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An industrial experimental methodology for the unsteady characterization of automotive turbocharger

Nicolas Cappelaere^{1,2}, Antoine Dazin¹, Gérard Bois¹



To improve the knowledge of the behavior of turbochargers under real conditions, it is needed to improve testing methodologies by making them realistic. Today, engine downsizing has lowered the size of the engine and reduced the number of engine's cylinders. The power loss has been compensated by the addition of a turbocharger. One consequence is that transient phenomena such as pulsations, are more and more important at the exhaust line, modifying the turbine environment, and behavior.

In this article, the author deals with the development of a new test methodology for turbocharger under pulsating conditions. A specific test bench has been developed to reproduce the behavior of gas at the exhaust of an engine. A transient instrumentation has been designed to evaluate the performance of the turbocharger.

Keywords

Turbocharger — Testing — Pulsating conditions

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INTRODUCTION

To reduce the fuel consumption of automotive internal combustion engines, turbocharger is nowadays a common part under the hood of a car. In order to optimize the turbocharger working points in agreement with the engine conditions, the knowledge of turbocharger's behavior in real exhaust manifold environment has to be improved [1]. Today's common turbocharger test benches work in stabilized conditions [2]. A significant improvement could be the ability to recreate a pulsating flow of an engine and to measure the turbocharger's transient parameters such as air velocity, pressure, temperature and mass-flow. Different approaches have been proposed in the literature, each one with its advantages and limitations [3] [4] [5]. In this article, the authors deal with development of a new testing methodology, based on the use of an industrial test bench adapted with a particular transient instrumentation for this specific characterization.

Following the trend of new engines architecture, and performances, it is possible to evaluate new specifications for the testing of turbocharger. Thanks to downsizing, today's engines are now mainly working with 4, 3, and even 2 cylinders, modifying the transient evolution at the inlet of the turbine. Thanks to particles emission reduction, the temperature at the outlet of engine is increasing.

These new specifications can then be summed up by:

- From 2 to 4 cylinders,
- P3 max 4bar
- Equivalent to 2000 rpm for a 4 cylinder 4 strokes
- Temperature of 700°C,

1. DEVELOPMENT OF A NEW TEST BENCH

1.1 Bench architecture

In order to answer to these challenges, a new architecture of test bench has been developed by CRITT M2A. The bench is based on a known KRATZER turbocharger gas stand. A gas burner supply hot air to the turbine reproducing the exhaust of an internal combustion engine. Variable pressure valves are used at the outlet of the compressor to reproduce the counter pressure generated by the engine admission. The bench is able to generate an air flow with the following characteristics:

- Pressure : 6 bar,
- Temperature : 1100°C,
- Mass flow: 0.5 Kg/s
- Power : 250 kW

The turbine and the compressor line is instrumented by steady sensors such as:

- Mass flow: inlet of the compressor and of the turbine,
- Pressure: inlet and outlet of the compressor and of the turbine
- Temperature: inlet and outlet of the compressor and of the turbine,

- Hygrometry: inlet of the compressor.

The turbocharger is also instrumented with a rotational speed sensor, and its oil circuit is instrumented with mass flow, pressure and temperature sensor. The global architecture of the bench is given in Figure 1

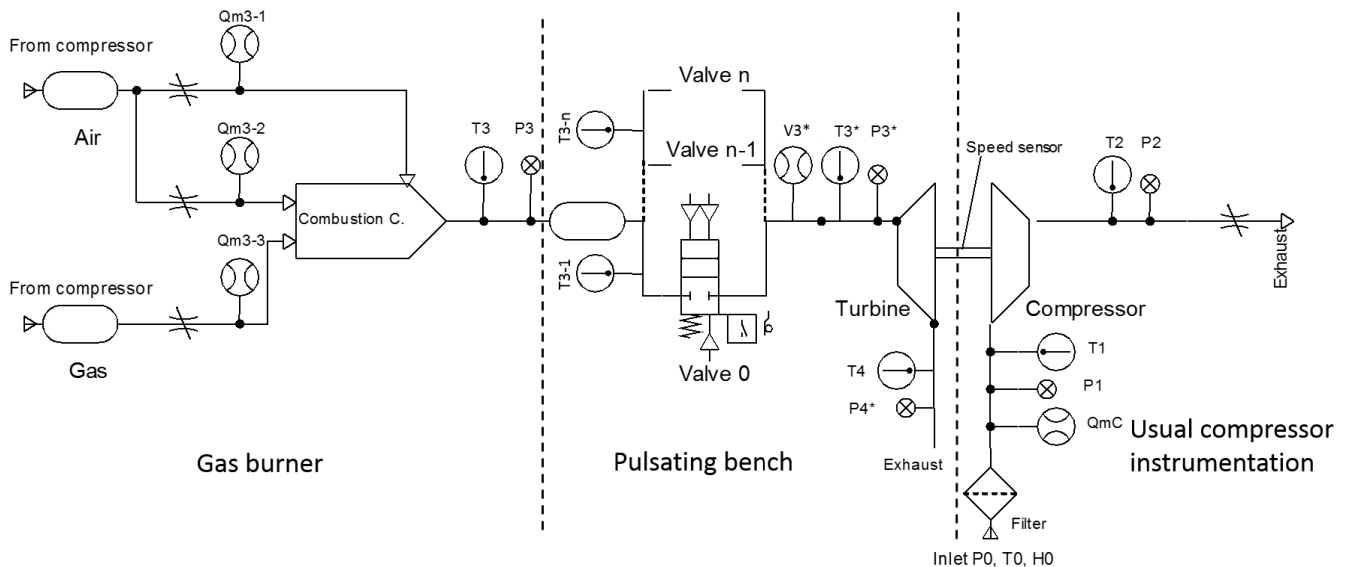


Figure 1. Bench architecture

1.2 Pulsating bench principle

In adjunction of this bench, a particular 'plug and play' system has been added to reproduce the pulsating behavior of the flow at the inlet of the turbine. This "flow pulsator" is composed of four valves whose architecture is quite similar to engine valves. These valves are placed between the gas burner and the turbocharger.

In order to reproduce the architecture of an engine the valves have been placed at a short distance of the turbocharger inlet exhaust manifold. The alternative opening and closing of these valves are synchronized and clocked as the cam law of an engine. The valves used for this application are TIAL waste gates (Figure 2). They are generally used on the exhaust manifold engines to by-pass the turbine, and so limit the pressure at its intake. They are built to stand to high temperature and pressure, therefore they fit for this application. The valve translation is controlled by two pneumatic chambers. A spring is also used to compensate the force applied by the air on the valve.

The control air is steered by electronic distributors at a pressure up to 7 bar (Figure 3). Thanks to an

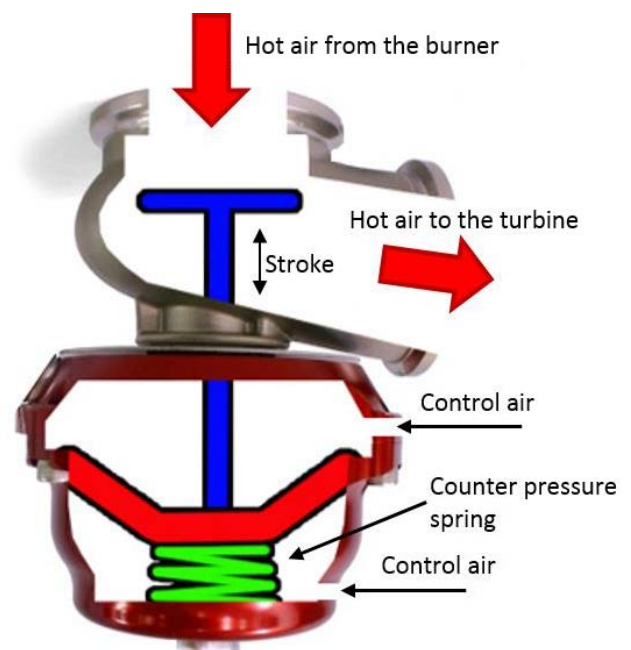


Figure 2. Waste gate cut

electronic control of the distributor and so of the valve, it is very simple to change the number of "active" valve(s). This makes it simple and fast to adapt the flow pulsator bench to the number of cylinders of the simulated engine.

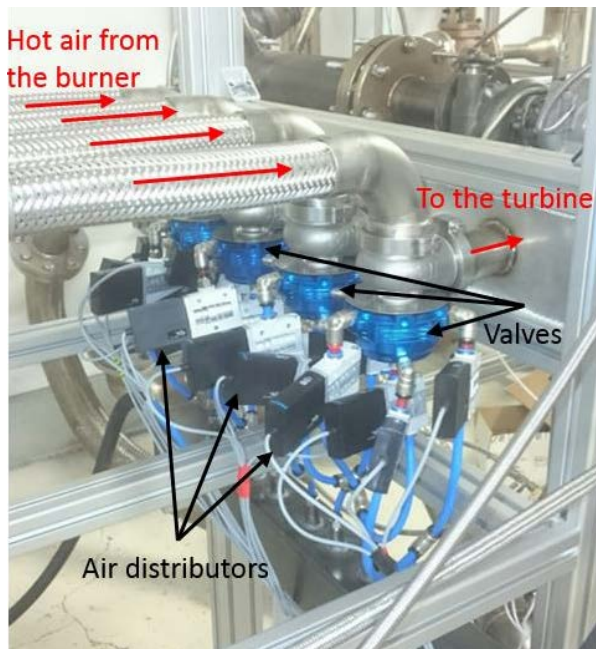


Figure 3. Valves assembly on the bench

1.3 Pulsating bench dynamic validation

The bench has the ability to reproduce the engine conditions up to a speed of 2000 rpm, equivalent to 15ms for the opening and closing of the valve(s). In order to match specifications, the dynamic response of the valve has been experimentally tested. The waste gate was instrumented with two accelerometers, the first on the moving valve, the other on the fixed bodypart measuring the impact when the valve hit the body at the end of its stroke. The waste gate was tested with an air control pressure up to 7 bar, and without air flowing through it. The stroke of the valve was restricted to around 9mm, similar to an engine valve. For an air control pressure of 5 bar, the average measured time for the opening and the closing of the valve is around 13ms fitting with the specifications.

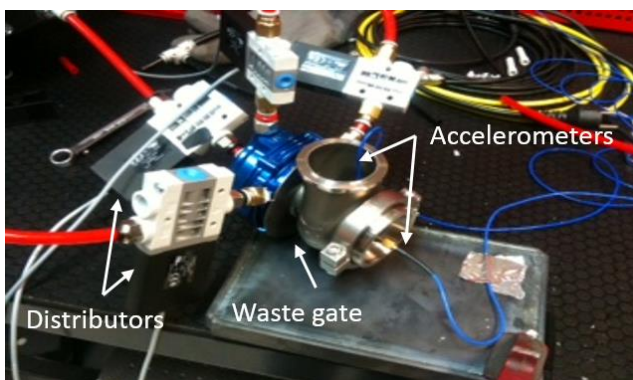


Figure 4. Dynamic valve testing

1.4 The pulsation isolation

Two plenums have been placed between gas burner and the valves in order to limit the pressure oscillations in the gas burner. Downstream the valves, an adaptation part has been realized to connect the valve with the exhaust manifold of the turbocharger. This part represents the cylinder head of the engine. During this test, the test sample is a turbocharger that is used on a small gasoline engine (4 cylinders, 4 strokes). One of the particularity of this turbocharger is that the turbine housing and the exhaust manifold of the engine are a single part.

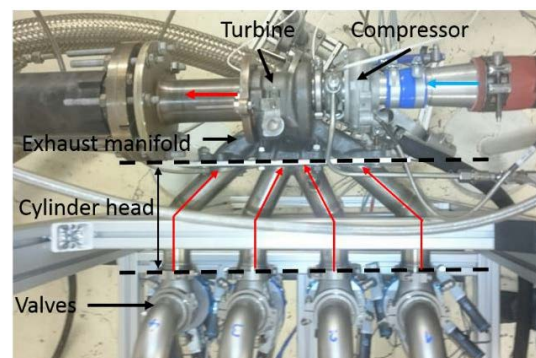


Figure 5. Interface bench/turbocharger

1.5 The bench control

The software of the bench has also been developed by CRITT M2A, it is running on a National Instrument PXI acquisition system. The program controls the pneumatic distributor(s), so the valve(s) opening and closing, and in the same time manage the acquisition of all the fast pressure sensors. All inputs and outputs of the bench are synchronized on the same high frequency clock. All the software is based on crank angle definition, and the pulsation generation is equivalent on the engine rotation speed and number of cylinder(s). So the data defined on an engine complete cycle is equal to: 720° crank angle for a four strokes engine, 360° crank angle for a 2 strokes engine. The usual crank resolution is 0,2° crank angle. The equivalence in term of frequency is given in the table:

Table 1. Speed to frequency equivalence.

Rotation speed	Frequency
rpm	Hz
1000	30007,5
1250	37509,3
1500	45011,2
1750	43760,9
2000	42867,8
Resolution of 0,2° on 720° crank	

2. THE SPECIFIC INSTRUMENTATION

Turbine's performances under steady conditions are characterized by an expansion ratio (total to static), a mass flow, and a rotation speed. So, under pulsating conditions, a transient measurement of the expansion ratio, and of mass flow are needed. High cut off frequency pressure sensor are nowadays usual in engine testing, they allow high speed pressure measurement, even at high temperature. On the other hand, few technologies are able to measure transient total pressure and transient mass flow at high temperature. Then a probe has been developed to perform these measurement.

2.1 Specifications

In order to evaluate the transient air velocity at the inlet of the turbine, a Pitot probe has been specially designed according to acoustic and thermal considerations. In fact, a Pitot probe can be considered as an acoustic quarter wave resonator if its section is constant. As consequences, the pipe of the probe is subject to acoustics resonances that could disturb the measure of the total pressure. A part of this article is dedicated to the conception of the geometry of the probe and to its acoustic sizing.

First of all, static pressure measures have been performed in the exhaust manifold of a 4 cylinders 4 strokes turbocharged engine, and a frequency analysis has been done. The goal of these measurements is to evaluate the generated frequencies that could be found at the entry of the turbine, and design the pitot probe so its resonance frequency won't match the generated frequencies. The engine use for this pressure experimentation is a Renault D4FT: 1.2l 16v turbocharged engine with a 4 to 1 exhaust manifold. Two fast pressure sensors were placed on the exhaust manifold, one in a branch, the other at the branch connection node.

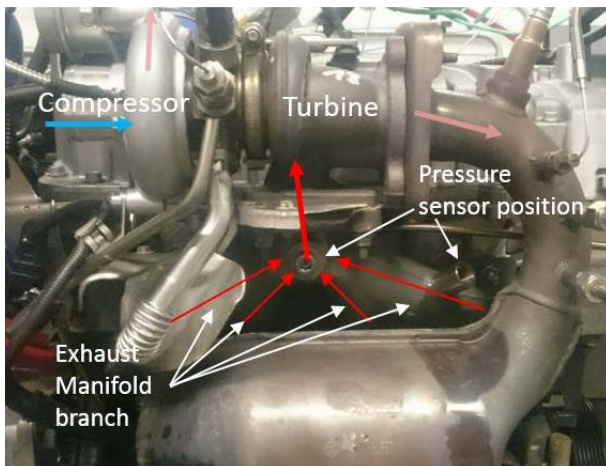


Figure 6. Pressure measurement points

Statics pressures have been measured at different engine operating points, from 1000 to 3000 rpm, and from 50% to 100% load. The results obtained with both sensors are quite similar. The FFT (Fast Fourier Transform) analysis has shown that the frequency spectrum is limited to 1000Hz for both sensor.

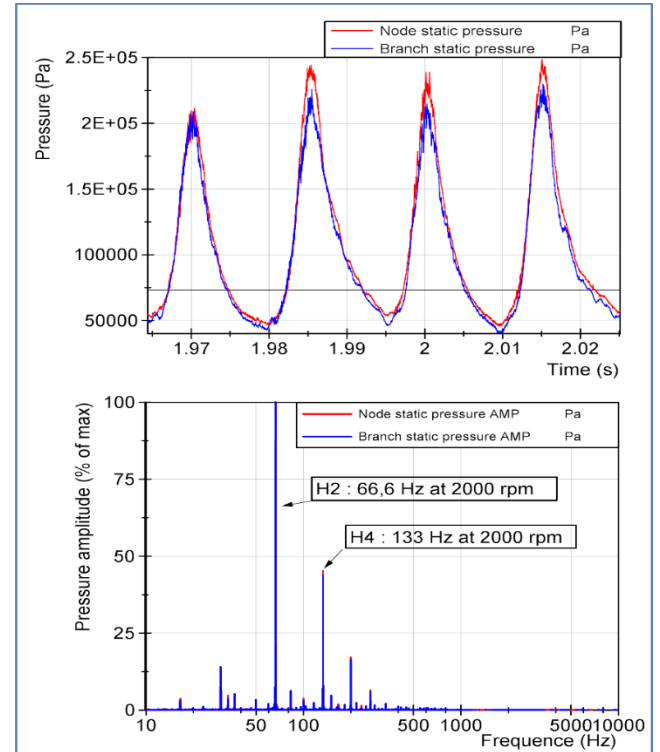


Figure 7. Frequency analysis of exhaust gas pressure

Only a very small difference in term of amplitude can be noticed between the two pressure measurements.

2.2 Theoretical considerations

The first model use for sizing the probe, is the model of an acoustic quarter wave resonator. Its resonance frequencies is given by the following formula:

$$f_r n = \frac{(1 + 2 \cdot n) \cdot c}{4 \cdot L} \quad (1)$$

$$\text{with } f_{r0} = \frac{c}{4 \cdot L} \quad (2)$$

$$\text{and } c = \sqrt{\gamma \cdot R \cdot T} \quad (3)$$

Application at 600°C for $f_{r0} = 3000\text{Hz}$, $L = 49.5\text{mm}$

During this study, acoustic results will be used in hot and cold conditions, therefore they have been corrected with the following formula obtained from (1) and (3):

$$f_r n / \sqrt{T_{\text{measured}}} = \frac{(1 + 2 \cdot n) \cdot \sqrt{\gamma \cdot R}}{4 \cdot L} \quad (4)$$

So they can be evaluated in other temperature conditions:

$$f_r n_{T_{ref}} = f_r n \cdot \frac{\sqrt{T_{ref}}}{\sqrt{T_{measured}}} \quad (5)$$

And:

$$f_r n_{T_{cor}} = f_r n \cdot \frac{1}{\sqrt{T_{measured}}} \quad (6)$$

2.3 Specifications

As the diameter of the pressure sensor used at the extremity of the Pitot probe is significant compared to the diameter of the exhaust manifold, the section of the probe has to be decreasing from the sensor to the extremity. In consequence, the model of quarter wave resonator is no longer perfectly valid, and so 3D simulations have been realized using the software VaOne, specialized in vibro-acoustic simulations.

In this model, the pitot probe inlet is placed at the center of a pipe, where its extremity are oscillating in a piston mode and so, generate an acoustic pressure wave. Acoustic pressure is measured at the inlet of the probe, and at its extremity where the pressure sensor is fixed. Then the transfer function of the probe has been measured by calculating the Gain and the Phase between the two sensors from 10 Hz to 5 kHz by step of 10 Hz. The diameter of the measuring pipe has been chosen so that the flat wave criteria is validated on all the frequency range.

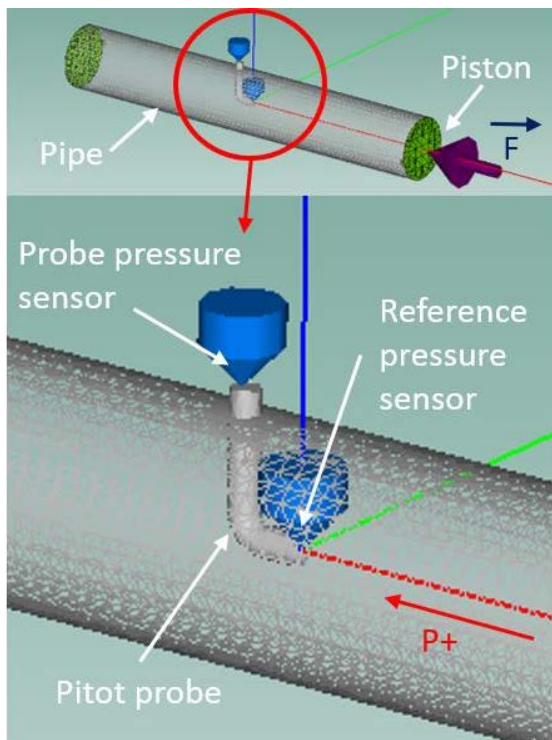


Figure 8. Pitot acoustic simulations

A sensitivity analysis has been performed, to validate the results of the simulation (Table 2).

Tableau 2. Sensitivity analysis

Variable factor		Reference		Level						Impact
Meshing	Pipe base	mm	10	5						No impact on the results(*)
	Pipe min	mm	5	2,5						
	Pitot	mm	1	0,5						
Temperature		°C	20	100	200	400	600	800	(**)	
Pressure level		Bar	1	2						No impact on the results (***)
Piston excitation force		N	1	10						
Absorption		%	0,1	1						Impact (****)

(*) No larger meshing was tested because the geometry of the probe was modified by the meshing.

(**) Using formula (6), all the results were similar.

(***) The software uses the hypothesis of linear acoustic so the results were predictable.

(****) Cannot be controlled.

Different probe geometries have been studied in order to find the geometry with the highest resonance frequency, and the most suitable geometry has been machined in stainless steel for thermal resistance considerations.

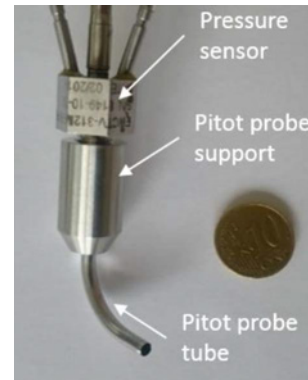


Figure 9. Pitot probe

2.4 Acoustic testing

Then, the simulation results have been correlated by experimental measurements by reproducing the simulation environment. The experiment setup is composed of a loudspeaker generating a white noise at the extremity of a pipe. The other extremity of the pipe was opened. All the setup was placed in a semi-anechoic room for the experimentation. The pressure are measured by acoustic microphone at the same positions as in the simulation. Different pitot probe positions on the pipe have been tested. This experiment has also been validated by testing 3 reference quarter wave resonators with the same instrumentation.

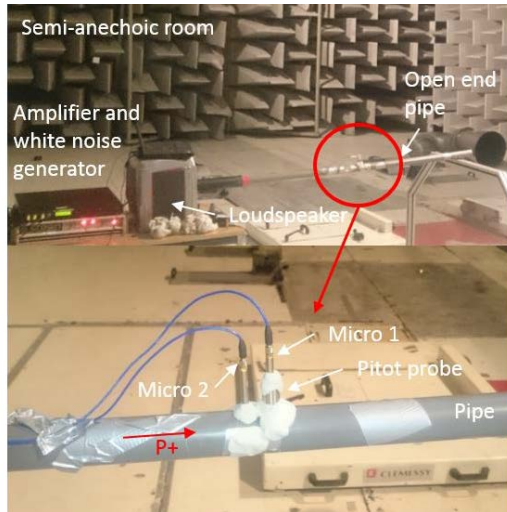


Figure 10. Pitot probe acoustic test

The results obtained by simulations are well correlated by the experimental results, and the acoustic transfer function fits well with pressure frequency that has to be measured, particularly at high temperature.

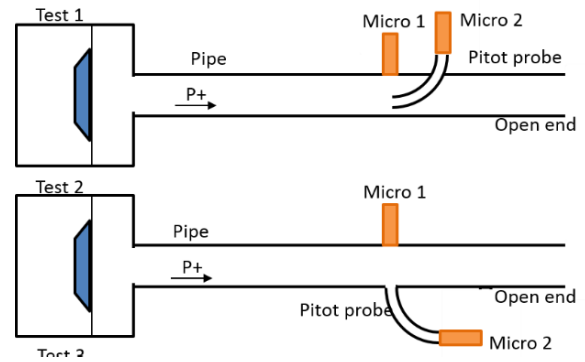


Figure 11. Pitot probe test position

As the acoustic transfer function is known, a low pass frequency filter has then been chosen to cut the frequencies above 1.1 kHz. The other particularity of this filter is that it has a constant phase equal to 0. Other ways, the temporal signal could have been delayed, resulting in a non-synchronization with the other measurements, particularly for the evaluation of the instantaneous velocity.

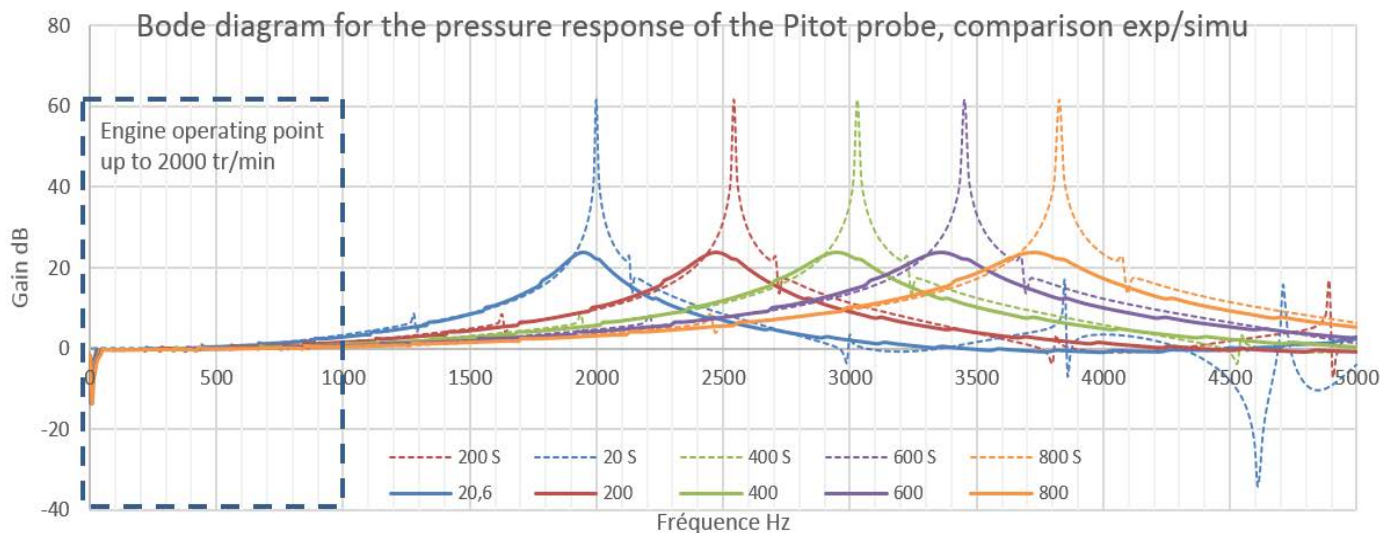


Figure 12. Pitot probe acoustic transfer function

2.5 Velocity measurement

At the inlet of the Pitot probe, the static pressure is measured by another fast pressure probe. The use of both pressure sensors (static and total) form a full Pitot probe. Moreover, due to the low Mach number ($Ma < 0.3$), the hypothesis of non-compressible gas is verified. Then, it is possible to evaluate the local air velocity from the Bernoulli formula:

$$P_{tot} = P_{stat} + \frac{1}{2} \cdot \rho \cdot v^2 \quad (7)$$

And so:

$$v = \sqrt{\frac{2}{\rho} \cdot (P_{tot} - P_{stat})} \quad (8)$$

With:

$$\rho = \frac{P_{stat} \cdot M}{R \cdot T}$$

3. TURBOCHARGER TESTING

The goal of this testing is to evaluate the performances of the bench and of the developed instrumentation. The testing was performed on a small turbocharger used on a 1.2L 16V 4 cylinders engine (not the same as in the first part of the article).

The turbocharger working points were defined on engine performances.

3.1 Instrumentation

In order to measure the behavior of the turbocharger in transient phenomena, KULITE transient pressure sensors have been used for both static and total measurements. The Pitot probe has been designed so that the pressure sensor can fit in the place of the acoustic sensor.

The full Pitot probe has been installed on the turbocharger to measure the pressure and the velocity at the inlet of the turbine (Turb.) volute. A K thermocouple (Tck) has also been placed nearby the Pitot probe. This exact same instrumentation has also been installed in the branch 2 of the cylinder head (Cyl. H.). The turbine outlet pressure is measured by another fast pressure sensor positioned 1mm downstream the turbine wheel.

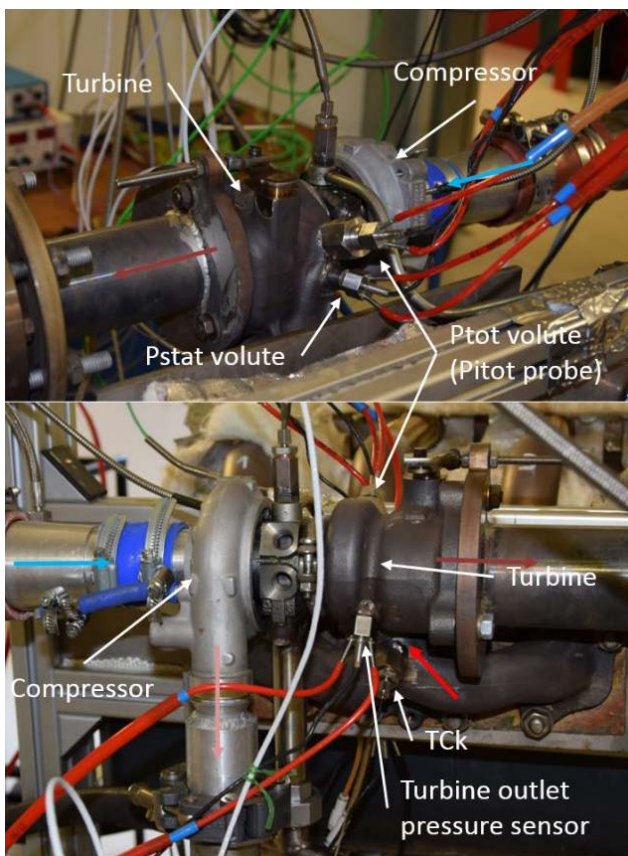


Figure 13. Specific instrumentation on the turbocharger

3.2 Test description

All tested points were measured at a temperature of approximately 500°C inside the turbine volute

A summary of the turbocharger tested point is given in the table 3:

Table 3. Tested point

	Unit	Point 1	Point2	Point 3
Engine rotation speed	Rpm	1250	1500	2000
Number of cylinders	-	4	3 and 4	4
Compressor	Mass flow	g/s	20	34
	Compression ratio tt	-	1,26	1,58 - 1,6
	Inlet pressure	hPa	1009	1008
	Inlet temperature	°C	24	21,2
Turbine	Mass flow (average)	Kg/s	0,021	0,035
			0,0436	
Turbo average rotation speed	rpm	97000	138000	171600

All the results have been averaged on 100 engine cycles, and are given in crank angle on 720°Crank.

3.3 Test results

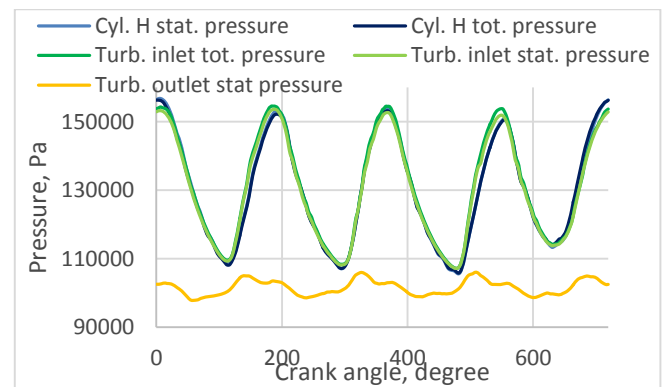


Figure 14. Pressure measurement around the turbocharger

The figure 14 represents the pressure measured at different points on the turbine. We can observe that the static pressure fluctuations are quite similar inside the cylinder head and inside the volute. Due to the absence of back pressure at the outlet of the turbine, its pressure outlet is fluctuating close to the ambient pressure. From these measures, the air velocity can be evaluated by formula (8). On the figure 15, the air velocity in the turbine volute is represented.

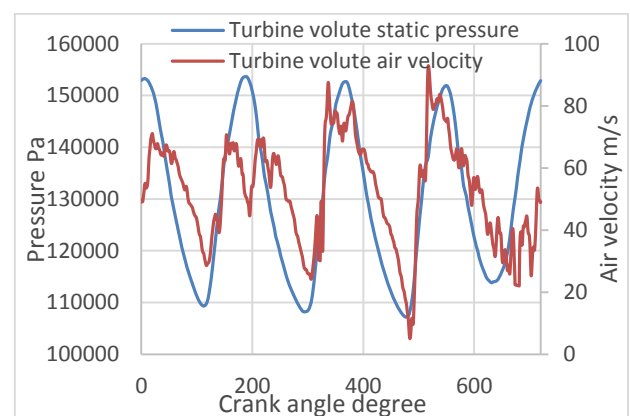


Figure 15. Air velocity measurement in the turbine volute

In order to verify the validity of the velocity measurement, the average mass flow through the volute has been calculated by the formula:

$$Q_m = \int_0^{720} \rho \cdot v \cdot S \cdot d\theta$$

The hypothesis of rectangular velocity profile through the volute section has been chosen. This verification has been realized on all the studied point and is summed up in the table 4:

Table 4. Comparison of mass flow measurement

	Point 1	Point 2		Point 3
mass flow Kg/s	4 cyl.	3 cyl.	4 cyl.	4 cyl.
Average	0,022	0,0346	0,031	0,0414
Burner	0,021	0,035	0,034	0,0436

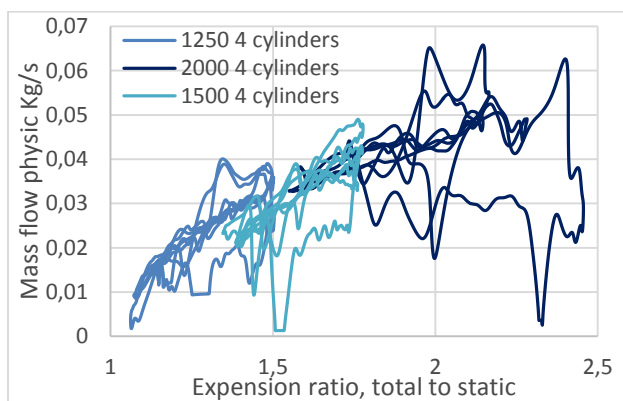


Figure 16. Turbine performances at different engine working points

The comparison of the mass flow measures shows a quite good agreement between the measurement techniques. The instantaneous mass flow measurement have been then used to evaluate the operating point of the turbine under different flow conditions.

The figure 16 represents the turbine operating points for 3 different engine rotation speed (partial load) for the same number on active cylinder. The results shows out that compared to steady results, the turbine operating point is far from been constant. The necessity to perform tests under unsteady conditions is then validated.

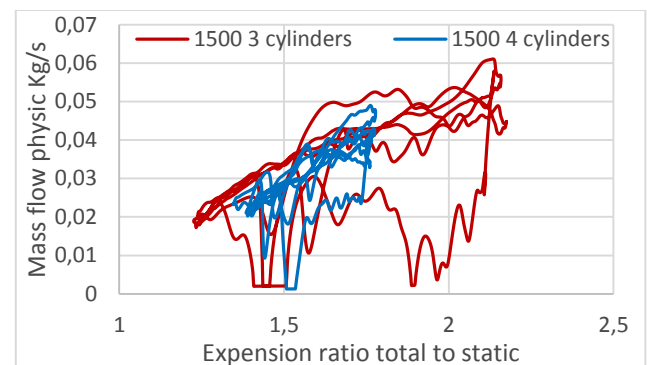


Figure 17. Impact of the number cylinders on the turbine performance for the same average point

On the figure, the impact of the number of cylinders is pointed out for almost the same turbocharger point. Even if the mass flow measurement shows a variation of 3 g/s, it is clear that reduction of the number of cylinders is widening the turbine operating point during one engine cycle.

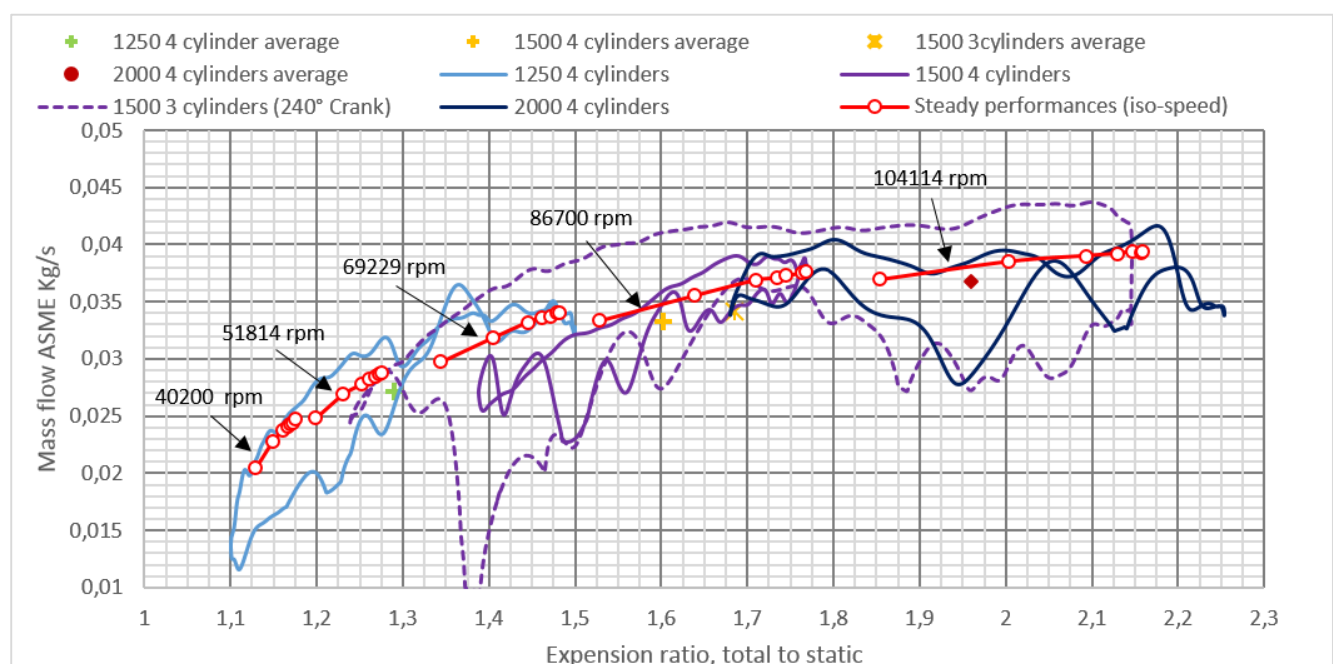


Figure 18: Comparison of turbine performances (steady/unsteady)

In figure 18, the turbine transient performances are compared to its performance measured in steady conditions. All the data have been corrected regarding ASME conditions (temperature: 288.15K, pressure 101325 Pa). We can observe that the turbine transient operating point follows the trend of the steady iso-speed.

CONCLUSION

During this study, the authors presented a new methodology to test a turbocharger under pulsating conditions. A test bench has been developed in order to better reproduce the engine exhaust conditions, and so to improve the approach of a real test condition. It was also been created to be flexible and representative of 2, 3, and 4 cylinders engines. A specific instrumentation has also been designed and tested to measure the turbocharger behavior in real conditions. It is composed of a full transient Pitot probe.

Then, this methodology has been tested on a small gasoline engine's turbocharger, to validate the proper functioning of the bench. The results demonstrate the real need of such a bench for the testing of turbocharger under pulsating conditions.

Table of symbol:

Symbol	Physical variable
Q_m	Mass flow
ρ	Gas density
v	Gas velocity
S	Pipe section
θ	Crank angle
P_{stat}	Static pressure
P_{tot}	Total pressure
M	Gas Molar mass
R	Ideal gas constant
T	Temperature
$f_r n$	Resonance frequency harmonic n
γ	Heat capacity ratio
c	Speed of sound
L	Distance

ASME CORRECTION FORMULA:

$$Q_{m\ ASME} = Q_{m\ Phy} \cdot \frac{P_{ref\ ASME}}{P_{tot}} \cdot \sqrt{\frac{T_{tot}}{T_{ref\ ASME}}}$$

With:

$$P_{ref\ ASME} = 101325\ Pa$$

$$T_{ref\ ASME} = 288.15\ K$$

ACKNOWLEDGEMENT:

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