International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 07 Issue: 09 | Sep 2020 www.irjet.net p-ISSN: 2395-0072

VIBRATION ANALYSIS ON SINGLE CYLINDER FOUR STROKE DIESEL ENGINE

S.Sakthi[1], K.Veerapandian[2]

[1]*Assistant Professor, Department of Mechanical Engg, SMR East Coast College of Engg.& Tech.,Tamilnadu, India*
[2]*Assistant Professor, Department of Mechanical Engg, Sir Issac Newton College of Engg.& Tech., Tamiladu, India*

---***---

Abstract: The purpose of this project is to mitigate the "vibration" in Diesel engines which deteriorate the engine performance adversely. The methodology introduced in the present work suggests a newly developed piston top land approach towards analyzing the vibration analysis of diesel engines. The method is based on fundamental relationship between the engine vibration pattern and the relative characteristics of the combustion process in each or different rpm. The combustion process mainly based on the piston combustion chamber structure. The restructured piston combustion chamber minimize the vibration forming inside the engine. Vibration in diesel engine is detected by measuring the vibration generated by the engine using The DC-11 FFT analyzer with accelerometer.

Key Words: *FEA, ANSYS, SOLID WORKS,*

1. INTRODUCTION

Vibrating in diesel engine always leads to the severe increase in pressure pulses inside the combustion chamber resulting a metallic noise which causes irreversible damage to the engine components and structure. The piston head is one of the most complex parts in the engine and it endures high thermal and mechanical stress during its working cycles. The good combustion may be influenced by proper designing of the combustion chambers of the piston and fuel burning in proper way manages the vibration of engine. So the design of the newly developed piston chamber gives better performance of the engine and vibration mitigation. "Design the Piston may damage often mainly due to Temperature, Wear, Fatigue etc. It is necessary to analyze the structure, strength and reliability of this zone .In this work, a three dimensional (3D) model of the Piston head is built up by using SOLID WORKS and the 3D FEM analysis ANSYS is carried out. The temperature distribution and stress distribution of the different piston head are carried out and to find out the best piston with optimum Thermal as well as Stress distribution. In this Paper, describes the stress and Thermal distributions on different Piston heads of I.C engine combustion chambers using FEA software. The main objective of this Project work is to analyze the vibration forming after top land modification of the Piston at real engine working condition at combustion process. The paper describes about vibration on the different regions on the engine head from mounting of the engine that is engine bed. Chamber portion of the top land of piston and fuel distributed in the chambers makes tumbler flow of the fuel intakes and also explains how the shape and structure in the piston design influenced in the combustion process in I.C engine. The FEA technique ANSYS is used to assign different load condition to the Combustion chambers in accordance with real time engine working conditions. Piston 3-D model is modelled by using SOLID WORKS software with real time I.C engine piston dimensions.

2. EXPERIMENTAL WORK

2.1. Experimental Set up

Experiment consists of a single cylinder DI-diesel single cylinder 5hp Kirloskar engine with compression ratio of 16.5. Newly developed piston assembled and the engine is fitted with eddy current dynamometer to apply different loads. The vibration generated by the engine is measured using vibration analyzer equipment.

Fig-1: Experimental Set up

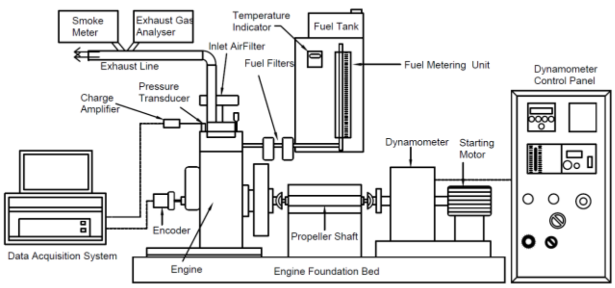

Fig-2: Experimental Block diagram

2.2. Vibration analyzer equipment

The DC-11 FFT analyzer is a digital spectrum analyzer and data collector specifically designed for machine condition monitoring, advanced bearing fault detection and measurement diagnostics. The following are the measurements that can be made by the instrument. DC-11.vibration analyzer.

Features

1. Time wave form (accelerometer) in OFF-ROUTE mode.

2. FFT auto spectra.

3. Envelop spectra selected by multiple band pass filters.

4. Rotation speed.

5. Amplitude and phase on rotation speed and its harmonics.

6. FrequencyRange1-2000Hz

7. Input Signal Range 100mV

8. Frequency Span 1-2000 Hz in 1 Hz resolution

9. Signal to Noise Ratio Greater than 70 dB

10. Amplitude Measurement Units Acceleration, velocity and displacement.

Fig-3: Equipment

Table-1: Specification

Frequency Range	1-2000Hz
Input Signal Range	100mV
Frequency Spam	1-2000Hz in 1 Hz resolution
Signal to Noise Ratio	Greater than 70dB
Amplitude Measurement Unit	Acceleration velocity and displacement

3. PISTON MODELING

3.1 Piston Design

The piston is designed according to the procedure and specification which are given in machine design and data hand books. The dimensions are calculated in terms of SI Units. The pressure applied on piston head, temperatures of various areas of the piston, heat flow, stresses, strains, length, diameter of piston and hole, thicknesses, etc., parameters are taken into consideration

3.2 Design Considerations for a Piston

In designing a piston for an engine, the following points should be taken into consideration:

- It should have enormous strength to withstand the high pressure.
- It should have minimum weight to withstand the inertia forces.
- It should form effective oil sealing in the cylinder.
- It should provide sufficient bearing area to prevent undue wear.
- It should have high speed reciprocation without noise.
- It should be of sufficient rigid construction to withstand thermal and mechanical distortions.
- It should have sufficient support for the piston pin.

3.3 Procedure for Piston Design

The procedure for piston designs consists of the following steps:

- Thickness of piston head (tH)
- Heat flows through the piston head (H)
- Radial thickness of the ring (t1)
- Axial thickness of the ring (t2)
- Width of the top land (b1)
- Width of other ring lands (b2)
- The above steps are explained as below:

GIVEN DATA :

Displacement = 149.89cc

Maximum power = 10.03 Ps @3000rpm

= 5 kw

SOLUTION:

Volume = π/4 d2 *l

149.89= π/4 d2*1.5 d {Assuming l=1.5d}

149.89= π/4 *1.5 d3

D= 5.174cm=6cm=60mm

3.4 PISTON HEAD OR CROWN:

The thickness of the piston head or crown is determined on the basis of strength as well as on the basis of strength as well as on the basis of heat dissipation and the larger of the two values is adopted.

(a) Thickness(tH) of piston head on the basis of strength

Material Selection: selecting material for piston as **aluminium alloy**.

Permissible bending [tensile] stress

σ=50 to 90 MPa

th=σ {taking p=45kg/cm2 f=500 kg/cm2}

tH=0.78cm=7.8 mm

3.5 Heat flowing through the piston head:

H=C*HCV*m*BP

C= Constant representing that portion of the heat supplied to the engine which is absorbed by the piston.

It varies from 5 to 20 %

HCV= higher calorific value of the fuel 47*103 KJ/BP/hr for petrol

m= Mass of fuel used in kg/BP/sec

H= 0.05*47*103 *41.7*10-6 *14

=1.37 kW

{Taking c=5% m=0.15 kg/BP/hr= 41.7*10-6 kg/BP/sec}

(b)Thickness of piston head on the basis of heat dissipation

tH=(1.37*103)/(12.56*174.75*75)

tH =0.0083m=8.32mm

Taking the larger of two values

tH=8.32 mm

k= thermal conductivity factor 174.75 W/M/

[Tc-Te]=temperature difference = 75

3.6 RADIAL RIBS:

Assumption: (a) 3 compression rings

(b) 1 oil ring

Radial thickness of piston rings t_1 = σ

T1=0.388=3.88mm

Taking pw=0.7 kg/cm2

σt =500 kg/cm2

t2=0.7t1=0.7*3.88=**2.716 mm**

The minimum axial thickness of the piston ring

tr =D/10nl=6/(10*4)=**1.5 mm**

Axial thickness of the piston ring as already calculated [tr=2.7] is satisfactory. Distance from the top of the piston to the first ring groove,i.e. width of top land

b1=(1 to 1.2)tr = 1.2*8.33=9.99≈1 cm

Width of the other ring lands

b2=(0.75 to 1)t2 =2.7 mm

The gap between the free ends of ring

G1=[3.5 to 4]t1

G1=4*3.8=**15.2 mm**

Gap when the ring is in the cylinder

G2=[0.002 to 0.004] D

G2=0.004*6=0.024 cm=**0.24mm**

3.7 Piston Barrel

Radial depth of the piston ring gloves

b=t1+0.4

=2.21+0.4=2.25mm

Maximum thickness of Barrel

t 3=0.03D+b+4.5

=(0.03*45)+2.25+4.5=8.1mm

Piston wall thickness towards the open end,

t 4=0.25 to 0.35 times t3

t4 =0.30*8.1=**2.43mm**

3.8 Piston Skirt

Let, l=length of the skirt, mm

The maximum side thrust on the cylinder due to gas pressure

R=μ*π/4*d2*p

=0.1*π/4*2025*4.414

=702.02N

Taking μ=0.1

P=45*9.81*(1/100)=4.414N/mm2

Side thrust due to bearing pressure on the piston barrel (Pb)

R=Pb*D*l

=0.45*45*l taking Pb=0.45Mpa

=20.25l N

20.25l=702.02N

L=34.67mm

Therefore total length of the Piston

L=length of the skirt + length of the ring section + top land

=l+[4(ti)+3b2]+b1

L =34.67+[4*1.5+3*2.7]+10=**58.77mm**

3.9 Piston Pin

d0=outside diameter of pin,mm

l1=Length of pin in the bush of the small end of the connecting rod,mm

Pb=Bearing pressure at the small end of the connecting rod bushing in N/mm2

For bronze bushing=25 N/mm2

Load on the pin due to bearing pressure,

=Bearing pressure * Bearing Area

=Pb1*d0*l1

=25d0*0.45*45 [l1=0.45D]

=506.25 d0 N

Maximum gas load =π/4 (D*D)*P

=π/4*2025*4.4

=6.998 KN

therefore 506.25 d0=6998 N

d0=13.824 mm ≈14mm

inside diameter of pin d1=0.6*d0

=0.6*13.824=8.3mm

d1=9 mm

Let the Piston pin be made of heat treated alloy steel for which the bending stress (σb) may be taken as 180 NPa

To check induced bending stress in pin

Maximum bending moment at center of pin

M=P*D/8

=7*45*10^3/8=39375

Also,the maximum bending moment

39375=π/32[d04-di4/d0]σb

93300= π/32[14^4-9^4/14] σb

σb=176.48 N/mm^2

Since the induced bending stress in the pin is less than the permissible value of 180 Mpa.

Therefore, the dimensions for the pin as calculated above (i.e.d0=14 mm & di=9 mm) are satisfactory.

Fig-4: Machined Piston

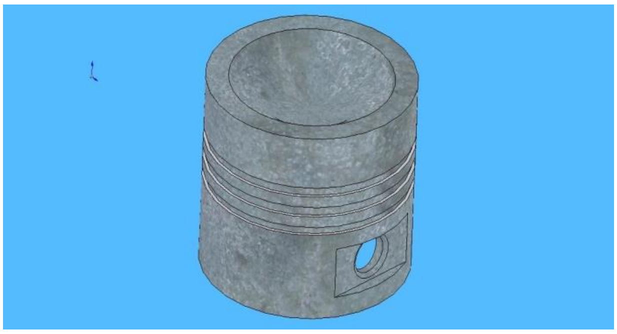

Fig-5: modeling piston

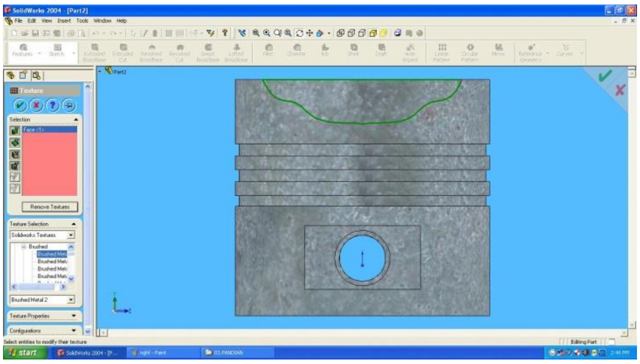

Fig-6: modeling piston

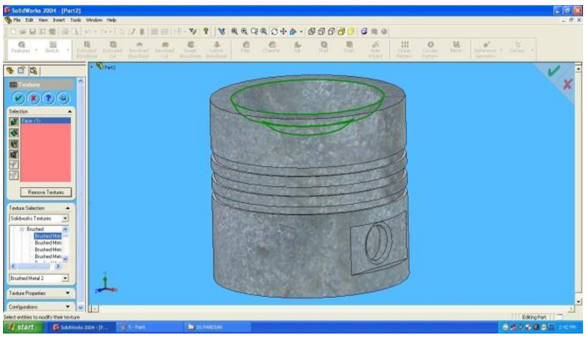

Fig-7: modeling piston

4. ANALAYSIS OF PISTON

The piston designed which is imported into the ANSYS software then it processed to analyse with mesh node method to get the piston strength and the stress distribution on the piston.

It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels.. As much as 60% of the total engine mechanical power lost is generated by piston ring assembly. The piston skirt surface slides on the cylinder bore. A lubricant film fills the clearance between the surfaces. The small values of the clearance increase the frictional losses and the high values increase the secondary motion of the piston. Most of the Internal Combustion (IC) engine pistons are made of an aluminium alloy which has a thermal expansion coefficient, 80% higher than the cylinder bore material made of cast iron. The thermal and geometric properties are as shown in below.

Table-2: Properties of Piston Alloy Material

Property	Aluminium Alloy	Zirconium
Young's Modulus	$70e^3$ Mpa	$220e^3$Mpa
Poisson Ratio	0.31	0.35
Thermal Conductivity	234W/mK	7W/mK
Co-eff of Thermal	$23e^{-6}$/K	$10e^{-6}$/K

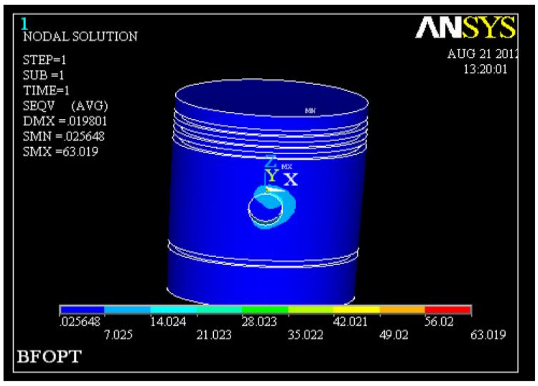

Fig-8: Deformation and Vonmisses Stress before Optimization

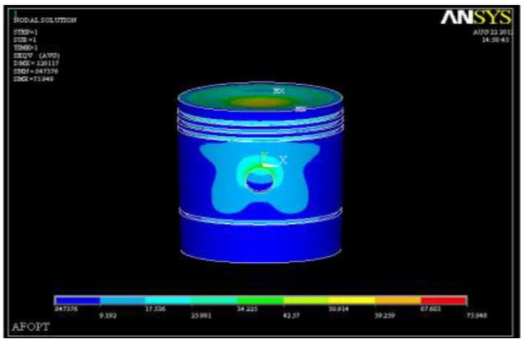

Fig-9: Deformation and Vonmisses Stress after Optimization

5. EXPERIMENTATION

5.1. Experimentation Procedure

- Experiment is carried out on single cylinder DI-diesel single cylinder 5hp kirloskar engine with compression ratio of 16.5.
- Experimentation is carried out at various engine loads applied through eddy current dynamometer and a spring balance.
- The experiment is carried with neat diesel.
- The load on the engine can be changed with the dynamometer control panel.
- Full load on the engine is equal to 40 kg on the spring balance. This dynamometer is popular for its stable and consistent readings.
  Engine cylinder vibration in FFT form is monitored at each load and for each ester simultaneously to compare the cylinder excitation frequencies with the base line frequencies.
- Time wave forms on the cylinder head are also recorded to analyze the combustion. Since the very combustion in the cylinder is the basic exciter, the vibration study of the engine cylinder through the measured FFT and time waveforms are the representatives of combustion propensity.

- Vibration accelerometer is mounted on the cylinder head, crank shaft bearing side and engine exhaust valve side preferably on the bolt connecting the engine bed and the cylinder to record the engine vibrations using DC-11 data logger which directly gives the spectral data in the form of FFT, the overall vibration levels.

5.2. Setup for Vibration Parameters

To measure the vibration occurring in the engine, an accelerometer is attached to the engine head through which the engine position with respect to time is measured.

The accelerometer is connected via a coaxial cable to an Analog to Digital (ADC) unit and then to the Data Acquisition System. The signals are captured using LabVIEW to interface the devices.

5.2.1. Accelerometer

An accelerometer is a device that measures the proper acceleration of the subject. This is not necessarily the same as the coordinate acceleration (change of velocity of the device in space), but is rather the type of acceleration associated with the phenomenon of weight experienced by a test mass that resides in the frame of reference of the accelerometer device.

An accelerometer measures weight per unit of (test) mass, a quantity also known as specific force, or g-force.

The Accelerometer used for this experiment is a DYTRAN model 3097A2. This unit has a sensitivity of 104.71 mV/g. It has a ceramic element with a planer shear element style that is housed in a Titanium housing. For detailed specifications,

6.2.2. Data Acquisition System

DAQ hardware interfaces between the signal and a PC. It could be in the form of modules that can be connected to the computer's ports (parallel, serial, USB, etc.)

5.2.3. Emission Parameters

Exhaust emissions of hydro carbons and carbon were measured on dry basis. A Crypton make 285 mode analyzer was used. The analyzer is a fully microprocessor controlled system employing non-destructive infrared techniques. The 285 measures CO, HC, CO_2 and another channel for measuring O_2 plus a further channel employing electrochemical measurement of oxygen,.

5.2.4. Smoke Testing

In cylinder smoke was measured using a standard HORIBA smoke measuring system. The measuring instrument consists of a sampling pump that sucks a definite quantity of exhaust sample through a white filter paper. The reflectivity of the filter paper was then measured using a evaluating a unit. The unit is mounted in a steel box that comprises a photoelectric cell pick up.

The photo electric cell pick up has a source of light which throws a beam of light on the soot impregnated filter paper disc held in the filter slide. The unabsorbed portion of light is reflected back on to the annular photoelectric cell that surrounds the light source. The intensity of the reflected light generates a current that is measured by a sensitive ammeter which gives the smoke reading.

Fig-10: Exhaust gas analyser

Table-3: Tabulation for emission test

Engine speed: 1500 rpm, Room temperature:30^0 c

% of Load	Calc of Load	Time taken 10cc(tavg)	EGT	CO	HC	CO_2	O_2	NO_X
			°C	% by vol	ppm	% by vol	% by vol	ppm
0	0	106	153	0.03	27	234	1660	461
20	4.71	100	185	0.04	28	238	1637	440
40	9.42	86	223	0.04	31	242	1648	422
60	14.13	65.5	268	0.04	35	320	1542	835
80	18.84	48	321	0.02	37	358	1582	1132

Table-4: Tabulation for emission test

Engine speed :1500 rpm

% of Load	Load Applied	Time taken 10cc of fuel consumption	Fuel consumption	Brake Power	Specific fuel consumption (SFC)	Fuel power	Break thermal efficiency
Kg	(Kw)	(T avg.) sec	(Kg/hr)	(Kw)	(Kg/kw/hr)	w	η%
0	0	300	0.36	0	0	4.2	0
20	4.71	220	0.49	0.73	0.66	5.73	12.91
40	9.42	172	0.62	1.47	0.42	7.33	20.19
60	14.13	140	0.77	2.21	0.347	9	24.65
80	18.84	112	0.96	2.95	0.32	11.25	26.29
100	23.55	94	1.14	3.69	0.31	13.40	27.59

Table-5: Engine vibration at normal condition

% of load	Load Applied	Engine rpm	Vibration (Hz) at different locations		
	Kw		Cylinder head	Cylinder bed	Flywheel shaft
0	0	1500	1289	1498	1410
20	4.71	1500	1367	1563	1504
40	9.42	1500	1486	1656	1617
60	14.13	1500	1570	1743	1699
80	18.84	1500	1468	1561	1523
100	23.55	1500	1440	1510	1468

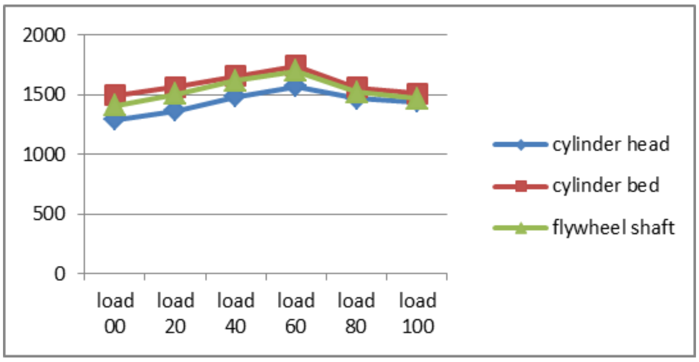

Chart-1: Engine vibration at normal condition

Table-5: Engine vibration at modified condition

% of load	Load Applied	Engine rpm	Vibration (Hz) at different locations		
	Kw		Cylinder head	Cylinder bed	Flywheel shaft
0	0	1500	1280	1512	1421
20	4.71	1500	1362	1570	1511
40	9.42	1500	1479	1661	1621
60	14.13	1500	1552	1745	1701
80	18.84	1500	1474	1572	1529
100	23.55	1500	1490	1518	1472

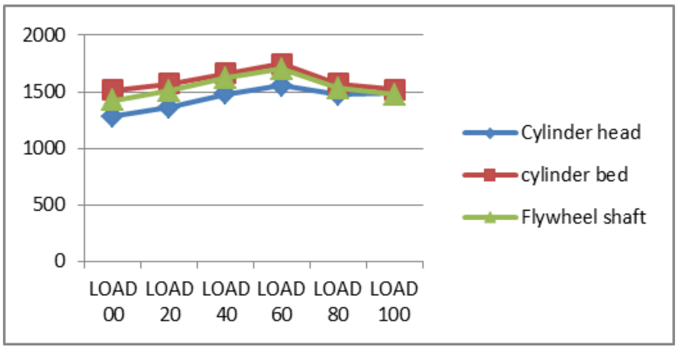

Chart-2: Engine vibration at modified condition

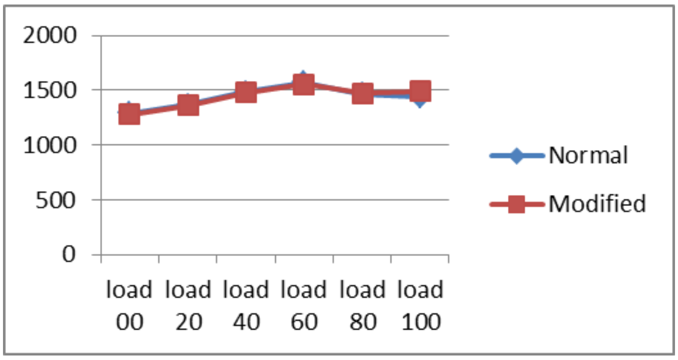

Chart-3: Comparison of Cylinder Head Vibration

6. CONCLUSIONS

The deflection of engine vibration at cylinder head in the critical load condition slightly improved. Even though other portion vibration increased conveniently. Improvement of piston top land designed structure can make the diminished vibration.

The stress distribution on the piston mainly depends on the deformation of piston. Therefore, in order to reduce the stress concentration, the piston crown should have enough stiffness to reduce the deformation. It has good sufficient strength in combustion process.

All the phases in this project given can be extended to the piston design with reduction of material at top land. The material is removed to reduce the weight of the piston so as to improve the efficiency. It is essential to obtain the optimized results for new piston with reduced material.

7. REFERENCES:

[1] Leppard, W. R. Individual-cylinder knock occurrence and intensity in multi cylinder engine, SAE Technical Paper, N°820074, 1982

[2] G. Brecq1 and O. 2Le Corre "Modeling of In-cylinder Pressure Oscillations under Knocking Conditions: Introduction to Pressure Envelope Curve .SAE2005

[3] Draper, C.S. The physical Effect of Detonation in a closed cylindrical chamber, Technical Report 493, NACA, 1935.

[4] G. A. Karim, S. R. Klat and N. P. W. Moore, Knock in dual-fuel engines. Proc. Inst. Mech. Engrs 181, 453466 (1967).

[5] Ziang Aihua" "Fault Diagnosis on Misfiring of Engine with Vibration Signal" world scientific e proceedings 2008.

[6] Jenkins S.H., "Analysis and treatment of diesel engine noise", Journal of Sound and Vibration Vol .43(2), pp 293-304, (1975).

[7] NwaforO.M.I., "Knock characteristics of dual-fuel combustion in diesel engines using natural gas as primary fuel", Department of Mechanical Engineering, Federal University of Technology, Vol. 27, Part 3, pp.375-382, (2002).

[8] WangYing, Zhou, Longbao. Investigation of the detection of knock and misfire of a spark ignition engine with the ionic current method. Proceedings of the Institution of Mechanical Engineers. 2003(7):617-621

[9] 'Gideon Goldwine, Gal deBotton, Boris Rivin and Eran Sher,"Assesment of the quality of combustion in compression ignition engine through vibration signature analysis-Fista world automotive congress, Barcelona2004

[10] "Condition Monitoring of a Small Four-stroke Petrol Engine using Vibration Signals. Songpon Klinchaeam KMITL Sci. Tech. J. Vol. 9,No. 1 Jan. - Jun. 2009.