# EFFICIENCY MEASUREMENT IN THE QINETIQ TURBINE TEST FACILITY WITH TEMPERATURE DISTORTION AND SWIRL: THE MASS FLOW RATE MEASUREMENT PROBLEM

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## ABSTRACT

This work describes the development and implementation of a system to perform accurate measurements of mass flow rate in the Turbine Test Facility (TTF) at QinetiQ Farnborough. The facility has recently been upgraded so that turbine aerodynamic efficiency measurements can be performed and the implementation of a system for mass flow rate measurement formed part of that upgrade. The measurement system is novel in that accurate measurements can be performed with both combustor representative turbine inlet temperature distortion (hot-streaks) and swirl.

The TTF is a short-duration (approximately 0.5 second run time) isentropic light piston turbine facility, which has been used for aerodynamic and heat transfer investigations of – primarily – high-pressure turbine stages, although it has also been configured to operate as a 1½ stage (HP stage with IP or LP vane) turbine. The MT1 turbine is a highly loaded unshrouded design relevant to modern military engine design, or future civil engine design. The turbine is engine scale, and all relevant dimensional parameters for aerodynamics and heat transfer are matched: Re, M, $N\big/\sqrt{T_{01}}$ , $T_{gas}\big/T_{wall}$ . Hot streaks are simulated in the TTF by the controlled mixing of hot and cold gas streams. The cold stream is introduced though a conventional sonic metering nozzle, from a large reservoir acting in blow-down mode.

In a transient facility, the accurate measurement of stage mass flow rate with combustor representative inlet temperature distortion and swirl presents an interesting problem, as the capacity of the ngv row is affected by both of these combustor representative inlet flow-fields. In the TTF, a mass flow rate measurement system has been developed using the exit contraction of the piston tube as a subsonic converging-diverging venturi: upstream $p_0$ and $T_0$, and throat $p$ are measured at a number of circumferential locations around the exit contraction to determine the mass flow rate of the hot stream. The effective area of the venturi was measured using a novel blow-down calibration technique which is described.

The bias error in the measurement of mass flow rate with and without temperature distortion was 1.37 per cent and 1.13 per cent respectively, of the same order as the accuracy associated with conventional tertiary devices. The precision uncertainty was 0.198 per cent in both cases. Accuracy is unaffected by the introduction of inlet swirl.

## INTRODUCTION

Transient, or short-duration, testing facilities were developed to allow combined heat transfer and aerodynamic measurements at engine representative conditions. More recently their use has been extended to performance (aerodynamic efficiency) measurements. Short duration facilities have a number of advantages over continuous steady state facilities, particularly the reduced cost of testing, and the possibility of performing transient heat transfer measurements such that true local Nusselt number can be determined. All non-dimensional parameters relevant to aerodynamics (Reynolds number, Mach number, pressure ratio, non-dimensional speed) and heat transfer (gas-to-wall temperature ratio) can be matched in the short-duration environment. Transient facilities have been developed with run times between 20 ms and 1s. Transient turbine testing facilities can be classified into two groups: those that include a braking system to maintain constant shaft speed, such as the QinetiQ TTF [1] and the MIT Blowdown facility [2]; and those that are un-braked and allow the turbine to accelerate through a range of non-dimensional speed during the test, such as the VKI Compression Tube Facility [3]and the Oxford Rotor Facility [4]. The QinetiQ ILPF is unique, in that it is aerodynamically braked: a turbobrake is mounted to the same shaft as the rotor, and is designed so that it develops an equal and opposite torque at the design point. Thus, very steady turbine speed is maintained during the 500 ms run.

In the TTF, hot streaks are simulated by the controlled mixing of hot and cold gas streams. The cold stream is introduced though a conventional sonic metering nozzle, from a large reservoir acting in blow-down mode. The hot stream is generated using a light free piston contained within a piston-tube, which, under the action of a driver gas, isentropically compresses and heats the working gas to the desired hot-gas temperature. Inlet swirl representative of a modern low NOx combustor is simulated in the ILPF using a passive modular ring containing 16 swirl introducing vanes. Peak swirl angles of ±45 degrees were obtained, giving a very good match to a modern engine profile, with good repeatability around the annulus.

In the last two years (2005-2007), as part of the European TATEF 2 programme, the QinetiQ TTF has been upgraded by Oxford University to allow aerodynamic performance (adiabatic efficiency) measurements of the single stage MT1 turbine. The turbine efficiency is to be characterised with and without pronounced inlet temperature non-uniformity (hot-streaks) and swirl. The target measurement uncertainty for efficiency was a bias uncertainty of ±2.0 per cent and a precision uncertainty of ±0.25 per cent. This is comparable with the target uncertainty of other facilities in which efficiency measurements have been attempted. This paper provides a detailed description of the mass flow rate measurement technique. The technique is unique in that the QinetiQ TTF is the first facility in which turbine efficiency measurements can be conducted with both hot-streaks and swirl, and the mass flow rate measurement system is unaffected by these disturbances at the turbine inlet plane.

Turbine aerodynamic efficiency measurements have been attempted in three other transient facilities in addition to the QinetiQ TTF. In each of these, the technique for mass flow rate measurement was different: the MIT facility uses a critical flow venturi downstream of the rotor with a correction to account for the change in stored mass in the intermediate volume [5]; the Oxford Rotor Facility uses the near choked turbine nozzle, which is calibrated using blowdown experiments of very short duration (so that appreciable changes in temperature do not occur), as a controlling area to meter mass flow rate [6]; the VKI Compression Tube Facility uses an indirect energy balance method to predict the pressure and temperature history and hence calculate the mass flow rate at the turbine inlet plane [7].

The use of the turbine nozzle as a venturi has much to recommend itself: the flow here is almost choked, and the capacity (effective area) of the nozzle throat is therefore a relatively weak function of the downstream conditions. A calibration experiment could be performed by a blowdown experiment from a vessel of known volume under well known starting conditions, and the influence of the rotor on the nozzle capacity could be computed using CFD. However in the QinetiQ TTF, the impact of inlet enhanced temperature distortion (EOTDF) and swirl on the HP turbine aerodynamic efficiency was to be investigated. Both temperature distortion and swirl affect the turbine nozzle capacity, and the use of the turbine nozzle as a flow meter was therefore impossible.

The use of a secondary nozzle situated downstream of the turbine stage is less desirable in transient experiments in which mass flow rate is required to high accuracy. For a transonic turbine stage, pressure ratios of between 2 and 3 are typical. For a choked turbine nozzle, an isentropic turbine would require a choked downstream nozzle with a throat area not smaller than 1.81 times (pressure ratio of 2) to 2.57 times (pressure ratio of 3) the size of the turbine nozzle throat. An unchoked downstream nozzle would naturally be even larger. In practice, to achieve even reasonable uncertainty, an upstream tube is required to condition the inlet flow to the nozzle. In short duration facilities, the requirement for this to be as long as possible is in conflict with the requirement to minimise the dead volume between the turbine and the flow meter, so that the unsteady correction term is minimised. Although, to some extent, the errors associated with the so-called installation effects could be eliminated by an in-situ calibration, there would still be uncertainties associated with the effect on the downstream nozzle capacity with, for example, residual swirl, non-uniformities in pressure and temperature associated with non-uniform work extraction, wakes, over tip leakage flow, hot-streaks etc., none of which could be replicated in a calibration experiment.

To achieve high accuracy in the mass flow rate measurement, the hot stream and cold stream gas paths were independently metered upstream of the point of mixing. The piston tube exit contraction was used as a subsonic converging-diverging metering nozzle for the hot gas path. Thus, no additional flow disturbance was introduced, and good flow conditioning was achieved at the turbine nozzle inlet. The cold gas path was metered using a conventional sonic venturi nozzle, which has a calibration traceable to a primary international (ISO) standard. A series of blowdown experiments have been performed to evaluate the effective area of the piston tube contraction. During each blowdown, the mass flow rate exiting the piston tube was calculated using a stored mass technique. A novel correction technique for the non-isentropic effects (Povey and Beard, [8]) allowed the calibration to be performed over longer time periods than would normally be permitted. Thus, calibration over a range of Re and M was possible in a single experiment. The development of this mass flow rate measurement system and its calibration are the subject of this work.

## THE QINETIQ ISENTROPIC LIGHT PISTON (TURBINE) FACILITY

The TTF at QinetiQ, Farnborough, is a short-duration facility which uses the light piston compression technique developed at the University of Oxford by Jones et. al [9]. A schematic of the facility is shown in Figure 1. Currently, the ILPF operates with the full-scale transonic high-pressure MT1 turbine. The parameters specifying the operating point of the turbine are summarised in Table 1.

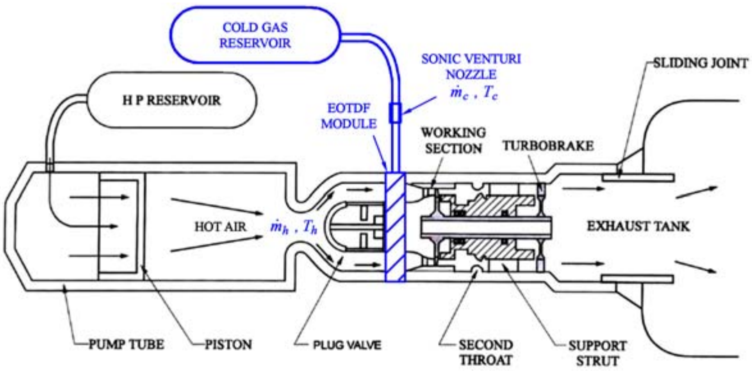



Figure 1: The QinetiQ ILPF with inlet temperature distortion system (EOTDF).

| PARAMETER | VALUE |
|---|---|
| Reynolds number based on NGV axial chord | 1.61 x 106 |
| Mach number at NGV exit | 0.879 |
| Capacity, $\frac{\dot{m}\sqrt{T_{01}}}{p_{01}}$ | 7.97 x 10-4 kg $K^{1/2}$ $Pa^{-1}$ |
| Corrected speed, $\frac{N}{\sqrt{T_{01}}}$ | 451.3 rpm K-1 |
| Inlet total temperature | 444 K |
| Inlet total pressure | 4.6 x $10^5$ Pa |
| Pressure ratio, $\frac{p_{01}}{p_{03}}$ | 2.5 |
| Gas-to wall temperature ratio, $\frac{T_{gas}}{T_{wall}}$ | 1.52 |

Table 1: Turbine MT1 operating point.

Prior to a test run, the plug valve is closed and the working section is evacuated to approximately 10 mbar. The rotor disc and turbobrake assembly is spun up at the turbine design speed (9500 rpm) using an air motor. With the turbine at the design speed, high pressure air is discharged into the piston tube (volume approximately 10 $m^3$) behind a free light piston. The piston moves down the piston tube, equalising the pressure differential across it, thereby isentropically compressing and heating the air in front of it. When the desired working section pressure and temperature are achieved in the piston tube, the test gas is allowed to discharge through the working section by opening the fast acting plug valve. By controlling the volumetric flow rate from the driver reservoir to equal that leaving the piston tube, the total pressure in the working section is kept constant through the test run. The stage exit pressure, and hence pressure ratio, is set by an adjustable downstream choked (second) throat. The test run ends as the piston reaches the end of the piston tube, achieving a quasi-steady run time of approximately 400 ms. An unique feature of the TTF is the aerodynamic braking system, or turbobrake [10], which absorbs the power produced by the turbine so that the rotational speed of the rotor is effectively constant during a test run.

The finite mass of the piston causes +/-1 per cent of mean level fluctuations in total pressure at a frequency of approximately 16 Hz. A reflected pressure wave also travels back and forth through the facility causing fluctuations of approximately +/-0.1 per cent of mean level at 80-100 Hz. All instrumentation is designed to accurately track these fluctuations in total pressure, and hence a minimum bandwidth of 200 Hz is imposed on all measurements.

## MASS FLOW RATE MEASUREMENT TECHNIQUE IN THE ILPF

The piston tube exit contraction, shown in Figure 2, acts as a converging-diverging venturi once the plug valve has opened. From continuity and compressible gas equations, the mass flow rate through such a venturi is given by,

$$dm/dt = p_0 A_{eff} \mathrm{M} \sqrt{\frac{\gamma}{RT_0}} \left(1 + \frac{\gamma - 1}{2} \mathrm{M}^2\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}} . \qquad (1)$$

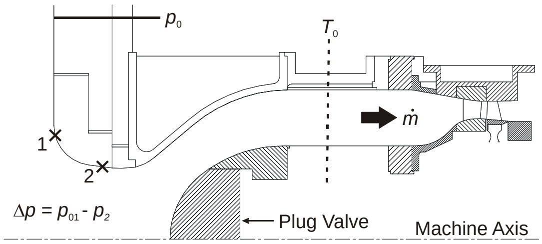



Figure 2: ILPF piston tube exit contraction with mass flow rate instrumentation.

The total temperature of the piston tube gas flow is measured using four evenly distributed rakes of five thermocouples just downstream of the contraction. To provide the required bandwidth for the measurement bare 12.7 µm k-type thermocouples are used. The total pressure upstream of the piston tube exit contraction is measured from four pneumatic tappings on the end-wall of the piston tube. The throat was instrumented with four evenly distributed differential pressure measurements. The geometry was such that a well defined mass mean static pressure measurement was difficult to define: CFD was used to optimize the location of the static pressure tapping as described below. Although the Mach number distribution is non-uniform across the throat section (see Figure 3a), the distribution is well defined, and a single Mach number evaluated using the static pressure at the throat can be used to compute mass flow rate,

$$\mathrm{M} = \sqrt{\frac{2}{\gamma - 1}\left\{\left(\frac{p_0}{p_0 - \Delta p}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\}} . \qquad (2)$$

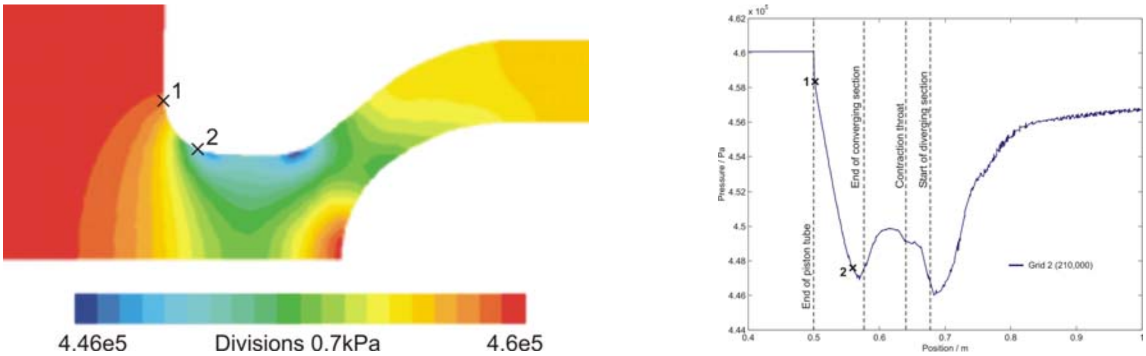
To correct for the non-uniformity, the effective area of the piston tube exit contraction is evaluated in a calibration experiment which is described later in this paper. It is assumed that the effective area is a function of Reynolds number, Mach number, Prandtl number and gas-to-wall temperature. The effective area term also allows for the displacement effect of viscous boundary layers.

## CFD MODELLING OF THE PISTON TUBE EXIT CONTRACTION

The location of the piston tube contraction pressure tappings was optimised through use of CFD. As the evaluation of turbine stage efficiency requires the accurate measurement of mass flow rate during solely the quasi-steady period of the test run, the CFD simulation was based on a steady simulation at the nominal turbine operating point.

The contraction and upstream and downstream plena were modelled using Fluent 3D. The code was run with the following settings: steady; compressible; viscous; Spallart-Allmaras turbulent model; ideal gas. The effect of compressibility was relatively small as the throat Mach number was approximately 0.16. The following inlet boundary conditions were used: $p_0 = 4.6\,\mathrm{bar}$ , $T_0 = 444\,\mathrm{K}$ , $\dot{m} = 2.9\,\mathrm{kg/s}$ , $\alpha = 0^\circ$ . The exit boundary condition was set as an outflow. A 60 degree section was modeled, and a tetrahedral mesh of 210,000 cells was used after a grid dependency study was conducted. Wall functions were imposed on all surfaces.

The CFD solutions for the flow static pressure in the piston tube exit contraction region and wall static pressure are shown in Figure 3a and Figure 3b respectively. The near wall flow over-accelerates (due to high surface curvature) ahead of the true throat, reaching the first – of two – minima at approximately 0.57 m. Between 0.57 m and 0.62 m, although the freestream flow continues to accelerate, the near wall flow is diffused slightly. In the region just downstream of the throat (> 0.64 m) the situation is reversed: the near wall flow accelerates again, while the mainstream is diffused. The pressure tapping locations (marked 1 and 2 in Figure 3) were chosen to maximize the pressure difference with the constraint that the throat tapping should be in a region of accelerating flow, so that it is not subject to unsteadiness associated with separation, or the Re dependency of such a region. The pressure difference was approximately 80 per cent of the maximum at the surface. Four independent pairs were used, distributed at 90 degree intervals around the circumference.




(a) (b)

Figure 3: Piston tube exit contraction, CFD results: a) Contours of static pressure for a 2D longitudinal slice of the modelled volume, b) wall static pressure.

## CALIBRATION OF THE PISTON TUBE BLOWDOWN PROCESS

In the method described by Povey and Beard [8], the mass flow rate exiting a vessel of known volume in a blowdown experiment can be computed to a high accuracy under conditions in which the flow is not in thermal equilibrium with the walls (provided the gas was initially in equilibrium). In practice this means that blowdown experiments can be used – in a manner analogous to the *thermodynamic* method – to calibrate flow meters. The method accounts for the heat transfer from the relatively warmer walls to the gas as the blowdown progresses: the correction for non-isentropic conditions. A good estimate of the uncertainty was obtained by directly comparing the calibration to a sonic venturi nozzle with a calibration traceable to an international (ISO) standard. The non-isentropic effect due to heat transfer is quantified in the form of a correction factor, which is expressed as the ratio of actual mass flow rate leaving the plenum to the mass flow rate that one would infer from a pressure measurement alone assuming an isentropic temperature trend. That is:

$$(3) \qquad \Pi_1 = \frac{\gamma R T_{initial}}{V_{PT}} \frac{dm/dt}{dp/dt} \left( \frac{p}{p_{initial}} \right)^{\frac{\gamma-1}{\gamma}} .$$

Blowdown experiments were performed in-line with a sonic venturi nozzle with a traceable calibration. Experiments were performed over a range of initial pressures (approximately 6 to 10 bar) and blow-down time constants (7 to 14 seconds). The correction factors for 44 experiments are plotted as a function of time in Figure 4. By definition the correction factor is unity at the origin, and grows steadily with time as the heat transfer to the enclosed gas becomes more significant.

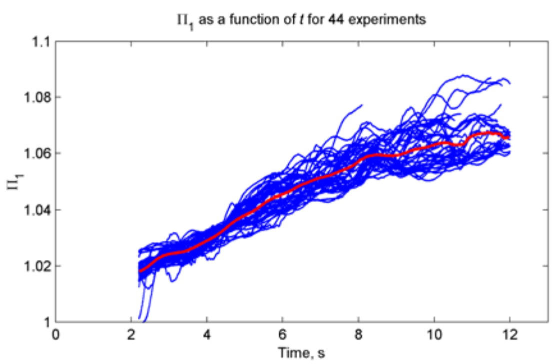



Figure 4: Correction factor as a function of time for 44 experiments.

Using equation (3) and the trendline shown in Figure 4, the inferred mass flow rate $dm/dt$ was calculated. It is emphasised that the only variables required to perform this calculation were $p_{initial}$, $T_{initial}$, $V_{PT}$, $\gamma$, $dp/dt$ and $\Pi_1 = f(t)$. Over very long timescales equivalent to the vessel discharging to half the initial pressure, the mean standard deviation based on 44 experiments was 0.57 per cent. Therefore, to a 95 per cent level of confidence the precision uncertainty in discharge mass flow rate was 1.14 per cent. The fact that accuracy comparable to conventional tertiary flowmeters can be achieved is a powerful demonstration of the utility of this technique.

## CALIBRATION OF THE PISTON TUBE CONTRACTION

The evaluation of the effective area of the piston tube exit contraction was conducted with a separate set of blow-down experiments. A schematic of the experimental arrangement is shown in Figure 5 below.

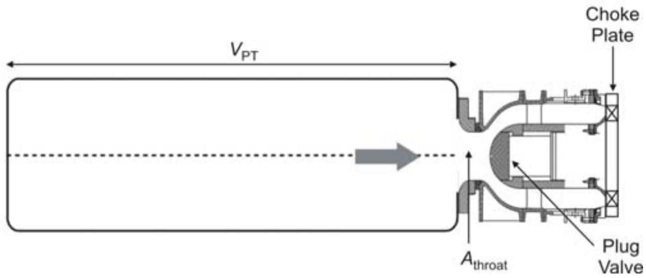



Figure 5: Schematic of effective area blow-down experimental arrangement

In these experiments the lightweight piston was removed from the piston tube, and the pressurised piston tube was vented through its exit contraction. The correction factor evaluated from the previous blow-down experiments was used to measure the mass flow rate exiting the piston tube during each discharge. This mass flow rate was also measured by considering equation (1) and by using the throat instrumentation previously described in this paper. A direct comparison of these allows the ratio of the effective area to the geometric area (or discharge coefficient) of piston tube exit contraction to be computed, thus:

$$\frac{A_{eff}}{A_{geo}} = \frac{(dm/dt)_{\text{inferred}}}{(dm/dt)_{\text{1D}}} = \frac{\Pi_1\left\{\dfrac{dp}{dt}\dfrac{V_{PT}}{\gamma\, RT_{initial}}\left(\dfrac{p}{p_{initial}}\right)^{\frac{1-\gamma}{\gamma}}\right\}}{p_0 A_{geo}\mathrm{M}\sqrt{\dfrac{\gamma}{RT_0}}\left(1+\dfrac{\gamma+1}{2}\mathrm{M}^2\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}}\ . \tag{4}$$

The effective area of a subsonic converging-diverging venturi, such as the piston tube exit contraction, can be assumed to be a function of Reynolds number, Mach number, Prandtl number and gas-to-wall temperature ratio. The pressurised air in the piston tube is vented from near ambient temperature for each blow-down and decreases as it is vented. Hence, the gas-to-wall temperature ratio is not matched during these experiments. With a lower temperature than in normal operation, the Prandtl number will also be different. Hence, the thermal boundary layer in the exit contraction will grow in a different manner; in fact the temperature profile in the thermal boundary layer will be reversed. A numerical boundary layer code has been used to estimate the effect of not matching these non-dimensional parameters in these experiments. The results show it has negligible effect. Hence, the effective area of the piston tube contraction is considered solely a function of Reynolds number and Mach number.

The turbine stage was replaced by a choke plate with adjustable open area to allow the effective area to be evaluated over a range of ± 10 per cent from the nominal throat values of both Reynolds number and Mach number. The open area of the choke plate determined the throat Mach number, which stays approximately constant while the plate is choked. As the piston tube is discharged the flow at the exit contraction throat is swept down through a range of Reynolds numbers, the maximum of which is set by the initial piston tube pressure (assuming the initial temperature is consistent at room temperature).

Once thermal equilibrium between the enclosed air and the piston tube was achieved, the discharge process was initiated by opening the fast-acting plug valve, which is the controlling valve during a normal ILPF test run. The time taken for the mass flow rate exiting the piston tube to reach a maximum was approximately 0.2 seconds.

During the blowdown experiments, pressure and temperature measurements were acquired at 8 locations along the centerline of the piston tube. All thermocouples were bare 0.025 mm k-type thermocouples, mounted in ceramic tubes. Piston tube wall temperature measurements were made using surface thermocouples at 4 evenly spaced locations. The piston tube exit contraction was instrumented as described previously in this paper and as shown in Figure 2.

*CALIBRATION RESULTS*

Calibration experiments were conducted at choke plate settings corresponding to -10%, -5%, -2%, 0%, +2%, +5% and +10% of the turbine nozzle throat area. As the flow at the piston tube exit contraction is approximately incompressible, this corresponds to a Mach number variation of ±10% at this location. Reynolds number was adjusted in the range ±10% of the design value by setting the initial pressure of the piston tube (during a discharge the Reynolds number decreases as the pressure decreases).

A discharge coefficient for piston tube exit contraction was defined as the ratio of the effective area based on the one-dimensional analysis to the geometric area of the piston tube exit contraction $A_{eff}/A_{geo}$. This was computed using equation (4), and is presented in Figure 6 as a function of Reynolds number and Mach number for the 70 calibration experiments. The nominal condition is shown by the red cross in Figure 6. The largest correction for non-isentropic effects in the *inferred* mass flow rate was 0.85 per cent. (approximately 0.6 per cent at the turbine design point). A plane was fitted to the data using principal components analysis (minimizing least squares). The standard deviation of the data from the plane was ± 0.30 per cent.

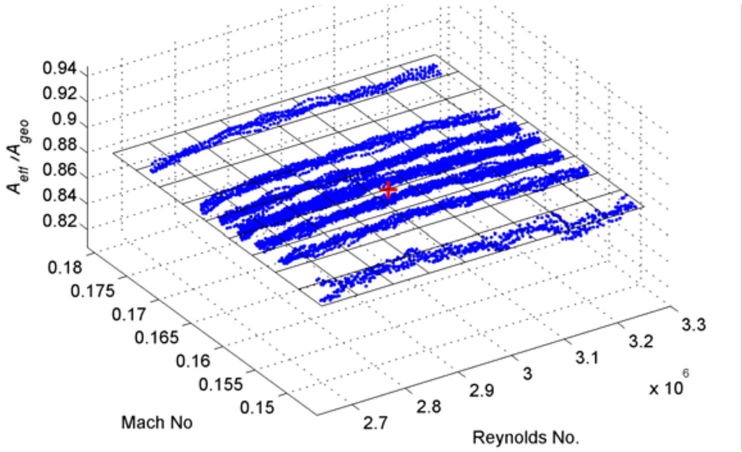



Figure 6: Piston tube exit contraction effective area as a function of Reynolds and Mach numbers for 70 calibration experiments, and the plane of best fit.

*EVALUATING PRECISION UNCERTAINTY*

The precision uncertainty of the mass flow rate measurement technique described is best evaluated using the piston tube exit contraction capacity C, defined by:

$$C = \frac{(dm/dt)_{throat}\sqrt{T_0}}{p_0} \quad (5)$$

In the subsonic situation $C = f\left(A_{eff}, \gamma, R, p_2/p_{01}\right)$. In the current experiments $\gamma$ and $R$ vary very little, and the run-to-run variation in $A_{eff}$ can be evaluated by plotting C as a function of $p_2/p_{01}$. Figure 7 presents a plot of the piston tube exit venturi capacity as a function of the upstream-to-throat pressure ratio for 10 runs nominally at MT1 design condition. The standard deviation of this data from a line of best fit is 0.099 per cent. Thus, the precision uncertainty to 95 per cent confidence is 0.198 per cent. This is a powerful demonstration of the quality of the calibration chain.

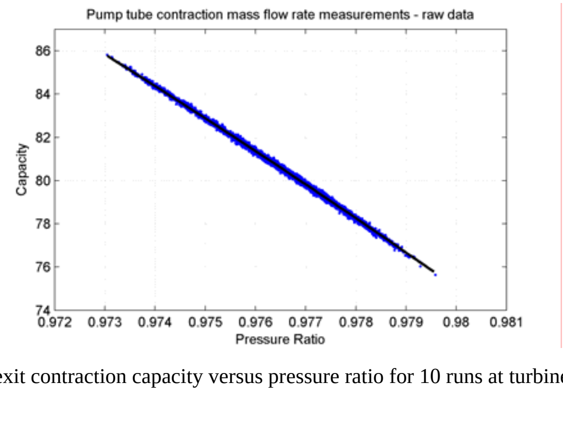



Figure 7: Piston tube exit contraction capacity versus pressure ratio for 10 runs at turbine design conditions.

*CORRECTION FOR INTERMEDIATE PLENUM*

Although the volumetric flow rate of the driver gas to the working gas exiting the piston tube is approximately matched during a run, perfect matching is impossible to achieve. Thus, variations in the inlet total pressure of order ± 1 per cent are typical. Because the light piston has finite mass, it is also typical that oscillations in the inlet total pressure at a natural frequency of approximately 16 Hz persist during the 400 ms run period. These are or order ± 1 per cent in magnitude.

As illustrated in Figure 1, there is a small intermediate plenum between the measurement plane (at the piston tube exit contraction) and the turbine nozzle. In a situation where there are fluctuations in the piston tube pressure and temperature, the stored mass in the intermediate plenum also varies with time, and an unsteady correction term is required to evaluate the mass flow rate through the turbine nozzle from that measured at the piston tube exit contraction. The magnitude and phase of the correction term can easily be shown to be a function of the frequency of the oscillation.

The corrected total mass flow rate through the turbine nozzle, $(dm/dt)_{total}$, is given by:

$$(6) \qquad (dm/dt)_{total} = dm/dt - \frac{V}{R}\frac{d}{dt}\left(\frac{p}{T}\right),$$

where the values of $p$, $V$ and $T$ correspond to the intermediate plenum.

## MASS FLOW RATE MEASUREMENT RESULTS

The mass flow rate measured at the pump tube exit contraction, $dm/dt$, is compared to the corrected mass flow rate through the turbine nozzle $(dm/dt)_{total}$ in Figure 8. Both the directly evaluated correction, $\dot{m}_{corr}$, (thermodynamic method) and the correction evaluated using the unsteady model, $\dot{m}_{model}$, are shown in Figure 8. The numerical model was one-dimensional and developed to predict the magnitude and phase of the unsteady correction term. In this model, air was admitted to the intermediate plenum via an upstream throat, and exhausted via a throat of the geometric area of the turbine nozzle. Downstream of the turbine nozzle throat a vacuum and a linear opening of the plug valve was assumed.In the settled period of the run the two correction terms agree very well.

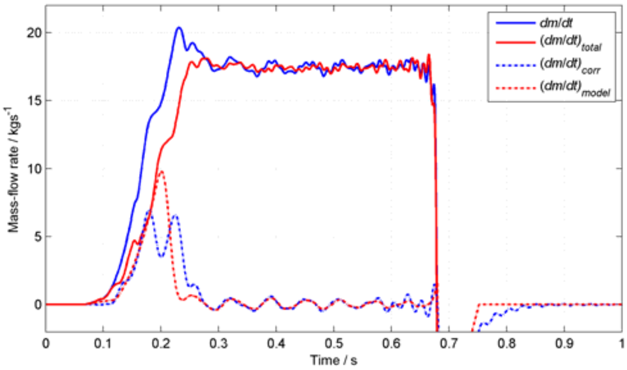


Figure 8: Typical ILPF test run mass flow rates and modeled unsteady correction term.

*MASS FLOW RATE MEASUREMENT WITH INLET TEMPERATURE DISTORTION*

As with the case of uniform inlet condition, a target mass flow rate of 17.4 kg/s, at a mass-averaged inlet total temperature and pressure of 444 K and 4.6 bar respectively were specified with inlet temperature distortion. To achieve these targets, the hot stream mass flow rate was 11.14 kg/s, at a temperature of 530 K, and the cold stream mass flow rate was 6.26 kg/s at a temperature of 290 K. The hot stream was provided by operating the piston tube at a higher compression ratio, with the same final total pressure (4.6 bar). The cold stream, which is at approximately ambient temperature, was introduced from an independent reservoir through a calibrated sonic venturi nozzle into a manifold ahead of the temperature distortion module.

With inlet temperature distortion, the hot and cold stream mass flow rates for a typical run are shown in Figure 9. The mass flow rates measured at the pump tube exit contraction (hot stream) and the calibrated sonic venturi (cold stream) are denoted by $\dot{m}_h$ and $\dot{m}_c$ respectively. The corrected mass flow rates are denoted by $\dot{m}_{h(corr)}$ and $\dot{m}_{c(corr)}$: these include the unsteady corrections required to account for the change in stored mass between the meters and the turbine inlet plane. The total mass flow rate through the turbine nozzle is denoted by $\dot{m}_{total}$. For a typical run, the unsteadiness in $\dot{m}_{total}$ was approximately 2 per cent during the test window, and was in phase with the fluctuations in turbine inlet total pressure.

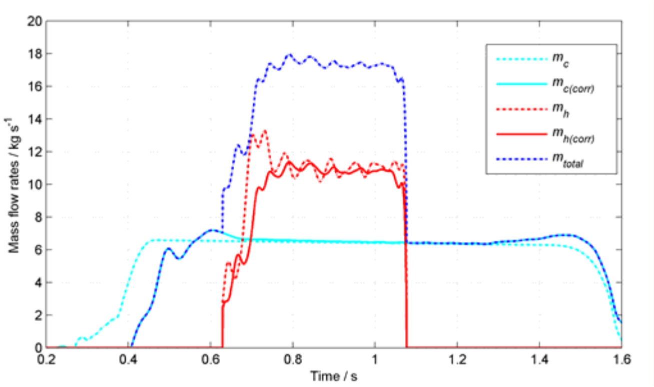



Figure 9: Hot and cold stream mass flow rates for a typical run

The uncertainty associated with the hot stream mass flow rate measurement is the same as in the case of uniform inlet temperature: 1.4 per cent bias error and 0.2 per cent precision error, to 95 per cent confidence. The overall bias error uncertainty associated with the cold stream mass flow rate was 0.7 per cent, of which 0.5 per cent arises from the uncertainty in the discharge coefficient of the sonic venturi nozzle. The precision uncertainty was 0.2 per cent. The combined bias uncertainty in mass flow rate with and without temperature distortion was therefore 1.37 per cent and 1.13 per cent respectively. That is, marginally more accurate with temperature distortion. The precision uncertainty was approximately 0.2 per cent in both cases.

*MASS FLOW RATE MEASUREMENT WITH INLET SWIRL*

Inlet swirl representative of a modern low NOx combustor was simulated in the ILPF using a passive modular ring containing 16 swirl introducing vanes. Peak swirl angles of ±45 degrees were obtained, giving a very good match to a modern engine profile, with good repeatability around the annulus. As the module was completely passive in operation the mass flow rate through the turbine nozzle was evaluated in a similar way to with uniform inlet conditions. Only an extremely small difference in the unsteady correction term is observed as a result of a drop in total pressure across the swirl simulator of approximately 0.3 per cent. Even so, with inlet swirl the intermediate volume between the pump tube exit and turbine inlet planes is modelled as two volumes (separated by the swirl module) to evaluate the unsteady mass flow correction term. Pressure and temperature measurements are acquired in both volumes. The uncertainty associated with mass flow rate was the same as in the case of uniform inlet temperature: 1.37 per cent bias uncertainty, and 0.2 per cent precision uncertainty.

## CONCLUSIONS

The development and implementation of a system to perform accurate measurements of mass flow rate in the Turbine Test Facility (TTF) at QinetiQ Farnborough has been described. The system is novel in that accurate measurements can be performed with both combustor representative turbine inlet temperature distortion (hot-streaks) and swirl. This is the first time such a system has been implemented in a transient turbine test facility.

Under conditions of uniform inlet temperature the use of the first stage turbine nozzle as a flow meter would have much to recommend itself. Under conditions of non-uniform inlet temperature and combustor representative swirl, however, this is impractical: variations in capacity are too large. Likewise, the use of a downstream nozzle gives rise to a large unsteady correction (in the transient case) and uncertainties associated with residual temperature distortion and swirl.

The flow measurement system implemented in the Isentropic Light Piston (Turbine) Facility (ILPF) at QinetiQ Farnborough uses two flow meters: a subsonic venturi meter for the hot stream, and a critical flow venturi meter for the cold stream. The combined bias uncertainty in mass flow rate with and without temperature distortion was 1.37 per cent and 1.13 per cent respectively. The precision uncertainty was approximately 0.2 per cent in both cases. Thus, accuracy comparable to standard tertiary flow meters has been achieved under the unsteady conditions typical of transient turbine testing.

In-situ calibration of the subsonic nozzle was performed using a novel blow-down calibration procedure. The calibration method has been validated against a nozzle with a traceable calibration. Thus, the uncertainty of the overall measurement chain is traceable.

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