Serrano, J.; Tiseira, A.; García-Cuevas González, LM.; Rodriguez-Usaquen, YT.; Guillaume, M. (2020). A methodology to study oil-coking problem in small turbochargers. International Journal of Engine Research. 21(7):1193-1204. https://doi.org/10.1177/1468087418803197

# A Methodology to study oil-coking problem in small turbochargers

J.R. Serrano[a], A.Tiseira[a], Luis Miguel García-Cuevas[a], T. Rodriguez Usaquén[a] , Mijotte Guillaume[b]

[a] CMT-Motores Térmicos, Universitat Politècnica de València, Spain

[b] Groupe Renault, Guyancourt, France

## Abstract

In compliance with oncoming emission directives, turbocharging and increasing complexity in the turbocharger system demands a great effort from researchers on the development of effective procedures and tools to cope with the new technological exigencies. This paper describes a methodology for studying oil coking influence in turbocharger performance. A preliminary evaluation and calibration is done. The aim of this work focuses on the development of methodologies and tools that help to evaluate and understand the consequences that degraded oils can generate in the bearing system during enhanced oil-coking procedure. Several experimental tests have been carried out in an engine test bench and using an independent lubrication system that only feeds the turbocharger. The test campaign is done under a specific engine cycle and using oil artificially contaminated at two different levels.

The work is divided into two parts. The first part provides a description and definition of test conditions for measuring of the maximum temperature in the bearing system and the second part tackles the measurement and post-processing of the main instantaneous parameters defining the engine and turbocharger behavior.

## 1. Introduction

Reciprocating petrol and diesel engines usually have a turbocharging system to increase the power, reduce consumption, reduce the emission of pollutants and adequate the size of the engine to required engine performance. Generally, the turbocharging system can have one or more turbochargers depending on the values of power or torque to be achieved by the engine [1].

In the turbocharger, a common shaft connects the turbine and the compressor. The movement of the shaft is limited to avoid metal-to-metal contact by radial and axial bearings. These bearings are tipically floating ring, semi-floating ring or ball bearings [2][3]. The bearing system supports the rotor assembly and is located in the central housing of the turbocharger. It must reliably position and support the sudden changes of rotor loads and speeds, the later can typically reach up to 280000 rpm in small turbochargers[4].

The bearing system has influence on critical rotor speeds, vibration and shaft instability. In recent years, the use of high-performance materials, load-capable design and robustness has been considered and implemented in new bearing technologies [5]. Whirl and frictional characteristics in floating ring bearings and ball bearings have been examined and evaluated [6] showing that both bearing present synchronous motions caused by vibrational modes and sub-synchronous motions presented in floating ring bearings as a result of oil film instabilities.

Oil degradation can be promoted due to thermal loads, liquid, gas or solid contamination. Ensuring an optimum lubrication quality is essential to ensure the successful operation of the turbocharger and the engine as a whole; if the system deteriorates, a decrease in engine performance can reach up to 20% in power and torque [7].

A big part of the energy from the exhaust gases that move the turbine can be lost due to friction [8], generating heat fluxes that increase the temperature of the oil that might even break the turbocharger. The temperature of the turbocharger environment also presents a challenge to the bearing system. If the engine is shut down immediately following a run at high power output, the turbine and turbine housing temperatures rise towards their upper limits, suddenly all gas flow through the turbine stops and all oil flow through the central housing stops, producing damage from overheating [9].

The common monitoring parameters for turbochargers are the oil pressure, turbine inlet gas temperature, compressor outlet temperature and rotating speed [10], [11]. When the turbocharger deteriorates, the engine control unit (ECU) can manage the turbocharger operation by VGT vane position [12] or waste-gate[13] , offsetting the problem and reaching the inlet pressure required but hiding any degradation that may gradually have the turbocharging system.

The oil is an element that must be carefully studied since it has influence on several of the systems of the engine. Several techniques have been developed to study engine wear sampling oil at regular intervals, as described in literature [14]. Of course, engine oil also suffers from degradation and it has to be changed periodically. The oil change is usually done as part of a preventive maintenance program when the engine odometer reaches a given threshold. Recently emerging technologies related to various sensors and wireless communication offer new opportunities to improve the efficiency of the operations of maintenance of automobiles, in particular, the implementation of predictive maintenance. The key point of predictive maintenance is to develop an algorithm that can analyze the state of degradation, level of contamination from engine wear, water vapour in the crankcase, acids and engine blow-by gasses and take preventive decisions. In one study for the application of predictive maintenance in an engine oil of a car, a new algorithm is proposed to determine the time of appropriate exchange of oil through the analysis of its degradation level[15].

A lubricant in service is subjected to a wide range of work that can degrade the oil base and additives. Heat, air, incompatible gasses, humidity, internal or external pollution, the combustion processes, the radiation and the unnoticed mixture with a different fluid influence such factors. The inclusion of combustion air increases the rate of elements that do not belong to the oil such as soot particles, nitrogen oxides, air etc. leading the oil oxidation, which is the primary mechanism of lubricant degradation [16].

Although most lubricants are formulated with antioxidants to control oxidation, this degradation cannot be avoided. Consequently, this can lead to an increase in viscosity and varnish formation. This condition can occur even when the oils are not especially old or contaminated, as well as in synthetic lubricants and hydraulic fluids thermally resistant. This type of degraded oil can leave traces in the turbine shaft and bearings affecting the dynamics of the turbocharger[17], [18].

The evaluation of the performance of an oil is difficult due to narrow access to the lubrication channels in turbochargers. However, in this work some parameters related to the dynamic of the turbocharger are evaluated to determine the impact that degraded oil has on the turbocharging system.

Furthermore, the results in this paper should contribute to further improvements in existing turbocharger heat transfer models [19]–[21] by adding a radial resolution of heat transfer phenomena in the turbocharger[22]. This should allow for fast calculations of the internal temperature distribution, and, thus the prediction of oil coke formation.

This work is divided into two parts. The first part describes the evaluation of thermal behavior in the bearing system by changing the engine speed and load, so the working conditions to perform endurance tests of oil coke in the turbocharger can be selected. The second part is the application of methodology and particular post-processing related to every instantaneous magnitude that is recorded during the different tests. The methodology allows studying the response of some variables selected of the engine and the turbocharger for understanding the complex problem of oil coking influence in turbocharger performance. Finally, the conclusions of the results obtained from the analysis are discussed.

## 2. Endurance test method

The journal bearings are generally lubricated by the engine oil, which is pumped through the turbocharger and acts as a lubricant and as a coolant. In this section, a methodology for measuring the thermal behavior to evaluate the maximum temperature in different points of the bearing housing under an oil-coking endurance cycle is shown. The proposed method allows passing through the engine map of speed and fuel consumption, in a relatively limited number of engine operating points (low, medium and high load), and, therefore, of tests.

### 2.1 System layout

The system layout has as main elements an engine (*E*), a turbocharger (*T*), and an independent lubrication system (*L*) that feeds the turbocharger and allows controlling mass flow, temperature and pressure of the oil. Figure 1 shows the fully instrumented test bench used for monitoring operating conditions of the engine and the turbocharger performance during hot stops. Table 1, shows the general parameters of the engine used.

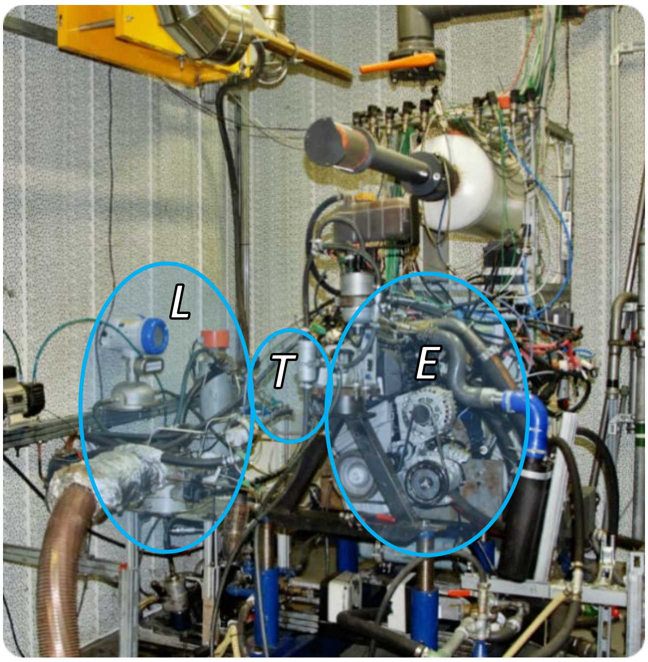


Figure 1. Engine test bench

*Table 1. General parameters of the engine*

| Technical specifications | |
|---|---|
| Engine Type | Compression ignition Direct injection |
| Displacement | 1.461 cc |
| Compression ratio | 15.2:1 |
| Injection system | Diesel common rail |
| Power | 110 hp |
| Max. torque | 260 N·m |

During each test, the parameters of engine speed, torque, turbine gas inlet temperature, as well as oil inlet temperature for the turbocharger have been controlled and held constant.

The input variables of pressure, temperature, mass flow and speed have been measured for the turbocharger. Table 2 synthesizes the absolute uncertainties of the sensors installed in the engine test bench.

*Table 2. Specifications of the transducers used in the experiments, including their expanded uncertainty.*

| **Variable Measured** | **Type** | **Range** | **Uncertainty** |
|---|---|---|---|
| Temperatures | K-Type thermocouples Diameter: 1.5 mm | 273 K to1100K | $\pm$2.2 K |
| Internal Temperatures | K-Type thermocouples Diameter: 0.5mm | 273 K to 1100 K | $\pm$1.5 K |
| Oil mass flow | Coriolis flow meter | 0 to 100 kg/h | 1 kg/h |
| Compressor and turbine mass flow | Thermal flow meter | 0 to 600 kg/h | $\pm$2% |
| Pressure | Piezo resistive | 0 to 500 kPa | 0.05% full scale |
| Turbocharger speed | Inductive | 0 to 400 krpm | 500 rpm |

## 2.2 Definition of test conditions and measurements

The definition of test conditions is done by combining a characterization procedure. This includes calibration of engine speed, fuel consumption, and time needed to let the turbocharger to reach thermal stabilization and the maximum temperatures in the bearing system, taking into account its thermal inertia.

The study evaluates the temperature evolution at different parts of the turbocharger by using the engine cycle shown in Figure 2. This sequence allows to study oil coke formation inside the turbocharger bearings, and parameters of the engine such as temperatures, pressures or torque. Automotive companies predefine these cycles in order to ensure that the studied parameters do not exceed some limits that are detrimental for the engine life.

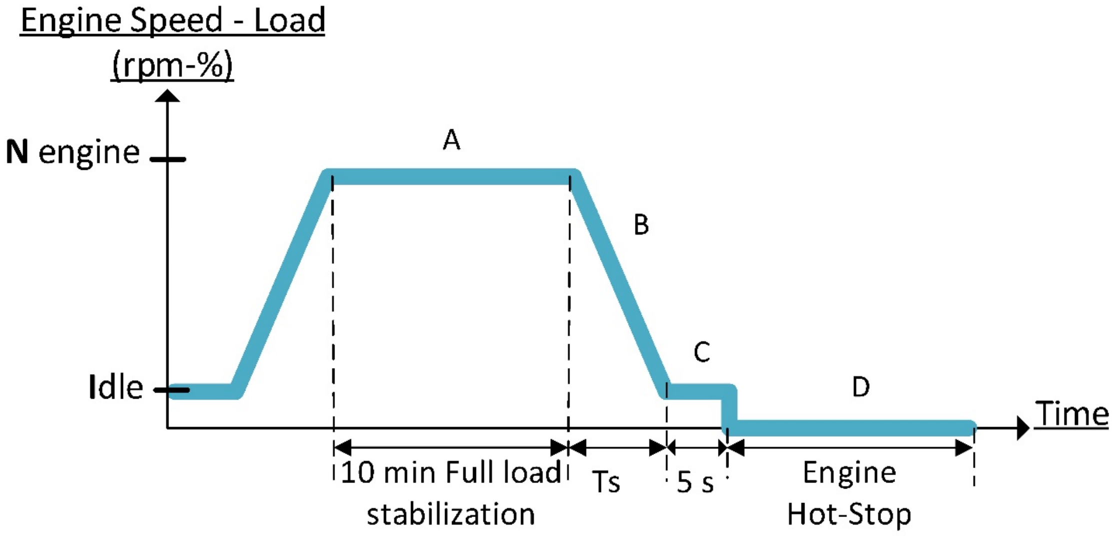


Figure 2. Engine test cycle

The engine cycle is divided in four main parts:

- A stabilization phase with steady state conditions (A in figure 2). At the end of this phase, the temperatures in the bearing system should be constant.
- A deceleration phase (B in figure 2) done with a calibrated duration according to the previous operating point.
- Idle conditions (C in figure 2). The engine is kept in these conditions during a small amount of time.
- A quick stop of the engine, in which circulation of water, oil and air through the turbocharger are cut for an allotted period (D in figure 2). This hot stop phase produces peak temperatures in the central housing and, thus, in the bearings sometime after stopping the engine.

This engine cycle is representative of conditions that resemble the real driving, such as the case of a hot stop after driving on a highway. These driving conditions can promote the formation of coke inside the turbocharger, due to the high temperatures reached in the oil trapped inside.

Figure 3 shows the method steps followed to study one of the most common failing conditions in turbochargers.

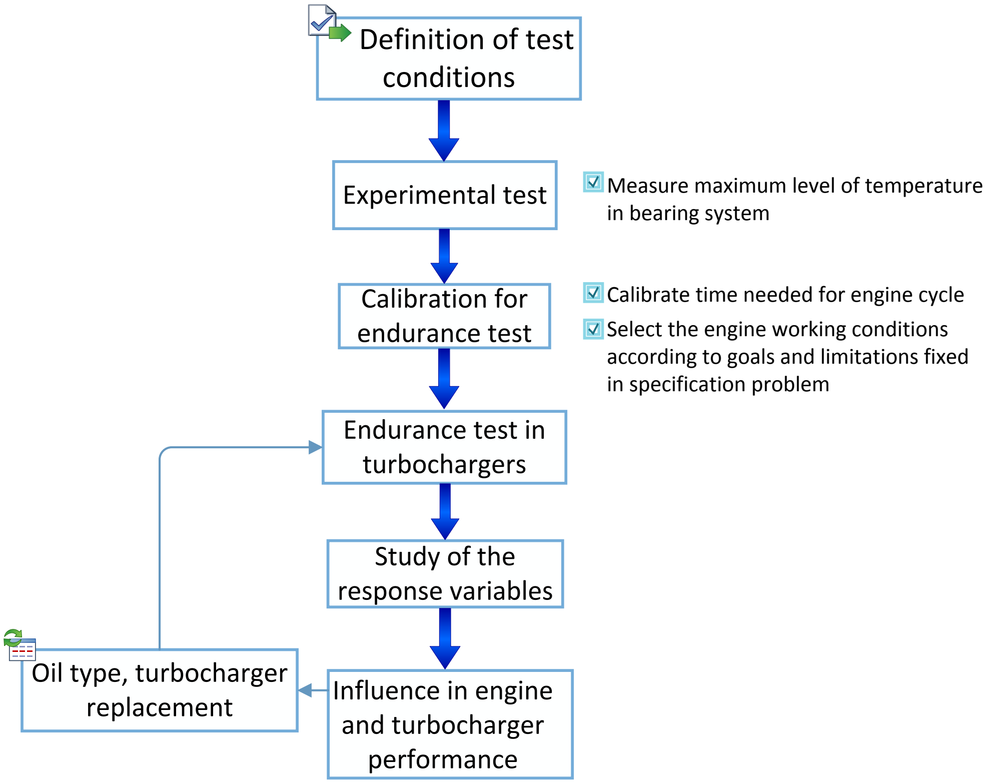


Figure 3. Steps to study oil coke formation in turbochargers

## 2.3 Calibration for endurance test in turbocharger.

A cycle as the one shown in figure 2 is repeated for different speeds and loads for characterizing the bearing system. Characterization results are considered valid if the temperatures of oil, bearings and housing have been stabilized and if under engine stop, no water, air or oil is flowing through the turbocharger.

Evolution of temperatures measured for the whole cycle (steady and transient conditions) are shown in figure 4 for two segments of the turbocharger: close to the radial bearing on the turbine side and at the bearing housing (drain wall), both which are influential in the coke formation due to high temperatures reached on them. Turbine and compressor temperatures as well as fluid temperatures have been monitored and stabilized before engine hot stop.

Temperatures are stabilized over 10 minutes (see figure 2 and figure 4). When the engine is stopped (Phase D in figure 2), the temperature in the bearing system rises and reaches its maximum peak values in the cycle (see figure 4). The heat exchanges during hot stop are mainly due to energy accumulated in the material and not due to the turbine gas temperature. The temperatures rise because of internal heat transfer through the turbocharger housing. These maximum temperatures must conform to the technical specifications of the component, or heat induced damage may develop.

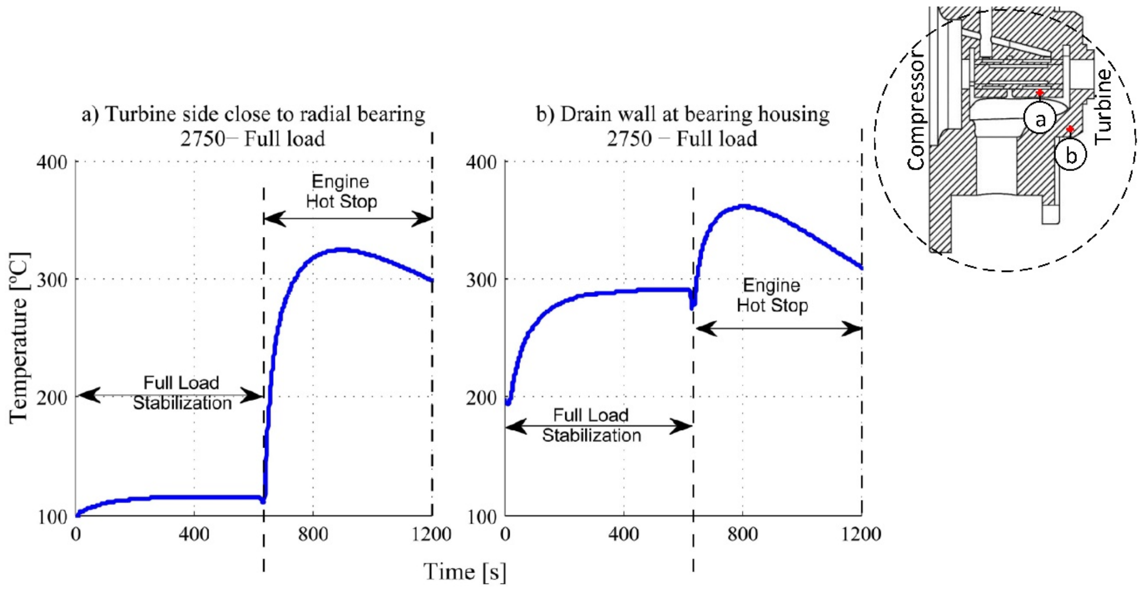


Figure 4. Turbocharger temperatures in engine cycle

Figure 5 shows the maximum wall temperature during engine hot stop (D in figure 2) for the sweep of engine operating points of engine speed and torque measured in the radial bearing (turbine side) and the bearing housing (drain wall).

The temperatures shown in Figure 5 are not correlated with exhaust temperature. During the engine hot-stops the temperatures in the radial bearing and drain wall rise due to internal heat transfer through the turbocharger. The exhaust temperature drops and can be correlated with the temperature in the turbine housing or back plate turbine that loss heat due to external convection and radiation once the engine is stopped. In addition, after 10 minutes of the engine shut down the turbocharger gets a thermal stabilization. The compressor side, central housing and Turbine side reach a convergence in temperatures.

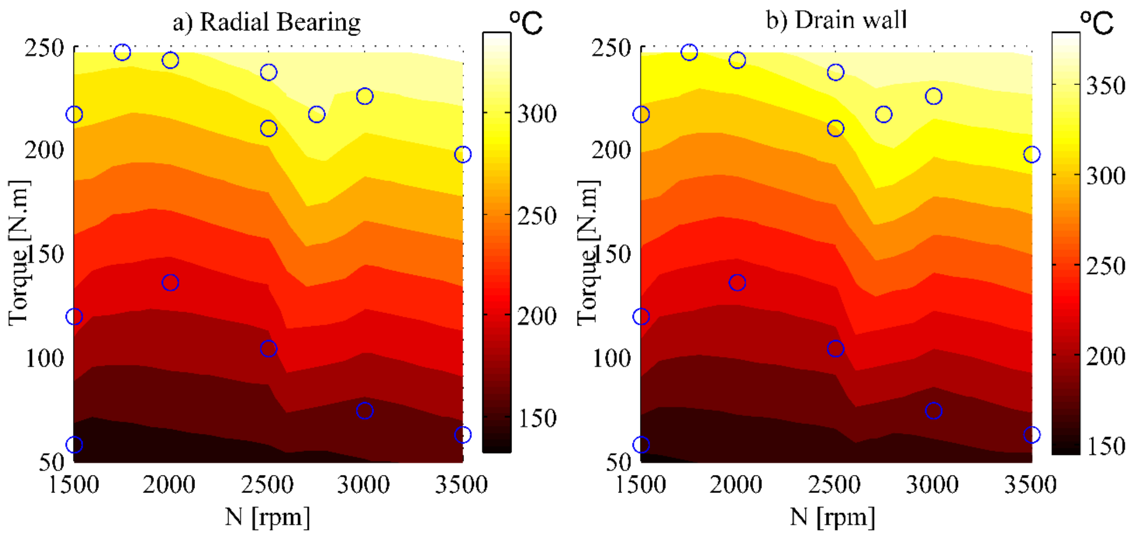


Figure 5. Turbocharger temperatures under hot stop

In order to illustrate the method, two steady operating points of the engine at high temperatures have been selected for testing and thereafter evaluate the oil coking influence in turbocharger performance. This selection process depends on the goals and limitations fixed in the specification problem. Table 3 shows the working conditions chosen and maximum temperature reached in the bearing housing system during an engine hot stop. Performance parameters of the engine and the turbocharger are monitored with the following operative conditions to study how they are influenced by oil coke formation.

*Table 3. Temperatures in the bearing system for two engine operating conditions*

| Engine Speed (rpm) | Torque (N.m) | Radial Bearing (ºC) | Drain Wall (ºC) |
|---|---|---|---|
| 2500 | 210.3 | 289 | 318 |
| 2750 | 216.6 | 324 | 367 |

## 3. Application of the method for studying oil coking influence in turbocharger performance

Current measurement techniques, even analyzers, do not offer suitable solutions to measure or quantify the level of fouling, or coke formation in the turbocharger. They are based on analyzing engine oils formulation and comparing with expected-behaviour correlations [23], which is usually the least expensive options, althoughintrusive inspection [17] is also used.

The use of this methodology intends to quantify the damage or loss in performance in a non-intrusive way with endurance tests in non-water cooled turbochargers. The endurance test in the turbocharger is performed by repeating one hundred times a hot stop engine cycle (shown in figure 2) under a given engine operative condition.

An engine-independent lubrication system exclusive for the turbocharger has been used for this purpose (L in figure 1). The system is connected with an oil pump that drives the fluid along the circuit; a valve opens and closes the system for the recirculation of oil. An electrical resistance heats the oil, and a pressure regulator controls the oil pressure flowing through the turbocharger. In order to prevent excessive deterioration or malfunction of the turbocharger, a filter in the circuit is implemented to prevent the passage of the larger pollutant particles that could be generated along the tests and appear in the oil circuit. The independent lubrication system prevents the oil coke formation to the rest of the engine.

The tests of resistance of coke require of the provisioning of oil of different levels of aging.
The carbonaceous material is a pre propellant for the formation of deposits. The more contained amount of soot in the oil, the more severe conditions for the tests. Two different multi-grade oils have been used: One SAE grade 5W30 (referred in this work as A) and another SAE 10W40 (referred in this work as C). In terms of quality specifications, A and C oils follow ACEA C4 and ACEA A3/B4-10 respectively. Both oil types are artificially contaminated at levels of 4% and 7% of impure carbon particles resulting from the incomplete combustion of hydrocarbons (soot).

Therefore, with higher levels of soot the possibility that oil coking problems will occur in a shorter test time increase  bearing in mind that turbochargers used in the endurance tests are new.

### 3.1 Measurements and post-processing of instantaneous variables from steady and transient tests

The instantaneous variables measured at constant engine speed are synchronously recorded with a sampling rate of 10 Hz during 20 minutes, so that 12000 data points are recorded every transient engine cycle. It is necessary to establish a group of control parameters, whose variation through the load transient test has to be recorded. Some of them have to be kept into a narrow range, namely engine speed and fuel temperature, in order to ensure the reliability of the acquired instantaneous signals. Moreover, ambient pressure and temperature are also monitored. Parameters such as engine speed, fuel consumption and turbine inlet temperature have been controlled to ensure repeatability in the set of test and shown in figure 6. Turbocharger conditions such as temperatures at the inlet and outlet of the compressor and turbine side, mass flow rates, VGT position and turbocharger speed have been also monitored during the experimental tests campaign.

From figure 6, on the left hand the instantaneous data recorded for an engine cycle are presented. On the right hand, the same engine variables are exposed as an average at the end of the stabilization phase (600 seconds) before engine shut down for the set of hundred cycles. There can be seen that engine variables have been held constant in order to ensure that endurance tests are performed correctly without fluctuation that may affect the main conclusions.

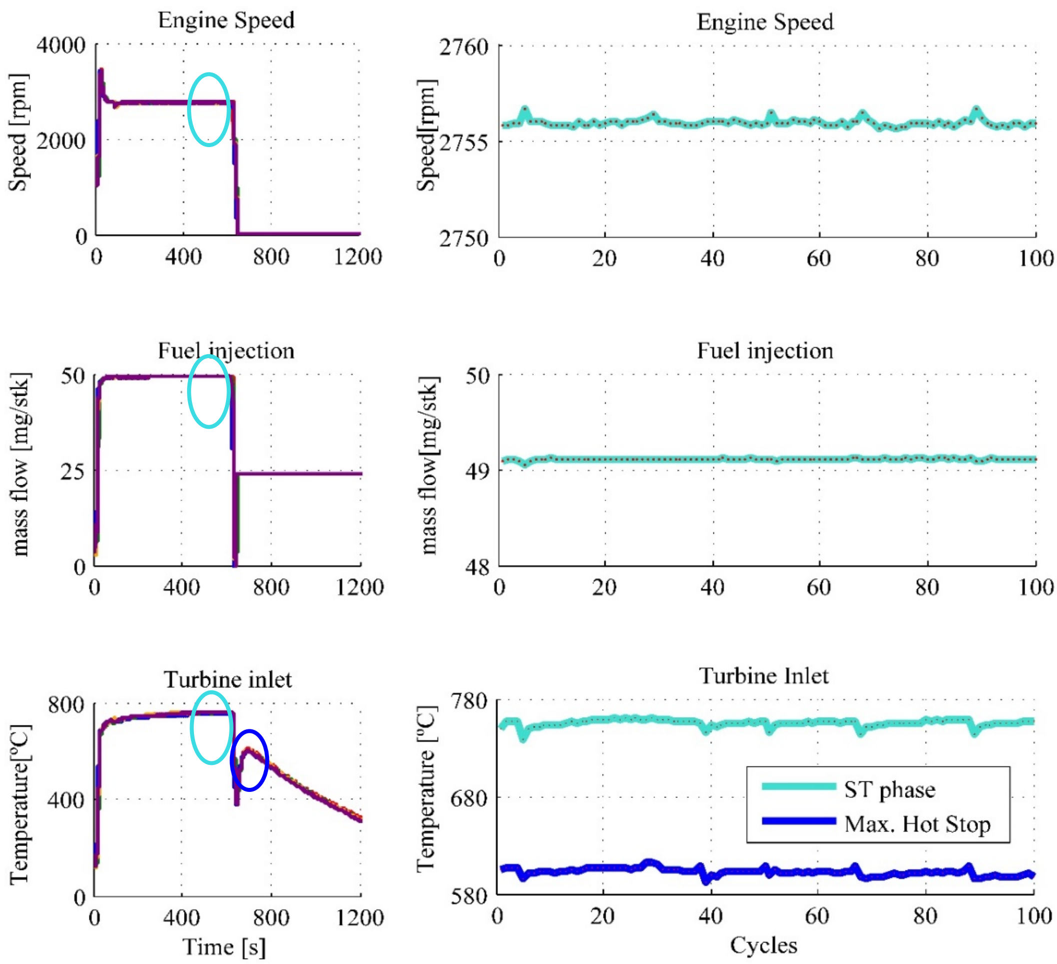

Figure 6. Engine variables control for endurance tests

Next, after testing the set of cycles, the independent lubrication system is cleaned, another type of oil is replaced and a new turbocharger unit of the same model is used.

### 3.2 Study of the dispersion of the response variables.

The general procedure to measure and post-process the fundamental magnitudes involved during the whole engine operation is described in the following paragraphs. The data is obtained and processed for eight turbocharger units.

A global analysis of the measured results is done. The engine torque, VGT position, turbine and compressor efficiencies are considered as response variables for the study of oil coking influence in turbocharger performance.

The compressor efficiency can be calculated by the ratio of the isentropic work of the compressor and the compressor work by using equation (1).

$$\eta_C = \frac{T_{iC} \cdot \left[ \left( \frac{P_{oC}}{P_{iC}} \right)^{(\gamma-1)/\gamma} - 1 \right]}{T_{oC} - T_{iC}} = \frac{\dot{w}_{Cs}}{w_C} \qquad (1)$$

From equation (1) P represents the pressures and T temperatures, the subscripts $oC$ and $iC$ means at the outlet and inlet of the compressor respectively.

Turbocharger efficiency is the product of the turbine ($\eta_T$), times the mechanical ($\eta_{mech}$) and times the compressor ($\eta_c$) efficiencies respectively. It has been calculated as shown in equation (2) by the ratio between compressor and turbine isentropic power.

$$\eta_{TC} = \eta_T \cdot \eta_{mech} \cdot \eta_C = \frac{\dot{w}_{Cs}}{\dot{w}_{Ts}} \qquad (2)$$

The isentropic power in the turbine side, can be calculated using equation 3, where $\dot{m}$ is the mass flow at the turbine inlet, $C_p$ the specific heat and the subscripts $oT$ and $iT$ means at the outlet and inlet of the turbine respectively.

$$\dot{w}_{Ts} = \dot{m}_{iT} . C_p . \ T_{iT} \cdot \left[ 1 - \left( \frac{P_{oT}}{P_{iT}} \right)^{(\gamma-1)/\gamma} \right] \qquad (3)$$

In order to make an analysis of efficiencies calculated in each turbocharger unit, a fitted polynomial has been used to filter abnormal values and getting clear trend patterns along the 100 cycles. Figure 7 a) shows a filtration example. The subplots b) and c) in Figure 7 shows that deviation between the measurements and trends of efficiencies filtered are less than 2% for the compressor efficiency and less than 4% in the turbocharger efficiency.

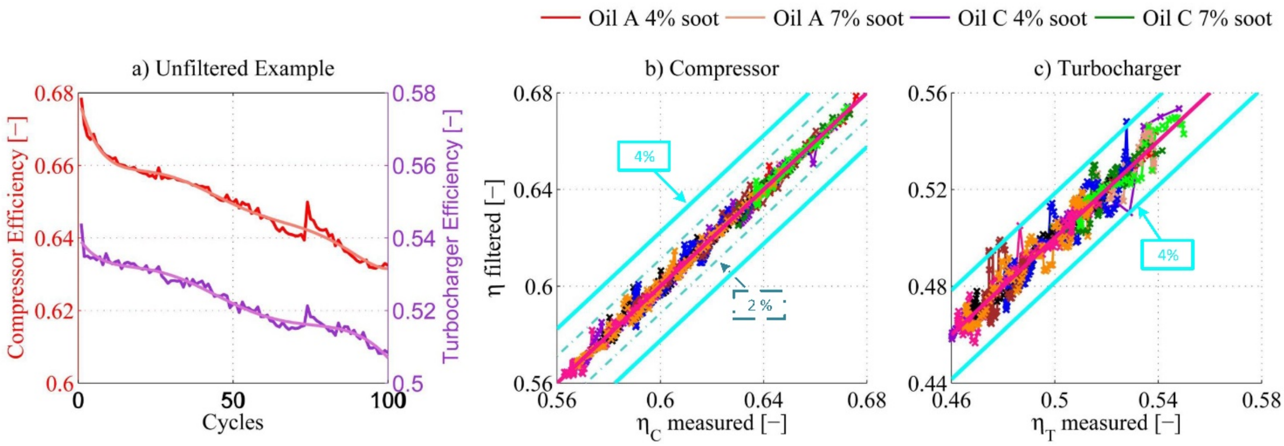


Figure 7. Deviation between measurements and results in compressor and turbocharger efficiencies

Figure 8 shows the transition along the 100 cycles and the main parameters evaluated in four turbochargers at engine speed of 2750 rpm and full load. Dots in figures represent an average of the last phase of thermal stabilization as shown in figure 6 (just before to proceed to the engine shut down). The first subplot a) indicates the engine torque; there can be seen a gradient of drop in the torque level that is similar for the four tests. Subplot b) indicates the vane position of the VGT vanes that are closing almost around 3 or 4 % for all turbochargers. Subplot c) indicates the turbocharger efficiency, which decreases probably due to oil coke trapped in the bearing system. The last subplot d) indicates the compressor efficiency that is being affected by the shaft motion producing an efficiency drop. As shown in figure 8, the VGT reacts to the torque level and turbocharger efficiency. This soot concentration and the oil coke deposits make the mechanical efficiency to deteriorate. The effect of accumulation of soot and coke could also affect the axial position of the shaft: more closed VGT rack positions and high turbine inlet pressures due to lower mechanical efficiencies can increase the compressor tip clearance as the shaft shifts axially towards the turbine, negatively impacting its efficiency. The latter, however, was not directly measured and it can only be inferred from the compressor efficiency results. In summary, if the mechanical and the compressor efficiency drops so it does the turbocharger efficiency (see equation 2). Since the control keeps the boost pressure ($P2$) constant, as the turbocharger efficiency drops, the VGT closes to increase the pressure ratio ($\pi_T$) as well as the turbine isentropic power($\dot{w}_{TS}$). As a consequence, the pressure at the turbine inlet ($P3$) increases and in this case, if $P2$ remains constant, the loss due to pumping mean effective pressure in the engine increases, therefore at constant $\dot{m}_{fuel}$ the effective torque decreases.

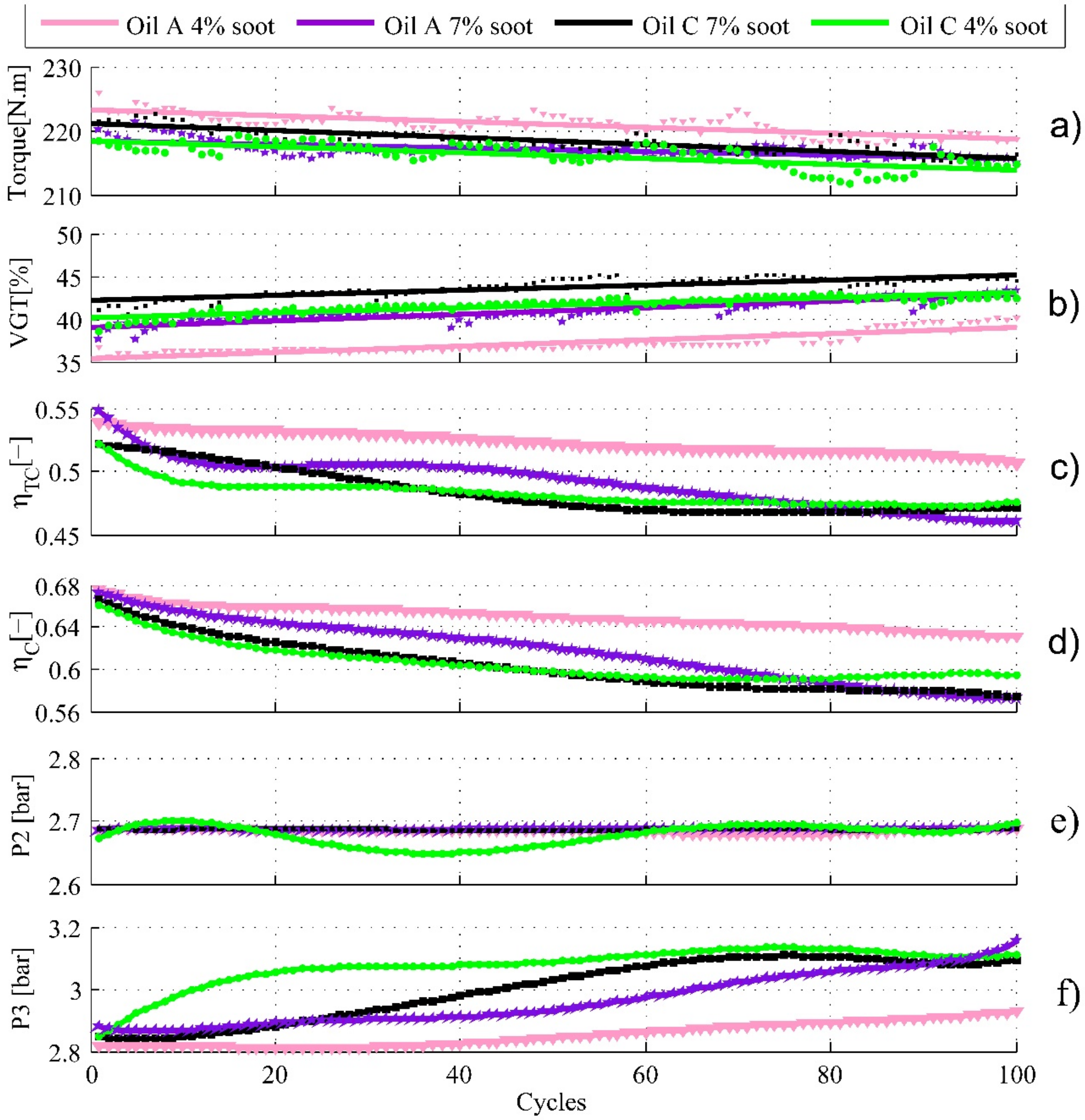


Figure 8. Response variables measured at 2750 rpm of engine speed and full load

Figure 9 shows the selected response variables for another four turbochargers measured under engine hot stop duty cycle at 2500 rpm and 75% of full load. In subplot a) the average engine torque is shown. A gradient of drop can be observed, which is grouped by the type of oil. This decrease in the engine torque level is less for this engine operative condition. Subplot b) represents the VGT vane position. The VGT vanes are closed more for the type C oil, suffering a greater torque drop. The third subplot c) shows the turbocharger efficiency. In this engine operative condition, the oil viscosity affects in a big proportion the efficiency deterioration. The oils with higher viscosity (oil type C) produce greater friction and therefore a higher drop of efficiency. The last subplot d) indicates the compressor efficiency. Higher loss of efficiency in the compressor is attributed by the authors of this manuscript to the increased compressor tip clearance, as a result of shaft motion due to coke deposits and increased turbine expansion ratio. An increase in the shaft motion and final axial position might generate imbalances in the compressor and even produce compressor blades damage, affecting its performance. As also shown in figure 9, the lower temperature in the bearing system produces less oil degradation. At this engine operative condition, there is a considerable difference between one oil quality and another.

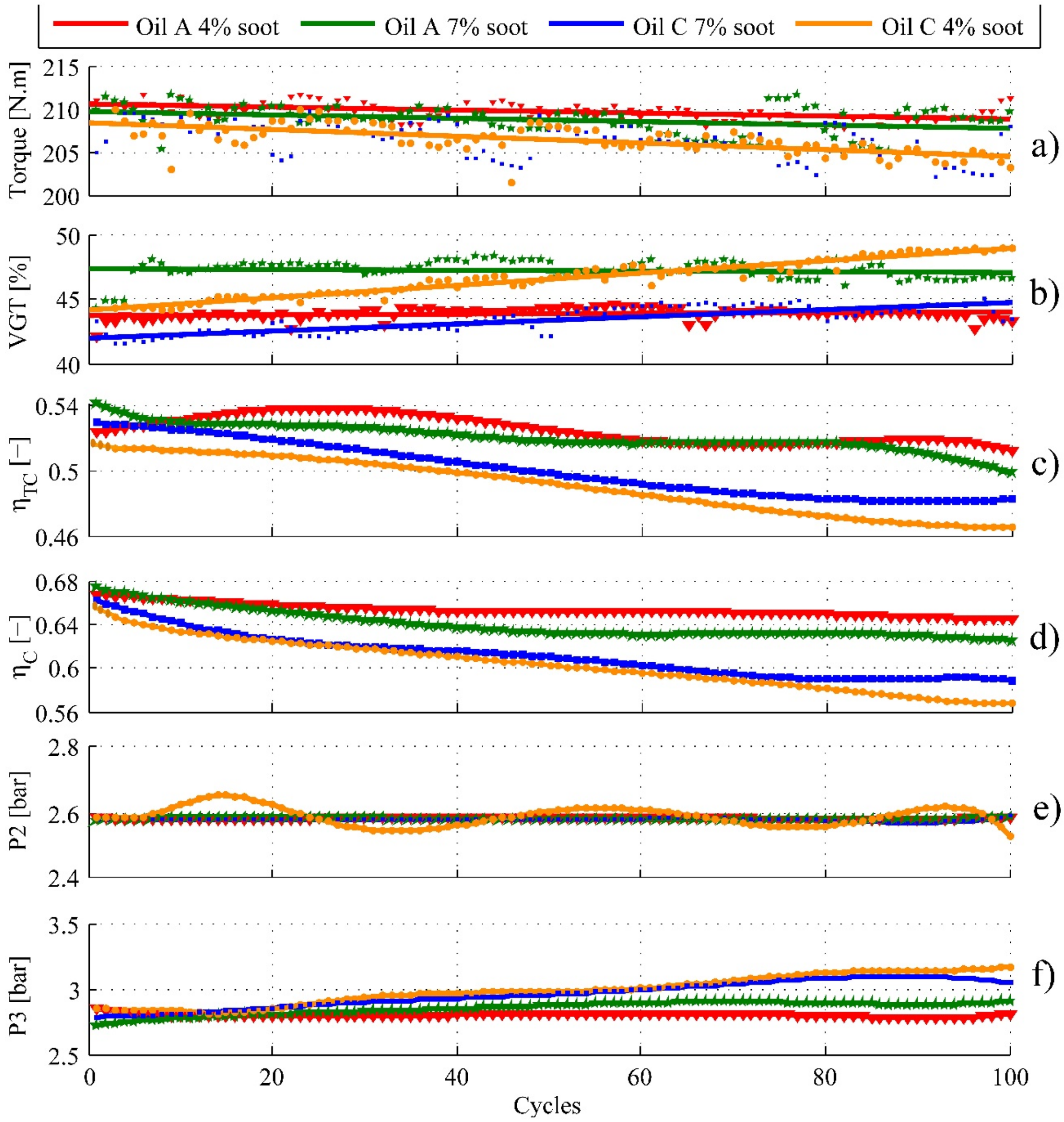


Figure 9. Response variables measured at 2500 rpm of engine speed and 75% of load.

Figure 10 shows oil coke deposits found for the turbocharger tested at high engine speed and load using oil type A at 7% of soot. The accumulation of soot and coke produces effects in the bearing rotation and, in fact, the variations observed in the results. The difference of weight before and after the endurance test was about 15.3 g.

Figure 10. Oil coke deposits in a turbocharger tested at 2750 rpm- full load and oil type A with 7% of soot.

Table 4 is a summary showing the highlighted numerical results to study oil-coking influence in turbochargers by using the method exposed through this paper. The table 4 shows the performance of the engine and the turbochargers at the end of the endurance tests in both engine operative conditions (high load and speed versus low load and speed) with the different oil qualities used. A greater and similar engine torque level drop is evidenced for the turbochargers tested at high load conditions. The loss in transient torque can be attributed to the increase in pumping mean effective pressure (PMEP), due to the increase of engine back-pressure by VGT vane position (derived through Engine Control Unit, ECU). The VGT vanes close a smaller amount for turbochargers measured at low load conditions and with less viscous oils (oil A).

The turbocharger efficiency losses are lower than the losses in the compressor side for all cases. The high soot concentration found inside the turbocharger tested at high engine speed and load using oil type A at 7% of soot has larger efficiency drops (bold in table): this may be attributed to oil coke and soot deposits that affects the shaft motion and consequently the performance of the turbocharger. The deterioration of the efficiencies in the turbocharger and compressor in which oil C with 4% of soot concentration has been used are higher than expected. Also, some of the compressor blades broke (see figure 11). The damages shown in figure 11 can be produced due to foreign-object damage, cyclic loading and crack growth or due to the proximity of the rotor tips to the compressor housing. In general, oil type A shows a better response to the endurance tests than oil C at low load and low engine speed (less temperature in the bearing system).

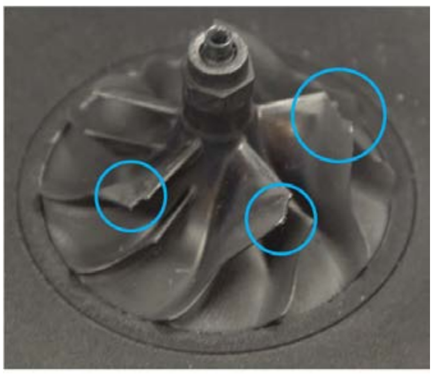

Figure 11. Compressor wheel in turbocharger tested at 2500 rpm- 75 %full load and oil type C with 4% of soot

*Table 4. Comparison of response variables after endurance test in all turbochargers tested*

| Oil type | Viscosity at 100°C (cSt) | EngineTorque drop (N.m) | | VGT (%) | | Turbocharger $\eta$ drop(Relative) | | Compressor $\eta$ drop (Relative) | |
|---|---|---|---|---|---|---|---|---|---|
| | | 2750 rpm 100%load | 2500 rpm 75%load | 2750 rpm 100%load | 2500 rpm 75%load | 2750 rpm 100%load | 2500 rpm 75%load | 2750 rpm 100%load | 2500 rpm 75%load |
| A 4%soot | 11.49 | 4.2 | 1.7 | -3.39 | -0.24 | 0.07 | 0.02 | 0.07 | 0.03 |
| A 7%soot | 13.01 | 5 | 1.9 | -3.73 | 0.30 | **0.17** | 0.06 | **0.16** | 0.07 |
| C 4%soot | 15.53 | 4.5 | 3.9 | -2.86 | -4.79 | 0.09 | **0.1** | 0.1 | **0.12** |
| C 7%soot | 16.18 | 5.1 | 3.7 | -4.11 | -3.22 | 0.1 | 0.09 | 0.14 | 0.11 |

## 4. Summary/Conclusions

A detailed method for the measurement and post-processing of steady and transient tests in a sweep of engine speed and load has been explained in this work. The method allowed calibrating the time in each phase of the engine cycle to generate the maximum temperature in the turbocharger bearing system taking into account its inertia and allowed to select the engine working conditions.

The method has been applied to study the oil coke influence in the turbocharger performance under engine hot stops. Two different oil qualities have been used, both artificially contaminated at levels of 4% and 7% of soot.

The treatment of the instantaneous measured variables from their acquisition, post-processing and synchronization show a good agreement. The global engine variables have been held constant ensuring that endurance tests were performed properly without any fluctuation that may affect in the main oil coking conclusions.

Some trends have been identified for the studied turbochargers. As coke accumulates inside the bearings, the turbocharger efficiency drops and the VGT vanes need to close to achieve the same boost pressure, making an increase in the pumping losses and thus lowering the engine torque.

The drop in the engine torque is higher and similar for the turbocharger tested at high load and high engine speed. The turbocharger efficiency decreases due to oil coke trapped in the bearing system.

The decrease in compressor efficiency can be associated to increased compressor tip clearance. As the mechanical efficiency drops, the turbine rack has to move to more closed positions to provide the power needed by the compressor, increasing the turbine inlet pressure. This can be translated into an axial movement of the shaft towards the turbine, increasing the compressor tip clearance and decreasing the compressor efficiency. A change in the compressor tip clearance have not been measured experimentally, however, more experiments are needed to confirm this hypothesis.

For the tests at low engine speed and low load, the oil quality affects and makes a noticeable difference in the performance of the engine and the turbocharger. At this operative condition, the VGT vanes position close and produce reaction in a big proportion to the torque level for turbochargers tested with oil type C. The higher viscosity of oil C produces greater friction and therefore higher drop in the turbocharger and compressor efficiencies.

In general, a good correlation has been obtained between the selected response variables. Some differences were observed in the study of the dispersion for the endurance test at high load and speed in which oil “A” at 7% soot has been used. The high temperature in the bearing system generates high concentration of coke inside this turbocharger producing a greater loss of performance in the turbocharger. The weight difference before and after one hundred cycles for this test was about 15.3 g. Differential weight analysis and other correlation factors associated with oil coke endurance tests in turbochargers will be further discussed in future studies.
Ensuring an optimum lubrication quality is essential to ensure the successful operation of the turbocharger and the engine as a whole; if the system deteriorates, a decrease in engine performance can reach up to 20% in power and torque.

## Acknowledgements

Authors want to acknowledge the ‘‘Apoyo para la investigación y Desarrollo (PAID)” grant for doctoral studies (FPI-2016-S2-1354). The work has been partially supported by FEDER and the Spanish Ministry of Economy and Competitiveness through grant number TRA2016-79185-R.

## References

[1] E. G. Giakoumis, “Review of Some Methods for Improving Transient Response in Automotive Diesel Engines through Various Turbocharging Configurations,” *Front. Mech. Eng.*, vol. 2, no. May, 2016.

[2] J. Šimek, “Turbocharger bearing support and rotordynamics,” Prague, 2015.

[3] K. Tanimoto, K. Kajihara, and K. Yanai, “Hybrid ceramic ball bearings for turbochargers,” *SAE Pap. 2000-01-1339*, no. 724, pp. 1–14, 2000.

[4] H. Nguyen-Schäfer, *Rotordynamics of automotive turbochargers*, Second edi. Springer International Publishing, 2012.

[5] D. Zeppei, S. Koch, and A. Rohi, “Ball Bearing Technology for Passenger Car Turbochargers,” *Mot. Zeitschrift - Springer*, vol. 77, no. 11/2016, pp. 26–31, 2016.

[6] M. D. Brouwer, “Whirl and Friction Characteristics of High Speed Floating Ring and Ball Bearing Turbochargers,” *J. Tribol.*, vol. 135, no. 4, p. 41102, 2013.

[7] J. a Addison and W. Needelman, “Diesel Engine Lubricant Contamination and Wear,” *Pall Corp. Engwear*, p. 12, 1986.

[8] J. R. Serrano, P. Olmeda, A. Tiseira, L. M. García-Cuevas, and A. Lefebvre, “Theoretical and experimental study of mechanical losses in automotive turbochargers,” *Energy*, vol. 55, pp. 888–898, 2013.

[9] D. Polichronis, R. Evaggelos, G. Alcibiades, G. Elias, and P. Apostolos, “Turbocharger Lubrication - Lubricant Behavior and Factors That Cause Turbocharger Failure,” *Int. J. Automot. Eng. Technol.*, vol. 2, no. 1, pp. 40–54, 2013.

[10] J. Galindo, J. M. Lujan, C. Guardiola, and G. S. Lapuente, “A method for data consistency checking in compressor and variable-geometry turbine maps,” *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.*, vol. 220, no. 10, pp. 1465–1473, 2006.

[11] J. R. Serrano, F. J. Arnau, V. Dolz, and P. Piqueras, “Methodology for characterisation and simulation of turbocharged diesel engines combustion during transient operation. Part 1: Data acquisition and post-processing,” *Appl. Therm. Eng.*, vol. 29, pp. 142–149, 2008.

[12] J. Black, P. G. Eastwood, K. Tufail, T. Winstanley, Y. Hardalupas, and A. M. K. P. Taylor, “The Effect of VGT Vane Control on Pumping Losses during Full-Load Transient Operation of a Common-Rail Diesel Engine,” *SAE Naples Sect.*, no. 2007-24–0063, 2007.

[13] M. Ghazikhani, M. Davarpanah, and S. A. M. Shaegh, “An experimental study on the effects of different opening ranges of waste-gate on the exhaust soot emission of a turbo-charged DI diesel engine,” *Energy Convers. Manag.*, vol. 49, no. 10, pp. 2563–2569, 2008.

[14] P. Sharma and P. Jayaswal, “WEAR RATE MEASUREMENT (IC ENGINE) USING LUBRICANT OIL TESTING METHOD,” *IJREAS Int. J. Res. Eng. Appl. Sci.*, vol. 2, no. 22, 2012.

[15] H. B. Jun, D. Kiritsis, M. Gambera, and P. Xirouchakis, “Predictive algorithm to determine the suitable time to change automotive engine oil,” *Comput. Ind. Eng.*, vol. 51, no. 4, pp. 671–683, 2006.

[16] F. Owrang, H. Mattsson, J. Olsson, and J. Pedersen, “Investigation of oxidation of a mineral and a synthetic engine oil,” *Thermochim. Acta*, vol. 413, no. 1–2, pp. 241–248, 2004.

[17] I. Miyata, S. Hirano, M. Tanada, and K. Fujimoto, “Mechanism of Turbocharger Coking in Gasoline Engines,” *SAE Int.*, no. JSAE 20159223, pp. 1–15, 2015.

[18] D. Deng, F. Shi, L. Begin, I. Du, and General Motors, “The Effect of Oil Debris in Turbocharger Journal Bearings on SubSynchronous NVH,” *SAE Int.*, vol. 2015-01–12, no. April, 2015.

[19] J. Serrano, P. Olmeda, F. J. Arnau, M. a. Reyes-Belmonte, and H. Tartoussi, “A study on the internal convection in small turbochargers. Proposal of heat transfer convective coefficients,” *Appl. Therm. Eng.*, vol. 89, pp. 587–599, 2015.

[20] R. D. Burke, P. Olmeda, F. J. Arnau, and M. Reyes-Belmonte, “Modelling of turbocharger heat transfer under stationary and transient engine operating conditions,” in *Institution of Mechanical Engineers - 11th International Conference on Turbochargers and Turbocharging*, 2014, pp. 103–112.

[21] J. R. Serrano, P. Olmeda, F. J. Arnau, A. Dombrovsky, and L. Smith, “Analysis and Methodology to Characterize Heat Transfer Phenomena in Automotive Turbochargers,” *J. Eng. Gas Turbines Power*, vol. 137, no. GTP–14–1352, pp. 1–11, 2015.

[22] J. R. Serrano, A. Tiseira, L. M. García-Cuevas, T. Rodriguez, and G. Mijotte, “Fast 2-D Heat Transfer Model for Computing Internal Temperatures in Automotive Turbochargers,” *SAE World Congr. WCX 17*, no. 2017-01–0513, 2017.

[23] T. W. Selby, “Turbocharger Deposits and Engine Deposits – A Duality : Correlative Bench

Test Studies of Turbocharger Deposits Shut-down,” *TAE Esslingen Conf.*, pp. 1–12, 2010.

## Definitions/Abbreviations

**ACEA** European Automobile Manufacturers' Association

**ECU** Engine Control Unit

**Max** Maximum

**Med** Medium

**PMEP** Pumping mean effective pressure

**SAE** Society of Automotive Engineers

**VGT** Variable-geometry turbocharger

**A** Monograde oil type

**C** Multigrade oil type

***E*** Diesel engine

***L*** Independent lubrication system

**N** Engine speed [rpm]

**P** Pressure [Pa]

**P2** Pressure at the compressor outlet [Pa]

**P3** Pressure at the turbine inlet [Pa]

***T*** Turbocharger

**T** Temperature [K]

$\dot{m}$ mass flow [kg/h]

$\dot{w}$ Isentropic power

$\eta$ Efficiency [-]

$\gamma$ Specific heat ratio

$\pi_T$ Pressure ratio

**Subscripts**

***c*** Compressor

**Cs** Compressor side

**i$\boldsymbol{C}$** Inlet compressor

**i$\boldsymbol{T}$** Inlet turbine

**mech** mechanical

**oC** Outlet compressor

**oT** Outlet turbine

**Ts** Turbine side

**TC** Turbocharger