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Nomenclature
Introduction

Pulse turbocharging is widely used in a variety of internal combustion engine applications in order to maximize the energy transfer between the hot exhaust gases and the turbocharger turbine. In such an application, a highly pulsating flow field resulting from the opening and closing of the reciprocating engine exhaust valve is delivered to the turbine inlet via a compact exhaust manifold. Although researchers are aware of the highly unsteady nature of the flow driving the turbine wheel, a large proportion of the experimental knowledge accrued over the past 40-50 years has been based on treating the turbine as a steady-flow, aerodynamic device. This is largely due to the inherent difficulty in making accurate time-resolved unsteady flow measurements in such a challenging environment. In addition, there appears to be a general feeling in some of the early research that the turbine did not vary significantly from quasi-steady behaviour. This presumes that over a pulse cycle, the unsteady performance of the turbine at every instance in time can be described without considering the previous history of the flow. Were this assumption valid, the turbocharger’s unsteady performance could be predicted as an integration of steady behaviour on an instant-by-instant basis.

With this in mind, early research into the unsteady performance of turbochargers such as Benson et. al [1,2], Wallace et. al [3,4,5], Kosuge, et al [6] and later Capabianco et. al [7,8,9] all sought to quantify the degree to which the turbocharger acted in a quasi-steady manner. Since fast response pressure transducers are easily implemented, a number of these researchers were able to directly measure instantaneous pressure at various locations along the turbine stage. The unsteady pressure ratio data in conjunction with averaged turbine wheel speed could then be matched with the corresponding steady-flow conditions to obtain a dynamic quasi-steady ‘equivalent’. However, since the other instantaneous unsteady parameters were not measured, the quasi-steady data could only be compared withfor different resonances without time consuming CFD calculations and potential blade excitations caused by the diffuser can be accounted for in the design process.

Background

The CFD simulations of a single stage centrifugal compressor have been carried out by Dickmann et al. [1] with the CFX5.7 code from ANSYS [7]. An impeller with 8 main and splitter blades, an inducer casing bleed system, a low solidity diffuser with 9 vanes, a 90 degree suction elbow and an asymmetric diffuser outlet static pressure distribution, typical for volutes was modelled. The asymmetric pressure distributions at the outlet of the diffuser allows doing away with the modelling of the volute so that the computational domain is smaller and less computational effort is necessary. Average values for total temperature and total pressure, a turbulence intensity of 5 % and flow vertical to the inlet boundary have been applied to the inlet boundary. One impeller revolution has been resolved by 256 time steps using a grid with 730’000 cells. The k-e turbulence model with scalable wall functions has been applied. 2nd order discretization in space has been used. Each time step is converged to 10-3 of all maximal residua after 10 internal time step loops. These maximal residua have been regarded as sufficient concerning the data of interest.

The compressor stage belongs to a turbocharger, which supercharges a state-of-the-art 1’100 kW Diesel engine. Investigations described here have been done on two off-design operating points on a speed line of 83 % design speed. OPcC corresponds to 80.4 % of design volume flow, 67.8 % design total pressure ratio and 103 % of design efficiency. OPcS to 60.0 %, 70.7 % and 100.0 % respectively (figure 1). The relative Mach numbers at the tip leading edge of the impeller are 1.50 for OPcC and 1.20 for OPcS. The absolute Mach numbers at the leading edges of the diffuser vanes are 0.65 for OPcC and 0.60 for OPcS.

The unsteady pressure data on the surfaces of the blades from transient CFD results were used to determine the harmonic excitations. When applied to an FE Model the amplitudes as well as the stresses in the impeller of the first eigenmode could be calculated. Results for the two operating points OPcC and OPcS on the same speed line are compared. The main statement of [1] is that different amplitudes of an inlet induced mode shape could be calculated for operating points close to choke and close to surge on the same speed line and that calculated amplitudes have been in good agreement with experimental research. These results act as a reference for the following analyses.
With this in mind, early research into the unsteady performance of turbochargers such as Benson et al. [1,2], Wallace et al. [3,4,5], Kosuge, et al [6] and later Capobianco et. al [7,8,9] all sought to quantify the degree to which the turbocharger acted in a quasi-steady manner. Since fast response pressure transducers are easily implemented, a number of these researchers were able to directly measure instantaneous pressure at various locations along the turbine stage. The unsteady pressure ratio data in conjunction with averaged turbine wheel speed could then be matched with the corresponding steady-flow conditions to obtain a dynamic quasi-steady ‘equivalent’. However, since the other instantaneous unsteady parameters were not measured, the quasi-steady data could only unsteady performance on the basis of time-mean values. The most convenient method of comparison was to calculate the quotient of the mean unsteady and quasi-steady values thereby reproducing parameters such as $I_m$ and $I_p$ which compare mass flow and output power respectively (Equations 1 and 2).

$$I_m = \frac{1}{TT} \int_0^{TT} \frac{m(t)}{Q_s} dt$$

$$I_p = \frac{1}{TT} \int_0^{TT} \frac{W(t)}{Q_s} dt$$

Kosuge et.al [6] is among the more comprehensive studies using this mean quasi-steady approach. This work reported a tendency of the quasi-steady prediction to under-predict both mass flow and power but demonstrated a contrasting influence of pulse frequency. The average quasi-steady mass flow appeared to approach the unsteady value as the frequency increased, but the measured power yielded the opposite trend, with the quasi-steady and unsteady averages diverging with increasing frequency. Like other authors [2,7,8], they proposed that the differences between quasi-steady and unsteady mean values were not only dependent on the frequency of pulsation, but also on the amplitude and pulse shape. Capobianco [8] also did considerable work on the influence of pulse amplitude and published results which suggested that the amplitude of the unsteady pulse had a greater impact on the departure from quasi-steady behaviour than the frequency.

The approach of comparing time-mean performance data has generally not been able to provide a clear consensus on the underlying causes of the departure of the unsteady behaviour from the steady case. Therefore, to begin to understand the physical processes which drive unsteady behaviour, time-resolved flow data must be acquired over a pulse cycle. Dale and Watson [10] were one of the first to comprehensively measure instantaneous unsteady quantities, followed by Baines et al. [11], Arcoumanis et.al [12,13], Karamanis et.al [14,15], Szymko et.al [16], Rajoo et. al [17,18] and Copeland et. al [19]. This body of work has yielded a number of insights into the effect of a pulsating flow field on the turbine operation. Some of the more fundamental discoveries are as follows:

1. The volume contained in the volute prior to nozzle entry acts as a reservoir which can accumulate and empty mass over a pulse cycle. In addition to this ‘filling and emptying’, there is a further influence of wave dynamics in the passage length. The balance of these two effects determines the dynamic behaviour of the flow entering the rotor and therefore the power it produces.

2. For the range of frequencies typically encountered in an IC engine, the rise in pressure is too rapid to allow the mass accumulation to follow. This hysteresis between pressure and mass flow generates an operating orbit that may or may not encircle the quasi-steady operating line. This effect has also been replicated in 1D wave action simulations [26].

The unsteady departure from the steady-state relationship between pressure and mass flow has been a topic of some debate and will be discussed further in this paper. Szymko et.al [16] attempted to explain this phenomenon by proposing three modes to describe the relationship between pressure and mass flow over a pulse cycle. The first involves purely quasi-steady behaviour where the pressure change over a pulse is sufficiently slow to permit a state of equilibrium to exist between mass flow and pressure. As the increase in frequency begins to produce the hysteresis between pressure and mass flow, a second mode is proposed where an operating orbit results which encapsulates the steady flow behaviour. A further increase in frequency results in the final mode where the turbine starts to depart from steady operation altogether, and the operating loop will no longer encapsulate the quasi-steady line. To distinguish between these three modes of operation, the authors suggested the Strouhal number as a useful indicator. The Strouhal number defines the rate of localized changes of a moving particle in comparison to its rate of propagation. This basic definition has been modified slightly in Equation 3 to include the parameter $\Phi$ which accounts for pulse events which are a fraction of the total wavelength.
In a standard fixed-nozzle turbine, the throat area is designed to pass the necessary mass flow for a particular application [20]. The influence of the nozzle opening is therefore quite straightforward to predict for steady flow. However, under unsteady flow, the addition of a nozzle ring produces some interesting effects that have yet to be clearly understood. Rajoo and Martinez-Botas [17, 18] were one of the first to comprehensively test unsteadiness in a nozzled turbine. They had the advantage of a variable nozzle turbine whereby the effect of different nozzle openings could be measured under steady and unsteady conditions. First, they noted that the nozzle acted to isolate some of the unsteadiness in the volute passage from the interspace volume upstream of the turbine rotor. Also, since the nozzle area controls the swallowing capacity of the turbine, it therefore determined how quickly the scroll volume can empty the mass that has accumulated over a pulse. Thus the nozzle geometry appeared to have quite a large impact on the relationship between mass flow and pressure measured at the turbine inlet. Most interestingly, if steady and unsteady operating lines were plotted together, the degree to which these behaviours coincided depended on the nozzle opening. This research suggested that the nozzle could also have some influence on the steady–unsteady disagreement.

The quasi-steady assumption becomes much harder to evaluate in a two-inlet scroll. Unless the unsteady pulsations arrive at the turbine completely in-phase, there will be two distinct flow conditions driving the turbine wheel over much of the pulse cycle. Since this unequal flow has been shown to produce significant performance implications under steady flow conditions [19,22,23], such test data should be used in the quasi-steady analysis. Benson and Scrimshaw [1] tested a double-entry, nozzled turbine exposed to two pulsating inlet conditions that were not in-phase. Nevertheless, they could only calculate averaged quasi-steady parameters based on either equal or partial admission steady flow maps. Their analysis was therefore not a true quasi-steady comparison. Further work by Baines, et.al [11] did cover out-of-phase pulsations but did not publish a quasi-steady comparison likely due to the prohibitive amount of steady data required. In fact, for this reason, it appears as if a dual-entry, quasi-steady comparison using the full range of unequal admission data has not been attempted until now.

\[
MS_t = \frac{fL}{U} \frac{1}{2\phi}
\] (3)

Experimental facility

Test facility

The turbocharger turbine test facility located at Imperial College London has been described by other researchers in some detail [16-19] but a brief overview is again provided here. The facility consists of an air supply and heating system, pulse generator, high speed dynamometer, data acquisition and control equipment. As shown in Figure 2, the main air compressors supply two independent sets of pipe-work which are ideally suited to dual-entry turbines. The mass flow rate in each limb is controlled independently through two valves thus allowing full or partial admission testing under steady or unsteady conditions. A motor drives a set of rotating ‘chopper’ plates which periodically cuts the flow to produce unsteady pulsations. The frequency is adjusted by varying the motor speed and the phase of the pulses between the two limbs is adjusted by repositioning the chopper plates. Although the air system also includes a heater to prevent condensation across the turbine, the test facility essentially operates with ‘cold’ flow. This allows the researcher to measure instantaneous flow parameters that would be very difficult to measure in the harsh conditions of a high-temperature exhaust flow. Using the principle of similitude, the cold flow of the test facility can be scaled up to hot exhaust conditions through the use of dimensionless performance parameters.
A sectioned view of the circumferentially divided, double-entry turbine is shown in Figure 1. This turbine was used in all the experimental testing presented in this paper. Each inlet supplies a separate 180° sector of the 24-blade nozzle ring. Between the exit to the nozzle and the inlet to the mixed-flow wheel there is a gap termed the nozzle-wheel interspace which spans the circumference of the turbine. After passing through the rotor, the flow expands into a straight duct to prevent an immediate expansion to atmosphere. Pressure is monitored at the two inlets measuring stations and on this exit duct. The turbine is coupled to a purposely built eddy-current dynamometer described in detail by Szymko et.al [21]. It allows testing of the turbine over a wide range of velocity ratios unavailable using standard compressor loading. Since it is very important to have a broad steady performance map as a basis of comparison with unsteady traces, this ability is indispensable to the research here. The dynamometer housing is mounted on a gimble arrangement which allows the average torque to be measured through reaction against a load cell. Instantaneous mass flow and pressure data is acquired at a measuring plane just prior to the turbine inlet. A new remote controlled dual hotwire system was installed to measure the instantaneous mass flow into each inlet simultaneously. On the strength of work by Szymko et. al. [16], instantaneous inlet temperature was calculated by using average thermocouple temperatures and assuming adiabatic compression/expansion. This was shown to provide a good approximation of the measured dynamic temperature. Lastly, dynamic torque was derived from the addition of the fluctuating torque calculated from the change in speed and average torque taken from the load cell reading. Each dynamic signal was post-processed first by ensemble averaging followed by a low pass filter to remove noise. Further detail into each of these measurements can be found in reference [16].

**Steady experimental results**

**Performance parameters**

The standard dimensionless and pseudo-dimensionless steady-flow performance parameters appropriate for a double-entry turbocharger are presented below. Since each inlet supplies a separate section of the nozzle vanes, a single averaged inlet pressure is not appropriate. Therefore, the two entries are treated separately by calculating independent pressure ratios and mass flow parameters with the measurements made at each inlet station. Equations 4 to 7 show the equations of pressure ratio, mass parameter, efficiency and velocity ratio respectively. The subscripts inner and outer refer to the lower and upper turbine inlet respectively (refer to Figure 1).

\[
P_{\text{inner}} = \frac{P_{01}}{P_4} \quad P_{\text{outer}} = \frac{P_{01}}{P_4} \quad (4)
\]

\[
M_{\text{inner}} = \frac{m \sqrt{T_{01}}}{P_{01}} \quad M_{\text{outer}} = \frac{m \sqrt{T_{01}}}{P_{01}} \quad (5)
\]

\[
\eta_{1-x,\text{total}} = \frac{W_{\text{act}}}{W_{\text{is,inner}} + W_{\text{is,outer}}} \quad (6)
\]

\[
U_{2} = \frac{\pi \cdot d_{2} \cdot N}{\sqrt{2 \left( \frac{W_{\text{is,inner}} + W_{\text{is,outer}}}{m_{\text{total}}} \right)}} \quad (7)
\]
Steady admission results

A complete map of the steady, equal admission performance for the current turbine has been published in reference [19]. In addition unequal admission performance at a constant velocity ratio of 0.65 was also provided. To better visualize the data, two-dimensional contour plots are interpolated from the data to show how a dependent variable such as efficiency varies with standard non-dimensional parameters. For equal admission data, relative efficiency is plotted in Figure 3 against velocity ratio on the ordinate and pressure ratio on the abscissa. For unequal admission data at a constant velocity ratio, the efficiency can be plotted against the pressure ratios across each of the inlets as in Figure 4. Thus, there are a total of three independent parameters needed to define the operating point of a dual-entry turbine – two pressure ratios and the velocity ratio. Consequently, the equal and unequal steady-flow characteristics can be combined on a single, three-dimensional plot as shown in Figure 5.

Since the turbine can experience a wide variation of all three parameters during a single unsteady pulse, in order to make a full comparison between steady and unsteady performance, the scope of available steady-flow test data must be broad enough to produce a reliable interpolation with little required extrapolation. Therefore, additional unequal admission tests to those reported in [19] were needed at velocity ratios equal to 0.8 and 0.5. The efficiency contours from these new tests are shown in Figure 4a and 4c, respectively. Since all these figures use the same colour map to represent relative turbine efficiency, it makes it easy to highlight areas of optimum turbine performance. For example, it is clear from Figure 3 and 4b that there is a band of high efficiency at a certain velocity ratio and pressure ratio.
efficiency in the range of velocity ratios between 0.6 and 0.7, as might be expected from a mixed-flow turbine wheel. It is also interesting to note from contours in Figure 4 that loss generation in steady, unequal flow conditions is essentially symmetric about the equal admission line (PR_{in} = PR_{out}). Moving off this line represents a greater pressure difference between the entries, leading to a penalty in turbine performance. This new data therefore confirms the conclusion that the losses introduced by unequal admission are a function of the ratio of pressures between the inlets but generally independent of which inlet condition is varied. This finding contrasts the behaviour of a meridionally divided volute which rarely produces similar symmetric behaviour under unequal admission conditions [22].

Unsteady experimental results

Performance parameters

Unsteady, time-resolved values of the standard performance parameters listed in Equations 4-7 are calculated by substituting instantaneous values of speed, pressure, mass flow, and temperature. From these time-resolved values, cycle-averaged parameters can also be derived which can prove useful for the overall performance analysis. In order to make a correct comparison between the different unsteady experimental cases, it was important to maintain a constant cycle-averaged velocity ratio, which was calculated as per Equation 8, giving an energy-weighted mean velocity ratio. In our investigation, this value was maintained at a constant 0.65. Note that if the value of instantaneous efficiency is substituted into Equation 8 in place of the velocity ratio, this will result in the definition of cycle-averaged efficiency given in Equation 9. This efficiency definition is very useful because it is defined as the quotient of total output and total input power, thus making it independent of any phase difference between the two measurements.

\[
\left( \frac{U}{C_{is}} \right)_{avg} = \frac{\sum_{t=0}^{T} \left[ U_{2}^{C_{is}}(t) \cdot \dot{W}_{is}(t) \right]}{\sum_{t=0}^{T} \dot{W}_{is}(t)} \tag{8}
\]

\[
\eta_{t-x,avg} = \frac{\sum_{t=0}^{T} \dot{W}_{act}(t)}{\sum_{t=0}^{T} \dot{W}_{is}(t)} \tag{9}
\]

Unsteady test cases

Table 1 lists the pulsating conditions that will be considered in this paper. There are a total of six test cases, since at each frequency, both in-phase and out-of-phase pulsations were tested. The rotor speed for all cases was 85% speed and the frequency of pulsations varied between 42-84 Hz. The wheel speed was selected on the basis of previous work in [19] which suggested that 85% speed would provide a turbine unsteady operating orbit that best coincided with the areas of optimum steady-state efficiency. Three values of \( \lambda = N/f \) were set, namely 10, 15, and 20. This parameter specifies the number of times the turbine wheel rotates in the time it takes the pulse generator to produce one complete pulse cycle. This is useful because it gives some sense of how the rate of change in flow conditions due to a single pulse compares with the rotor process.

<table>
<thead>
<tr>
<th>Speed</th>
<th>Phase</th>
<th>P.R. (avg)</th>
<th>Freq. (Hz)</th>
<th>U/C_{is} (avg)</th>
<th>Equiv. 6 cyl rpm</th>
<th>( \lambda )</th>
</tr>
</thead>
<tbody>
<tr>
<td>85%</td>
<td>in</td>
<td>1.64</td>
<td>42.1</td>
<td>0.658</td>
<td>1684</td>
<td>20</td>
</tr>
<tr>
<td>85%</td>
<td>out</td>
<td>1.67</td>
<td>42.2</td>
<td>0.654</td>
<td>1688</td>
<td>20</td>
</tr>
<tr>
<td>85%</td>
<td>in</td>
<td>1.72</td>
<td>56.3</td>
<td>0.660</td>
<td>2252</td>
<td>15</td>
</tr>
<tr>
<td>85%</td>
<td>out</td>
<td>1.74</td>
<td>56.5</td>
<td>0.665</td>
<td>2260</td>
<td>15</td>
</tr>
<tr>
<td>85%</td>
<td>in</td>
<td>1.74</td>
<td>83.3</td>
<td>0.660</td>
<td>3332</td>
<td>10</td>
</tr>
<tr>
<td>85%</td>
<td>out</td>
<td>1.80</td>
<td>83.3</td>
<td>0.644</td>
<td>3332</td>
<td>10</td>
</tr>
</tbody>
</table>

Table 1: Six unsteady test cases

![Steady-flow interpolated relative efficiency contours plotted in 3-D space](image)
One of the difficulties in making sense of instantaneous quantities in a turbine is the incongruity between the energy measured at the volute inlet in terms of mass flow, pressure and temperature, and the power that is imparted to the turbine wheel a moment later. The generally accepted approach is to shift the phase of the inlet waveform to ensure a common time frame with output power. This was done by calculating the time difference using the isentropic pressure wave velocity (bulk flow + sonic speed) and the length of the flow path from volute inlet to a representative wheel entry point. Since the double entry configuration has different entry lengths, two flow paths for each entry were measured from the inlet measuring station to a point 90° downstream of the corresponding tongue. The unsteady data presented in Figures 6 and 7 has been phase-shifted in this manner. As Figure 6c demonstrates, owing to the different passage lengths, pulses arriving at the turbine inlets in-phase will be slightly out-of-phase at the wheel inlet. Naturally, this approach presumes that the shape of the pulse is not subject to wave interference in the volute. This assumption might seem too prescriptive, given the reality of the flow in this type of turbine; in any case, note that a precise phase shift is generally only critical when calculating the instantaneous turbine efficiency. To avoid some of the possible phase shift issues, cycle averaged efficiency values are frequently employed.

Figures 6a and 7a show the unsteady operating orbits in three dimensional space for in-phase and out-of-phase operation respectively. Since these traces can be somewhat difficult to follow in three dimensions, they are reproduced from an equal admission perspective in Figures 6(b,c) and 7(b,c). The steady-state efficiency characteristics are also plotted in the background with a different colour scheme than used in early figures to provide a clear view of the unsteady operating traces. From these figures, as the frequency is increased, the variation of both velocity ratio and pressure ratio over a pulse is reduced, resulting in a more compact unsteady orbit. Figure 9 demonstrates this more clearly by showing a reduction in amplitude of both the pressure ratio and resulting wheel speed for the 42 and 84 Hz in-phase pulses. In addition, the dynamics of a reflecting pressure wave at 84Hz appears to create secondary peaks in the pulse shape. Since the length of the two volute passages is different, the influence of such pressure wave dynamics will differ for each inlet resulting in an asymmetric operating trace at higher frequencies (compare the 42Hz and 84Hz traces in Figure 7c). Thus these diagrams appear to indicate that the increase in frequency is linked to an increase in wave dynamics but also a reduction in the variation of pressure ratio and velocity ratio over a pulse cycle (in terms of amplitude). This suggests that despite the wave interaction at higher frequency, the turbine is, in fact, less dynamic in terms of the overall variation of instantaneous quantities.

Unsteady and quasi-steady comparison
In order to understand how far the unsteady operation is from the steady equivalent, a quasi-steady analysis was carried out. As already illustrated, in order to fix the operating point of the turbine in a double-entry turbine, three non-dimensional parameters are needed: \( PR_{\text{inner}} \), \( PR_{\text{outer}} \) and \( U/C_u \). A three-dimensional interpolation was needed to calculate the steady-state turbine efficiency that corresponded to the three non-dimensional parameters; the accuracy of this procedure clearly depends on the amount of data available from the steady tests. Thus, a total of 272 steady-state test points were measured encompassing a broad range of equal and unequal admission conditions. A program was then used to fit a hypersurface of the form \( w = f (PR_{\text{inner}}, PR_{\text{outer}}, U/C_u) \) to these non-uniformly spaced experimental points using Delaunay Tesselation-based linear interpolation. The algorithm interpolates the hypersurface at the points specified by a uniform Cartesian grid with a resolution of 0.01. Since at each test point, steady-state mass flow, power, and efficiency were measured or calculated, any of these variables could be interpolated as required. Since some of the unsteady cases produced operating conditions outside the bounds of the steady-flow data, some extrapolation was necessary. Fortunately, there was very little unsteady operation that was not covered by the steady data considerably reducing the uncertainty of the analysis. The validity of the interpolation was checked against the original steady efficiency data to assure consistency. The interpolated data yielded a root mean square deviation of 0.3% from the measured points, thus providing sufficient confidence in the method. From this analysis it is possible to select a point in time during an unsteady pulse and compare it to the steady operation under the same condition, giving quantitative indication of departure from quasi-steady operation. This quasisteady analysis can then be integrated over a pulse cycle to test its validity in predicting overall unsteady performance.
6 Steady-state efficiency contours, with unsteady operating traces superimposed, (a) 3D representation (b) equal admission perspective (c) unequal admission perspective (in-phase)

7 Steady-state efficiency contours, with unsteady operating traces superimposed, (a) 3D representation (b) equal admission perspective (c) unequal admission perspective (out-of-phase)

8 (a) insteady [US] and quasi-steady [QS] cycle-averaged efficiency (b-c) unsteady/quasisteady ratios of averaged mass flow and power ($\dot{m}$ and $P$), (b) in-phase (c) out-of-phase
Figure 8a illustrates the values of unsteady cycle-averaged efficiency calculated using Equation 9 for the conditions listed in Table 1. In the evaluation of these average values, it was not necessary to shift any measurements in time. That is, the quantities are averaged first and then divided; thus providing a value of unsteady turbine efficiency regardless of the dynamics occurring within the stage. Quasi-steady efficiency was also calculated in a similar manner by substituting the interpolated steady values into Equation 9 that match the instantaneous unsteady conditions. Bearing in mind that the values of equal to 10, 15 and 20 represent frequencies 84Hz, 56Hz, and 42Hz respectively, it is apparent from this figure that there is a reduction in the unsteady efficiency of the turbine as the frequency decreases. Although the quasi-steady efficiency shows a similar trend, clearly the degree of the decrease is much less than the measured unsteady values.

To understand the source of these efficiency trends, the un-steady values of mass flow and power were time-averaged and compared to the corresponding time-averaged quasi-steady values in Figures 8b and 8c using parameters $I_M$ and $I_P$, presented in the Introduction to this paper. With the exception of mass flow at 84 Hz, these figures show that the quasi-steady averages generally under-predict the corresponding unsteady values. Similar to the results of Kosuge et al. [6], there appears to be a contrasting influence of pulse frequency on the output power parameter $I_P$ and the mass flow parameter $I_M$. The former decreases with decreasing frequency, while the latter increases. Thus one might explain the reason for the low unsteady efficiency at the low frequencies (high lambda) by the high mass flow rate of the inlet pulse when compared to the quasi-steady value; the turbine then seems unable to convert this energy into output power.

### Quasi-steady Interpretation

Although the pattern of decreasing cycle-averaged efficiency at lower frequency has been measured by other researchers [16,24,25], it might seem somewhat counterintuitive at first. It could be supposed that the increase in flow unsteadiness associated with higher frequency should lead to a greater degree of loss. The Strouhal number, defined by Equation 3, can be used as a dimensionless criterion to define the onset of unsteady effects in a flow. Table 2 lists the modified Strouhal numbers (MSt) of the three in-phase conditions using the velocity of the bulk flow and the length of each of the two inlet passages. If the velocity of the pressure wave propagation is used in place of the bulk flow, the Pressure Modified Strouhal number (PMSt) results. Based on experimental data, Szymko et al. [16] suggested a value of PMSt $\approx 0.1$ as an upper limit to steadiness. Referring to Table 2, by this criterion the pressure field can only be considered unsteady at $\lambda = 10$ and 15 in the longer outer entry. This is corroborated by the additional features in the 84 Hz pressure pulse shape postulated earlier as arising from the dynamic behaviour of the pressure wave in the outer limb. Yet, how does one reconcile the fact that the increase in pulse frequency produces significant flow unsteadiness caused by wave action in the volute, with the fact that the turbine also seems to operate more efficiently? To start, perhaps it is useful to clarify what is often meant by ‘unsteadiness’. This term has been used to denote a condition in which the frequency of the incident waveform is high enough to produce significant pressure wave dissemination resulting from the superposition of incident and reflected waves. That is, where wave dynamics (a term that is frequently used in this paper) gain significance thus making a simple ‘filling and emptying’ approach inappropriate due to the importance of momentum continuity. This should lead to a condition where the pressure field is significantly removed from its steady-state

<table>
<thead>
<tr>
<th>Frequency</th>
<th>n/f</th>
<th>Modified Strouhal (MSt)</th>
<th>Pressure modified Strouhal (PMSt)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L=0.24$m (inner entry–measurement plane to rotor inlet)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>42.2</td>
<td>20</td>
<td>0.146</td>
<td>0.032</td>
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<tr>
<td>56.5</td>
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<td>0.200</td>
<td>0.044</td>
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<td>10</td>
<td>0.304</td>
<td>0.056</td>
</tr>
<tr>
<td>$L=0.54$m (outer entry–measurement plane to rotor inlet)</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>42.2</td>
<td>20</td>
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<tr>
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</tr>
<tr>
<td>83.7</td>
<td>10</td>
<td>0.683</td>
<td>0.146</td>
</tr>
</tbody>
</table>

Table 2: Strouhal numbers (in-phase)
relationship with the mass flow in the volute. Costall et. al. [26] made an interesting contribution by defining an unsteadiness criterion based on the deviation of the gradient between the steady and unsteady MFP vs PR curves. However, as pointed out earlier, while lower frequencies might have less unsteadiness, there is a greater variation of instantaneous quantities revealed by the amplitude of the unsteady traces shown in Figures 6 and 7. This would seem to suggest that these large amplitude variations (i.e. low frequency) have a greater influence on the loss in the turbine than signified by pressure wave dynamics (i.e. high frequency).

The quasi-steady efficiency values plotted in Figure 8a seem to substantiate this interpretation by also showing a performance penalty with reducing frequency albeit quite a minor one. This trend is not surprising considering the behaviour of the traces in Figures 6 and 7. The high frequency orbits largely remain in an area where peak efficiency is expected from steady considerations. However, as frequency is reduced, the operating point starts to encroach on regions of increasing loss (lower efficiency). Despite the similar trend, there is clearly a large discrepancy between the measured unsteady efficiency loss and that predicted by the quasi-steady assumption. To demonstrate this point, the instantaneous input and output power used to calculate the cycle-averaged efficiency have been plotted in Figure 10. First of all, it is worth noting that the 42 Hz output power derived from the unsteady torque measurement closely resembles the quasisteady output power resulting from interpolated steady flow values. This suggests that the method of phasing is correct, and that the wave dynamics have a low importance at this frequency. The measured output power trace no longer matches the quasi-steady behaviour at higher frequencies, signifying a more complex situation under these conditions.

10 In-phase, isentropic and actual power using unsteady and quasi-steady data
As indicated earlier in Figures 8b and 8c, the quasi-steady analysis under-predicts the energy of the input pulse. Since the quasi-steady data is derived by matching the steady and unsteady pressure and velocity ratios, the difference in input power must be due to a difference in mass flow. Thus, Figure 11 was produced to enable a comparison between the mass flow measurement and the quasi-steady mass flow interpolated from equivalent conditions. In a sense, this proposes the following thought experiment. At an instant in the pulse cycle, if the same entry pressure ratios are applied to a turbine operating with the same velocity ratio and allowed to reach steady-state, what would the swallowing capacity be? Figure 11 shows that the swallowing capacity of this hypothetical quasi-steady case is much less than the real mass flow measurement at the peak of the pulse. This difference can be attributed to the filling and emptying of the turbine volume. Upon the arrival of an unsteady pulse, the volute volume will have had time to empty some of the mass it accumulated from the previous pulse. Thus, the turbine can swallow more mass compared to the steady-state case in order to fill this ‘empty’ volume. The influence of frequency further complicates matters. As pulse frequency increases, the time available to empty the volume through the nozzles/rotor gets shorter, meaning one of the following may occur:

1. To maintain the same amount of filling and emptying in a shorter time span, the mass flow through the nozzles must increase. However, this will be limited by the flow capacity of the nozzle.
2. The amount of filling and emptying decreases - leading to less of a variation of mass contained in the volute. This more ‘constant’ amount of mass contained in the volume could be considered closer to steady-state operation.

Although a deeper analysis of this phenomenon is necessary before further conclusions are made, the above interpretation suggests a close link between the pulse frequency, the nozzle swallowing capacity, and the significance of the filling and emptying. Applying this line of reasoning even further, the following series of events can be suggested from Figure 11. At a lower frequency, the turbine appears to have sufficient time to swallow the mass introduced by the pulse. This can be noted by the high peak mass flow filling the volute (A) and subsequent close match between the quasi-steady and unsteady mass flows after the peak of the pulse (B). As already mentioned, as the frequency increases, the time available to swallow all of the mass introduced by the pulse is less, which may result in a volume with a more constant amount of mass. Thus, the arrival of a pulse introduces more mass into this already ‘full’ volume.

11 In-phase, unsteady mass flow versus quasi-steady mass flow (outer entry)
than the nozzle opening is able to swallow. This is suggested by the plateau in the mass flow peak at 84Hz (C), followed by the quick drop in mass flow below the quasi-steady line (D). As Figure 8(b,c) demonstrates, the end result is an unsteady, averaged mass flow value that is closer to the quasi-steady value ($I_{n, \rightarrow 1}$) at higher frequencies. Thus, although the above interpretation of the mass flow traces is clearly speculative, whatever the interpretation, it is apparent that the turbine is able to swallow more mass than the quasi-steady assumption predicts at lower frequencies but does not translate into greater shaft power. Given the distance between the turbine volute entrance and the rotor inlet, there is a time delay between the two planes. This means that the turbine power developed cannot be directly compared with the available power measured at the turbine volute inlet.

Thus far, the issue of out-of-phase unsteady pulsations has not been explicitly considered. It is important since it is the condition most representative of the true unsteady flow supplied to the turbine by an internal combustion engine. Figure 7 (a-c) illustrates the unsteady operation of the turbine when the pulses arrive at the turbine inlet 180° out-of-phase. This results in the two sectors of the nozzle ring feeding the turbine with two very different flow conditions for most of the pulse cycle. Although some mixing is expected in the nozzle-wheel interspace, a rotor passage will thus experience a periodic change in flow conditions as it rotates from one sector to the next. Further more, Figure 7c shows that at the peak of the pulse, the turbine is operating in a region corresponding to decreased performance due to unequal admission which should result in a much lower quasi-steady efficiency value. This is indeed the case as shown in Figure 8a where the out-of-phase quasi-steady efficiency is 3 – 6 percentage points lower than the in-phase quasi-steady prediction. Bearing in mind that the turbine wheel rotates 10-20 times over a pulse cycle, it seems reasonable to expect that the out-of-phase, unsteady efficiency should similarly suffer. Interestingly however, the measured out-of-phase values show very little difference from the inphase unsteady efficiency. This would seem to suggest that the losses associated with steady, unequal admission are not as prevalent in conditions of steady, out-of-phase admission.

The work presented in this paper will be complemented by future work which will include a detailed analysis of the flow conditions in steady, unequal and unsteady admission using 3D CFD methods. The major emphasis of this work is the postulation of loss mechanisms that can explain the complex energy conversion and efficiency traces presented in this paper. The work will then move to the suggestion of a more realistic 1D model of the turbine for implementation in engine gas dynamics matching codes. The understanding of the loss of performance due to unsteadiness based on physical understanding will be most valuable in the development of the turbine model, thus removing the need to “downgrade” the turbine performance by a single penalty unsteady factor. This understanding will open the way for the selection of the turbocharger with consideration of unsteady flow phenomena.

Conclusions
A range of experimental results from a double-entry, circumferentially divided turbocharger turbine have been presented in this paper. Building on the work reported in [19], additional unequal admission, steady-state data was measured at velocity ratios equal to 0.5 and 0.8. The symmetry of this new data strengthened the conclusion that the performance loss due to unequal flows in a double-entry arrangement is largely independent of which inlet condition is varied, but correlated to the ratios of pressure between them.

A total of six unsteady conditions were presented in this work at a constant cycle-averaged velocity ratio of 0.65. By maintaining a constant 85% wheel speed, the effect of three different pulse frequencies introduced both in-phase and out-of-phase was investigated. The increase in pulse frequency had two visible effects on the unsteady operating traces. First, the amplitude of variation for both pressure ratio and velocity ratio decreased with increasing frequency, resulting in a more compact unsteady orbit. Second, there appeared to be an increase in the pressure wave unsteadiness resulting from the increasing influence of wave dynamics in the volute passages. Due to the two different passage lengths, the effect of this pressure wave interaction was noted to differ for each entry, resulting in an increasingly asymmetric operating trace at higher frequency.

The input power integrated over a pulse cycle was divided by the shaft power to give a value of the cycle-averaged efficiency. Plotting the results against frequency factor $\lambda = n/f$ demonstrated that the unsteady efficiency of the turbine suffered most at the lowest frequency. Using interpolated steady-state data, the averaged quasisteady efficiency was also calculated and compared to the unsteady values. Despite displaying a similar decreasing trend, the quasi-steady efficiency could not replicate the magnitude of unsteady efficiency drop with frequency. The reason for this departure at lower frequency was investigated further by calculating the quotient of unsteady and quasi-steady, time averaged mass flow and output power ($I_{u, n}$ and $I_{p}$).
These parameters showed that the quasi-steady assumption underpredicted the magnitude of mass flow, especially at lower frequency. Therefore, the energy of the input waveform was greater than the quasi-steady value at lower frequency, but this did not result in more output power.

For further understanding, the waveforms of isentropic input power and mass flow were plotted in time. It was apparent from these plots that the isentropic power measured from the instantaneous inlet quantities was greater than the quasi-steady equivalent at the peak of the pulse. This difference could only be attributed to a difference in mass flow, which was conjectured to result from the effect of 'filling and emptying'. Although somewhat speculative, the behaviour of the mass flow was explained in the light of the influence of pulse frequency, volume effects, and the nozzle flow capacity. It was suggested that at lower frequencies, the time available to swallow the mass introduced from a pulse cycle is greater, thus encouraging a greater flow field variation in the volute. At higher frequencies, the time for the nozzle and turbine to accept the mass contained in a pulse is less, perhaps resulting in a volume that exhibits less of a variation of total mass in the volume. In conclusion, the experimental data has demonstrated that a significant efficiency loss can be attributed to high amplitude, low frequency exhaust pulsations.

Since one of the main objectives of pulse turbocharging is to preserve as much of the exhaust gas energy to the turbocharger inlet as possible, it is important to assess whether or not the turbocharger turbine can then convert this energy into useful power supplied to the compressor. The results presented here suggest that despite the increase of pressure wave unsteadiness, the turbine is better able to convert high frequency, lower amplitude pulsations into usable power. It also suggests that while the quasi-steady assumption can reflect some of the rough trends of the unsteady performance, it is not capable of giving quantitative predictions of the turbine efficiency with unsteady admission.

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References


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Contact us

For more information please contact:

ABB Ltd
Affolternstrasse 44
CH-8050 Zurich, Switzerland

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