

A LARGE-SCALE HIGH PRESSURE AND TEMPERATURE AIR SYSTEM TO ENABLE REPRESENTATIVE METAL EFFECTIVENESS IN MULTIPLE GAS TURBINE FACILITIES

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ABSTRACT

This paper details the design of a 1 MW compressed air system and infrastructure upgrade to supply heated air at 327°C (600 K) and 50 barg to two world-leading gas turbine test rigs: the Oxford Turbine Research Facility (OTRF) and the Engine Component Aero Thermal (ECAT) facility. These are two established, high technology readiness level (TRL) facilities for high pressure turbine research and have been constantly developed over 40 years. This high temperature air supply system represents the latest such development program. The motivation for this is to reduce the development time for novel gas turbine cooling technology, by providing complimentary test facilities with a high degree of engine similarity that can evaluate component thermal performance before committing to expensive full-scale engine testing. The new air supply system enables both facilities to match engine mainstream-to-coolant temperature ratio with run times over 60 seconds, in addition to previously matched Reynolds number, Mach number, combustor temperature, swirl and turbulence profile.

The new air supply system consists of four air storage tanks with a total volume of 136m³, three of which are electrically trace heated across 23 individually PID controlled zones. The system heat-up time is 29 hours from ambient and the final temperature stability is better than 5 °C/hour. Up to three liquid-cooled multistage reciprocating compressors deliver 2400 sm³/hr at 50 barg. Desiccant dryers remove water vapour to -30 °C and oil content is reduced to less than 0,01 mg/m³ at 20°C. A single 300 kW inline electric heater raises the air temperature from 35 °C to 327 °C before entering the storage tanks. The connecting pipework is designed to accommodate nozzle loadings imposed during 8 different operating configurations, depending on whether individual tanks are heated or operated at ambient temperature. The discharge from the tanks feeds one facility at a time, through 8" NB flanged pipework that is also trace heated and maintained at the setpoint. Traceable critical flow venturi nozzles provide flow rate measurements up to 30 kg/s dry air (at 327°C) with an accuracy better than 0.5 % actual reading. Details are provided on the overall system design, the commissioning tests and performance measurements of the flow metering system.

KEYWORDS

Test, Facility, Design, Turbine, Aerothermal, Metal Effectiveness

NOMENCLATURE

p	Pressure	[Pa]
T	Temperature	[K]
p_0	Stagnation pressure	[Pa]
T_0	Stagnation pressure	[K]
PETAL	Purdue Experimental Turbine Aerothermal Laboratory	
ECAT	Engine Component Aerothermal Test facility at Oxford University	
ECAT+	Upgraded ECAT heated working air up to 600 K	
HAS	Heated Air Supply (system used to operate two turbine test rigs ECAT and OTRF)	
OTRF	Oxford Turbine Research Facility at Oxford University	
OTRF+	Upgraded OTRF heated air up to 600 K	
TRL	Technology readiness level (a major design metric)	
PID	Proportional-integral-derivative (closed-loop control system)	

INTRODUCTION

Experimental data from gas turbine test facilities provide vital insights into understanding aerothermal-flow phenomena. Although there have been significant developments in the numerical simulation of turbine flow dynamics, accurate and consistent prediction of thermal performance in fully cooled high-pressure turbines remains a significant challenge that still requires experimental validation data. To achieve stringent emission controls and for alternative sustainable fuels to be viable, there is a continuing need to improve jet engine efficiency and cooling technologies. Therefore, gas turbine test facilities, such as those detailed herein, remain necessary to evaluate new designs in an environment closely representative of real engine conditions.

Rather than operating at full engine temperatures, Hans-Jürgen Render et. al. (2017) point out that turbine facilities that operate at lower temperature have the advantages of higher instrument density, as well as facilitate the application of modern measurement techniques. Notwithstanding the lower temperatures used, these facilities still match important parameter ratios such as Mach number, Reynolds number, non-dimensional rotational speed and mainstream to cooling stream temperatures.

Turbine test facilities can be broadly categorised into short duration operation (run duration typically 0.1 – 2 s) and continuous operation. The Oxford Turbine Test Facility (OTRF, see Chana et al. 2013 & Falsetti et al. 2022) and the Von Karman Institute Compression Tube Facility (Paniagua et al. 2013) are short duration isentropic light piston facilities. Both facilities compress, and thereby heat, test air by injecting high pressure air behind a light free piston contained in large piston tube. Once the required pressure is achieved, fast acting valves open to initiate the flow through the turbine test section, the rotor of which is pre-spun in vacuum. The transient test nature of such facilities is ideal for heat transfer measurement and the assessment of quantities such as heat transfer coefficient, adiabatic wall temperature and Nusselt number. They are also cost and time effective in yielding valuable unsteady heat transfer and aerodynamic data. However, the run time is too short to allow measurements such as metal effectiveness as turbine walls are near isothermal during the test period.

Anthony & Clark (2013) describe a similar short duration rig, namely the Air Force Research Laboratory (AFRL) turbine research facility. They highlight further advantages of transient testing compared to continuous testing, which are: far less energy required, an order of magnitude lower cost to run, and a greater selection and density of instrumentation available. Using relatively large blowdown tanks, this facility is able to achieve blowdowns of the order of several seconds.

Operating in the short-duration mode, the Purdue Experimental Turbine Aerothermal Laboratory (PETAL) blowdown rig described by Paniagua, et al. (2018) features a high-pressure compressor system delivering dry air at 138 barg and a flow rate of 0.45 kg/s to storage tanks with a total volume of 56 m³. The facility blows down into a 10 mbar vacuum tank with a storage capacity of 283 m³. This allows a maximum flow rate of 23 kg/s for unheated air and 4 kg/s for air heated to 1090 K using a natural-gas fired air heater. To heat the air directly in this configuration would require at least 3166 kW of heat input. Using an air mixer, test air is delivered to the working section between 270 K and 700 K at a maximum flow rate of 6 kg/s. This facility features two parallel test sections, one that is linear and the other has an annular geometry aimed at TRLs of 1-2 and 3-4 respectively.

Based on a typical thermal time constant for turbine blades of 5s, the measurement of metal effectiveness (non-dimensional metal temperature akin to a engine thermal paint test data) requires a test duration an order of magnitude longer, such as the continuous operation Next Generation Turbine Test Facility (NG-Turb) (Rehder et al. 2017). However, the maximum shaft power alone is 3.7 MW, resulting in a substantial increase to running costs compared to short duration facilities.

Between the extremes of transient and continuous running, it is possible to have a semi-transient run time of order minutes, which is long enough for metal effectiveness measurements with moderate power consumption. This running mode has been demonstrated by the ECAT facility (Kirolos et al. 2017) operating with a gas-to-coolant temperature ratio of 1.28. This motivated the upgrade of the OTRF and ECAT facilities to incorporate the new Heated Air Supply (HAS) system capable of achieving run times of 60 seconds at engine representative gas-to-coolant temperature ratio of ~2.0.

Table 1 summarises these existing facilities in terms of run time and compares relevant parameters to the HAS system which is described next.

	AFL TRF (Anthony & Clark, 2013)	HAS		PETAL (Paniagua, et al., 2018)	NG-Turb Rehder et al. (2017)
		OTRF+	ECAT+		
Turbine stages	1.5	1.5	Vane only	Vane + non-rotating rotor (fixed by load cell)	2
Run time	2 – 2.5 s	1 min	1 min	0.1 - 0.5s / continuous	Continuous
$p_{0,inlet}$ [barg]	6.9	0.5 - 15	0.5 - 15	0.5 - 6	0.1 – 1.95
$T_{0,inlet}$ [K]	400	273 - 560	273 - 560	270 – 700	300 – 530
Mass flow rate [kg/s]	37 (N ₂)	30 @ 560 K	25 @ 560 K	6	9.0
High pressure [barg]		55	55	138, 55 / 40	
Metal effectiveness	No	Yes	Yes	Yes	Yes

Table 1: Comparison of existing turbine test facilities and Oxford HAS system facilities

HEATED AIR SYSTEM DESCRIPTION

The main elements of the Heated Air Supply (HAS) system depicted in Figure 1 are: compressors, air dryers and filters, storage tanks (vertical air receivers), piping, air heater, electric trace heating system, and bespoke pressure regulator systems. The system operation begins by heating the tanks and pipework to the set point temperature which takes up to 27 hours. Subsequently, the compressors

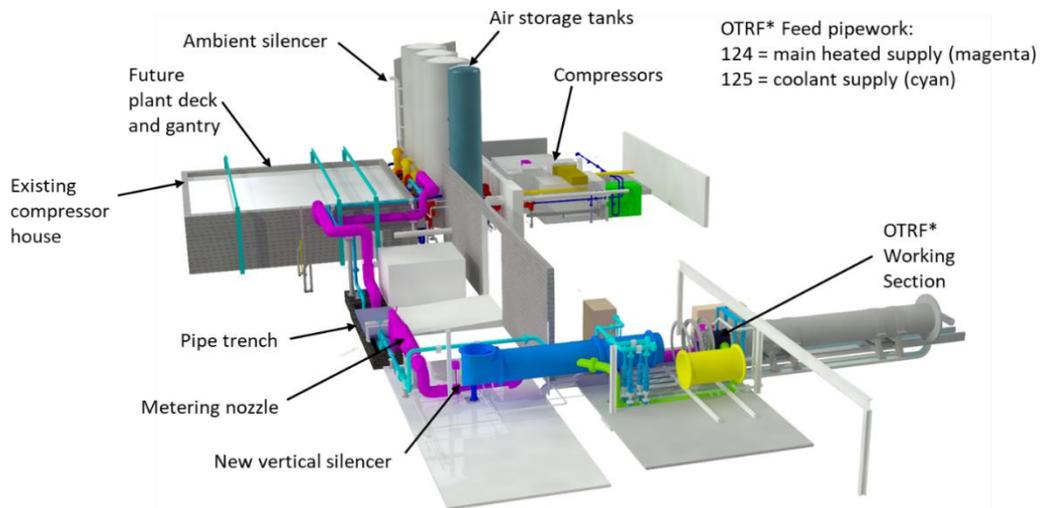


Figure 1: Rendered View Hot Air System and OTRF+

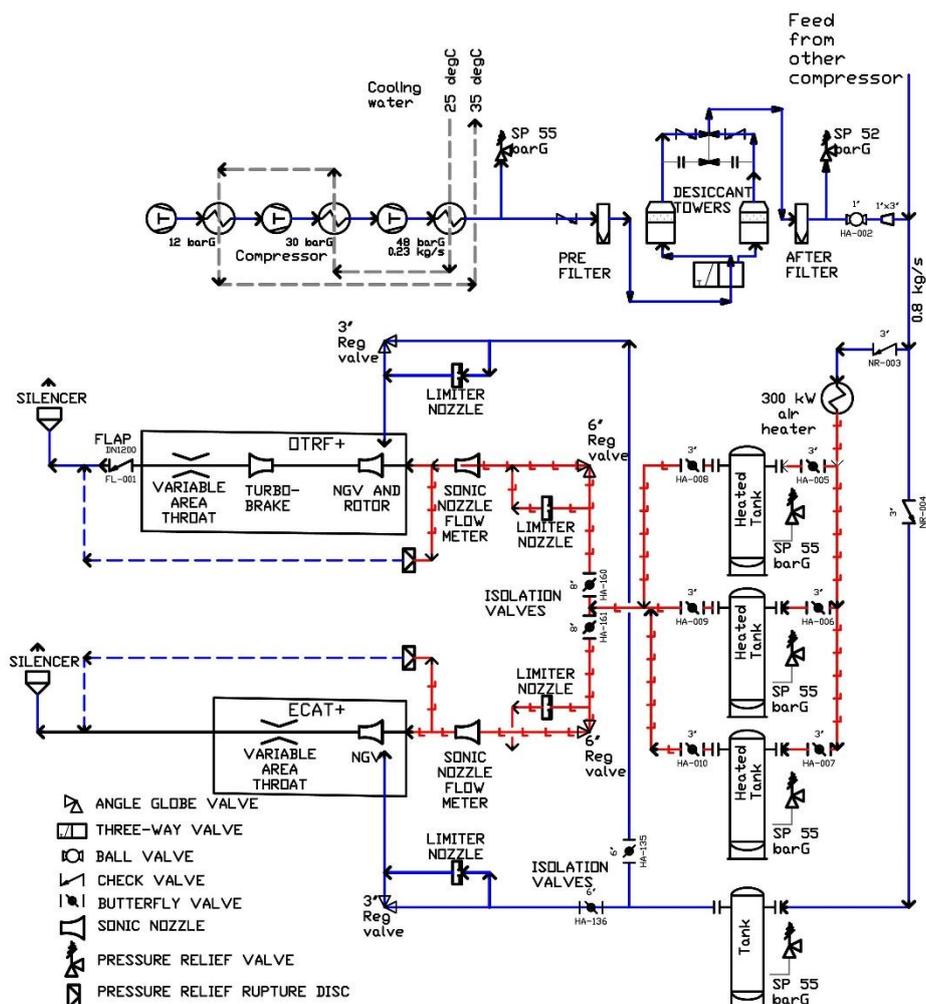


Figure 2: Schematic of Heated Air Supply system (only one compressor shown for brevity)

are started pressurising the system. An inline air heater heats the hot air feed up to the temperature setpoint before entering the tanks. Once the setpoint pressure has been achieved, the system is maintained until discharged during a test run. Thereafter, the system is recharged automatically. A simplified piping and instrumentation schematic is presented in Figure 2 and pertinent details of each sub-system are detailed below.

Compressor system

Up to three water-cooled multi-stage reciprocating compressors operating in parallel are used to deliver 2490 sm³/hr up to 50 barg. Two compressors are currently installed, although provision has been made for the third identical unit which would allow a charging flow rate of up to 0.8 kg/s of dry air, achieving a charge time of approximately 95 minutes (140 minutes for two compressors) from atmospheric conditions. While the isentropic compressor power required is 166 kW, the power consumption of each electric motor is 190 kW, resulting in an isentropic efficiency of 87%. Considerations influencing the choice of compressor type include the inter- and after-cooling water cooling between the three stages of compression. This is important not only for efficiency and material considerations, but also for the final delivered air temperature that needs to be less than 40°C to facilitate effective moisture removal using desiccant-type air dryers.

The dedicated air dryers each feature pre- and after-filters and two 70 litre towers filled with hygroscopic desiccant (Figure 2). These effectively remove oil carry over to less than 0,01 mg/m³ at 20°C, as well as remove moisture content until the dew point is less than -30°C. Three-way control valves divert flow across one tower at a time so that while one tower is absorbing, the desiccant in the second tower can be dried by passing a small amount of scavenged dry air over the desiccant to recharge it.

Adiabatic air-cooled heat exchangers situated external to the building are used to provide cooling water to the compressors at 25°C with a flow rate of 4.2 l/s. The flow rate was measured using an ultrasonic type flow meter and coincides with the pressure/flow rate curve supplied for the compressor cooling circuit by the original equipment manufacturer. A glycol-water mixture of 25% prevents freezing in the colder months, resulting in a heat duty of 171 kW for each cooler. The coolers are designed to achieve this set point whilst operating at an ambient temperature of 35°C and wet bulb temperature of 20°C, which are realistic maximums for Oxford. These important considerations necessitated the choice of adiabatic systems when selecting the cooling system for the compressors. Further heat exchange from the compressor motor to the environment is made possible by a dedicated ventilation circuit designed to reject 30 kW per compressor from the acoustic enclosure.

Air receivers

Compressed, dry air heated up to 327°C and 50 barg is stored in three air receivers, each with a volume of 38 m³, as pictured in Figure 3. Dry ambient temperature air at 50 barg is stored in a fourth air receiver of volume 20 m³ providing a source of coolant test air.

Dual pressure relief valves protect each tank from over-pressure, set at 55 barg and 57.5 barg respectively (design pressure: 55 barg @ 387°C). The relief valves are connected via flexible joints to bespoke vertical exhaust stacks that discharge the heated air at a safe height and within 98 dB at ground level. The exhaust is fixed to each air receiver at the lowest support, and features two sliding joints further up the vessel to accommodate the 12 mm of thermal growth of the tank as it is heated.



Figure 3: Photograph of trace heated hot air storage tanks adjacent to compressor house

Each air receiver has a steel wall thickness of 55 mm that is insulated with 150 mm thick mineral wool. Beneath the insulation, trace heating elements are coiled around the tanks. There are six independently temperature-controlled zones along each tank, each approximately 2m high with three independent heating circuits (creating 18 circuits in total per tank), which maintain isothermal conditions (and thus counters natural convection effects) typically within $\pm 3.0^{\circ}\text{C}$.

Triple-offset metal-seated butterfly valves are fitted upstream and downstream of each air receiver, allowing individual tanks to be operated in either ambient or heated states which gives rise to a total of 8 different operating configurations as listed in Table 2. Although this functionality provides maximal operational flexibility – e.g. ability to use any of tanks 1-3 as an extra coolant supply for combustor simulators, or isolation / inspection / repair of an individual heated air reservoir whilst continuing testing – this adds considerable complexity to the piping design. The thermal nozzle displacements on each vessel are considerable, and therefore, major expansion and contraction loops were incorporated in the piping design to relieve thermal stress and comply with design codes.

Configuration	TANK 1	TANK 2	TANK 3	TANK 4
Operation 0	C	C	C	C
Operation 1	H	C	C	C
Operation 2	C	H	C	C
Operation 3	C	C	H	C
Operation 4	H	H	C	C
Operation 5	C	H	H	C
Operation 6	H	C	H	C
Operation 7	H	H	H	C

Table 2: Different system operating modes depending on tank heating and configuration showing each tank heating is independent (H = heated to 660 K, C = ambient of 273 K)

Pipework Design

The rigid carbon-steel pipework throughout the feed system had to be designed with sufficient flexibility to accommodate the thermal growth between ambient and heated states (design temperature differential of 387 K). The detailed flexibility and stress analyses were completed using the software package: ROHR2 Sigma® – see Figure 4(a) for example. The highest nozzles on the tanks, namely 117, 118, and 119 in Figure 4(b) have a vertical displacement of 12.6 mm. Flexible joints were avoided due to cost, maintenance and inspection considerations. Alternatively, expansion loops and spring-loaded pipe supports were utilised.

Pipework diameter was sized to limit the pressure drop throughout the system. A line size of 3” nominal diameter connects the compressors and tanks, whilst the discharge piping between the tanks and the rigs is 8” nominal diameter to accommodate the discharge mass flow rate of up to 35 kg/s. The discharge piping is classified according to the Pressure Equipment Directive as category III, necessitating 10% radiograph testing of the weld quality, as well as 100% hydrotesting. Since the working fluid is dry air with a dew point of -30 °C, the internal pipe surfaces were uncoated and a corrosion allowance of 1.6 mm was provided. Due to the large thermal mass of the steel pipework and tanks, these were electrically trace heated as described next.

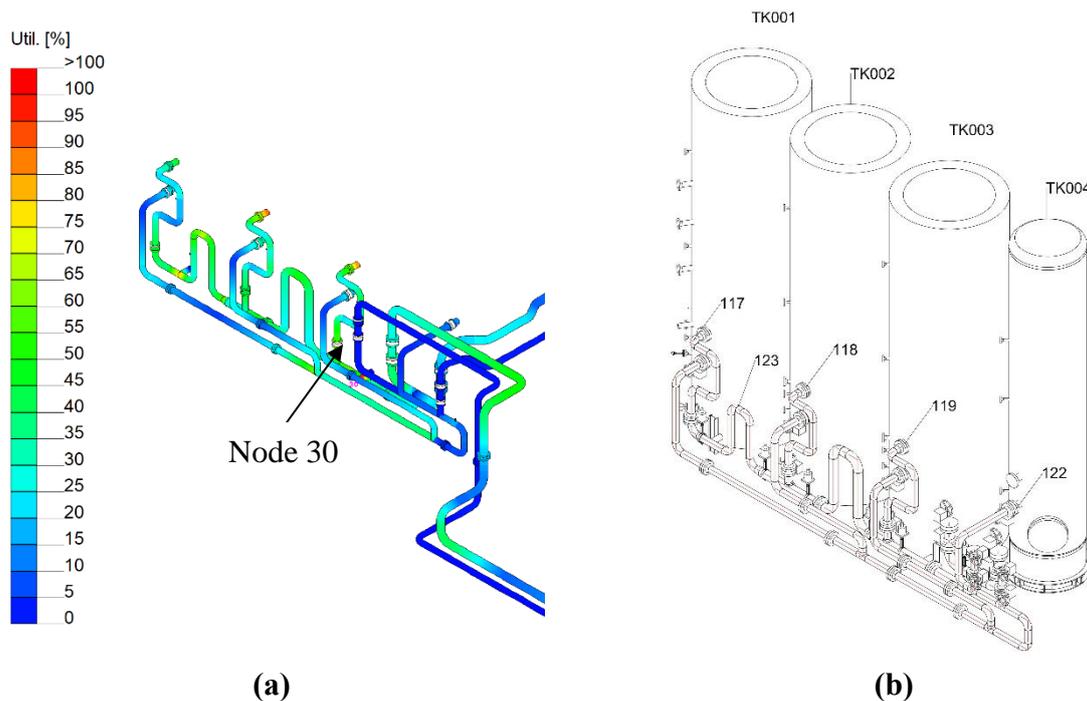


Figure 4: (a) Stress analysis to EN 13480-3: utilization analysis, max utilization 96% at node 30, (b) Isometric view of discharge piping showing expansion loops

Trace Heating and Air Heater

The trace heating system installed on the heated air receivers utilises mineral insulated heating elements coiled around the tank cylinder and shell ends. Steel mesh (50 mm x 50 mm) is banded over these elements to ensure good thermal contact, before aluminium foil is applied and finally encased in 150 mm thick mineral wool insulation. The insulation thickness was increased from the

certification requirement (100 mm, based on maintaining a surface temperature below 60 °C) to reduce the heat loss during steady state operation, and hence reduce power consumption and the facility running costs. The steady state temperature of the steel is maintained within $\pm 3,0$ °C of the setpoint.

Pressure and Temperature Measurements throughout the System

Safety interlocks are integrated into a dedicated PLC control system that monitors pressures and temperatures throughout the system. In particular, gas and metal surface temperatures are monitored using Pt100 resistance temperature detectors inserted into the pipework and air receivers. In the event of any of these temperatures exceeding 610 K, the system is automatically shut down disabling the compressors, air heater, and trace heating systems. Protection against system over-pressure as a result of heating is also ensured by restricting compressor and trace heating operation unless the following criteria is satisfied:

$$1.05 (p/T) < (p/T)_{design} = 55/660 \quad (1)$$

Bespoke Pressure Regulation to Control Mainstream and Coolant Mass Flow Rates

The mainstream (heated) and coolant (ambient) mass flows are controlled by regulating the pressure upstream of traceable (EN ISO 9300:2015) choked venturi nozzles. Following isolation valves, pressure regulation is achieved by the parallel connection of a flow limiter and pressure control valve (or regulator) in each facility feed (see Figure 2). The role of the limiting nozzle, which is replaceable with different nozzle diameters, is to provide roughly the correct total mainstream or coolant mass flows at the start of the test (when the tanks are at maximum pressure). Subsequently, the control valves operate over the maximum stroke, maximising test duration.

To maximise the safety margin on the test section burst relief discs, the transient peak pressures during start-up are minimised by tuning the opening time of the run-initiation (butterfly) valves downstream of the choked nozzles to at least 5 seconds. Thereafter, as the pressure in the tanks decays, a closed loop controller opens the control valve to maintain a constant pressure upstream of the flow meters. As the flow nozzles remain choked, the various facility mass flows are almost steady over the test run. Multiple modes of valve control are possible. These consist of pure PID pressure control, pre-programmed opening mode, and an option to correct the nozzle upstream pressure to account for reducing tank temperature during a run (due to isentropic expansion of the gas inside of the tanks).

The use of a motorised control valve actuator was deliberate. A motorised actuator is slower than an equivalent pneumatic actuator, taking approximately 30 seconds to fully open. This was found to be an advantage to prevent nuisance bursts of the test section rupture disks in the event of a control system error demanding the valve to open fully at maximum tank pressure. In this event, 30 seconds of air would be lost before the valve is fully open, limiting the maximum flow rate through the test section. Motorised actuators also have higher precision of valve opening compared to the equivalent pneumatic actuator.

Mass Flow Rate Measurement

Mainstream mass flow rate measurement is performed using a dedicated 14" critical flow venturi nozzle (according to standard EN ISO 9300:2015) with a nozzle throat diameter of 3.5" upstream of each test rig. Four bare-bead T-type thermocouple probes are installed circumferentially at 90°

intervals one pipe diameter upstream of the nozzle to measure the average upstream total temperature (uncertainty ± 0.65 K or $\pm 0.2\%$ at 600K). Four Druck 5000 absolute pressure transmitters measure the average upstream total pressure (uncertainty $\pm 0.14\%$ at 35 bar). A similar pressure transmitter measures the pressure downstream of the nozzle.

All coolant flows are also individually metered prior to entering the test facilities by dedicated choked venturi meters to a similar accuracy. The OTRF+ facility includes 6 separate coolant streams.

Power requirements and cost per test

An estimate of the total power consumption (including the compressors, coolers, air heater and trace heating) is given in Table 3, and compared to the power required to directly heat the mainstream air at 30 kg/s. Both estimates assume a 60 second period test occurring every hour. The HAS estimate also assumes a single system heat up and 5 tests per day are achievable.

Number of tests runs	Total power consumption for HAS [MWh]	Direct heating [MWh]
1	11.7	10.4
5 (1 day)	19.0	52.0
10 (2 days)	27.4	104.0
25 (1 week)	51.6	260.1

Table 3: Estimated total power consumption compared to direct heating

To heat the system from ambient to 327°C (600K) takes approximately 29 hours at full power (300 kW). Between 15-20% of full power is required to maintain the system temperature depending on the prevailing weather conditions. Comparing the estimates in Table 3, a reduction in total power consumption of 63% is achieved for a single test day with five tests. The consumption savings continue to increase with the length of test campaign, reaching 80% for 1 week. As the heat capacity of the system reservoirs and pipes is orders of magnitude higher than for the test air, it is clear that reducing the number of total system heating cycles is beneficial to total power consumption.

TRANSIENT MASS FLOW AND THERMAL MODELS

Transient pressure and thermal models were developed early in the design phase to enable prediction of the system performance, and hence assist with design decisions. Messenger & Povey (2020) describe these models in detail. Commissioning data was used to validate both models.

The thermal model was used to capture the thermal response of the system over a full test run (of the order minutes). The pipework system is divided into a series of discrete thermal masses with transient radial heat transfer. This accounts for heating or cooling of the gas as it passes through the pipework, the majority of which is pre-heated to tank temperature. The tank is modelled as a volume with heat transfer from the walls of the tank to the stored fluid. During a test, the tank temperature reduces due to expansion of the contained gas as mass is removed. The flow rates in the thermal model are assumed to be quasi-steady, i.e. equal flow rate at all points in the system at a given time.

The transient pressure model was used to capture the transient pressures experienced during the initial start-up of the facility (typically the first 10 seconds). The HAS system is reduced to a series of restrictions (valves, nozzles) and simplified volumes (pipes, tanks). Isentropic flow equations are used to solve the mass flow rate into and out of a particular volume during each time step. An energy balance is applied to each volume to determine the change in temperature and pressure as the mass of gas in each pipework volume changes time step to time step. In this model, it was assumed that

there was no heat transfer to or from the pipework metal. This is justified by considering that the thermal time constant for the system is considerably greater (minutes) than the pressure time constant (seconds). Different to the ECAT+ operation, the OTRF+ features an exhaust flap that allows the working section and volume up to the run valve to be vacuumed to 0.05 bara prior to testing. This allows the rotor to be spun up in a vacuum which greatly reduces the torque required to overcome the windage resistance. This is represented in the model as a piece-wise step in mass flow, i.e. below the prescribed opening pressure of 0.5 barg the flap remains closed and thereafter opens fully.

RESULTS

Initially the individual components of the HAS system were commissioned, i.e. the compressors, trace heating systems, air heater and control system. Then the entire system was commissioned by operating the ECAT+ facility in blowdown tests. The OTRF+ facility was still being manufactured at this time, and it made sense to commission the system using the non-rotating facility (lower risk).

Key results from the ECAT+ commissioning are presented to demonstrate the capability of the new system and to validate the design models.

Commissioning of pressure regulation system

The correct functioning of the pressure regulation system is important to ensure the correct test conditions are achieved. Initial design calculations regarding the flow regulation system were based around the manufacturer supplied flow coefficients, which did not include the effect of the particular HAS pipework installation on the flow through the valve. Therefore, the control valves were characterised at a number of valve positions as part of the initial system commissioning.

The valve was characterised by an equivalent area, which simplifies compressible flow calculations as opposed to flow coefficients (units of volumetric flow) typical of standard industrial valve sizing equations (such as BS EN 60534). The valve characteristic of the mainstream control valve is shown in Figure 5(a). Note that the characteristic is nonlinear, and is typical of an “equal percentage” valve trim.

The updated control valve model was implemented in the design code, and a high mass flow rate test simulated. The predicted pressures and valve opening are plotted in Figure 5(b) and Figure 5(c) respectively. Refer to Figure 6 for station numbers. As can be observed in Figure 5(c), the predicted

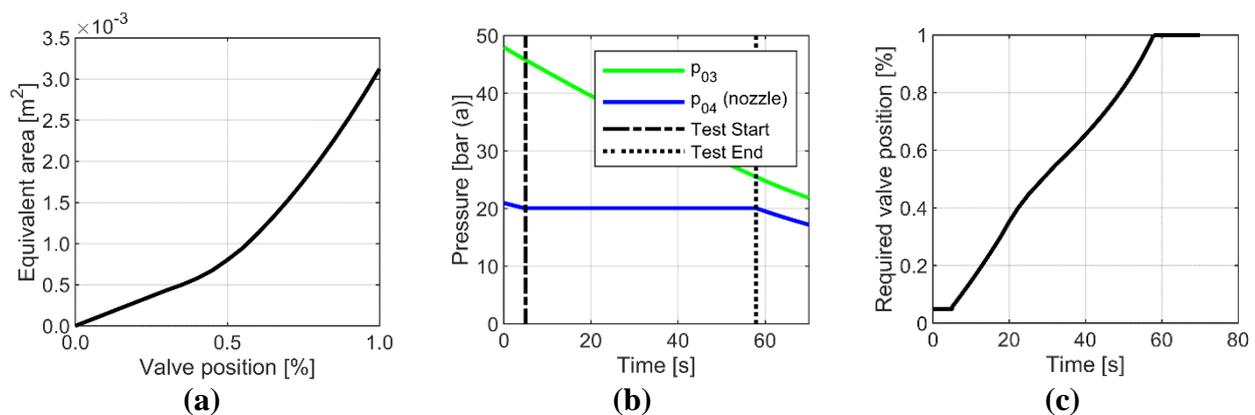


Figure 5: (a) Equivalent area of DN150 angle globe control valve, (b) Simulated pressures - high mass flow, (c) Simulated valve opening - high mass flow

valve opening is expected to be approximately linear in time when regulating to a constant downstream pressure. Equal percentage valve trims are typically designed for this purpose.

Due to the motorised actuator previously described, there is a practical limit to the maximum speed the control valve can open. The maximum full stroke opening time for this model of valve is 30 seconds, which is more than sufficient for the high flow test expected (approximately 55 seconds to open from 5% to 100 %). It was found to be advantageous to start with the control valve 5% open as there is a relatively small change in equivalent area with valve position at small valve openings.

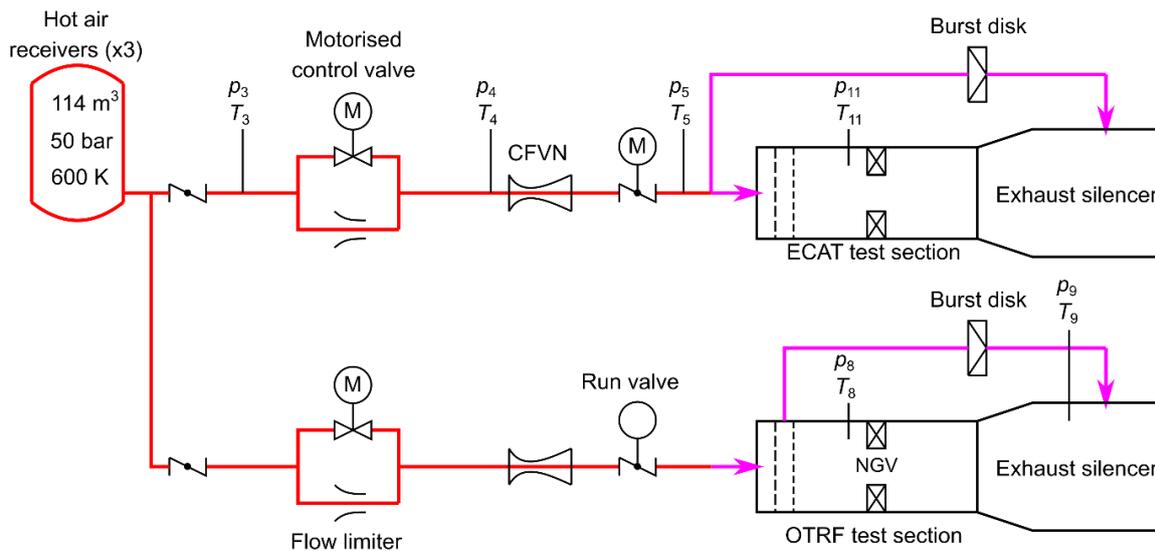


Figure 6: Simplified schematic diagram of mainstream (hot) air supply to ECAT+ and OTRF+ facilities

Commissioning of HAS with ECAT+ test facility

Data from the initial commissioning and test campaign conducted using the HAS and ECAT+ facility is shown in Figure 7. The purpose of this is to demonstrate the capability of the HAS system to supply the facility and to validate the design models. For these tests, the ECAT+ facility did not include the back-pressure control module, reducing the initial commissioning pressure inside the test section for safety reasons. As a result, the test duration is longer than the ‘high mass flow’ test simulated in Figure 5(b). Subsequently, the design models were used to predict the performance of the OTRF+ and ECAT+ facilities at different operating conditions. A thorough presentation of the specific ECAT+ and OTRF+ commissioning data will be the subject of future publications.

Figure 7(a) shows the key pressures throughout a typical ECAT+ commissioning run. Initially, the pipework was charged to the tank pressure up until the ‘run valve’. Once the run valve was opened, the pressure upstream of the critical flow venturi nozzle (CFVN) decreased, and the pressure increased in the test section, showing the initial start-up pressure transient due to filling and venting of the pipework volumes. Five seconds later, the pressure control system was automatically activated to maintain a constant pressure. The PID controller exhibited a slight overshoot in pressure, but steady conditions were reached approximately 20 seconds after the run valve was opened.

An extensive number of runs were performed at low temperature to tune the PID controller coefficients using a trial and improvement procedure. Temperature had little effect on the required

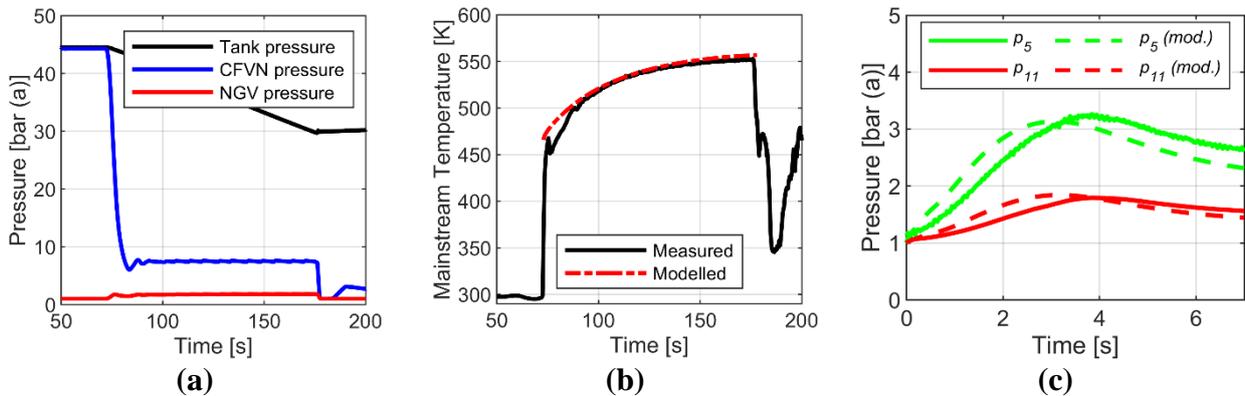


Figure 7: Typical ECAT+ high temperature run data: (a) reservoir pressure $p_3(t)$, CFVN upstream pressure $p_4(t)$ and NGV pressure $p_{11}(t)$, (b) mainstream gas temperature $T_{11}(t)$, (c) transient test section pressures $p_5(t)$ and $p_{11}(t)$

coefficients, but they were sensitive to the initial system pressure and controller set pressure. Therefore, the system would need to be re-tuned for different test conditions and/or hardware should they be significantly different.

Figure 7(b) shows the gas temperature upstream of the NGVs under test (during the same run as Figure 7(a)). There was a very rapid increase in gas temperature as the run valve opened, followed by a steady increase in temperature before a maximum temperature was reached at the end of the test. At the end of the test the temperature was almost steady at approximately 550 K, corresponding to a maximum achieved temperature ratio (TR) of 1.90. There was also a small variation in temperature observed after the valve was opened due to the start-up pressure transient (compression and expansion of the gas in the pipework and test section). Also included is the prediction from the transient thermal model. This shows very good agreement with the experimentally observed temperature, and validates the modelling approach for this system.

To validate the transient pressure model of the system, the pressures inside the test section, and downstream of the run valve are plotted in Figure 7(c). Good agreement in the peak pressures was observed, although there is a small time shift between the predicted and measured peak pressures. The most likely reason is that the transient model does not include acceleration of the valve, i.e. it assumes a constant opening speed. This is unimportant, however, as it is the magnitude of the pressure inside the test section that is of interest.

Application of models to predict OTRF+ performance

The transient pressure and thermal models were applied to predict the performance of the OTRF+ facility. The predicted transient thermal response at the OTRF+ inlet is shown in Figure 8(a). Note that at the higher flow rate, the maximum temperature occurs sooner in the test period than at the lower flow rate simulated and tested in Figure 7. For the lower flow rate tests conducted during ECAT+ commissioning, the test was stopped as maximum mainstream temperature was reached. If the test was allowed to proceed, a decrease in mainstream temperature would have been observed beyond the maximum.

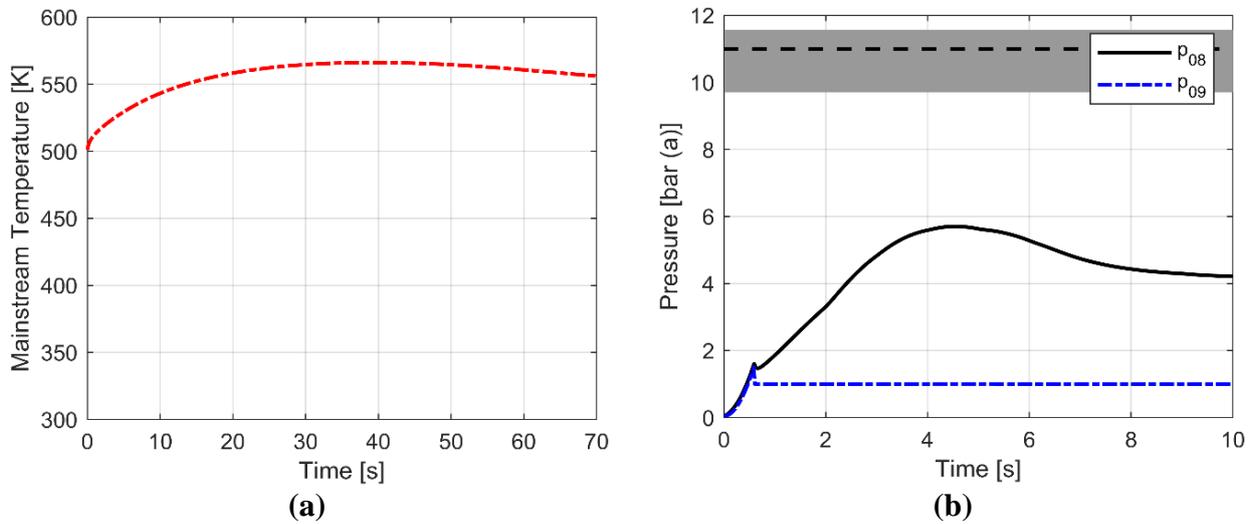


Figure 8: Predicted OTRF+ test conditions: (a) thermal transient temperature at test section inlet $T_8(t)$, (b) Start-up transient pressure $p_8(t)$

The predicted transient pressure response of the OTRF+ is shown in Figure 8(b). The shaded grey area is the range of pressure at which the rupture disk is likely to burst, due to manufacturing tolerance (+/- 5%) and the de-rating of the rupture pressure due to increased temperature. The nominal rupture pressure is 10 barg. The run valve opening time was set to 10 seconds.

Stable conditions of pressure and temperature for a duration of at least 60s are predicted after an initial 10 s transient, ideal for metal effectiveness evaluation.

CONCLUSIONS

The infrastructure upgrade to two world-leading gas turbine test rigs comprised of a 1 MW compressed air system has been designed, built and commissioned. Dry air is supplied at 327°C (600 K), and 50 barg to each rig individually in a blow-down operating mode that provides run times of order 60 s. This will allow the measurement of metal effectiveness of gas turbine hardware at highly representative engine conditions. Furthermore, the semi-transient operation of this system provides a cost and time effective method to measure metal effectiveness compared to continuous mode facilities and actual engine testing with thermal paints.

Highly accurate measurements of mainstream and coolant mass flow rates are achieved by dedicated critical flow venturi nozzles distributed throughout the system. Bespoke high temperature pressure regulators are effective in providing a stable upstream pressure to these flow nozzles, despite decaying tank pressure.

Pertinent design considerations and decisions using simple fluid and thermal models are covered in this paper and compared to the experimental commissioning data. The relatively simple models are able to predict the performance of the pressure regulator system and resulting system mass flow rate. These results may be of interest to other researchers as well industry partners, particularly those involved with facility design and testing of aerothermal components.

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