# 745. Analysis of block vibrations induced by combustion chamber pressure in a diesel engine

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**Abstract.** The paper describes an experimental methodology devised to study engine block displacement of internal combustion engine in the radial direction due to combustion force. This force produced in the cylinder varies with the chamber profile since flow parameters are influenced by the combustion chamber profile. Combustion effectiveness fluctuates depending upon flow parameters. To correlate between combustion and injection and to detect faults in the injection system it is necessary to supervise the combustion in the individual cylinders. This can be accomplished by evaluating the crankshaft speed at the flywheel. Speed is directly related to combustion by means of indicated pressure and torque. Different combustion chamber profiles were taken for the analysis along with speed and load as the design variables. The displacement and time-domain frequency for all the profiles were compared. The results provide displacement induced by the combustion pressure, uncertainty in combustion processes and nonlinear vibration of the engine block. This new approach in engine parameter design gives insight on the vibrational processes in the case of different chamber profiles as well as sources of noise in the diesel engine.

**Keywords:** combustion pressure, piston profiles, displacement, peak amplitude.

**Nomenclature**

$A$, $B$ Lame's constants
$A'$ Vibration constant
$\Lambda$ Cross sectional area of cylinder ($\pi D^2/4$)
$D$ Bore diameter
$E$ Young's modulus
$F_r$ Forces due to reciprocating masses
$F_v$ Forces due to valve actuations
$F_{kc}$ Forces due to kinematic chain
$IP$ Indicated power
$P$ Indicated mean effective pressure
$R_1$ Inner radius of engine block
$R_2$ Outer radius of engine block
$r$ Radius in between $R_1$ and $R_2$
$u$ Amplitude
$\varepsilon$ Strain
$\mu$ Poison ratio
$\omega$ Natural frequency
$v$ Excited frequency
$\delta r$ Displacement
$\varsigma$ Damping ratio
$\sigma_a$ Axial stress
$\sigma_c$ Circumferential stress
$\sigma_r$ Radial stress

<u>Figure notations</u>

DC Double concave
HS Hemispherical
EL Elliptical
ED Ellipsoid
N11100 rpm
N21200 rpm
N31300 rpm
N41400 rpm
N51500 rpm
E Experimental
T Theoretical

## 1. Introduction

The main objective of this work is to investigate the in-cylinder pressure contribution to engine block vibration and the impact of piston profile on the vibration. The measurement of in-cylinder pressure has been object of the study from the beginning of the internal combustion engine. Tremendous amount of useful information can be extracted from the cylinder pressure signal for engine combustion control. However, physical cylinder pressure sensors are undesirably expensive and their health needs to be monitored for fault diagnostic purpose as well [1]. So two neural network-based independent cylinder pressures related variable estimators were developed and verified at steady state. The development of combustion in diesel engines is strictly dependent on combustion chamber and injection parameters. Vibration signal analysis and mitigation of vibration amplitude are important research tasks since 18th century and rely upon different methods and devices. This type of work was carried out for all the machinery and equipment. The impact of inlet, exhaust, and fuel injection parameter has been presented. Short term Fourier transform is used to identify different sources of internal combustion engine block vibration from single point acceleration measurements taken with a commercial knock sensor [2]. Even slight modification or change in the engine parameter will induce vibration, and abnormality such as knocking, combustion instability, thermo-acoustic vibration and squat performance. There is a possibility of using engine block vibration as a mean diagnostic tool for the combustion modifications induced by the injection parameters, and accelerometer positioned at the engine block has been analyzed [3]. Piston slap is a cause of complex transient vibration response related to the impact excitation inside the engine, and the piston-slap impact with the slap-induced vibration was correlated with the engine dynamic behavior simulation and working condition monitoring [4]. Some origin of mechanically induced noise caused by various forces which act on the moving parts of the engine to accelerate them across their running clearances and thus cause mechanical impacts. One of these sources is piston slap and dominates in high-speed medium engines [5]. Analytical treatment was conducted to investigate piston motion in a diesel engine and a computer program was written to predict optimum designs for high mechanical efficiency and low noise and vibration excitation due to offset settings of the crankshaft, piston-pin, and piston centre of gravity [6].

Excitation vibration of the engine due to piston slap, friction, speed, injection parameter, and load condition are analyzed in various methodologies. One is impact excitations inside the engine are reasonably analyzed, based on an analytical model of the non-stationary engine vibration. At this juncture time-varying transfer properties was developed and discussed in details on its time domain and time-frequency domain characteristics [7]. Even Least mean squares algorithm with a deflation method is used to separate and identify several vibration sources [8]. In the same way a diagnostic methodology for internal combustion engines combustion was proposed by means of non-invasive measurements on the cylinder head, such as acoustic and vibration related to the internal indicated mean effective pressure [9]. Internal variables are studied using recurrence plots, recurrence quantification analysis and continuous wavelet transform [10]. Similarly, a practical application of nonlinear autoregressive moving average polynomial models with exogenous inputs technique was proposed to model pressure dynamics inside the cylinder of a direct injection compression ignition engine [11]. Macian proposed different alternative detection principles for the detection of slight unevenness between cylinders in the injection process for a turbocharged diesel engine, and for that the selected parameter are the instantaneous exhaust manifold pressure, the instantaneous turbocharger speed and the mean temperature at the exhaust cylinder ports [12]. In-cylinder pressure derivatives are also used to detect the combustion conditions. Sudden changes in the chamber pressure have amplified by the pressure derivative, and related to thermodynamic phenomena within the cylinder [13]. From the analysis of acoustic signals and vibration signals, an investigation of the noise source identification in a diesel engine was presented, and the noise sources of engine front were identified [14]. The decision of number of cycles to be measured is performed on the basis of the cycle-to-cycle standard deviation of the signal, and the spectrum analysis allows the definition of cut-off frequency for the filter design. In general, lower number of cycles is needed in diesel engines due to its lower cycle-to-cycle dispersion [15]. Furthermore, the fluctuations of effective pressure for different fuels are studied [16]. Detailed multi-body numerical nonlinear dynamic model of a single cylinder internal combustion engine has been formulated and comprehensive noise, vibration, and harshness investigation of the engine was performed [17]. The vibrational characteristics of an internal combustion engine crankshaft have vital role for cylinder health diagnostics.

A crank-angle domain numerical model of the crankshaft dynamics for a six cylinder industrial diesel engine is adopted to establish the effects of continuous low-power production in individual cylinders of a multi-cylinder engine [18]. Minimization of the transmission of engine vibration in the case of a mounted four cylinder engine is performed by choosing the individual balancing of masses and associated phase angles as the design variables [19]. Element model of different internal combustion engine is also a possible one to analyze the characteristics vibration of individual components. In the above survey authors are taken so many parameters to moderate the engine vibration, and to formulate new method for characterizing the engine vibration phenomena. Energy around 3 kHz and 4 kHz can be linked to common rail injection signature to accelerometer signature; it was discussed for the signal transfer [20]. In order to select the frequency bandwidth of interest, we combine the time frequency analysis with the coherence function analysis. The value of the coherence function between the knock signal and the cylinder pressure gives frequencies, where the information of combustion can be extracted from the vibration trace [21]. From the survey it is concluded that none of them used integral part of engine to mitigate vibrations. The combustion chamber profile is intact for moderating vibration or the thermo-acoustic characterization. So the author considered different piston profile and the corresponding vibration phenomena for investigation.

## 2. Time-frequency analysis

The most commonly used device for vibration measurement is the piezoelectric accelerometer, which gives an electric signal proportional to the vibration acceleration. This signal can readily be amplified and analyzed. The engine was treated as elastic system with effective damping system. The engine was supported by an elastic system of stiffness $k$ and effective viscous damping coefficient $c$. The amplitude $u$ is given as:

$$u = A'\left(\nu/\omega\right)^2 / \sqrt{\left(\left(1-(\nu/\omega)^2\right)^2 + 2\varsigma(\nu/\omega)^2\right)} \quad (1)$$

The engine in-cylinder pressure correlation with the displacement of the engine block in the radial direction was taken from the following theoretical Lame’s [22] equations using the in-cylinder pressure/indicated mean effective pressure $P$ [23], inner radius $R_1$, outer radius $R_2$ and at any radius $r$:

$$A = PR_1^2 / (R_1^2 - R_1^2) \quad (2)$$

$$B = PR_1^2 R_2^2 / (R_2^2 - R_1^2) \quad (3)$$

$$\sigma_c = (B / r^2) + A \quad (4)$$

$$\sigma_r = (B / r^2) - A \tag{5}$$

$$\sigma_a = PR_1^2 / (R_2^2 - R1^2) \tag{6}$$

$$P = (0.3IP / L\Lambda N) \tag{7}$$

$$\varepsilon = (1/E)(\sigma_a - \mu(\sigma_c - \sigma_r)) = (\delta r / r) \tag{8}$$

Theoretical displacement $\delta r$ was calculated by using $\delta r = \varepsilon r$, but the experimental value measured was given in eq. (9) as a function of the parameters:

$$\delta r = f(\varepsilon r, F_r, F_{kc}, F_v) \tag{9}$$

The displacement in radial direction was calculated for all four categories of pistons, and the results were discussed in the following sections.

## 3. Engine test programme and experimental data acquisition

A single cylinder compression ignition engine was used to generate the test data. This direct-injection 4-stroke diesel engine was fitted with fixed valve timing and speed control. The engine was coupled with a mechanical dynamometer. Knock sensor is used to measure engine block vibrations. The engine specifications are listed in Table 1.

**Table 1.** Specification of the engine

| Engine Parameters | Value |
|---|---|
| No. of Cylinder (*n*) | One |
| Bore *D* (mm) | 85 |
| Stroke *L* (mm) | 110 |
| Maximum Speed *N* (rpm) | 1500 |
| Connecting rod length (mm) | 235 |
| Compression ratio | 18:1 |
| Intake pressure (bar) | 1 |
| Intake temperature (K) | 300 |
| Inner radius of engine block $R_1$ (mm) | 44 |
| Outer radius of engine block $R_2$ (mm) | 112 |

### 3. 1 Selection of data sampling

Two different sampling methods were available, namely: crank-angle-based sampling and time-based sampling. Crank-based sampling is generally the most suitable for sampling engine test data. Most combustion events occur at different angular positions of the crankshaft and therefore can easily be located with crank-based sampling. However, control over the sampling frequency is not possible, this is determined by the engine-crank speed and by the pulse per revolution of the encoder. Crank-based data sampling is therefore not readily suitable for identifying fixed-step discrete-time mathematical models. Temporal sampling by contrast, involves fixed-time-step sampling controlled as usual by an internal clock, such that the sampling rate is independent of the crank speed. In this method the combustion events cannot be readily referenced to crank position; therefore the number of data samples varies from one cycle to another cycle owing to slight fluctuations in engine speed. The major advantage of temporal sampling is that the data is readily suitable for identifying discrete-time mathematical models. For vibration analysis involving combustion events, high sampling-rates are needed for capturing acceleration data. This is because activities such as valve-impact and injector-pulse forcing are high-frequency events. For these reasons, constant time-sampling was used instead of crank-based-sampling.

### 3. 2 Collection of vibration data and performance parameter

The experimental programme was divided into four categories. For the entire category the engine was loaded using dynamometer. The load starting from 0-15 kg in steps of 5 kg, and for each load the speed of the engine has varied from 1100 rpm to 1500 rpm in steps of 100 rpm. Firstly, the double convex profile piston is fitted in the engine and runs in normal mode. Secondly, the hemisphere profile piston is fitted, the load and speed is varied and then the engine is run in typical way. Similarly, in the third category engine was running with elliptical profile piston, and finally the ellipsoid piston was fitted with the engine, and running for the normal test conditions. For the above set of categories using Sendig 911-Vibrometer acceleration was picked up through accelerometer vibration pickup and stored in data collector as shown Fig. 1. The accelerations were picked up around cylinder block, and then MCME software was used for analyzing the collected data for various correlations. For entire categories the performance parameters were recorded, and the equations (2) to (8) were used for the calculation of displacement.

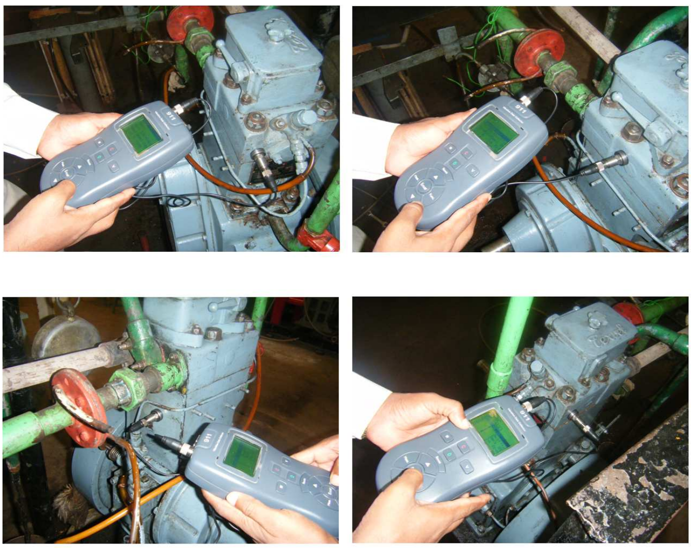

**Fig. 1.** Engine experimental setup with Sendig

### 4. Results and discussion

Experimental values for all four pistons shown in Fig. 2 were taken, and analyzed for the pressure contribution, Time-domain analysis and the combined contribution of moving masses. The results of displacement and Time-domain correlation are explained in the sub sections.

### 4. 1. Results for double concave profile piston at different speeds

The contribution of in-cylinder pressure induced displacement and experimental displacement is shown in Figs. 3-5 and the experimental values are following the same trend of the theoretical values. The inlet valve, exhaust valve and fuel injection are common for all the four type of piston fitted engines. Deviation of experimental values from the theoretical values is due to the unbalanced reciprocating forces in the engine ($F_r$), thrust force exerted by all connecting kinematic chains in the engine ($F_{kc}$) and valves impact ($F_v$). At low speed the fuel consumption was high, and part of the power produced was exerting forces on side walls of the cylinder. The damping takes too much time than the combustion cycle time of the diesel engine, because of the cycle-to-cycle variation in the combustion dominates in the particular piston engine. At low load the cycle-to-cycle phenomena cause periodic behavior in combustion timing; together with cylinder deviations this is found responsible for decreasing the operating regime. The combustion from the other cylinder noise sources are identified from Fig. 6(a), the peak amplitude marks the combustion at the top dead centre. The other cylinder noise is in between the valley and peak amplitude over the whole engine mapping for the double concave combustion chamber piston engine.

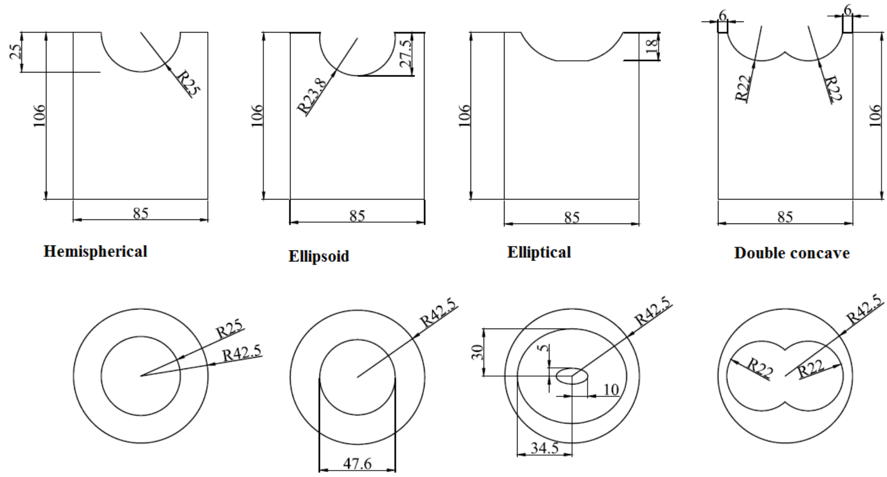


**Fig. 2.** Different piston profiles

### 4. 2. Results for hemispherical profile piston at different speeds

Hemispherical profile piston engine is normally used in most of the internal combustion engines and theoretical displacement at 1200 and 1300 rpm was slightly diverging (Figs. 3-4). The combustion phenomenon in the chamber is adverse to produce a constructive pressure increase in the cylinder. But it is used to increase the unbalanced excess forces on the kinematic chain. Consequently, it produces the vibration and then the amplitude is increased. An off-line time-frequency analysis of the cylinder pressure and the resulting knock signal determines the frequency range where the information about the combustion can be extracted (Fig. 6(c)). One can observe in the first angular-frequency represents the pressure signal within the same combustion cycle where the combustion process occurs. Low-frequency component may be related to the auto ignition and appears few crank angle degrees before the top dead centre. Fig. 5 indicates that the combustion chamber pressure induced displacement amplitude increases rapidly to the experimental value. These effects are due to the combustion pressure generated in the cylinder.

### 4. 3. Results for Elliptical profile piston at different speeds

Elliptical profile piston engine exhibits different vibration behavior that is provided in Figs. 3-5. Theoretical value of displacement at N4 and N5 speed are nearly drawn-out at the same point, the vibration amplitude due to combustion in cylinder pressure are stabilized in this speed. Time-domain value in Fig. 6(b) gives the displacement amplitude as cyclic variation with time. The energy released by the combustion as the instantaneous energy is the estimated source of vibration. This corresponds to the squared sum of Fourier coefficients. The theoretical displacement variation for speed and load was very close to each other. It indicates the energy release and combustion of the engine is uniform and follows quasi-adiabatic heat release. This pressure becomes the source of vibration and noise.

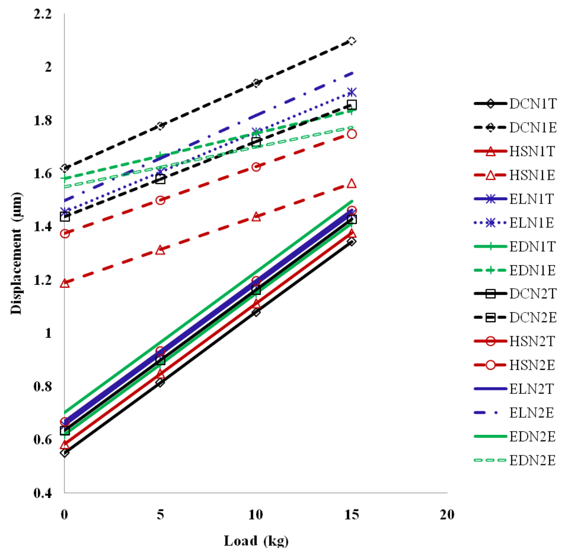


**Fig. 3.** Displacement of various pistons at N1 and N2 speed

## 4. 4. Results for Ellipsoid profile piston at different speeds

In-cylinder pressure contribution to the displacement for ellipsoid profile piston engine is shown in Figs. 3-5. At low speed the amplitude is minimal and it is increased with speed and load. Energy release and combustion phenomena are constantly increasing for the load as well as speed. The ellipsoid profile piston engine exhibits the habitual changes for the entire range of operating conditions. Noise production in the cylinder due to instable combustion is lower. The amplitude was smaller as depicted in Fig. 6(d) because of stable combustion, the unbalance forces acting on the reciprocating elements, kinematic chain and rotary elements are reduced. Minimization of the maximum value of the frequency function of the primary system is very important. Oscillation of the ellipsoid piston engine has very stable motion over long combustion cycle, but the rest of piston engines, oscillation decay for long period and growth for long period shows uncertainty of combustion cycle. The time-domain analysis the peak is better as four major peaks are occurred, that is equivalent of four combustion cycle. So the cycle-to-cycle variation was not affecting this type of engine like other engines.

## 5. Conclusions

The in-cylinder pressure induced displacement and time domain wave form has been used to identify the uncertainty of combustion from single point measurements average engaged with a vibration pickup instruments. Time-domain wave form was suitably accompanied with the in-cylinder pressure induced displacement for detecting vibrations of reciprocating and rotary elements. The difference between the theoretical and experimental values is due to the reciprocating and rotary masses. Elliptical profile and ellipsoidal profile are the two geometric differences from its major axis and depth of the profile, these two piston profiles are common under rankine half body, which is combination of circular and elliptical geometry favorable to combustion chamber. The vibration amplitude was lower for the streamlined piston profile, in ellipsoidal piston it has lower vibration and better combustion stability in comparison to other piston profiles. Thus, vibrations induced by different piston profiles were used for analysis to acquire better combustion chamber, which is competent in both vibration and combustion. Time domain wave form of rapidly-decaying and growing responses for vibration amplitude is an easy method to identify the instability in combustion and vibration knock. A new-fangled approach is proposed to identify the cycle-to-cycle variation of in-cylinder pressure for different chambers and is correlated to combustion stability, vibration and other sources of noise.

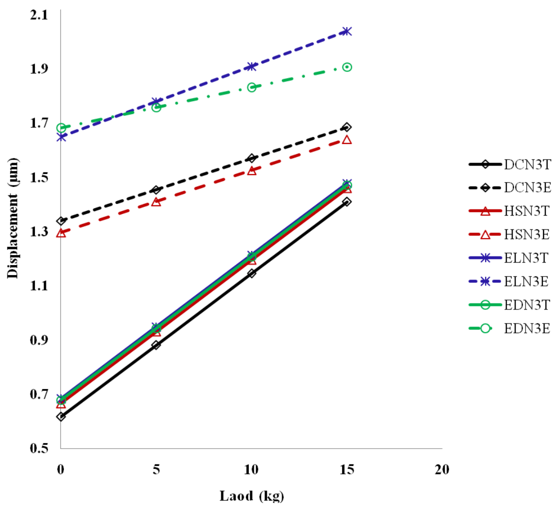


**Fig. 4.** Displacement of various pistons at N3 speed

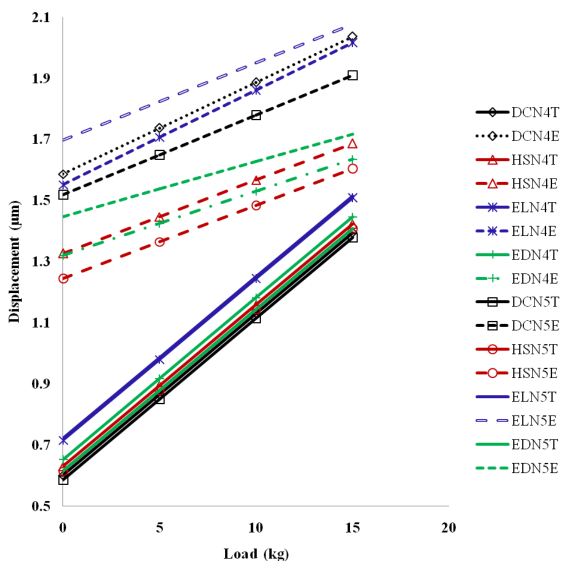


**Fig. 5.** Displacement of various pistons at N4 and N5 speed

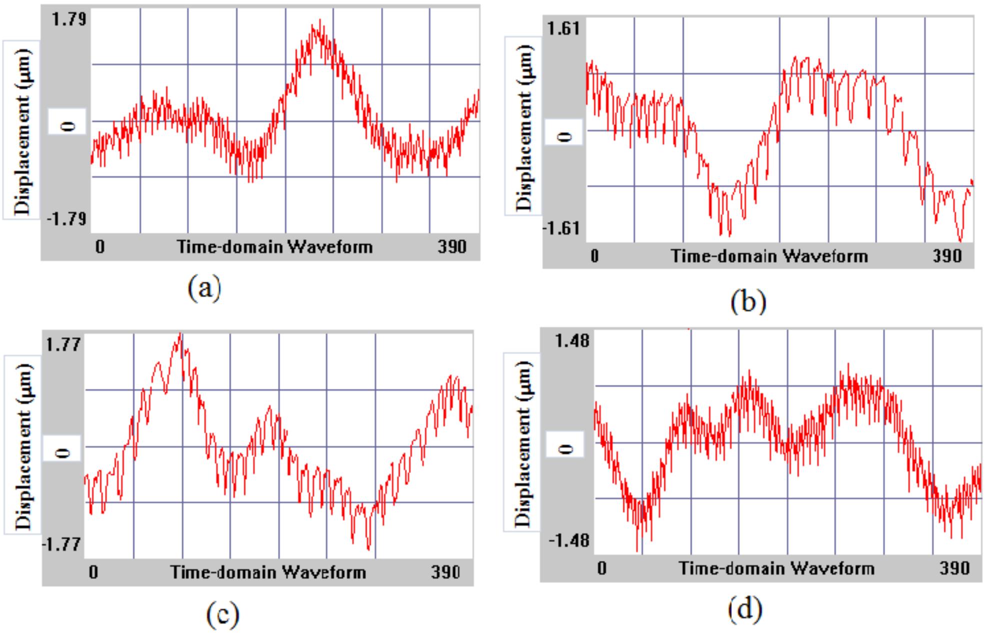


**Fig. 6.** Time-domain wave form of engine with various pistons:
(a) double concave, (b) elliptical, (c) hemisphere, (d) ellipsoid

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