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Energy Procedia 101 (2016) 630 – 637

Energy Procedia

71st Conference of the Italian Thermal Machines Engineering Association, ATI2016, 14-16 September 2016, Turin, Italy

Effect of Circuit Geometry on Steady Flow Performance of an Automotive Turbocharger Compressor

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Abstract

Downsizing and turbocharging are today considered an effective way to reduce CO_2 emissions in automotive gasoline engines. To this aim, a deep knowledge of turbocharger behavior could be a key solution to improve the engine-turbocharger matching calculation. The influence of the intake system geometry on the surge line position is an important aspect to guide the project of the intake manifold, enlarging the compressor stable zone. This aspect has a considerable impact on engine performance, especially during transient operation. A wide experimental investigation was carried out at the turbocharger test facility of the University of Genoa on a small turbocharger compressor. Compressor characteristic curves measured considering an automotive intake circuit are compared with standard maps provided by turbocharger's manufacturer. This information allows the optimization of 1D model implementing more realistic maps of compressor. The influence of three different layouts has been investigated varying overall circuit volume and length, keeping values in a range compatible with passenger cars packaging constraints. In the paper, the main results of the experimental campaign are presented taking into account the influence of geometry variations on compressor map and surge line position.

Peer-review under responsibility of the Scientific Committee of ATI 2016.

Keywords: Turbocharging; centrifugal compressor; internal combustion engine; surge; unstable zone; experimental.

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Peer-review under responsibility of the Scientific Committee of ATI 2016.
doi:10.1016/j.egypro.2016.11.080

1. Introduction

The goal of CO_2 reduction imposed by the European Commission for new registered passenger cars leaded manufacturers to develop technological solutions to limit vehicles exhaust emissions and to reduce engine fuel consumption [1], especially considering vehicle real driving conditions and cold start [2]. Downsizing and turbocharging techniques seem to be a key technology to reach this target. Since 1D models are usually employed for matching calculations, an accurate definition of turbocharger (TC) performance when coupled to the engine is required to properly accomplish numerical simulation models [3-5]. Some typical drawbacks are linked to the performance maps typically provided by turbocharger manufacturer and implemented in 1D model:

- Only steady flow maps are available, while TC coupled to the engine operates under unsteady flow conditions
- Maps are usually defined over a limited range requiring strong extrapolation [6]
- Experimental characteristic curves are commonly measured in steady state using layout strongly different to the intake automotive circuit
- The methodology adopted to define the surge line position is usually not specified
- No information is provided about sensors location, thus increasing the uncertainty in the compressor efficiency calculation.

Therefore, the availability of detailed experimental information on turbocharger performance under both steady and unsteady flow conditions is an essential requirement to improve simulation models [7, 8].

The reduction of engine displacement with a reduced number of cylinders and the adoption of Variable Valve Actuation systems drives the compressor to work under unsteady flow conditions [9-11]. Compressor unsteady behavior also occurs near the surge line, thus reducing mass flow rate and inducing local unstable operation. The surge phenomenon, i.e. the periodic return of the compressed fluid through the compressor towards its inlet, can limit the pressure ratio capability, especially when the engine is working at low speed and high load. Car manufacturers limit boost pressure level to prevent surge condition in order to avoid a deterioration of rotor blades, a significant drop in the efficiency and a fluctuation of the supplied engine power. Compressor stability is a very complex issue and several studies are available in the open literature to deepen the dynamic of this phenomenon and the surge line definition properly controlling compressor instability [11-14]. In 1976, Greitzer [11] proposed a mathematical model to reproduce the dynamic response of compression system as a function of the circuit geometry [11] representing the starting point for many studies about compressor surge. In [15] Capon and Morris compared two compressor maps, one measured in standard steady-state operation [16], the other measured coupling the compressor inlet to an automotive intake circuit. These results highlighted the importance of dedicated experimental tests to reproduce the real compressor behavior, correctly defining the position of the surge line.

The strict packaging requirements for the engine and its accessories generally produce unfavorable flow at the compressor inlet. Different studies to investigate the influence on compressor performance of the flow conditions at its inlet have been developed, proposing solutions to improve the flow field through different diffuser shapes [17] or bleed channels [14] to obtain map improvements. In [18] and [19] the influence of the compressor inlet design on compressor map has been investigated, while pre-whirl flow at compressor inlet has been studied in [20, 21] in order to reduce negative effects of an inlet flow perpendicular to the compressor axis. Particularly, Galindo et al. [21] proposed an inlet swirl generator device (SGD) that can be used also to extend the surge margin by modifying its blades depending on the engine operative conditions.

Even if the influence of compressor intake circuit has been deeply investigated, a lack of information can be observed concerning the influence of the compressor downstream circuit.

In the paper, the results of a wide experimental campaign in steady state are presented. The investigation allowed to extend compressor maps definition considering the typical engine circuit geometries, thus reducing extrapolation errors in simulation models and guiding circuit design to achieve an optimization of turbocharger behavior. A comparison between compressor maps measured adopting an automotive intake circuit and turbocharger manufacturer's map is reported. Through a specific variable volume plenum integrated in the downstream compressor circuit, three different automotive intake circuit arrangements were tested, following specific requirements by a car manufacturer. The last stable point of the compressor was fixed monitoring the instantaneous signal measured at the compressor outlet section for each turbocharger rotational speed levels considered, in order to univocally define the surge limit position in the range of interest.

Nomenclature

L_{eq}	compressor downstream pipe length [m]
M_c	compressor mass flow rate [kg/s]
n	turbocharger rotational speed [rpm]
p_0	pressure reference condition, equal to 0.981 [bar]
p_{T1}	compressor inlet total pressure [bar]
p_{T2}	compressor outlet total pressure [bar]
$p_{2\ inst}$	compressor outlet instantaneous static pressure [bar]
T_0	temperature reference condition, equal to 293.15 [K]
T_{T1}	compressor inlet total temperature [K]
T_{T2}	compressor outlet total temperature [K]
T_{T2is}	compressor outlet total temperature for an isentropic process [K]
V_{eq}	volume between compressor and downstream valve [m^3]

2. Experimental set-up

The experimental activity aimed at a better understanding the effect of downstream circuit capacity on compressor surge line position, was carried out using different outlet circuit geometries.

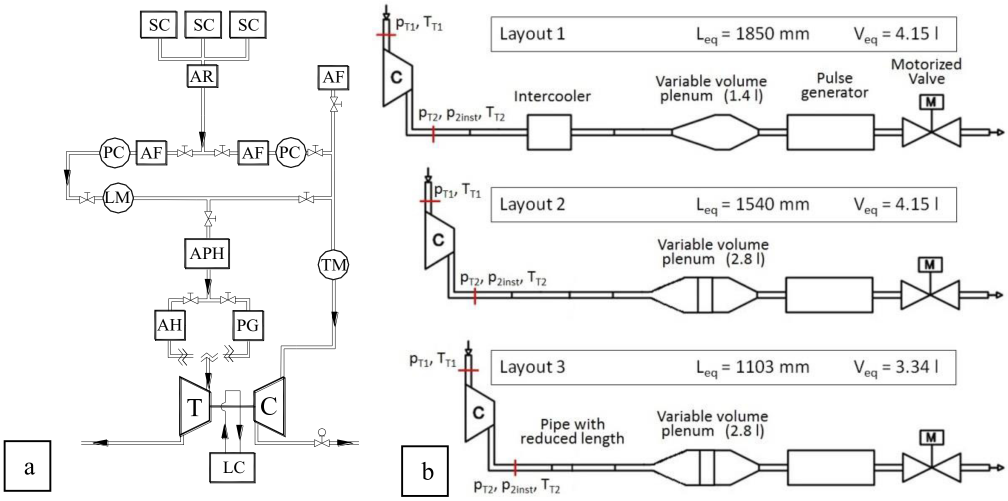
The turbocharger test facility of the University of Genoa (Fig.1a) is an experimental compressed-air apparatus that allows to performing investigations on intake and exhaust automotive components under both steady and unsteady flow conditions. Dry clean air is delivered by three screw compressors, which could supply a total mass flow rate of 0.6 kg/s at a maximum pressure of 8 bar.

Through an electrical air heating station it is possible to reach turbine temperature up to 750 °C, allowing “cold” and “hot” experimental investigations. A more detailed description of the test bench is presented in [10, 22].

A specific circuit with a variable volume plenum was integrated in the compressor outlet circuit, in order to study the effect of three different configurations on steady compressor performance with special reference to the surge line position. The first arrangement is similar in volume and length to the reference automotive intake circuit layout adopting the automotive intercooler. The second layout is characterized by different length with the same capacity of the first one, removing the intercooler. The last configuration without intercooler as well, is characterized by lower values of length and volume, as can be observed in Fig.1b, where the arrangements adopted are shown together with the equivalent values of length and volume and with measuring station used to evaluate compressor performance (section 1 and section 2). In each configuration a pulse generator consisting in a rotating valve was adopted in order to study also the effect of pulsating flow on compressor map. However, results on this subject will be discussed in further works. It’s important to underline that every change in length and volume has been obtained maintaining the same circuit from the compressor outlet to the main measuring station downstream the machine (where p_{T2} and T_{T2} were measured). This choice was adopted to obtain comparable data and to avoid different heat transfer effect, which could be affect temperature levels distorting compressor efficiency evaluation [22]. Even if the experimental investigation was performed at low air temperature at the turbine inlet, both the turbocharger and the relevant connecting pipes were thermally insulated to estimate with good accuracy turbocharger performance reducing heat transfer phenomena [22].

As regards the average levels of thermodynamic quantities, measurements were performed through a PC-controlled data acquisition system, using interactive procedures developed in LabVIEW® environment.

A high frequency response pressure sensor was used to detect instantaneous signal downstream the compressor volute exit to identify surge occurrence. The pressure transducer was installed close to the wall to avoid signal distortion due to the connecting pipe and a fast response data acquisition card was used [23].



AF	Air Filter	LM	Laminar Flow Meter
AH	Air Heater	PC	Pressure Control
APH	Air Pre-Heater	PG	Pulse Generator
AR	Air Reservoir	SC	Screw Compressor
C	Compressor	T	Turbine
LC	Lubricating Circuit	TM	Thermal Mass Flow Meter

Fig. 1. Schematic layout of University of Genoa test facility (a) and downstream compressor circuit layouts (b).

Different average parameters were measured, as follows:

- Static pressures were measured at the inlet and outlet section and across the intercooler (if used) through strain-gauge and piezoresistive transducers characterized by an accuracy of ±0.15% of the full scale.
- Air temperature levels were measured at the compressor inlet and downstream the intercooler (if adopted) through platinum resistance thermometers (Pt 100 Ohm) with an accuracy of 0.15 °C ±0.2% of measured value. At compressor outlet section a K type thermocouple was adopted.
- Turbocharger rotational speed was detected by an inductive probe mounted close to the compressor wheel with an accuracy of ±0.009% of the full scale.
- Compressor mass flow rate was measured through a thermal mass flow meter with an accuracy of 0.9% of measured value and ±0.05% of the full scale.

To univocally locate the surge line position, instantaneous compressor outlet pressure was recorded [23]. Since its forcing frequency is affected by surge occurrence, monitoring the time based signal it was possible to state when it turned from the almost flat stable condition shape to a clearly sinusoidal wave typical of the surge condition. In the same time, compressor mass flow rate was registered and compared with the last recorded value measured in the stable zone. In order to define a reliable and repeatable method, the surge line was then located by decreasing the last stable mass flow rate value of 2.5% of the difference between the two mass flow levels. The corresponding pressure ratio was then evaluated considering a third order interpolating equation, obtained using measured data.

The experimental investigation was performed on a small MHI turbocharger for downsized Spark Ignition engine application, characterized by an impeller diameter of 40 mm. Steady flow characteristic curves were measured at different levels of corrected rotational speed (80000, 100000, 120000, 140000 rpm) for each layout configurations, from the choking region to the surge line. The definition of compressor curves over a wide range was performed by changing the external circuit characteristic through the motorized valve located downstream the compressor (Fig.1) and by properly controlling the turbine work output.

All quantities are scaled based on the conventional non-dimensional groups to properly take into account Mach number similitude at the compressor inlet.

Compressor maps here reported refer to the following parameters:

- Corrected rotational speed [rpm]

$$n_{cr} = \frac{n \cdot \sqrt{T_0}}{\sqrt{T_{T1}}} \quad (1)$$

- Total to total compression ratio [-]

$$\beta_{cTT} = \frac{p_{T2}}{p_{T1}} \quad (2)$$

- Corrected mass flow rate [kg/s]

$$M_{cr} = \frac{M_c \cdot p_0 \cdot \sqrt{T_{T1}}}{p_{T1} \cdot \sqrt{T_0}} \quad (3)$$

- Isentropic total to total efficiency [-]

$$\eta_{cTT} = \frac{T_{T2is} - T_{T1}}{T_{T2} - T_{T1}} \quad (4)$$

As an example of the procedure followed to identify surge phenomenon, in Fig.2 a single compressor iso-speed is shown with special reference to a stable operating condition (called "a"), to the last stable point recorded before surge occurs (called "b") and to an operating condition in unstable zone (marked in red). Looking at instantaneous pressure signals (Fig.3) recorded at the compressor outlet section for the three conditions above defined (Fig.2), it is apparent the different shape of signals when the compressor works in stable zone and in unstable operating condition, this last characterized by a sinusoidal shape typically occurring in the deep surge zone.

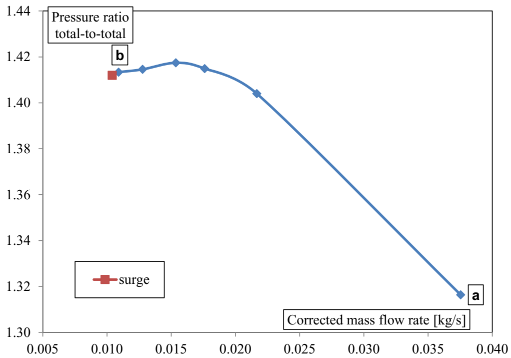


Fig. 2. Compressor isospeed (12000 rpm) with considered operating points.

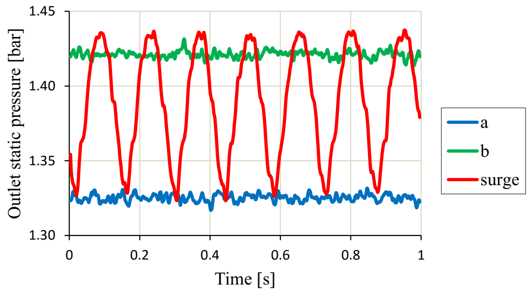


Fig. 3. Pressure signals recorded downstream the compressor.

Using a Fast Fourier Transform (FFT) analysis, it is possible to monitor and detect the typical force frequency helping the identification of surge phenomenon. In Fig.4 FFT analysis obtained for the operating point "b" is reported on the left side, while the result obtained in the unstable zone is shown on the right side. By comparing each FFT analysis the typical force frequency peak related to the surge occurrence can be clearly observed.

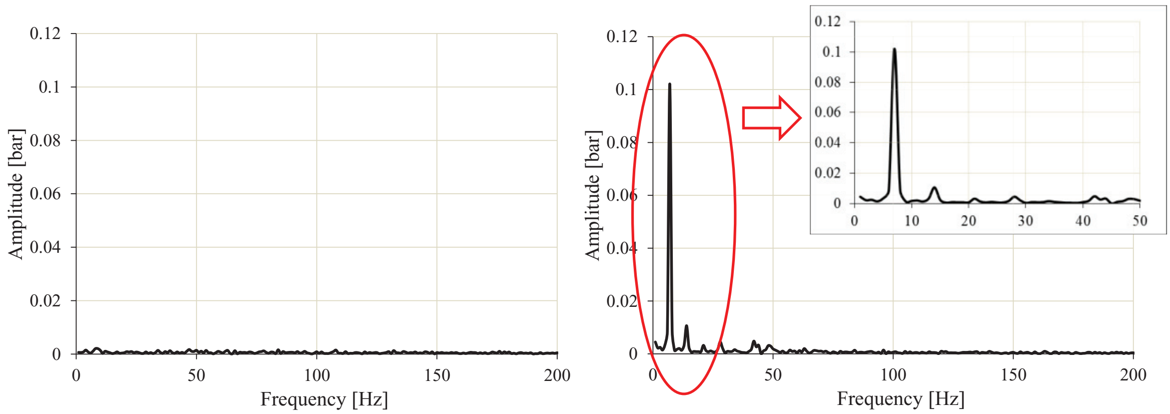


Fig. 4. FFT analysis on compressor outlet pressure signal referring to the operating point “b” (on the left) and to surge condition (on the right).

3. Experimental results

In Fig.5a a comparison between experimental compressor pressure ratio curves versus corrected mass flow rate referred to each arrangements is shown together with pressure ratio compressor map provided by turbocharger manufacturer (in black line). Besides, in Fig.5b, total-to-total compressor efficiency curves are shown.

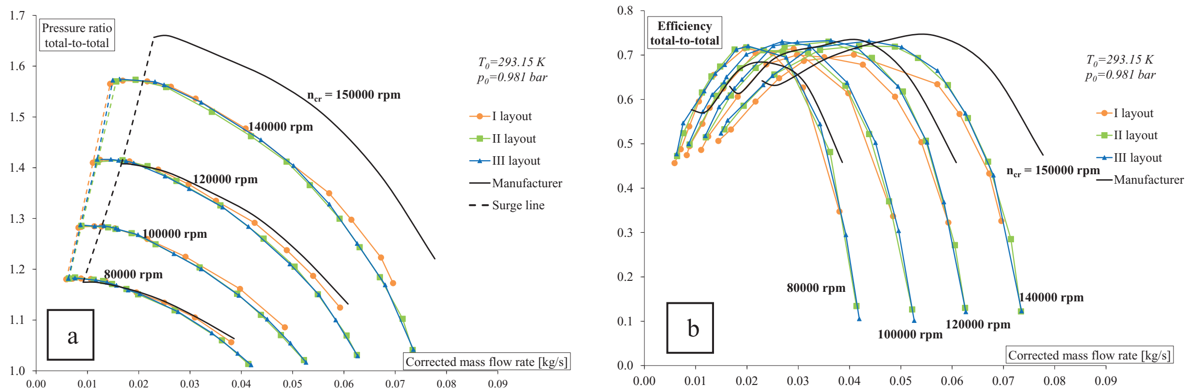


Fig. 5. Compressor pressure ratio map (a) and total-to-total efficiency map (b).

Considering the pressure ratio map, slight deviations of steady state characteristic curves measured for each circuit configurations can be observed, particularly in the choking region and in the surge line position. Looking at higher level of mass flow rate, the first configuration seems to be characterized by a higher pressure ratio for the same level of corrected mass flow rate, thus improving the boosting capacity. Considering the left side of the map, compressor stable zone seems to be slightly wider in the case of the first layout giving an improvement in air mass flow ranging from 4 to 7 per cent. In particular, the stable zone enlargement seems to be related to the damping effect of the intercooler and to the lower volume of the plenum. Negligible differences can be observed between the second and the third layout considered, both of them characterized by the absence of the intercooler and by the same volume of the plenum. Comparing the surge line position for each layout with respect to the manufacturer curves (black line in Fig.6) a strong difference in the stable zone width can be observed. Particularly, considering the first layout, an improvement in air mass flow of about 35 per cent can be noticed, guaranteeing better engine operating conditions.

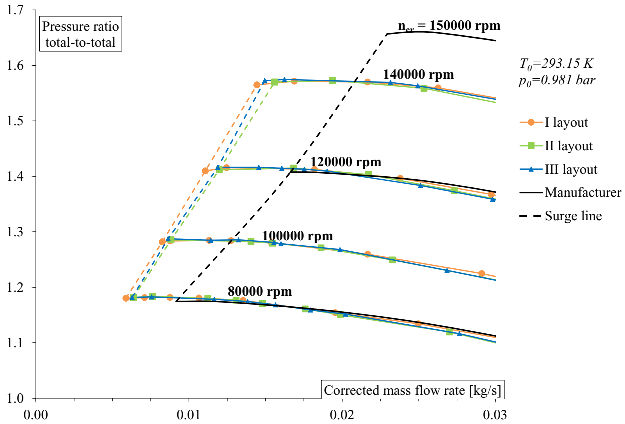


Fig. 6. View of surge margin for each layout considered and in the case of turbocharger manufacturer.

Concerning the strong differences between manufacturer's and University of Genoa's surge line position, the circuit geometry and sensors position play a significant role. While the surge lines for each arrangements have been measured adopting the same methodology, no details are available about manufacturer's procedure followed during tests. It seems that TC manufacturer is more conservative in the surge line location leading to a widest unstable zone. However, the enlargement of the stable area obtained considering the real intake automotive circuit coupled to the compressor, confirms the importance of experimental tests here reported to improve engine-turbocharger matching calculation

As regards compressor total-to-total efficiency, in Fig.5b a comparison between University of Genoa's and manufacturer's data is shown. Slight differences can be detected between each configurations considered, while noticeable deviations can be observed in the case of manufacturer map, probably related to the different location of measuring stations. Considering the efficiency equation followed (Eq.4), a change in the position of measuring station at compressor exit and a different insulating materials adopted, should modify compressor outlet temperature, thus affecting efficiency evaluation. Indeed, to guarantee the comparison between the experimental efficiency curves, the measuring stations have been maintained in the same position for each layout. It must be remarked that the arrangements considered was limited by automotive engines constraints in order to investigate only realistic automotive intake layout.

4. Conclusions

In the paper an experimental study on the effect of circuit geometry on steady flow performance of an automotive turbocharger compressor has been presented. Two different aspects have been analyzed:

- The influence of volume and length variations in the outlet compressor circuit, maintaining total values in the range allowed by engine constraints
- The differences between turbocharger manufacturer's map and characteristic curves experimentally measured considering a real intake automotive circuit.

Experimental results show slight differences in the compressor maps measured considering three different outlet arrangements, with special reference to the choking area and to the surge line position. This means that no appreciable improvement in turbocharger performance seems to be achievable through a slight modification of compressor outlet geometry. Considering turbocharger manufacturer's map, prominent differences can be noticed, especially in the surge line position maybe due to the different procedures followed to define it and to different volumes of the experimental layout. Manufacturer's data are usually referred to typical steady test bench arrangement [16], strongly different from automotive application. Results obtained confirm the importance of dedicated experimental investigation to improve simulation models, especially considering the real intake automotive circuit.

Acknowledgements

The research leading to these results has received funding from the European Union Seventh Framework Programme [FP7/2007-2011] under Grant Agreement n° 234032 (EU Project "POWERFUL"). The Authors wish to thank Eng. Giulio Marmorato for the collaboration in carrying out experimental measurements.

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