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DECOMPOSITION OF TURBULENT VELOCITY SIGNALS USING MOVING AVERAGE FILTER

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ABSTRACT

In this study, turbulent velocity signals of the Ricardo E6/T variable compression engine are investigated using moving average filtering technique. An important characteristics of a turbulent flow is its irregularity or randomness. Statistical methods are therefore used to define such a flow field. In this study, turbulent velocity signals are smoothed using moving average filter for different engine speeds and configurations.

I. INTRODUCTION

The turbulent flow field in an engine plays an important role in determining its combustion characteristics and thermal efficiency. The research about automotive engineering has shown that the variations in the both combustion chamber shape and inlet system geometry, change the turbulent flow field; influence emissions, fuel economy and the lean operating limit of an engine [1]. In turbulent flows, the rates of transfer and mixing are several times greater than the rates due to molecular diffusion. This turbulent "diffusion" results from the local fluctuations in the flow field. It leads to increased rates of momentum, heat and mass transfer; moreover it is essential for the satisfactory operation of spark ignition and diesel engines. An important characteristics of a turbulent flow is its irregularity or randomness. Statistical methods are therefore used to define such a flow field. Basically, there are organized and turbulent motions associated with DC component in an unsteady turbulent flow. But, in engines, the flow pattern changes during the engine cycle. In addition, while the overall features of the flow repeat itself at each cycle, the details do not; because the organized motions can vary significantly from one engine cycle to the next cycle. There are both cycle-to-cycle variations in the organized motion at any point in the cycle, as well as turbulent fluctuations about that specific cycle's organized motion.

In the literature, the decomposition of in-cylinder flow into mean and turbulence velocities has remained as unsolved problem [2]. In stationary flows, the mean flow component is associated with the energy in the DC component flow, and the flow energy in the higher frequency components is associated with the turbulence. Engine flows are, however, non-stationary such that the frequencies associated with the mean and turbulence flow processes overlap. It is possible to characterize engine flow based on high turbulence intensities and quasi-periodic variations in the mean [2]. Because of the overlapping scales and the quasi-periodic variations in the mean, there is no clear-cut method to differentiate between mean and turbulent flow energies [3].

There are many methods to investigate the signal of in-cylinder flows. One of the most frequently used approaches in engine velocity measurements is ensemble-averaging method. But, in-cylinder flows are exposed to cycle-to-cycle variations, thus the turbulence is underestimated by this effect [4]. Other most used methods are low-pass and high pass filtering that is strongly depends on cut-off frequency [5]. In this study, moving average filter, which is very popular for smoothing time series data, is used to decompose turbulent velocity signals.

In the next sections; first, the experimental work is explained, then a brief theoretical background of moving average filter is given. The results of smoothing turbulence signals for shrouded valve and standard valve are discussed in the following section. Finally, a conclusion that comprise the comparison of shrouded and standard inlet valve operating under different engine speeds, is given.

II. EXPERIMENTAL

The turbulent velocity signal measurements were carried out in a Ricardo E6/T variable compression engine [6]. The Ricardo E6/T variable compression engine has a single cylinder, poppet valve, four strokes; and it can be operated as spark ignition or diesel engine. It was used with a spark ignition engine head in this study. The combustion chamber is cylindrical in shape, the ends being formed by the flat surfaces of the cylinder head and piston. The engine was used in a motored engine conditions. Two types of inlet valves, standard and shrouded, were used to investigate the effectiveness of the inlet swirl on the combustion. The inlet valve opens 9 degrees before Top Dead Center (TDC), and closes 36 degrees after Bottom Dead Center (BDC), and valve clearance is 0.15 mm. The exhaust valve opens 42 degrees before TDC, and closes 7 degrees after TDC, and valve clearance is 0.20 mm. A Prosser Scientific Instruments, Model 6110 anemometer bridge module, was used to measure velocity. The anemometer is essentially a Wheatstone bridge with the probe resistance forming the active element. The bridge can be driven either by a constant current (CCA) or more usually, as used in this study, by a constant temperature (CTA) using a feedback amplifier. The in-cylinder velocity measurements were carried out in a radial and circumferential plane; 100 cycle were considered, and each cycle has 1306 measurement.

III.MOVING AVERAGE FILTER

Moving average filter is a signal-averaging filter that is used for smoothing rapid fluctuations in a signal. This type of filtering is very popular especially for smoothing time-series data generated by physical, economic or biological processes [7]. A moving average is obtained by accumulating the average over the previous N samples. By taking the average of each successive group of samples, the new samples are included in the average and the oldest samples are removed. Consequently, the accumulated signal is always the average of the N most recent samples. This nonrecursive process can be described by the following equation,

$$\begin{aligned} y(n) &= \frac{1}{N}\{x(n)+x(n-1)+\ldots+x(n-[N-1])\} \\ &= \frac{1}{N}\sum_{k=0}^{N-1} x(n-k) \end{aligned} \quad (1)$$

The larger N is, the smoother the data is obtained. This filter can also be designed as weighted moving average filter described by

$$y(n) = \frac{G}{N}\sum_{k=0}^{N-1} w(k)x(n-k) \quad (2)$$

with different weighting sequences, *w(k)*. As seen from the last equation, the response of the moving average filter is a weighted average of the past and present inputs.

From the design point of view, there are two fundamental types of digital filters: finite impulse response (FIR) and infinite impulse response (IIR). As the terminology suggest, these classifications refer to the filter's impulse response. By varying the weight of the coefficients and number of filter taps, virtually any frequency response characteristics can be realized with an FIR filter. The moving average filter is known as the most elementary form of an FIR filter. The digital filter implementation of moving average filter is beyond the scope of this paper and can be found digital signal processing texts [8]. Moreover, Matlab is used for the implementation of moving average filtering in this study.

IV. RESULTS AND DISCUSSION

The energy spectrum of the measured circumferential velocities is shown in Figure 1 for shrouded inlet valve with 1500 rpm. A strong peak is observed at 25 Hz, and a secondary peak can also be observed at 75 Hz. These peaks represent a strong mean flow (DC component and organized motion) resulted from the shrouded valve configuration. At higher frequencies, total energy spectrum decreases rapidly, and it is in consistency with Kolmogorov's -5/3 slope. In Figure 2, the first cycle of the raw and filtered circumferential motions are shown for the shrouded inlet valve configuration with the engine speed, n=1500 rpm. As it is seen from Figure 2, the mean flow (filtered signal) decreases until Inlet Valve Closing (IVC) reaches at 216 degree crank angle. After that it starts to increase steadily up to the interval of 270-300 degree crank angle. Then, the direction of the signal again changes and start to decrease as the piston approaches Top Dead Center (TDC). As seen from the figure, mean flow increases slightly around TDC. The rapid decay just prior to TDC can be due to viscous and turbulent dissipation, or due to non-uniformity [1]. In Figure 3, filtered circumferential velocities for shrouded valve is shown versus engine crank angle both for 1500 rpm and 2000 rpm of the engine speeds. When the piston approaches to the Top Dead Center, there is a difference between filtered circumferential velocities for 1500 rpm and 2000 rpm. It can be concluded that, mean flow is higher around TDC, when the engine speed is increased. In Figure 4, radial raw and filtered velocities are shown for the standard inlet valve. It can be observed from this figure that radial velocities for standard valve shows the same behavior up to about 330 degrees crank angle as the behavior of circumferential velocities of shrouded valve. But they continue to decrease after TDC although the circumferential velocities of standard valve increase as seen in Figure 2. In Figure 5, circumferential smoothed velocities filtered by the moving average method for standard valve decrease before TDC, and then increase around TDC. Note that, DC component of the circumferential velocities for standard valve are lower than that of the shrouded inlet valve. As shown in Figure 6, there is a huge difference between circumferential turbulence energy of shrouded and standard inlet valve at n=1500 rpm. In other words, turbulence energy for shrouded inlet valve (shown in Figure 6.a) is higher than the turbulence energy of standard inlet valve (shown in Figure 6.b) for the same engine speed.

The DC component, Reynolds number and turbulence intensities are listed in Table 1 for the circumferential velocities, and in Table 2 for the radial velocities. In Table 1 (2), the DC component of the circumferential (radial) velocity for the shrouded valve case is approximately 3.4 (1.2) times greater than the one for the standard valve case. It can be seen from the tables that, the turbulence intensities and Reynolds numbers for the shrouded valve case are higher than the ones for the standard valve case. In addition, turbulence intensity and DC component of velocity are increased with the engine speed. Hence, this results complies with the other studies [9].

V. CONCLUSION

Two types of inlet valves, standard and shrouded, were used to investigate the effectiveness of the inlet swirl on the combustion. The in-cylinder velocity measurements were carried out in a radial and circumferential plane with different engine speeds. After smoothing the velocities using a moving average filter, it was found that the DC component of the velocities and turbulence intensities are higher for the shrouded valve case than that of standard valve. Furthermore, the DC component and turbulence intensity increase as engine speed gets faster.

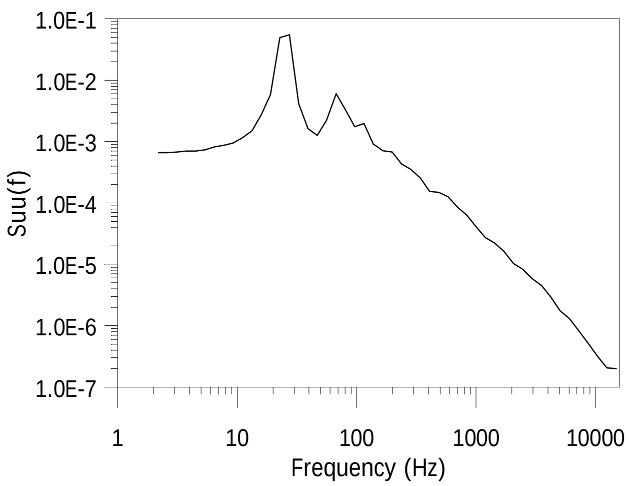

Figure 1 The energy spectrum of the circumferential measured velocities for shrouded valve (ε=8:1, n=1500 rpm, $\overline{u} = 17{,}1$ m/s, Re_{ℓ}= 4142)

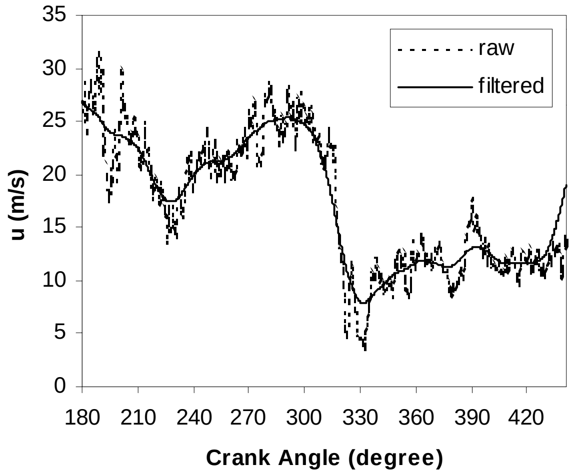

Figure 2: Circumferential raw and filtered velocities for shrouded valve (ε=8:1, n=1500 rpm, $\overline{u} = 17{,}1$ m/s, Re_{ℓ}= 4142)

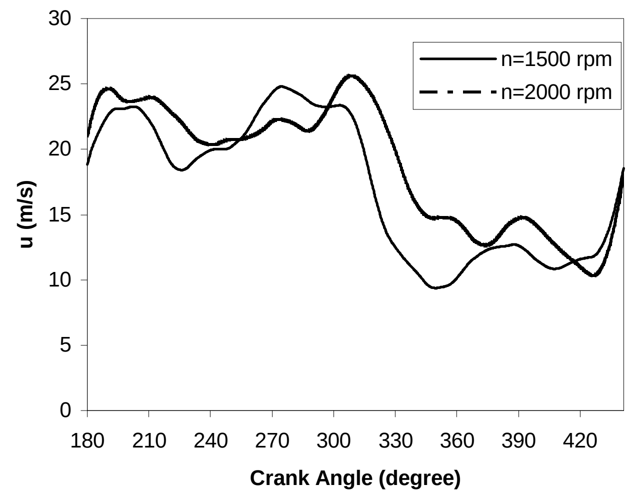

Figure 3. Filtered circumferential velocities for shrouded valve (ε=8:1, , $\overline{u} = 17{,}1$ m/s, Re_{ℓ}= 4142 for n=1500 rpm; $\overline{u} = 18.3$ m/s, Re_{ℓ}= 4406 for n=2000 rpm)

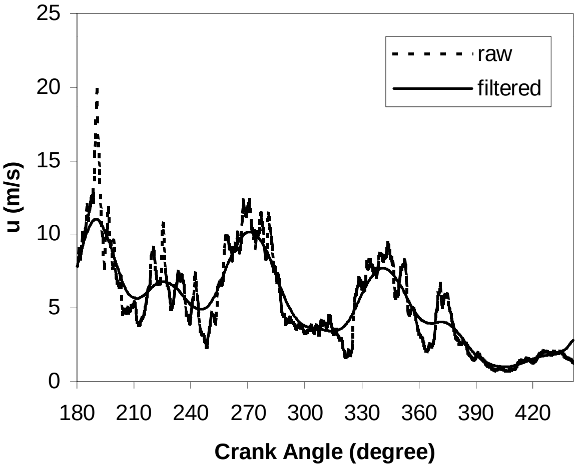

Figure 4 Radial raw and filtered velocities for standart valve (ε=8:1, n=1500 rpm, $\bar{u} = 5.51$ m/s, Re_ℓ= 1330)

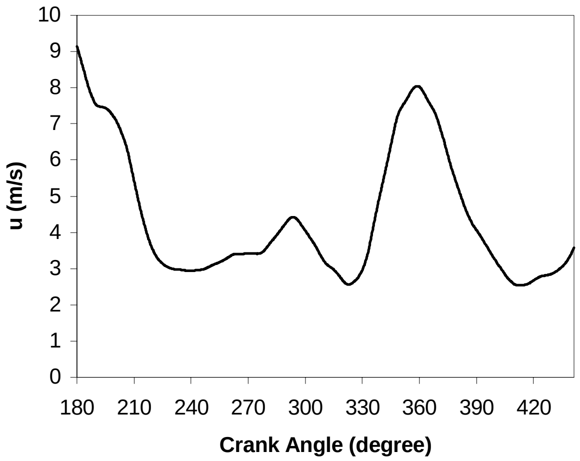

Figure 5 Circumferential raw and filtered velocities for standard valve (ε=8:1, n=1500 rpm, $\bar{u} = 5.02$ m/s, Re_ℓ= 1216)

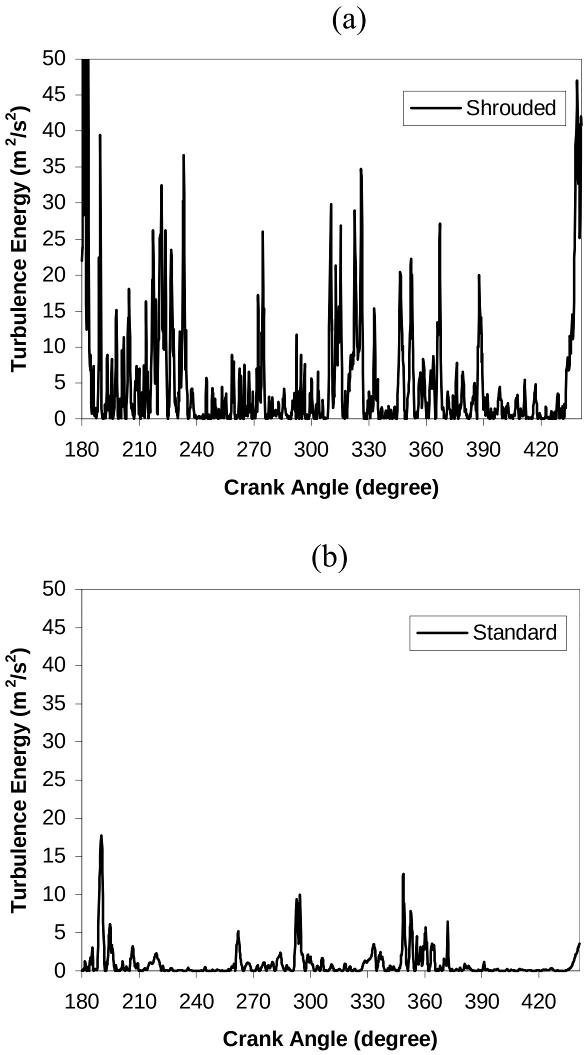

Figure 6 Circumferential turbulence intensities for standart and shrouded valve (ε=8:1, n=1500 rpm, $\bar{u} = 5.02$ m/s, Re_ℓ= 1216 for standart valve; $\bar{u} = 17{,}1$ m/s, Re_ℓ= 4142 for shrouded valve)

Table 1. Turbulence characteristics deduced by moving average method for the circumferential velocities, (CR=8:1).

	Shrouded Valve n=1500 rpm	Shrouded Valve n=2000 rpm	Standart Valve n=1500 rpm
DC component (m/s)	17.1	18.3	5.02
Re_ℓ	4142	4406	1216
Intensity (m/s)	2.16	2.23	1.17

Table 2. Turbulence characteristics deduced by moving average method for the radial velocities, (CR=8:1).

	Shrouded Valve n=1500 rpm	Shrouded Valve n=2000 rpm	Standart Valve n=1500 rpm
DC component (m/s)	8.37	10.91	5.51
Re_{ℓ}	2029	2628	1330
Intensity (m/s)	1.86	2.35	1.24

REFERENCES

1. R. J. Tabaczynski, R. J., Turbulence and Turbulent Combustion in Spark-Ignition Engines, Selected Papers from Progress in Energy and Combustion Science, Edited by N. Chigier, Pergamon Press, First Edition, 1979.
2. R. Ancimer, P. Sullivan, J. Wallace, The Influence of Turbulent Flows on the Natural Gas Combustion Process as seen through Discrete Wavelet Transforms, Vol. 23, pp 41-50, 2000.
3. P. Sullivan, R. Ancimer, J. Wallace, Turbulence averaging within Spark Ignition Engines, Exp. Fluids, Vol. 27, pp. 92-101, 1999.
4. T.M. Liou, D.A. Santavicca, Cycle Resolved LDV Measurements in a Motored IC Engine, Transactions of the ASME, Vol 107, pp. 232-239, 1985.
5. A.C. Enotiadis, C. Vafidis, J.H. Whitelaw, Interpretion of Cyclic Flow Variations in Motored Internal Combustion Engines, Experiments in Fluids, Vol. 10, pp.77-86, 1990.
6. K. AYDIN, Turbulent Combustion in Spark Ignition Engines, Ph. D. Thesis, The University of Liverpool, Liverpool, 1993.
7. R.D. Strum, D.E. Kirk, Contemporary Linear Systems Using Matlab, PWS Publishing Company, 1995.
8. Mixed Signal and DSP Design Techniques, Published by Analog Devices, Inc. 2000.
9. R.E. Winsor, D.J. Patterson, Mixture Turbulence, A Key to Cyclic Variation, SAE Paper No. 730086, 1973.