) occurs at peak frequency of 63 Hz for 2 kg load, C.R. 12. After a sudden fall in frequency till 250 Hz one can observe a smaller second peak at 500 Hz for all the attempted combinations of compression ratios and loads.

- B20: For B20 value of SPL in frequency spectrum varies from 58.4 dB(A) to 91.9 dB(A). Its maximum value 91.9 dB(A) is obtained at peak frequency of 63 Hz for 0 kg load, C.R. 18. As observed from fig 6.25 to 6.44 the pattern for B20 frequency spectrum is similar for B20 also. After a sudden fall in frequency till 250 Hz one can observe a smaller second peak at 500 Hz for all the attempted combinations of compression ratios and loads for the case of B20.

- B30: For B30 value of SPL in frequency spectrum varies from 57.2 dB(A) to 92.1 dB(A). Its maximum value of 92.1 dB(A) occurs at peak frequency of 63 Hz for 2 kg load, C.R. 14. The pattern followed by frequency spectrum of B30 is similar to pattern followed by B10 and B20. After a sudden fall in frequency till 250 Hz one can observe a smaller second peak at 500 Hz for all the attempted combinations of compression ratios and loads.

- With peak frequency occurring at 63 Hz and its corresponding maximum SPL of approximately 92 dB(A) for all the combinations attempted one can conclude that blends do not have much effect on frequency spectrum.

CHAPTER 7

CONCLUSION AND SCOPE FOR FUTURE WORK

7.1. CONCLUSION

The objective of the present work is to collect the data based on two parameters i.e. compression ratios and loads using different blends of Pongamia oil and compare it with pure diesel as fuel.

On the basis of present study following conclusions are drawn:

a) With increase in compression ratios SPL increases for different blends of biodiesel as well as for pure diesel fuel.

b) For all the fuels tested with increase in loads SPL increases till 6 kg for A,B,C and D points. Then one can find attenuation in SPL till load 8 kg for all the biofuels blends as well as for pure diesel.

c) Acoustic power increases as load increases for almost all the Compression ratios for different blends as well as for pure diesel fuel.

d) Peak values of sound pressure level are on lower frequency range for different combinations attempted for different blends. The frequency spectrum gets it peak frequency at 63 Hz and then falls down till 250 Hz. One can observe another smaller peak at 500 Hz.

e) Among all the 5 points A, B, C, D and Exhaust. The maximum SPL in (dB (A)) is at exhaust, so one can rate exhaust as the worst point where the values of SPL are maximum for different blends as well as for pure diesel fuel.

f) Among the 4 points A, B, C and D surrounding the engine it is found that the maximum sound pressure level is at point B because B is nearest to engine in comparison to A, C and D and minimum SPL is for point A in comparison to B, C and D because A is farthest from engine.

g) On the basis of observations of data collected it is concluded that B20 is the best blend among the three blends B10, B20 and B30 tested on CI engine. This result is only on the basis of noise study on diesel engine so when it comes other engine considerations like overall efficiency, brake mean efficiency, specific fuel consumption it may lead to different observations.

7.2. SCOPE FOR FUTURE WORK

The present work may be extended in one of the following ways. Directions in which the work can be further carried out in future are:

a) There are other biofuels used like Jatropha oil, Bioethanol, etc in the use today, using different fuels one can study the noise results and choose the most appropriate fuel for CI Engines.

b) The results can also be observed for more combination of blends B5, B15, B25 etc, and even for higher blends till B50 or even more because with crude oil depleting there are chances that engines could be run on higher percentage of blended fuels.

c) All the measurements can be repeated at least 2-3 times to cover more variations for varying parameters and to get more accurate results.

d) It is concluded from results that maximum SPL is at 31.5, 63, 126 Hz frequencies. Results can be identified better by using Intensity probe. By using the probe one can identify the points of the engine from where the maximum SPL is generated. One can identify its causes and make certain changes to attenuate the effect of SPL on engine.

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Referred Pictures from laboratory

35. **An overview of dynamometer in IC engine lab where experimentation was done**
36. **Variable compression ratio engine set up**
37. **4 points at a distance of 1 meter each, surrounding the engine boundary.**

APPENDIX

APPENDIX-A-1

OBSERVATIONS IN SPL (dB(A)) FOR VARYING COMPRESSION RATIOS AND LOADS AT 20% BLEND OF PONGAMIA OIL

TABLE NO.7.1 FOR C.R.12 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 12									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	86.5	85.8	86.1	86.3	86.3	87	86.2	86.3	86
B	86.3	86.6	87.1	87.6	88.6	88.4	86.7	88.7	87.7
C	87	87.9	88.2	88.4	88.6	87.6	88.3	88.1	89.2
D	88.3	88	88.7	88.3	88	88.7	88.7	88.5	88
EXHAUST	82	81.4	85.7	82.4	83.3	83.8	84.9	85.8	86.5
GRID POINTS									
1	83.6	83.5	84.5	84.6	84.5	84.8	84.4	84.5	83.9
2	83.9	84.7	84.5	84.8	85.6	85.9	85.5	85.6	86.1
3	84.3	83.9	84.7	86.2	86.4	86.5	86.5	85.6	86.5
4	84.7	85.1	85.1	86.1	86.3	86.6	87.4	87.1	86.9
5	84.9	86.5	85.4	86.3	86.7	86.2	86.3	88.1	86.6
6	85.2	87	85.4	86.4	86.9	86.2	86.4	88	86
7	84.9	84.7	85.3	85.5	85.6	86.4	85.6	86.4	85.8
8	85.4	85.2	86.3	86.5	86.2	86.6	86.3	87	86
9	85.7	86.4	87	87.7	87.5	86.9	87	86.6	88.8
10	85	83.8	85.8	85	84.6	85.2	85.7	85.3	86.5
11	84.4	84.1	84.4	85.5	85.3	86.9	86.1	87.3	86.6
12	83.3	85	84.8	85.9	86.4	86.3	86.3	86.2	86.2
13	87.1	86.6	87.8	87.9	88.8	87.7	88.5	88.5	88.2
14	85.8	85.9	85.3	86.8	86.6	86.7	87.6	87.5	87.4
15	86.4	86.7	86.4	88	87.2	87.7	88.7	87.2	87.9
16	84.9	85.4	85.5	85.5	85.5	86.6	85.8	86.9	86.6
17	87.7	88.4	88	88.7	88.6	85.8	88.2	88.5	88.3
Ls	**85.33**	**85.57**	**86.00**	**86.38**	**86.52**	**86.57**	**86.69**	**86.99**	**86.90**

TABLE NO.7.2 FOR C.R.12.5 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 12.5									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	83.8	84.9	85.9	86.5	86.4	87.3	84.9	87	86.4
B	86.9	85.6	87.5	88.8	88.5	88.6	86.9	87.2	87.7
C	86.2	87.1	87.2	86.9	87.4	87.5	88.5	87.8	87.8
D	86.5	87	87.2	88	88.9	87.6	87.6	87.6	87.8
EXHAUST	85.2	85.1	85.6	85.2	84.6	84.9	83.8	84.2	84.8
GRID POINTS									
1	81.5	82.9	83.8	84.3	83.9	84.3	84.1	83.9	83.9
2	83	83.8	84.4	84.7	85.4	84.6	84.6	84.9	85.4
3	83.4	84.8	84.2	84.9	85.8	86.4	86.1	86.2	84.4
4	84.4	85.2	85.2	86.1	86.2	86.7	86.7	86.7	86.2
5	84.5	85.2	84.6	85.6	85.9	84.8	85.7	85.8	86.9
6	84.8	85.5	85.8	86.8	86.9	85.2	86.5	86.2	86.4
7	83.5	84.6	84.7	84.9	85.6	85.6	85.8	85.8	85.3
8	84.4	85.2	85.7	85.6	85.9	86	85.9	85.8	85.4
9	86.2	86	86.3	86.9	85.6	86.4	85.7	86.9	87.5
10	83.7	83.1	84.3	83.7	84.8	85	83.9	83.9	85.6
11	83.9	84.7	84.3	84.8	85.3	85.8	85.7	85.7	85.4
12	84.4	85.5	83.5	85.1	84.3	84	86.1	87	83.5
13	85.8	86.1	87.2	88.2	87.7	87.7	87.8	87.4	88
14	84.1	86.7	86	86.2	86.7	85.7	87.6	88.6	88
15	85.4	87	86.8	87.8	87.6	85.9	87.2	87.1	87.6
16	84	85	84.1	85.9	84.9	85.8	84.6	85.4	84.9
17	87.7	86.5	88.7	88.1	88.1	88.8	88.3	87.8	87.7
Ls	**84.70**	**85.34**	**85.59**	**86.14**	**86.20**	**86.12**	**86.09**	**86.31**	**86.21**

TABLE NO.7.3 FOR C.R.13 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 13									
LOADS (Kg)	0	1	2	3	4	5	6	7	8
POINTS									
A	83.7	86.5	84.8	85.7	86.3	86.1	86.1	85.4	85.2
B	85.4	86.2	87.2	86.7	87.3	86.4	87.6	87	87.5
C	84.5	85.7	86.8	86.8	87.5	87.6	87.3	87.7	87.8
D	86.2	87.4	87.4	87	87.8	87.6	87.7	87.7	87.1
EXHAUST	85	85.6	86.8	86.7	87.2	85.2	88.2	89.3	91.5
GRID POINTS									
1	82.5	83.2	84.4	84	83.9	84.3	84.9	84.5	83.7
2	83.7	83.2	84.2	84.5	84.9	84.9	84.8	85.1	85.4
3	83.2	84.7	84.1	85.3	85.1	86.1	85.6	85.7	85.7
4	83.9	84.1	85	85.8	86.2	86.1	86.5	86.2	86
5	84.7	85.2	86.1	85.6	86.8	87.4	87.5	87.1	87
6	84.6	84.8	85.6	85.9	86.1	87	86.2	85.7	85.8
7	84	83.8	84.4	84.7	86	85.8	86.1	85.4	86.8
8	88.5	85.1	85.2	84.9	85.9	86.1	86.1	85.9	85.7
9	84.8	84.7	86.2	86.1	86.8	86.8	86.9	87	86.6
10	82.8	84.8	83.7	83.7	84	84.6	84.5	84.7	84.4
11	84.2	84.4	85.2	85.1	85.8	86.7	86	85.8	85.5
12	84.2	84.6	85.5	85	85	85.9	86.3	85.9	85.6
13	85.7	86.2	86.5	87.4	87.9	88.4	88	88.8	88.6
14	84.9	85.7	85.5	85	85.6	86.2	86.8	86.9	86.6
15	85.9	85.8	86.7	87.5	88.2	88.1	88.2	89	88.4
16	83.8	83.7	84.7	84.7	85.5	85.8	86	85.5	88.6
17	84.4	88.1	88	87.8	88.2	87.6	88.7	87.9	88.5
Ls	**84.57**	**85.16**	**85.64**	**85.72**	**86.27**	**86.40**	**86.64**	**86.55**	**86.73**

TABLE NO.7.4 FOR C.R.13.5 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 13.5									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	84.3	84.8	86.5	85.9	86.1	86	86.4	85.5	86.4
B	85.5	86.2	86.4	86.6	87.6	87.4	87.2	87.8	86.7
C	85.6	86.1	86.5	86.3	87.3	86.4	87.8	87.7	88
D	85.8	85.8	86.4	87.4	86.6	87.1	87.6	88.3	88.6
EXHAUST	86.2	84.3	85.2	84.6	85.9	86.7	90.1	90.5	91.5
GRID POINTS									
1	82	83.2	84.3	84.6	85	84.3	85	83.9	84.6
2	82.1	83.1	84.3	84.6	85.2	84.7	85.4	84.2	85.6
3	83	83.5	84.7	85.3	86	86	85.9	85.8	86.1
4	83.8	83	84.8	86.3	86.5	86.3	86.1	86	86.2
5	84.2	83.9	85.4	86	85.8	85.7	87	86.6	87.8
6	84	85.6	86.1	85.9	86.6	86.7	87	86.8	87.2
7	83.8	84.2	84.7	85.8	86	85.6	86.1	85.9	86.2
8	84.2	84.3	85.1	85.5	86.3	85.9	85.6	86.6	86.1
9	84.6	85.8	85.8	86.9	86.9	86.6	87.1	87	87.3
10	83.1	83.7	83.5	85.1	85.2	85	85.6	84	85.8
11	83.8	83.5	85.1	85.8	85.9	85.8	85.2	86.2	86.4
12	82.7	83.3	83.9	85.7	86.6	85.5	86.2	86	85.9
13	85	83.4	86.3	87.9	88.2	87.8	88.1	88	88.2
14	84.8	84.9	84.8	85.6	85.9	85.7	85.9	87	87.5
15	85.7	86.7	86.3	86.6	87.6	87.9	88.2	87.1	87.2
16	83.8	84.3	84.5	85.5	86.2	84.8	86.4	86.5	86.9
17	87.3	86.2	85.3	86.4	87.3	88.1	88.4	88	88.5
Ls	**84.33**	**84.54**	**85.27**	**85.92**	**86.40**	**86.18**	**86.74**	**86.61**	**87.03**

TABLE NO.7.5 FOR C.R.14 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 14									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	85.9	84.7	85.9	86.3	87.6	86.4	86.2	86.3	86.3
B	87.3	87.1	87.8	87.6	87.8	88.3	87.7	88	88
C	87.2	86.7	87.4	88.4	88.8	88.3	88.6	88.1	87.8
D	86.6	87.9	87	87.2	87.9	88.7	87.7	87.7	87.6
EXHAUST	84.7	84.9	85.3	88.5	88.5	87.6	89.5	89.5	90.1
GRID POINTS									
1	81.9	82.9	84.6	84.2	84.9	85.1	85.5	85.1	84.6
2	81.3	83.7	84.6	85.3	85.6	85.8	85.8	85.5	86
3	82.8	83.6	84.3	84.9	86.2	86.9	86.1	86.4	86.4
4	83.6	85	86.2	85.7	86.2	87.4	87	86.8	86.3
5	83.8	86.4	85.9	86.4	86.4	87.8	87.6	87.7	87.3
6	85.2	86.1	85.4	86	86.2	86	86.1	87.9	87.6
7	83.1	85.6	84.8	85.4	86.2	86.4	86.1	85.9	86.4
8	83.8	86.9	86	86.5	85.5	86.1	86.5	85.7	85.6
9	84.7	85.8	86.2	87.4	87.3	87	87.7	86.9	86.9
10	83.5	84.1	84.3	84.2	85.5	84.9	86.1	86.4	84.8
11	84.9	85.3	84.9	86	86.1	86.5	86.6	85.9	86.5
12	83	84.3	84.3	85.4	85.4	85.6	86.1	86.2	85.7
13	85.9	86.8	87.6	87.7	88.6	88.8	88.4	89.5	88.2
14	85.3	85.9	86.1	87.1	86.4	86.2	87	88.1	86.2
15	85.7	87.1	87	87.5	88.2	89.1	88.6	88.7	87.9
16	84.7	85	85.3	86.5	85.9	87	86.2	85	86.4
17	87	88.7	87.8	88	88	88.6	87.8	87.8	87.8
Ls	**84.63**	**85.66**	**85.85**	**86.46**	**86.78**	**87.02**	**87.04**	**87.05**	**86.84**

TABLE NO.7.6 FOR C.R.14.5 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 14.5									
LOADS (Kg)	0	1	2	3	4	5	6	7	8
POINTS									
A	85.4	84.8	86.7	87.2	86.9	87.6	86.9	87	86.3
B	84.5	86.1	87.6	87.7	88.5	89	89	88.6	88.3
C	85.2	85.9	87.1	88.6	88.5	88.5	88.8	88.1	89.8
D	85.2	86.8	87.4	86.8	87.9	88.2	88.6	87.7	88.1
EXHAUST	86.8	87.1	87.7	88.3	89.7	91.6	92.1	91.6	91.9
GRID POINTS									
1	81.6	83.2	84.4	84.6	85.2	85.5	86	85.2	85.5
2	82.2	83.5	84.8	87.4	87.6	87.2	86.5	88.1	85.7
3	86.4	83.4	84	85.7	86.2	85.6	86.6	87.2	85.6
4	83.9	85.1	85.9	85.6	87.5	86.8	86.8	86.9	87.2
5	84	85.1	85.3	86.2	87	87.8	87.7	88.8	87.5
6	84.7	85.2	85.5	87.2	87.4	88.7	87.7	86.9	87.4
7	83.1	83.7	84.7	86	86.7	86.3	87.1	86.6	87
8	83.4	84.3	85.1	86	86.2	87.2	86.9	88.2	85.8
9	84.5	86.2	86.4	87.6	87.3	88	87.4	86.9	87.6
10	82.9	82.6	84.5	85.2	86.4	86	85.3	85	87.5
11	83.7	84.6	85.1	87.7	87.5	86.5	87.1	86.1	86.6
12	85.3	85.2	84.8	85.8	86.1	86	86.4	87.8	85.5
13	85	86.2	87.9	87.7	88.6	89.1	88.7	89.9	88.9
14	84.8	86.5	86.8	86.1	86.5	85.7	87.4	88.3	88.2
15	85.5	87.1	87.4	88.6	88.8	87.6	89.2	89.4	89.2
16	83.5	84.9	85	85.7	86.6	86.4	88.1	86.8	87
17	87.3	87.9	87.6	87.8	87.9	89.3	88.6	88.8	88.6
Ls	**84.50**	**85.25**	**85.99**	**86.80**	**87.32**	**87.48**	**87.68**	**87.72**	**87.51**

TABLE NO.7.7 FOR C.R.15 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 15									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	82.8	84.7	86.1	86.8	87.1	86.3	86.3	86.2	87
B	83.7	86.3	86.8	87.2	86.9	87.4	87.2	87.6	88
C	85.6	86.8	87.3	88.4	87.5	87.7	87.4	88.9	88.6
D	85.7	86.4	87.1	87.5	88.1	88.4	87.9	88	88.3
EXHAUST	89.6	88.4	93.3	93.4	91.2	92.5	91.6	95.6	93.7
GRID POINTS									
1	83.8	83.4	84.5	84.7	86.1	85.7	85.9	84.8	85.5
2	85.4	85.8	86.2	85.4	87.9	86	85.2	85.9	85.1
3	82.5	83	84.2	86.7	87.7	86.5	85.9	86.8	86.5
4	83.6	84.4	85.2	85.4	87.2	86.6	86.8	86.8	87.3
5	84.1	84.5	85.5	86.7	87.5	87.3	86.6	87.4	87.1
6	86.1	86.4	86.3	86.3	87	87.3	88.3	87	86.6
7	83.5	84.2	84.5	86.3	86.3	88.2	86.9	86.7	86.4
8	83.7	84.8	85.7	86.5	86.2	86.5	85.9	85.8	86.6
9	84.9	85.2	86.1	87.7	87.1	87.8	88.2	87.9	87.8
10	85.1	84.6	85.3	85.8	86.5	86.8	87.2	85.9	86.5
11	84.5	84.2	85.6	86.3	86.7	86.3	85.7	87.3	86.3
12	82.1	83.4	85.1	85.1	86.1	86.5	86.5	86.3	86.7
13	85.7	86.4	87.4	87.9	88.5	88.2	88.9	88.7	89.5
14	84.4	85.2	85.9	86.4	86.9	86.6	87.7	87.6	87.3
15	85.2	86.2	86.8	88.7	88.8	88.1	89	88.5	88.4
16	83.3	84.3	85.8	86.6	86.1	87.1	85.7	86.3	86.3
17	87.5	87.3	86.9	88.2	88.4	88.6	88	88.1	88.5
Ls	**84.67**	**85.27**	**86.25**	**87.00**	**87.35**	**87.38**	**87.22**	**87.46**	**87.45**

TABLE NO.7.8 FOR C.R.15.5 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 15.5									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	83.6	85	86.2	87.3	86.9	87.5	88.1	87.7	87.2
B	84.7	86.8	87.7	88.7	89.4	89.1	88.6	89	87.4
C	86.7	86.4	87.9	88.8	88.7	89	89.4	88.5	89.7
D	85.7	86.9	89.4	89.1	88.9	87.6	87.3	89.6	88.4
EXHAUST	87.3	88.6	87.2	91.1	91.9	90.7	94.4	93.5	96.8
GRID POINTS									
1	82.5	83	84.5	85.9	85.9	85.4	86.2	86.3	87.2
2	82.5	83.2	84.9	87.4	86.6	86.3	86.8	88.4	88.8
3	82.3	84.2	85.5	86.3	86.4	86.4	87.1	86.9	87.6
4	84.1	84.9	85.4	86.9	87.3	87.4	87.4	87.2	86.9
5	84.3	84.3	86.6	87.5	87.4	88	88.7	88.4	88.3
6	85.1	86.3	88	88.4	89	87.6	88.3	87.2	88.8
7	83.7	84.7	85.7	86.3	90.1	87.3	88.2	86.1	86.4
8	83.9	85.1	85.6	87.4	86.9	86.3	87.9	87.2	86.4
9	84.5	86.9	87.2	89.3	88.5	89.3	89.2	88.9	89.1
10	82.6	84	84.6	86.6	85.9	86.4	86.5	86.8	87.4
11	82.8	84	86.1	87.5	87	86.1	87.6	88.4	87.6
12	84	85.3	85.1	86.5	86.8	87.1	87.3	86.6	87.2
13	85.6	87.1	87.4	89.6	89.1	89.7	90.4	89.5	88.5
14	84.7	85.4	86.2	86.5	87.3	87.4	88.3	88.2	87.8
15	85.5	86.7	88.1	88.9	89.9	89.4	90.2	90.1	89.3
16	83.8	84.7	86.2	87.2	88.3	87.2	88.5	87.3	87.5
17	87.2	87.7	87.2	88.9	89.1	89.3	90.1	89.3	89.8
Ls	**84.41**	**85.51**	**86.49**	**87.82**	**88.06**	**87.75**	**88.48**	**88.23**	**88.37**

TABLE NO.7.9 FOR C.R.16 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 16									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	84.8	84.9	85.8	86.8	86.5	87.1	87	86.1	86.6
B	83.6	85.6	87.7	87.2	86.8	88.2	87.5	87.1	87.4
C	85.2	85.3	86.6	87.3	88.4	88.7	90.8	89.8	90.3
D	85.3	85.8	84.4	88.5	90	89.3	88.2	88.2	88.6
EXHAUST	83.3	83.9	85.5	89.5	89.7	92.6	90	96.1	94.7
GRID POINTS									
1	81.7	81.9	84.2	84.4	84.8	85.4	86.5	84.5	85.1
2	83.4	83.1	84.9	85.9	85.7	85.5	88.6	85.1	85.3
3	83.4	83.6	84	85.1	85	86.6	87.5	85.3	85.5
4	84.8	83.5	85	86.1	86.2	86	86.4	87.8	86.1
5	83.2	83.8	84.8	86.3	86.9	87.1	87.4	87.8	89.9
6	85.4	85.5	85.1	87.2	87.9	88.4	87.2	88.2	86.8
7	85.6	83.6	84.6	85.9	85.2	86.9	86.9	86.4	85.7
8	87	83.7	84.6	85.6	86.1	86.4	86.8	85.8	85.9
9	87.8	85.7	86.6	87.6	88.4	88.5	88	88.4	87.8
10	82.6	83.4	84.2	83.8	86.5	87.3	87	85.6	85.4
11	84.1	85.7	85.6	86.8	87.1	87.6	87.6	85.5	85.9
12	83.7	83.4	84.2	84.3	86.3	87.4	86.9	85.6	85.7
13	85.3	85.6	86.9	88.6	89	88.9	88.2	87.7	88.8
14	84.3	85.1	85.1	85.3	86.8	86.8	87.3	87.6	86.2
15	85.2	85.6	86.5	87.6	87.8	87.9	88.3	90.1	88.4
16	85.2	83.8	84.3	84.5	86.5	85.9	87	86.7	86.7
17	88.4	86.4	86.5	88.7	88.6	89	89.2	88.8	89.1
Ls	**84.70**	**84.50**	**85.32**	**86.50**	**87.10**	**87.61**	**87.74**	**87.46**	**87.36**

TABLE NO.7.10 FOR C.R.16.5 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 16.5									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	83.8	85.7	86.7	86.6	86.9	86.7	86.8	86.3	85.9
B	84.8	87.4	87.6	87.7	89.2	87.9	87.9	88.4	87.6
C	86.3	88	89.2	88.5	88.6	86.6	87.5	88.8	86.9
D	85.8	86.9	86.7	87.6	88.8	86.8	86.6	86.8	86.8
EXHAUST	89.1	90.5	89.9	92.3	89.3	89.5	96.2	101.5	97.3
GRID POINTS									
1	82.3	83.9	84.9	85.1	85.6	85.8	85.7	86.6	85.2
2	83.9	83.9	86.1	86.8	86.6	85.8	85.8	87	85.7
3	83.7	84	85	85.7	86.4	85.9	85.5	87.2	86.1
4	83.6	85.8	85.4	86	87.4	86.4	86.4	86.5	86.5
5	84.9	85.6	86.1	86.6	86.8	87.4	87.3	87.2	87.1
6	85.6	86.2	87	88	88.1	88.2	86.9	86.9	88.2
7	83.5	84.9	85.4	86.5	87.3	86	86.4	85.7	85.6
8	83.4	85.6	85.8	85.8	86.6	86.7	87.6	85.9	86.8
9	84.3	86.9	86.8	88.5	88.2	88.7	87.8	87.6	88.1
10	83.7	84.5	84.3	85.2	85.5	85.4	85	86.1	84.9
11	83.6	85.6	85.8	86.9	86.5	86.7	86.3	87.1	86.3
12	83.4	84.6	86.2	85.7	86.8	86.3	86.4	87.5	85.8
13	85.4	87.1	87.2	88.6	89.8	89.1	88.9	88.2	88.4
14	84.5	86	85.9	85.4	85.9	86.6	86.4	86.9	86.9
15	84.6	86.9	86.9	88.1	87.9	88.4	86.9	87	88.9
16	84.2	84.4	85.7	86.5	86	86.5	87.3	86.3	86.4
17	87	88.2	87.3	88.9	88.4	88.9	88.7	88.6	88.9
Ls	**84.61**	**86.03**	**86.45**	**87.14**	**87.39**	**87.10**	**87.29**	**87.73**	**87.29**

TABLE NO.7.11 FOR C.R.17 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 17									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	82.8	84.1	85.5	85.6	86.3	86.2	86.1	86	86.9
B	84.6	85.4	87.5	88.9	88.5	88.3	87.4	88.1	86.7
C	84.9	85.7	87	87.1	87.9	87	88.8	87	88.1
D	84.9	85.7	87	86.9	88.4	86.9	88.9	87.8	88.6
EXHAUST	86.8	88.7	89.9	94.5	93.5	96.3	93.6	98.2	97.2
GRID POINTS									
1	81.2	83.1	83.7	85.2	85.5	85.7	85.4	84.8	84.9
2	81.1	83.4	84.6	86.1	85.5	86.7	85.3	85.1	85.6
3	81.5	82.9	84.6	85.7	86.6	86.6	85.5	85	84.3
4	82.8	84	85	86.3	86.7	86.9	86.7	86.7	86.4
5	83.5	83.5	84.7	86.3	87.5	87.4	86.6	87.4	86.9
6	83.4	84.4	87.1	87.5	87.7	88.4	86.2	86.1	86.9
7	82.9	84.4	84.6	85.7	86.8	86.4	86.3	84.8	85.2
8	83.8	83.9	84.9	86.2	86.5	86	86.3	86.1	86.7
9	83.5	84.9	87.4	87.3	87.8	88.1	87.4	88.2	87.6
10	81.5	82.6	83.6	84.6	85.6	85	85.5	85.3	84.1
11	83	84.3	85.2	86.3	87.4	87.2	86.8	86	84.9
12	82.5	83.6	83.6	86.7	86.7	86.7	84.5	84.7	84.4
13	84.7	86.2	87.2	88.6	89.7	89.4	88.7	89.9	88.4
14	83.3	83.2	85	85.1	85.7	87	87.6	87.5	86.2
15	84.9	85.1	86.9	88.5	89.4	88.2	88.3	87.1	87.4
16	84.1	84	84.3	85.8	86.4	86.7	86.4	85.5	85.2
17	84.9	86.6	88.1	87.6	89.1	87.8	89.6	89.8	88.9
Ls	**83.48**	**84.53**	**85.79**	**86.93**	**87.51**	**87.50**	**87.18**	**87.14**	**86.89**

TABLE NO.7.12 FOR C.R.17.5 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 17.5									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	83.2	84.1	85.5	86.8	86.3	85.9	86.8	86	85.9
B	85.1	86.7	86.8	88.4	87.8	88	89.2	87.2	87.5
C	86.1	86.4	86.4	87.9	87.4	88.8	88.1	88.2	88
D	86	86.5	86.5	88.3	87.9	8.7	87.6	86.8	87.8
EXHAUST	88	93	88.7	90.6	89.8	93.1	92.7	93.8	99.5
GRID POINTS									
1	80.9	83.1	84	85.3	85.6	85.2	84.8	85.5	86.3
2	80.5	85.3	84.7	86.4	87.7	85.8	85.7	85.6	85.1
3	81.2	83.8	85	86.3	86.7	86.8	86.2	86.1	85.3
4	82.4	84.6	85.6	87.3	88	87.4	87.5	86.2	86.6
5	83.7	84	87.6	87	87.2	87.7	88.1	87.5	86.6
6	83.6	85.6	86.9	88.2	88.8	88.8	88	88.7	86.7
7	82.1	84.8	85.8	86.1	86.7	86.8	87.1	86.5	86.5
8	83.4	84.1	85.2	85.1	86.6	86.5	86.6	86.5	86.7
9	83.8	85.7	86.8	88.2	88.5	87.8	88.9	87.3	87.2
10	81.5	83.4	86.2	85.8	85.4	85.9	85.8	85.8	84.7
11	82	85.2	85.3	86.7	87	88.3	86.8	85.9	85.3
12	82.6	83.7	84.6	85.1	86.6	87.2	85.5	86.8	86.1
13	85.3	87	87	89.3	89.2	88.9	87.9	89.1	88.2
14	84.2	84.2	85.7	84.8	85.6	85.3	86.9	86.5	86
15	83.7	85.1	86	87.1	88.8	88.2	88.8	88.4	89.4
16	84.2	83.9	84.9	86.4	86.5	86.6	86.9	86.6	86.3
17	87.6	86.8	87.4	87.8	88.4	89.5	88.8	88.8	87.6
Ls	**83.69**	**85.32**	**86.03**	**87.04**	**87.39**	**83.96**	**87.49**	**87.26**	**87.24**

TABLE NO.7.13 FOR C.R.18 AT 20% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 18									
LOADS (Kg)	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
POINTS									
A	83	84.8	86.1	87.5	89	86.8	88.1	87	87.5
B	86.4	86.4	88.6	87.8	89.2	89.8	88.5	89.2	87.7
C	85.4	86.9	87.6	88	89.5	88.3	89.9	88.2	88.4
D	85.1	86.4	86.1	89.2	89	89.4	89.6	88.3	89.4
EXHAUST	88.8	87.2	90.2	93.8	89.1	91.5	89.8	92.3	90.7
GRID POINTS									
1	81.1	83.3	83.9	84.6	85.4	86.2	85.1	86.3	84.9
2	81.5	82.4	85.3	86.5	87.8	86.9	86	85	85
3	81.9	85.4	85.8	87.4	87.7	87.8	87.2	87	85.5
4	81.7	84.5	86	87.3	87.4	87.9	85.2	87.5	86.2
5	82.9	85.5	85.9	86.2	86.8	87.1	87.2	87.9	87.3
6	85.1	86.3	86.8	88.3	87.7	88.2	87.6	86.7	86.7
7	83.1	84.5	85.2	85.9	88	88.9	85.7	88.8	87.7
8	83.5	84.3	85.1	84.9	86.6	87.9	86.6	86.3	86.2
9	84.8	86.5	86.8	88.6	89.9	89.4	89.4	89.1	89.2
10	81.2	84.4	84.8	85.6	87	84.9	89.3	85.2	84.8
11	82.4	84.8	86.5	87.3	87.7	86	85.6	85.8	86.8
12	83.9	85.2	83.8	86.2	87	88.4	86.2	86.4	86
13	84.3	86.6	86.6	88.8	89.3	88.5	87.1	89.4	89.1
14	82.9	85.3	86.2	86.7	87.4	87.5	89.1	88.5	87.6
15	86.2	86.7	87.4	88.3	88.4	88.5	87.3	88.5	88.9
16	83.5	84.9	85.3	86.6	87.1	87.6	85.8	85.9	85.7
17	87.6	88	87.6	88.8	87.4	87.5	90.4	86.9	88.5
Ls	**83.92**	**85.47**	**86.25**	**87.47**	**87.93**	**87.95**	**87.58**	**87.55**	**87.26**

APPENDIX-A-2

OBSERVATIONS IN SPL (dB(A)) FOR VARYING COMPRESSION RATIOS AND LOADS AT 10% BLEND OF PONGAMIA OIL

TABLE NO.7.14 FOR C.R.12 AT 10% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 12					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	86.5	86.9	86.2	85.9	86.4
B	87.8	88.4	89.5	88.9	88.5
C	88.9	87.5	88.3	88.5	88.4
D	87.2	88.3	88.4	88.7	87.9
EXHAUST	86.7	88.9	91.9	91.1	87
GRID POINTS					
1	81.1	83.3	83.9	84.6	85.4
2	81.5	82.4	85.3	86.5	87.8
3	81.9	85.4	85.8	85.4	87.7
4	81.7	84.5	86	86.1	87.4
5	82.9	85.5	85.9	86.2	86.8
6	85.1	86.3	86.8	88.3	87.7
7	83.1	84.5	85.2	85.9	88
8	83.5	84.3	85.1	84.9	86.6
9	84.8	86.5	86.8	88.6	89.9
10	81.2	84.4	84.8	85.6	87
11	82.4	84.8	86.5	85.9	87.7
12	83.9	85.2	83.8	86.2	87
13	84.3	86.6	86.6	86.8	89.3
14	82.9	85.3	86.2	86.7	87.4
15	86.2	86.7	87.4	88.3	88.4
16	83.5	84.9	85.3	86.6	87.1
17	87.6	88	87.6	88	87.4
Ls	**84.30**	**85.85**	**86.51**	**86.99**	**87.58**

TABLE NO.7.15 FOR C.R.14 AT 10% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 14					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	85.8	86	86.7	85.9	86
B	86.1	87.6	88.6	89.2	88.7
C	86.7	87.4	87.5	88.1	88.4
D	86.5	86.9	87.9	87.9	87.4
EXHAUST	92.8	92.9	91	95.6	92.9
GRID POINTS					
1	83.2	84.3	84.6	85	84.3
2	83.1	84.3	84.6	85.2	84.7
3	83.5	84.7	85.3	86	86
4	83	84.8	86.3	86.5	86.3
5	83.9	85.4	86	85.8	85.7
6	85.6	86.1	85.9	86.6	86.7
7	84.2	84.7	85.8	86	85.6
8	84.3	85.1	85.5	86.3	85.9
9	85.8	85.8	86.9	86.9	86.6
10	83.7	83.5	85.1	85.2	85
11	83.5	85.1	85.8	85.9	85.8
12	83.3	83.9	85.7	86.6	85.5
13	83.4	86.3	87.9	88.2	87.8
14	84.9	84.8	85.6	85.9	85.7
15	86.7	86.3	86.6	87.6	87.9
16	84.3	84.5	85.5	86.2	84.8
17	86.2	85.3	86.4	87.3	88.1
Ls	**84.27**	**84.99**	**85.85**	**86.31**	**86.02**

TABLE NO.7.16 FOR C.R.16 AT 10% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 16					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	83.2	86.1	87.6	86	86.1
B	85.7	88.1	88.7	89.1	88.5
C	86.1	87.9	87.6	88.5	87.6
D	85.6	87	88.6	87.9	87.2
EXHAUST	90.1	89.9	92.2	92.9	95.1
GRID POINTS					
1	83.7	85.9	85.4	86.2	86.3
2	83.2	86.6	86.3	86.8	88.4
3	84.2	86.4	86.4	87.1	86.9
4	84.9	87.3	87.4	87.4	87.2
5	84.3	87.4	88	88.7	88.4
6	86.3	89	87.6	88.3	87.2
7	84.7	90.1	87.3	88.2	86.1
8	85.1	86.9	86.3	87.9	87.2
9	86.9	88.5	89.3	89.2	88.9
10	84.8	85.9	86.4	86.5	86.8
11	84.4	87	86.1	87.6	88.4
12	85.3	86.8	87.1	87.3	86.6
13	87.1	89.1	89.7	90.4	89.5
14	85.4	87.3	87.4	88.3	88.2
15	86.7	89.9	89.4	90.2	90.1
16	84.7	88.3	87.2	88.5	87.3
17	87.7	89.1	89.3	90.1	89.3
Ls	**85.46**	**87.75**	**87.79**	**88.32**	**88.06**

TABLE NO.7.17 FOR C.R.18 AT 10% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 18					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	83.2	84.9	86.2	86.8	86
B	83.9	87.6	89.2	89.9	89.1
C	84.5	86.2	87.5	89.3	88.6
D	84.8	86.4	88.2	88.2	88.9
EXHAUST	87.9	89.1	91.2	93.2	95.7
GRID POINTS					
1	85.6	85.8	85.7	86.6	85.2
2	86.6	85.8	85.8	87	85.7
3	86.4	85.9	85.5	87.2	86.1
4	87.4	86.4	86.4	86.5	86.5
5	86.8	87.4	87.3	87.2	87.1
6	88.1	88.2	86.9	86.9	88.2
7	87.3	86	86.4	85.7	85.6
8	86.6	86.7	87.6	85.9	86.8
9	88.2	88.7	87.8	87.6	88.1
10	85.5	85.4	85	86.1	84.9
11	86.5	86.7	86.3	87.1	86.3
12	86.8	86.3	86.4	87.5	85.8
13	89.8	89.1	88.9	88.2	88.4
14	85.9	86.6	86.4	86.9	86.9
15	87.9	88.4	86.9	87	88.9
16	86	86.5	87.3	86.3	86.4
17	88.4	88.9	88.7	88.6	88.9
Ls	**86.55**	**86.95**	**87.16**	**87.53**	**87.46**

APPENDIX-A-3

OBSERVATIONS IN SPL (dB(A)) FOR VARYING COMPRESSION RATIOS AND LOADS AT 30% BLEND OF PONGAMIA OIL

TABLE NO.7.18 FOR C.R.12 AT 30% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 12					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	89.5	90.2	90	88.7	88.6
B	91.8	92.8	91.5	91	90.9
C	91.7	92.7	92	90.9	91.2
D	90.8	91.2	90.6	90.3	90.3
EXHAUST	87.5	88.1	87.2	92.4	90.9
GRID POINTS					
1	84.6	85	84.3	85	83.9
2	84.6	85.2	84.7	85.4	84.2
3	85.3	86	86	85.9	85.8
4	85.4	86	85.8	85.7	87
5	86.1	85.9	86.6	86.7	87
6	84.7	85.8	86	85.6	86.1
7	86.9	86.6	87.1	87	87.3
8	85.2	85	85.6	84	85.8
9	85.9	85.8	85.2	86.2	86.4
10	86.6	85.5	86.2	86	85.9
11	86.3	87.9	88.2	87.8	88.1
12	84.8	85.6	85.9	85.7	85.9
13	86.3	86.6	87.6	87.9	88.2
14	84.5	85.5	86.2	84.8	86.4
15	85.3	86.4	87.3	88.1	88.4
16	86.2	84.8	86.4	86.5	86.9
17	87.3	88.1	88.4	88	88.5
Ls	**86.70**	**87.09**	**87.22**	**87.25**	**87.44**

TABLE NO.7.19 FOR C.R.14 AT 30% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 14					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	84.9	85.1	86.4	86.4	85.8
B	85.6	88	88.2	88.2	87.9
C	88.8	89	89	89.4	88.3
D	86.3	87.9	88.1	88.2	88.2
EXHAUST	87.1	94.4	95.7	93.1	96.4
GRID POINTS					
1	83.2	84.4	84.6	85.2	85.5
2	83.5	84.8	87.4	87.6	87.2
3	83.4	84	85.7	86.2	85.6
4	85.1	85.9	85.6	87.5	86.8
5	85.1	85.3	86.2	87	87.8
6	85.2	85.5	87.2	87.4	88.7
7	83.7	84.7	86	86.7	86.3
8	84.3	85.1	86	86.2	87.2
9	86.2	86.4	87.6	87.3	88
10	82.6	84.5	85.2	86.4	86
11	84.6	85.1	87.7	87.5	86.5
12	85.2	84.8	85.8	86.1	86
13	86.2	87.9	87.7	88.6	89.1
14	86.5	86.8	86.1	86.5	85.7
15	87.1	87.4	88.6	88.8	87.6
16	84.9	85	85.7	86.6	86.4
17	87.9	87.6	87.8	87.9	89.3
Ls	**85.34**	**86.35**	**87.20**	**87.49**	**87.56**

TABLE NO.7.20 FOR C.R.16 AT 30% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 16					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	84.4	85.8	87.6	86.7	86.1
B	85.5	87.4	87.8	88.2	88.2
C	86.3	87.2	87.9	88.5	87.5
D	86.2	87.8	88.4	87.9	87.7
EXHAUST	89.1	92.5	90.7	93	96.4
GRID POINTS					
1	81.2	83.1	83.7	84	84.8
2	81.1	83.4	84.6	84.7	85.1
3	81.5	82.9	84.6	85	85
4	82.8	84	85	85.6	86.7
5	83.5	83.5	84.7	87.6	87.4
6	83.4	84.4	87.1	86.9	86.1
7	82.9	84.4	84.6	85.8	84.8
8	83.8	83.9	84.9	85.2	86.1
9	83.5	84.9	87.4	86.8	88.2
10	81.5	82.6	83.6	86.2	85.3
11	83	84.3	85.2	85.3	86
12	82.5	83.6	83.6	84.6	84.7
13	84.7	86.2	87.2	87	89.9
14	83.3	83.2	85	85.7	87.5
15	84.9	85.1	86.9	86	87.1
16	84.1	84	84.3	84.9	85.5
17	84.9	86.6	88.1	87.4	89.8
Ls	**83.09**	**84.12**	**85.32**	**85.81**	**86.47**

TABLE NO.7.21 FOR C.R.18 AT 30% BIODIESEL BLEND

OBSERVATIONS IN SPL (dB(A)) FOR COMPRESSION RATIO 18					
LOADS (Kg)	**0**	**2**	**4**	**6**	**8**
POINTS					
A	82.8	86.2	87.1	86.7	86.3
B	84.8	87.2	88.8	89.7	88.7
C	84.6	87.2	87.8	89.1	88
D	85.8	86.2	88.4	88.8	88
EXHAUST	87.1	89.5	91.4	90.4	97.2
GRID POINTS					
1	80.9	85.3	85.6	84.8	85.5
2	80.5	86.4	87.7	85.7	85.6
3	81.2	86.3	86.7	86.2	86.1
4	82.4	87.3	88	87.5	86.2
5	83.7	87	87.2	88.1	87.5
6	83.6	88.2	88.8	88	88.7
7	82.1	86.1	86.7	87.1	86.5
8	83.4	85.1	86.6	86.6	86.5
9	83.8	88.2	88.5	88.9	87.3
10	81.5	85.8	85.4	85.8	85.8
11	82	86.7	87	86.8	85.9
12	82.6	85.1	86.6	85.5	86.8
13	85.3	89.3	89.2	87.9	89.1
14	84.2	84.8	85.6	86.9	86.5
15	83.7	87.1	88.8	88.8	88.4
16	84.2	86.4	86.5	86.9	86.6
17	87.6	87.8	88.4	88.8	88.8
Ls	**83.10**	**86.64**	**87.25**	**87.08**	**86.93**

1. MEASUREMENT OF ACOUSTIC POWER

For measuring the acoustic power one has to identify the length, breadth and height of the sound source (i.e. engine) and then calculate the area by the given formula

Area = ab + 2(bc + ca)

Where, "a" is Length =1.71 m

"b" is Breadth =0.7 m

"c" is Height =1m

Area =1.71*0.7+2(0.7*1+1*1.71)

= 6.017 m^2

For calculation of Acoustic power (Lw):

Lw = Ls +10$\log_{10}(S/S_0)$

Where, Lw = Acoustic power, (dB (A) ref. to 10^{-12} watt)

Ls = Sound pressure level, dB (A)

S = Hypothetical surface area

S_0 =Reference area, 1m^2

Lw = Ls +10$\log_{10}(6.017/1)$

Lw = Ls +7.79 ………………………………………………………………………… [equ. (1)]

Put all the values of Ls for different compression ratios and at different loads from the table no.7.1 to table no.7.21 from appendix A in equation no. (1).

1. OBSERVATIONS FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) at 20% BLEND OF PONGAMIA OIL

TABLE NO.7.22 FOR ALL COMPRESSION RATIO'S AT 20 % BLEND

LOADS	0	1	2	3	4	5	6	7	8
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 12									
Lw	93.12	93.36	93.79	94.17	94.31	94.36	94.48	94.78	94.69
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 12.5									
Lw	92.49	93.13	93.38	93.93	93.99	93.91	93.88	94.10	94.00
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 13									
Lw	92.36	92.95	93.43	93.51	94.06	94.19	94.43	94.34	94.52
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 13.5									
Lw	92.12	92.33	93.06	93.71	94.19	93.97	94.53	94.40	94.82
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 14									
Lw	92.42	93.45	93.64	94.25	94.57	94.81	94.83	94.84	94.63
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 14.5									
Lw	92.29	93.04	93.78	94.59	95.11	95.27	95.47	95.51	95.30
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 15									
Lw	92.46	93.06	94.04	94.79	95.14	95.17	95.01	95.25	95.24
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 15.5									
Lw	92.20	93.30	94.28	95.61	95.85	95.54	96.27	96.02	96.16
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 16									
Lw	92.49	92.29	93.11	94.29	94.89	95.40	95.53	95.25	95.15
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 16.5									
Lw	92.40	93.82	94.24	94.93	95.18	94.89	95.08	95.52	95.08
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 17									
Lw	91.27	92.32	93.58	94.72	95.30	95.29	94.97	94.93	94.68
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 17.5									
Lw	91.48	93.11	93.82	94.83	95.18	91.75	95.28	95.05	95.03
OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 18									
Lw	91.71	93.26	94.04	95.26	95.72	95.74	95.37	95.34	95.05

2. OBSERVATIONS FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) at 10% BLEND OF PONGAMIA OIL

TABLE NO.7.23 FOR ALL COMPRESSION RATIO'S AT 10 % BLEND

LOADS	0	2	4	6	8

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 12

Lw	92.09	93.64	94.30	94.78	95.37

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 14

Lw	92.06	92.78	93.64	94.10	93.81

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 16

Lw	93.25	95.54	95.58	96.11	95.85

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 18

Lw	94.34	94.74	94.95	95.32	95.25

3. OBSERVATIONS FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) at 30% BLEND OF PONGAMIA OIL

TABLE NO.7.23 FOR ALL COMPRESSION RATIO'S AT 30 % BLEND

LOADS	0	2	4	6	8

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 12

Lw	94.49	94.88	95.01	95.04	95.23

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 14

Lw	93.13	94.14	94.99	95.28	95.35

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 16

Lw	90.88	91.91	93.11	93.60	94.26

OBSERVATION FOR ACOUSTIC POWER (dB(A)ref. to 10^{-12} watt) FOR COMPRESSION RATIO 18

Lw	90.89	94.43	95.04	94.87	94.72

1. OBSERVATION OF SOUND PRESSURE LEVEL FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT DIFFERENT BLENDS OF PONGAMIA OIL

TABLE NO.7.24 FOR COMPRESSION RATIO 12 AT 10 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 12										
FREQUENCY (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	83.5	91.1	86.7	80.3	83.8	84.5	80.1	74.5	66.1	57.3
2	82.1	92.8	87.3	79.2	83.4	84.2	80.1	75.3	67.3	58.5
4	81.6	91.2	82.3	76.6	85.5	84.7	82	76.1	67	59
6	81.8	91.3	84	78.8	86.9	85.5	81.9	75.2	67.1	59.4
8	82.6	92.1	87.6	78.6	85.1	85.2	81.7	75.5	70.9	61

TABLE NO.7.25 FOR COMPRESSION RATIO 14 AT 10 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 14										
FREQUENCY (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	81.7	90.5	86.9	79.9	82.9	83.6	78.7	74.5	67.1	58.9
2	81.6	92.5	87.8	79.8	85.8	84.9	78.6	74.7	67.4	60.4
4	81.6	91.5	83.6	80.2	86.9	85.7	80.3	74.9	67.5	60.6
6	81.9	91.4	84.4	79.8	86.8	85.8	80.8	74.8	67.3	60.6
8	82.3	91.4	87.5	80.5	86.8	83.7	80.2	73.9	71.2	62

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 16										
FREQUENCY (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	80.4	91.4	85	77.4	83.2	82	76.8	73.8	66.8	58.6
2	80.8	92	84.7	77.7	85	83.9	79.1	74.3	67.3	60.8
4	80.7	90.6	83	77.3	86.3	85.2	80.4	75.5	67.6	61.3
6	81.6	90.9	83.8	78	87.2	85.7	79.6	73.9	67.8	60.2
8	82.9	91.6	84.8	77.5	87.5	84.3	79.6	74.4	73.1	64.7

TABLE NO.7.26 FOR COMPRESSION RATIO 16 AT 10 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 18										
FREQUENCY (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000
LOAD (Kg)										
0	80.1	91.4	83	75.6	81.7	80.5	76.5	73.2	66.2	58.1
2	80.7	91.2	82.6	76.4	84.6	83.5	78.5	74.7	68.1	60.5
4	81.2	90.5	84.2	76.9	87.3	85.1	79.4	74.9	66.9	59.9
6	81.8	91.6	84.5	78.2	89.1	86.1	79.9	75.3	70.3	60.7
8	82.4	91.7	81.1	77.6	87.5	84	79	74.5	74	66.2

TABLE NO.7.27 FOR COMPRESSION RATIO 18 AT 10 % BLEND

TABLE NO.7.28 FOR COMPRESSION RATIO 12 AT 20 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 12										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	77.3	89.1	84.3	77.6	88	87.9	84.5	80.3	70.5	61.2
2	79.4	89.2	83.3	79.8	90.1	88.5	85.2	80.9	69.1	59.1
4	80.9	90.2	82.9	76.4	90.2	87.5	84.2	80.7	69.3	60.9
6	79.9	90.8	86.4	78	90.7	87.7	85.6	80.3	68.6	58.9
8	81	90.7	88.7	78.2	88.5	88	84.1	79.2	70.5	60.2

TABLE NO.7.29 FOR COMPRESSION RATIO 14 AT 20 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 14										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	79.7	90.5	84.9	79.9	82.9	83.6	78.7	74.5	67.1	59.2
2	80.6	90.5	86.9	79.2	84.2	84.9	78.6	74.1	66.4	59.4
4	80.4	91.5	82.3	80.2	85.1	85.7	79.9	74.2	66.2	59.2
6	80.9	90.9	83.7	79.5	85.3	85.8	80.8	74.8	66.8	60.6
8	81.3	91.2	87.5	80.5	86.8	83.7	80.2	73.9	70.6	61.3

TABLE NO.7.30 FOR COMPRESSION RATIO 16 AT 20 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 16										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	77.9	89.8	82.3	76.1	78.8	82.9	76.4	73.4	63.5	58.4
2	80.6	91.1	80.1	75.5	83.4	85.9	80.1	72.9	66.5	60
4	80.5	91.6	84.5	76.5	86.9	85.2	80.5	75.5	67.2	59.8
6	82.6	91.9	82.8	77.2	87.5	83.5	79.6	72.9	67	60.2
8	82.1	90.5	84.3	76.5	85.9	86.3	79.1	74.8	72.7	63.4

TABLE NO.7.31 FOR COMPRESSION RATIO 18 AT 20 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 12										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	79.8	90.3	85	77.1	81.9	80.1	77.2	73.6	64.3	58.7
2	80.9	90.6	82.8	77.4	84.5	85.9	78.1	74.2	67.5	59.2
4	80.7	90.3	82.9	78.3	86.8	85.5	80.1	75.1	66.7	59.9
6	81.9	91.7	87.8	79.2	86.3	84.5	82.1	73.8	69	59.1
8	81.2	91.9	85.2	77.8	85.6	82.7	77.6	74.2	73.2	62.8

TABLE NO.7.32 FOR COMPRESSION RATIO 12 AT 30 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 12										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	79.3	90.1	84.3	77.6	88	87.9	85.4	81.1	70.3	60.2
2	80.4	91.2	83.3	79.8	90.1	89.5	85.2	80.9	69.5	60.4
4	80.9	90.6	82.9	76.3	90	88.5	85.7	80.3	69.7	60.9
6	81.9	91.2	86.4	78	90.7	88.7	85.6	80.3	69.6	59.6
8	83	91.7	88.7	78.2	88.5	88.3	84.6	79.7	71.2	61.2

TABLE NO.7.33 FOR COMPRESSION RATIO 14 AT 30 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 14										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	79.2	90.8	86	78.7	80.8	82.9	78.4	73.6	65.8	57.2
2	81	92.1	83.9	78	86.4	86.1	79.1	74.6	67.2	59.7
4	81.6	91.6	81.8	77.6	83.7	84.4	79.6	73.8	65.9	58.4
6	82	91.5	86.8	78.6	89	86.1	80.7	74.2	65.9	58.7
8	82.4	92.3	86.8	77.6	86.8	85.6	80.3	73.9	70.8	61.5

TABLE NO.7.34 FOR COMPRESSION RATIO 16 AT 30 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 16										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	78.9	90.4	83.2	76.1	79.8	82.9	77.3	73.4	65.9	57.7
2	80.6	91.5	81.8	76.5	84.4	85.3	79.1	74.6	67.5	60.4
4	81.5	91.2	81.2	75.9	85.3	86.3	80.3	75	67.1	59.6
6	82.1	91.6	87.9	79.2	86.2	84.8	79.5	74	67.6	59
8	82.5	91.8	87.9	76.2	86.5	85.9	80.4	74.6	70.5	60.5

TABLE NO.7.35 FOR COMPRESSION RATIO 18 AT 30 % BLEND

OBSERVATION FOR FREQUENCY SPECTRUM IN 1-1 OCTAVE BAND AT COMPRESSION RATIO 12										
FREQUENCY (Hz)	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**16000**
LOAD (Kg)										
0	79.8	91.3	84	77.5	80	81.5	77.5	73.3	65.7	57.6
2	80.9	91.6	82.8	77.4	84.5	83.9	78.1	74.2	67.5	59
4	81.7	91.3	82.9	78.3	86.8	85.5	80.1	75.1	66.7	58.9
6	82.2	91.7	87.8	79.2	86.3	84.5	80.1	73.8	69	59.1
8	82.9	91.9	88.2	78.6	86.1	83.1	79	74.2	72.7	64.8