Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 45 (2014) 1116 – 1125

Energy Procedia

68th Conference of the Italian Thermal Machines Engineering Association, ATI2013

Highly flexible hot gas generation system for turbocharger testing

Rodolfo Bontempo[a], Massimo Cardone[a], Marcello Manna[a]*, Giovanni Vorraro[a]

[a]Università degli Studi di Napoli – Federico II – Dipartimento di Ingegneria Industriale, via Claudio 21, 8015, Napoli, Italy

Abstract

The paper presents the design and prototyping of a gas generator system for turbocharger experimental testing capable of delivering a wide range of flow rates with adequate thermodynamic characteristics. The system feeds the turbocharger test section with an hot gas stream of prescribed mean and time varying pressure and temperature in order to fully span the operating domain of the device with controlled accuracy. Compared with the more conventional gas combustor system, it allows for a safer rig operation ensured by the structural robustness of a four stroke diesel engine and an easier turbine inlet flow control. The latter is achieved by means of an external supercharging station and a modern ICE electronic control unit so that mass flow rate, pressure and temperature values can be set independently.
The steady compressor and turbine performance maps can be obtained operating the rig according to a conventional procedure, i.e. collecting a set of flow rate pressure ratio data points ($\dot{m}_g$, π) for given hot gas properties. Alternatively using more advanced operating modes unsteady testing is possible to reproduce the complexities characterizing the driving cycles required by the latest European regulations.

Selection and peer-review under responsibility of ATI NAZIONALE

Keywords: turbocharging, hot gas generation system, rig

1. Introduction

Latest trends emerging in the internal combustion engine industry have witnessed the renaissance of turbocharging as the technology capable to bring significant changes to several areas of automotive engineering.

* Marcello Manna. Tel.: +39-081-768-3287; fax: +39-081-239-4165.
E-mail address: marcello.manna@unina.it

Selection and peer-review under responsibility of ATI NAZIONALE
doi:10.1016/j.egypro.2014.01.117

Pros of the application of turbocharger to ICE are well known and, in the downsizing concept, involve a reduction of the brake specific fuel consumption (BSFC) and of the pollutant emissions. In order to achieve these benefits, an accurate matching is requested, so that reliable operating maps of both compressor and turbine are very important. Unfortunately, turbocharger manufacturers usually offer few data of poor quality to their customers, disregarding, for instance, the potential occurrence of compressor instabilities (rotating stall and surge) or the effects of pulsating flow conditions at turbine inlet. Very often information about the turbine operating conditions are provided with a single continuous line with no details about the effects of the turbocharger rotational speed and even efficiency data are typically not detailed. Sometimes turbine data have been measured in continuous and cold flow rigs, whose flow conditions differ very much from the real ones.

Commercial test rigs such as the Austrian AVL or the German FEV [1,2] are both based on the use of a combustion chamber, and with their large flow rate and temperature span offer the possibility to obtain accurate performance data, typically in steady flow conditions. Different solutions have been proposed by other research centers [9,10] where unsteadiness is typical introduced inserting on the turbine inlet manifold a rotating valve.

The Spanish rig of the CMT-Motores of the University of Valencia is instead based on the use of an internal combustion engine as hot gas generator, and unsteadiness at turbine inlet is accounted for inserting on the manifold a valve train system of a reciprocating ICE [3,4].

Similarly to the previously discussed rig, the present facility is also based on the use of an internal combustion engine as hot gas generator, while differences mainly concern the cold gas generator system feeding the ICE. The rig is capable of producing accurate steady and unsteady data according to a fully automated electronically controlled procedure.

Nomenclature

AC	alternate current
BMEP	brake mean effective pressure
BSL	best straight line
CGG	cold gas generator
DC	direct current
DCU	data acquisition and process control unit
FS	full scale
HGG	hot gas generator
ICE	internal combustion engine
$\dot{m}_a$	air mass flow rate
$\dot{m}_f$	fuel mass flow rate
$\dot{m}_g$	gas mass flow rate
p_a	air pressure
p_g	gas pressure
PID	proportional-integrative-derivative controller
PLCU	part load control unit
T_a	air temperature
T_g	gas temperature
TC	turbocharger
TIT	turbine inlet temperature
TS	test section

Greek

π	pressure ratio
θ	dimensionless temperature T/T_{ref}

Subscripts

a	air
f	fuel
g	gas
in	inlet
out	outlet
ref	reference

2. The rig design targets

As already mentioned in the introduction the latest engine developments heavily relies on the use of turbochargers in nearly all ICE applications both in the propulsion and industrial context. For this reason all manufacturers produce family of turbochargers, characterized by different sizes and performance maps. In order to maximize the flexibility of a test rig, all of these peculiarities have to be taken into account at the design stage. Therefore a preliminary in-depth study of the rig operating envelope has to be carefully performed before carrying out an executive design of the rig.

As previously described, the easiest way to test turbochargers is based on the use of a commercial rig like the Austrian AVL [1] or the German FEV [2]. The core device of these systems is a combustion chamber where hot gas to be sent to the turbocharger are produced with the help of a diffusion flame, and pressurized by a separate compressor.

The design of the latter type of rig begins with the definition of the range of the most important thermo-fluid dynamic characteristics of the hot gas fed to the test article, that is, the mass flow rate $\dot{m}_g$, the temperature T_g and the pressure p_g. Obviously, the maximum thermal power and fuel mass flow rate $\dot{m}_f$ mainly depend on the choice of the ($\dot{m}_g$, T_g) pair, and they have to comply with the thermal and mechanical resistance of the materials. Generally the gas flow rate is provided by a speed controlled compressor, while the hot gas temperature is controlled by different air/fuel settings.

While simple to build and rather diffused in the scientific community, this type of rig has some severe limitations. First of all, the thermo-fluid dynamic characteristics of the hot gas entering the test section are difficult to be selected independently. Then, at every operating condition a minimum gas temperature must be guaranteed to allow for a regular combustion process. Moreover, the gas pressure and flow rate are steady in nature and therefore they are far from the typical operating conditions of a turbocharger. Finally, from a safety point of view a combustion chamber requires the adoption of special and costly measures.

For these reasons a different methodology was followed during the early design stages of the present rig and several objectives were looked for, namely:

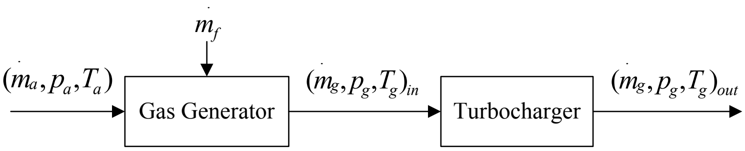

Fig. 1 – Conceptual hot gas generator layout

- independent choice of the mass flow rate, temperature and pressure of the hot gas,
- temporal modulation of the above characteristics,
- complete control and ease of operation of the rig in a fully automated fashion.

On account of the above requirements it was quickly concluded that the best solution was provided by an Internal Combustion Engines operated through a dedicated control strategy.

While gasoline engines provide higher exhaust gas temperature compared to diesel engines, the latter has been preferred on the basis of safety consideration as well as lower operating and maintenance costs. The gap in the turbine inlet temperature (TIT) can be filled with the adoption of an after-burner to be inserted on the exhaust manifold and taking advantage of the lean exhaust gas mixture. Such a device will enhance the gas generator flexibility offering the possibility to adjust the TIT according to the test article requirements, a feature that would be lost adopting a gasoline engine as gas generator.

Therefore the gas generator device is based on a modern light-duty turbocharged, direct injection diesel engine which has been deprived of its turbocharger to preserve the exhaust gas energy content and has been externally supercharged with an ad-hoc designed compression station, ensuring the attainment of the original performance of the ICE, in terms of brake mean effective pressure (BMEP). Once more to secure the largest possible rig flexibility, the compression station has been conceived as a truly independent (from the ICE) system with an autonomous mechanical driving unit.

3. The rig layout

The experimental facility is located in the laboratories of the DII – Dipartimento di Ingegneria Industriale of the Università di Napoli Federico II. The lab is newly built and complies with the most actual standards for safety, pollution control and energy saving. The turbocharger test rig can be considered consisting of four main sections, namely a cold gas generator (CGG), a hot gas generator (HGG), a test section (TS) and a data acquisition and process control unit (DCU) (see Fig. 2).

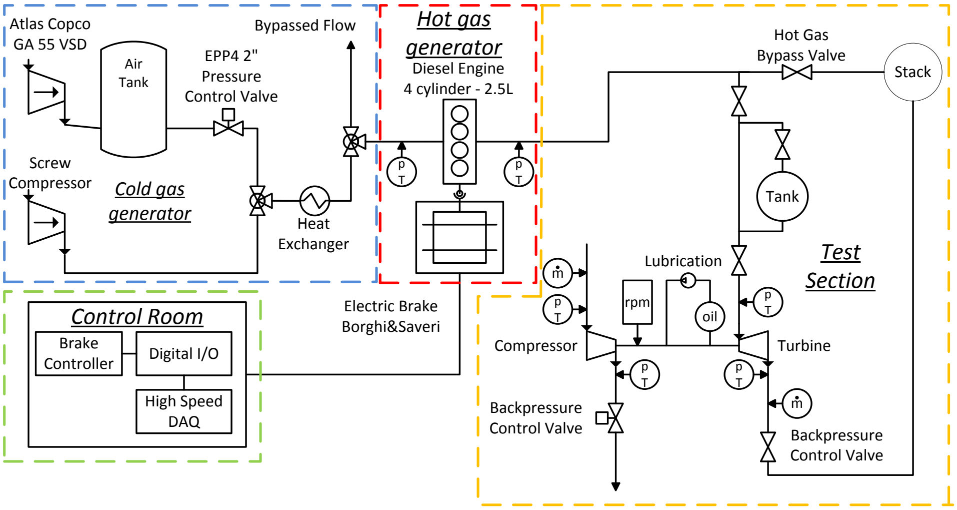

Fig. 2 – The test rig general layout

3.1. The cold gas generator

As mentioned in §2, the CGG should provide the ICE with the requested air flow rate at the appropriate boost pressure and temperature levels. Those data were collected through a detailed experimental campaign carried out on the turbocharged engine, and served as design target for the CGG conception. The CGG control procedure will be dealt with in §4.

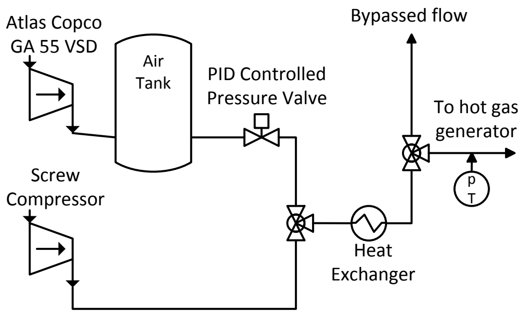

Fig. 3 – The cold gas generator layout

The core of the cold gas generator device consists of two medium size screw compressors, viz. a two stage, intercooled Atlas-Copco GA55VSD and a one stage all-purpose special screw compressor. The Atlas-Copco is of the oil injected type and disposes of a post processing unit separating the lubricating oil of the compressed air, and of a 2000 liters reservoir used as capacitive unit. At full power (55 kW) the two intercooled stages can deliver up to 3500 kg/h at a maximum gauge pressure of 12 bar. The station is operated with a closed loop control system setting the appropriate rpm value of the variable speed AC driving motor as to match the target flow rate.

Downstream the reservoir a PID controlled pressure regulator is located. This device sets the boost pressure level requested by the HGG and it is able to cover a wide range of operating conditions with an high frequency response. Input data to the CGG control system come from the HGG control system.

The all-purpose special screw compressor is driven by a 37 kW AC electric motor and operated at variable speed through a frequency converter. It can deliver up to 600 kg/h at maximum pressure of 2.5 bar. Similarly to the Atlas-Copco, the compressor is operated with a closed loop control strategy in order to match the target pressure and flow rate as requested by the HGG control system.

When the hot gas generator is disabled, the test section can be directly fed by the CGG still offering a remarkable enthalpy drop to the turbine. In these circumstances, due to the low temperatures reached during expansion, a dehumidifier should be used to prevent condensation.

3.2. The hot gas generator

As mentioned in §3 the hot gas generation is granted by a direct injection light duty diesel engine which has been deprived of its turbocharger to maximize the turbine inlet temperature and pressure. The ICE power is delivered to a Borghi&Saveri FE 350SA water cooled eddy current brake which has been specifically designed for reliable transient and steady long time operation. The brake whose maximum speed, torque and power are 10000 rpm, 1500 Nm and 295 kW respectively, is equipped with a digital controller and permits accurate, smooth and pre-defined braking at constant or variable speed. When operated at constant torque (resp. speed) the accuracy is better than 0.5% FS (±3 rpm). The brake cooling water is treated by a 240 kW cooling tower operating in closed loop, benefitting of a 10 m^3 storage tank.

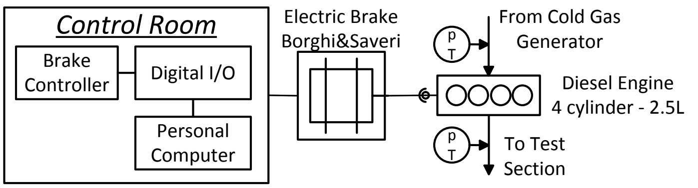

Fig. 4 – The hot gas generator layout

3.3. The test section

The test section is physically separated from the HGG to avoid heat transfer interference between the turbocharger and the HGG, and it can accommodate turbochargers of different sizes, the largest dimension being dictated by the HGG characteristics. The TC compressor and turbine are inserted in two separate circuits each of which is equipped with the necessary instrumentation. The turbine inlet piping connects the HGG exhaust manifold to the turbine inlet flange through a double branch thermally insulated circuit. A 150 liters cylindrical reservoir is installed on one of the two branches in order to dump out the pressure oscillation produced by the HGG, if desired. A set of automated remotely controlled valves determines the quality and amount of flow the turbine is fed with. More precisely, a by-pass and a backpressure valve define the flow rate and the expansion ratio seen by the turbine for each prescribed value of the ICE BMEP. The turbine outlet piping connects the turbine outlet flange to the stack, a 300 mm large, 20 m long insulated stainless steel circular duct equipped with an extraction centrifugal fan so that the whole exhaust pipe is slightly below atmospheric pressure. On the compressor side, the suction and delivery pipes are connected to the test cell and to the stack respectively, in an open loop scheme. A closed loop configuration, typically adopted to enlarge the compressor operating envelope, can easily be implemented. The turbocharger is lubricated and cooled by means of a closed loop lubrication system. The circuit is supplied with a 10W40 synthetic oil from the discharge side of a volumetric pump and it is then cooled before being returned to a 30 liters reservoir. The volumetric flow rate is controlled through a bypass system on the basis of the pressure signal collected by means of a Druck PTX-600 transducer located on the pump delivery flange, and sent to the DCU.

Performance of both compressor and turbine are assessed through direct real time measurements in terms of the relevant thermo-fluid dynamic properties: inlet-outlet temperatures and pressures, mass flow rates and TC rotational speed. Temperatures on the cold and hot manifolds are measured with either type K thermocouples or PT-100 thermo-resistances with range and accuracy of 0-1260 °C (type K), 0-350 °C (PT-100), ±2.8 °C (type K), better than 0.25% FS (PT-100), respectively.

Mean pressures in a 0-6 kPa range are measured with silicon deformable membrane transducer with an accuracy better than 0.25% FS BSL. Two high-frequency response Kulite WCT-312M pressure transducers, with a maximum frequency response of 72kHz are used for transient analysis.

Mean flow rates on the hot and cold sides are measured with two Rosemount 3095MFA pressure based meters with an operative range of 900kg/h (scalable up to 5000 kg/h with different sensor housing) and accuracy better than 0.8% FS BSL. Time dependent flow rates on the cold side are measured with a Bosch HFM-5 hot film meter with an accuracy better than 3% FS and a frequency response of 30Hz.

The turbocharger rotational speed is acquired with an eddy current sensor flush mounted on the compressor casing near the impeller inlet shroud, detecting the blade passing frequency corresponding to a maximum rotational speed of 200000 rpm.

All transducers signals are routed into the control room via cables electrically and magnetically insulated to limit the external noise thus maximizing the signal to noise ratio.

The hardware and software devices, as well as the brake controller, are managed with a high speed data acquisition system collecting all transducer signal digitally. The rig control parameters are handled with the help of several dedicated I/O modules such as the National Instruments PCI-6133 Data Acquisition and the cFP-1808 Programmable Automation Controller units, also accounting for safety issues. More precisely, the PCI-6133 and the cFP-1808 deal with the high frequency response and the steady state signals, respectively. Finally, all I/O hardware is controlled via a Virtual Instrument home-developed code running under LabVIEW standard, that permits to acquire, control and save all the operative parameters of the rig in a nearly automated manner.

4. The rig control strategy

One of the key point to address when operating the HGG is related to the supercharging strategy to adopt in order to preserve and/or maximize the enthalpy drop the turbine can take advantage of. The CGG relying on the two compression systems (the general purpose screw compressor and the Atlas Copco GA55VSD) described in §3.1 is used to supercharge the ICE following an ad-hoc designed control strategy run by the DCU.

When the general purpose screw compressor serves as CGG, the DCU generates a DC signal which is sent to the frequency converter, setting the desired speed of the electrical motor whose maximum power is 37kW (see Fig. 5).

By so doing different operating conditions of the compressor and therefore different boost pressure levels can be defined. In other words changing the AC motor rpm it is always possible to match exactly the engine supercharging requirements through a single control parameter.

Different operating modes are instead possible with the Atlas Copco GA55VSD compressor. This requires a more in-depth description of the system. The Ga55VSD screw compressor is a two-stage intercooled machine

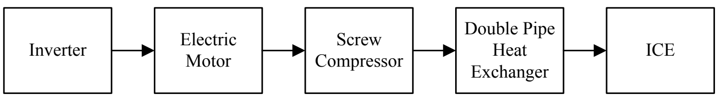

Fig. 5 – The general Screw compressor flow chart

powered by a 55kW variable frequency AC motor, whose delivery flange is connected to a 2000 liters reservoir.

The compressed air system control strategy is mainly aimed at defining the system dynamics, i.e. the changes in demand over time in the most efficient way. Here the primary concern is related to the capability of the system to meet the demand requirements promptly, without running into operating inefficiencies and/or malfunctioning. Therefore, part load operations are achieved through a high performance pressure regulator system which is fully controlled electronically by the DCU. The system is based on the Parker EPP4 architecture which is equipped with a programmable PID. The PID parameters are manipulated with a proprietary software and a USB connection in order to adjust the dynamic response of the system. Taken collectively the compression station can deliver any mass flow between 0 and 3500 kg/h at any pressure between 1 and 12 bars. A counter flow double-pipe heat exchanger is placed downstream the junction of the two compression station and it may be used to cool the compressed air using as cold sink the water flow rate processed by the cooling tower.

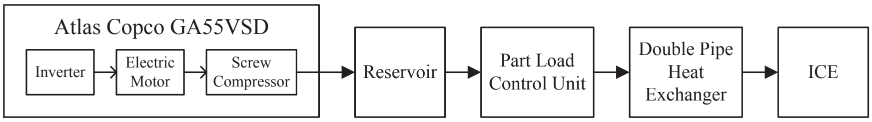

Fig. 6 – Atlas Copco GA55VSD screw compressor layout

5. Operating the rig

In this section we present a few relevant results concerning both the CGG and the HGG to demonstrate the potential of the proposed system. Before removing the turbocharger from the ICE, the latter has been characterized carrying out a series of dedicated extensive tests aimed at qualifying the performance of the turbocharging system, and more precisely of the compressor. The objective is to define the boost pressure level ensured by the turbocharger for each value of the mass flow rate swallowed by the engine. This is best understood through Fig. 7 where the operative envelope of the original compressor matched to the ICE is presented. It is worth noting that every $(\dot{m},\pi)$ pair lying outside the shaded area of Fig. 7 cannot be obtained with the original setup simply because they do not respect the turbocharger-ICE matching conditions. Fig. 8 presents the external supercharging conditions offered by the general purpose single stage screw compressor replacing the turbocharger compressor on the engine intake side. It can be seen that thanks to the variable speed operation allowed by the frequency converter, the original operating map of the engine, defined by the shaded area, is readily spanned. However, since the $(\dot{m},\pi)$ pairs on the operating map result from the compressor-ICE matching, and the controlling element to operate the compressor at the part-loads is its speed, it is difficult to set a pre-determined value of both $\dot{m}$ and π without resorting to a PID controller. This limitation is overcome by the Atlas Copco compressor, which, thanks to PLCU, induces the operating map presented in a Fig. 9. The advantages offered by the PLCU system are clearly enlightened by the Figure where new $(\dot{m},\pi)$ pairs lying outside the shared area are present.

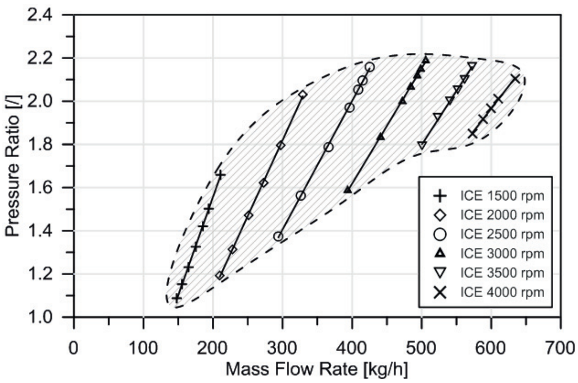

Fig. 7 – The original compressor envelope matched to the ICE

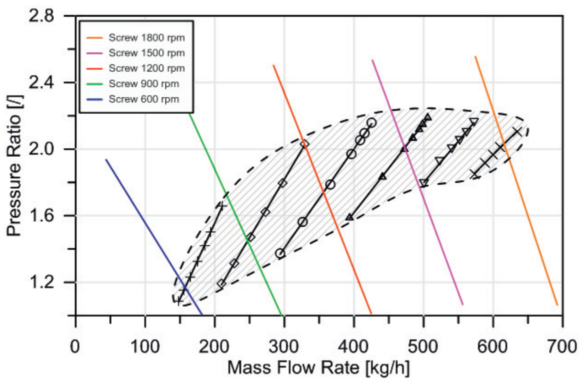

Fig. 8 – The general purpose screw compressor-Ice matching

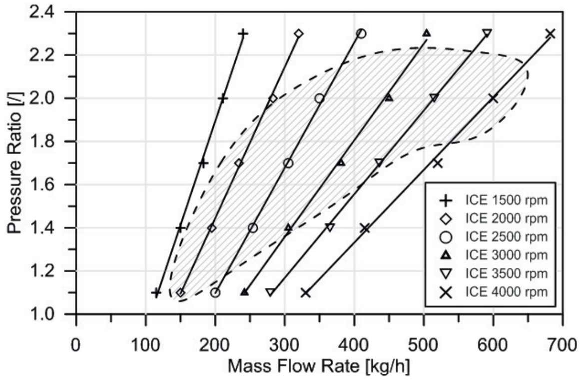

Fig. 9-The Atlas Copco extended operating envelope

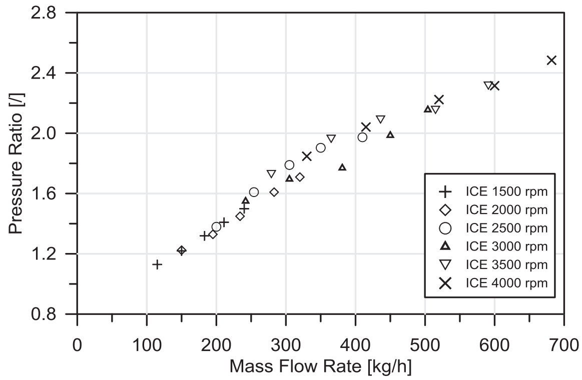

Fig. 10- The hot gas operating plane at 30% ICE load

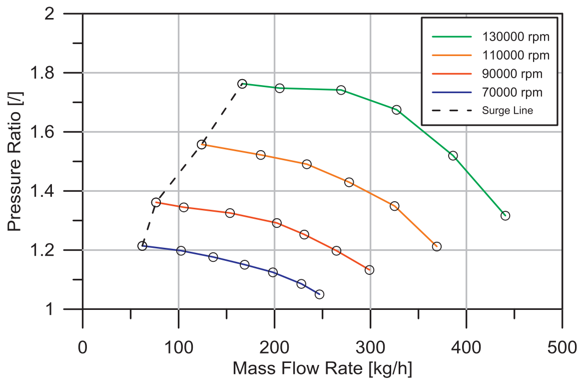

Fig. 11- Compressor performance map: pressure ratio

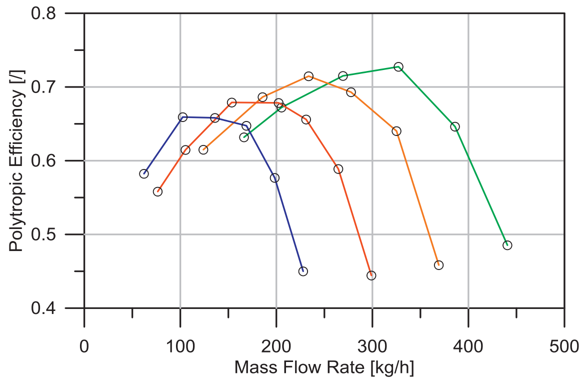

Fig. 12- Compressor performance map: polytropic efficiency

On the hot side the state space is three-dimensional, i.e. the hot gas characteristics are described in terms of $(\pi_g,\dot{m}_g,\theta_g)$ triplets, the turbine inlet temperature being a parameter depending on the ICE load. Fig. 10 reports the operating plane on the hot side obtained with a 30% load; higher pressure ratios for identical flow rates can be obtained increasing the engine load (results not shown).

Figures 11 and 12 show a typical steady state result obtained with the automated data acquisition system, reporting pressure ratio and polytropic efficiency vs. flow rate of the compressor of a medium size turbocharger for automotive applications. The dashed line in Fig. 11 denotes the occurrence of surge as detected by the high frequency pressure transducers and defined once the pressure signals begin to exhibit appreciable unsteadiness qualified in terms of its rms values.

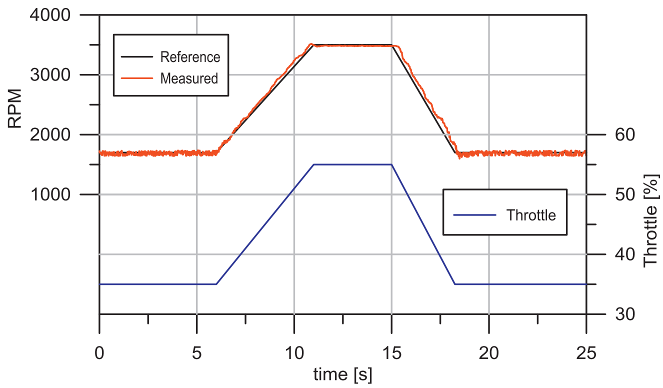

Fig. 13- The HGG transient maneuver

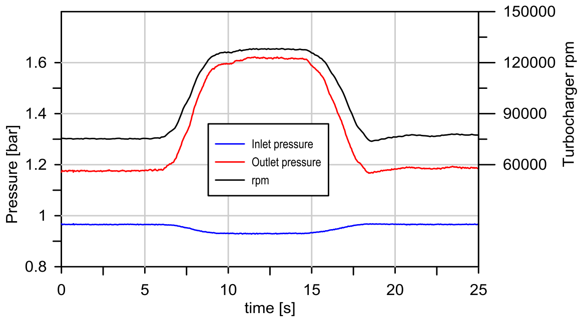

Fig. 14- Compressor pressure and rpm time histories during maneuver: 45% throttling

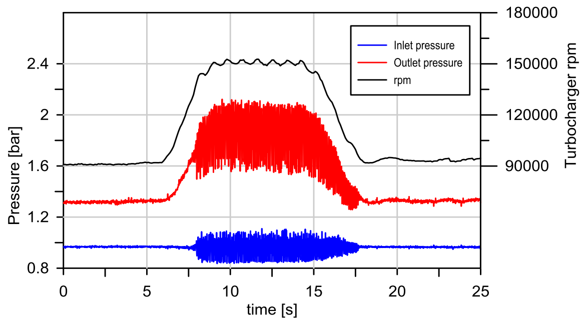

Fig. 15- Compressor pressure and rpm time histories during maneuver: 22% throttling

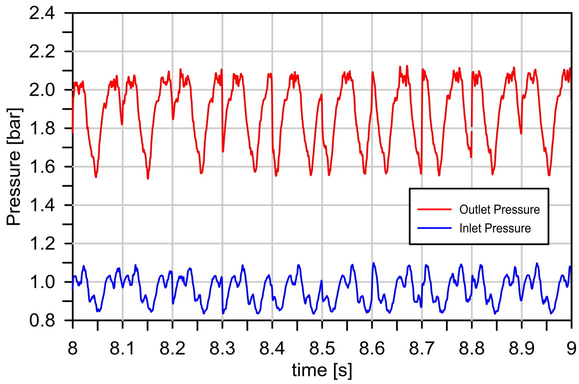

Fig. 16-Closed up view of Fig. 15

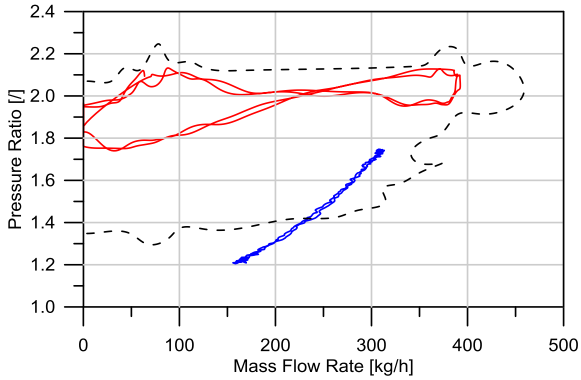

Fig. 17-Compressor operating line during maneuver: red line, 0.1s sampling , 22% throttling; blu line , 25s sampling, 45% throttling; dashed line envelope 25s maneuver

To demonstrate the potential of the rig in terms of its unsteady capabilities, we present in Figures 13-17 the effects of a simple test maneuver carried out on the HGG on the performance of the compressor. More precisely the ICE rotational speed and the brake torque are simultaneously changed in time as illustrated in Fig. 13 by the black (rpm) and blue (throttle) curves. More complex maneuvers are of course possible. The response of the system in terms of ICE rpm as measured by the eddy current encoder, showing a perfect correspondence with the imposed signal, is also reported on the same figure (red curve). The effects of the maneuver on the compressor performance are documented in figures 14 and 15 presenting the temporal traces of the suction and delivery pressure signals together with the turbocharger rpm for 45% and 22% backpressure valve throttling, respectively (100% corresponds to full opening). While for 45% throttling the pressure ratio exhibit a smooth behavior in time, in the 22% throttling case the situation is different and the occurrence of deep surge flow conditions are clearly visible. It is a periodic phenomenon with a neat dominant frequency of about 19 Hz which is best visualized in Fig. 16. In terms of global performance the effects of the two maneuvers are also reported in Fig. 17 in the $\dot{m}$-π plane. While for the 45% throttling the trajectory of the ($\dot{m}$,π) pair is practically a single-valued curve, in the 22% throttling case, the occurrence of deep surge flow conditions determines hysteresis cycles, whose envelope is described by the dashed line of fig. 10. Therefore quasi-steady approach would yield reasonable results in the first case only.

6. Conclusions

The paper has presented the conception, design and prototyping of a hot gas generation system for turbocharger experimental testing. Thanks to its flexibility, the HGG has been shown capable of delivering a wide range of flow rates with adequate pressure and temperature values allowing for an easy and complete spanning of the operating domain of the tested equipment with controlled accuracy. Steady experimental results of the compressor performance map have confirmed the quality of the control system of the air compression station serving as CGG. Unsteady results referring to a time varying maneuver obtained simultaneously changing the quality and quantity of hot gas sent to the tested article demonstrated the potential of the rig reporting accurate data of the compressor characteristic under deep surge flow conditions.

Acknowledgements

The authors are indebted with Cav. Gianpaolo Borghi of Borghi&Saveri for supporting this work with first class equipment.

References

[1] https://www.avl.com/turbo-charger-testing

[2] http://www.fev.com/what-we-do/engineering-services/turbocharging/

[3] Lujian JM, Bermudez V, Serrano JR, Cervellò C. Test bench for turbochargers groups characterization. SAE Paper 2002-01-0163

[4] Galindo J, Serrano JR, Guardiola C, Cervello C. Surge limit definition in a specific test bench for the characterization of automotive turbocharger. Elsevier Experimental Thermal and Fluid Science 30 (2006) 449-462

[5] Zinner K. Supercharging of internal combustion engines: fundamentals, calculations, examples. Springer-Verlag; 1978

[6] Watson N, Janota MS. Turbocharging the internal combustion engine. Macmillan; 1982

[7] Naundorf D, Bolz H, Mandel M. Design and implementation of a new generation of turbo charger test benches using hot gas technology. (2001) SAE Paper 2001-01-0279.

[8] Stemler E, Lawless P. The design and operation of a turbocharger test facility designed for transient simulation. (1997) SAE Paper 970344,

[9] Rajoo S, Romagnoli A, Martinez-Botas RF Unsteady performance analysis of a twin-entry variable geometry turbocharger turbine. Energy 38 (2012) 176-189

[10] Capobianco M, Marelli S. Experimental analysis of unsteady flow performance in an automotive turbocharger turbine fitted with a waste-gate valve. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering (2011) 225: 1087