# EVALUATION OF THE FLOW WITHIN THE COMBUSTION CHAMBER OF SPARK-IGNITION ENGINES

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***Abstract:*** *By turbocharging the spark-ignition engine one is able to reduce the specific fuel consumption as well as the emission of certain pollutants (Vianna et al, 1989). That is, by means of a turbocharger, the equivalence ratio may be reduced without losses in output power. Two engines, normally aspirated and turbocharged, have been compared and the results show this reduction to be true. The reduction remains unexplained.*

*In order to infer on the dynamics of the flow during the first two strokes, measurements of pressure, temperature and velocity within the combustion chamber of an 1.3l petrol engine have been carried out. A piezoelectric pressure transducer, a constant current and a constant temperature anemometers have been employed respectively. The analysis has been limited to the first two strokes because the characteristics of the burning process are a consequence of phenomena occurring prior to the release of the sparkle.*

*As compared to the normally aspirated engine, the turbocharger brings about higher temperature and velocity turbulent intensities along the whole scope of analysis. This can be intuitively expected since a greater amount of mixture is admitted into the combustion chamber. The mixing process and consequently the quality of the burning of the fuel are both improved.*

*Keywords: SI-engine, turbocharging, anemometry, combustion chamber.*

## 1. Introduction

The internal combustion engines are held greatly responsible for the waste of energy and environmental damage. Their efficiency is limited to only about 30% (Benson – 1982). Still, it is impractical, if not irresponsible, to simply ban this type of engine. The viable short term alternative would be to increase its efficiency, including production, distribution and consumption of fuels.

In a previous work, Vianna *et al* (1989) have compared the performance of a turbocharged engine and its normally aspirated counterpart. Their study showed that from partial to full loads, at speeds ranging from 1500 to 3250rpm, the specific fuel consumption (S.F.C.) of the turbo engine is about 8% lower than that of the normally aspirated (Fig. 1). Besides that, the turbo engine operates at lower air/fuel ratios, that is leaner mixtures. These two results together lead to the fact that by means of the turbocharger it is possible to produce a greater amount of power with the same quantity of fuel than would be produced by the normally aspirated engine.

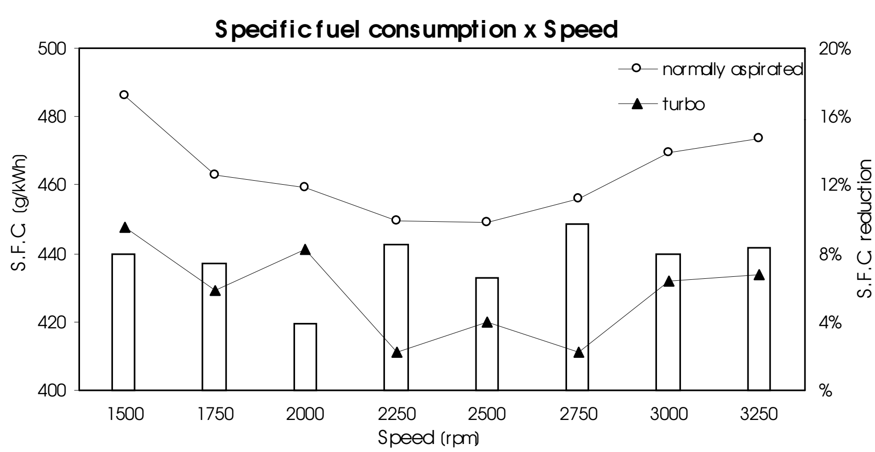


Figure 1. Specific fuel consumption for the normally aspirated and turbocharged engines.

Besides the fact that more air/fuel mixture is admitted into the combustion chamber, thus increasing the pressure and temperature at the end of compression stroke, what could be the other possible reasons for the reduction observed in the S.F.C.? One of the first conclusions that can be intuitively drawn is that when both higher pressure and air/fuel mixture are combined there are evident shear stress and consequently turbulence intensity increases. This last concept requires a particular explanation which is made in the next section.

The hypothesis presented is that the turbocharger brings about higher pressure, temperature and turbulence intensity which result in an improved burning process and consequently lower S.F.C.. This work is an attempt to experimentally test this hypothesis in terms of these three variables.

## 2. In-cylinder flow

This section is a brief description of the in-cylinder flow based on the work of Benson (1982).

There are two main structures of the flow inside the cylinder: swirl, whose rotation axis is that of the piston, and tumble, whose rotation axis is perpendicular to the cylinder axis. The manifold/valve configuration employed in this work is of the tangential type: it imposes such an orientation to the movement of the mixture that the swirl ratio is much greater than the tumble ratio.

During the intake stroke, while the inlet valve is still open two main toroidal vortices are generated: one in the vicinity of the valve, called external vortex, and the other covering the rest of the cylinder, called internal vortex. As the piston moves closer to B.D.C., these two structures begin to decay due to the reduction of inlet flow, to turbulence shear stress increase and the reversion of the piston movement.

The decay of the two main flow structures continue along the compression stroke. The turbulence reduction rate is lower than that of the intake stroke and the distribution is more homogeneous, which is associated to convection and diffusion. Even though dissipation does take place, the turbulence is reasonably constant due to shear and compression stresses generated by the piston movement.

At the beginning of the compression stroke, the integral turbulent length scales are practically as large as the volume of the cylinder. As the piston approaches the T.D.C., these scales are reduced. The smallest structures are generated by the mean flow, and as it decays, they are consumed by dissipation.

About 30º before T.D.C. the sparkle is released, which marks the beginning of the combustion. The small flame region formed, takes advantage of the turbulence and propagates through the unburnt region. The energy is transported into the unburnt region by heat transfer (conduction, radiation), compression against the cylinder walls, and diffusion of the hydrogen atoms. The higher the turbulence intensity, the larger will the flame front contact area be and the better will it propagate.

The ordinary unleaded gasoline is formed basically by light hydrocarbons with 5 to 20 carbon atoms. It is obtained by physico-chemical processes such as separation, catalytic cracking, etc.. In Brazil, the gasoline is added 20-25% of alcohol (ethanol – $C_2H_5OH$). A consequence of the ethanol addition is that it becomes more knock resistant at the same time that allows higher compression ratios. Therefore, it is necessary to either change the ignition timing or reduce the size of the combustion chamber.

### 2.1 Turbulence Experimental Approach

Turbulence is traditionally defined as randomly occurring variations of a variable at one spot of the flow along the time. Its evaluation requires statistical analysis, by which the instantaneous value of that variable is equal to the sum of a mean value and a fluctuation about the mean:

$$K(t) = \overline{K} + k(t) \tag{1}$$

$K(t)$ - instantaneous value of K

$\overline{K}$ - time-average of K

$k(t)$ - fluctuation of K about $\overline{K}$

When it comes to internal combustion engines the mean value and the fluctuation evaluated along the time do not have a clear meaning due to the alternating characteristic of the flow. An adaptation of the expression shown above is therefore required. The instantaneous value of the variable is defined in terms of a ensemble crank angle average and a fluctuation about the mean.

$$K(\theta) = \overline{K} + k(\theta) \tag{2}$$

$K(\theta)$ - instantaneous value of K at angle $\theta$
$\overline{K}$ - ensemble average of K
$k(\theta)$ - fluctuation of K about $\overline{K}$

The turbulence intensity is equal the r.m.s. of $k(\theta)$ divided by $\overline{K}$ and is expressed in percentage.

There is another approach which considers another component of the instantaneous value of the variable nevertheless, the regular ensemble average approach has been employed in this work.

**3. The Experimental Apparatus**

Pressure, temperature and velocity have been measured inside the combustion chamber along the first two strokes. The measurement set up comprised of a pressure transducer, a constant current and a constant temperature anemometers. The three devices have been adequately calibrated according to the requirements of this study. The data acquisition was done by means of a dynamic signal apparatus with a floppy disk drive.

**3.1 The Piezoelectric Pressure Transducer**

A comparison pressure device has been employed for the static calibration of the piezoelectric pressure transducer. The charge amplifier output signal was associated to the reading of a 40bar reference manometer which is regularly calibrated. Four curves have been constructed and a cubic curve has been fitted to the curve of mean values. The results are shown on Fig. (2).

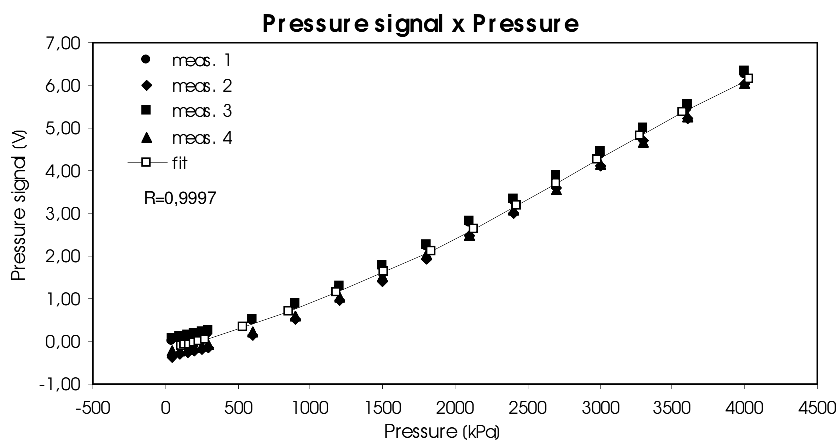


Figure 2. Static calibration curves of the piezoelectric pressure transducer.

**3.2 The Constant Current Anemometer**

The wire anemometer associates the changes in the variable of interest, either temperature or velocity, to the heat transfer from the heated wire to the fluid. This transfer occurs mainly by convection. Once the convection heat transfer coefficient is a function of both the temperature and velocity of the fluid, it is necessary to separate these two properties when analyzing the anemometer temperature output signal. The easiest way is to adjust the current of the constant current anemometer so that it becomes insensitive to velocity. This is done by calibrating the probe at different flow velocities and subsequently comparing the results. If the curves constructed are in good agreement, e.g. their dispersion is considerably low, then the current chosen is such that it allows for greatest temperature sensitivity without sensing the flow velocity. The results are shown in Fig. (3).

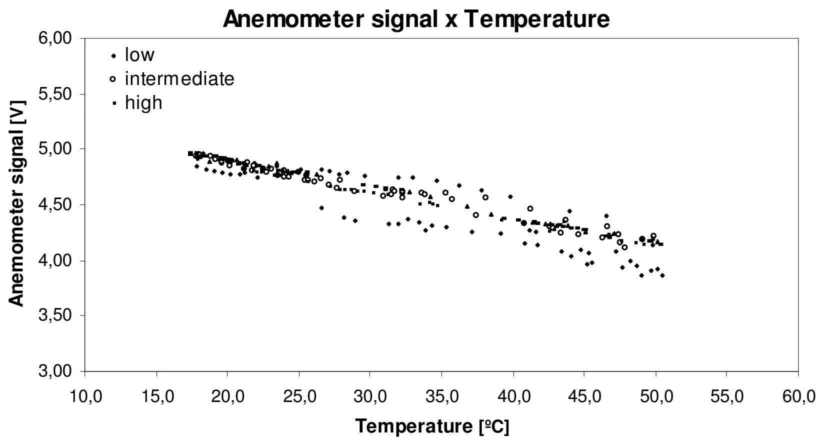


Figure 3. Constant current anemometer current adjustment calibration curves at low, intermediate and high flow velocities.

These results lead to the conclusion that the signal obtained at this particular current value refers exclusively to temperature changes while measuring air flows.

The next step for the preparation of the constant current anemometer is to calibrate the wire probe. In order to do so, a hot bath and a Pt100 type reference thermometer have been employed. The anemometer output signal has been associated to the temperature values to plot the calibration curve.

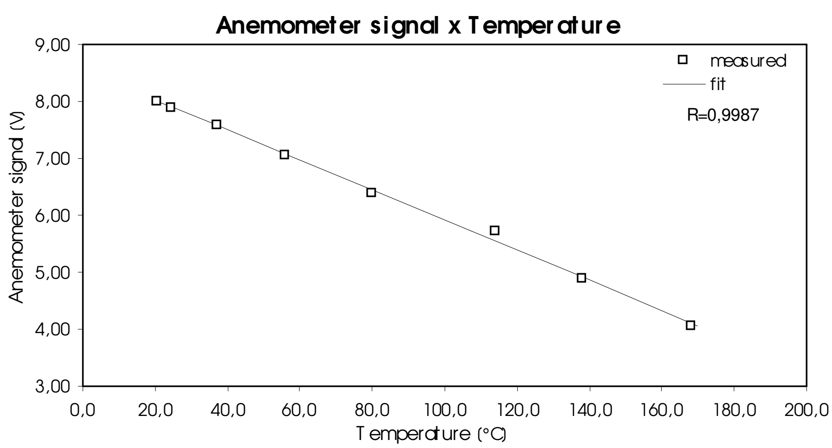


Figure 4. Constant current anemometer calibration curve.

The probe has been calibrated in air and the measurements were to be made in air/fuel mixture. In order to estimate the consequent error from this fluid difference a series of calibrations in alcohol, water and mixtures of both have been made. Figure (5) shows the curves obtained. Taking the water curve as the reference, the maximum error is about 1.64% which is acceptable for combustion chamber measurements.

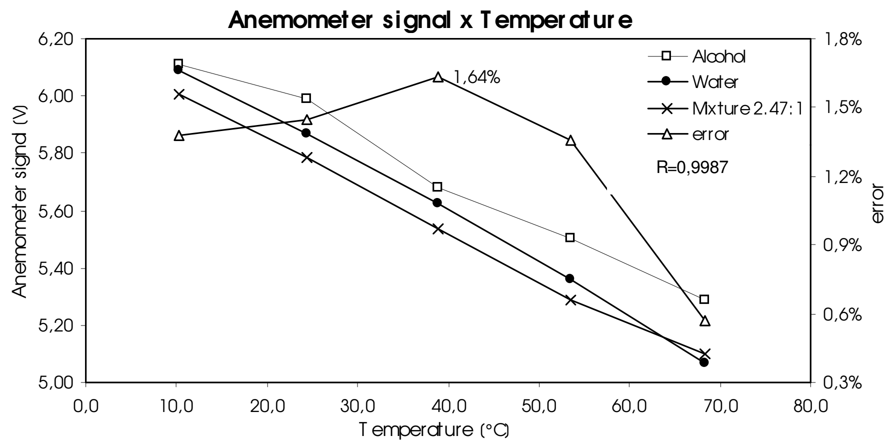


Figure 5. Constant current anemometer calibration curves obtained in alcohol, water and one mixture of both.

The anemometer is known for showing very fast response to temperature step stimuli. The probe has been submitted to a temperature step by immersion. Figure (6) shows the behavior of the signal and its corresponding normalized cumulative integral which allowed for the determination of the beginning of the step. The time constant is equal to 0.08ms.

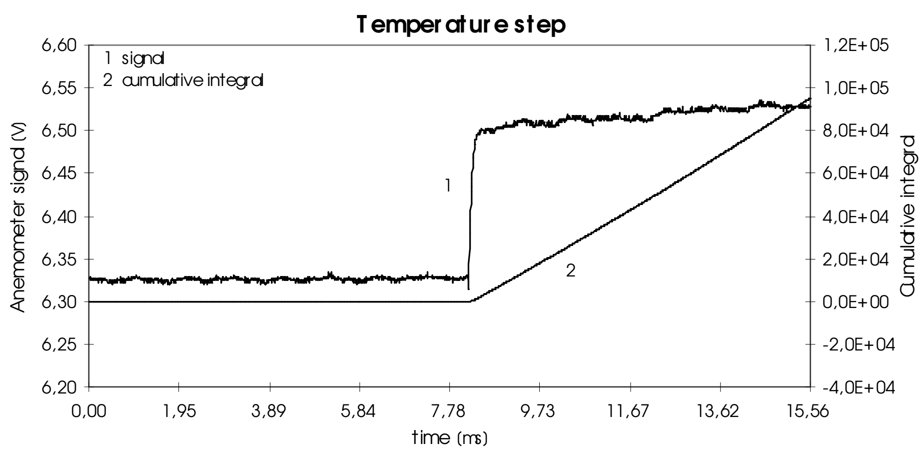


Figure 6. Anemometer response to a temperature step and corresponding normalized cumulative integral.

**3.3 The Constant Temperature Anemometer**

The constant temperature anemometer is used for measuring the velocity normal to the probe wire. As with the temperature measurements, it requires a static calibration. This has been done by means of a calibrated nozzle and a flow controller. Figure (7) shows the calibration curve obtained.

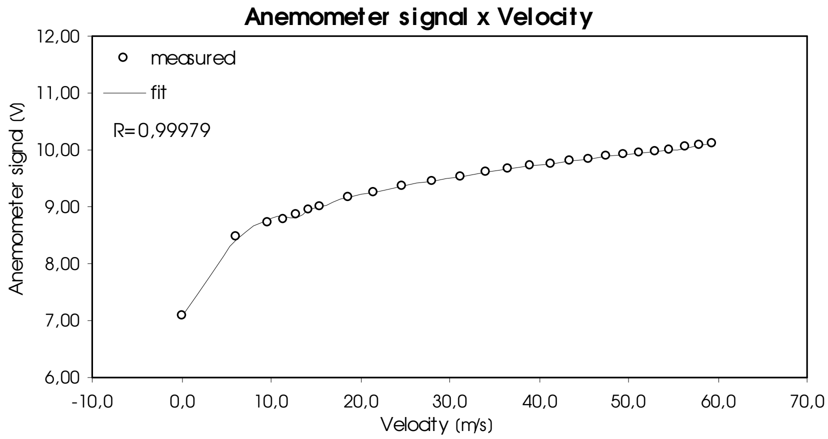


Figure 7. Constant temperature anemometer calibration curve.

There is a sudden leap between the first and second points which is peculiar to constant temperature anemometers. This behavior justifies the difficulties of measuring low velocities. Several attempts have been made in order to smoothen the curve around these points but have all turned out unsuccessful given the means available. In addition to this, previous experience shows that the velocity range to be measured was not in that region.

The simplest and perhaps most crucial test to be made is to certify the integrity of the wire under real motoring conditions. It can not withstand combustion, still the flow conditions within the combustion chamber are quite severe. No wire breakage of the probe has been observed but it has to be cleansed with isopropilic alcohol every 20 minutes.

### 3.4 The Dynamic Signal analyzer

All data has been acquired by means of a dynamic signal analyzer with a 3"½ disk drive. This device allows two channel simultaneous acquisition at higher sampling rates than required by this study. The rate used was of 2048 samples in 127ms. The data has not been filtered in any way as it would alter the fluctuation results.

### 3.5 The Test Engine

A 1.3 litter carbureted engine has been used for this study. The turbocharger system required the insulation of the carburetor, a stronger fuel pump and a heat exchanger for the oil line. The engine has been set up on a dynamometric bench which allowed the measurement of torque, speed and fuel consumption.

## 4. The Experimental Procedure

Wire probes require a direct access to the flow. The probe used has been adapted to fit the spark plug hole and therefore the results are relative to a non burning environment at the spot where the combustion first begins. Even though there is no combustion and this work aims to somehow infer on this very process, the measurements allow the understanding of part of the preparation of the mixture for the burning process which takes place during intake and compression.

The piezoelectric pressure transducer has been screwed to the side of the combustion chamber and had access to its interior through a $\phi$2mm channel.

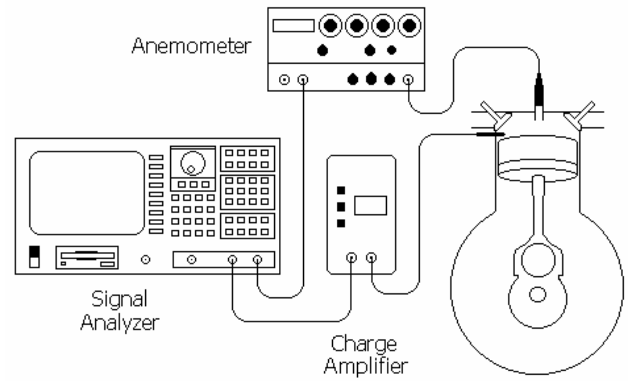


Figure 8. Experimental set up.

Prior to any test, the engine was worm up for about 20 minutes and then the sensors which had been previously put in stand by for several hours were installed. Care was taken to use the same cables during calibration and engine tests. As mentioned before, every 20 minutes the wire probe had to be cleansed to remove any eventual deposits.

The test engine parameters were kept constant during tests for both engine types: 2500rpm, 75% of normally aspirated full load which corresponded to 79,4±1N read on a load cell.

## 5. Results and Discussion

Figure (9) shows the ensemble average pressure and temperature curves obtained for the normally aspirated and turbocharged engines as a function of crank angle. The pressure peak values coincide in time and since there is no combustion they both occur at the T.D.C.. The coincidence of the pressure peaks allowed the use of this signal for aligning the respective temperature of velocity curves.

As expected, during intake the pressure inside the cylinder of the turbocharged engine is higher than the local atmospheric pressure while for the normally aspirated it is lower.

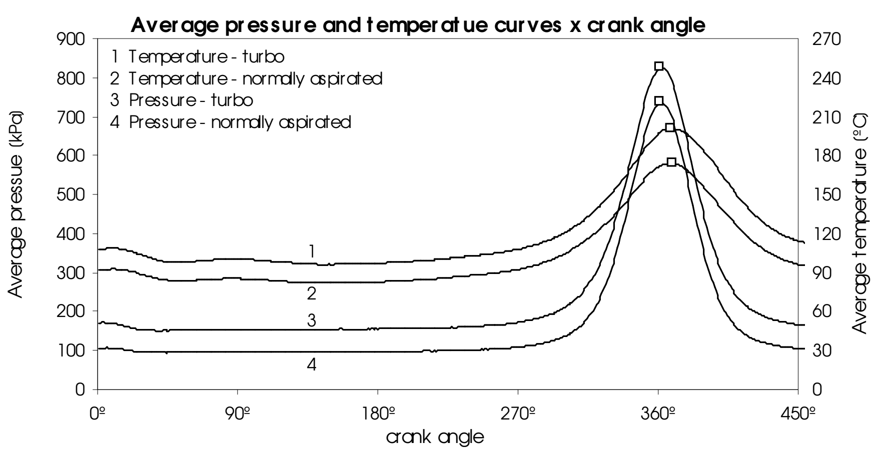


Figure 9. Ensemble average pressure and temperature curves.

An unexpected behavior of consecutive pressure curves has been observed. As the measurements went on, consecutive curves maintained their shape but shifted progressively by a constant factor along its extension to lower values. The possibility of sensor overheat was diminished by cooling it with an air jet. As new series of measurements were performed the first curves tended to coincide with the first curves of previous series, characterizing a modifier input. The first curve of each series of measurements has been used as a reference for the displacement of the subsequent curves by a factor equal to the average of the shift evaluated at each point.

The peak temperature for the turbocharged engine is outside of the calibration range (refer to Fig. 4). Previous experiments have shown that the behavior of the constant current anemometer is sufficiently linear to allow extrapolations up to close values. The calibration range has been reduced due to physical constraints of the probe.

Similarly to what occurred to pressure, the temperature peaks are coincident but shifted in crank angle value by about 5º after T.D.C.. There are two possible causes for this non coincidence between pressure and temperature peaks. The first is that the heat transfer between wall and wire is significant. The wall absorbs heat during compression and releases it during intake. The second is that the sensor could be either within the thermal boundary layer or within a region of relevant recirculation, where the heat flow is very different from the free stream region.

Another observation to be made is that the fundamental gas law may not be directly applied to engines. The non coincidence of the pressure and temperature peaks would be then unacceptable. It is possible to conclude that the behavior of the constant current anemometer is not determined by the pressure, once again certifying the adequacy of the current adjusted.

From all the measurements so far the one requiring most caution is the velocity. The wire temperature is about 600ºC which makes it prone to breakage.

The velocity signal is more sensitive to pressure than the temperature signal. That is why a correction relation based on the Nusselt and Reynolds numbers is required when the values of velocity in m/s are to be obtained. For such a correction it is necessary to measure the pressure, temperature and velocity simultaneously. The use of averaged temperature values would lead to erroneous fluctuation values and consequently misleading turbulence intensity results. None of the correction methods available is feasible with the means available.

Due to material limitations the results obtained are based on the analysis of the velocity signal in volts instead of real values in m/s. The turbulence intensity (T.I.) has been normalized by the ensemble average of the signal and expressed in percentage value. Figure (10) shows the results obtained.

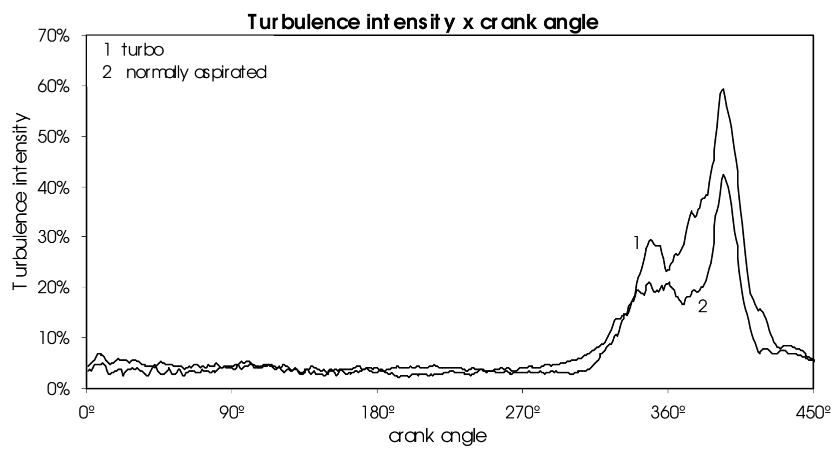


Figure 10. turbulence intensity curves.

Up to the T.D.C. the T.I. curves for both engines is very similar. The difference increases by the end of the compression stroke. According to Lancaster (1976) the r.m.s. fluctuation is greater during intake but the comparison of the results is not possible once no percentage data is presented. The ignition timing is about 30º before T.D.C.. At this point the difference in T.I. is about 10%.

From what has been presented the conclusion is that the air/fuel mixture undergoes a more intense homogenization process in the turbocharged engine, which along with higher pressure and temperature results in a better preparation of the mixture for combustion.

**6. Conclusions**

In this work, the flow within the combustion chamber of S.I. engines has been evaluated so that inferences on the influence of the turbocharger on part of the combustion process could be made. Some characteristics of the flow in terms of pressure, temperature and velocity during the intake and compression strokes have been studied.

- Even though more precise measuring devices do exist, the anemometer is suitable to this analysis due to its time response and repeatability;
- The value of current adjusted to the temperature measurements is adequate;
- The linearity of the constant current anemometer allows for the extrapolation of the calibration curve;
- The adequate comparison with other works requires the correction of the velocity signal;
- The turbulence intensity, based on the r.m.s. fluctuation and the ensemble average o f the velocity signal indicated that the excitation of the air/fuel mixture is more intense in the turbocharged engine. Although it is

observed along the whole intake process, it is more intense later in the compression stroke, where it reaches 10%;

- In order to state that the increased turbulence intensity together to higher pressures and temperatures are the actual reason for lower S.F.C. in the turbocharged engine, measurements of CO emissions have to be performed. These measurements would allow the testing of the combustion efficiency of both engines.

## 7. References


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