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THEORETICAL AND EXPERIMENTAL STUDY OF COMPRESSION IGNITION ENGINE VIBRATION AND NOISE

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ABSTRACT:

This paper presents an analytical model of the pressure force and vibratory response of the cylinder induced by the piston movement of compression ignition engine. In this method, the equation of motion for the coupled system of piston and cylinder is derived, taking account of three – degree of freedom system of the piston to simulate accurately time of the pressure force and vibratory response. The characteristics under different engine torque conditions of acoustic emission and vibration signals of the compression ignition engine from a baseline test are presented in this work. The purpose of this research is to investigate relation the engine vibration and noise with engine performance parameters (indicated specific fuel consumption, indicated power, indicated pressure and indicated thermal efficiency), and the effect of different pressure forces on vibrations in a small diesel engine. The investigated parameters are indicated engine performance parameters, sound pressure level (SPL) and vibration generated from engine are calculated using cylinder pressure measurements. A MATLAB program is developed to get the pressure forces and vibration amplitudes in the three – dimensions. It was found that the peak amplitude of acoustic emission root mean square (RMS) signals correlating to the impact like combustion related events decreased in general as the engine torque increases. It is also noticed that the peak amplitude of the acoustic emission RMS attributing to indicated specific fuel consumption increases as the engine torque decreases. The calculation performance of the combined system is 76.72% when tested on the validation (theoretical) set and 75.47% on the final test set. The calculated results by MATLAB program show that the pressure forces and the piston vibration amplitudes agree well with measured results with relative true error of 3%.

Keywords: Performance, Noise, Vibration, Diesel engine, Pressure.

دراسة نظرية وعملية لاهتزاز وضوضاء محرك إشعال بالضغط

الخلاصة:

تم في هذا البحث عمل نموذج تحليلي لقوة الضغط واستجابة الاسطوانة الاهتزازي الذي حدث لحركة مكبس اسطوانة محرك ذو إشعال بالضغط. تم في هذه الطريقة اشتقاق معادلة الحركة للنظام الثنائي (المكبس والاسطوانة) ولثلاث محاور لحركة المكبس داخل الاسطوانة وذلك لتخمين وقت قوة الضغط والاستجابة الاهتزازية بدقة . وان الخواص تمت تحت عزم متغير للمحرك لانبعاث الضوضاء وإشارات الاهتزاز لمحرك إشعال بالضغط من خط الأساس للاختبار المعين في هذا العمل. إن الغرض من هذا البحث هو دراسة علاقة اهتزاز وضوضاء المحرك مع معاملات أداء المحرك (الاستهلاك النوعي البياني للوقود – القدرة البيانية – متوسط الضغط الفعال البياني في الاسطوانة والكفاءة الحرارية البيانية)، وتأثير قوى الضغط المختلفة على اهتزازات محرك الديزل الصغير. إن المعاملات المدروسة وهي معاملات أداء المحرك البيانية، مستوى ضغط الصوت SPL والاهتزاز الناتج من المحرك تم حسابها باستعمال قياسات ضغط الاسطوانة. ولقد تم تطوير نظام MATLAB للحصول على قوى الضغط ومديات الاهتزاز في ثلاثة اتجاهات. لقد وجد إن المدى العالي لانبعاث شارات الصوت RMS (مربع متوسط الجذر) مرتبطة بتأثير مجريات الاحتراق التي تقل بصورة عامة بزيادة عزم

المحرك. كما لوحظ أن المدى العالي لانبعاث الصوت لإشارات RMS ينسب إلى الزيادة في قيمة الاستهلاك النوعي البياني للوقود مع نقصان عزم المحرك. إن حسابات الانجاز للنظام المركب كان 76.72 % عندما أجريت التجربة على مجموعة التصحيح (النظري) و 75.47 % على مجموعة الاختبار النهائي. إن النتائج المحسوبة بواسطة برنامج MATLAB لقيم قوى الضغط ومدى اهتزاز المكبس تتفق جيدا مع النتائج العملية المستخلصة، وان نسبة الخطأ النسبية كانت 3%.

INTRODUCTION:

The main drawback of vibration technique is effective when the vibration signal generated by a development fault of an engine is well above the background noise level. Thus, applications of the technique in diesel engine monitoring could be severely hindered by the relatively low vibration energy generated by incipient engine faults and the more dominant high energy noise and vibration from various mechanical events such as combustion, valve opening, closing and auxiliary devices [Lin et al, 2011]. It is known that diesel engines generally deliver less power for a given size and they produce more noise and vibration. Lately, the noise and vibration have been decreasing toward a level that is comparable to that of gasoline engines grace of technological improvements of the fuel injection system [Mihai et al, 2000].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 [Chirag, 2004]. The vibration induced in any machine due to its moving parts is only of lower frequency. But some high frequency vibrations are also present in internal combustion engine due to abnormal combustion of charge (fuel – air mixture). Vibrations produced in diesel engine are mainly in two directions: vibrations in lateral direction and vibrations in longitudinal (axial) direction [Jindal, 2012]. Condition monitoring technology for diesel engines has been developed in recent years. Diesel engine manufactures and third – party vendors have developed engine monitoring techniques to determine performance of diesel engine operation. A typical monitoring system is applied microprocessor based instrument with analysis software to monitor engine combustion characteristics, intake / exhaust valve operation, piston motion, cylinder and liner conditions[Songpon et al, 2009]. Several researchers worked on the some field, but with several point of views. Among the diagnostic techniques applied to internal combustion engines, those based on the analysis of accelerometer data have earned a greater success. Chun measured oscillations at the upper part of the cylinder block center for knock in S.I.E (spark ignition engines) [Chun et al, 1994]. Zurita rebuilt in – cylinder pressure history through the signal provided by an accelerometer placed externally [Zurita et al, 1999]. Antoni used vibrations to indicate malfunctioning [Antoni et al, 2002], which has been proved by Carlucci that injection pressure and injected quantities, over an energy release threshold, really affect the vibration signals in a peculiar way, injection timing affects the engine block vibration in a less evident way [Carlucci et al, 2005]. Gideon used vibration measurement to identify malfunctioning in a multi cylinder engine [Gideon et al, 2004]. The cylinder pressure developed within an internal combustion engine can be considered to be the pulse of the engine. In conjunction with the periodic events such as inlet and exhaust valve operations and fuel injection timings, it provides valuable information about the combustion characteristics of the engine [Chun et al, 1994]. Many investigations have been done concerning the relation of the diesel engine combustion noise to the engine operating and design parameters [Sun et al, 2007]. Typical frequencies seen in vibration power spectra measured on the engine block include low frequency harmonic components at firing frequency and its multiplies from 1Hz up to 400Hz,bending frequencies in the range of 400 - 800 Hz and combustion chamber resonances in the range from 800 – 4000 Hz [Ren et al, 1999].

The aim of this research is to investigate relationship between the diesel engine vibration and noise with engine performance parameters such as (indicated power, indicated specific fuel consumption, indicated thermal efficiency and indicated mean effective pressure), and the effect of pressure force (which is produced from by ignition of air – fuel mixture in the cylinder) on vibration and sound pressure level (SPL) in a small diesel engine. The calculation performance of the combined system was tested on the validation (theoretical) set and on the final test set. Also comparison with effectiveness and sensitivity of different vibration and noise analysis was done.

THEORY:

In a direct injection, there are two main excitatory forces which are the unidirectional force due to combustion and reversible forces which are mechanically induced. Due to the large heat release during combustion, the pressure rise is extremely rapid in this zone and causes oscillations in the cylinder pressure [Ren et al, 1999].
The pressure force fluctuations, Fp (t) as shown in **Fig. 1** , cause the engine structure to vibrate and this response could be modeled as a linear spring – mass system of mass (M), damping coefficient (C), and dynamic stiffness (K) in degree of freedom dimensional system as in [Ren et al, 1999].

$$Ma + Cv + Kx = Fp\,(t) \tag{1}$$

Where : x: displacement
v : velocity
a : acceleration

In loading the engine, the equation of motion of diesel engine which is used in this research in three degree of freedom DOF system becomes:

$$m\ddot{x} + C\dot{x} + Kx = Fx(t) \tag{2}$$

$$m\ddot{y} + C\dot{y} + Ky = Fy(t) \tag{3}$$

$$m\ddot{z} + C\dot{z} + Kz = Fz(t) \tag{4}$$

$$\begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & m \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{z} \end{bmatrix} + \begin{bmatrix} C & 0 & 0 \\ 0 & C & 0 \\ 0 & 0 & C \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{y} \\ \dot{z} \end{bmatrix} + \begin{bmatrix} K & 0 & 0 \\ 0 & K & 0 \\ 0 & 0 & K \end{bmatrix} \begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{Bmatrix} Fx \\ Fy \\ Fz \end{Bmatrix} \tag{5}$$

Eq. (5) can be reduced to unit matrix as:

$$m\begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \\ \ddot{z} \end{bmatrix} + C \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{y} \\ \dot{z} \end{bmatrix} + K \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{Bmatrix} Fx \\ Fy \\ Fz \end{Bmatrix} \tag{6}$$

Where unit matrix is : $I = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$

Substituting this unit matrix I in eq.(6):

$$m.I.\begin{bmatrix}\ddot{x}\\ \ddot{y}\\ \ddot{z}\end{bmatrix} + C.I.\begin{bmatrix}\dot{x}\\ \dot{y}\\ \dot{z}\end{bmatrix} + K.I\begin{bmatrix}x\\ y\\ z\end{bmatrix} = \begin{Bmatrix}Fx\\ Fy\\ Fz\end{Bmatrix} \tag{7}$$

In order to solve eq.(7), the following assumptions are used:

$x = X \sin \omega t$

$y = Y \sin \omega t$

$z = Z \sin \omega t$

Substituting these values in eq.(7) and rearranging in terms of unit matrix I:

$$- m.I.\begin{bmatrix}X\\ Y\\ Z\end{bmatrix}.\ \omega^2 \sin \omega t + C.I.\begin{bmatrix}X\\ Y\\ Z\end{bmatrix}.\omega.\cos \omega t + K.I.\begin{bmatrix}X\\ Y\\ Z\end{bmatrix}\sin \omega t = \begin{Bmatrix}Fx\\ Fy\\ Fz\end{Bmatrix} \tag{8}$$

Rearranging eq.(8) becomes as:

$$(C.\cot(\omega t) - m.\omega + \frac{K}{\omega})\begin{vmatrix}X\\ Y\\ Z\end{vmatrix}.\ \omega \sin(\omega t).I = \begin{Bmatrix}Fx\\ Fy\\ FZ\end{Bmatrix} \tag{9}$$

MATLAB PROGRAM:

Eq. (9) is used to determine vibration pressure force at different engine torque range (0 N.m to 10 N.m) and at engine speed range N = 1100 – 2500 r.p.m or ω =115 – 262 rad/s is shown in following program, where:

$$\omega = \frac{2\pi N}{60}$$

Therefore by using experimental results at engine torque T = 0 N.m.

```
>> % Calculate vibration pressure force at T = 0 N.m
>>  t = 385.7;
>>  m = 35 ;
>>  C = 47 ;
>>  K = 344 ;
>> ω = 115 ;
>>  x = 10.2 ;
>>  y = 27 ;
>>  z = 25.3 ;
>> B = C * cot (ω * t) – m* ω + (K / ω)  ;
>> D = B * [ x; y; z];
>>  J = (ω / sin (ω * t))* [ 100; 010; 001]* (1/10^6);
>>  F = J*D
>>  F
        80.2136                                    Fx = 80.2
        134.4071         ------->     it means     Fy = 134.4
        243.6315                                   Fz = 243.6
```

The results of MATLAB program is represented in **Tables 1, 2 and 3**. This program is used to determine vibration pressure forces in three axes and to calculate theoretical vibration amplitude in three coordinates. In automotive applications, it is generally assumed that:

a. The engine is decoupled from the load torque.
b. The crankshaft is sufficiently rigid.
c. The engine torque is changed.

The assumption that the engine is decoupled from the load is usually valid in automotive applications, since the nominal rotational speed of the engine is typically significantly higher than the first natural frequencies of the crankshaft.

EXPERIMENTAL SETUP:

The tests were carried on the small internal combustion diesel engine test bed at the research lab of automotive workshop in Institute of Technology - Baghdad. The engine is four stroke, single cylinder naturally aspirated compression ignition engine combined with scale data acquisition system with following specifications:

Engine Type	TD202
Fuel Type	Diesel
Displacement(Engine Capacity)	232cm3
Compression Ratio	22:1
Bore Diameter	69 mm
Stroke	62 mm
Absolute Maximum Power	3.5 kW (4.8 hp) at 3600 rev/min
Continuous Rated Power	3.1 kW at 3000 rev/min
Maximum Speed	3600 rev/min

The present system is digitally controlled, and capable to store the experimental data instantaneously. The system as shown in **Fig. 2** provides a facility to conduct engine performance tests at different engine speeds (1100 – 2500) r.p.m. Two signals acquired using Lab / View program as vibration and noise signals. An aluminum clamp was used to hold - Bruel (PCB) & Kjoer (JCP) piezoelectric accelerometers at the three axis x, y and z are used. It can be seen that the vibration signals show other small and unclear events, which may come from other sources inside the engine such as ignition process, combustion process, piston movement and auxiliary equipment. The noise measurements were effected in accordance with instrument STAS 7150 – 77, concerning measurement method for the noise level. The microphone being mounted in the position where the engine in working conditions. SLM (Sound Level Meter) was calibrated and the calculating errors were removed. It was calibrated to 75 dB(A) on the lower side and 100 dB(A) on the upper side. The engine connected to hydraulic type dynamometer for loading. These signals were interfaced to the computer. Provision was also made available for interfacing indicated specific fuel consumption, indicated power, indicated pressure and load measurements. The compression ratio was constant (C.R = 22:1) and the engine torque was varied from 0 N.m to 10 N.m., fuel measuring unit, pressure measurement, time measurement, process indicator and engine indicator were used in this test. The sensors are type (Kistler) high temperature pressure sensor and an (PCB) JCP piezoelectric accelerometers. Signals acquired by these sensors at the normal engine signal running condition various engine torques are analyzed for the characterization and to establish the baseline information of the diesel engine. This baseline information would be utilized for fault detection in the fault simulation test of the diesel engine. Cylinder pressure is measured using piezoelectric pressure transducer as shown in **Fig. 3**. The sensing element is located in a hole drilled through the cylinder head into the combustion chamber. The sensing element consists of a metal diaphragm which deflects under pressure, and is connected by an optical fiber to a light source and a detector. The intensity of the reflected light is converted to a voltage which is proportional to the pressure.

RESULTS AND DISCUSSION:

Fig. 4 shows the relationship between pressure force and engine speed. It is seen that the engine speed has no effect on the resulted pressure force. From **Figs. 5 & 6** the engine structure get excited with lower order harmonics which has higher vibration amplitudes, and the pressure is increased parallel to the vibration amplitudes at engine torques 0 N.m and 10 N.m., also, it can be seen from these figures, the effect of different engine torques on the total engine vibration amplitudes. **Fig. 7** shows the stability zone decreases as RMS of total vibration amplitude and mean cylinder pressure are increased, so it is noticed the minimum zone of vibration at torques 2 N.m and 8 N.m. The calculated results by MATLAB program of theoretical vibration amplitudes agree well with experimental results with relative true error was 3%. In this research, the diesel engine speed used is in the range of (1100 – 2500) r.p.m. The changes in indicated mean effective pressure (IMEP) and total engine vibration amplitudes would be proportionally slower in the slow engine speed, thus making it easier to record data with a much higher resolution. **Fig. 8** shows that the small amount of heat energy is produced in the chamber of the compression ignition engine, which needs low thermal efficiency with low RMS vibration and then thermal efficiency is increased with however, increasing RMS vibration. The relationship between sound pressure level SPL and engine speed, is represented by **Fig. 9**, noise was in the range of 80 – 88 dB, it means not so high noise is produced from the engine, because of SPL generated from cylinder pressure is due to different rates of fuel burning, pressure wave generated with different process inside engine cylinder due to piston motion, fluctuations in heat release rate, mixing process of burnt and unburnt gases, compression and expansion of gases inside the cylinder due to piston motion. Noise SPL signals are due to their respective operating frequency ranges. Therefore noise SPL is increased clearly with increasing engine speed. Important vibrations are generated by the forces of pressure in **Fig. 10** which developed in the cylinder block, that depend on the mass and dimensions of the different moving engine parts. The forces of pressure and their torque can be generally, entirely or partly balanced. The engine sound pressure level SPL varies with loading torque variations with 2 – 4 dB, but at the same time working characteristics vary in large intervals. It is concluded, that engine torque has nearly no effect on sound pressure level SPL. In **Fig. 11** it is obviously noticed that SPL is decreased with increasing indicated power. **Fig. 12** shows typical curve, which indicated specific fuel consumption ISFC represents the maximum value at low and high sound pressure level SPL. While at the mean value of SPL, ISFC was reached the minimum value. At the same time ISFC was decreased with engine torques increasing. ISFC at the point in **Fig. 12** meets the upper value of the thermal efficiency at the same value of SPL generated from combustion process. **Fig. 13** shows typical curve, the thermal efficiency at low speed 1100 r.p.m begins to increase as SPL is increased till 1800 r.p.m, which represents ideal speed of the diesel engine. However, that gives a high thermal efficiency, because of the combustion process is completed of the fuel in the combustion chamber. While SPL is continuous to increase till 2500 r.p.m, which gives the low value of thermal efficiency. Comparison between the averaged noise RMS and vibration RMS at various engine torque condition, revealed the noise is a better technique than vibration method for condition monitor of diesel engines due to its ability to produce high quality in a noisy diesel engine environment. Engine torque influences on the noise level which appears in the combustion period for a diesel engine is relatively small. The aim in ensemble creation is that of finding a set of nets that exhibit a minimum number of overlapping errors tested on a validation set. In the light of research on combining nets, an attempt was made to improve performance by combining a set of nets each trained respectively on pressure, vibration or fused pressure and vibration data.

In this work, it is considered only the calculation method of performance of the combined system was 76.72% when tested on the validation (theoretical) set and 75.47% on the final test set as shown in **table 4** which contains the performance of the final system comprising of sets trained on pressure, vibration and fused (calculated) data. The final system is correctly classified 317 out of 420 test cases with a confidence level of 62,71%, a level of generalization was 76.72%.

The suggestion to the next research is the necessity to develop researches in order to determine the best medium pressure, which satisfies the demand to increase the power performance and to decrease the vibration and noise levels.

CONCLUSIONS

It can be concluded the followings:

1- The characteristic of pressure force determine the rate of increase with engine speed.

2- Combustion system determines the rate of increase of noise with engine speed about 2.5 dB. It can be seen that at low speeds there is a possibility of reducing noise by as much as 10 dB(A) by smoothing the development of the pressure force.

3- The ideal engine speed was 1800 r.p.m which it considers reasonable efficiency of the engine about 32.8% and at this speed can be also get the interested conversion of vibration zone to stable zone with accepted fuel consumption.

4- Diesel engine noise is independent of the indicated power produced, and the indicated thermal efficiency.

5- The best calculation performing showed a level generalization performance of 57.6% pressure and 57.70% vibration respectively.

6- SPL generated from cylinder pressure increases with the increasing of engine speed due to increase of pressure force, heat release outer and turbulence intensity in engine cylinder.

Table (1) represents MATLAB results, experimental and theoretical data at engine torque T = 0 N.m

Speed N [r.p.m]	Angular Speed ω [rad/s]	Freq. f [Hz]	Indicated thermal effic. η [%]	Exp. indicated mean effective pressure p [bar]	Indicated Power I.P [kW]	Indicated Specific Fuel Consumption I.S.F.C [kg/kW.hr]	Exp. Total Vibration Amplitude in 3- dim. [m/s²]	Theor. Total Vibration Amplitude in 3- dim. by MATLAB [m/s²]	Theor. Total Pressure Force in 3- dim. Fp by MATLAB [N]	Sound Pressure Level SPL [dB(A)]	Engine Torque T [N.m]	Time t [sec]
1100	115	18.3	27.96	2.04	0.45	0.33	38.3	37.93	17.9	83	0	325.7
1150	120	19.1	28.22	2.12	0.56	0.327	47.3	47.52	23.9	83.5	0	301
1500	157	25	31.87	2.21	0.64	0.31	51.4	51.18	21.9	84	0	261
1750	183	29.1	32.08	2.37	0.81	0.287	70	70.86	95.8	86	0	222
2000	209	33.2	34.3	2.43	0.94	0.266	67.7	72.66	118.8	85.5	0	178.9
2250	236	37.2	33.5	2.51	1.07	0.276	70	57.35	141	86	0	164.1
2500	262	41.7	36.31	2.95	1.43	0.259	77.8	70.84	299	88.5	0	133.1

Table (2) represents MATLAB results, experimental and theoretical data at engine torque T = 4 N.m

Speed N [r.p.m]	Angular Speed ω [rad/s]	Freq. F [Hz]	Indicated thermal effic. η [%]	Exp. Indicated mean effective pressure P [bar]	Indicated Power I.P [kW]	Indicated Specific Fuel Consumption I.S.F.C [kg/kW.hr]	Experimental Total Vibration Amplitude in 3- dim. [m/s²]	Theor. Total Vibration Amplitude in 3- dim. by MATLAB [m/s²]	Theor. Total Pressure Force in 3- dim. Fp by MATLAB [N]	Sound pressure level SPL [dB(A)]	Engine Torque T [N.m]	Time t [sec]
1100	115	18.3	32.05	5.20	1.12	0.288	32.6	39.09	33	82	4	165.2
1150	120	19.1	32.12	4.89	1.19	0.2866	40	37.11	21	81.5	4	181.5
1500	157	25	32.21	4.23	1.26	0.2857	46.2	46.23	42.5	81	4	158.8
1750	183	29.1	28.68	4.17	1.4	0.321	38.8	31.98	96	82	4	129
2000	209	33.2	32.224	4.38	1.68	0.2863	62.8	65.72	143.3	83.5	4	117.5
2250	236	37.2	35.8	4.89	2.02	0.258	70.3	61.19	169.4	84.5	4	112.6
2500	262	41.7	32.67	5.22	2.5	0.282	93.7	83.78	190.6	85.5	4	92.9

Table (3) represents MATLAB results, experimental and theoretical data at engine torque T = 10 N.m

Speed N [r.p.m]	Angular Speed ω [rad/s]	Frequency f [Hz]	Indicated thermal efficiency η [%]	Exp. indicated mean effective pressure P [bar]	Indicated Power I.P [kW]	Indicated Specific Fuel Consumption I.S.F.C [kg/kW.hr]	Exp. Total Vibration Amplitude in 3- dim. [m/s²]	Theor. Total Vibration Amplitude in 3- dim. by MATLAB [m/s²]	Theoretical Total Pressure Force in 3- dim. Fp by MATLAB [N]	Sound pressure level SPL [dB(A)]	Engine Torque T [N.m]	Time t [sec]
1100	115	18.3	28.85	6.66	1.59	0.32	42	42.29	19.5	81	10	95.1
1150	120	19.1	27.51	7.17	2.02	0.332	49.3	52.89	25	81.5	10	81.4
1500	157	25	26.6	7.81	2.23	0.34	55.2	53.94	48.6	82	10	62.5
1750	183	29.1	26.74	7.58	2.6	0.345	67.3	64.82	80	83	10	53.9
2000	209	33.2	29.76	7.95	3.19	0.31	79.6	77.72	244.4	84	10	48.9
2250	236	37.2	28.75	7.97	3.48	0.321	85	85.44	247	86.5	10	43.3
2500	262	41.7	28.53	8	3.72	0.323	94.1	92.84	860.3	87	10	40.2

Table (4): performance of final system

Data Source	performance	
	Experimental	Validation (theoretical)
Pressure (p_i)	50 – 87%	57.61%
Vibration (v_j)	57.70%	56.78%
P&v ($p_i + v_j$) fused (calculated)	62.71%	62.28%
Ensemble performance	75.47%	76.72%

Table (5): represents the relation between SPL and vibration at torque T = 4 [N.m]

Engine Speed N [r.p.m]	Exp. Total Vibration Amplitude in 3- dim. [m/s²]	Theor. Total Vibration Amplitude in 3- dim. [m/s²]	Sound Pressure Level SPL [dB(A)]
1100	32.6	39.09	82
1150	40	37.11	81.5
1500	46.2	46.23	81
1750	38.8	31.98	82
2000	62.8	65.72	83.5
2250	70.3	61.19	84.5
2500	93.7	83.78	85.5

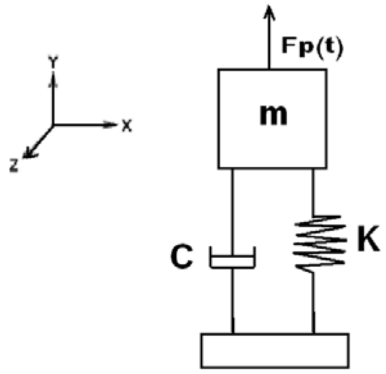

Fig. (1) : Mechanical model of piston and cylinder.

Fig.(2): Photograph of TD202 Bench

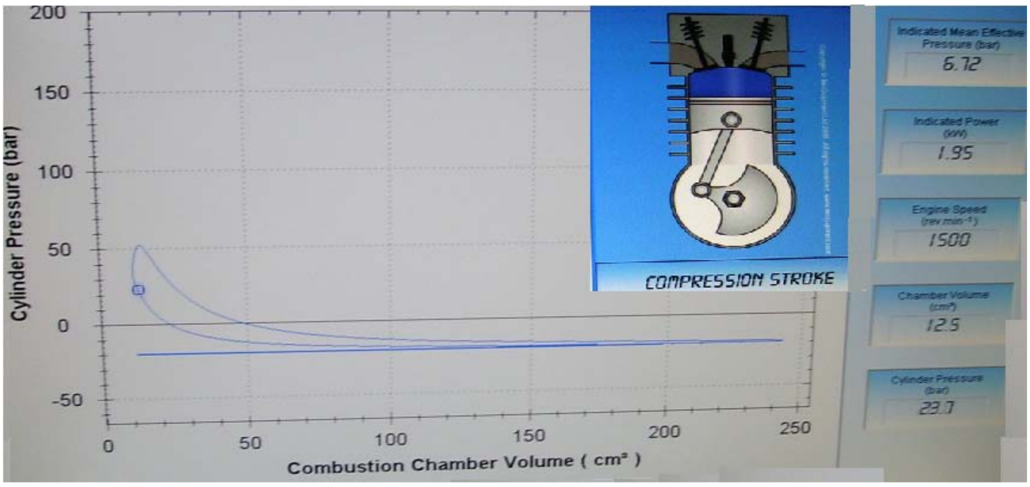

Fig. (3): Cylinder Pressure Traces of all Conditions Vs Combustion Chamber Volume

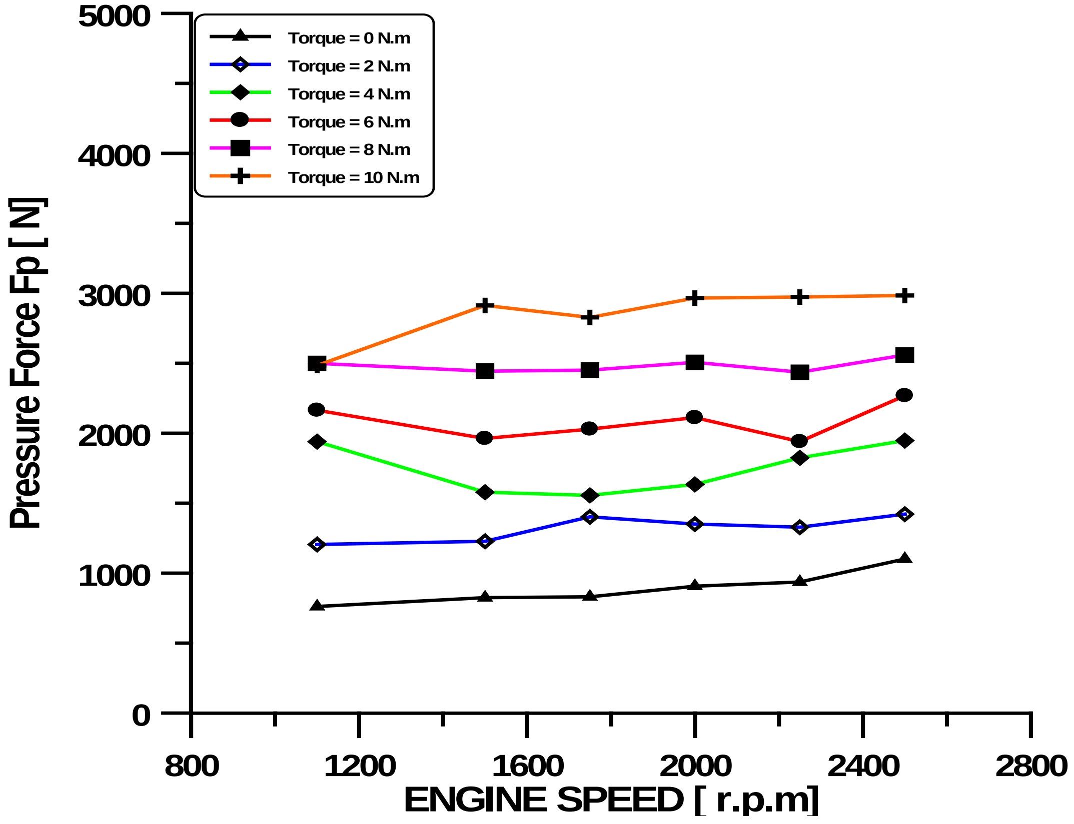

Fig. (4): Pressure Force for Different Engine Torques

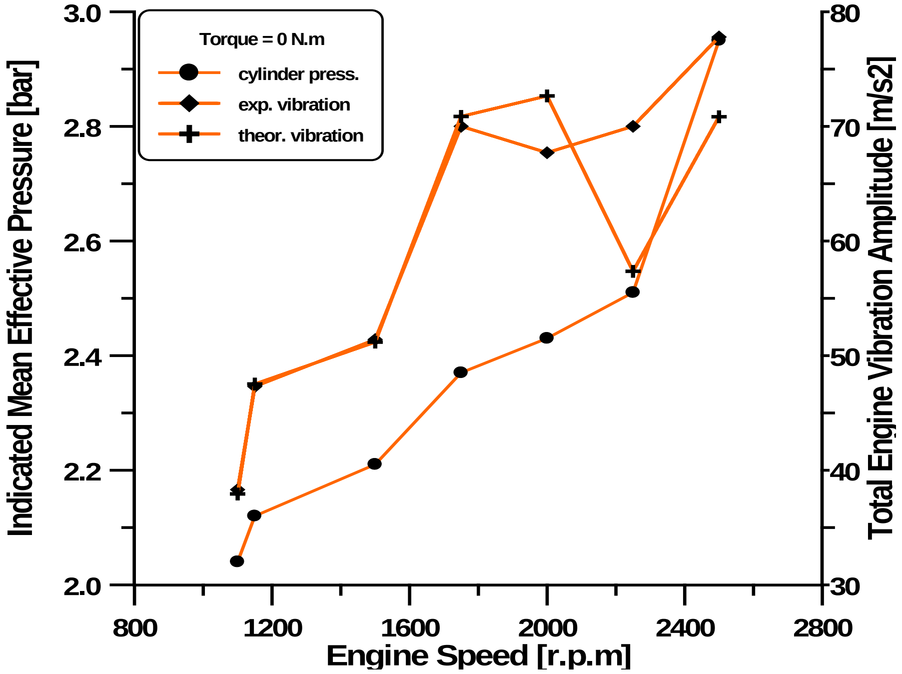

Fig. (5): Cylinder Pressure, Experimental and Theoretical Vibration Amplitude for different Engine Speeds

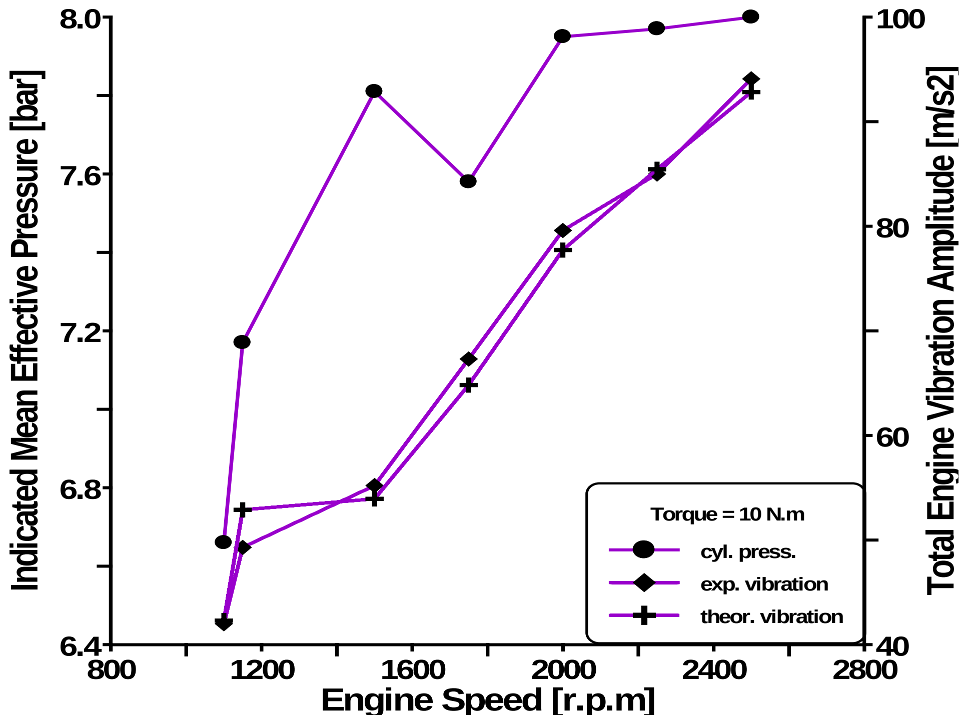

Fig. (6): Cylinder Pressure, Experimental and Theoretical Vibration Amplitude for Different Engine Speeds

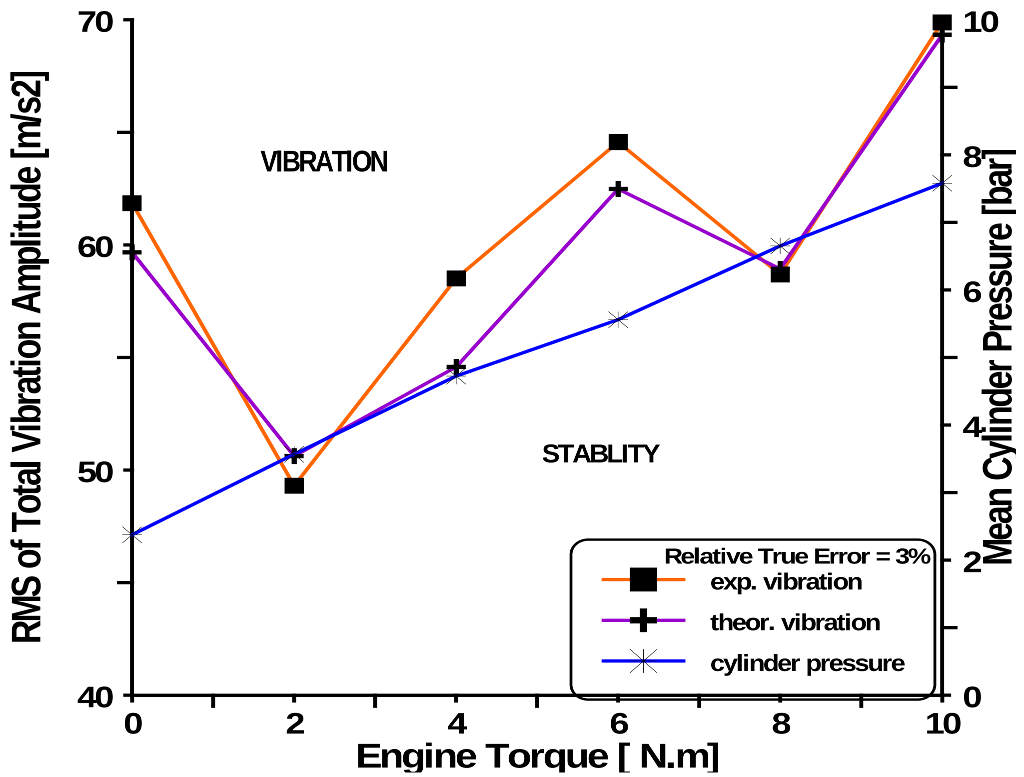

Fig. (7): RMS of Total Vibration Amplitude and Mean Cylinder Pressure for Different Engine Torques

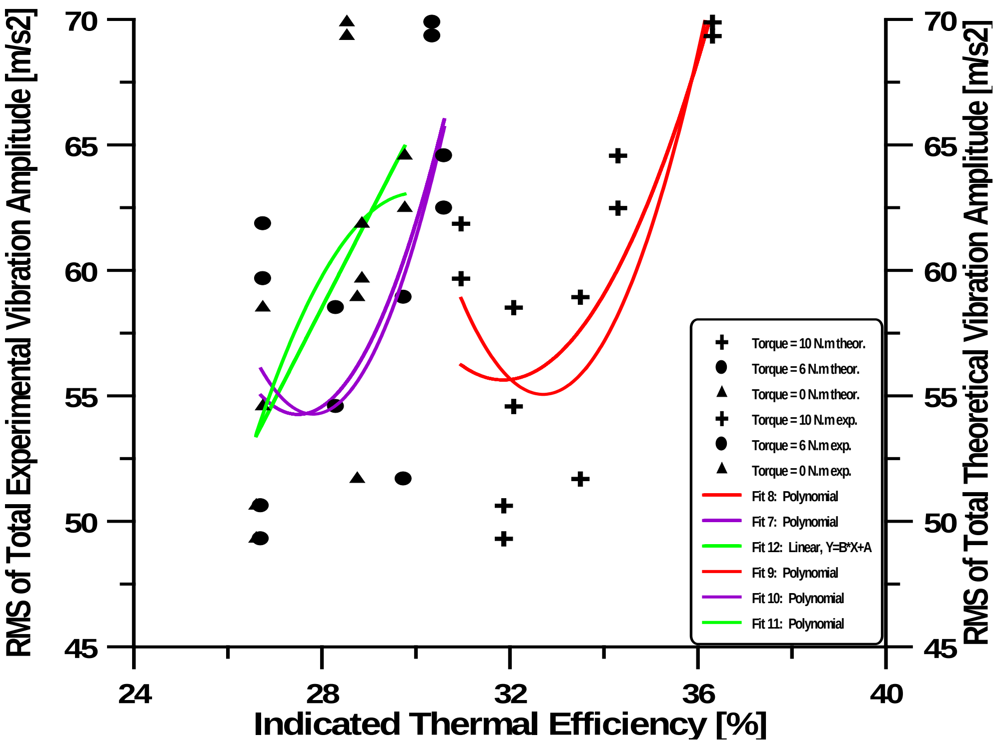

Fig. (8): Indicated Thermal Efficiency, RMS of Total Experimental and Theoretical Vibration Amplitude for Different Engine Torques.

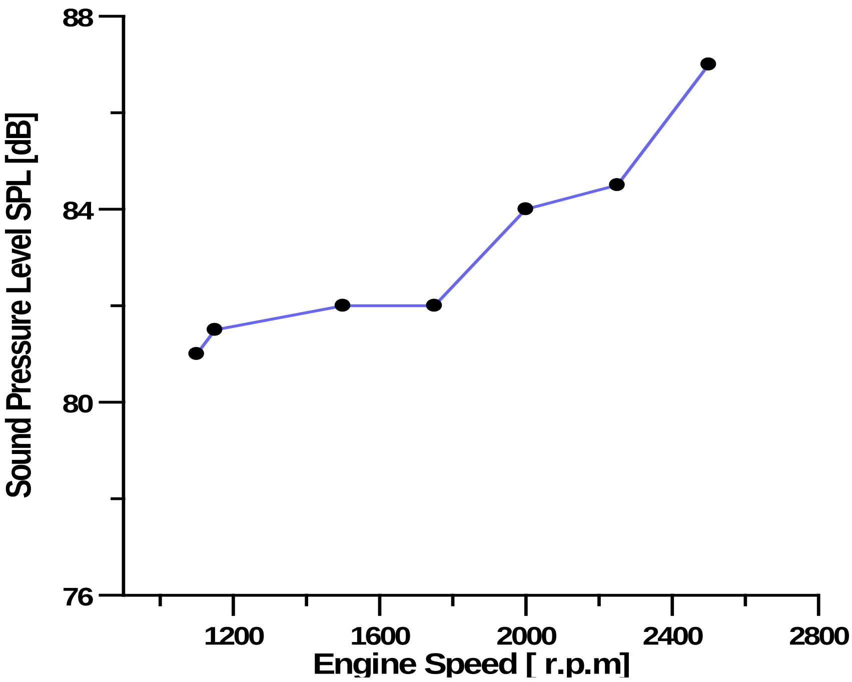

Fig. (9): Sound Pressure Level for Different Rotational Engine Speed at Torque T = 6 N.m

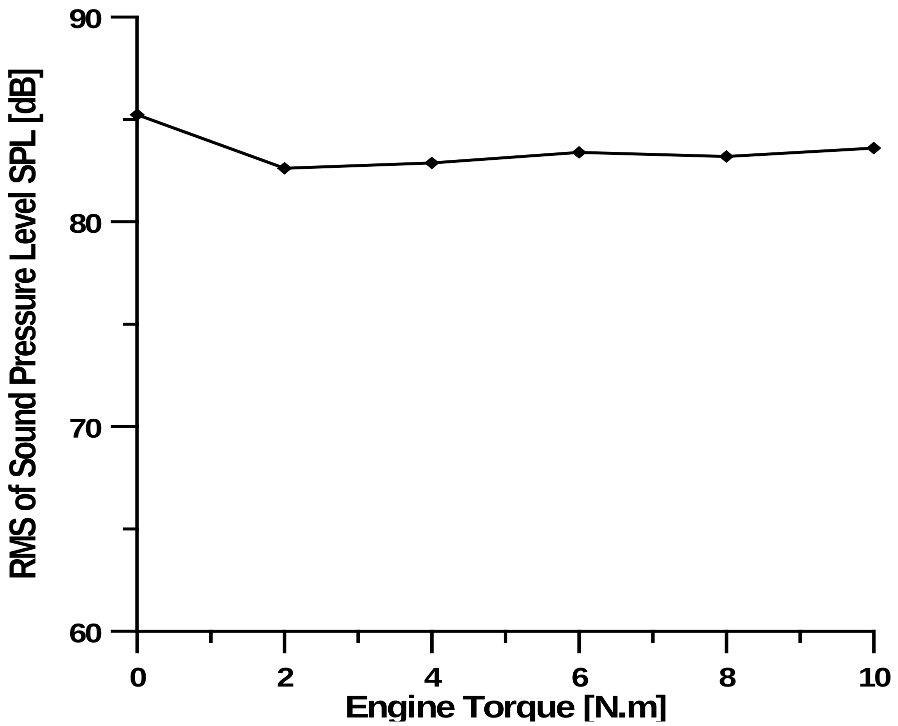

Fig. (10): RMS of Sound Pressure Level for Different Engine Torques

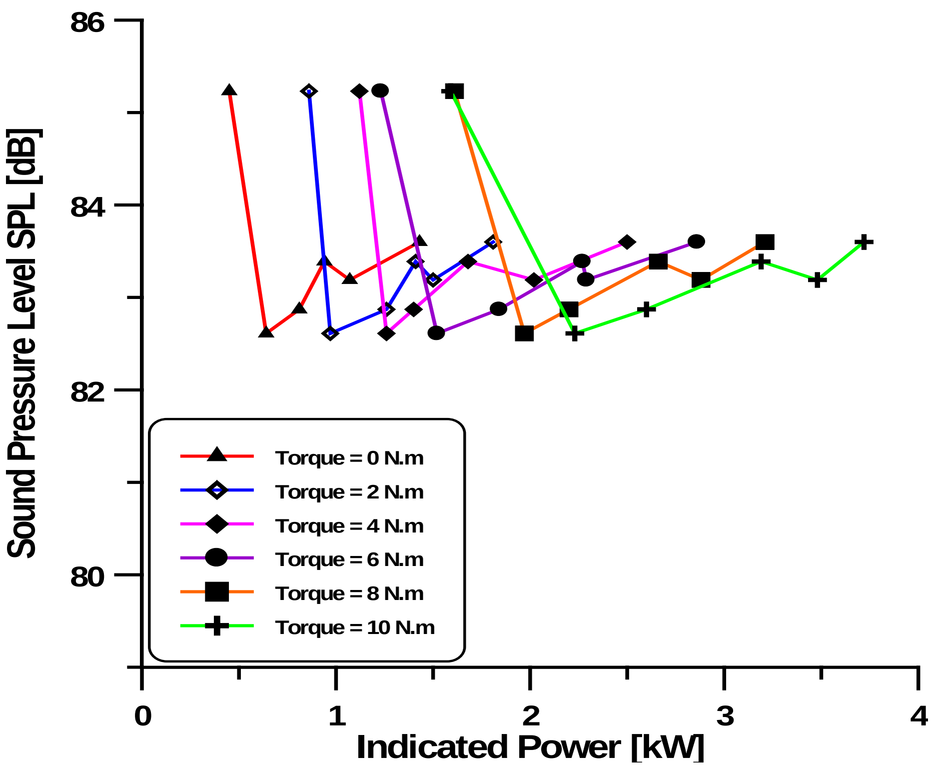

Fig. (11): Sound Pressure Level and Indicated Power for Different Engine Torques.

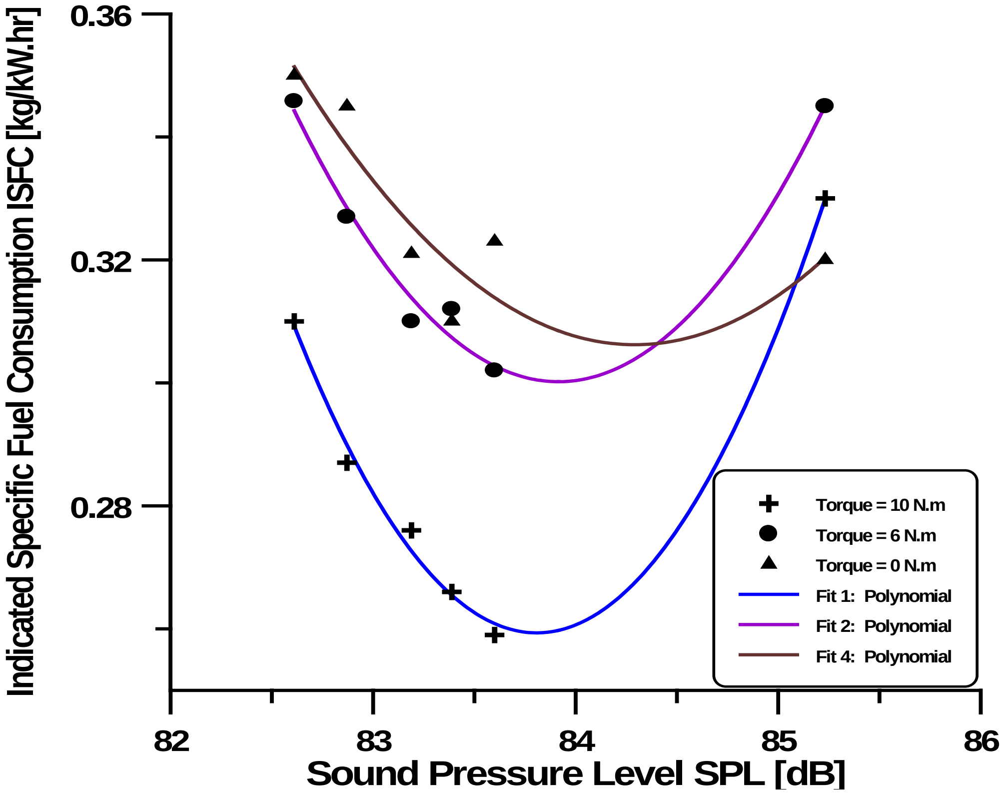

Fig. (12): Indicated Specific Fuel Consumption and Sound Pressure Level for Different Engine Torques.

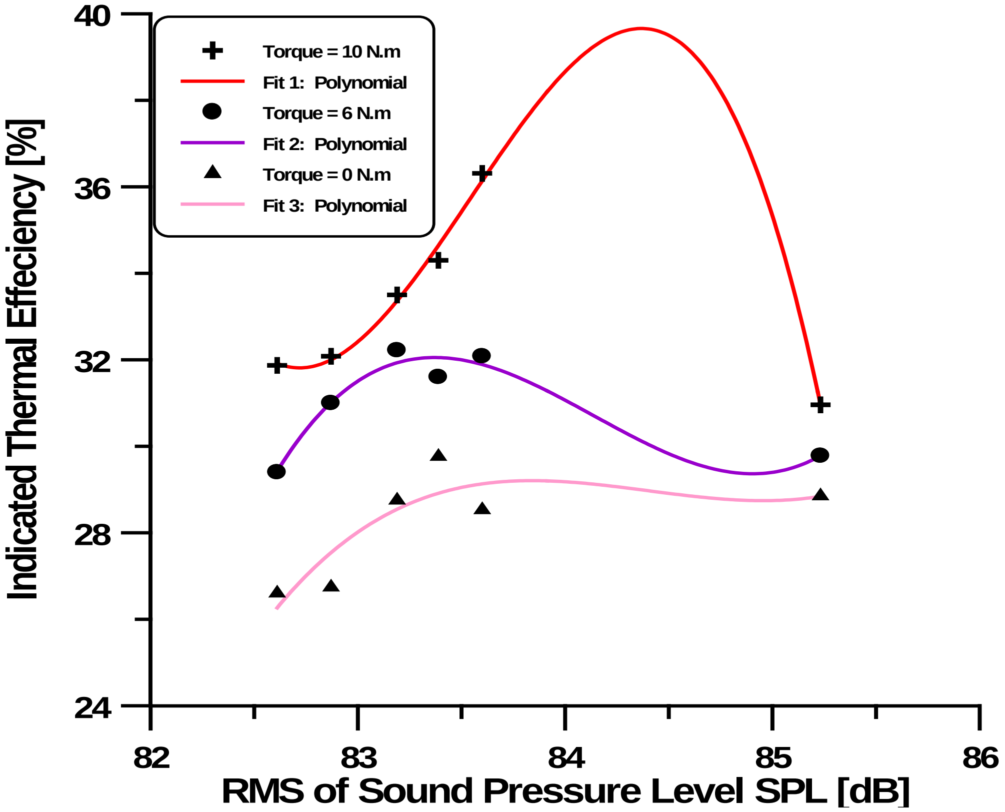

Fig. (13): Indicated Thermal Efficiency and RMS of Sound Pressure Level for Different Engine Torques

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