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BACK PRESSURE EFFECTS ON VARIABLE GEOMETRY TURBINE PERFORMANCES

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ABSTRACT
Turbochargers are widely used in applications to increase specific power and decrease fuel consumption. However, recent anti-pollution regulations have became stricter and pressed automotive engineers to find new solutions to reduce Nox emissions. Two of these solutions are the catalytic converter and the intercooler system. All these modifications will change the initial matching of the turbocharger performance characteristics to the engine requirements. In this paper, several compressor wheel sizes are investigated to evaluate the turbine/compressor matching. The intercooler and catalytic converter back pressure induced are respectively modeled by a lower duct section downstream the compressor stage and a variable valve downstream the turbine stage. The influences of the different modifications are identified through the loading and the flow coefficients and also on classical turbine performance maps. First, an analogy between compressor wheel size and back pressure effects is underlined. Second, it is shown that initial control settings of turbine nozzle vanes are no longer appropriate with a catalytic converter.

INTRODUCTION
Turbocharger becomes an important part of diesel engines with a significant impact on engine power [1]. However, in recent years, it also proved its ability to improve specific fuel consumption and combustion. Recently, there is a growing trend worldwide toward strengthening emissions regulations for all vehicles. As a result, it becomes an essential condition for diesel engines to be coupled to turbochargers. Conventional technologies may not be able to cope with regulatory requirements as they become progressively stricter each year with emissions regulations [2]. Indeed, Euro 3 regulations (2000) imposed on all automotive vehicles a limit of 0.5 g/km for Nox rejection. This proportion was reduced to 0.25 g/km in 2005 for Euro 4 regulations. Thereby, new technologies have to be adopted such as variable geometry nozzle [3] used with EGR (Exhaust gas recirculation) [4]. One of these systems is the catalytic converter. This system, patented in 1969 captures the carbon monoxide from exhaust gas, and converts it into nitrogen and oxygen atoms. Nowadays, it is massively used to reduce the Nox inside the converter and avoid rejection (cf. Figure 1) in spark-ignition engines. Another solution consists in fitting an intercooler system between the compressor and the intake manifold. This reduces the inlet temperature of the engine, thus increasing the air density which permits to reach a higher specific power [5].

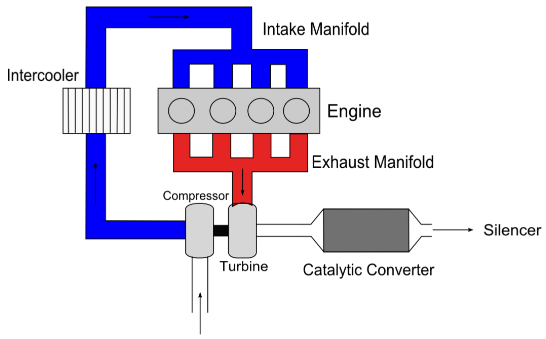


Figure 1: Schematic representation of the global system

For turbocharged engines, the two components (turbine and compressor) must be carefully chosen so that the good performance range of speed and mass flow coincide with those of the cylinder of the engine [6]. However, each modification of the engine configuration such as catalytic converter or intercooler cannot lead to the definition and the conception of a new turbocharger. Engineers have to modify existing systems and need to predict new turbine and compressor operating points [7]. In order to fit the basic configuration, the modifications induced by both the intercooler and the catalytic converter have to be characterized. The intercooler system imposes a pressure drop of about 0.3 bars in the intake manifold ($\Delta P=P_{2e}-P_{2i}$). This implies that for the same intake manifold pressure P_{2e}, the initial target compressor boost pressure is not reached.

$$\boldsymbol{P}_{compressor}=\frac{Q_a C_p T_{i1}}{\eta_c}\left[\left(\frac{Pi_{2i}}{Pi_1}\right)^{\frac{k-1}{k}}-1\right] \quad (1)$$

Thus, the power delivered by the compressor has to be increased for a same back-pressure. As compression work is mainly done by the rotation, this means a higher rotational speed. Due to power balance between turbine and compressor, the turbine operating point may change to non optimal functioning regions.

In the same way, the catalytic converter imposes a higher back-pressure than a simple exhaust system. The value of static-pressure-increase caused by the converter depends on the mass-flow rate going through the turbine. At high engine power output, the value is about 0.3 bars. Therefore, the total-to-static turbine stage pressure ratio (Pi_3/P_4) is no longer obtained at the same given mass flow rate, imposed by the engine. As a consequence, the power developed by the turbine cannot drive the compressor to reach the target boost pressure. The paper describes the evolution of the turbine VNT (variable nozzle turbine) operating point in two different turbine map definitions. The aims are to understand the impact of compressor and turbine back pressure on turbine performances. First, compressor back-pressure and dimension are studied to understand the evolution of the turbine performance. Then, the turbine back-pressure effects are discussed and the interaction with the compressor is assessed.

NOMENCLATURE

Q	[g/s]	Mass-flow
Qrt	[g/s]	Corrected turbine mass flow
Qrc	[g/s]	Corrected compressor mass flow
Nrt	[rpm/$K^{1/2}$]	Corrected rotational speed
C_p	[J/kg/k]	Specific heat at constant pressure
D	[m]	Diameter
H	[J/kg]	Specific Enthalpy
P	[bar]	Static Pressure
π	[bar]	Total Pressure
π_c		Compressor total-to-total pressure ratio
π_t		Turbine total-to-static pressure ratio
r		Ideal gas constant
T	[K]	Static Temperature
Ti	[K]	Total temperature
U	[m/s]	Tip speed

Special characters

ϕ	[-]	Flow coefficient
ψ	[-]	Loading coefficient
η	[-]	Efficiency
$\boldsymbol{\Phi}$	[Watt]	Power
VNT		Variable Nozzle Turbine

Subscripts

a	air
c	compressor
thm	overall
t	turbine
ts	Total-to-static reference
1	Compressor inlet
2i	Intercooler inlet
2e	Intercooler outlet/cylinder inlet
3	Turbine inlet
4	Turbine outlet

TURBOCHARGER TESTS

The tests were conducted on a test rig which ensures global functioning and instrumentation of a complete radial turbomachine (turbine fitted to its compressor) [8]. In order to widen the turbine and compressor performance maps, the compressor and turbine air-flow lines are separated. Therefore, the mass-flow rate and the pressure-ratio of the turbine and compressor can be set independently. The turbine is driven at steady flow conditions and the mass-flow rate is regulated. Another valve at the exit of the turbine stage is used to simulate back-pressure due to the converter. A VNT turbine is used for all the tests with five nozzles at different opening angles from fully closed (1/5) to fully opened (5/5). The variable geometry nozzle allows the turbine to best match diesel engine for different conditions by changing the stage capacity, through the throat area of the nozzle (cf. Figure 2). A detailed description of the VNT can be found in [9].

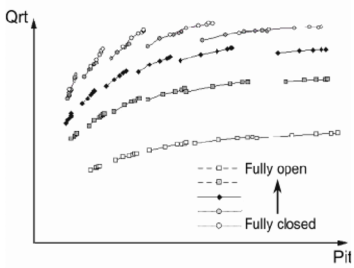


Figure 2: Effects of variable geometry nozzle on turbine map

The compressor inlet receives air from atmospheric conditions. A valve is positioned at the exit of the compressor stage to control the delivery pressure and mass-flow rate. Four different wheel diameter compressors are tested to evaluate turbine/compressor matching (cf. Tab 1).

Compressor names	Compressor wheel diameter (D/D_{ref})
TC1	0.89
TC2 (reference compressor)	1
TC3	1.06
TC4	1.42

Tab 1: Compressor wheel size

Two different test campaigns are conducted. Intercooler and compressor wheel size effects are firstly underlined, and then catalytic converter effects are studied.

For intercooler effects, 6 pressure-ratio lines (from 1.4 and 1.9) are described by acting on the compressor valve. These six pressure-ratios correspond to engine operative range of mass-flow as it can be seen in previous paper [9]. Tests are reproduced for each stator opening angle and for each compressor diameter. The inlet temperature is limited to ambient temperature (290 K). This is due to the excessive high rotational speed expected for TC1 at high temperature.

For turbine back-pressure effects, tests are realized with reference compressor (TC2) for an inlet temperature of 680K. Even if the level of rotational speed reached with TC2 is about 180 000 rpm, this is acceptable for the installation. In order to model the catalytic converter, the valve position downstream the turbine stage is set to obtain 3 different pressures: 1.3 bars, 1.6 bars and 1.8 bars which is the maximum acceptable considering the increase of the global pressure in the stage. The choice of these back-pressures corresponds to the pressure drop expected on real exhaust system with a catalytic converter. Practically, at higher pressure ratio (1.9), the valve position is set to obtain the desired pressure at the exit. Then for this same valve position, measurements are made for 2 other pressure ratios (π_{ts}=1.5 and π_{ts}=1.7).

PRESSURE-RATIO LINES

Results are not only presented in classical turbine performance maps (i.e. corrected mass-flow against pressure ratio and efficiency against pressure ratio), but also on an alternative representation consisting in plotting results in ψ-ø maps. The definitions of these two parameters are given below in equations 2 and 3.

$$\phi = \frac{QrT_{i3}}{U_3 P_{i3} \frac{\pi}{4} D_3} = \frac{240}{\pi^2}\frac{Qrt}{Nrt} \tag{2}$$

$$\psi = \frac{\Delta H}{U_3^2} = \frac{C_p T_{i3}\left(1-\left(\frac{1}{P_{it}}\right)^{\frac{\gamma-1}{\gamma}}\right)}{\eta_{ts} U_3^2} = \frac{1}{2}\eta_{ts}\left(\frac{C_s}{U_3}\right)^2 \tag{3}$$

These parameters are mainly used with Smith charts [10] to begin a new design but Binder [11] showed that loading and flow coefficients can be useful for a one-dimensional analysis of turbine performances. In fact, pressure-ratio lines are expected to be linear in ψ-ø map (cf. Figure 3).

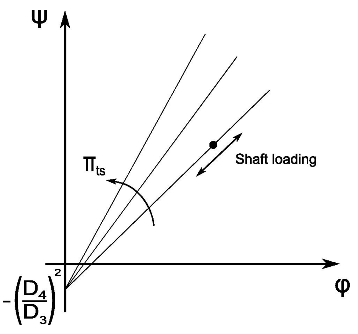


Figure 3: ψ-ø map

The intercept is the trim $(D_4/D_3)^2$ of the rotor. The slope increases with pressure ratio. Therefore, off-design operating conditions are then reflected by a displacement along the pressure line or the modification of its slope. In this case, the influence of intercooler system and catalytic converter on mass-flow can be characterized and predicted. This diagram can also be used to locate speed lines and best efficiency regions. Indeed, best efficiency regions are theoretically expected for $\psi \approx \eta_{\max}$ (cf. Figure 4).

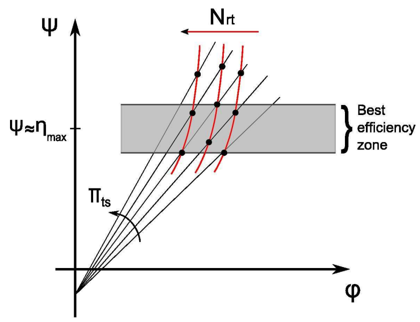


Figure 4: Influence of parameters in Psi-phi map

Indeed, loading coefficient is linked to efficiency (cf. eq. 3) and it is commonly admitted that best efficiency regions occur at U/Cs equal to 0.7. Considering that efficiency is nearly constant when U/Cs is fixed:

$$\psi = \frac{1}{2}\eta_{ts-\max}\left(\frac{C_s}{U_3}\right)^2 \approx \frac{1}{2}\eta_{ts-\max}\left(\frac{1}{0.7}\right)^2 \approx \eta_{ts-\max} \tag{4}$$

RESULTS

First, the influence of the compressor modifications is studied then the turbine back-pressure effects are discussed. Lastly, analogy between compressor and turbine back-pressure is presented.

Compressor back-pressure and wheel size effects

First, the evolution of the three dimensionless parameters (Qrt, Nrt and η) is considered.

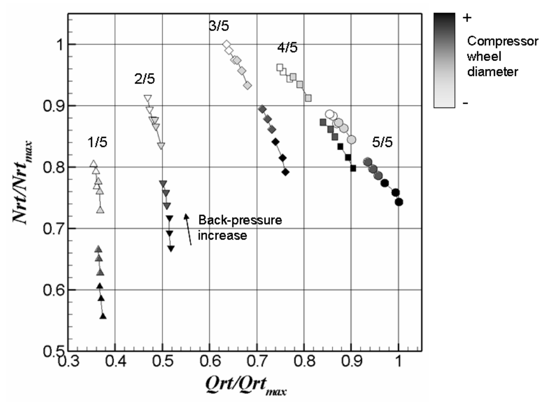
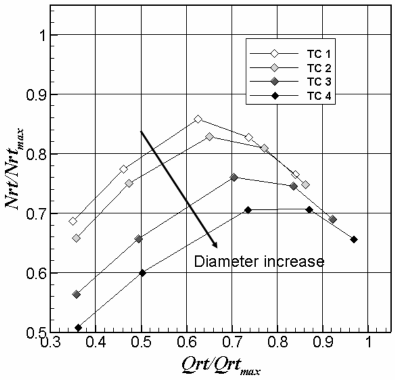


Figure 5: Evolution of Nrt for Pit=1.6

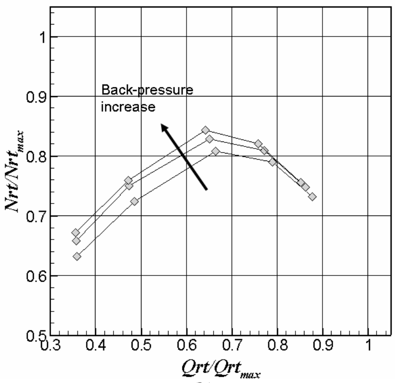
In Figure 5, Nrt is plotted against Qrt for all compressors and all turbine nozzles at nominal pressure ratio (1.6). The different symbol shapes correspond to the different nozzle opening angles (from 1/5 to 5/5) and each symbol colour corresponds to a different compressor (from TC1 to TC4). The three connected points represent the three different back-pressures tested for each configuration. An increase of compressor back-pressure imposes a higher rotational speed due to a displacement in compressor performance map. In the same way, compressor wheel size influences the reduced speed. The work done by the compressor can be written for an axial inlet as:

$$W = U_2 C_{\theta 2} - U_1 C_{\theta 1} = R_2 \omega_2 C_{\theta 2} \quad (4)$$

Consequently, the highest wheel diameter R_2 is, the lowest rotational speed will be for a given turbine work. These two results are classical for compressor stage. However, it appears that the displacement in turbine map remains on the same line for each nozzle (the junction between different compressors depends on the explored range). This continuity points up an analogy between the two effects (compressor wheel size and back-pressure) for the turbine stage as illustrated on Figure 6. It shows the evolution of Nrt for the 4 different compressors at constant back-pressure valve position (a) and for the same compressor (TC2) (b) at different back-pressures.



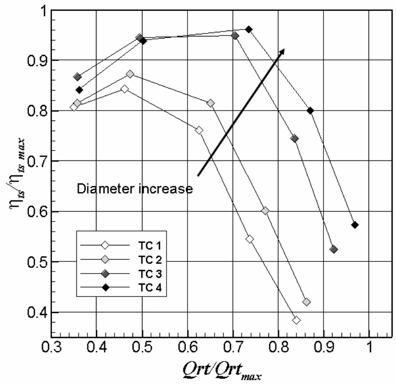
(a)



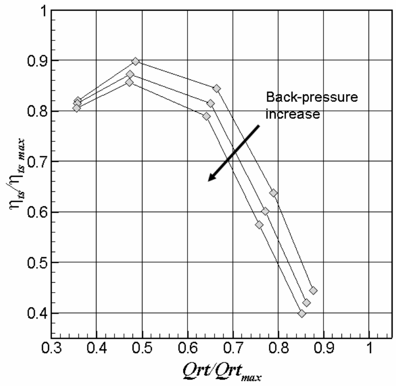
(b)

Figure 6: Influence of compressor (a) and back-pressure (b) on reduced speed

Wheel or back-pressure compressor modifications are equivalent in term of turbine speed. This means the evolution is mainly driven by the power balance between compressor and turbine stages. The efficiency of the turbine stage has now to be considered to evaluate the effects on performances.



(a)



(b)

Figure 7: Influence of compressor (a) and back-pressure (b) on efficiency

As it can be expected, the modification of efficiency is also due to a displacement in the turbine map than a real compressor functioning influence. This enforces the importance of turbine to compressor matching as shown in Korakianitis **[7]** compared to the separated effects of the two stages. Thus, the influence of intercooler system can be modelled by a change of compressor wheel diameter.

Results are now plotted in alternative map (ψ-ø) on Figure 8.

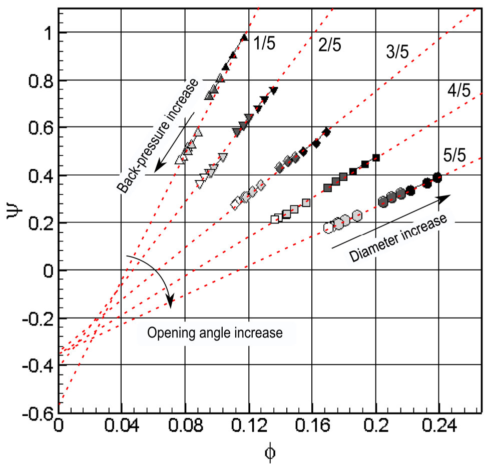


Figure 8: Influence of compressor modifications and turbine nozzle

As expected, pressure-ratio lines are straight lines but the intercept is not fixed as predicted. The trim is fixed by the geometry at 0.67. A correction has to be brought to predict the stage behavior. The origin of this mismatch should be found in the second hypothesis of theoretical approach considered for the integration of Euler equation. Actually, the angular momentum is not conserved at the outlet of a turbine stage, particularly at off-design conditions. An analysis of these results should give indications on the correction to bring, and help to choose the correct outlet radius considered for the integration.

Values of ψ and ø decrease along lines due to the intercooler system modeled by a back-pressure. Thanks to this diagram, effects of modifications allow to predict easily the displacement of turbine operating point. Intercooler effect is in opposition with compressor wheel size effect as seen previously. This map gives a tendency of efficiency evolution with an intercooler system (cf. Figure 4).

Turbine back-pressure effects

To simulate a catalytic converter, different turbine back-pressures have been investigated for the five nozzles. Results are plotted on Figure 9. Turbine back-pressures (0.8, 0.6 and 0.3 bars) have been set at high pressure ratio (Pit =1.9) to be representative of an exhaust system with a catalytic converter.

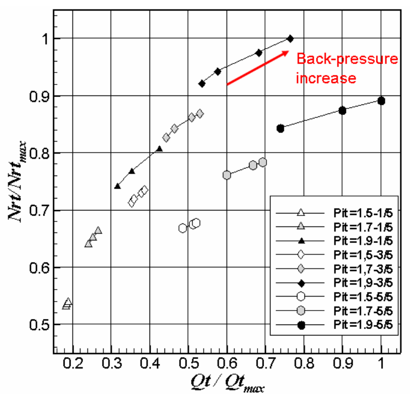


Figure 9: Evolution of Nrt against Qt

It appears there are very few modifications on speed for the points at Pit =1.5. The influence of back-pressure decreases with the exit pressure of the turbine stage due to the experimental protocol previously described. However, for the other pressure ratio, the influence is significant and speed increases with back-pressure. This is due to a higher inlet total pressure which imposes a higher density and so an increase of mass-flow Q_t . The turbine power can be written as:

$$\boldsymbol{P}_{turb} = Q_t C_p T_{1t} \eta_{ts} \left(1 - \left(\frac{1}{\pi_{ts}} \right)^{\frac{\gamma-1}{\gamma}} \right) \tag{5}$$

Thus, the power delivered by the turbine increases for a given pressure ratio. A higher rotational speed is reached with a variation of 10000 rpm for the nominal speed for all nozzles (8% of the nominal speed).

The variations of turbine overall efficiency are now considered (cf. Figure 10). At high turbine temperature, thermal losses overestimate expansion work transfer and then turbine efficiency so turbine overall efficiency (turbine efficiency times turbocharger mechanical efficiency) is preferred to turbine efficiency. Moreover, it takes into account the bearing and compressor losses. As previously, turbine back-pressure effects are significant at high pressure ratio. For Pit=1.7 and Pit=1.9, the turbine overall efficiency decreases due to a displacement in the turbine map as compressor back-pressure effects. This point will be discussed later.

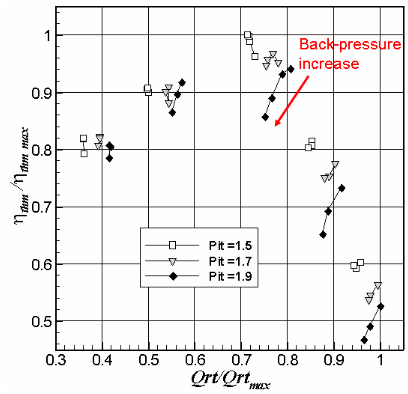


Figure 10: Influence of turbine back-pressure on turbine overall efficiency

Due to speed modifications, compressor operating points will be modified as it can be seen on Figure 11.

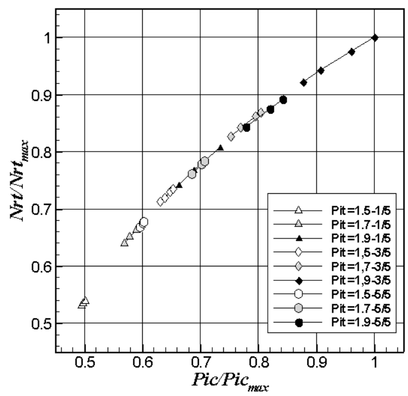


Figure 11: Evolution of Pic for three nozzles (a=1/5, b=3/5 and c=5/5)

Turbine back-pressure does not affect compressor functioning because compressor points remain on the same trajectory. Changes of nozzle opening angle have the same influence. This means the influence of catalytic converter on compressor operating point can be reset by a change of nozzle opening angle. The initial setting of compressor to engine matching imposes an optimal value of Pic that will be altered by the converter. A correction on pressure regulations of variable geometry system will enable to preserve good settings.

On Figure 12, turbine back-pressure tests are plotted in ψ-ø map with best efficiency contours.

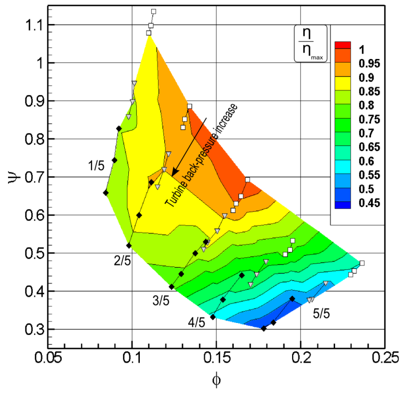


Figure 12: Influence of turbine back-pressure in Psi-Phi map

Catalytic converter modifies the turbine point in the same way as intercooler system. Values of ψ and ø decrease with back-pressure and the slope of curves depends on nozzle opening angle. Therefore, there is no continuity as previously between iso pressure-ratio lines. In terms of performance, catalytic converter system, modelled by a turbine back-pressure, moves turbine points to lower efficiency regions.

CONCLUSION

Test campaign allows characterizing the effects of intercooler system and catalytic converter. An analogy between compressor wheel size and compressor back-pressure is point up which helps to predict intercooler influence. This is essential because most of vehicles will be equipped with such systems either diesel or gasoline engines. This will provide a basis to know how to adapt existing system.
An alternative map is used to give easily tendencies on what is expected in terms of mass-flow, speed and efficiency. Then, the influence of catalytic converter does not enable to use the same settings as a simple exhaust system. Engineers have to modify the setting of nozzle opening angle to keep good engine functioning. In this study, effects of turbine back-pressure on engine have not been taken into account. A higher inlet turbine pressure corresponds to a higher outlet engine pressure. It can have a harmful effect on engine power if boost pressure is not increased in higher proportion due to the increment of engine pumping losses.
Further tests on turbine inlet temperature will permit to study the influence of inlet conditions. A description of different turbine maps can also be made to understand the influence of temperature on power balance.

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