UNSTEADY FLOW TURBINE PERFORMANCE IN TURBOCHARGED AUTOMOTIVE ENGINES

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ABSTRACT

Turbocharging is becoming a key technology for both Spark Ignition (SI) and diesel automotive engines. As regards gasoline engines, turbocharging can help to reduce CO\textsubscript{2} emissions when used in conjunction with other technologies, such as engine downsizing, direct injection and variable valve actuation, all managed on the basis of appropriate control strategies.

However, the successful application of turbocharging to SI engines must address different problems related to the specific operating environment (exhaust gas temperature level) and to functional aspects (torque curve configuration, transient response). Research work on this subject is required, particularly as regards the unsteady flow operation typical of automotive applications. To this end, a great deal of information can be provided by measurements made on dedicated test facilities as these allow the correlations between unsteady flow characteristics and turbocharger behaviour to be investigated over a broad range without the operating restrictions imposed by the engine.

The intake and exhaust components test rig operating at the University of Genoa can be used to investigate automotive turbochargers under steady and pulsating flow conditions, analysing the effect of unsteady flow parameters, circuit geometry and exhaust valve actuation strategies on turbine performance.

This paper presents the results of an extensive investigation performed on small turbocharger turbines for gasoline engines, focusing on unsteady flow operation. Initially, the assessment of turbine unsteady flow performance is deepened as regards the evaluation of available energy at the turbine inlet and the procedures to calculate turbine efficiency. The influence of the main pulsating flow parameters on the amount of available energy at the turbine inlet is then analysed, taking into account both theoretical and experimental waves. Lastly, the results of an experimental test programme investigating instantaneous turbine mass flow behaviour and the efficiency levels provided by various different procedures are presented.
### NOMENCLATURE

#### Notations
- \( c \): specific heat
- \( f \): pulse frequency
- \( h \): specific gas enthalpy (referred to mass unit)
- \( k \): specific heat ratio
- \( n \): rotational speed, number of time intervals
- \( p \): pressure
- \( t \): time
- \( M \): mass flow rate
- \( P \): power
- \( T \): temperature, pulse period
- \( \alpha \): waste-gate opening (driving shaft angle)
- \( \epsilon \): turbine expansion ratio
- \( \eta \): efficiency
- \( \Delta \): amplitude, difference
- \( \Phi \): pulse duty cycle

#### Subscripts
- \( 0 \): ambient condition
- \( 3 \): turbine inlet
- \( 4 \): turbine outlet
- \( a \): apparent
- \( i \): referred to time interval
- \( m \): mean
- \( p \): constant pressure
- \( r \): referred to actual process
- \( s \): referred to isentropic process
- \( t \): turbine
- \( NSF \): non steady flow value
- \( QSF \): quasi-steady flow value
- \( SF \): steady flow value

### INTRODUCTION
Increasing energy efficiency has recently become a priority in automotive applications due to the rising cost of fuel and the need to reduce CO\(_2\) emissions. This will require the development of high efficiency powertrain systems whilst complying with near-zero pollutant emissions and customer demand for excellent vehicle performance and driveability (Bandel et al., 2006).

The automotive diesel engine has been substantially improved in recent years due to the introduction of various breakthrough technologies, especially electronically-controlled fuel injection systems and advanced turbocharging circuits (usually fitted with variable geometry turbines). As a result, modern diesel engines for car applications have achieved high specific power levels with low fuel consumption. However, a further reduction in exhaust emissions (especially nitrogen oxides and particulate) in order to comply with stricter future legislation may be a difficult task requiring a great deal of research and probably generating considerable increases in both the cost of engines and fuel consumption.

On the other hand, the spark ignition (SI) engine must significantly reduce fuel consumption (especially at part load operation) in order to achieve CO\(_2\) emission targets, whilst maintaining the advantages of a conventional aftertreatment process, high power output and excellent vehicle...
driveability. In order to achieve this goal, a global approach should be used by integrating both available and innovative technologies managed on the basis of appropriate control strategies. Downsizing is a promising means of improving SI engine efficiency as it allows engine operation to be shifted towards higher loads where its efficiency is typically better. A potential efficiency increase of between 10 and 30 percent can be achieved if the reduction in engine displacement is associated with the application of different technologies, such as charge boosting, gasoline direct injection (Lecointe and Monnier, 2003) and variable valve actuation (Denger and Kapus, 2002). Since very high specific performance should be reached by downsized engines, a mechanically driven compressor is too power consuming and is incompatible with the low fuel consumption target. As a consequence, turbocharging seems to be the best solution for this application. In any case, the trend towards a more complex configuration of the engine intake and exhaust circuit is evident, requiring suitable subsystem control strategies to be developed.

In this context, turbocharging is becoming a key technology for both gasoline and diesel automotive applications. However, successful application of exhaust turbocharging to downsized SI engines needs to solve various problems, related both to the specific operating environment (exhaust gas temperature level) and to engine performance, focusing on low-end-torque and transient response (Wirth et al., 2000; Lake et al., 2004; Petitjean et al., 2004). Dedicated investigations into the turbocharging system are therefore required in order to obtain better knowledge of its performance, particularly in the typical unsteady flow conditions occurring in the exhaust circuit of automotive engines. To this end, measurements performed on fully flexible test facilities are particularly useful as they allow to work independently of the engine, focusing on different aspects, such as the effect of the main pulsating flow parameters on turbine performance, the development of suitable comparison criteria between steady and unsteady flow results and the validation of results provided by theoretical models and quasi-steady flow prediction procedures (Winterbone et al., 1991; Szymko et al., 2005; Costall et al., 2006).

An intake and exhaust components test rig is available at the University of Genoa, which allows to investigate automotive turbochargers under steady and pulsating flow conditions. This experimental facility was recently upgraded as regards unsteady measuring capabilities (Capobianco and Marelli, 2006) in order to analyse both the effect of the main pulsating flow parameters on turbine performance and engine exhaust subsystem behaviour.

This paper presents the results of an extensive investigation performed on small turbocharger turbines for gasoline engines, focusing on unsteady flow operation. Initially, turbine unsteady flow performance is analysed as regards the evaluation of available energy at the turbine inlet and the procedures to calculate turbine efficiency. The influence of the main pulsating flow parameters on the amount of available energy at the turbine inlet is then analysed, taking into account both theoretical and experimental waves. Lastly, the results of an experimental test programme investigating instantaneous turbine mass flow behaviour and the efficiency levels provided by various different procedures are presented.

**EXPERIMENTAL SYSTEM AND INVESTIGATION PROGRAMME**

Experimental investigations were performed on the test rig operating at the University of Genoa, the layout of which has been fully described in previous papers (Capobianco and Marelli, 2006). It is a continuous flow apparatus which allows tests to be performed on individual components and subassemblies of automotive engine intake and exhaust circuits both under steady and unsteady flow conditions. The facility is particularly suitable for performing investigations on exhaust turbochargers.

Two dedicated compression stations provide air to both turbocharger fluid machines. The turbine circuit includes an electrical heater which moderately raises air temperature (up to 400 K) in order to avoid condensation and freezing problems during expansion, while the compressor (which acts as
a dynamometer) can be fed with air at controlled pressure in order to extend the investigation of turbine characteristics.

Two arrangements of the turbine supply circuit are available (Figure 1), each fitted with a pulse generator system used to investigate different aspects of unsteady flow turbocharger operation. The two line configurations are connected to the upstream portion of the feeding circuit through a plenum (which acts as a damping element and flow distributor) and can be easily interchanged by modifying a few connections. The same turbine inlet and outlet measurement stations are used with the two line arrangements.

The first circuit layout (Figure 1a) was designed to perform parametric studies on the effect of the main unsteady flow parameters on turbine performance (Capobianco et al., 1990; Capobianco and Gambarotta, 1990; Capobianco and Gambarotta, 1992; Capobianco and Gambarotta, 1993; Capobianco et al., 1995). Tests on single and two-entry components can be developed, independently controlling thermodynamic parameters at each entry. In both feed branches, pulsating flow is generated by a diametral slot rotating valve. The main pressure pulse parameters (amplitude and mean value) at each turbine entry can be controlled by correctly mixing two flow components (steady and pulsating) in a Y-junction and adjusting upstream plenum pressure. For two-entry devices, unequal admission and out-of-phase pulses can be reproduced (Capobianco and Gambarotta, 1993).

The second arrangement of the turbine feeding circuit (Figure 1b) was designed to more precisely replicate turbocharger unsteady flow operation when matched with an automotive engine and to extend experimental investigations to subsystem level. In this alternative layout, heated compressed air at the damping plenum outlet enters a flow distributor emulating the reference engine cylinder block. Pulsating flow is generated by a Variable Valve Actuation (VVA) cylinder head mounted on the flow distributor. Any valve opening profile can therefore be reproduced, thus modifying unsteady flow characteristics in the exhaust circuit, the geometry of which can be easily changed by installing different manifolds. Dedicated throttle valves in the flow distributor allow to simulate engine load transients (referring to mass flow rate).

For both pulsating flow generator systems, a variable rotational speed electrical motor is used to adjust pulse frequency within the typical range of automotive engine intake and exhaust systems (10-200 Hz).
The test rig is fitted with a PC-controlled automatic data acquisition system, using interactive procedures in LabVIEW® environment. Average wall static pressures are evaluated by strain gauge and capacitive transducers, while temperatures are measured by platinum resistance thermometers and turbocharger rotational speed and pulse frequency by inductive probes. Mean turbine and compressor mass flow rates are estimated by a laminar flow meter and a sharp edged orifice respectively. The turbocharger waste-gate opening degree was referred to the angular position of the relative driving shaft and was measured with a variable resistance transducer.

In the event of unsteady flow, instantaneous pressures and turbine mass flow rate are also evaluated. To this end, high frequency response strain gauge pressure transducers are mounted near the duct walls in order to eliminate inaccuracies deriving from the damping effect of the signal connecting lines. Instantaneous mass flow rate is estimated through a constant temperature hot-wire anemometric system providing measuring frequencies up to 10 kHz. Two high-speed data acquisition cards are used to simultaneously acquire instantaneous signals which are then processed using different filtering techniques. A suitable triggering signal allows data acquisition to commence at a definite position of the pulse generator (rotating valve or camshaft).

Within the present study, the assessment of turbine unsteady flow performance was first deepened, focusing on the evaluation of available energy at the turbine inlet and the calculation of turbine efficiency. Several procedures were considered, each requiring a different amount of instantaneous measured data.

Extensive experimental investigations were then performed, referring to two different turbochargers matched for application to 4-cylinder gasoline engines. Both units were fitted with a single-entry nozzleless radial flow turbine and used a waste-gate valve as a turbine mass flow control device. The turbochargers were installed at the UNIGE test rig and connected to the feeding circuit by means of dedicated adaptors. The measurements referred to in this paper were performed using the rotating valve pulse generator system.

Firstly, the effect of pulse parameters on turbine inlet energy was analysed, referring both to theoretical and measured pressure waves. The study then went on to investigate the behaviour of turbine instantaneous mass flow rate, comparing measured levels with those calculated according to a quasi steady flow assumption. Lastly, the results of several unsteady flow turbine efficiency calculation procedures requiring different amounts of experimental data were considered.

ASSESSMENT OF TURBINE UNSTEADY FLOW PERFORMANCE

In high speed automotive engines, the turbocharger turbine usually operates in unsteady flow conditions and its performance is substantially affected by feeding flow characteristics (Winterbone and Pearson, 1999). It is therefore important to correctly assess the main parameters used to characterise turbine unsteady flow behaviour, i.e.: mass flow rate, available inlet energy and conversion efficiency.

The available gas energy at the turbine inlet in pulsating flow operation is affected by the exhaust valve opening profile and the manifold geometry. The relative specific level (referred to mass unit) over the pulse cycle ($\Delta h_{0\text{ NSF}}$) can be calculated as the integral of the instantaneous gas energy, referring to an isentropic expansion from turbine upstream conditions (for which reference is usually made to static levels) to downstream ambient pressure ($p_0$), thus avoiding to take into account real turbine outlet pressure ($p_4$), the level of which can be affected by the particular geometry of the downstream circuit or by the operation of specific devices, such as waste-gate valves (Capobianco and Marelli, 2007).

If the pulse period is split into a number of short time intervals, according to (Zinner, 1978; Lujan et al., 2001), $\Delta h_{0\text{ NSF}}$ can be calculated as the ratio of the sum of gas energy in each time interval and the accumulated mass of gas flowing through the turbine during the entire pulse period:
To evaluate available gas energy at the turbine inlet, it is therefore necessary to know the instantaneous turbine mass flow rate, inlet pressure and temperature (assumed to be constant within each time interval). Pressure diagrams can be measured with good degree of accuracy by using high frequency response transducers and suitable pressure taps in the inlet measurement plane, while the experimental evaluation of instantaneous mass flow rate and temperature is more critical. Alternatively, calculated levels of these quantities can be used, referring to 1D simulation models the results of which can be validated by comparing measured and calculated instantaneous pressure and average mass flow rate and temperature (Lujan et al., 2001).

In the investigation performed at the University of Genoa, instantaneous temperature at the turbine inlet was not directly measured and was calculated on the basis of the adiabatic process of an ideal gas, using mean measured pressure and temperature as a reference:

$$T_{3i} = T_{3m} \cdot \left( \frac{p_{3i}}{p_{3m}} \right)^{\frac{k-1}{k}}$$  \hspace{1cm} (2)

The results provided by this procedure are usually assumed as a reasonable approximation of instantaneous temperature level if this quantity cannot be measured (Szymko et al., 2005).

As regards instantaneous mass flow rate, two different approaches were followed: the experimental level of this parameter at the turbine inlet was detected through a hot wire probe. Since this technique is not usually applicable to engine investigations, instantaneous mass flow rate was also evaluated starting from measured pressure diagrams and turbine steady mass flow characteristics, using a quasi-steady flow (QSF) assumption.

Turbine efficiency is another important performance parameter to be evaluated in unsteady flow conditions. Various approaches can be used to assess this quantity, depending on the amount of available experimental information. If no by-pass system operates in parallel to the turbine rotor (Capobianco and Marelli, 2007), average cycle efficiency ($\eta_{t, NSF}$) can be calculated referring to the integral over the pulse period of instantaneous actual ($\Delta h_{tr}$) and isentropic ($\Delta h_{ts}$) turbine specific work:

$$\eta_{t, NSF} = \frac{\int_{0}^{T} \Delta h_{tr}}{\int_{0}^{T} \Delta h_{ts}}$$  \hspace{1cm} (3)

Assuming a number of short time intervals over the pulse period, average isentropic work ($\Delta h_{ts, NSF}$) can be expressed by:
which is formally similar to equation (1) when replacing ambient pressure \( (p_0) \) with turbine downstream pressure \( (p_{4i}) \). In this investigation, \( \Delta h_{ts\ NSF} \) was calculated referring to measured turbine inlet and outlet pressure diagrams and calculated instantaneous inlet temperature (using equation (2)). As regards turbine mass flow rate, both measured and QSF calculated instantaneous levels were used to evaluate specific isentropic work and efficiency. It was therefore possible to assess the acceptability within the considered calculation of the QSF approach, the results of which are often assumed to be acceptable (Capobianco and Gambarotta, 1990), even though noticeable deviations of the instantaneous unsteady mass flow rate from quasi-steady behaviour were indicated by some authors at higher pulse frequencies (Szymko et al., 2005).

It is not an easy task to determine instantaneous turbine actual work \( (\Delta h_{tr}) \) since it is difficult to directly measure instantaneous torque for high speed rotating components. Some attempts to determine this quantity are presented in open literature (Dale and Watson, 1986; Winterbone et al., 1991; Szymco et al., 2005), based on the measurement of turbine mean torque (from compressor work or using a suitable dynamometer) and the estimation of the fluctuating torque component according to the mass moment of inertia of the rotating assembly and its angular acceleration, calculated from accurate measurements of instantaneous rotational speed.

If no information on instantaneous turbocharger rotational speed is available, the cycle average turbine actual work \( (\Delta h_{tr\ NSF}) \) is usually calculated starting from mean measured levels of turbine power \( (P_{tr\ NSF}) \) and mass flow rate \( (M_{t\ NSF}) \):

\[
\Delta h_{tr\ NSF} = \frac{P_{tr\ NSF}}{M_{t\ NSF}}
\]  

If a high speed dynamometer is not available on the test rig, turbine actual work is generally evaluated on the basis of measurements made on the compressor side of the turbocharger. As a consequence, an overall efficiency level (turbine isentropic efficiency multiplied by turbocharger mechanical efficiency) is estimated through equation (3).

Turbine unsteady average efficiency \( (\eta_{ta\ NSF}) \) calculated according to the above procedure can be compared with steady flow value \( (\eta_{SF}) \) and with the level provided by a simplified approach based on the evaluation of reference isentropic work starting from mean measured pressure and temperature values. The resulting apparent turbine unsteady flow efficiency \( (\eta_{ta\ NSF}) \) can therefore be expressed as:

\[
\eta_{ta\ NSF} = \frac{\Delta h_{tr\ NSF}}{c_{pm} \cdot T_{3m} \cdot \left( \frac{1 - \left( \frac{p_{4m}}{p_{3m}} \right)}{k} \right)}
\]
Finally, if a quasi-steady flow approach is followed, turbine unsteady efficiency can be evaluated on the basis of the reference steady flow curves. To take into account the distribution of turbine mass flow rate over the pulse period, in this investigation a modified calculation method was considered: QSF cycle average efficiency ($\eta_{t\text{QSF}}$) was evaluated as a weighted average value, assuming the QSF mass flow rate in each time interval ($M_{t\text{QSF}}$) as weight:

$$
\eta_{t\text{QSF}} = \frac{\sum_{i=1}^{n} (\eta_{\text{QSF}}, M_{\text{QSF}} \cdot \Delta t_i)}{\sum_{i=1}^{n} (M_{\text{QSF}} \cdot \Delta t_i)}
$$

(7)

EFFECT OF PULSE PARAMETERS ON TURBINE INLET ENERGY

In an initial step of the investigation, the effect of the main pulsating flow characteristics on available energy at the turbine inlet was analysed. Since several unsteady flow parameters proved to be simultaneously affected by the change of turbine operating conditions in real operation, the behaviour of isentropic specific gas energy was first analysed referring to theoretical waves. To this end, sine and square pressure waves at the turbine inlet were considered.

The available gas energy at the turbine inlet was calculated according to Eq. (1), assuming an isentropic correlation for the evaluation of instantaneous inlet temperature $T_{3i}$ (Eq. (2)) and a quasi-steady flow (QSF) assumption to assess turbine mass flow rate in each time interval ($M_{t\text{QSF}}$). Reference was made to a typical turbocharger turbine for automotive gasoline engines, operating at a fixed rotational speed factor level ($n/\sqrt{T_3} = 4000 \text{ rpm/} \sqrt{\text{K}}$).

In the case of sinusoidal pressure waves, Figure 2 shows the effect of pulse amplitude on specific energy available at the turbine entry, for different average pressure levels. A trend towards higher energy values when increasing oscillation amplitude can be seen for each mean inlet pressure, with reduced impact at higher $p_{3m}$. This behaviour confirms that available energy is mainly related to flow unsteadiness, which is usually associated with the ratio between pulse amplitude and the relative mean level (Zinner, 1978).

The influence of inlet temperature oscillation amplitude on turbine available energy was also investigated, at constant pressure diagram, by correlating instantaneous temperature with the relative mean value through different polytropic relationships. A small impact of temperature oscillation amplitude was found, particularly at moderate mean inlet temperatures (as experienced
on the UNIGE test rig). This result confirms that the use of an approximate procedure to assess instantaneous turbine inlet temperature when the experimental value is not available does not give rise to substantial errors in the determination of specific inlet energy.

Depending on the design of the engine exhaust manifold and of the turbine admission (single or twin-entry), pressure oscillations at the turbine inlet can be extended over a different fraction of the period. It can therefore be useful to introduce a proper pulse duty cycle factor ($\Phi$), defined as the pulse length of the wave as a fraction of the wavelength (Costall et al., 2006). Referring to sinusoidal pressure oscillations with the same amplitude and mean level, Figure 3 shows the effect of the pulse duty cycle on available specific energy (with reference to the condition $\Phi=1$). It is apparent that a wider pulse extension over the period generates an increase in turbine inlet energy: at constant pressure oscillation amplitude and mean level, the maximum specific energy content is achieved with a pulse extended over the entire engine operating period. This situation is typical of most automotive engine arrangements fitted with single-entry turbocharger turbines.

Since experimental tests revealed significant modifications in the pressure oscillation shape when changing pulse frequency (and therefore engine rotational speed) (Capobianco and Marelli, 2005), the inlet energy levels related to sine and square waves with the same mean pressure value were compared. Figure 4 shows the relative results plotted against pulse amplitude and referred, for each wave configuration, to the energy level calculated at the minimum pulse amplitude considered ($\Delta p_3=0.2$ bar). It is evident that the available energy increase at higher pulse amplitudes is greater in the case of the square pressure wave, for which the instantaneous shift from the average level over the pulse period is more significant. It is therefore apparent that the ratio between pulse amplitude and mean value is not sufficient to characterise flow unsteadiness which should also be related to oscillation shape.

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As mentioned above, measured pressure diagrams at constant turbine rotational speed and mean inlet pressure were significantly affected by pulsating flow frequency, due to wave action in the turbine feeding circuit (Capobianco and Marelli, 2006). However, a general trend towards higher pressure oscillation amplitudes was found when increasing the frequency (Figure 5). The specific inlet gas energy calculated proved to be substantially dependent on pulse amplitude, as shown in Figure 6, confirming the trend highlighted by theoretical waves analysis.

At constant pulse frequency, pressure profile shapes were similar when changing turbine inlet pressure level and the correlation between inlet pulse amplitude and turbine mean inlet pressure proved to be almost linear (Figure 7). The behaviour of mean available gas energy is the outcome of the combined effect of both these parameters (Figure 8).

TURBINE UNSTEADY MASS FLOW AND EFFICIENCY CHARACTERISATION
The study was then extended to the analysis of instantaneous turbine mass flow rate in unsteady flow conditions. To this end, several measurements were performed on a small IHI turbocharger (model RHF3), matched to a downsized SI automotive engine with a capacity of approximately 1.4 litres, which was installed on the UNIGE test rig. The relative turbine mass flow rate and efficiency steady state curves were previously defined over an extended range for different rotational speeds and waste-gate opening degrees (Capobianco and Marelli, 2007).

During the unsteady flow tests, turbine inlet and outlet pressure diagrams and inlet instantaneous mass flow rate were measured simultaneously using high frequency response strain gauge transducers and a hot-wire probe respectively. The anemometric system calibration curve was formerly defined in steady flow conditions with the probe installed in the measurement plane, assuming as a reference the mass flow level supplied by the laminar flow meter. The analysis was extended to different turbine expansion ratio levels, whilst keeping average rotational speed factor \(n/\sqrt{T_3}=4000 \text{ rpm}/\sqrt{\text{K}}\) and pulse frequency \(f=66.67 \text{ Hz}\) constant. The waste-gate valve was kept closed during this investigation.

The assumption of instantaneous steady flow behaviour of the turbine operating under pulsating flow conditions, often used within simulation models to predict unsteady flow performance, was also considered. Using this approach, instantaneous QSF turbine performance was calculated starting from inlet and outlet pressure diagrams and referring to steady flow characteristics, according to the procedure first proposed in (Benson, 1974).
Figure 9 shows instantaneous measured and calculated mass flow rate and inlet pressure diagrams over the pulse period for three different average levels of turbine expansion ratio (\(\varepsilon_m\)). All experimental signals were acquired for several complete cycles (generally 50) and appropriate averaging techniques were then applied to the measured data. Experimental mass flow rate profiles proved to be affected by higher frequency components when compared with pressure and calculated mass flow diagrams. A Fast Fourier Transform (FFT) analysis highlighted significant harmonic components up to a frequency of approximately 600 Hz, confirming the results of a preliminary investigation developed at a different pulse frequency level (Capobianco and Marelli, 2007). Higher frequency oscillations are probably related to flow disturbances in the measuring section. On the contrary, QSF mass flow rate profiles proved to be strictly related to measured pressure diagrams, due to the computation procedure used. Experimental and calculated diagrams were almost in phase at a lower turbine expansion ratio, while a noticeable phase shift (about 1/10 of the oscillation period) was observed when increasing the expansion ratio and consequently average mass flow rate level, probably due to more substantial filling and emptying effects in the turbine volute casing. Similar behaviour was observed, at a constant turbine overall expansion ratio, when opening the waste-gate valve, thus increasing the mass flow level and the magnitude of unsteady flow phenomena (Capobianco and Marelli, 2007).

Mean measured and QSF mass flow rate values proved to be similar in these operating conditions; calculated levels slightly underestimated experimental ones, with deviations generally below 5%. This result confirms the conclusions of previous investigations performed on different turbocharger turbines (Capobianco and Gambarotta, 1990; Capobianco and Marelli, 2006). Different behaviour, instead, was observed as regards wave amplitude. The correlation between inlet pressure oscillation amplitude and turbine mean expansion ratio was confirmed (see Figure 7), but experimental mass flow rate oscillation amplitude also proved to increase significantly at a higher average pressure ratio, due to the higher mean mass flow level and the relative inertial effects. On the contrary, calculated mass flow rate profiles highlighted an almost constant oscillation amplitude when changing the turbine expansion ratio: this result confirms that a QSF
approach is not adequate to describe system behaviour when unsteady effects become prominent (Szymko et al., 2005).

The deviation of unsteady flow turbine performance from steady state behaviour is apparent if instantaneous measured mass flow parameter is plotted against the turbine expansion ratio, as in the case of steady flow characteristics (Figure 10). The filling and emptying of the volute is highlighted by the hysteresis loop surrounding the steady state curve, the area of which was found to increase at a higher turbine expansion ratio, where flow unsteadiness is more significant (Figure 9). It is confirmed that, at typical pulsating flow frequencies occurring in the exhaust system of automotive engines, the pulse is so rapid that mass flow does not have enough time to incrementally fill the volute volume with pressure, hence causing the hysteresis observed between measured mass flow rate and pressure. Besides, the steady state curve is not totally encapsulated by the unsteady mass flow loop. As reported by other authors (Szymko et al., 2005), this may be due to the fact that there is not enough time to fill the volute volume before the peak of the pulse is achieved and suggests that the filling and emptying action is influenced not only by gas velocity oscillation but also by the unsteadiness of pressure wave. Experimental analysis on this subject will be extended at a later stage of the investigation, taking into account a wide pulse frequency range and the influence of the waste-gate setting.

Figure 11 shows the results obtained from the application of the above approaches to assess turbine unsteady efficiency. The steady state efficiency curve measured on the UNIGE test rig using the same measurement inlet and outlet planes and proper data processing is also plotted as a reference. It is apparent that, all test conditions being uniform, steady state efficiency values were always higher than the unsteady results provided by the different procedures, confirming the fact that pulsating flow deteriorates turbine efficiency (Watson and Janota, 1982).

Unsteady efficiency levels provided by the various evaluation methods proved to be different from each other, thus confirming the importance of an appropriate procedure for assessing this parameter.

Apparent turbine unsteady efficiency ($\eta_{ua, NSF}$), calculated on the basis of mean measured pressure and temperature levels, was lower than measured steady state efficiency. The reductions, amounting
to approximately 7-8%, confirmed the trend observed by other authors (Lujan et al., 2001) who measured differences of about 13% referring to a twin-entry turbocharger turbine for a heavy-duty diesel engine application.

The procedure based on the evaluation of instantaneous inlet and outlet parameters provided the most reliable turbine unsteady efficiency levels. In this case, the calculated efficiency values were about 12-13% less than the steady state efficiency levels measured at the same average expansion ratio. No significant differences were found between unsteady efficiency levels ($\eta_{t NSF}$ and $\eta^*_t NSF$ respectively) evaluated using measured or calculated instantaneous mass flow rate to estimate average turbine isentropic work. This result may be related to comparable average levels of experimental and QSF mass flow rate. The shifts observed between mass flow profiles (Figure 9) probably affect instantaneous efficiency level, though to evaluate this, turbine actual work must also be assessed.

The modified QSF procedure, based on the calculation of a weighted average efficiency level, did not provide satisfactory results in the selected operating conditions. The relative levels proved to be lower than steady state values but higher than those provided by the other procedures for evaluating turbine average unsteady efficiency. This result seems related to the prevailing influence of steady flow efficiency levels in the calculation procedure.

CONCLUSIONS AND FURTHER STEPS
This paper presented the results of an extensive investigation on automotive turbocharger turbine unsteady flow behaviour. Several tests were performed with a dedicated test rig operating at the University of Genoa on two different turbochargers matched for application to 4-cylinder gasoline engines.

Firstly, the assessment of turbine unsteady flow performance was analysed. The available specific gas energy at the turbine inlet was evaluated starting from instantaneous levels of pressure, temperature and mass flow rate. Different options for determining these quantities were considered, taking available experimental data into account.
Various approaches towards assessing turbine unsteady efficiency were then presented, each differing as regards the procedure used to estimate isentropic turbine specific work. To this end, a method based on the evaluation of instantaneous parameters affecting this quantity (both measured or calculated) was proposed as an alternative to the simplified approach based on the evaluation of reference isentropic work starting from mean measured pressure and temperature levels. A modified quasi-steady flow approach was also considered by means of which turbine unsteady efficiency can be evaluated as an average weighted level referring to steady state curves.

The effect of unsteady flow characteristics on the available energy at the turbine inlet was then studied, referring both to theoretical inlet waves and to pressure profiles measured on the UNIGE test rig within an experimental campaign performed on a small automotive turbocharger turbine. A trend towards higher energy values when increasing the amplitude of oscillation was found at constant mean inlet pressure, with reduced magnitude at a higher average level. This behaviour confirmed that available energy is mainly related to flow unsteadiness, usually associated with the ratio between pulse amplitude and the relative mean level.

A small impact of temperature diagram amplitude on turbine inlet energy was found, particularly at moderate mean inlet temperature. On the contrary, both pulse extension over the oscillation period and the relative shape significantly affected the available specific gas energy level when comparing theoretical pressure waves with different characteristics. At constant pulse amplitude and mean level, the maximum specific energy content was achieved with a pulse extended over the whole period, as usually occurs in automotive engine arrangements fitted with single-entry turbocharger turbines. Moreover, available energy increased in the case of inlet pressure profiles affected by a significant shift of instantaneous level from the average value over a large fraction of the oscillation period.

Measured pressure diagrams were significantly influenced by pulsating flow frequency, due to the wave action in the turbine feeding circuit. However, a general trend towards higher pressure oscillation amplitude and specific inlet gas energy was found when increasing pulse frequency. On the contrary, pressure profile shape was found to be similar when changing turbine inlet pressure level at constant pulse frequency and mean available gas energy proved to be affected by the combined effect of pulse amplitude and mean value.

The study went on to analyse turbine unsteady mass flow rate and efficiency performing measurements on a small turbocharger for downsized gasoline engines. Instantaneous mass flow profiles measured through a hot wire probe were compared with the levels calculated using a quasi steady flow (QSF) approach. Measured diagrams proved to be affected by higher frequency components, probably related to flow disturbances in the measuring section. On the contrary, QSF mass flow rate profiles were strictly related to measured pressure diagrams, due to the computation procedure used. Slight differences were found between measured and calculated average mass flow levels, while significant differences were observed as regards wave amplitude, confirming that a QSF approach is not adequate to describe system behaviour when unsteady effects become prominent.

The deviation of unsteady flow turbine performance from steady state behaviour was confirmed when plotting instantaneous measured mass flow rate against the turbine expansion ratio. In this case the filling and emptying of the turbine volute volume was highlighted by a significant hysteresis loop surrounding the steady state curve, the area of which was found to increase at a higher turbine expansion ratio.

All test conditions being equal, steady state efficiency values were always higher than the unsteady results, confirming the fact that pulsating flow deteriorates turbine efficiency. In any case, substantial differences between average unsteady efficiency levels calculated using different methods were found, thus confirming the importance of an appropriate procedure for assessing this parameter in the typical operating conditions occurring at the exhaust of automotive engines.
Efficiency calculated on the basis of mean measured pressure and temperature levels was lower than steady state efficiency but proved to be significantly higher than the values provided by a more reliable procedure based on the evaluation of instantaneous inlet and outlet turbine parameters. Referring to this method, no significant differences were found when using measured or calculated instantaneous mass flow rate to estimate average turbine isentropic work. On the contrary, a modified QSF procedure based on the calculation of a weighted average efficiency level did not provide satisfactory results.

The investigation will be further developed at a later stage in order to assess instantaneous turbine actual work in unsteady flow conditions starting from the measurement of instantaneous turbocharger rotational speed. This experimental information, together with the measurement of instantaneous mass flow rate, will make it possible to estimate turbine efficiency diagrams over the pulse period, highlighting the operating conditions in which a QSF approach gives a poor approximation of real turbine behaviour. At the same time, the analysis will be extended to include a wide pulse frequency range and different waste-gate valve settings.

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