Turbocharger turbine pulse flow performance and modelling – 25 years on

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ABSTRACT
The difference in turbine performance observed between tests on a gas stand and operation on the engine has long been recognized, and has been attributed at least in part to the pulsating nature of the engine exhaust gas, as compared with the steady flow conditions of a gas stand. Efforts to understand the influence of inlet flow pulsations are limited by the experimental difficulties in measuring unsteady flow performance and fluid flow. Even today, very few facilities exist where this work can be done. Increasingly, however, it is being augmented by unsteady flow analysis using CFD. Attempts are also being made to develop analysis methods that are suitable for use in engine simulations. In this paper, an historical survey is used to give insight into the present understanding of pulse flow in a turbine, the factors that influence the turbine performance, and the benefits of various modelling methods. Methods of quantifying pulse flow performance are also discussed.

1. INTRODUCTION
It is now almost 25 years since the very first comprehensive study of the performance of a radial turbine under pulsating flow conditions was published, and this seems an appropriate time to take stock of progress towards the goals of being able to understand pulse flow performance, and of modelling the turbine as part of a turbocharged engine simulation.

2. DEVELOPMENTS IN THE UNDERSTANDING OF PULSE FLOW PERFORMANCE
Early efforts to understand the influence of inlet flow pulsations were limited by the experimental difficulties in measuring unsteady flow performance and fluid flow. It was normally possible to measure only the inlet pressure as a time-varying quantity, while the mass flow rate and power output were measured as time-mean quantities. Perhaps not surprisingly, therefore, the comparisons of steady and pulse flow capacity and efficiency reported by various authors are quite contradictory. Moreover, such results do not necessarily reflect only a change in efficiency of the turbine under pulse flow conditions, but also the available energy from the engine exhaust due to the high pressure ratios associated with the pulse peaks (1, 2).

2.1 Zero-dimensional (0D) models
The first serious attempt to understand this problem by testing a turbine under simulated engine exhaust pulsations, with all necessary parameters measured on a time-varying basis, was done by Dale and Watson (2, 3) at Imperial College, London, and continued by Baines and co-workers (4–6). This program has been summarized elsewhere (7) and will not be repeated in detail. The major findings may be summarized as follows:

1. There are differences in flow capacity and aerodynamic efficiency of a turbine tested under steady and pulsating flow conditions (Figure 1). These differences clearly demonstrate that under pulse flow, the turbine does not operate in a quasi-steady manner.

2. The fluid state and velocity measured at the inlet to the rotor during steady flow, and at the same, turbine inlet conditions encountered instantaneously during pulse flow, are very similar (Figure 2). This strongly suggests that the rotor component of the turbine is operating in a quasi-steady manner during pulse flow. The time-varying effects must occur principally, and possibly exclusively, in the stator components of the turbine.

3. An attempt was made to model the pulse flow performance of the turbine by treating the volute as a volume that introduces a residence time into the gas flow. This is the "filling and emptying" or zero-dimensional approach to unsteady flow modelling. The rotor was treated as quasi-steady, and conventional steady flow one-dimensional modelling was used in this component. Some success in predicting the performance of a single-entry, nozzleless turbine was achieved by this method (Figure 3).

An effort was made to extend this modelling approach to a twin-entry volute (8), but this was not pursued far. The zero-dimensional approach was originally used for engine system modelling but has now been surpassed by solutions of the one-dimensional unsteady gas dynamic equations, which are able to predict the spatial and time variations in the fluid state and velocity in pipes and manifolds more successfully than 0D methods, as shown, for example, in (9). Since the turbocharger turbine is a component of the engine exhaust system, it would seem appropriate to use a consistent methodology for the complete system, for better modelling, and also for mathematical and numerical consistency. The latter argument, of course, only has force providing it can be demonstrated that the one-dimensional (1D) unsteady method is capable of predicting the turbine performance to a necessary standard of accuracy.

2.2 Simple one-dimensional models

Winterbone developed a similar pulse flow test rig to the Imperial College facility (10, 11) and was able to measure the turbine flow capacity and efficiency under pulse flow. No internal velocity measurements were reported, but the pressure at the turbine exit was measured and showed a small but significant variation through the pulse cycle. This is not inconsistent with the quasi-steady rotor finding, but does indicate that care must also be taken with the downstream boundary condition for any turbine model.

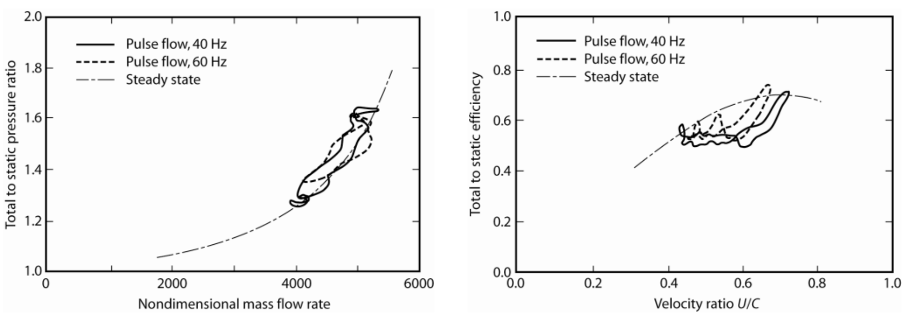


Figure 1. Turbine performance measured under steady and unsteady flow conditions, at two pulse frequencies (3)

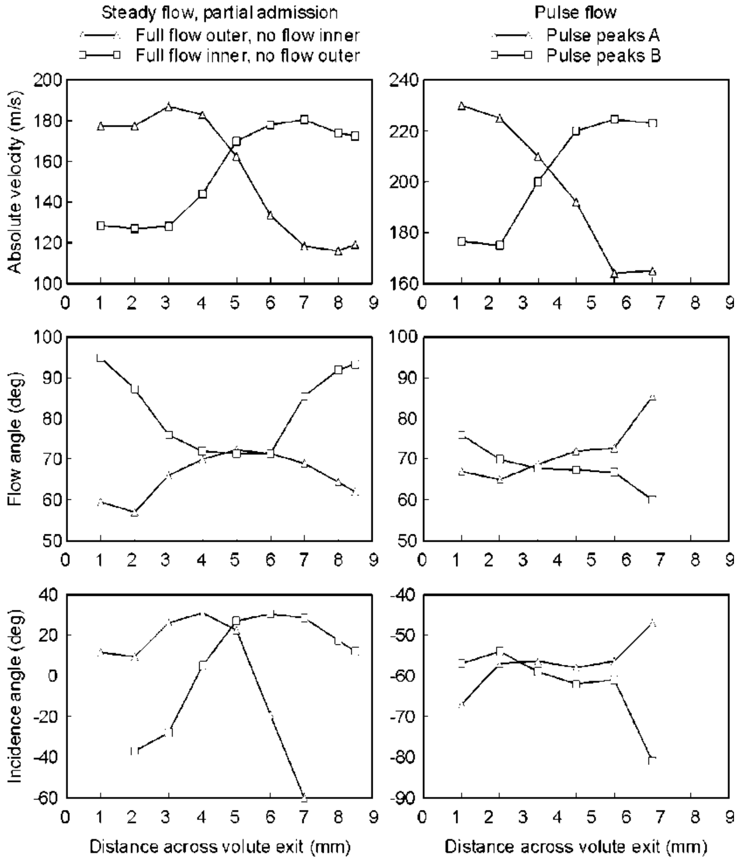


Figure 2. Comparisons of velocity traverses at rotor inlet for a twin-entry nozzleless turbine, showing (left) steady flow at the extremes of partial admission, and (right) instantaneous pulse flow conditions that correspond to the steady flow conditions (i.e., pulse maximum at one inlet and minimum at the other). Based on data from (4, 5)

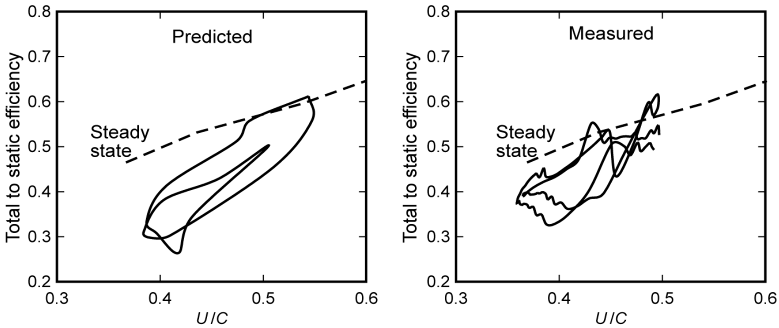


Figure 3. Comparison of measured efficiency in pulse flow, and predictions using a 0D volute model and quasi-steady rotor model. Single-entry, nozzleless turbine (6)

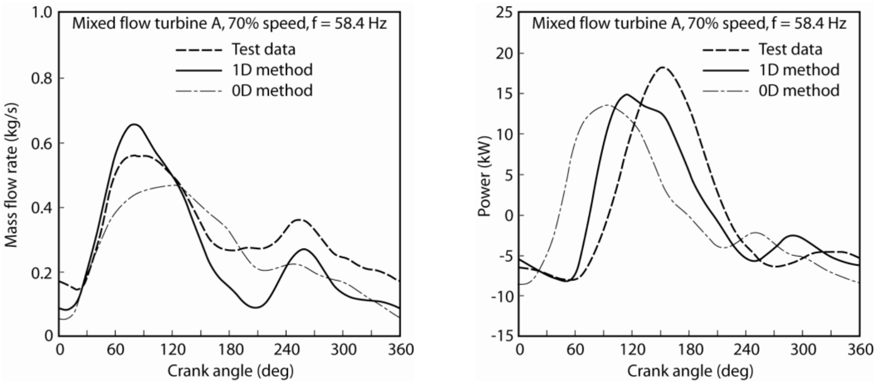


Figure 4. Turbine performance through one pulse cycle. Comparison of measurement with 0D and 1D model predictions. Based on Chen *et al.* (13)

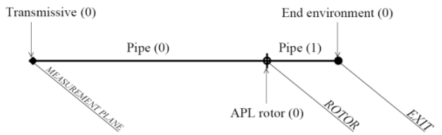
Based on this work, Chen (12, 13) developed a model of a nozzleless turbine, by treating the volute as a tapered pipe of a certain length, and solving the 1D unsteady gas dynamic equations for this pipe.

The rotor was assumed to be quasi-steady and was modelled as such. The length of the pipe was taken to be the path length of the centres of area of the volute sections from the tip of the tongue to a point at 180° removed from the tongue. The pipe area was set to give the correct volute volume. The results shown in Figure 4 do indicate that this model is more successful than the 0D model at reproducing the turbine performance characteristics, although some significant differences remain.

This method was developed by Abidat *et al.* (14), who retained the quasi-steady rotor, but modelled the volute as a curved pipe whose inlet section is the volute section at the tip of the tongue, whose exit section is at a radius equal to the mean of the volute section centroid radii and at the azimuth angle at which this occurs, and whose area is equal to the section area at this point. Intermediate radii are set equal to the centroid radii of the volute sections, and the area of the curved pipe is assumed to vary linearly with azimuth angle.

The curved pipe is divided into a network of finite volumes and the one-dimensional unsteady equations of flow, momentum, and energy are discretized and solved using a second-order accurate solution method (as compared with the Chen method that is only first order accurate). Only a limited amount of information was published, but the results do suggest slightly better prediction of mass flow rate than those shown in Figure 4. The model was also used to investigate the effect of pulse frequency and amplitude. The results suggest that the pulse frequency has no significant influence on the average mass flow rate, but there is a clear effect of pulse amplitude, and a decreasing mass flow rate with increasing amplitude. The turbine output power, however, increases with both pulse amplitude and frequency. These findings have not been properly checked against test data.

This modelling approach was adopted and developed by Costall *et al*. (15, 16). Figure 5 shows the one-dimensional model used to simulate pulse flow in a single-entry turbocharger turbine. Pipe (0) represents the total volume of the test section and turbine housing between the measurement plane and the rotor inlet station. Though the form of this 1D duct is far removed from that of a volute, the objective was to construct the simplest model possible that could resolve filling-and-emptying and wave action effects. Its length is set from the measurement plane to a point 180° downstream of the volute tongue, following the work of Chen (13). Costall



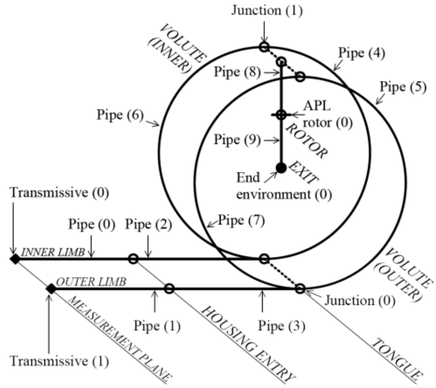


Figure 5. Single-entry 1D unsteady flow model (15)

Figure 6. Twin-entry 1D unsteady flow model (16)

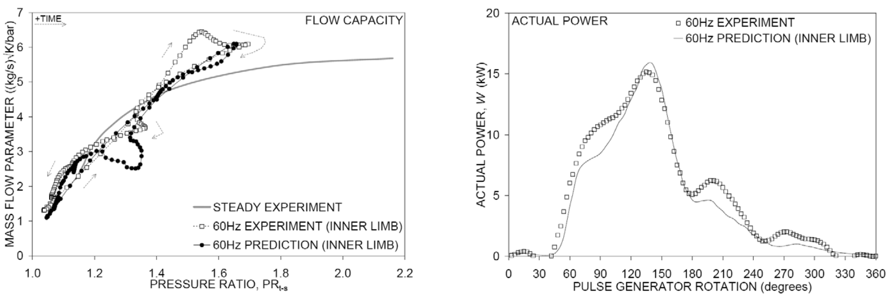


Figure 7. Single-entry turbine predicted and measured unsteady mass flow rate and power at 60 Hz pulse frequency (16)

argues (16) that the end point should ideally be based on mass flow consumption, e.g., so that half of the steady mass flow is consumed within the chosen volute azimuth angle. This would be much harder to implement because the distribution of mass flow about the periphery of the volute exit is not known initially.

Pipe (0) has a constant diameter such that its volume matches the total volume between the measurement plane and the rotor inlet. The lack of curvature along this duct is the most obvious limitation, given that it is intended to represent a volute. To simplify model calibration, the entire stage pressure drop was assumed to be concentrated at the rotor boundary. The pressure loss is modelled as a function of rotor speed and exit Mach number, but must be calibrated using steady-state test data. Pipe (1) represents the volume of the rotor passages and the remaining volume of the exhaust diffuser or pipe before the stage exhausts to an open pipe end boundary. Its length and volume are based on the test setup.

Figure 6 shows a development of this model for a twin-entry turbine. Stator losses are admitted at Junctions 0 and 1, in addition to the rotor loss. Figure 7 illustrates some results for the single-entry model, showing good agreement. Predictions for a twin-entry turbine are at present less satisfactory, but further work is in progress.

The prediction accuracy of methods such as this also relies on the existence and quality of appropriate test data to model the quasi-steady rotor. In turbocharger turbines, even if this data is available, it often covers only a small range of operation. In practice considerable extrapolation is usually necessary (just as it is when modelling engine operation using steady-state turbine characteristics), introducing additional uncertainty. Alternatively, results of a steady-state turbine simulation may be used as a basis. Such simulations are typically one-dimensional, and also require input of suitable loss data or correlations, such as those described in (17–19), are generally couched in terms of accessible fluid dynamic or thermodynamic parameters such as total pressure or enthalpy loss.

2.3 Complex one-dimensional models

One-dimensional unsteady flow models of the complete turbine, including the rotor, have been developed by Hu (20–22) and King (23), based on the work of Ehrlich (24, 25). The volute is discretized into a set of streamwise elements, so that the flow field is solved along a mean streamline. Each nozzle passage is treated as a separate element, and the inlet conditions to each passage are determined by interpolation of the adjacent volute stations. Each rotor passage is simplified into a radial and an axial element. The rotor passage flow is solved in the relative frame, while using an absolute frame for the stator components, basically a "sliding plane" approach. This avoids the need to add artificial terms to the equations to account for the lift force on the blade and work transfer in the rotor. Mathematically it means that a consistent set of equations and solution can be applied to all components of the turbine. Treating each nozzle and rotor passage separately also means that the model is capable of calculating the one-dimensional time accurate flow field within individual rotor passages.

The model suffers a number of limitations, some more fundamental than others. For nozzleless turbines it is necessary to assume a constant rotor inlet absolute flow angle, which is tantamount to including a nozzle component. A twin-entry turbine is handled in a very simple manner, essentially as two turbines working in parallel with no mixing of the two streams. It would appear that both these objections could be overcome with suitable developments of the discretization and the transfer of information between components.

A more fundamental problem is in the handling of losses. The model solves the 1D Euler equations, and non-isentropic effects are introduced as centrifugal body and surface forces. In (20–23), suitable values of these forces were obtained by comparing the measured steady-state mass flow rate and efficiency of the turbine with their isentropic values. For a comprehensive accounting of losses, it would be necessary to have suitable test data for each component of the turbine, but in practice this is rarely available and usually only the overall stage performance is measured. The problem of ascribing a fraction of the loss to each component was solved in very simple fashion by assuming that all of the loss occurs in the rotor.

The data on turbine steady-state loss and efficiency, whether from experiment or modelling, is generally couched in terms of accessible fluid dynamic or thermodynamic parameters such as total pressure or enthalpy loss. To make such information available to this unsteady flow model, it must first be converted into body and surface forces, which in turn requires considerable geometric information about the turbine. Such force terms are far from intuitive and outside general experience, so that engineering designers will not find it easy to judge whether values and solutions are sensible.

2.4 Three-dimensional models

In principle, any problem in pulsating flow can be modelled accurately and solved by means of the time-varying Navier-Stokes equations, and this has been attempted by a number of researchers. The first published reference is Lam *et al.* (26), which gives considerable detail of the approach taken. A complete turbine stage with a straight inlet duct was modelled using multiple reference frames (MRF). Here the domain is divided into stationary and rotating subdomains, and no account is taken for the relative motion of one subdomain with respect to the other.

This is the "frozen rotor" method, and the authors justified its use by noting that the pulse frequency is very much lower than the rotor passing frequency.

No comparisons with pulse flow test data were shown, although the general form of the predicted time-varying flow rate and torque output was consistent with that seen elsewhere. The results indicated that the volute has a significant damping effect on the variation of mass flow rate with time, which is consistent with temporary mass storage occurring in the volume. Little damping, however, occurs to the variation of total temperature. The instantaneous efficiency of the rotor alone deviates to only a small extent from the steady-state efficiency (Figure 8) during the pulse cycle. At first sight, this would appear consistent with quasi-steady flow and performance of the rotor, but the use of a "frozen rotor" computing model might predicate this result, and further investigations appear to be warranted.

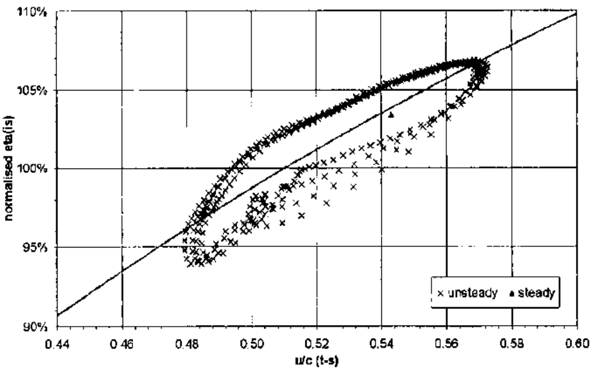


Figure 8. Predicted variation of instantaneous rotor efficiency through a pulse cycle, compared with steady-state efficiency (26)

In the work of Palfreyman and Martinez-Botas (27), the turbine rotor was explicitly rotated during the calculation, the stationary and rotating meshes being coupled at each instant in time through a sliding plane at the stator-rotor interface. Comparisons with measured unsteady efficiency and mass flow rate are shown in Figure 9. The general form of the efficiency variation through the pulse cycle is reproduced although there are significant differences in magnitude over much of the cycle. The mass flow rate is much less well predicted, although this might be the effect of a different location of measurement.

The differences between steady and unsteady efficiencies in this study are much greater than predicted in (26). This may be due to the choice of sliding plane rather than a frozen rotor model, but as Hellström (31) points out, Palfreyman and Martinez-Botas modelled a nozzleless turbine, whereas Lam *et al.* modelled a turbine with a nozzle, which might reduce the variation of flow conditions at rotor inlet during the engine cycle.

In Figure 9 it will be noticed that the unsteady flow efficiency is predicted to rise far above 1.0, and below 0.0, in the course of the engine cycle. These are physically unrealistic results that stem from the misapplication of the conventional definition of isentropic efficiency. This matter is discussed in greater depth in Section 4.

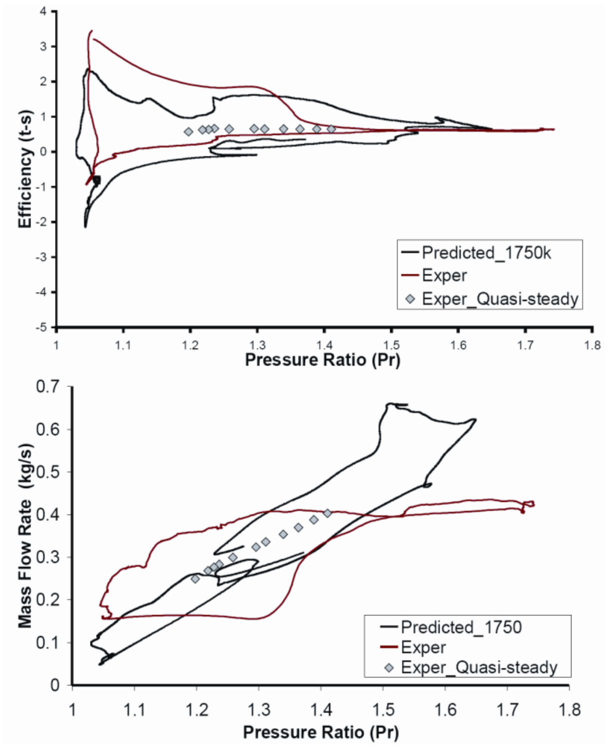


Figure 9. Comparison of turbine performance through a pulse cycle (27)

An interesting point is made in (27) concerning the calculation of torque, and hence power and efficiency, using this method. The torque exerted by the rotor is calculated by integrating the pressure forces on the blades, which results in a very unsteady curve of torque against time through the pulse cycle. In reality, the measured shaft torque is damped by blade displacement, damping of the bearing system, and inertia of the rotating assembly.

The CFD-based analysis of Abidat and Hachemi (28) is described in much less detail. A comparison of the inlet mass flow rate with test data shows agreement that is not significantly better than 1D approaches (Figure 10). The variations of mass flow rate with time at the rotor inlet and exit were predicted to be very similar, and the authors argue that this implies that the rotor is quasi-steady.

The 3D CFD analysis of a nozzleless turbine by Hellström and Fuchs (29–31) is without supporting test data, but nevertheless achieves some interesting results that shed light on some of the problems of this form of modelling. The authors argue that in pulse flow, the large scales of the turbulence are important, and this is handled by large eddy simulation (LES); explicitly computing the large turbulence scales, while modelling the effects of smaller scales. A major role of the small scales is to dissipate the turbulent energy that is transferred (in the average sense) from the larger scales to the small ones through the so-called energy cascade. By resolving the large scales, coherent structures can be separated from the random turbulence eddies. A sliding mesh is used to model the stator-rotor interface.

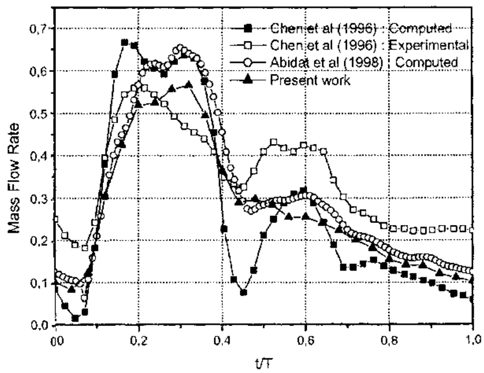


Figure 10. Comparison of instantaneous inlet mass flow rate, as measured, and using three different calculation methods (28)

In (29), two different pulse frequencies were investigated for the same mass flow and temperature at the inlet to the turbine. The effects of a ten times higher frequency was that the maximum shaft power was slightly lower (due to a larger incidence angle at this point in the cycle), but during the deceleration phase, the shaft power was instantaneously higher. This suggests that the relationship between pulse frequency and output power is more complicated than the results of Abidat *et al.* (14) suggest.

In (30), a variety of non-pulsatile boundary conditions was used at inlet to the volute, including uniform flow with and without turbulence, and several combinations of turbulence with swirl and streamwise vortices. The undisturbed inlet profile gave the highest time-averaged power output, with a variation of nearly 20% in time-averaged power across all cases predicted. The authors attribute this to a lower bulk flow velocity through the wheel due to flow perturbations, inlet swirl giving rise to vortices in the volute and unfavourable conditions at inlet to the wheel, and the wake of the tongue being enhanced by inlet vortices. These results demonstrate the important effect that fine detail of the turbine inlet conditions has on the predicted turbine performance, but do require confirmation by experiment.

3. FLUID DYNAMICS OF PULSE FLOW IN A TURBINE

A common theme in much of the early literature on this subject was the question of whether the effects of pulse flow are caused by the action of pressure waves or the convection of mass flow through the turbine. This understanding should have an important bearing on the choice of modelling technique, because if it can be shown that pressure wave effects dominate, this strongly indicates that unsteady flow modelling is necessary, whereas if convection is more important, a simpler zero-dimensional approach might be adequate for the purpose. This question can be answered if sufficient detail of the time-varying flow field is available so that the phase relationships of different events can give insight into the time differences between causes and effects. In practice the early researchers were occupied with the considerable problems of measuring the overall performance of the turbine stage under pulse flow conditions without the extra complications of flow field measurements. In the circumstances, the only phase relationship generally available was between the incoming pressure, measured upstream of the turbine inlet flange, and the torque developed at the turbine shaft. There was, unfortunately, sufficient ambiguity in the various results obtained that no definitive answer emerged at that time.

Winterbone and Pearson (32) discuss the comparison of pressures measured at the inlet flange of the turbine and some distance downstream in the exhaust pipe. The pressure wave is considerably reduced in amplitude, but not completely damped, in its passage through the turbine, but the phase relationship of the two traces is consistent with pressure wave motion without any additional delay in the turbine. A more detailed set of static pressure measurements at a number of points upstream of the inlet flange and at various angular distances around the volute were made by Karamanis *et al.* (33) which were again consistent with wave propagation.

Measurements of the inlet and exit conditions to a turbocharger turbine operated on-engine rather than in a test rig were made by Ehrlich (21, 22). In this study, significant temperature variations were evident over the pulse cycle, which would not be present in cold flow experiments. Both the temperature and pressure fluctuations at the turbine inlet were as much as 40% of the mean values, and gave rise to significant density variations. Figure 11 compares the mass flows in and out of the turbine at the same instants in time, and shows large momentary mass storage and expulsion in the turbine. The results of this study suggest that the pressure propagates by wave action, but because the temperature variations are larger than would be expected from a compression wave, convection of the hot gas also occurs, and the results shown in Figure 11 are a combination of these effects.

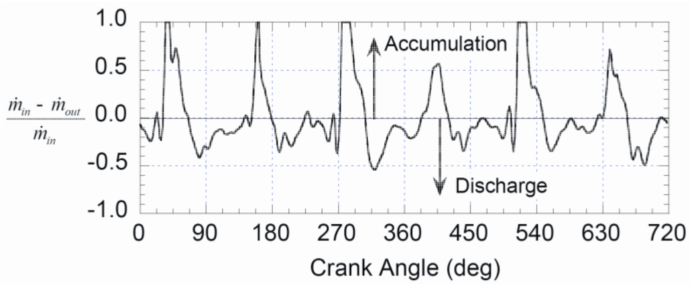


Figure 11. Unsteady mass storage in a turbine through one engine cycle (22)

An equally important question is whether the rotor can be treated as quasi-steady, as is assumed in the zero and simple one-dimensional models. Even those one-dimensional models that treat the rotor in a time-varying manner, such as Hu (17–19) and Costall *et al.* (15), are not truly unsteady, in that they use steady-state data as a basis for simulating the rotor losses. The only experimental evidence in support of this is that of Baines and Yeo (3), who demonstrated the similarities in rotor inlet velocity profiles under steady and pulsating flows, shown in Figure 2. Some limited corroboration also comes from the unsteady CFD results of Lam *et al.* (26) shown in Figure 8, and Abidat and Hachemi (31).

For many practical purposes, quasi-steady behaviour of the rotor is a necessary assumption in order to produce a workable model suitable for industrial design and analysis requirements. No test data, simple models, or correlations exist (nor are they likely to become available in the near future) for unsteady flow in a radial turbine rotor. It is thus probable that one-dimensional unsteady flow models, incorporating a quasi-steady rotor model, will continue to be of importance in such applications. Three-dimensional unsteady CFD models are capable of simulating both the convected and wave effects in all parts of the turbine, but the ability to predict the turbine performance by such methods is currently outstripping the availability of test data to corroborate them. Even with modern developments in computing power, they still require large computing resources and run times.

4. THE REPRESENTATION OF PULSE FLOW PERFORMANCE

Modern engine simulation programs work by discretizing the engine cycle into small intervals of crank angle, and thus need to access the turbine performance many times during a cycle. This is normally done using a turbine map or look-up table, whose primary parameters are mass flow rate, pressure ratio, speed, and efficiency. To represent the effects of pulse flow when using a steady-state map obtained from gas stand testing, empirical pulse flow correction factors of mass flow rate and efficiency may be used. These are normally obtained by comparing the performance of the turbine on the gas stand and on the engine (34).

The models described here could, at least in principle, be used to calculate similar correction factors, by integrating the instantaneous mass flow and power output through the cycle (35), but such an approach is inefficient and inconvenient. The boundary conditions of pressure and temperature at the turbine inlet, and pressure at the turbine exit, required for any turbine model, vary through the engine cycle and must be obtained from the engine simulation. Thus the procedure becomes one of iterating between the engine simulation and the turbine pulse flow model until all of the cycle-averaged parameters have converged. Furthermore, the problem is not as simple as stated, because as discussed in Section 2, the available evidence indicates that the turbine flow development is a dynamic process, and the effects of pressure waves and mass storage in the turbine are such that the flow capacity and power developed by the turbine at any instant in time are not simple functions of the boundary conditions at that moment, but depend on the history of the boundary conditions for some time previous to the moment of calculation.

Thus it can be seen that a "pulse flow performance map" for a turbocharger turbine, that is sometimes sought in the hope that it could be used as a direct replacement for the current steady flow maps, can only be an approximation at best. The instantaneous turbine performance is determined by a history of prior events, and for true realism it is necessary to consider a pulse flow turbine model working within a larger simulation of the engine exhaust (or the complete engine, or powertrain), operating dynamically to predict the turbine performance at each moment in time through the engine cycle.

It is therefore necessary to decide how such a model would interact with the engine simulation, because this will not be the same as the simple map or table look-up used with steady-state turbine data. The outputs of a conventional steady-state turbine model, and of turbine test data, under circumstances where the inlet conditions are defined and the exhaust pressure and speed are known, are the mass flow rate, often represented as a flow capacity $m\sqrt{T_0}/p_0$, and the efficiency. These parameters also form the basis of steady-state maps. The analogous instantaneous flow capacity can be defined in identical fashion, because the inlet conditions p_0 and T_0 are specified and the mass flow rate at the inlet to the turbine should be calculable as a function of time in the model. The instantaneous efficiency is defined as:

$$\eta(t) = \frac{W_x(t)}{\Delta h_{0s}(t)} \tag{1}$$

where $W_x(t)$ is the shaft work output per unit mass flow rate at time t. This has been measured directly in pulse flow laboratory testing, and it is a practical requirement of any pulse flow turbine model that it can be calculated. For 0D and 1D models using a quasi-steady rotor, it can be calculated from the change of total enthalpy across the rotor, and this is consistent with the quasi-steady assumption. However, such models require input of empirical loss terms, and must be calibrated against appropriate, but steady-state, test data. For CFD-based methods, a calculation based on enthalpy change is less reliable because the inflow and outflow processes are not coincident in time. A better method is to determine the torque at each instant in time by spatially integrating the tangential component of the pressure force exerted on the rotor blades.

The denominator of Equation (1) is the instantaneous isentropic work output, which can be defined in terms of the inlet and exit conditions using standard gas dynamics:

$$\Delta h_{0s}(t) = C_p T_{00}(t)\left[1-\left(\frac{p_{exit}(t)}{p_{00}(t)}\right)^{(k-1)/k}\right] \tag{2}$$

Equations (1) and (2) reduce the expression of instantaneous efficiency to its simplest form. Westin *et al.* (36–38) expand the instantaneous work term as the sum of the compressor work and the work done to accelerate the rotating system of the turbocharger

$$W_x = \left(\frac{2\pi}{60}\right)^2 JN\frac{dN}{dt} + \frac{1}{\eta_m} m_a \int_{T_{01}}^{T_{02}} C_p dT \tag{3}$$

where J is the rotating inertia and N is the rotational speed. All terms in this equation are instantaneous, but it is implied in this equation that the compressor operation does not vary through the cycle. This equation avoids a separate calculation of the turbine work output from the turbine operating conditions, but does rely on knowledge of the instantaneous speed of the turbocharger. As such, it is suited to the analysis of test data, where the rotational speed can be measured at sufficiently high frequency for the purpose, but for engine system modelling, it is the turbine work output that determines the turbocharger speed variation, rather than the reverse.

In their work, Westin *et al.* use Equation (2) to calculate the instantaneous isentropic turbine work. Winkler *et al.* (39) elaborate on this for a twin-entry turbine by summing the contributions of the two inlet flows and note significant interaction between the two inlets, consistent with the work of Dale and Watson (2).

Equation (2) implies that all of the time-varying quantities must be measured at the same instant in time. Logically, this definition is unsatisfactory because work is developed in the rotor, whereas p_{00} and T_{00} are measured at the turbine inlet, or possibly some distance upstream of the inlet, and p_{exit} is measured at or downstream of the turbine exit. In steady flow conditions, this has no significance, but when the flow is unsteady, the timing of events occurring at different locations in the expansion system becomes very important. This effect is the cause of physically unrealistic values of isentropic efficiency seen in test data and predictions – Figure 11 is a particularly striking example of this but there have been many others. Other inconsistencies have also been seen, for example, in their numerical experiments, Hellström and Fuchs (29) show five different cases of a turbine in which the time-average power output varies by almost 20%, but the isentropic efficiency, based on time-average quantities, varies by only two percentage points.

Some experimenters have attempted to reconcile this by calculating the efficiency using measurements of these quantities at different instants in time, the time differences being those appropriate to compensate for the spatial separation of the measuring points. The choice of the correct time differences thus becomes very important to the accuracy of the solution. Noting the discussion in Section 3 concerning acoustic and convective propagation of fluid properties, it can be seen that the choices are not obvious. This was actually the original source of debate mentioned above about whether the time should be based on the acoustic or mass transfer velocity, which was only subsequently picked up in modelling.

As noted above, the weight of evidence suggests that both effects are present in a turbine. Wave propagation may predominate in cold laboratory testing, but in engine operation the situation is less clear. Whatever the final conclusion, the result is that there is currently no clear and accepted definition of unsteady turbine efficiency. In a twin-entry turbine, and indeed any multiple entry, turbine, the problem is even more complicated, because at any moment in time, the conditions at the rotor inlet are determined by the history of events in both (or all) inlets to the turbine.

In their CFD-based calculations, Hellström and Fuchs (29, 30) show that there is a non-constant phase shift between the mass flow, temperature and pressure at the inlet to the turbine and the shaft power. The phase shift varied during the pulse, but also depended on the frequency of the pulsations. In (30), where computations were performed for a geometrical configuration consisting of a 4-1 manifold and a turbine, the phase shift was quite small, which shows that it also depends on the geometry upstream of the turbine, and its effect on the flow. In this case, the manifold will give a highly turbulent flow into the turbine but this is also due to the fact that the manifold is damping the pulses.

One solution would be to define the isentropic process from rotor inlet to rotor exit, rather than turbine inlet to turbine exit. Providing the rotor is quasi-steady, it should be possible to achieve a satisfactory definition and measurement of rotor efficiency. In practice this is difficult to do experimentally in a small turbine. The static pressures at these points can be measured, but total pressure and total temperature, even steady-state, require extremely small probes in a number of positions in both locations. Considerable problems have been encountered with this even under steady flow conditions.

For both logical and practical reasons, therefore, it appears to be very difficult to provide a useful definition of unsteady efficiency, in the sense of one that can be used to provide accurate engine simulations, and also to guide turbine designers in ways to optimize the turbine design for this form of operation. The concept of a "pulse flow efficiency" is actually just borrowed and adapted from steady-state turbine performance mapping, and since it can now be seen that a different approach is necessary for pulse flow modelling, it must be rejected. The problem can be solved if the other output of the turbine model (in addition to mass flow rate) is specific work, shaft torque or power output, rather than efficiency. A model that can provide this information on a time-varying basis through the engine cycle will meet all the requirements for engine simulation.

It appears that this approach, based on instantaneous power output rather than efficiency, will be completely satisfactory for engine simulation where the turbocharger turbine is one component model of the full system. Its main disadvantage is that it does not provide a simple figure of merit for designers about the performance of the turbine itself, particularly in comparison with other turbine designs used in the same application. Hellström and Fuchs (29) have proposed a "utility factor" κ, defined as the ratio of the shaft power to the available power at the inlet boundary (measured by the incoming flux of total enthalpy), both integrated over the engine cycle. Their numerical results suggest that this correlates well with the time-average power produced. However, it must be understood that any such parameter is only useful for comparing turbines in the same application. Comparisons between turbines in different applications will not have the same validity, because what is being measured is a combination of the exhaust energy available to the turbine and the efficiency with which that energy is converted.

5. CONCLUSIONS

The operation of radial turbines in pulse flow conditions is a complex problem that has not yielded simple and readily applicable solutions. There is as yet no agreement about definitions of figures of merit that can be used to characterize turbine performance and inform future turbine development for this type of operation. It appears that simple concepts of "pulse flow efficiency" and "pulse flow performance maps" will be, at best, approximations, and at worst, very misleading for both engine simulation and turbine design purposes.

A more rigorous engine simulation will require a dynamic approach to turbine modelling, in place of current steady-state map interpolation and extrapolation. Suitable turbine models will most likely be based on one-dimensional unsteady flow methods with quasi-steady rotor models, because of the data preparation and running time requirements of unsteady 3D simulations. The latter will have an important continuing role in research into pulse flow energy conversion in turbines, and in turbine design.

At present, the ability to compute the problem has outstripped the data available to check and validate the computer solutions, and while this continues, important questions about the appropriate choice of solution method, boundary conditions, turbulence models, and so on, will remain unresolved.

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