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Design of an Acoustic Enclosure for a 12.5kVA Diesel Engine Electric Generator.

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ABSTRACT

The study focused on the design of an acoustic enclosure for 12.5 kVA diesel generator with an objective to minimize the noise level to a moderately loud sound level. Factors which affect noise reduction and heat management were considered. The study revealed that the loss in transmitted sound amounted to 20dB from the initial 95dB without an enclosure, with the insertion loss being 48.6%. The analysis of the transfer of the sound wave revealed that the frequency of the incident wave was lower than the critical frequency of the enclosure, this signifies that the enclosure will not resonate during operation. The heat generated within enclosure is 19280.806kW and this will cause the maximum temperature of the cylinder head to be exceeded. To avert this, acoustic holes where designed to allow a mass flow rate of 0.57kg/sec of air to pass through the enclosure, conducting the excess heat away.

Introduction

Epileptic power supply in Nigeria has become a norm rather than an anomaly. Despite heavy investments in the sector, gains experienced are disproportionate with the investments made. According to [1], the Transmission Company of Nigeria's (TCN) generation report disclosed that the nation witnessed total system collapse on June 28, 2016 and partial collapse on July 10. Overall, in 2016, the power grid collapsed 22 times – 16 total and five partial – up from 13 and 10 in 2014 and 2015 respectively. This has forced a large majority of the populace to acquire electric power generators in a bid to provide power for their daily activities spanning from industrial to domestic purposes. These generators have the advantages of been cost effective, user friendly and reliable than other energy options. However, sound generated from these generating set as a result of the vibration of the various parts of the generator causes noise pollution, hence, the need to curb the menace caused by the generator sound becomes imperative. It is against this background that the work aims to design an acoustic enclosure for a 2.5kVA diesel powered generator with adequate thermal management.

Research Elaborations

Sound is formed when vibrating objects in air induce a pressure wave that falls on human ear. When this sound is not desired, it is termed as noise. The noise source of a generator include: Engine noise, exhaust noise, alternator noise, vibrating structure, and change in load demand.

This polluting noise is known to cause side effects such as hearing loss, sleeplessness, psychological trauma, physiological changes, and pain. According to [2] by using a base-10 logarithmic scale, the whole range of human hearing can be described by a more convenient number that ranges from 0 dB (threshold of normal hearing) to 140 dB (the threshold of pain).

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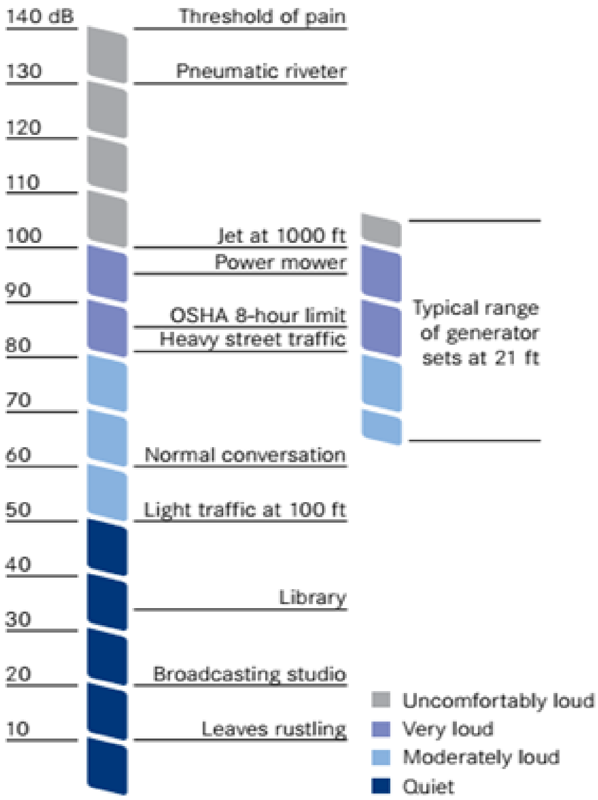

Figure 1. Decibel levels of different sounds [2].

Figure 1 indicates that the decibel level of the sound from the generating sets is usually very loud. Hence there is a need to attenuate the sound. Attenuation of sound can be done in many ways such as: the use of acoustic barriers, acoustic insulation, vibration insolation, attenuation of cooling air noise, exhaust silencers and maximizing the distance from the source [2].

Since noise is effectively controlled at the source and along its path of transmission, an acoustic enclosure, was used to attenuate the sound, with attention given to suppressant materials. The passive sound attenuation system intends to strike a balance between the control of the noise generated by the generator using an acoustic enclosure and the heat that ensues within the enclosure as a result of trapped heat energy. This heat causes the engine to exceed its operating cylinder head temperature, leading to eventual failure of the engine.

With the aid of a sound meter, the sound level of the generator without an enclosure was taken in an open test field of 75ft by 75ft and the average sound level was found to be 95dB.

Material Selection

Materials such as, oil palm fiber, coir fiber and Rockwool which can serve as good sound absorbers were considered, and Rockwool was selected due to the following reasons [3]:

- Iginition Temperature (850°C)
- Heat insulating
- Airflow resistance (9 kPa*s/m^2)
- Dynamic stiffness
- Surface protection of the product
- Density (100 kg/$\boldsymbol{m^3}$)
- Non-combustible
- Moisture and water repellent
- Sound-absorbing (up to 5000Hz)
- Thermal conductivity (0.033W/mK)
- Tensile Strength (10kN/m^2)

E-glass fiber was used as the casing with the following specifications:

- Physical properties (Type: chopped strand)
- Length of strand: 7mm
- Density: 2.58g$\boldsymbol{cm^{-3}}$
- Diameter of a strand: 13.0m.
- Tensile strength of 3445 Mpa. [3]
- Thermal Conductivity: 0.05W/mK. [4]

Fabrication of the enclosure

There are two aspects of the design. The first aspect involves fabricating the composite panel boxes using E-glass fiber composite which form the inner and the outer casings of the enclosure. The reasons for making a composite material our choice are due to their sound and vibration damping properties as well as their light weight which makes the enclosure portable. The second aspect of the design is the insertion of Rockwool as the sound absorbing material in the 50mm gap between the two casings. This gap maximizes the sound insulation for airborne sound. The combination of high quality glass fiber/polyester composite and Rockwool as the absorption material gives the enclosure good absorption characteristics. The inner casing is designed to improve absorption of sound waves by the absorbing material. The dimension of the enclosure is 120cm x 90cm x 70cm.

Design Calculation

Sound consideration parameters

Transmission Loss (TL): The transmission loss of the panel indicates the amount of air-borne insulation provided by the panel or acoustic enclosure. It is directly proportional to the sound insulation. It also increases with frequency, but is found to be at its least value of the critical frequency. It is expressed as:

$$\boldsymbol{TL = 20 \times log_{10}\left[\frac{1}{\tau}\right]} \quad (1)$$

Where $\boldsymbol{\tau}$ = transmission loss coefficient.

Transmission Loss Coefficient $\boldsymbol{\tau}$*:* This indicates the amount of incident acoustic wave that passes through the acoustic enclosure and appearing at the secondary section of the panel. It is unity at critical frequency. The higher its value (above 1), the higher the transmission loss. It is expressed as:

$$\boldsymbol{\tau = \frac{Incident\ pressure}{Transmitted\ pressure}\frac{P_i}{P_T}} \quad (2)$$

Transmitted pressure, $\boldsymbol{[P_T] = P_{rms}\sqrt{2}}$ (3)

Where: Incident pressure,$\boldsymbol{(P_i) = P_{rms}\sqrt{2}}$ (4)

$\boldsymbol{P_{rms}}$ = Pressure root mean square = $\boldsymbol{10^{(level\frac{dB}{20})}}$ (5)

Sound Pressure Level/Acoustic Pressure (SPL): this is the pressure at which the sound travels in air. It is expressed as :

$$SPL = \boldsymbol{20\ log_{10}\left[P^I_{rms}/P_{ref}\right]} \quad (6)$$

Where: $\boldsymbol{P_{ref} = 2 \times 10^{-5} pa}$ in other media [5]

Insertion Loss, (IL): The loss gives the effect of the enclosure on the noise source. It is a comparism of the sound pressure level without enclosure to the sound pressure level with enclosure. It is expressed as:

$$IL = 10\ \boldsymbol{log_{10}\left[E^I/E^I_T\right]} \quad (7)$$

Where: $\boldsymbol{E^I}$= Sound pressure level without enclosure

$\boldsymbol{E^I_T}$ = Sound pressure level with enclosure

Critical frequency of the enclosure, $\boldsymbol{(F_C)}$*:* This is the lowest frequency at which the enclosure begins to vibrate at an amplitude almost equal to the amplitude of the air particles in the incident wave. In other words, the panel seems to resonate by transmitting a wave that is nearly as intense as incident wave. Mathematically, this will occur when

$\boldsymbol{F \geq F_C}$. This phenomenon is known as acoustics as coincidence.

The critical frequency of the enclosure is expressed as:

$$\boldsymbol{F_C = \frac{0.556\,C^2}{t}\sqrt{\frac{\rho_p}{E_p}}} \quad (8)$$

Where $\boldsymbol{\rho_p}$ = density panel

$\boldsymbol{E_p}$ = Young's modulus

t = thickness of panel

Incident wave velocity, $\boldsymbol{(U^I)}$*:* This is the velocity at which the incident wave is projected in air. It is expressed as:

$$\boldsymbol{U^I = \frac{SPL \times x}{\rho_0 C_0}} \quad (9)$$

Where: $\boldsymbol{\rho_0}$ = density of air; $\boldsymbol{C_0}$ = speed of sound in air at $\boldsymbol{20}$°C, x = perimeter of enclosure.

Acoustic Mach number, M: it's a ratio of the velocity of the incident wave to the velocity of the sound in 2m. It is expressed as

$$\boldsymbol{M = \frac{U^I}{C_0}} \quad (10)$$

Bending wave velocity, $\boldsymbol{(C_B)}$*:* This is the velocity at which the absorbed sound wave is propagated in the panel. Hence, when $\boldsymbol{C_B \leq U^I}$, coincidence will occur.

$$\boldsymbol{C_B = \sqrt{\frac{1.8tfE_P}{\rho_P}}} \quad (11)$$

Frequency of the incident wave, (f): This can be evaluated using the relation

$$\boldsymbol{f = \frac{1}{2\pi}\sqrt{\frac{\rho_0 C_0^2 m_1 m_2}{(m_1 m_2)g}}} \quad (12)$$

Where: $m_1 =$ surface density of E-fiber, m_2 =surface density of Rockwool

m = ρ x t

g = distance between the composite panels.

Intensity Level, (∅): The sound intensity level can be evaluated using the relation established by [5]

$$\emptyset = 10\ log_{10}\left[{}^{\mathrm{I}}/_{\mathrm{I}_{ref}}\right] \quad (13)$$

Where: I $=\mathbf{P}_{rms}/\rho_0 C_0$, $\rho_0 C_0 = 4 \times 10^2_{kg}/m^2_{sec}$ for air under atmospheric condition

$\mathrm{I}_{ref} 10^{-12} W/M^2$ Is related to $\mathrm{P}^I_{ref} = 2 \times 10^{-5}$ pa in air.

Acoustic Impedance, (z): This is the opposition to the acoustic flow offered by the enclosure as a result of an acoustic pressure applied on the enclosure. It is expressed as:

$$\mathrm{Z} = \frac{complex\ acoustic\ presure}{complex\ particle\ velocity} = \frac{SPL}{U^I} \quad (14)$$

Angle of incidence, (φ): This is the angle at which the incident pressure wave strikes the enclosure panel. It is expressed as:

$Sin\varphi = \frac{\lambda}{\lambda_{\mathrm{B}}}$. When $\varphi = 90°$, coincidence wave emerges/occurs.

$\lambda =$ Incident wave length.

$\lambda_{\mathrm{B}} =$ Bending wave length.

Heat consideration parameters

In calculating the heat exchange associated to this enclosure, the following consideration where made:

Heat trapped within the enclosure: This is due to the heat generated during the operation of the diesel engine, radiation and heat loss due to enclosure wall. The technical data of the diesel engine is given in table 3.1.

Heat generated by diesel engine operation: The input energy which the engine runs on comes from fuel. Hence, Energy input,

$$(E_d) = \frac{LCV(diesel) \times mass\ of\ fuel}{operation\ \ time\ (\mathrm{sec})} \quad (15)$$

$$Mass\ flow\ rate = \rho_{diesel} \times volume\ flow\ rate$$

It was observed that when the engine runs at ¾ of the rated power output, the volumetric flow rate of fuel is 0.002Litres/hr.

The efficiency of the alternator is 90%

$$\eta_{alt} = \frac{Electrical\ Energy\ generated\ (E_e)}{Actual\ mechanical\ Energy\ supplied\ (E_m)} \quad (16)$$

Hence; Heat trapped in enclosure due to the engine operation

$$\mathrm{Q}_{\mathrm{engine}} = E_d - E_m \quad (17)$$

Where:

$$E_d = Heat\ Energy\ input, \quad (18)$$

$$E_m = \ Actual\ mechanical\ Energy\ operation \quad (19)$$

Heat trapped due to solar radiation:

The heat transfer by solar radiation is given as:

$$Q_{solar} = E\sigma A\left[T_{surf} - T_{amb}\right]^4 \quad (20)$$

Where: E = emissivity constant,

A = Area of enclosure,

$\sigma =$ Stefan-Boltzman constant,

$T_{surf} =$ Temperature of enclosure surface,

$T_{amb} =$ Ambient temperature.

Heat loss due to wall of the enclosure: The heat loss due to wall of the enclosure: The heat loss due to the composite wall on the 5 sides of the enclosure is expressed as:

$$Q_{comp} = \frac{T_{C.H} - T_{amb}}{R_{ToT}} \quad (21)$$

$$R_{ToT} = \frac{1}{h_i.A} + \frac{1}{K_A A} + \frac{1}{K_b A} + \frac{1}{K_A A} + \frac{1}{h_0 A} \quad (22)$$

h_i & $h_0 =$ Heat transfer coefficients between the inside and outside surfaces of the wall and surrounding air respectively.

K_A & $K_B =$ Thermal conductivities of E-glass fiber and Rockwool respectively,

A = Area of enclosure

Where: $R_{ToT} =$ Total thermal resistance across the composite wall

Table 1. A Technical Data of the Diesel Engine.

Model	ZS195NM
Type	Single cylinder, Horizontal, 4 stroke, Diesel
Combustion system	Swirl
Bore x stroke	95x115(mm)
Displacement	0.815L
Compression ratio	20:1
Rated output	9.7kW
Speed	2200 Rpm
Cooling system	Radiator
Lubrication system	Combined pressure and splash
Starting system	Electric start, key switch

Total heat trapped in the enclosure: The total heat trapped in the enclosure can be estimated with the relation given in equation (3.22)

$$Q_{enclosure} = \mathrm{Q}_{\mathrm{engine}} + Q_{solar} - Q_{comp} \quad (21)$$

The value of $Q_{enclosure}$ is the heat that should be conducted away from the enclosure in order to avoid the cylinder head temperature from exceeding its maximum limit.

To achieve this, forced convection was considered with the aid of a fan.

Mass of cooling air: The mass of air needed for cooling can be obtained using the relation in equation (22)

$$Q_{enclosure} = MC\Delta T \quad (22)$$

Where: C = specific heat capacity of air at constant volume

ΔT=$T_{enclosure} - T_{amb}$

$T_{enclosure} =$ Temperature of enclosure,

$T_{amb} =$ Ambient temperature.

Sizing of fan:

The size of fan needed can be obtained using the relation given in equation (23)

$$V = \frac{m}{\rho_{air}} \quad (23)$$

Where: m = mass flow rate of air.

ρ_{air} =Density of air

Thickness of panel (t): This is the thickness needed for maximum attenuation of the sound. Oldham *et.al,* (1991) expressed it as:

$$t = \frac{\mathrm{P}_i}{\alpha TLE} \quad (24)$$

Where: $\mathbf{P}_i =$ Incident sound pressure

$\alpha =$ Absorption coefficient

TL= Transmission loss

E= Average energy density for acoustic material.

Design Analysis

Transmission loss coefficient: The transmission loss coefficient was evaluated using equation (2). The incident pressure and transmitted pressures were first determined using the equations (3) and (4) respectively. In analyzing this, the $\mathbf{P}_{rms}$ was determined using equation (5). For incident pressure, the sound level was measured to 95dB.

Hence, $p_i = 10^{(95/20)} \times \sqrt{2}$ =79527.07μPa

$P_T = 10^{(75/20)} \times \sqrt{2}$ =7952.71$\mu Pa.$

$$\tau = \frac{79527.07}{7952.71} = 9.99$$

Transmission loss: This indicates the effectiveness of the attenuation. It was determined using the relation in equation (1)

$$TL = 20 \times \log_{10}\left[\frac{1}{9.99}\right] = -19.99$$

Sound pressure level (SPL): This gives the pressure at which the generated sound will travel in air. It was determined using equation (6)

$$SPL = 20\log_{10}\left[\frac{56234.13}{2\times10^{-5}}\right] = 188.98\mu Pa.$$

Insertion loss: It measures the effectiveness of the enclosure on the sound source, equation (7) was employed to determine its value for the study.

$$E_T^I = 20\log_{10}\left[\frac{5623.41}{2\times10^{-5}}\right] = 168.98\mu Pa$$

$$E^I = \text{SPL} = 188.98\mu pa$$

$$\therefore IL = 10\log_{10}\left[\frac{188.98}{168.98}\right] = 0.486 \text{ or } 48.6\%$$

Thickness of panel: Equation (24) was used to determine the panel thickness needed for maximum attenuation. Since diesel engine sound has a low frequency, enclosure was designed to attenuate the lowest low frequency sound. This is to enable the enclosure to serve in similar sized generating set.

Hence, sound absorptivity coefficient corresponding to the lowest frequency on table 3.2 was used in equation (24)

$$t = \frac{79527.07\ pa}{0.46\times19.99\times272160} = 0.0318\text{m} = 31.8\text{mm say } 35\text{mm}$$

Critical frequency of the enclosure, (f_c): This was determined using equations (8)

$$f_c = \frac{0.556\times343^2}{0.035}\left(\sqrt{\frac{100+2580}{10000+344510^6}}\right) = 1648.42H_Z$$

Incident wave velocity, (U^ι): The incident wave velocity is determined using equation (9)

$$U^\iota = \frac{188.98\times10^{-6}\times2[1.2\times0.9+1.2\times0.15+0.9\times0.15]}{400}$$

$$= 1.06\times10^{-6}m/sec$$

Mach number, (M): This is evaluated using equation (10)

$$M = \frac{1.06\times10^{-6}}{343} = 3.1\times10^{-9}.\ since\ m < 1,$$

The sound is subsonic

Frequency of incident wave: Equation (12) was used to determine the incident frequency.

$$F = \frac{1}{2\pi}\sqrt{\frac{1.20(343^2)3.18\times85.044}{(3.18+85.044)0.001}}$$

But $m_1 = 100\times0.0318 = 3.18\frac{N}{m^2}$

$$m_2 = 2580\times0.0318 = 85.044\frac{N}{m^2}$$

$$g = 0.001m$$

$$f = 1047.02H_Z$$

Bending wave velocity, (C_B): Equation (11) was applied in order to determine the bending wave velocity.

$$C_B = \sqrt{\frac{1.8\times0.0318\times1047.02(+3445\times10^6)}{2680}}$$

$$= 8777.18m/s$$

Acoustic impedance (z): Applying equation (14), we get

$$Z = \frac{188.98\times10^{-6}}{1.06\times10^{-6}} = 178.28\ \text{pa.s/m}$$

Heat generated by the diesel engine (E_d): The mass flow rate of the diesel fuel (m_f) was determined as:

$$m_f = \rho_{diesel}\times V_f$$

$$= 800\times0.00056 = 0.44\text{k}g/sec$$

Energy input $= LCV_{diesel}\times m_f$

$$= 43400\times0.44 = 19288.89KJ/sec$$

Table 2. Temperature of cylinder head without and with enclosure.

Operating time (min)	0	10	20	30	40	50	60
Temperature of cylinder head without enclosure (°C)	45	88	105	167	210	219	219
Temperature of cylinder head with enclosure (°C)	48	92	119	178	221	228	228

Actual mechanical Energy supplied, ($\mathbf{E_m}$):

Electrical load applied, (E_e):=3/4× power output

$$=3/4\times9700 = 7275W$$

$$\therefore E_m = \frac{E_e}{\eta_{alt}} = 8083.33\ W$$

Heat trapped in enclosure due to diesel engine operation,

$Q_{(diesel)} = E_d - E_m$

$$= 19288.89 - 8.083 = 19280.806\text{k}W$$

Heat trapped due to solar radiating, (Q_{solar}): Equation (18) was applied in order to determine (Q_{solar}).

$E = 1$(Assuming the sun to be a perfect block body)

$\sigma = 5.67\times10^{-8}W/m^2k^4$

$A = 3.48m^2$

$T_{surf} = 38$℃

$T_{amb} = 32$℃

$Q_{solar=1\times5.67\times10^{-8}3.48[311-305]^4}$

$Q_{solar=2.56\times10^{-4}w}$

Heat loss due to wall of enclosure, Q_{comp}

Where $T_{C.H} = 183$℃

$T_{amb} = 32$℃

$h_1 = 16.5W/m^2k$

$h_0 = 7.4W/m^2k$

$$R_{TOT} = \frac{1}{16.5\times3.48} + \frac{1}{0.05\times3.48} + \frac{1}{0.05\times3.48} + \frac{1}{0.05\times3.48} + \frac{1}{7.4\times3.48} = 21.13k/w$$

Hence, $Q_{comp} = \frac{183-32}{21.13} = 7.14w$

Total heat trapped in the enclosure

$Q_{enclosure} = 19280.806\text{k}W$

Mass flow rate of cooling air needed

$$m = \frac{Q_{enclosure}}{C\Delta T}$$

$$Temperature\ of\ enclosure = 68℃$$

$$m = \frac{2869336.67}{1000\times(68-32)} = 0.57\text{kg/sec}$$

Hence to determine the size of fan needed for forced convection, equation (3.24) is employed.

$$V = \frac{m}{\rho_{air}} = \frac{0.57}{1.20} = 0.47m^3/sec$$

Discussion

The aim of an acoustic enclosure is to ensure that the sound that ensues as a result of the vibrations associated with the operation of any machinery is reduced to a level that is in accordance with the local acceptable standard.

In this design, sound level productions such as the transmission coefficient, Transmission loss, Sound pressure level, Insertion loss, etc., were considered. From the analysis, the transmission coefficient is found to be 9.99, approximately

10, indicating that it is ten times above coincidence point. The enclosure was designed to ensure that the transmission loss for the low frequency sound generated by the generator will be 20dB. This led to a design thickness of 35mm for the enclosure. In order to enhance sound attenuation a composite wall enclosure was chosen; at a transmission loss of 20dB, the insertion loss which evaluates the efficiency of the sound insulation material was found to be 48.6%.

The critical frequency of the incident frequency of the enclosure was determined to be 1648.42Hz which is 1.6 times greater than the incident frequency of the sound wave. At this condition the enclosure is not susceptible to coincidence effect, hence will effectively serve the purpose of attenuation. The vibration encountered along exhaust pipe was considered as this also reduces the transmission loss.

To reduce this effect caused by the exhaust pipe vibration, a muffler was installed at the exhaust pipe external section.

Considering the enclosed space, the heat generated is expected to accumulate and at a point where the cylinder head temperature exceeds the operating temperature specified by the manufacturing company the engine fails to operate. Hence, heat considerations analysis was carried out on the engine. The heat trapped within the enclosure is $\mathbf{19280.806kW}$, from the analysis, a fan with a volumetric flow rate of 0.47$\boldsymbol{m^3/\mathrm{sec}}$ and above will effectively conduct this heat away. Hence, in order to ensure that the maximum operating temperature of the cylinder head specified by the manufacture as 287^{0}C is not attained, a fan with 0.5m^3/sec volume flow rate was installed. Figure 2 shows that the maximum cylinder head attained is 228°C.

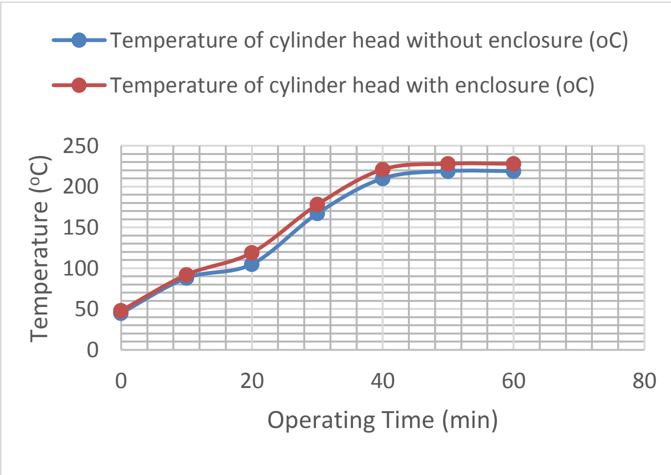

Figure 2. Cylinder Head Temperature variation with time when the generator is with and without an enclosure.

Conclusion

The study carried out an effective noise attenuation on a 12.5kVA diesel powered electric generator, using an acoustic enclosure. Material selection was considered, and the choice of Rockwool and E-glass fiber were made as the sound absorbing material and the panel casing respectively. Sound parameters considered includes: Transmission loss, Transmission loss coefficient, Sound pressure level, Insertion loss, Critical frequency of the enclosure, Bending wave velocity, etc. the effect of the heat generated as a result of the enclosure was also considered because its accumulation will cause the operating cylinder head temperature to exceed its safe operating limit. The design achieved a reduction in sound level from 95db (very loud) to 75dB (moderately loud).

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