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Accurate Instantaneous Engine Speed Recording by Employing an Optical Measurement System - Application to a Typical Low Power Industrial Engine

D.- N. Pagonis, G. Theotokatos and G. Livanos
Dpt. of Naval Architecture TEI of Athens

ABSTRACT

The presented work concerns the development of a novel measurement system for determining the instantaneous rotational speed of an engine with high accuracy. The developed system is mainly based on a commercially available optical sensor and appropriate data acquisition / post-processing procedure. The accuracy of the system is high; speed recording with a resolution of one degree of crank angle has been succeeded when measuring the speed of a one cylinder four stroke S.I. motored engine. The deduced experimental results were compared with the corresponding theoretical ones obtained by appropriate simulations, validating the proper functionality of the developed system. Furthermore, the system was also integrated into a typical four cylinder low power industrial engine successfully. Key-features of the proposed measurement configuration are accuracy, simplicity and low-cost suggesting numerous potential applications.

INTRODUCTION

Optimizing the performance of an engine has been the goal of numerous research efforts [1], since proper prediction of its main operation parameters (such as torque fluctuation during various loading etc.) can significantly contribute to efficient fault diagnosis, fuel economy, gas emissions reduction and optimum engine performance in general. The value of the instantaneous rotational speed of the crankshaft is an important parameter employed widely in engine operation simulations [2,3,4], engine operation analysis [5,6,7,8,9] and fault diagnosis [10,11,12,13] since it is relatively easily accessible employing low cost equipment. Most of the modern engine rotational speed monitoring systems are based on a “magnetic pick-up” principle of operation [13]. In brief, a metallic wheel carrying a given number of dents of appropriate geometry on its surface is attached to the rotating shaft, in order to trigger an inductive sensor accordingly; the rotational speed is deduced from the sensor's output signal. We should note that the maximum resolution of the above system (i.e. maximum number of velocity measurements during a single rotation of the shaft) is mainly affected by the minimum resolution of the sensor, i.e. the minimum size required for each dent in order to be magnetically detected. A measuring set-up as described above, employed in four cylinder low power industrial engine, as the one installed in the Department of Naval Architecture at TEI of Athens [14], has a maximum resolution of 120 measurements per cycle. In order to increase that figure, one should increment the number of dents available on the rotating disc, resulting to an increase of its dimensions; thus, the maximum resolution of such a system is limited -especially in the case of low power engines- by the available space provided for the disk. The developed measuring system presented in this work overcomes the above restriction by employing successfully an optical sensing methodology, details of which are presented in the following sections.

EXPERIMENTAL RESULTS

The basic idea investigated in the proposed measuring system was to use a standard low-cost optical sensor, in order to overcome the limit - posed by the space available, at the maximum resolution of the speed measurement, when a standard magnetic system is used. Therefore, instead of a metallic disc with dents on its surface, the specific system employs a disc with a flat surface, on top of which an appropriate reflecting pattern is situated. This pattern includes a sequence of identical geometrical sets consisting of a "Dark" and a "Light" region, appropriately dimensioned (figure 1); for each set to correspond to one degree of crank angle, the total number of patterned regions on the reflecting surface is 720. By using a contrast optical sensor (IFM contrast sensor, type O5K500), the contrast difference between the two successive geometrical patterns that comprise each set is detected and the corresponding measurement of the rotational speed of the disc (i.e. of the shaft) is deduced. We should note at this point that the minimum dimensions of each pattern to be sensed by the contrast sensor are significantly smaller than the corresponding ones of a "dent" that has to be implemented on the metallic disc, in a typical magnetic measuring system. Thus, the "reflecting" disc is considerably smaller, which makes it suitable to be mounted on the shaft of even a very low power engine, where the available space is extremely limited. Furthermore, as we will see in the following experimental sections in the case of a typical industrial low power engine there is no necessity for mounting an extra disc since the reflecting pattern can be easily applied directly on the shaft / shaft coupling of the engine.

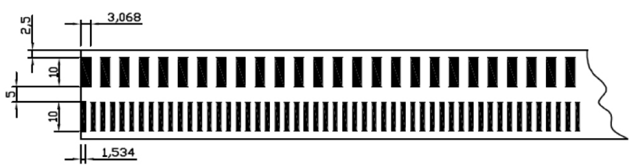

Figure 1. Schematic of the reflecting pattern situated on top of the disk's surface, consisting of consequent sets of a "Dark" and a "Light" region (dimensions are in mm).

Deducing the Rotational Speed from the Sensor's Output Signal

The contrast sensor's output is digital providing a high pulse on sensing a dark region and a low pulse on a light region or vice-versa. In general, the methods of measuring rotational speed can be categorized into two main groups: timer/counter-based and ADC-based [15]. The developed system is of the first type; therefore, the signal of the optical sensor is treated as a pulse-train and is employed to trigger an appropriate timer. By measuring the time elapsed between two successive pulses, one can obtain the corresponding frequency and hence the rotational speed (rpm) of the shaft for the specific time interval. The signal of the sensor is acquired by a standard Data Acquisition Card (National Instruments, type 6212) which is controlled by a PC running LabView. The resulting frequency is obtained by appropriate post-processing of the signal. In order to ensure that the sensor's output is at a compatible voltage level to the data acquisition card a simple circuitry is also present.

A one cylinder four stroke S.I. engine was employed initially in order to demonstrate the operation of the system. The reason for applying the system into the specific engine was the ability to easily vary the engine's speed (from as low as 30 rpm to more than a thousand) by motoring it. In more detail, a three-phase asynchronous motor was coupled to the engine's shaft while an appropriate inverter (LS, type SV015iC5-1) was used in order to control its speed, controlling effectively the engine's speed as well. Note that the cylinder spark plug and the valve kinematic mechanism have been removed thus no compression is occurring during the engine's cycle. As a result, the monitored instantaneous speed curve will be basically characterized by the inertia of the moving parts inside the engine (i.e. piston, flywheel etc.). The overall layout and top-view of the developed experimental set-up are presented in figure 2.

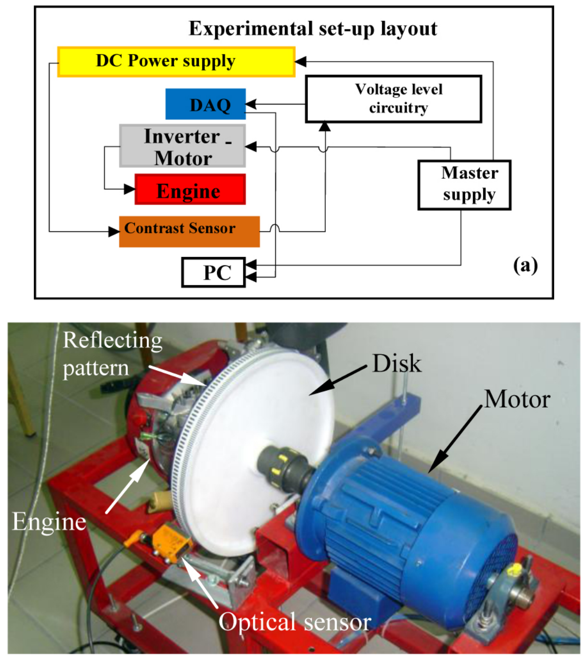

Figure 2. Overall layout (a) and top-view (b) of the developed experimental set-up for the case of a one cylinder four stroke S.I. engine (motored).

Initial Measurements - Post-Processing of the Obtained DATA

A typical recorded rotational speed variation of the engine is represented in figure 3. Observing the specific curve, leads to the conclusion that a significant amount of electronic noise is induced into the system, deteriorating the measurement results.

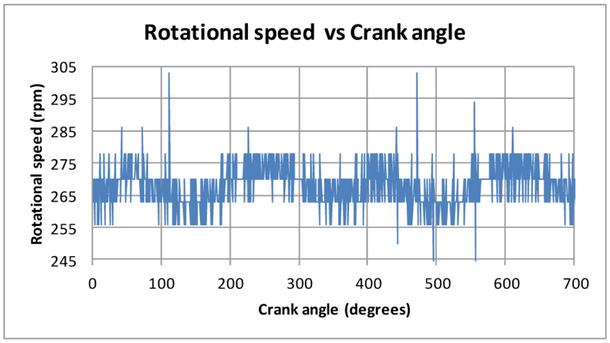

Figure 3. Initial experimental results; a significant amount of noise is present in the measurement data which becomes dominant as the motoring speed increases

The specific problem was addressed by appropriate post-processing of the initial measurement data, in order to remove effectively most of the noise induced into the system. In more detail, Fast Fourier Transform analysis was initially performed at the obtained signal representing the instantaneous rotational speed (i.e. at the signal representing the frequency variation which is deduced from the sensor's output pulse train). The analysis exposed a significant amount of noise present which was mainly contributed by high-frequency harmonics shown in figure 4. By employing a typical low-pass filter, the specific harmonics were removed from the original signal, revealing the underlying measurement. Figure 5 shows the recorded variation of the rotational speed of the engine before and after removing successfully the noise.

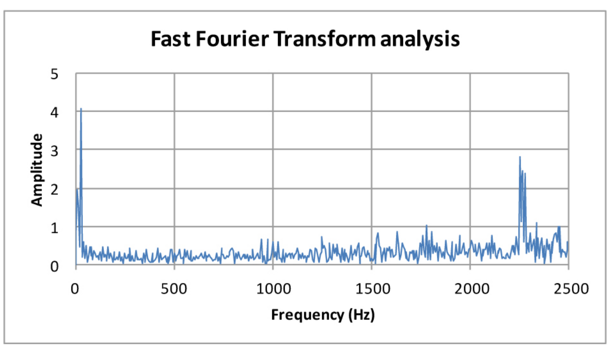

Figure 4. Fast Fourier Transform analysis of the signal representing the instantaneous rotational speed.

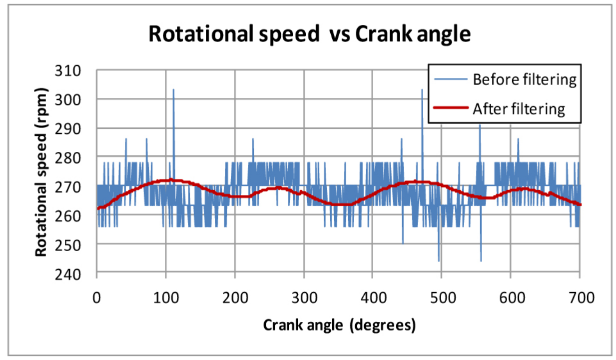

Figure 5. Recorded variation of the rotational speed of the engine before (a) and after (b) removing successfully the noise

As it was revealed by the FFT analysis the specific noise is mainly comprised of high frequency harmonics; therefore it is very likely that it is induced into the system by the inverter regulating the speed of the motor. As we will see in the following sections, the above observation is verified in the case where the system is integrated into a typical low power industrial engine which was running instead of motored.

Instantaneous Rotational Speed Recording on a Motored Engine

The instantaneous speed of the engine was obtained successfully employing the developed sensing system, for values ranging from as low as 50 rpm, up to 700 rpm. Typical results are presented in figure 6. As already mentioned above, in the specific case the engine is motored thus there is no compression present. The recorded variation of the rotational speed is originated mainly due to the inertia of the moving parts of the engine and the developed friction of the engine parts. We should also note that when the speed raises above 710 rpm, the optical sensor reaches its maximum switching frequency -that is, the transition between two consequent “Dark” and “Light” patterns is occurring at a frequency that is larger than the maximum switching frequency that the sensor is capable to operate at, resulting to an invalid measurement. In order to overcome the specific limit, one should decrease appropriately the resolution of the system by setting each “Dark” and “Light” set to correspond to more than one degree of crank angle (i.e. 1.5 or 2 degrees); this would still provide though a high resolution measurement.

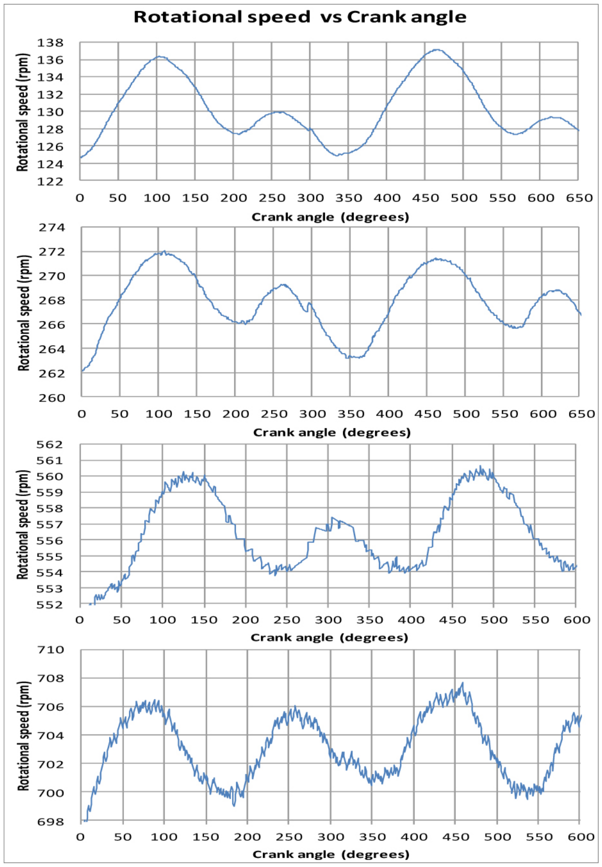

Figure 6. Typical obtained measurements of the instantaneous rotational speed of the engine employing the developed system.

Validating the Functionality of the System

In order to perform an initial validation of the proper functionality of the developed measuring system, the experimentally obtained instantaneous rotational speed profile was compared with the corresponding theoretical one, which was derived based on appropriate modeling of the combined system. In more detail, the engine crankshaft rotational speed is calculated by integrating the angular momentum conservation equation, which can be written as:

$$\frac{dN}{dt} = \frac{30(T_e - T_m)}{\pi(I_m + I_e)} \tag{1}$$

Owing to the fact that the engine is rotated using the electric motor, the electric motor produces torque (thus is considered to be negative), whereas the engine kinematic mechanism absorbs torque. The electric motor inertia is considered to be constant, whereas the engine kinematic mechanism inertia is calculated by the following equation, which was derived taking into account the energy conservation in the rotating and reciprocating masses of the engine:

$$I_e = I_r + m_l r^2 \left(\frac{c}{r\omega}\right)^2 \tag{2}$$

with

$$\frac{c}{r\omega} = \sin\phi \left(1 + \frac{\cos\phi}{\sqrt{(l/r)^2 - \sin^2\phi}}\right) \tag{3}$$

where c is the instantaneous piston velocity, is the crank angle, l is the connecting rod length, r is the crank radius, ω is the engine angular velocity, I_r is the polar moment of inertia of the engine kinematic mechanism rotating parts (including shaft inertia) and ml is the engine reciprocating mass. The engine torque is calculated by [1]:

$$T_e = \left((p_{cyl} - p_{cr})A_p r - m_l r^2 \omega^2 \left(\frac{b}{r\omega^2}\right)\right)\frac{c}{r\omega} - T_{fr} \tag{4}$$

with

$$\frac{b}{r\omega^2} = \cos\phi + \frac{(l/r)^2 \cos 2\phi + \sin^4\phi}{\left((l/r)^2 - \sin^2\phi\right)^{3/2}} \tag{5}$$

where p_{cyl} is the cylinder pressure, p_{cr} is the crankcase pressure, A_p is the piston area and b is the instantaneous piston acceleration. The engine friction torque, T_{fr} is calculated using the model developed by Rezeka and Henein [16]. This model accounts for losses sourcing from the piston ring viscous and mixed lubrication regimes, piston skirt hydrodynamic lubrication, valve train friction, auxiliaries and loaded bearings friction. As is also explained below, in the examined single cylinder engine all auxiliaries and valve train mechanism were removed, in order to focus on the piston - crank - slider mechanism response under motoring conditions. As a result, the relevant friction components, separately described in the used friction model, were not taken into account.

For the conducted experiments, the cylinder spark plug and the valve kinematic mechanism have been removed. Both were modeled by considering the engine cylinder connected upstream and downstream to valves of constant area. The cylinder pressure is calculated considering the mass, energy conservation and the ideal gas law equations for the cylinder volume [17]. The working medium mass flow rates through the valves were calculated using the flow equation for an orifice [17]. The crankcase pressure was considered to be constant. The model was implemented in MATLAB/ Simulink environment.

Simulation runs were performed for several values of mean engine rotational speed and the derived results superimposed to the measured data are presented in figure 7. As it can be deduced from this figure, the simulation results are in adequate agreement to the measured data. The observed variation is owing to the following reasons: a) the dynamics of engine/motor shafts coupling was not modeled, b) the used friction model has been developed for the normal operation of engine (normal speed region and firing condition), whereas in this work it was used under engine motoring conditions in low engine speeds, c) the motor produced torque was considered constant whereas it may exhibit a slightly oscillating behavior.

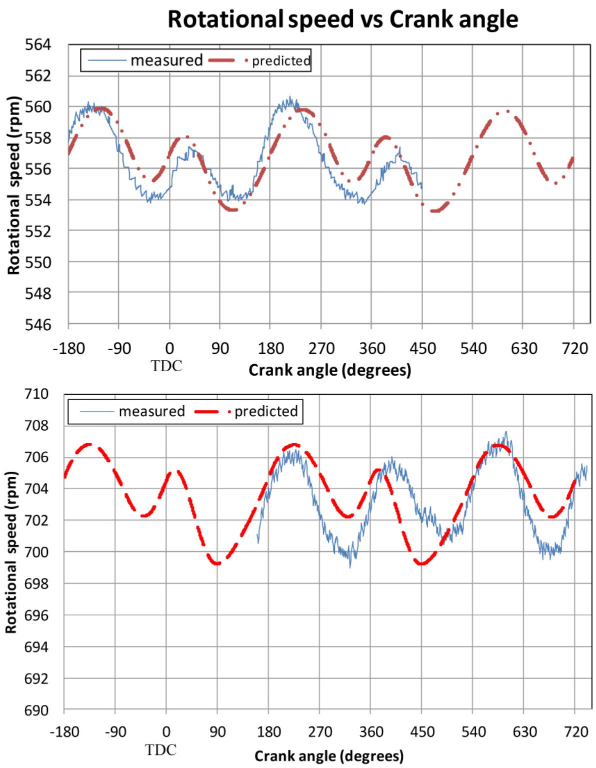

Figure 7. Typical simulation results in comparison with the corresponding experimental ones

However, the obtained results seem to validate the proper operation of the proposed measuring configuration and also the functionality of the developed engine model.

Integrating the System into a Typical Industrial Low Power Engine

After having successfully validated the functionality of the proposed system, a typical four cylinder low power industrial engine was employed in order to test it under normal running conditions instead of motoring. In more detail, the engine employed was an IVECO N45 MST with rated power of 93 kW (125 HP) at 2200 rpm. As a first approach the system was set to run at a resolution of four degrees (i.e. 90 measurements per cycle). The reason for choosing this level of accuracy was to avoid the need of the extra plastic disc to be coupled to the engine's shaft and to embed the necessary reflecting pattern directly on the outer surface of the engine/dynamometer coupling. In this way, the adjustments / alterations made to the engine that were necessary to be performed in order for the measuring system to be integrated successfully were kept to minimum; i.e., they included only the fixing of the optical sensor and the adhesion of the reflecting pattern; both, very simple and fast procedures. Figure 8 presents the resulting set-up. Note that the reflecting pattern is situated on top of the coupling connecting the intermediate shaft with an Eddy current dynamometer.

Figure 8. The developed system integrated on the IVECO N45 MST engine

As a first trial the engine was running at idle speed while the speed variation was captured. The results are shown in figure 9. Note that in this configuration there is no significant noise present at the derived signal compared to the previously studied case of the one cylinder four stroke S.I. engine, which was motored by the electric motor. In that case, the origin of the specific large amount of high frequency noise was due to the presence of the inverter. However, in the case of the IVECO engine, the high frequency observed oscillations, are basically owing to the quality of the patterned reflecting surface. In order to remove the observed oscillations, a low pass filter was applied to the recorded data; the deduced results are also shown in the same figure. From the data presented, the engine cylinders firing effects are clearly recognized at every 180 CA degrees.

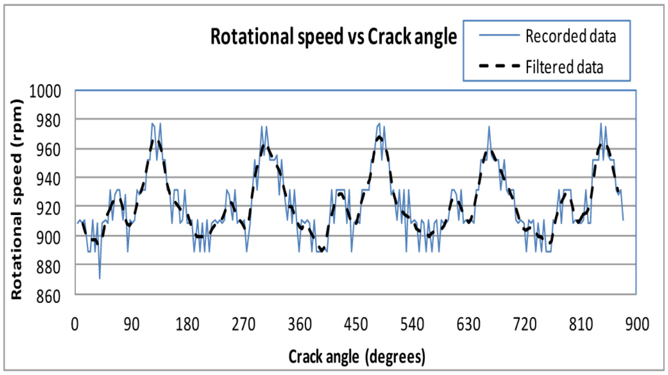

Figure 9. Recorded variation of the rotational speed of the engine running at idle speed

The filtered engine rotational speed signal in the case of engine idle speed operation (figure 9) for one engine cycle (720 degrees CA) was modified by subtracting its mean value and then analyzed using Fast Fourier Transform (FFT). The magnitude of engine rotational speed FFT as function of the ratio of the calculated signal frequencies to the frequency that corresponds to the mean engine speed is presented in figure 10. The average engine rotational speed for the examined engine cycle was calculated at 922.2 rpm.

From FFT analysis we know that the minimum frequency resolution is equal to:

$$\Delta f = f_{sampling} / K \tag{6}$$

where K is the total number of measurements.

In the specific case K=180 since the speed is measured every 4 degrees of CA while the rotational speed was captured for 720 degrees of CA. The sampling frequency $f_{sampling}$ can be easily approximated by taking into account the calculated mean value of the recorded rotational speed (i.e. 922.2 rpm) and the total number of measurements for each shaft rotation(i.e. 90); hence:

$$f_{sampling} = 90 / t_{\sin gle cycle} \tag{7}$$

where: t_{single_cycle} is the time corresponding to a single shaft rotation (on which 90 measurements are performed)

Since the mean value of rotational speed was found 922.2 rpm, we can easily deduce that:

$$t_{\sin gle cycle} = 60 / 922.2 = 0.065062\,s \tag{8}$$

Thus we should expect that:

$$\Delta f = 7.685\,Hz\,or\,461.1\,rpm \tag{9}$$

As we can notice the calculated value for Δf is verified in the FFT analysis presented in figure 10, where the first frequency was found to be 460.8 rpm -corresponding to approximately the half of the average engine rotational speed value, which is equal to 461.1 rpm. Thus, the calculated frequency ratio values are 0, 0.5, 1, 1.5, 2, etc. The highest amplitude value is calculated for the frequency that corresponds to the double engine rotational speed value (ω/ω_1=2). Large magnitude values are also calculated for its next harmonics (ω/ω_1=4, 6, 8). This is explained as follows.

The main contribution to the engine rotational speed variation is attributed to the engine firings. Since the engine cycle is four-stroke, one engine cylinder firing occurs for every engine cycle, i.e. for every two rotations of engine shaft. In addition, since the engine has four cylinders, the firings contribution to the instantaneous rotational speed will occur at the frequency that corresponds to the double engine rotation speed (4N/2) and at its higher harmonics.

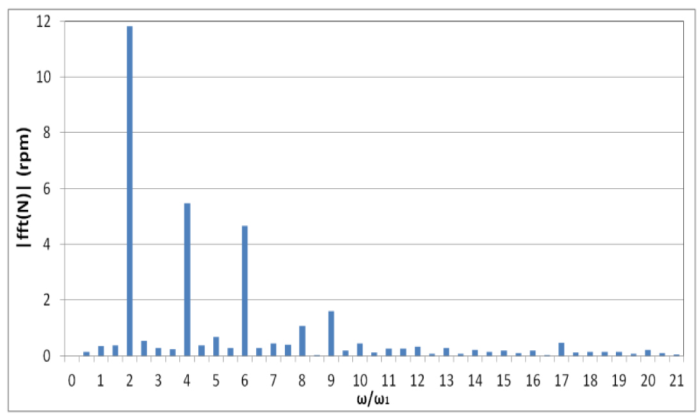

Figure 10. FFT of the engine rotational speed signal for one engine cycle

In order to ensure the repeatability of the proposed measuring system, the engine speed was recorded for a number of consecutive engine cycles. The results for the case of engine running at 1150 rpm unloaded are presented in figure 11. As It is observed for figure 11 the deduced speed measurement exhibits adequate repeatability for three consecutive engine cycles. Respective results concerning the filtered data were also derived for other engine operating conditions, not presented here.

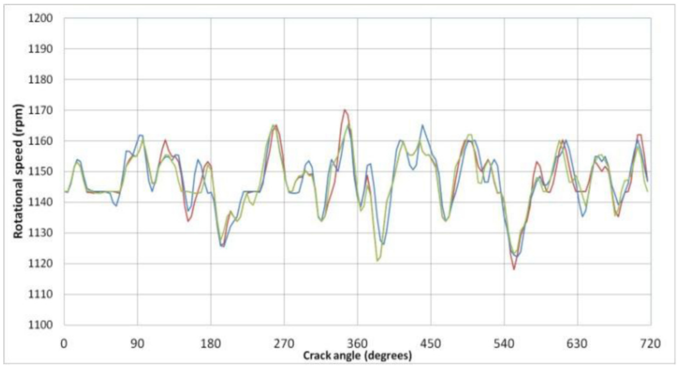

Figure 11. Filtered engine speed signal for three consecutive engine cycles at 1150 rpm -unloaded engine operation

In addition, the engine rotational speed was also recorded in the case of engine shutting down, i.e. from 850 rpm till the engine stop. Note that the engine stopping time is greatly affected by the total polar moment of inertia of the engine / Eddy current dynamometer rotating parts; the specific results are presented in figure 12.

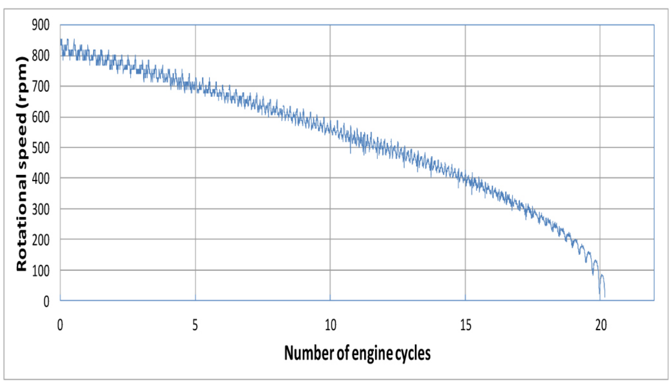

Figure 12. Recorded variation of the rotational speed of the engine during engine shutting down

CONCLUSIONS

In the current work, a novel system for measuring the instantaneous rotational speed of an engine is presented. The system is comprised mainly of a low-cost commercial optical sensor and appropriate data acquisition/post-processing. The functionality of the system was successfully indicated by measuring the variation of the rotational speed of a one cylinder four stroke S.I. engine which was motored and comparing the obtained profile to the respective one derived using an appropriate simulation model. Furthermore, the proposed configuration was easily integrated successfully into a typical industrial low power engine for recording its speed variation profile during running at various unloaded conditions and shutting down. The obtained results were analyzed using FFT and the findings were explained based on fundamental principles governing the instantaneous engine speed variation. The repeatability of the measured signal was also tested and found to be adequate for the cases of engine operation unloaded at various speeds. The recording of the engine speed variation during engine shutdown revealed the effect of engine/dynamometer rotating parts polar moment of inertia on the required engine stopping time.

Significant advantages of the proposed system include simplicity of the necessary set-up, reduced fabrication cost, trivial process for integrating the system into a typical engine, combined with high measurement accuracy. Ongoing studies are focused on modifying appropriately the developed system in order to increase the maximum value of rotational speed which is capable of recording, at a resolution of one degree of crank angle and also to determine how effectively the developed sensing arrangement can be employed in optimizing an engine's performance and in real-time fault diagnosis monitoring.

Acknowledgment

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