

## **Effect of supercharging pressure on internal combustion engine performances and pollutants emissions**

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### **ABSTRACT**

The first advantage of turbocharging is the increase of specific power. Automotive engine requires a high torque at low engine speed in order to improve acceleration of the vehicle that means to have a high inlet pressure.

Nevertheless, this high pressure is difficult to get with conventional or variable geometry turbocharger, the effect of supercharging for low engine speed has been studied, in order to estimate the advantages or disadvantages obtained in terms of performances and pollution. The experimental study is led on a 2.1 liter turbocharged indirect injection engine. Performances and following gas emissions CO, HC, NO<sub>x</sub>, CO<sub>2</sub>, O<sub>2</sub>, smoke have been measured. The most significant results are presented in this paper. Results can lead to an evaluation of the benefits expected.

### **NOMENCLATURE**

AFR	Air fuel ratio	$Q_{L\text{VH}}$	Lower heating value of fuel
Bmep	Brake mean effective pressure	Rpm	Rotation per minute
Bsfc	Brake specific fuel consumption	SN	Smoke number Bosch unit
FID	Flame Ionisation Detector	Tem	Exhaust manifold temperature °C
HC	Unburned Hydrocarbon	$\phi$	Equivalence ratio: $\text{AFR}_{\text{stochio}}/\text{AFR}$
MPA	Magnetopneumatic Analyser	$\eta_g$	Gross efficiency
N	Engine speed in rpm	$\eta_v$	Volumetric efficiency
NDIR	Non-dispersive Infrared Analyser	$\rho_{\text{ic}}$	Density of air at engine inlet
NO <sub>x</sub>	Nitrogen oxydes		
P <sub>ic</sub>	Relative pressure at engine inlet		

## INTRODUCTION

It is usually considered that turbocharging leads to fuel consumption reduction and diminution of pollutant emissions. This argument is not applicable to the whole area of engine functioning.

For the automotive turbocharged Diesel engine, it is required a high torque at low engine rotation speeds. Hence, the turbine wheel is oversized in order to give a noticeable power to the compressor in this range. So, this arrangement leads to extreme exhaust counter-pressure at high loads and high speeds, whatever the means used for the control of the boost pressure (waste-gate (Anada et al 1997 [1]), variable scroll (Toussaint et al 1999 [2]), adjustable guide-vanes (Descombes et al 1999 [3])). In spite of this oversizing, the available power is at very low speed insufficient to give a substantial boost pressure.

Even if the means to obtain this substantial boost pressure are difficult to achieve, it is of a major interest to estimate its effect on engine performances to appreciate the potential benefits.

The study of the effect of boost pressure can't be done on a usual turbocharged engine due to the link between the compressor and the turbine and more generally between the compressor, the engine and the turbine. So, to overcome this difficulty, the engine is fed with compressed air at different pressure levels.

This experimentation has been done on low engine speed. Most significant parameters in terms of performances and pollutant emissions have been registered.

## 1. EXPERIMENTAL SETUP

Testes are performed on a standard test bench, fitted with a flywheel in order to simulate vehicle acceleration (Podevin et al 1999 [4]). The general layout of the test rig is presented in Figure 1.

In order to get an adjustable boost pressure, the following alterations have been done to the engine:

- only the standard turbine scroll has been fitted to the exhaust manifold. The remaining hole has been obstructed.

- an additional turbocharger has been set close to the engine. The turbine is supplied with dry compressed air controlled by a pneumatic air regulator. This design has been retained as we have at our disposal a large amount of compressed air (700 m<sup>3</sup> at 25 bar).

The benefit of such arrangement is that the original setting and functioning of the engine is nearly kept. Furthermore, the use of this additional turbocharger allows to set the boost pressure in a very accurate and easy way.

The main characteristics of the engine are:

- Turbocharged Diesel
- 4 cylinders - 12 valves
- Displacement: 2088 cm<sup>3</sup>
- Compression ratio: 21.5: 1
- Power: 81 kW at 4300 rpm
- Peak torque: 248 N.m at 2000 rpm

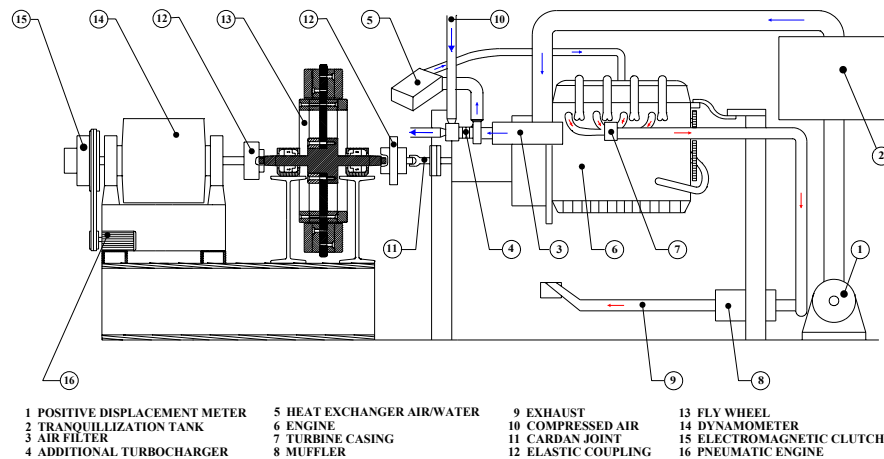


Fig. 1 - Sketch of the test bench

The engine was fully instrumented for temperature and pressure measurements, including in pre-chamber pressure. Gas emissions capability included:

- CO<sub>2</sub>, CO, NO<sub>x</sub> (NDIR analyzer)
- HC (FID analyzer)
- O<sub>2</sub> (MPA analyzer)
- smoke measurements:
  - Smoke Number (AVL 409 Smoke Meter)
  - In line Opacity Meter (Celesco 107)

## 2. TEST CONDITIONS

The main interest of this study concerns the potential benefit we can obtain with an increase of boost pressure at low engine speeds. So, this research has been done for speeds 1500, 2000, 2500 rpm with different pressure rates:

Engine speed (rpm)	Inlet engine pressure (mbar)
1500	0, 200, 400, 600
2000	0, 300, 600, 900
2500	0, 250, 500, 750, 1000

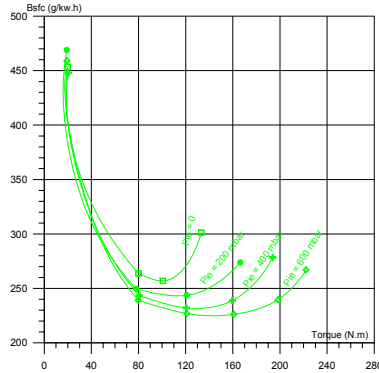
For these experiments, the exhaust gas recirculation device EGR was disconnected. No adjustments have been done to the injection system.

## 3. RESULTS

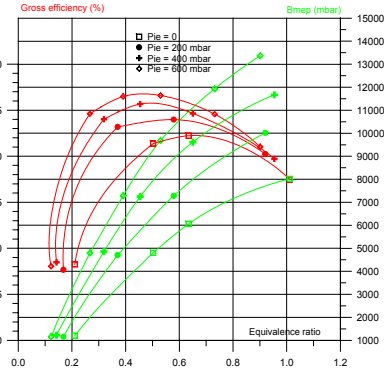
### 3.1 Influence of boost pressure on fuel consumption

In Figure 2, is represented the brake specific fuel consumption B<sub>sf</sub>c versus torque at 1500 rpm for different inlet pressure. When the inlet pressure rise, the benefits in torque and

Bsfc can be clearly stated. Increase of torque seems logical because more energy (more fuel) can be put when the boost pressure rises. Improvement in Bsfc can be demonstrated. For this purpose, it seems useful to express the engine performances as a function of brake mean effective pressure Bmep, gross efficiency  $\eta_g$  and the equivalence ratio  $\phi$  (Figure 3).



**Fig. 2** – Bsfc versus torque for different relative engine inlet pressure at  $N = 1500$  rpm



**Fig. 3** – Bmep and  $\eta_g$  versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm

The brake mean effective pressure can be expressed by the following equation (Magnet 1998 [5]):

$$\text{Eq. } Bmep = \frac{\eta_v \rho_{ie} \eta_g Q_{LVH}}{AFR} \quad (1)$$

For example, we will compare the brake mean effective pressure at equivalence ratio 0.8 for point A with relative boost pressure 0 and point B with 600 mbar.

$$\text{Eq. } \frac{Bmep_B}{Bmep_A} = \frac{\rho_{ieB} \eta_{vB} \eta_{gB}}{\rho_{ieA} \eta_{vA} \eta_{gA}} \quad (2)$$

Bsfc and  $\eta_g$  can be read in Figure 3, extra experimental value necessary for this calculation are given below:

Point	Bmep bar	$\rho_{ie}$	$\eta_v$	$\eta_g$
A	7.1	1.192	0.85	0.31
B	12.5	1.855	0.9	0.33

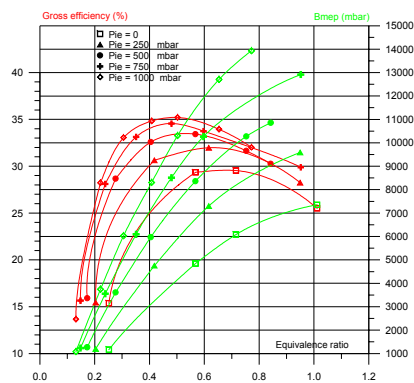
So, we have:  $\frac{Bmep_B}{Bmep_A} = 1.556 \cdot 1.059 \cdot 1.065 = 1.754$  which is in accordance with

the ratio  $\frac{Bmep_B}{Bmep_A} = \frac{12.5}{7.1} = 1.76$

This result shows that increase of brake mean effective pressure is not only due to the rise of air density but also to the improvement of volumetric efficiency and gross efficiency. The rise of these two last terms is also the reason of lower brake specific fuel consumption. This result is logical:

- for high inlet pressure there is a better filling of the cylinder, so a better volumetric efficiency,
- increase of gross efficiency is due to the fact that a certain amount of power is given by compressed air. This point will be discussed later on.

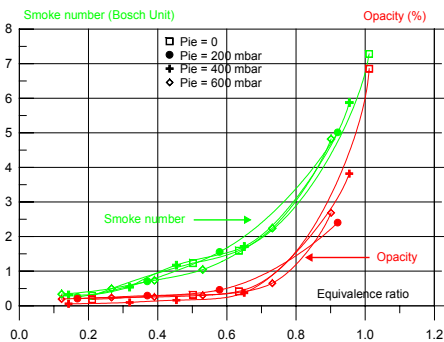
In Figure 4, same kinds of results are shown for 2500 rpm engine rotation speed.



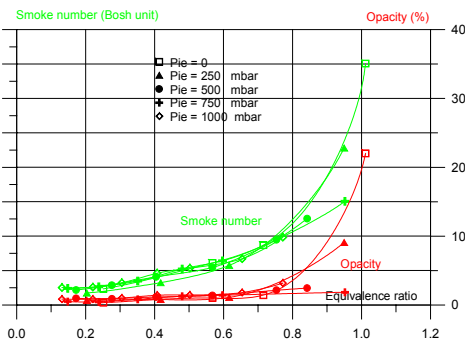
**Fig. 4** – Bmep and  $\eta_g$  versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 2500$  rpm

### 3.2 Influence of boost pressure on smoke emissions

As for the last curves, we choose to represent smoke emissions versus equivalence ratio. Results are presented in Figures 5 and 6 respectively for 1500 and 2500 rpm engine speed.



**Fig. 5** – Smoke number and opacity versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



**Fig. 6** – Smoke number and opacity versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 2500$  rpm

Both smoke number and opacity curves have been drawn even they are representatives of the same phenomena.

At 1500 rpm, it can be noticed that smoke emissions are close whichever the inlet engine pressure.

At 2500 rpm, some differences appear for equivalence ratio greater than 0.75 .

Equivalence ratio 0.75 can be considered for a diesel engine as maximum admissible value. Greater values lead to a quick increase of black smoke emission. So, if smoke emissions is consider as representative of combustion effectiveness, results show that combustion depends particularly on the equivalence ratio rather than on the boost pressure.

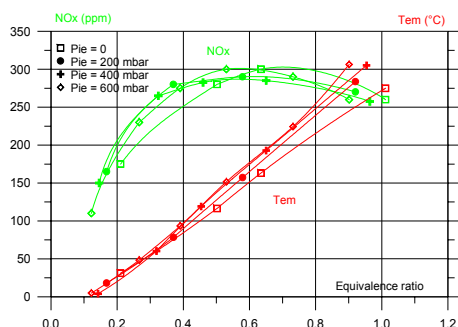
### 3.3 Influence of boost pressure on gas emissions concentration

#### 3.3.1 Nitrogen Oxide: NO<sub>x</sub>

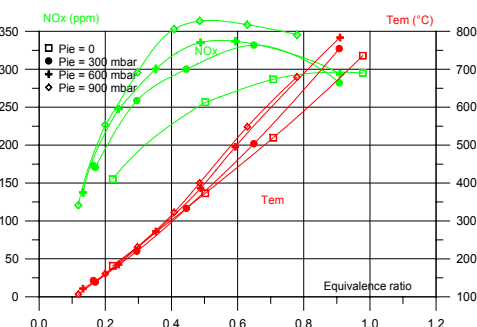
NO<sub>x</sub> depends mostly on equivalence ratio and on maximum temperature in combustion chamber. In consequence, NO<sub>x</sub> emission is indirectly linked with manifold exhaust temperature.

For engine speed 1500 rpm (Figure 7), the variation of NO<sub>x</sub> versus equivalence ratio is about the same for the different boost pressure.

At 2000 rpm there is a slight difference. A higher production of NO<sub>x</sub> when the boost pressure increase (Figure 8). The same phenomenon appears at 2500 rpm but this tendency is not so obvious.



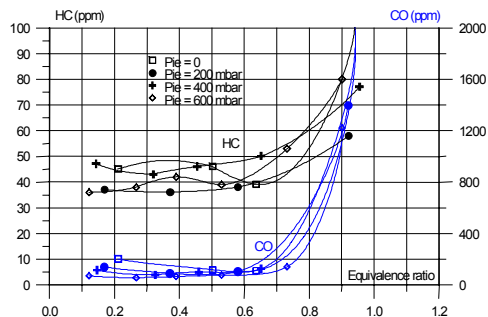
**Fig. 7** – NO<sub>x</sub> concentration and exhaust manifold temperature versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



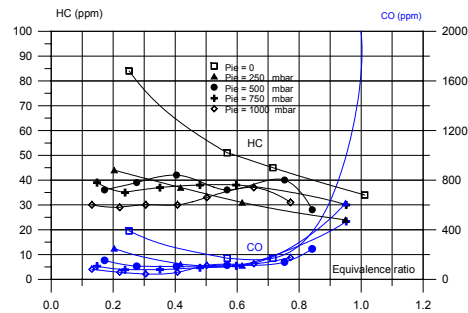
**Fig. 8** – NO<sub>x</sub> concentration and exhaust manifold temperature versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 2000$  rpm

#### 3.3.2 Hydrocarbon and Carbon Monoxide: HC, CO

HC and CO curves versus equivalence ratio are close (Figures 9 and 10). For the relative pressure Pie = 0, it is observed an increase of HC and CO at low equivalence ratio, especially for 2000 and 2500 rpm. This effect is probably due to the counter pressure at engine outlet. Pie = 0 means that boost pressure is equal to 0. In reality, this experiment corresponds to a test without supercharging, so there is a negative relative pressure at engine inlet and a positive pressure at engine outlet. Hence, we have a bad filling of the engine and exhaust gas recirculation.



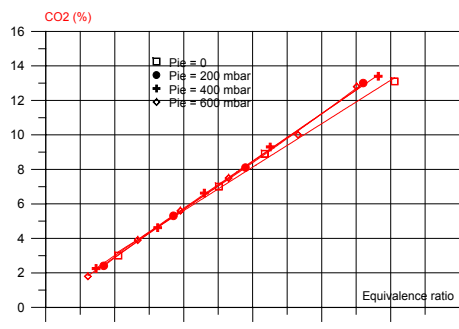
**Fig. 9** – HC and CO concentration versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



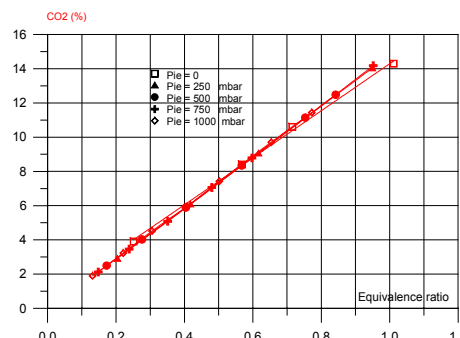
**Fig. 10** – HC and CO concentration versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 2500$  rpm

### 3.3.3 Carbon Dioxide: CO<sub>2</sub>

CO<sub>2</sub> emissions are proportional to equivalence ratio and independent of boost pressure as indicated in Figures 11 and 12.



**Fig. 11** – CO<sub>2</sub> concentration versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



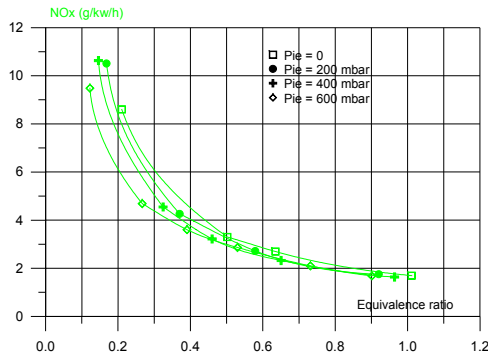
**Fig. 12** – CO<sub>2</sub> concentration versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 2500$  rpm

### 3.4 Influence of boost pressure on specific gas emissions

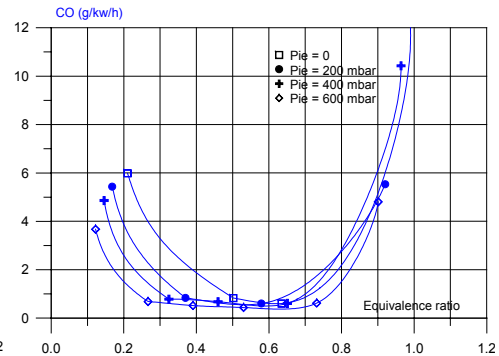
As noticed in chapter 3.1, for the same equivalence ratio brake specific fuel consumption improves when boost pressure increase and especially at low ratio. This effect has a direct influence on specific gas emission and we have approximately for all the gas considered:

- same level of specific gas emission for equivalence ratio between 0.4 and 0.8
- lower level with increasing of boost pressure for equivalence ratio less than 0.4.

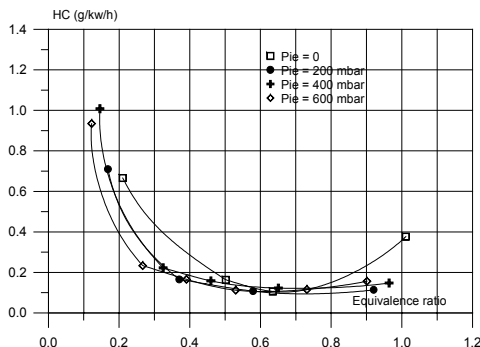
These results are represented for 1500 rpm in Figures 13 to 16.



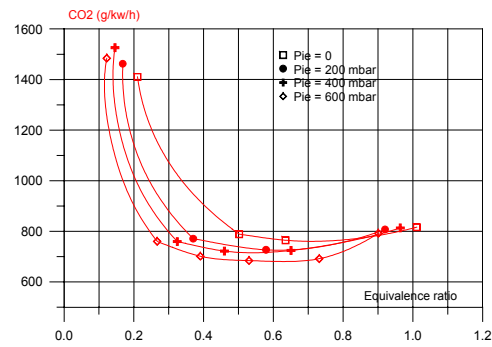
**Fig. 13** – NOx specific emission versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



**Fig. 14** – CO specific emission versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



**Fig. 15** – HC specific emission versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm



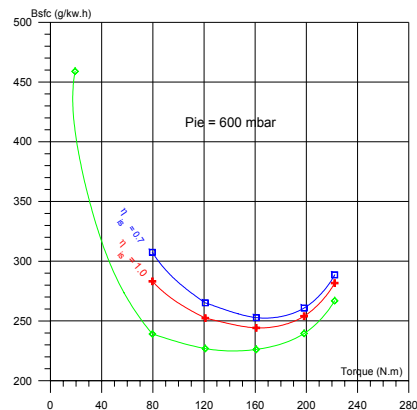
**Fig. 16** – CO2 specific emission versus equivalence ratio  $\phi$  for different relative engine inlet pressure at  $N = 1500$  rpm

#### 4 DISCUSSION

Results show that boost pressure have no significant influence on emissions for the same equivalence ratio, except for NOx where there is a slight tendency of increase with boost pressure for engine speed 2000 and 2500 rpm. If specific emissions are considered, we find lower level of pollutants for low equivalence ratio, which is due to improvement of brake specific fuel consumption with the increase of boost pressure.

This benefit is doubtful because in calculating Bsf<sub>c</sub> of the engine, it has not been taken into account the power required to produce the supercharging. In Figure 17 is represented the variation of Bsf<sub>c</sub> assuming an efficiency of compressor of 1.0 and 0.7 .

So, it appears that this advantage can be cancelled by a drastic increase of Bsf<sub>c</sub> at low load for high boost pressure. It is logical because the power needed to produce the compressed air become equivalent to the power produced by the engine (about 2.5 kW are necessary to get an inlet engine pressure of 600 mbar at 1500 rpm).



**Fig. 17** – Bsf versus torque for compressor efficiency 0.7 and 1.0 at  $N = 1500$  rpm

## CONCLUSION

Increases of boost pressure at low engine speeds don't lead directly to significant improvement of pollutant emissions or brake specific fuel consumption.

However, opportunities can be found in supercharging:

- supercharged engine has a higher torque, so regarding pollution vehicle can be more easily use in a convenient functioning area.
- instead of using compressed air for increasing engine power, it can be forwarded to use this air for treatment of exhaust gas as for example post combustion in order to improve the catalyst or burn of particulate.

This turbocharging concept oriented towards decreasing pollution emissions is already used in Diesel engines and it's now the object of interest for new direct injection engines.

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