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RESEARCH ARTICLE

TURBULENCE INTENSITY VARIATION WITHIN THE COMBUSTION CHAMBER OF A SI ENGINE
DUE TO TURBOCHARGING

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ABSTRACT

By turbocharging the SI-engine, one is able to reduce the specific fuel consumption as well as the emission of certain pollutants without power losses (Vianna *et al.*, 1989). In order to determine the causes for these reductions, measurements of pressure, temperature and velocity inside 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. As compared to the normally aspirated engine, the turbocharger brings about higher pressure, temperature and turbulent intensity along the whole scope of analysis. The mixing process and consequently the quality of the burning of the fuel are improved

Key words:

Turbocharging,
SI Engine,
Turbulence Intensity.

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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%. Still, it is impractical 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. Although there are theoretically several possibilities to improve engine efficiency, their implementation is limited by fuels, internal aerodynamics and even construction materials. In this context, the careful use of turbochargers has successfully improved reciprocal engine efficiency. In a previous work, Vianna *et al.* (1989) have compared the performance of a turbocharged engine and its normally aspirated version. 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 turbocharged engine is about 8% lower than that of the normally aspirated, Fig. 1. Besides that, the turbocharged engine operates at higher 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. Besides the fact that more mixture is admitted into the cylinder, 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. 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.

In-cylinder flow

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

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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 intake process continues, 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.

mean value and a fluctuation about the mean. 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 alternative characteristic of the flow. An adaptation of the definition presented above is therefore required. The instantaneous value of the variable is then defined in terms of an ensemble crank angle average and a fluctuation about the mean.

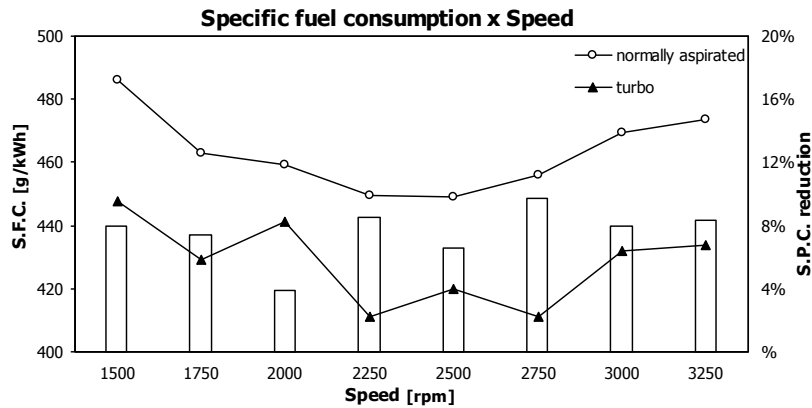


Figure 1. Specific fuel consumption for the normally aspirated and turbocharged engines

The decay of the main 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 (Benson, 1982). In 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. The ignition timing, which marks the beginning of the combustion, is about 30° before TDC. The small flame region formed, taking advantage of the turbulence 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 surface be and the better will it propagate.

The regular unleaded gasoline is formed basically by light hydrocarbons with 5 to 20 carbon atoms. It is obtained by physical-chemical processes such as separation, catalytic cracking, etc.. In Brazil, the gasoline is actually added 25% of alcohol (ethanol – C₂H₅OH). A consequence of the ethanol addition is that it becomes more knock resistant requiring higher compression ratios. Therefore, it is necessary to alter the ignition timing once compression ratio changes would require physical modifications.

Turbulence Experimental Approach

Turbulence is traditionally defined as randomly occurring variations of a variable in 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

$$K(\theta) = \bar{K} + k(\theta) \dots\dots\dots(1)$$

$K(\theta)$, \bar{K} and $k(\theta)$ are the instantaneous value of K , its ensemble-average and the fluctuation about \bar{K} , respectively. The turbulence intensity k' is equal to the r.m.s. of $k(\theta)$ normalized by \bar{K} and expressed in percentage. There is still another approach which considers two components of fluctuation about the mean but the regular ensemble average approach has been employed in this work instead.

The Experimental apparatus

Pressure, temperature and velocity have been measured inside the combustion chamber along intake and compression strokes under motoring conditions. Three cylinders of a 1.3 liter four-cylinder petrol engine were kept active while the one left served as test cylinder. 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 analyzer with a floppy disk drive.

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 40 bar reference manometer which is regularly calibrated. Four curves have been constructed and the third degree polynomial fitted to the mean curve. The results are shown on Fig. 2.

The Constant Current Anemometer

The wire anemometer associates changes in the variable of interest, temperature or velocity, to heat transfer from the

heated wire to the fluid. This transfer occurs mainly by convection. Since the convection heat transfer coefficient is a function of both the temperature and velocity of the fluid it is necessary to be able to distinguish these two variables when analyzing the anemometer temperature output signal. The most efficient way is to adjust the current of the constant current anemometer so that it is 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.

These results lead to the conclusion that the signal obtained at this particular current value refers exclusively to temperature changes when measuring air flows. The next step on 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. 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 have been made. Fig. 5 shows the curves obtained.

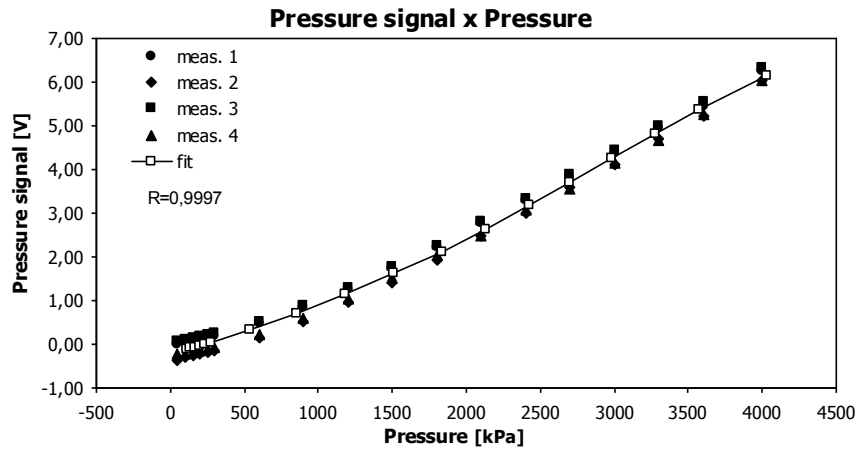


Figure 2. Static calibration curves of the piezoelectric pressure transducer

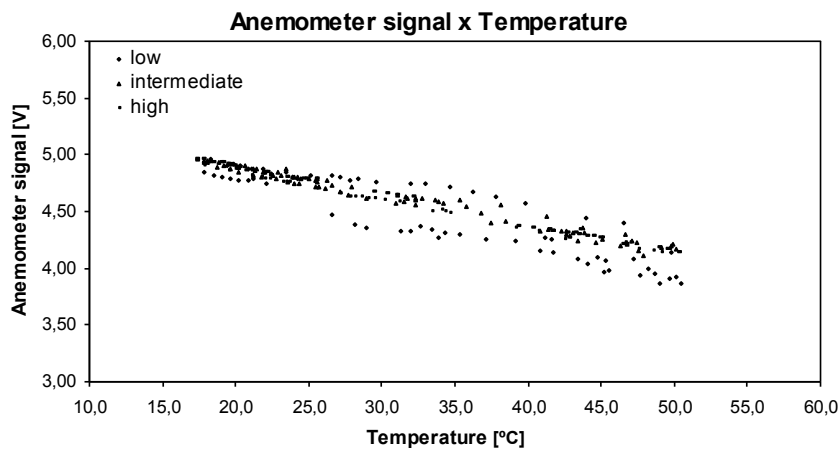


Figure 3. Constant current anemometer current adjustment calibration curves at low, intermediate and high flow velocities

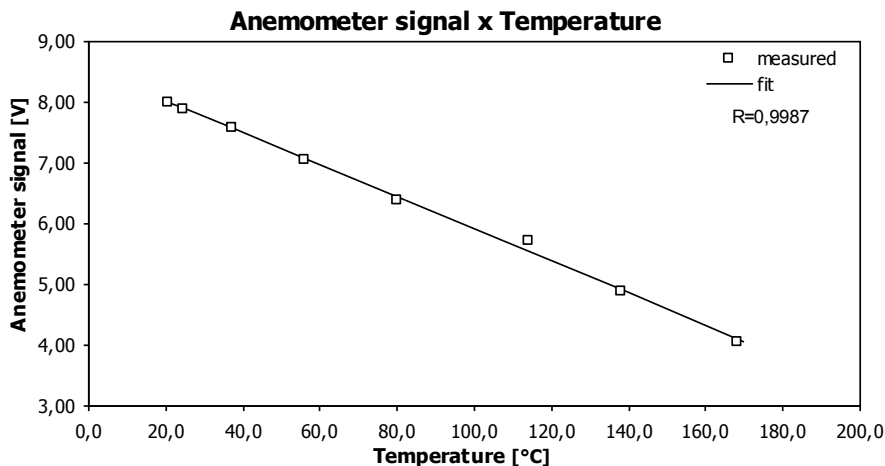


Figure 4. Constant current anemometer calibration curve

Having the water curve as the reference, the maximum error is only about 1.64% which is acceptable for in-cylinder measurements. The anemometer is known for showing very fast response to dynamic stimuli. The probe has been submitted to a temperature degree by immersion. Fig. 6 shows the behavior of the signal and its corresponding normalized cumulative integral which allowed for the determination beginning of the step. The time constant is equal to 0.08 ms.

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. It has been done by means of a calibrated nozzle and a flow controller. Fig. 7 shows the calibration curve obtained.

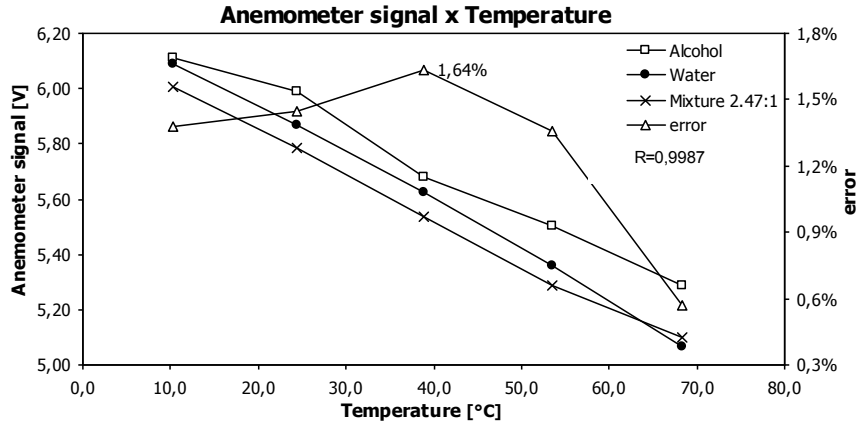


Figure 5. Constant current anemometer calibration curves obtained in alcohol, water and one mixture of both

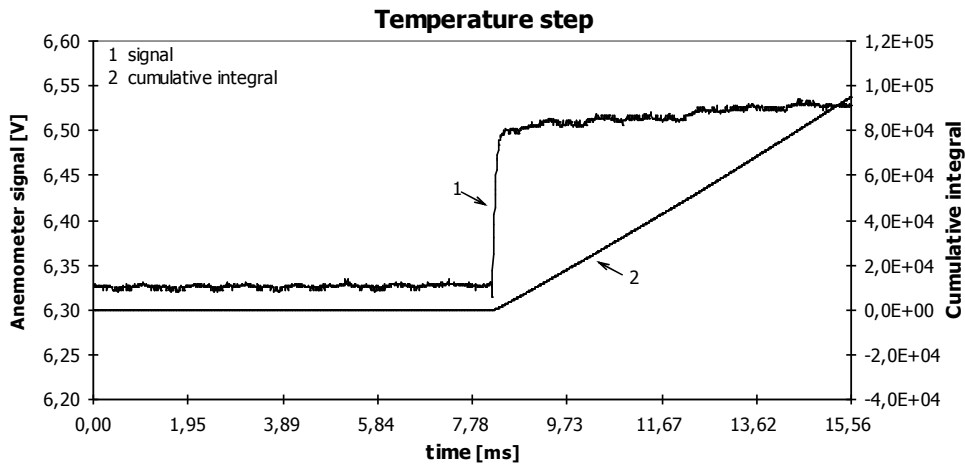


Figure 6. anemometer response to a temperature step and corresponding normalized cumulative integral

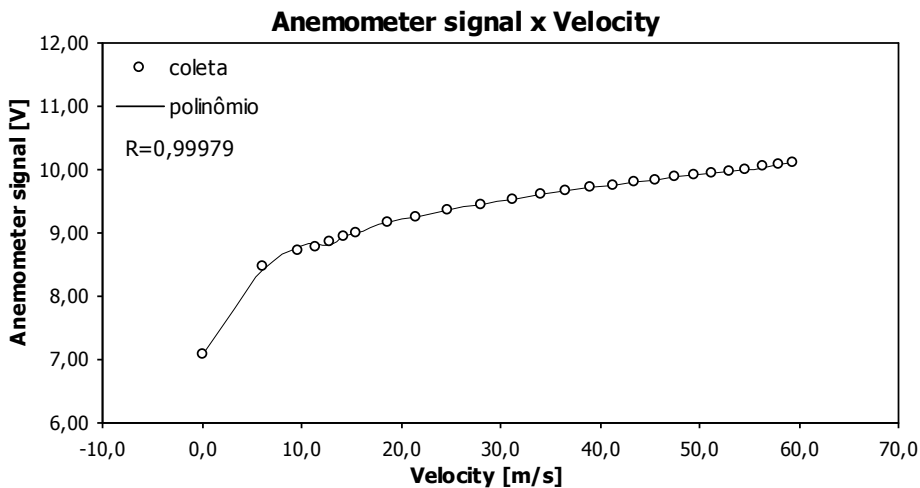


Figure 7. Constant temperature anemometer calibration curve

There is a sudden leap between the first and second points of the calibration curve 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. Still, 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 attest the integrity of the wire under motoring conditions. It can not withstand combustion, still the flow conditions within the combustion chamber are quite aggressive. After the tests no wire breakage has been observed but it has to be cleansed with isopropyl alcohol every 20 minutes.

The Data Acquisition System

All data has been acquired by means of a dynamic signal analyzer with a floppy 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 127 ms. The data has not been filtered in any way as it would alter the fluctuation results.

The Test Engine

A 1.3 liter four-cylinder 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 allows for torque, speed and fuel consumption measurements.

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 motoring conditions at the spot where the combustion under normal conditions begins. Even though there is no combustion and this work aims to somehow infer on this very process, the measurements allow for 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 cylinder and had access to its interior through a $\phi 2$ mm hole.

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 four hours were installed. The anemometer was kept operating from calibration to engine use except during transportation from one spot to another which did not take longer than five minutes. 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 depositions. The test engine parameters were kept constant during tests for both engine types: 2500 rpm, 75% of normally aspirated full load which corresponded to $79,4 \pm 1$ N read on a load cell.

RESULTS AND DISCUSSION

Fig. 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 and 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. 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 made 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, Fig. 4. Previous experiments have shown that the behavior of the constant current anemometer is sufficiently linear to allow extrapolations up to close values.

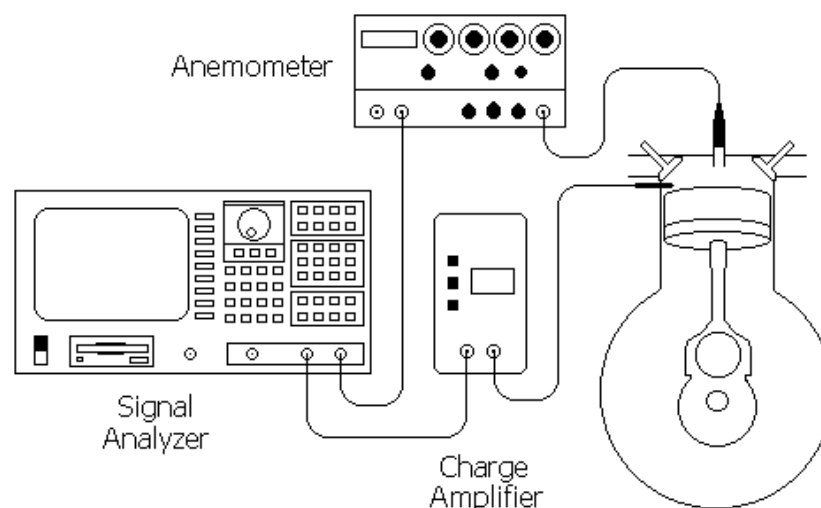


Figure 8. The experimental set up

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 5° after T.D.C. There are two possible causes for this non coincidence between pressure and temperature peak times. 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 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 only be applied to quasi-static processes, which is not the case with engines. From all the measurements so far the one requiring most caution is the velocity.

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 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 conclusions. None of the correction methods available is feasible given the means available. The turbulence intensity (T.I.) has then been normalized by the ensemble average of the signal and expressed in percentage value. Fig. 10 shows the results obtained. 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

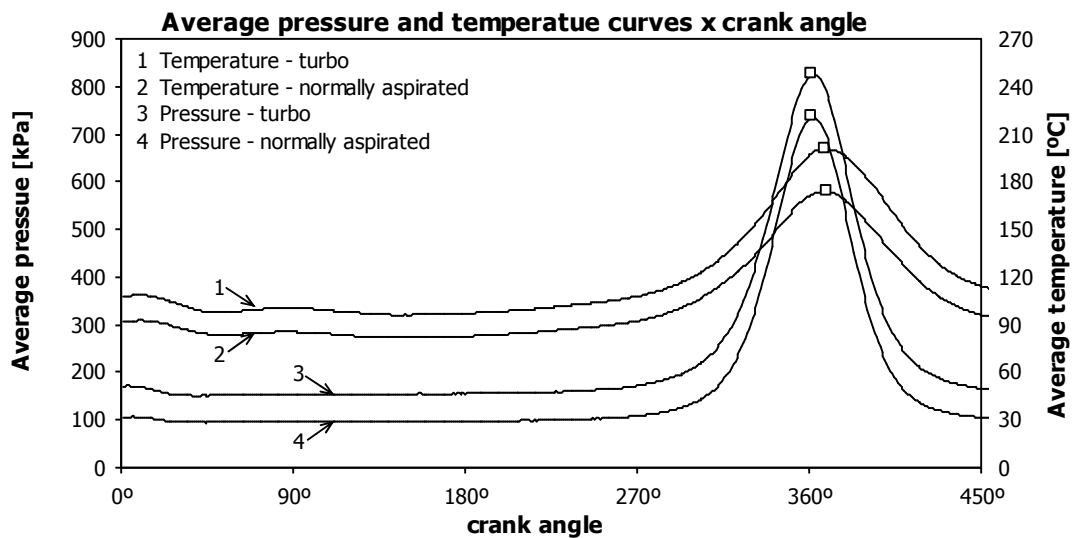


Figure 9. Ensemble average pressure and temperature curves

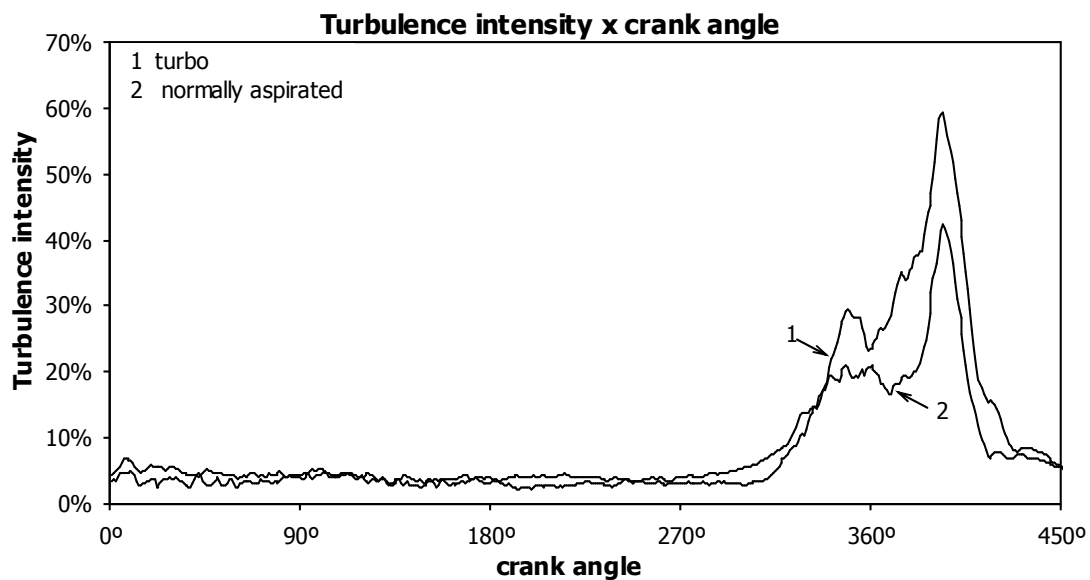


Figure 10. turbulence intensity curves

The wire temperature is about 600°C which makes it even more suitable to breakage. The velocity signal is more sensitive to pressure than the temperature signal. That is why a

data is presented. At the ignition timing 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 result in better preparation of the mixture to combustion.

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 turbulence intensity 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 turbulence intensity, based on the r.m.s. fluctuation and the ensemble average of the velocity signal indicated that the excitation of the mixture is more intense in the turbocharged engine. It is observed along the whole intake process but is more intense late in the compression stroke where it reaches 10%

- In order to affirm that the increased turbulence intensity together with higher pressures and temperatures are the actual reason for lower S.F.C. in the turbocharged engine, measurements of CO emissions had to have been performed. These measurements would allow the testing of the burning quality of both engines.

REFERENCES

- Benson, C., 1982. "The Thermodynamics and Gas Dynamics of Internal-Combustion Engines", Clarendon Press, *Oxford – Ing.*
- Cruz, P. T. A., 2003. "Avaliação do Escoamento no Interior da Câmara de Combustão de Motores do Ciclo Otto", Universidade de Brasília.
- Vianna, J.N.S. Fraga, A.G., reis, A.V. Oliveira, A.B.S., Sousa, M.T., 2005. "Reduction of Pollutants Emissions on SI Engines - Accomplishments With Efficiency Increase", *Revista Brasileira de Ciências Mecânicas*, 2005 (to be published).
