Proceedings of the 1st Global Power and Propulsion Forum
GPPF 2017
Jan 16-18, 2014, Zurich, Switzerland
www.pps.global

GPPF-2017-136

COMPRESSOR SURGE FOR FULLY AND SEMI FLUCTUATING FLOWS IN AUTOMOTIVE TURBOCHARGERS

Calogero Avola
University of Bath
c.avola@bath.ac.uk
Bath, UK

Colin Copeland
University of Bath
c.d.copeland@bath.ac.uk
Bath, UK

Tomasz Duda
University of Bath
t.l.duda@bath.ac.uk
Bath, UK

Richard Burke
University of Bath
r.d.burke@bath.ac.uk
Bath, UK

ABSTRACT

Performance of radial compressors are gathered in maps correlating mass flow parameter, compressor speed, total-to-total pressure ratio and efficiency. In automotive turbochargers, a radial compressor is coupled to an expansion turbine via a common shaft. In these turbomachines, compressors performance are measured in steady flow gas-stand following specific code of standards where normalisation for intake pressure and temperature are applied. Compressor operations are delimited by surge and choke conditions due to instabilities onset. In this study, the focus is on the analysis of surge, corresponding to the low mass flow limit of the compressor. Surge is characterised by flow recirculation at the inlet, across the compressor wheel, leading to instability of the turbomachine due to extreme pressure and mass flow oscillations.

In order to adopt turbochargers with internal combustion engines, pressure and mass flow oscillations at turbine inlet and compressor outlet should be considered due to the fluctuation caused by the reciprocating engine. Therefore, a specific engine gas-stand has been developed for powering the turbine and generating real mass flow oscillations due to the motion of the engine exhaust valves. This is referred in the paper as semi fluctuating flow condition of the turbocharger, with turbine solely subjected to pulsating flows. In addition, the compressor is disconnected from the engine intake and the load is controlled by a back-pressure valve, to reflect semi fluctuating flow conditions. In this research study, the Fast Fourier Transform (FFT) and fluctuation of compressor outlet pressure and mass flow have shown the reduced effects of turbine pulsations on compressor stability. In order to capture the recirculation at the compressor inlet, FFT and temperature rise to the ambient intake temperature has been monitored through a thermocouple placed 25mm away from the compressor wheel.

Moreover, the map generated for compressor operating under steady flow with solely flow pulsation at the turbine inlet has been used as a benchmark to exploit surge behaviour of the compressor for fully fluctuating flows in turbochargers. In fact, the compressor outlet has been connected to the engine intake in parallel with the external boost rig. In this layout, the external boost has been supplied in order to control the load on the turbocharger by providing flow to the engine and throttling the compressor, simultaneously. Compressor surge has resulted different between the two configurations, extending to lower corrected mass flow values in fully fluctuating flows in the turbocharger. Investigation has been performed, evaluating compressor pressure and mass flow variations and compressor inlet temperature rise to the ambient conditions in the proximity of the compressor ducting.

INTRODUCTION

The demand for small and efficient powertrain architectures to power transportation vehicles has required the increase of power output from naturally aspirated internal combustion engines [1]. The inclusion of compressors to maximise deliverable engine torque and turbines to recover exhaust energy is a significant strategy towards fuel economy improvements of powertrains [2]. Due to the flexible operations of an internal combustion engine, turbochargers are requested to deliver high pressure ratios at changing air flow conditions using variable turbine and two stage technologies [3, 4]. Therefore, turbochargers selection is fundamental for power and efficiency optimisation.

In relation to the compressor characteristics, an extreme pressure ratio could be demanded at low mass flow, pushing the compressor to operate over an instable region where the compressor is not able to convert energy with adequate efficiency. This is represented by the surge limiting area of the compressor map, which causes the flow to recirculate at the compressor inlet [5]. In axial [6, 7] and radial [8] compressors, surge process has been modelled and analysed against experimental data [9], resulting in a system dependency process, inversely proportional to the Helmholtz resonator frequency [10]. The model has been implemented in steady [11] and unsteady [12] flow conditions and the importance of experimental data is necessary to define and represent the compressor surge behaviour.

The volume and size of downstream compressor influence the surge dynamics in relation to the operating mass flow [13]. The reduction of volume at the compressor outlet allows the measurement of close to zero mass flow conditions in the compressor performance map [14]. In addition, 1D compressor models have been able to simulate surge dynamics once specific characteristic factors of the system and compressor time delay are included [15]. The oscillation of mass flow and pressure ratio during compressor surge can be recorded in rising values of standard deviation from the mean value under steady flow conditions [16]. However, Fast Fourier Transform (FFT) of pressure signals are not suggested for pulsating flow application due to possible engine frequency interactions [16].

Furthermore, Galindo et al. [17] proved that pressure oscillation at the outlet of the compressor could extend the surge line towards lower mass flow conditions when compared to steady gas-stand measurements. Moreover, a similar approach has been adopted by Marelli et al. [18]. In this study, the stability range of the compressor differed between the steady map and the pulsating rig. However, no significant variation could be detected for the averaged surge line generated in the same rig when pulsation are introduced and the highest sampling frequency of 80Hz was adopted [18].

In the current research study, the compressor surge behaviour of an automotive turbocharger has been investigated in an engine gas-stand. In this layout, steady compressor performance can be monitored due to the absence of pulsating flows. Therefore, surge has been analysed as variation of pressure and mass flow. In addition, the air injection technique [19, 20] has been implemented for exploring compressor surge under fully pulsating turbocharger (compressor and turbine), as in a parallel two-stage turbocharging system [21]. In fact, a fixed boost supply could generate the necessary amount of compressor back-pressure, causing instabilities without reducing enthalpy at the turbine inlet. Conclusively, the temperature rise at 25mm from the radial compressor wheel in relation to the ambient intake conditions is captured by a 1.5mm K-type thermocouple, in order to improve the information on surge onset.

EXPERIMENTAL FACILITIES

The compressor behaviour is monitored for semi fluctuating flows, in the engine gas-stand layout of figure 1, and fully fluctuating flows , in the engine + boost rig arrangement of figure 2. In both approaches the same automotive turbocharger with variable turbine geometry has been investigated. In figure 1, the 49mm radial compressor has been mapped under steady conditions with a minimum amount of pulsation through the turbocharger shaft as highlighted by the FFT analysis in the results and discussion section. The rig allows controlled turbocharger operations via engine speed and boost, fuel flow, variable geometry (VGT) position and compressor back-pressure (CBP) [22].

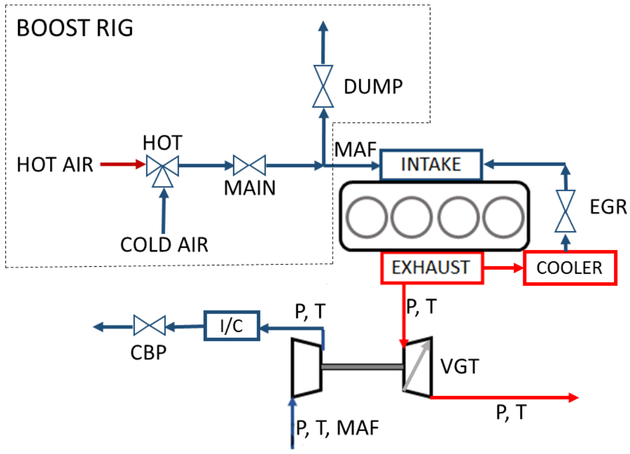


Figure 1. Engine gas-stand layout for semi fluctuating flow in turbocharger

In order to control compressor surge for complete powertrain systems, the boost supply has been connected in parallel to the compressor outlet ducts, including a non-return valve. In this schematic of figure 2, the external boost supply can increase load and reduce mass flow on the compressor, simultaneously. Due to boost limits of the engine intake, the compressor outlet pressure has to be restricted with a compressor restricting flow (CRF) valve. More importantly, the volume of the ducting at compressor outlet remains constant between the two layouts of figures 1 and 2 for a total of about $0.48m^3$, including the intercooler. This refers to ducting from compressor outlet to CBP in figure 1 and to three-way junction in figure 2. In particular, 51mm ducts have been adopted at compressor inlet and outlet.

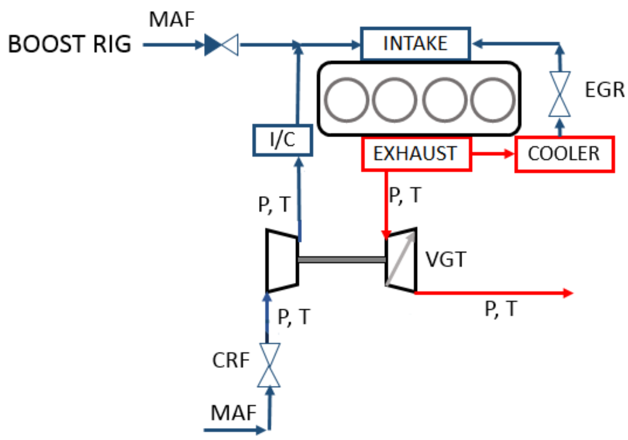


Figure 2. Engine + boost rig layout (air injection technique) for fully fluctuating flow in turbocharger

Pressure and temperature across the compressor is captured through averaging pressure rings and platinum resistance temperature (PRT) sensors, respectively. The positioning of the measuring points reflects the instructions given by ASME [23] and SAE [24] standards. In figure 3a, the averaging pressure ring for monitoring pressure at compressor inlet and outlet, as well as the turbine boundary conditions. In figure 3b, the PRT sensors positioned radially at the compressor outlet are shown. In this case, the four sensors tips are positioned at 1/2, 1/3 and 1/4 of the duct diameter. At the compressor inlet, two PRT sensors have tips protruding at 1/3 of the duct diameter. Information and accuracy on the complete list of sensors adopted in the engine gas-stand can be accessible in Avola et al. (2016) [22]. Specifically, the ducts from the turbocharger to the measurement sections are thermally insulated to reduce heat transfer.

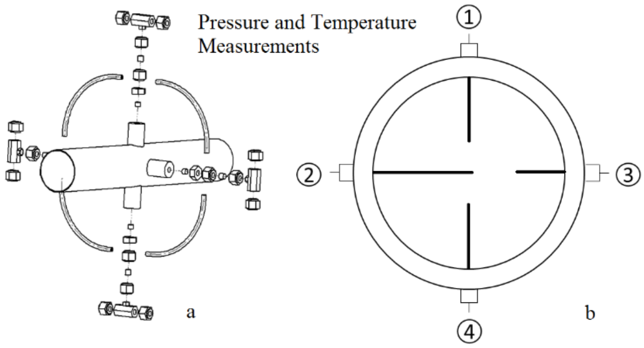


Figure 3. Pressure (a) and temperature (b) sensors adopted in the engine gas-stand to measure compressor performance

Experimental analysis

In particular, in both layouts, mass flow and pressure outlet signals are monitored to control surge onset and the elevated instabilities for the compressor. The maximum sampling frequency for the compressor mass flow sensor is 80 Hz, while the pressure sensor has a capability of 500 KHz. In addition, the acquisition system represents the real limit due to a resolution of 0.1 engine crank angle degrees (CAD), reducing the maximum sampling frequency for the pressure signal. In equations 1, 2 and 3, standard deviation σ, coefficient of variation c_v and maximum amplitude θ are shown and adopted for the investigation of the compressor surge dynamics.

$$\sigma = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(x_i - \mu)^2} \quad (1)$$

$$c_v = \frac{\sigma}{\mu} \quad (2)$$

$$\theta = (M - m) \quad (3)$$

In relation to the above equation, N is the total number of x_i samples, μ is the mean value, higher and lower values of the samples are indicated as M and m, respectively. Furthermore, FFT of compressor outlet pressure and mass flow are analysed. In particular FFT of compressor mass are limited to 40 Hz due to possible aliasing distortion [25].

RESULTS AND DISCUSSION

Compressor surge for semi fluctuating flows in turbocharger

The compressor map has been generated under steady flow conditions in the engine gas-stand facility and is visible in figure 4. Circle points represent stable operations at several corrected compressor speeds and surge points at three compressor speeds are also plotted. Therefore, operating the compressor at low mass flow causes recirculating flows at the inlet, resulting in instabilities as the surge loops of figure 4.

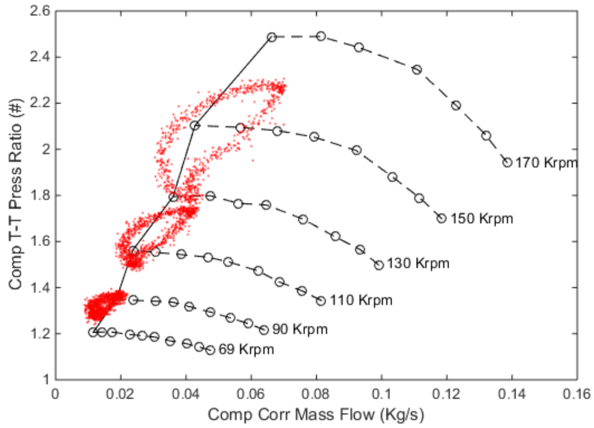


Figure 4. Compressor map and surge points for semi fluctuating flows in turbocharger

Surge loops at 69, 110 and 150 Krpm are shown, resulting in wide pressure ratio and mass flow oscillations. It is clear that the magnitude of oscillation increases directly with compressor speed for the pressure ratio and the corrected mass flow. The data points considered have a sampling frequency of 80 Hz due to limits on mass flow measurements. The average total-to-total pressure ratio and corrected mass flow are plotted in figure 5 as a result of 10 Hz, 80 Hz and higher frequencies for the pressure data. It is evident that the variation of frequency has small impact on the location of the surging conditions, suggesting that 10 Hz sampling frequency would be adequate for capturing flow and pressure oscillations. Therefore, a more accurate analysis of the experimental measurements is required to have a clear vision of the surge onset.

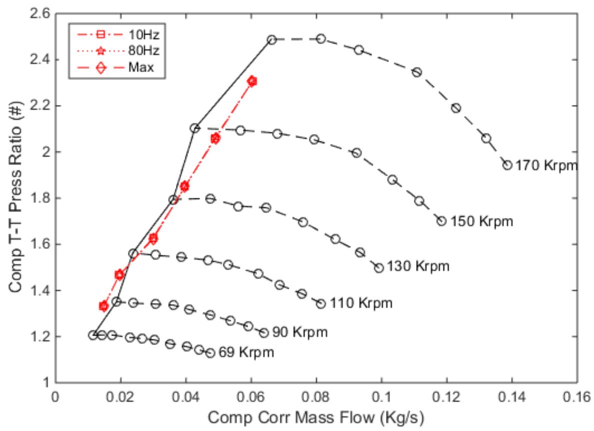


Figure 5. Influence of sampling frequency on the compressor surge points of semi fluctuating flows in turbocharger

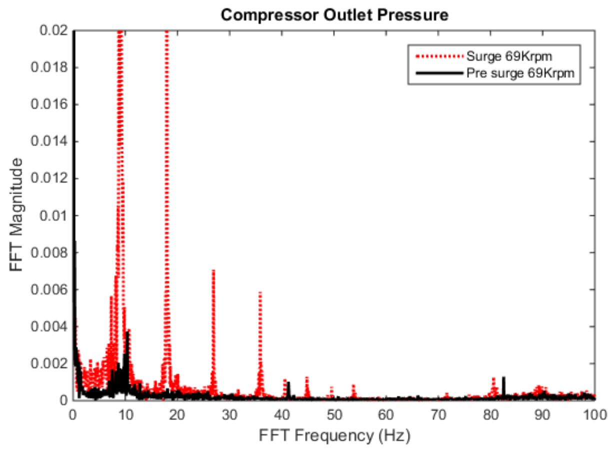
Pressure data investigation

As the acquisition system is dependant of the engine speed, surge at 69, 110 and 150 Krpm is captured at 75, 90 and 120 KHz, respectively. In table 1, standard deviation, coefficient of variation and maximum amplitude for compressor outlet pressure are considered at the last stable point before surge onset and at surge conditions. The point before surge is limited to the engine gas-stand operability. It can be noted that c_v values of about 0.6% are common for stable operations, instead of, a minimum of 2.6% for surging conditions. The trend of parameters at the three compressor speeds for the last stable point and the instable operation is similar. Therefore, monitoring the pressure signal at the compressor outlet could provide significant information on surge onset.

75-90-120 KHz	σ	c_v	θ
Pre Surge 69krpm	0.84 KPa	0.64%	19.2 KPa
Surge 69krpm	3.43 KPa	2.63%	26.4 KPa
Pre Surge 110krpm	1.04 KPa	0.62%	21.94 KPa
Surge 110krpm	8.6 KPa	5.44%	46.47 KPa
Pre Surge 150krpm	1.21 KPa	0.56%	23.42 KPa
Surge 150krpm	17.04 KPa	8.59%	68.16 KPa

Table 1. Standard variation, coefficient of variation and amplitude of compressor outlet pressure for semi fluctuating flows in turbocharger

As suggested by the FFT analysis of the pressure signal, pressure oscillations at around 10 Hz are present for surging conditions, as shown in figure 6. The last stable point recorded before surge shows a small increase of fluctuation in the frequency of about 10 Hz. In fact, while moving towards compressor surge conditions, amplitude of pressure oscillations at 10 Hz increases until the inoperability of the compressor is reached. Moreover, it is important to notice that, although, the turbine is subjected to pulsating flows, the FFT signals of figure 6 present insignificant peaks at 41.67, 50 and 66.67 Hz. These frequencies correspond to the motion of engine exhaust valves at the three different compressor speeds analysed in figure 6.



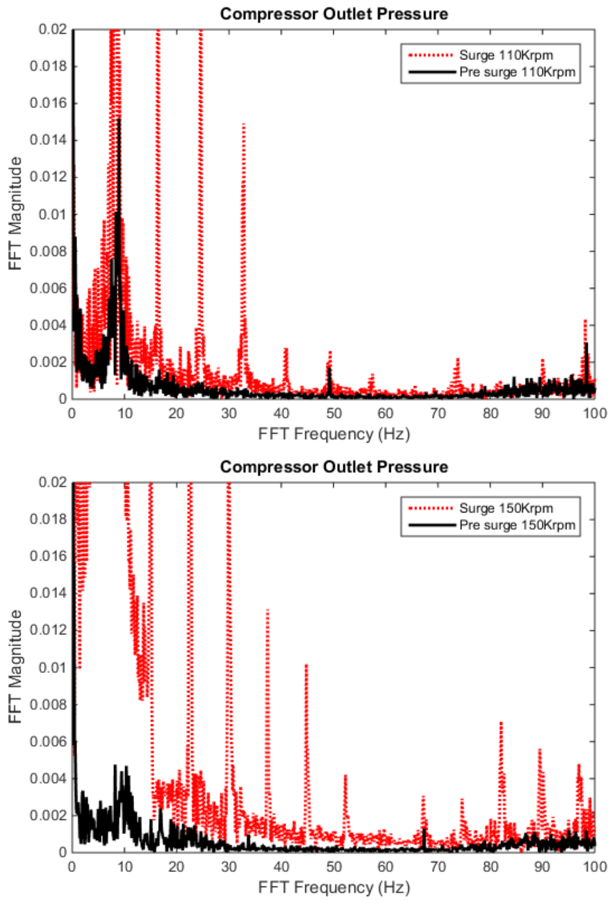


Figure 6. FFT of compressor outlet pressure for surge and pre surge at 69, 110 and 150 Krpm for semi fluctuating flows in turbocharger

Mass flow data investigation

The mass flow sensor at the inlet of the compressor has been considered for the definition of instabilities and surge onset. In fact, data in table 2 show variations higher than 20% in mass flow measurements at every compressor speed considered at instable conditions. Due to the maximum sampling frequency of 80 Hz, FFT magnitudes have been considered acceptable up to 40 Hz, in according with the Nyquist-Shannon sampling theorem [25]. In figure 7, the pre surge conditions are not able to capture significant peaks of FFT. This could be due to the wide distance between the mass flow sensor and the compressor wheel. Moreover, harmonics of the surging frequency event are recognisable in figure 7.

80 Hz	σ	c_v	θ
Pre Surge 69krpm	0.001 Kg/s	9.18%	0.006 Kg/s
Surge 69krpm	0.0032 Kg/s	21.61%	0.0132 Kg/s
Pre Surge 110krpm	0.0011 Kg/s	4.9%	0.0091 Kg/s
Surge 110krpm	0.0072 Kg/s	24.55%	0.0255 Kg/s
Pre Surge 150krpm	0.0011 Kg/s	2.55%	0.0052 Kg/s
Surge 150krpm	0.0116 Kg/s	24.56%	0.0398 Kg/s

Table 2. Standard variation, coefficient of variation and amplitude of compressor mass flow for semi fluctuating flows in turbocharger

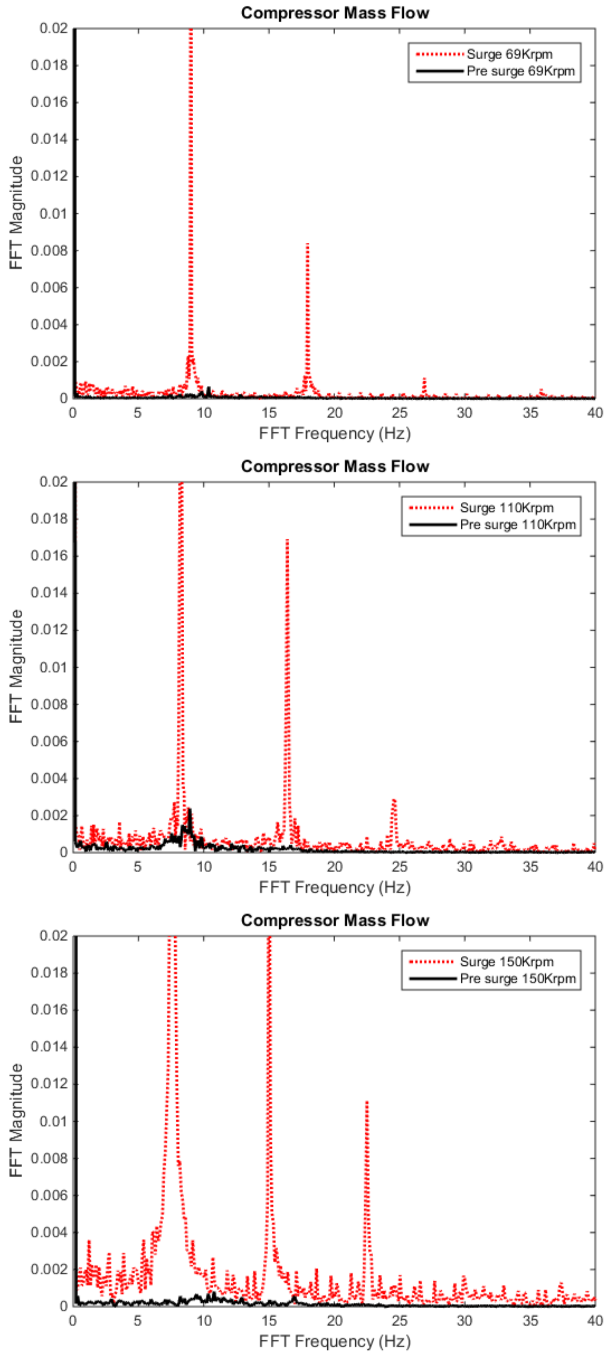


Figure 7. FFT of compressor mass flow for surge and pre surge at 69, 110 and 150 Krpm for semi fluctuating flows in turbocharger

Compressor surge for fully fluctuating flows in turbocharger

In order to study the characteristics of compressor surge for fully pulsating flows in turbocharger, including pressure pulsations at compressor outlet and turbine inlet. The engine gas-stand of figure 1 has been reconfigured in order to generate flow pulsations at the compressor outlet owing to the motion of engine intake valves. Therefore, in the layout of figure 2, the boost rig has been adopted to control load on turbocharger and engine intake pressure. In this scenario, a small operating area of the compressor in the proximity of surge has been investigated.

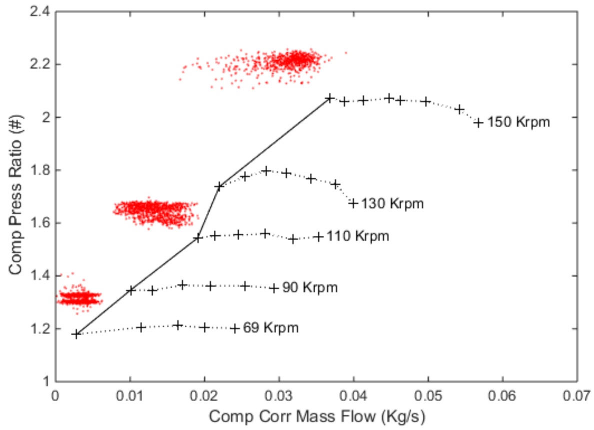


Figure 8. Compressor map and surge points for fully fluctuating flows in turbocharger

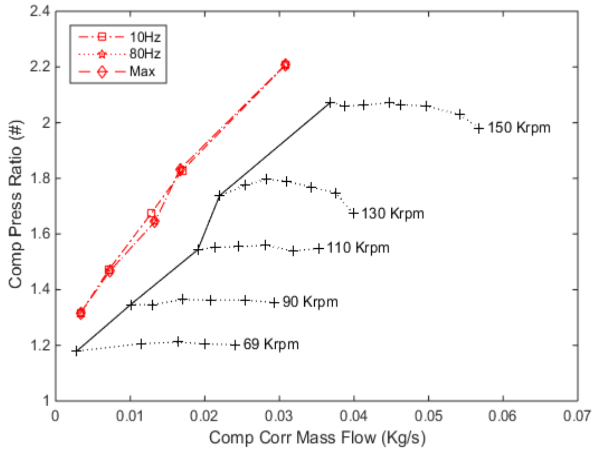


Figure 9. Influence of sampling frequency on the compressor surge points of fully fluctuating flows in turbocharger

In figure 8, crosses show the time averaged stable operations of the compressor and the cyclic surge data points at 69, 110 and 150 Krpm, laying outside the operating map. Experimental data of figure 8 and 9 are collected at different engine speeds in order to keep constant turbine inlet temperature (TIT). The intake valve motion causes pulsations at 41.67, 50 and 66.67 Hz for compressor speeds of 69, 110 and 150 Krpm, respectively. In addition, the sensitivity study on sampling frequency shows a small influence on the averaged surge points of the compressor map. In particular, it seems that an acquisition frequency of 10 Hz is less adequate to represent surge points under pulsating conditions, in according to figure 9, due to the variation at 110 Krpm.

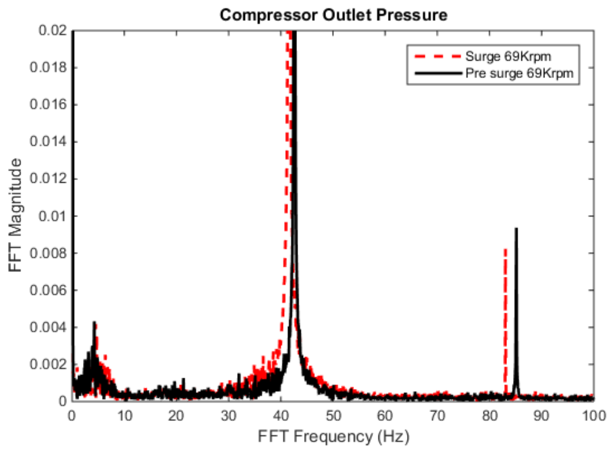
Pressure data investigation

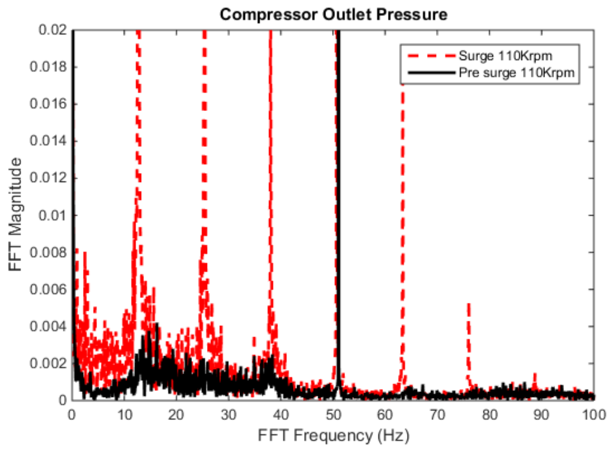
The identification of surge is extremely important in order to limit the compressor operations when connected to an internal combustion engine to avoid power and efficiency losses. Therefore, pressure oscillations at the compressor outlet have been controlled at surging and stable pre surging conditions. Unlike semi fluctuating flow conditions in turbocharger operations, c_v values of table 3 have resulted lower at surging conditions than in table 1. Moreover, it has resulted that surge onset would be difficult to capture at the compressor speed of 69 Krpm for the fully fluctuating turbocharger. In conjunction with the information provided by table 3, this is suggested by the FFT analysis of figure 10.

At 110 and 150 Krpm, surge frequencies have resulted slightly higher than under steady compressor operations in the FFTs. This can be correlated to the variation of compressor outlet ducts and volumes due to the presence of the boost rig. As well as at semi fluctuating flow conditions, the low frequency peaks increase while deteriorating the compressor stability for fully fluctuating flows in the turbocharger. Furthermore, engine intake valves motion causes significant pulsations of the compressor outlet pressure at fully fluctuating flow conditions. According to figure 10, these pulsations and surging flow have disparate frequencies.

75-90-120 KHz	*σ*	c_v	*θ*
Pre Surge 69krpm	2.21 KPa	1.73%	28.29 KPa
Surge 69krpm	2.28 KPa	1.78%	27.97 KPa
Pre Surge 110krpm	1.3 KPa	0.86%	22.78 KPa
Surge 110krpm	2.14 KPa	1.41%	25.21 KPa
Pre Surge 150krpm	0.96 KPa	0.5%	14.71 KPa
Surge 150krpm	2.1 KPa	1.09%	21.32 KPa

Table 3. Standard variation, coefficient of variation and amplitude of compressor outlet pressure for fully fluctuating flows turbocharger





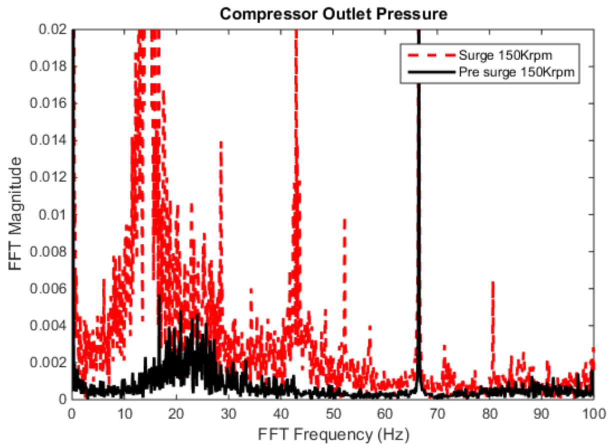


Figure 10. FFT of compressor outlet pressure for surge and pre surge at 69, 110 and 150 Krpm for fully fluctuating flows turbocharger

Mass flow data investigation

Dissimilarly to pressure data in table 3, compressor mass flow variations of table 4 are more severe at 110 and 150 Krpm when compressor is under surge. In addition, a clearer distinction between stable and unstable conditions could be made. Additionally, a difficult indication of surge is existent for the lower compressor speed analysed. Due to the 80 Hz sampling rate of the mass flow sensor, FFTs of compressor mass flow are interested to possible aliasing distortion for flow variation due to engine intake. For the expressed reason, the analysis has not been found of high relevance.

80 Hz	*σ*	c_v	*θ*
Pre Surge 69krpm	0.001 Kg/s	31.33%	0.0064 Kg/s
Surge 69krpm	0.001 Kg/s	68.74%	0.0056 Kg/s
Pre Surge 110krpm	0.001 Kg/s	6.03%	0.0057 Kg/s
Surge 110krpm	0.0023 Kg/s	18.33%	0.011 Kg/s
Pre Surge 150krpm	0.001 Kg/s	3.34%	0.0064 Kg/s
Surge 150krpm	0.0028 Kg/s	10.54%	0.0184 Kg/s

Table 4. Standard variation, coefficient of variation and amplitude of compressor mass flow for fully fluctuating flows in turbocharger

Effect of pulsations on compressor surge

The FFT analysis of compressor outlet pressure data has served for the distinction of stable and unstable conditions. This statement has had significant validity for both semi and fully fluctuating flow conditions. In fact, this is highlighted by the fact that surging frequencies appear at different rates compared to engine generated events, as the motion of intake valves. In this way, it is possible to monitor the surge onset and recirculating flows at the compressor inlet. Due to the better controllability of the system of figure 2, it has been possible to reduce compressor mass flow intervals between fixed operating points for the fully fluctuating flows in turbocharger. In addition, the presence of pulsations at compressor outlet has varied the stability range of the compressor before surge, as shown in figure 11.

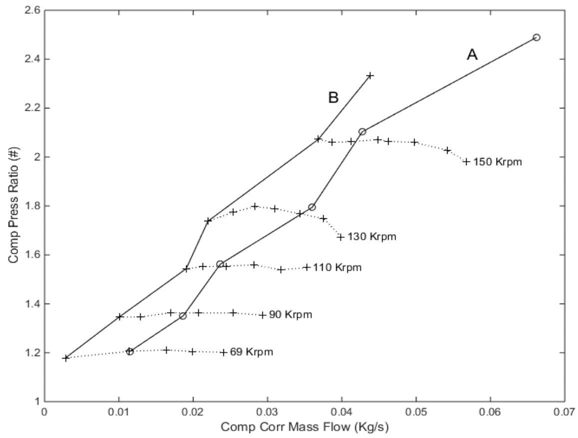


Figure 11. Last stable points before compressor surge for semi (A) and fully (B) fluctuating turbocharger

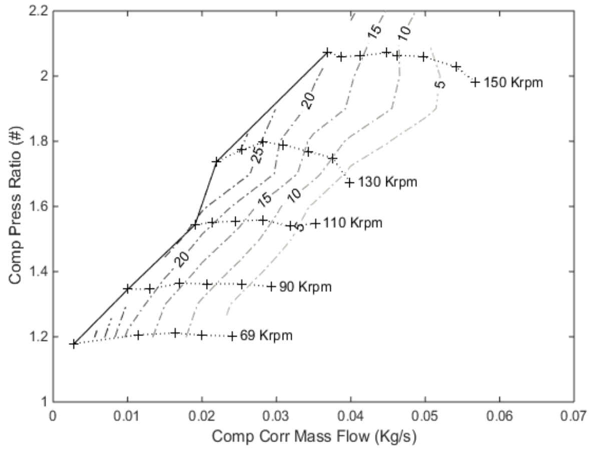


Figure 13. Temperature rise (degC) at 25mm from compressor wheel to main intake conditions for fully fluctuating flows in turbocharger

Inlet temperature analysis

In order to support the identification of surge and the proximity to instable operations, a temperature rise investigation on the compressor inlet temperature has been performed. A 1.5mm thick k-type thermocouple placed 25mm away from the compressor wheel is measuring the flow temperature at a depth of half the compressor blades. The temperature measured at the compressor inlet is compared to the ambient intake temperature conditions. Although, the thermocouple is kept at the same position for semi and fully fluctuating flow conditions in the turbocharger, temperature rise has been higher with the presence of pulsating flows at the compressor outlet, as it can be noted in figures 12 and 13.

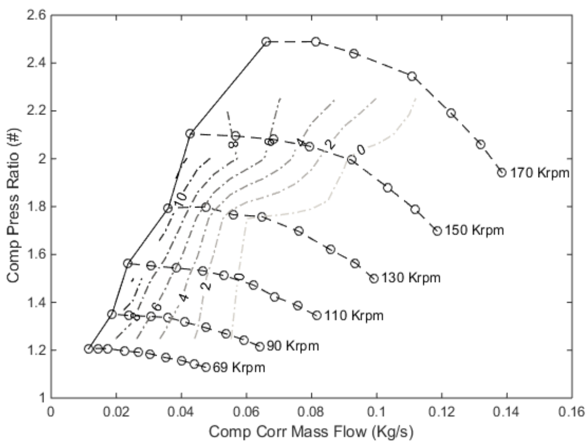


Figure 12. Temperature rise (degC) at 25mm from compressor wheel to main intake conditions for semi fluctuating flows in turbocharger

In figure 13, the higher temperature rise recorded at the compressor inlet could suggest that a larger flow recirculation is allowed in the presence of full turbocharger fluctuations. In fact, the presence of steady flows at the compressor seems to restrict the stability range at low corrected mass flow values. Furthermore, it is important to consider that the compressor outlet volume has been slightly increased in order to accommodate the boost rig supply and generate information for figure 13. According to Hansen et al. [8], this should worsen the compressor surge tolerance in comparison to the semi fluctuating flows setting. It seems that the effect of pulsating flows has a significant benefit in improving tolerance to compressor surge.

CONCLUSIONS

The analysis of surge onset for an automotive compressor has been performed for semi and fully fluctuating flows in the turbocharger. Compressor surge and operations have been analysed at steady and pulsating flows at the compressor outlet, while the turbine is fed with oscillating flows. In the first condition, the compressor load has been controlled through a back-pressure valve. In the second pulsating flow approach, an external boost supply has allowed the monitoring of surge onset. Therefore, the research investigation has allowed the following findings:

- Turbine pulsations have a small influence on the compressor map and surge onset. A steady flow assumption on the compressor could be made for semi fluctuating turbocharger.
- Variations of compressor outlet pressure and mass flow could equally identify surge onset in the presence of solely turbine pulsations. The pressure c_v shifts clearly from 0.6% to a minimum of 2.6% for surging conditions and a θ minimum value of 26 KPa is recorded at surge. Additionally, a c_v higher than 20% represents instable conditions of the compressor mass flow.
- In FFTs of compressor outlet pressure under semi fluctuating flows, a clearer rising peak at about 10 Hz is visible in the proximity of surge

and it is well distinguishable at surging flows. The development of surge is not as clear in the mass flow FFTs.

- The fully fluctuating flow conditions allow the compressor to extend the operability area at lower mass flow and the FFT of pressure becomes extremely important for identifying surge onset. Amplitude peaks at about 10 Hz are visible in figure 10, apart from the 69 Krpm case. This is not possible for mass flow measurements in the specific implemented layout.
- The temperature rise at the compressor inlet has provided useful information on the surge tolerance increment owing to the higher temperature recorded for pulsating flows at the compressor. Moreover, engine frequencies are different from surging phenomena.

NOMENCLATURE

CAD	Crank Angle Degree
CBP	Compressor Back-Pressure
CRF	Compressor Restricting Flow
c_v	Coefficient of variation
EGR	Exhaust Gas Recirculation
FFT	Fast Fourier Transform
I/C	Inter-cooler
M	Higher sample value
m	Lower sample value
MAF	Mass Air Flow
N	Total samples
P	Pressure
PRT	Platinum Resistance Temperature
T	Temperature
T	Temperature
TIT	Turbine Inlet Temperature
VGT	Variable Geometry Turbine
x_i	Sample
θ	Maximum amplitude
μ	Mean value
σ	Standard deviation

ACKNOWLEDGMENTS

The Authors would like to acknowledge the technical staff at the Powertrain and Vehicle Research Centre for the support received in implementing the experimental facility.

REFERENCES

[1] Turner, J. W. G., Popplewell, A., Patel, R., Johnson, T. R., Darnton, N. J., Richardson, S., Bredda, S. W., Tudor, R. J., Bithell, C. I., Jackson, R., Remmert, S. M., Cracknell, R. F., Fernandes, J. X., Lewis, A. G. J., Akehurst, S., Brace, C., Copeland, C., Martinez-Botas, R., Romagnoli, A., and Burluka, A. A., 2014, "Ultra Boost for Economy: Extending the Limits of Extreme Engine Downsizing," SAE Technical Paper(2014-01-1185).

[2] Watson, N., and Janota, S., 1982, Turbocharging the internal combustion engine, The Macmillan Press Ltd, London.

[3] Tang, H., Pennycott, A., Akehurst, S., and Brace, C. J., 2014, "A review of the application of variable geometry turbines to the downsized gasoline engine," International Journal of Engine Research, 16(6), pp. 810-825.

[4] Avola, C., Copeland, C., Burke, R. D., and Brace, C. J., 2016, "Numerical Investiagtion of Two-Stage Turbocharging Systems Performance," ASME ICEF 2016, ASME, Greenville, SC, USA.

[5] Harley, P. X. L., Spence, S. W. T., Early, J., Filsinger, D., and Dietrich, M., 2014, "Inlet recirculation in automotive turbocharger centrifugal compressors," 11th International Conference on Turbocharger and Turbocharging, IMechE, London, pp. 89-100.

[6] Greitzer, E. M., 1976, "Surge and Rotating Stall in Axial Flow Compressors. Part I: Theoretical Compression System Model," ASME Journal of Engineering for Power, APRIL 1976, pp. 190-198.

[7] Greitzer, E. M., 1976, "Surge and Rotating Stall in Axial Flow Compressors. Part II: Experimental Results and Comparison With Theory," ASME Journal of Engineering for Power, APRIL 1976, pp. 199-211.

[8] Hansen, K. E., Jorgensen, P., and Larsen, P. S., 1981, "Experimental and Theoretical Study of Surge in a Small Centrifugal Compressor," ASME Journal of Fluids Engineering, 103, pp. 391-395.

[9] Fink, D. A., Cumpsty, N. A., and Greitzer, E. M., 1992, "Surge Dynamics in a Free-Spool Centrifugal Compressor System," Journal of Turbomachinery, 114(April 1992), pp. 321-332.

[10] Greitzer, E. M., Tan, C. S., and Graf, M. B., 2004, Internal flow: concepts and applications, Cambridge University Press, USA.

[11] Dehner, R., Selamet, A., Keller, P., and Becker, M., 2011, "Prediction of Surge in a Turbocharger Compression System vs. Measurements," SAE International Journal of Engines, 4(2), pp. 2181-2192.

[12] Theotokatos, G., and Kyrtatos, N. P., 2001, "Diesel Engine Transient Operation with Turbocharger Compressor Surging," SAE Technical Paper(2001-01-1241).

[13] Galindo, J., Serrano, J. R., Guardiola, C., and Cervelló, C., 2006, "Surge limit definition in a specific test bench for the characterization of automotive turbochargers," Experimental Thermal and Fluid Science, 30(5), pp. 449-462.

[14] Marelli, S., Carraro, C., Marmorato, G., Zamboni, G., and Capobianco, M., 2014, "Experimental analysis on the performance of a turbocharger compressor in the unstable operating region and close to the surge limit," Experimental Thermal and Fluid Science, 53, pp. 154-160.

[15] Galindo, J., Serrano, J. R., Climent, H., and Tiseira, A., 2008, "Experiments and modelling of surge in small centrifugal compressor for automotive engines," Experimental Thermal and Fluid Science, 32(3), pp. 818-826.

[16] Andersen, J., Lindström, F., and Westin, F., 2008, "Surge Definitions for Radial Compressors in Automotive Turbochargers," SAE Technical Paper(2008-01-0296).

[17] Galindo, J., Climent, H., Guardiola, C., and Tiseira, A., 2009, "On the effect of pulsating flow on surge margin of small centrifugal compressors for automotive engines," Experimental Thermal and Fluid Science, 33(8), pp. 1163-1171.

[18] Marelli, S., Carraro, C., and Capobianco, M., 2012, "Effect of Pulsating Flow Characteristics on Performance and Surge Limit of Automotive Turbocharger Compressors," SAE International Journal of Engines, 5(2), pp. 596-601.
[19] Galindo, J., Tiseira, A., Arnau, F. J., and Lang, R., 2013, "On-Engine Measurement of Turbocharger Surge Limit," Experimental Techniques, 37(1), pp. 47-54.
[20] Galindo, J., Arnau, F., Tiseira, A., Lang, R., Lahjaily PhD, H., and Gimenes Md, T., 2011, "Measurement and Modeling of Compressor Surge on Engine Test Bench for Different Intake Line Configurations," SAE Technical Paper.
[21] Chesse, P., Hetet, J. F., Tauzia, X., Roy, P., and Inozu, B., 2000, "Performance Simulation of Sequentially Turbocharged Marine Diesel Engines With Applications to Compressor Surge," ASME Journal of Engineering for Gas Turbines and Power, 122, pp. 562-569.
[22] Avola, C., Dimitriou, P., Burke, R., and Copeland, C., 2016, "Preliminary DoE Analysis and Control of Mapping Procedure for a Turbocharger on an Engine Gas-stand," ASME Turbo EXPO 2016, ASME, Seoul, South Korea.
[23] ASME, 1997, "Performance Test Code on Compressors and Exhausters," ASME Standards.
[24] SAE-International, 1995, "SAE J1723 Supercharger Testing Standard," Society of Automotive Engineers, Inc.
[25] Lee, E. A., and Varaiya, P., 2011, Structure and Interpretation of Signals and Systems, LeeVaraiya.org.