

EXPERIMENTAL TURBINE AERO-HEAT TRANSFER STUDIES IN ROTATING RESEARCH FACILITIES

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ABSTRACT. The present paper deals with the experimental aero-heat transfer studies performed in rotating turbine research facilities. Turbine heat transfer research had significant progress in the last few decades. Since the full-scale operational conditions of a modern gas turbine dictate high temperatures well in excess of 3600 ° F and pressure ratios ranging from 20 to 50, experimental forced convection heat transfer research on the gas side of a rotating turbine environment is a technically challenging task. The current paper provides a limited review of turbine heat transfer research in various facilities including short-duration blow-down and large-scale/low-speed turbine systems. Since the final status of any forced convection heat transfer problem is closely related to the detailed structure of momentum transfer in highly 3D, unsteady, rotating, and turbulent viscous flow environment, emphasis will also be placed on pertinent turbine aerodynamic features existing in turbines. The most significant parameters to simulate in a rotating aero-heat transfer facility can be listed as Reynolds number based on the blade chord, Mach number for compressibility and shock wave effects, tip speed, intensity and scale of free-stream turbulence, Strouhal number for the unsteady wake passing effects, free-stream to wall temperature ratio, coolant to free-stream temperature ratio, specific heat ratio, molecular Prandtl number of the operating gas and a rotation number for turbine aero heat transfer work performed under rotational conditions. A flow coefficient and a loading coefficient defined by the actual turbine hardware is typically maintained during laboratory testing.

NOMENCLATURE

c_p	local specific heat	T^*	$(T-T_o)/(T_w-T_o)$ non-dim. temperature
c_{p0}	specific heat at reference conditions	V_i^*	V_i/V_o non-dim. velocity component
c_p^*	c_p/c_{p0} non-dimensional specific heat	V_o	reference velocity
k	local thermal conductivity	x_i	independent space variable (index not.)
k_o	thermal conductivity at ref. conditions	λ	second coefficient of viscosity
k^*	k/k_o non-dimensional therm. Conduct.	μ	absolute viscosity
L	reference length	μ_o	reference viscosity
p	pressure	μ^*	μ/μ_o non-dimensional viscosity
p_o	reference static pressure	Φ	viscous dissipation function
p^*	$(p-p_o)/\rho_o V_o^2$ non-dim. static pressure	Φ^*	$L^2 \Phi/\mu_o V_o^2$ non-dim. dissip. Function
Re	$V_o L / \mu_o$ Reynolds number	ρ	local density
T_∞	free-stream total temperature (gas temp.)	ρ_o	density at reference conditions
T	temperature	ρ^*	ρ/ρ_o
t	time	ν	kinematic viscosity
t^*	$t V_o / L$ non-dimensional time	ν_o	ref. kinematic viscosity
V_i	velocity component (u,v,w)		
T_o	reference temperature		

INTRODUCTION

The experimental rotating research rigs for gas turbine aero-heat transfer studies range from full-scale gas turbine demonstrators to short duration facilities, from rotating coolant passage simulators

to large-scale/low speed turbine research facilities, from rotating disk cavity research rigs to tip leakage flow simulators. While instrumented full-scale gas turbine demonstrators are excellent candidates to generate very realistic gas turbine heat transfer data, the initial investment made to construct them, their extremely high operational costs and the technological challenges in performing reliable and high-resolution aero-thermal measurements limit their current use.

The short duration facilities or blow down type rigs have been extremely popular choices mainly because of their relatively reduced cost of operation and excellent scaling available for Mach number and Reynolds number sensitive aero-thermal phenomena. Although they may not operate at a full-scale gas temperature, they can generate an accurately measurable amount of heat transfer from the gas side to turbine blades in a linear cascade, annular cascade or turbine stage arrangement where there is usually a realistic rotor operation. The heat transfer coefficients resulting from short duration rotating rigs are considered to be reasonably high quality in terms of experimental uncertainties. Although heat transfer instrumentation in short duration facilities require advanced heat flux sensor making, signal transmission and signal processing capabilities; the results generated in these facilities have extremely high technical value, during and after a turbine design sequence. The most visible short duration heat transfer facilities were constructed and operated by Oxford University, Von Karman Institute for Fluid Dynamics, Ohio State University (Calspan previously), MIT Gas Turbine Lab, Wright Patterson Air Force Base and QinetiQ Corporation.

Large-scale and low speed research facilities are also frequently employed in aero-heat transfer research because of their low operating costs, their high reliability and operational safety characteristics. One of the first low speed turbine research rigs was operated by United Technologies Research Center at Connecticut. The currently operational low speed, large-scale turbine research rigs are at the Aero-Heat Transfer Laboratory of the Pennsylvania State University, Purdue University and Texas A&M University. There are also a number of low-speed/large-scale turbine research facilities currently operational in Germany and Switzerland mainly used for the improvement of stage aerodynamic features.

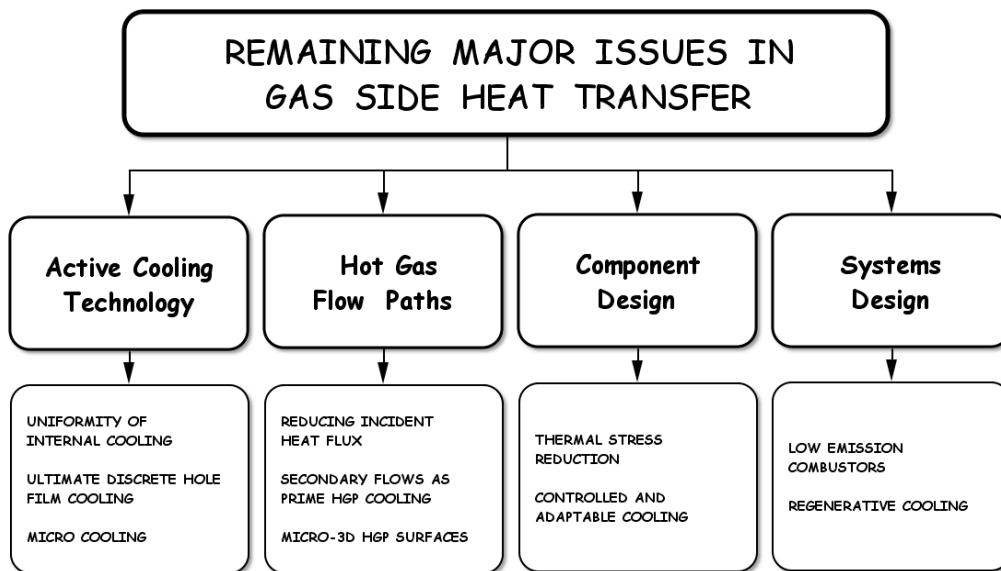


Figure 1. Major gas side heat transfer issues in turbines, Bunker [2006]

The current paper will provide an overview of presently operational turbine heat transfer research facilities that have a fully operational rotor. In addition to major rotating rigs, the research results from the Axial Flow Turbine Research Facility AFTRF at Penn State, operated by the author of this paper will be presented. Since the final status of any forced convection heat transfer problem is determined by the detailed structure of momentum transfer in highly unsteady, rotating and turbulent viscous flow environment, emphasis will be placed on pertinent turbine aerodynamic features existing in rotating facilities.

CURRENT GAS SIDE HEAT TRANSFER ISSUES

The ten major issues in gas turbine aero-thermal research that remain today were presented by Bunker [2006]. He mentioned that effective solutions to these ten issues are required to address the remaining gas turbine problems, in realizable engineering. Figure 1 shows a general classification of the challenging problems. The present day gas turbine heat transfer technology usually meets the strict requirements of modern gas turbine engines. The improvements in this technology in the areas of efficiency and life came with small increments over a long duration. The last significant achievements in the technological plateau reached today was the widespread application of shaped film cooling holes and the implementation of thermal barrier coatings. More recent advancements in the last few years were in the area of the understanding of detailed flow physics and in the development and use of advanced analytical/computational analysis tools for aero-thermal problems in the gas side of turbine engines. Bunker [2006] considers several important questions to be answered in pushing aero-thermal technology further. His first question is whether to cool a component only at selected hot spots or globally as a whole component. Whether the overall shielding performance against incident heat load could be improved over the level reached by TBCs is his second question. He considers to maintain a component temperature closer to hot gas temperature as one way of minimizing the heat loads from the gas side. A more isothermal component is likely to contain a less significant internal thermal stress field. Most modern turbine systems are effectively cooled and they usually operate well above the maximum allowable material temperature. High heat loads in modern systems and a simultaneous implementation of effective cooling result in higher through wall thermal gradients and higher in plane thermal gradients. Moving the thermal energy to somewhere else and dealing with it at its new location is also considered by Bunker [2006]. A comprehensive treatment of convective heat transfer and aerodynamics in axial flow turbines can be found in an IGTI Gas Turbine Scholar Lecture by Dunn, [2001].

RESEARCH SIMULATION ISSUES

Proper scaling of the gas turbine operating environment requires the replication of Reynolds number, molecular Prandtl number of the gases, Mach number, wall to free-stream temperature ratio, specific heat ratio of the gases. The simulation parameters listed here can be directly obtained from the non-dimensional forms momentum conservation equation and thermal energy equation.

$$\rho^* \frac{\partial V_i^*}{\partial t^*} + \rho^* V_j^* \frac{\partial V_i^*}{\partial x_j^*} = - \frac{\partial p^*}{\partial x_i} + \frac{1}{Re} \frac{\partial^2 V_i^*}{\partial x_j^* \partial x_j^*} \quad \text{where} \quad Re = \frac{\rho_o V_o L}{\mu_o} = \frac{V_o L}{v_o} \quad (1)$$

$$\rho^* c_p^* \frac{DT^*}{Dt^*} = \frac{V_o^2}{(T_w - T_o) c_{p_o}} \frac{Dp^*}{Dt^*} + \frac{k_o}{V_o \rho_o c_{p_o}} \nabla^2 T^* + \frac{\mu_o V_o}{L \rho_o c_{p_o} (T_w - T_o)} \Phi^* \quad \text{where} \quad \Phi^* = \frac{\Phi}{\mu_o V^2 / L^2} \quad (2)$$

Viscous dissipation rate in its dimensional form is defined as follows:

$$\Phi = u \cdot \left[2\left(\frac{\partial u}{\partial x}\right)^2 + 2\left(\frac{\partial v}{\partial y}\right)^2 + 2\left(\frac{\partial w}{\partial z}\right)^2 + \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)^2 \right] + \lambda \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)^2 \quad (3)$$

$$\rho^* c_p^* \frac{DT^*}{Dt^*} = Ec \cdot \frac{Dp^*}{Dt^*} + \frac{1}{Re \cdot Pr} \nabla^2 T^* + \frac{Ec}{Re} \cdot \Phi^* \quad \text{where } Ec = \frac{V_o^2}{(T_w - T_o)c_{p_o}} \quad \text{and } Pr = \frac{\mu_o c_{p_o}}{k_o} \quad (4)$$

The final form of the energy equation as shown in equation 4 indicates that a proper simulation of energy transfer in any environment where forced convection heat transfer is significant; one needs to simulate Reynolds number Re, Prandtl number Pr, and Eckert number Ec. Eckert number represents the ratio of the kinetic energy of the mean flow to the thermal energy carrying capability of the molecular domain. Eckert number may be easily substituted with “free-stream gas to wall temperature ratio”, Reynolds number and Mach number in the region of interest. The equation of state $p_\infty = \rho RT_\infty$ and the local speed of sound $a = \sqrt{\gamma RT_\infty}$ can be re-arranged to show $a^2 = \gamma \cdot p_\infty / \rho_\infty = c_{p_o} (\gamma - 1) \cdot T_\infty$. After re-arrangement, Eckert number can be expressed as follows :

$$Ec = \frac{\frac{V_o^2}{a^2}}{\frac{(T_w - T_o)c_{p_o}}{c_{p_o}(\gamma - 1) \cdot T_\infty}} = (\gamma - 1) \cdot M^2 \cdot \frac{T_\infty}{(T_w - T_o)} = (\gamma - 1) \cdot M^2 \cdot \frac{\frac{T_\infty}{T_o}}{\left(\frac{T_w}{T_o} - 1\right)} \quad \text{where } \frac{T_o}{T_\infty} = [1 + (\gamma - 1) \cdot M^2] \quad (5)$$

The final form of the energy equation suggests that one needs to simulate Eckert number by operating at a realistic Mach number M, free-stream to wall temperature ratio T_∞/T_w and specific heat ratio γ . This condition is in addition to Re and Pr, “for proper simulation ability” in a forced convection heat transfer problem. Simulation for a gas turbine environment also needs to include a proper mass flow function $\dot{m} \sqrt{R \cdot T_o} / D^2 P$, corrected speed $N \cdot D / a_o = N \cdot D / \sqrt{\gamma \cdot R \cdot T_o}$ and a rotation number that is also called as Rossby number $Ro = U / \Omega D$. The last set of requirements directly result from a dimensional analysis effort performed on stagnation enthalpy change, power output and efficiency of a turbomachinery system.

The scaling issues considered up to this point only considered a laminar flow environment in a momentum and convective heat exchange problem. In a modern gas turbine, combustor exit flow contains a significant amount of turbulence and free-stream unsteadiness. The turbulence intensity and length scale of turbulence are the two parameters that should be considered very carefully. The rotor blades of a gas turbine continuously pass through the highly turbulent and low axial momentum wakes of nozzle guide vanes generating a highly unsteady flow in the rotor passages. In nozzle guide vanes with a transonic/supersonic exit flow, the rotor blades may also chop through shock waves originating from the NGV trailing edges. In the relative flow environment of the rotor exposed to hot gases, blade surfaces see the periodic passing of NGV wakes, shock waves and high momentum fluid originating from the central sections of the upstream passage. The wall shear stress generation on gas turbine surfaces and heat transfer to these surfaces is highly three-dimensional and unsteady. When the gas turbine is film cooled, the typical additional scaling parameters are the blowing rate which is the coolant to free-stream mass flux rate ratio ($M = \rho_c U_c / \rho_\infty U_\infty$) and the coolant to free-stream temperature ratio T_c / T_o . The ratio of the coolant to free-stream momentum flux rate is also a popular cooling performance parameter ($I = \rho_c U_c^2 / \rho_\infty U_\infty^2$). A mass flow rate ratio of the coolant to free-stream $\dot{m}_c / \dot{m}_\infty$ for a stage or a complete turbine may also be used as a cooling performance indicator.

KEY SIMULATION PARAMETERS

The most important scaling parameter in a gas turbine heat transfer problem is the free-stream gas to wall temperature ratio. The very high temperature level of gas temperatures in a modern turbine makes full-scale investigations technically challenging and expensive. Figure 2 shows the combustor exit temperature and pressure for a number of advanced gas turbine configurations that are successfully operated in the past. Instead of working with the absolute level of the free stream gas temperature, operating with the “the free-stream gas to wall temperature ratio”, the momentum transfer and heat transfer in a gas turbine could be effectively simulated under relatively manageable testing conditions. In general, reducing the absolute level of the gas temperature has a very advantageous effect in the design of a turbine test facility. Lowered free-stream gas temperatures require a lower operating pressure when the Reynolds number is fully matched between the actual operation and the test facility.

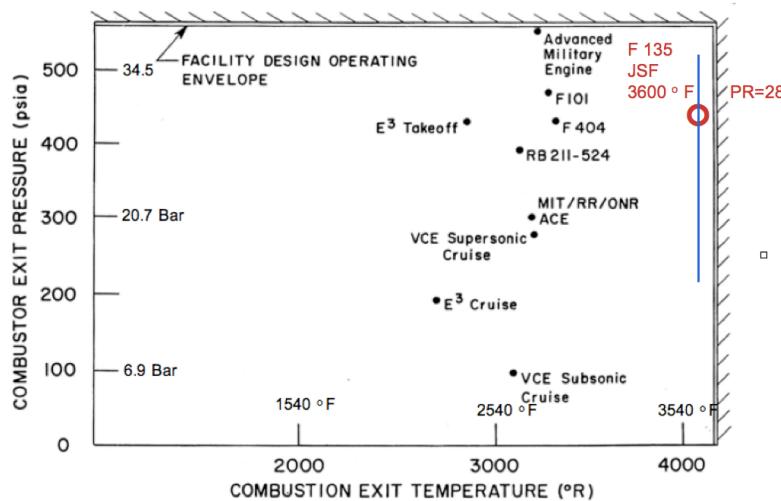


Figure 2. Typical gas turbine entry temperatures and pressures from existing gas turbine systems

Lowered gas temperature operation has another significant benefit. The rotational speed of the test facility is also reduced when the absolute level of the gas temperature is reduced. The mechanical stresses on the test facility and rotating instrumentation system are simultaneously reduced simplifying the facility design and operation. The reduction in rotational speed reduces the frequency response requirements of instrumentation especially for time-accurate and phase-locked measurements in the rotor frame of reference, Epstein et al. [1985]. The measurement of unsteady heat transfer, wall shear stress, aspirating probe measurements for simultaneous total pressure&temperature measurements, hot wire measurements, hot film measurements, time accurate tip clearance measurements, particle image velocimetry measurements and pressure sensitive paint measurements all fall into a category that benefits from relatively reduced frequency response operation at lowered rotational speed in facilities realistically simulating present day turbines.

Although the low speed operation of a research turbine may not be sensitive to the specific heat ratio $\gamma=c_p/c_v$, a proper matching of this fluid property can not be ignored for transonic and supersonic turbines. The specific heat ratio γ is a strong property of the gas composition that can be adjusted by using a combination of low and high γ molecules. Increasing the molecular weight of the gases in a turbine decreases the operating pressure required at a selected Reynolds number and decreases the speed of sound of the gas thus reducing the turbine rotational speed. The heaviest mono-atomic gas which is readily available is Argon, Epstein et al., [1985]. For example, a 75 % Argon and 25 %

Freon-12 mixture is an ideal composition for scaled turbine testing. Freon-14 itself may be an effective candidate as turbine cooling gas, however the usage of Freon is currently restricted due to environmental reasons. The rotational speed of the turbine disk is usually adjusted to keep the corrected speed $N \cdot D / \sqrt{\gamma \cdot R \cdot T_o}$ constant. Scaled turbine operation requires an operation at significantly reduced mass flow rate and power level.

AERO-HEAT TRANSFER RESEARCH IN SIMULATED TURBINE RIGS

The rotating turbine research rigs for aero-heat transfer studies range from full-scale gas turbine demonstrators to short duration facilities, from rotating coolant passage simulators to large scale/low speed turbine research facilities, from rotating disk cavity research rigs to tip leakage flow simulators. While instrumented full-scale gas turbine demonstrators are excellent facilities to generate very realistic gas turbine heat transfer data and performance evaluation; the initial investment made to construct them, their extremely high operational costs and the technological challenges in performing reliable and high-resolution aero-thermal measurements limit their current use.

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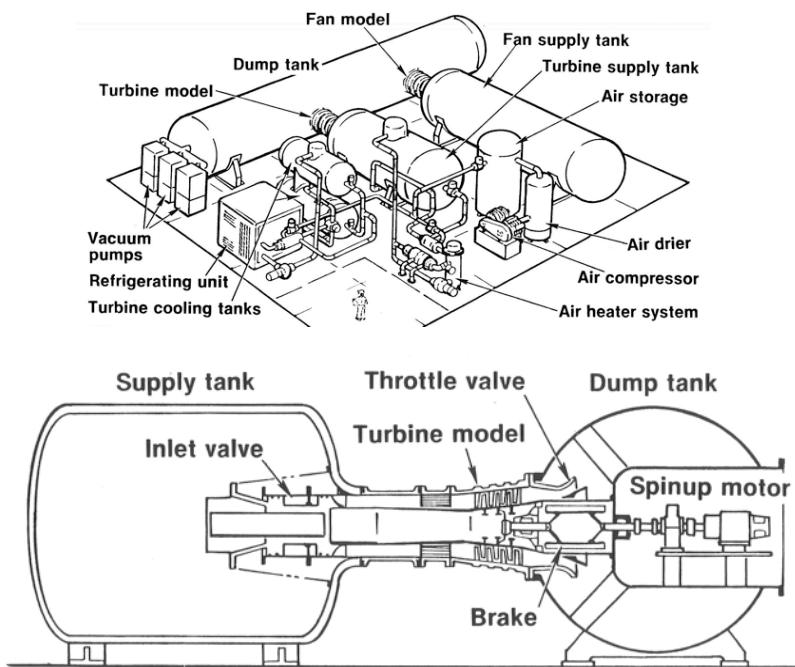


Figure 3. Proposed turbine/fan test facility at UTRC, Case [1985] (not constructed)

Large-scale and low speed research facilities are also frequently used in aero-heat transfer research because of their low operating costs, high reliability and operational safety characteristics. One of the first turbine research rigs was operated at United Technologies Research Center at Connecticut. The currently operational low speed, large-scale turbine research rigs are at the Aero-heat Transfer Laboratory of the Pennsylvania State University, Purdue University and Texas A&M University. There are also a number of low-speed/large-scale turbine research facilities currently operational used mainly for the improvement of stage aerodynamic features.

A Proposed Blow Down Turbine/Fan Test Facility at UTRC: A blow down research facility was designed by UTRC for aero-heat transfer studies in turbines and fans, Case, [1985]. Although this facility was never built, the design had a unique property of being able to test both a turbine or a fan in a short duration mode. Figure 3 shows the turbine model and fan model connected between a common supply tank and a large dump tank. The turbine test section was designed to have a multi-stage and cooled turbine including an adjustable throttle valve, rotor spin-up motor and an eddy-current-brake for torque absorption and speed control. The facility allowed the use of up to 4 turbine stages with a maximum diameter of 37 inches. The quick acting inlet valve operating in less than 50 ms located inside the supply tank was one of the most important components in this turbine test rig, allowing a total test duration of 1 second. This 1 second long test duration included the initial transient disturbances due to quick opening of the inlet valve. The facility inlet had provisions to simulate inlet pressure and temperature profiles and a heat exchanger section for turbine inlet temperature level control. A 100 HP spin up motor located in the initially evacuated dump tank brought the turbine rotor to its rotational speed before the opening of the quick acting inlet valve. The pressure ratio between the supply tank and the dump tank controlled the Mach number simulation in this facility. The level of the supply tank initial pressure and the throttle valve controls the mass flow rate and Reynolds number in the turbine rig. The facility design allowed testing every 20 minutes and had provisions for boundary layer bleed at turbine inlet.

TABLE 1
TURBINE DESIGN PARAMETERS

Turbine loading, AH/U^2	-2.3
Total pressure ratio	4.2
Velocity ratio, C_d/U	0.63
Rotor aspect ratio	1.5
NGV exit Mach No.	1.18

TABLE 2
MIT BLOWDOWN TURBINE SCALING

	Full Scale	MIT Blowdown
Fluid	Air	Ar-Fr 12
Ratio specific heats	1.28	1.28
Mean metal temperature	1118°K (1550°F)	295°K (72°F)
Metal/gas temp. ratio	0.63	0.63
Inlet total temperature	1780°K (2750°F)	478°K (400°F)
True NGV chord	8.0 cm	5.9 cm
Reynolds number*	2.7×10^6	2.7×10^6
Inlet pressure, atm	19.6	4.3
Outlet pressure, atm	4.5	1.0
Outlet total temperature	1280°K (1844°F)	343°K (160°F)
Prandtl number	0.752	0.755
Eckert number†	1.0	1.0
Rotor speed, rpm	12,734	6,190
Mass flow, kg/sec	49.0	16.6
Power, watts	24,880,000	1,078,000
Test time	Continuous	0.3 sec

* Based on NGV chord and isentropic exit conditions

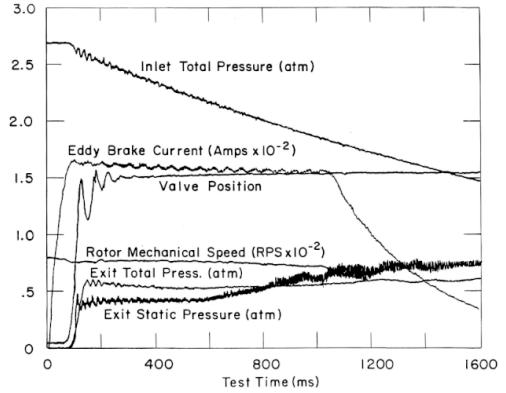
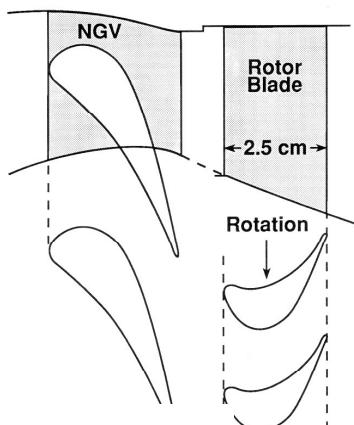


Figure 4. MIT blow down rig design parameters, turbine scaling, and timing, Epstein et al. [1984]

MIT Blowdown Turbine Test Facility: MIT turbine test rig is a short duration facility that is capable of testing film-cooled, high work aircraft engine stages for a typical duration of 400 ms, Epstein et al. [1984]. This facility can generate test rig conditions that are properly simulating turbine operation up to 40 atm inlet pressure and 4000 °F (2500 °K) turbine inlet temperature. 0.5 m diameter test facility is intended for the investigation of three-dimensional and unsteady turbine flow and convective heat transfer. A full experimental simulation of a turbine stage including wall

to free-stream temperature ratio, coolant to free-stream temperature ratio, Reynolds number, Mach number, Rossby number, corrected speed, weight flow and specific heat ratio is possible in a relatively large diameter stage in which detailed unsteady flow, heat transfer and performance indicating instrumentation is available. Figure 4 shows major turbine design parameters and turbine scaling characteristics and a mid-span blade-to-blade view of the turbine stage. Epstein et al. [1984] mentions that the instrumentation access, initial cost and high time response instrumentation are the prime concerns in the design of this facility. The simulation characteristics presented in Figure 4 indicates that a realistic free-stream gas to wall temperature ratio of 0.63 can be realized at 295 ° K for a full scale operation of 1780 ° K (2750 ° F). A full-scale coolant temperature level of 790 ° K is reduced to 212 ° K in the test rig. The lowered free-stream gas temperature in the rig allows a turbine disk operation at almost half speed ($N=6190$ rpm) of the full-scale operation (12,734 rpm). MIT blow down turbine facility operates at a turbine inlet total

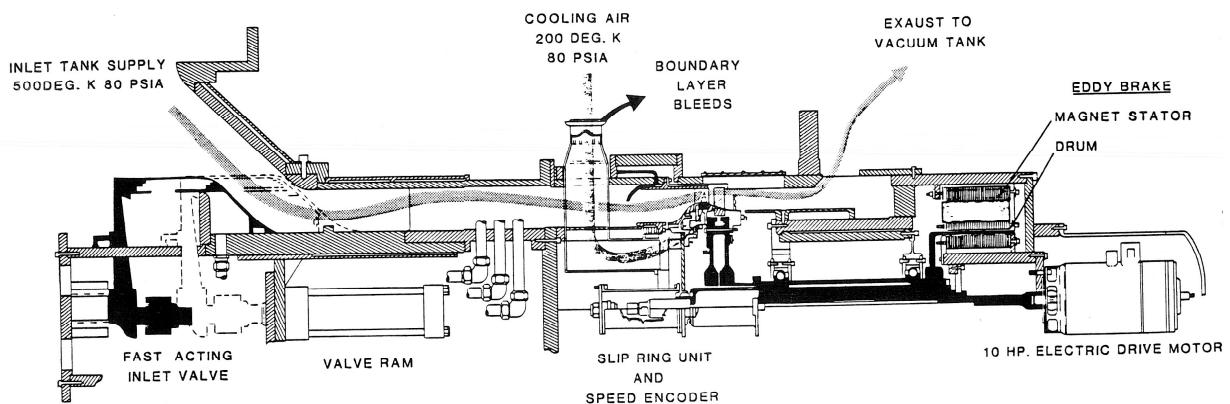
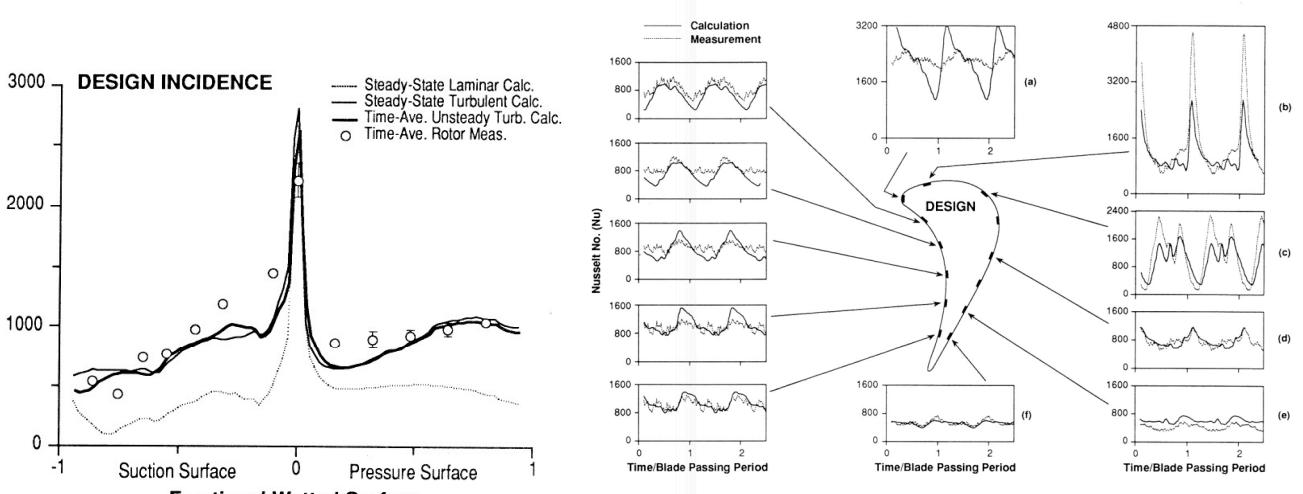


Figure 5. MIT turbine test section and fast acting inlet valve, Epstein et al. [1984]

pressure of 4.3 atm (64 psia) whereas the full-scale hardware runs at 19.6 atm (289 psia). Other important test rig parameters in comparison to the full-scale hardware are shown in Figure 4. The specific heat ratio γ in the facility is adjusted using a combination of a mono-atomic gas Argon and other heavier molecular weight gases that are refrigerant gas mixtures. The fast opening valve



isolating the test section from the supply tank opens fully in 50 ms and is designed for a supply tank pressure of 10 atm when the test section is near vacuum conditions. The supply tank temperature Figure 6. Rotor blade heat transfer measurements in MIT turbine facility, Abhari et al. [1991]

can be adjusted to a level as high as 530 ° K. A 0.9 m diameter annular cone shaped plug seals against the inner and outer annuli of the flow path as shown in Figure 4. The cone slides forward to open on graphite rings along its shaft to form a smooth annular contraction from the supply tank to the test section. The contraction is a part of the boundary layer control system that also includes a boundary layer bleed arrangement. Time resolved turbine rotor blade heat transfer measurements performed in MIT blowdown turbine rig are compared with computations by Abhari et al. [1991]. The influence of hardware manufacturing tolerances on rotor heat transfer variation is discussed. A multi-layer thin film heat flux gauge using a 25 μm thick polyimide insulator is adhesively bonded to the blade profile. The sensing area is limited to 1 x 1.3 mm. Above 1 Khz, the 25 μm thick insulator appears infinitely thick to the top surface. The entire frequency domain in heat flux measurements from d.c. to 100 Khz is reconstructed where the blade passing frequency is 3.6 Khz. The heat flux signals are digitized at 100 KHz sampling rate and ensemble averaged for 360 vane passing periods. The absolute calibration accuracy of the time-accurate heat flux sensors are quoted to be about 10 %, Abhari et al [1991].

A comparison of the time averaged measured rotor heat transfer, the unsteady NGV-rotor calculation and the steady state solution for the rotor alone are also compared in Figure 6 for design incidence. The Nusselt number is defined using the blade axial chord, heat flux, inlet relative total minus local blade temperature and thermal conductivity at blade local temperature. The difference between the steady state turbulent and time averaged unsteady calculations is shown to be minimal. The measured heat transfer rates in general, follows the fully turbulent computation levels. As a whole blade average, the predicted heat transfer rate is about 11 % less than the measured value. Time resolved heat transfer data at design incidence are presented in Figure 6 along both the suction and pressure surfaces. The unsteady flow influence on heat transfer signals is shown. Near the stagnation point, the computations using a Baldwin-Lomax turbulence model show a much higher degree of unsteadiness when compared to the measurements. The unsteadiness of heat transfer data is a result of NGV wake impingement on the rotor blade surfaces and potential coupling that includes moving shock patterns emanating from NGV trailing edges. A very interesting conclusion from Abhari et al.'s [1991] study was that the prediction of heat transfer in highly loaded turbines is more sensitive to the details of the inviscid flow prediction than to the particulars of the turbulence model employed. Another interesting observation in his study was the small difference between the steady fully turbulent rotor calculation and the time average of the unsteady heat transfer calculation. The unsteadiness played a very small role in the specific highly loaded transonic turbine even though both the measurements and computations show very significant unsteady fluctuations in time accurate heat transfer data.

UTRC's Large-Scale Rotating Rig (LSRR): United Technologies Large-Scale Rotating Rig (LSSR) was one of the first rotating turbine research facilities that performed significant amount of aerodynamics and heat transfer research in a low speed turbine environment where the first three rows of the test rig represented a high by-pass ratio aircraft engine. General characteristics of this rotating test facility is described in Dring and Joslyn [1981]. This 1 ½ stage low speed facility operated from laboratory air conditions at about 1 atm inlet total pressure and 294 K inlet total temperature with a flow coefficient of 0.78. The experimental turbine having a 0.8 hub/tip ratio had 22 first vanes, 28 first blades and 28 second vanes. The model airfoil chord lengths were about five times full-scale hardware chord lengths. The rotor speed was about 410 m/s and a typical airfoil Reynolds number was approximately 560,000 which was typical of high pressure turbine airfoils. The effects of turbulence and stator-rotor interactions on turbine heat transfer was studied by Blair et al. [1988 a,b]. Figure 8 presents the Stanton number distribution on the suction side of the LSSR rotor blade in comparison to Sharma et al.'s [1990] computations. A very good match between the unsteady computations and rotor heat transfer measurements was shown. An important result from

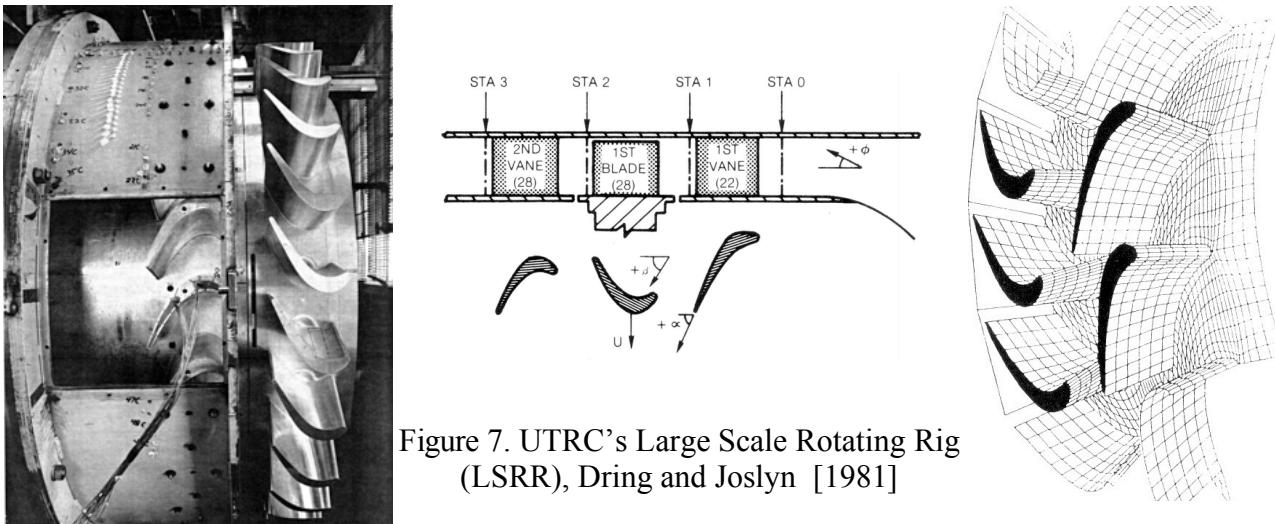


Figure 7. UTRC's Large Scale Rotating Rig (LSRR), Dring and Joslyn [1981]

Sharma's unsteady numerical simulations indicated that time-averaged loads on airfoil surfaces (and endwalls) are not influenced by unsteadiness even in the presence of temperature streaks. Unsteady computations clearly show that the pressure sides of turbine airfoils operate at higher temperatures than the suction sides in the presence of circumferentially non-uniform temperature profiles.

Blair, Wagner and Steuber [1991] developed two interesting liquid crystal techniques to obtain steady-state heat transfer data on the gas side of rotating turbine airfoils in LSSR. In one study they developed a liquid crystal based method to measure heat transfer rates on the rotating blade surfaces. The second study was geared towards heat transfer measurements in rotating turbine coolant passages. Figure 9 shows the sample liquid crystal temperature patterns on the rotor airfoil at two different rotational speeds ($N=407$ rpm and 160 rpm). This approach used slightly non-rectangular “constant heat flux surfaces” for the measurement of heat transfer coefficients. The deviations from the constant heat flux value due to slightly non-rectangular heater shapes were computed by analytically calculating non-uniform current density functions in the actual heater shapes. The liquid crystal images were obtained in the stationary frame of reference by using a strobe light during the passage of the rotor blades. The rotor was instrumented by arrays of 89, 124, and 102 thermocouples on the pressure side, suction and endwall surfaces respectively. Thermocouple signals and heater electrical connections from the rotor to the stationary frame of reference were transmitted by using a slip-ring device.

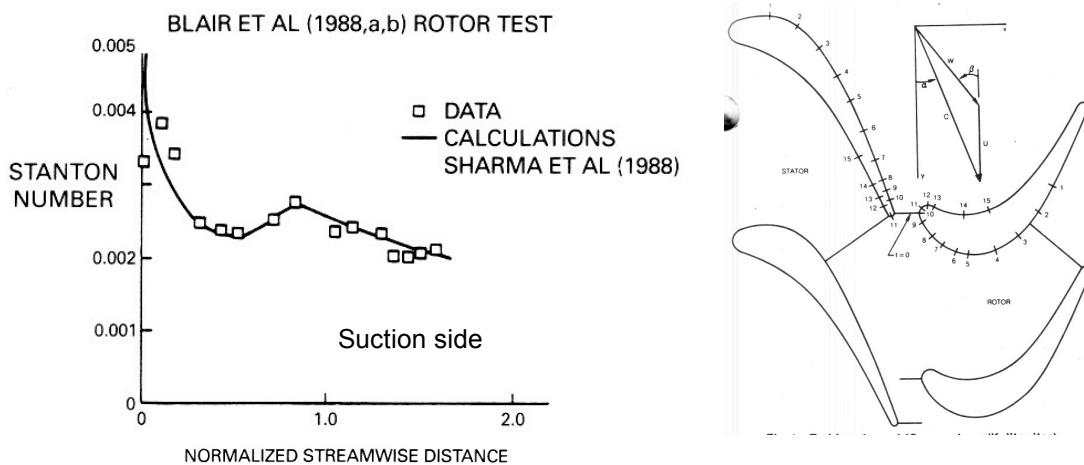


Figure 8. Comparison of LSRR rotor heat transfer data with computations Blair et al. [1988,a,b]

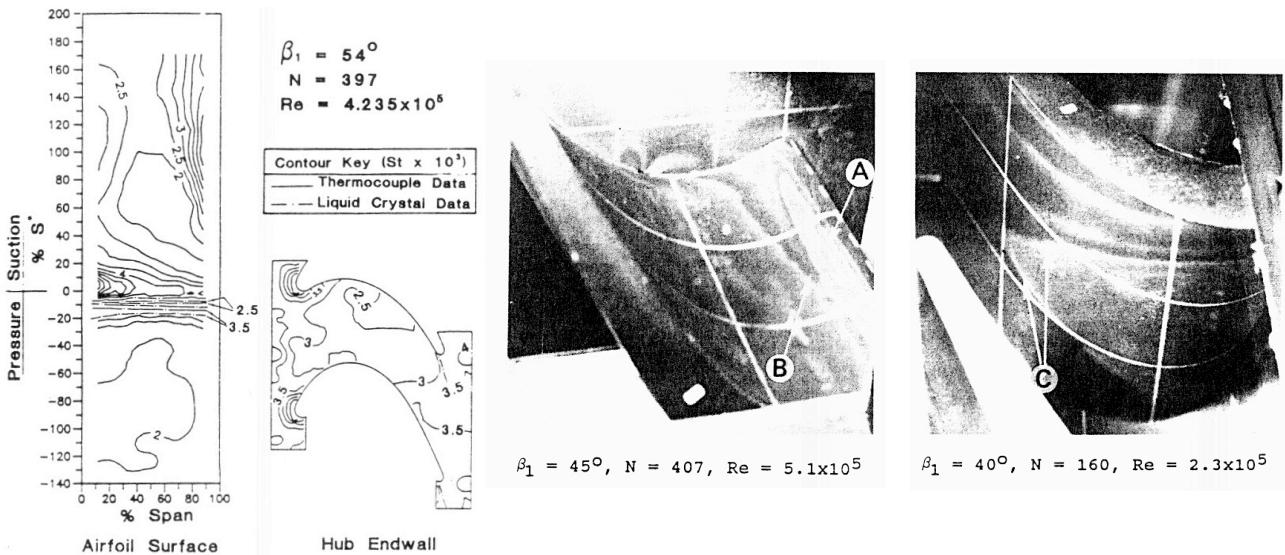


Figure 9. Stanton number measurements on the rotor surfaces of (LSRR) Blair et al. [1991]

Figure 9 shows the contour maps of Stanton numbers on the suction side and pressure side by using a rectangular projection of the unwrapped airfoil surface. Liquid crystal patterns were used to supplement the thermocouple data in regions where extremely localized effects existed. A separation bubble from the heat transfer measurements was clearly visualized. A highly three dimensional pattern near the suction side, especially near the tip shows the influence of the tip leakage vortex and the secondary flows on the surface heat transfer distribution. Strong variations in Stanton number contours near the suction side of the leading edge can be attributed to the horseshoe vortex formation due to the momentum deficit in the inlet endwall boundary layer.

Turbine Stage in the Oxford Rotor Facility ILPT: Turbine stage in the Oxford rotor facility as shown in Figure 10 uses an Isentropic Light Piston compression Tunnel ILPT to obtain well simulated gas turbine conditions. Schultz et al. [1977] showed that the temperature differences encountered in gas turbine components could be scaled down in light piston compression tube experiments. The validity of this approach was established by Hilditch and Ainsworth [1990]. The turbine stage is a transonic shroudless turbine of 0.5 m tip diameter as explained in Ainsworth et al.

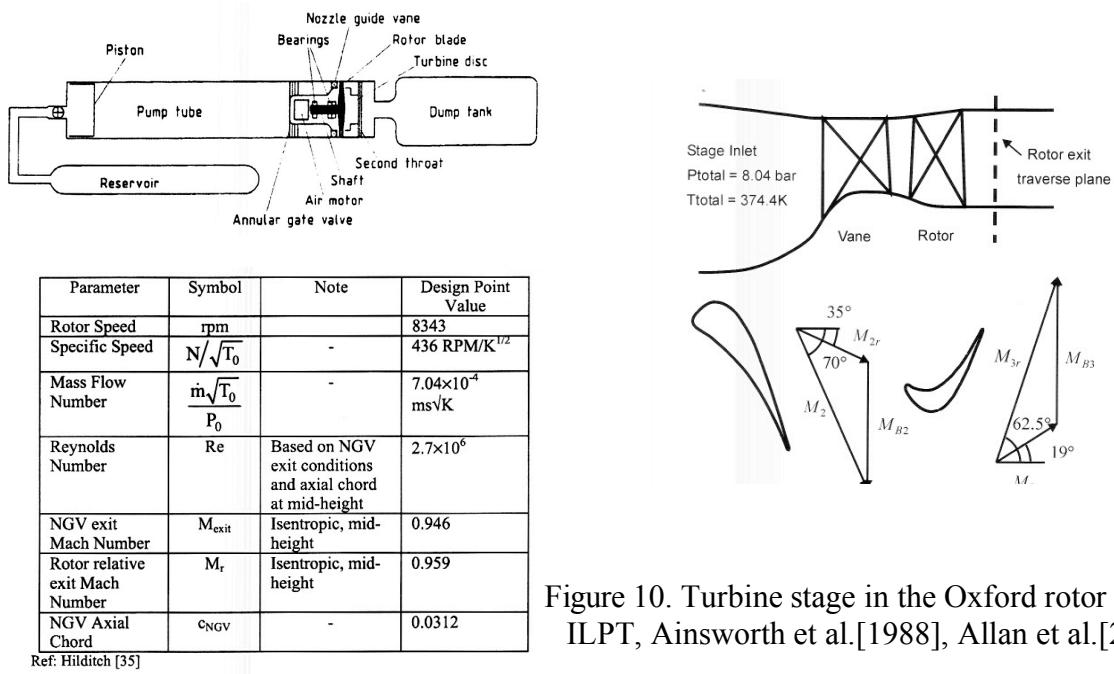


Figure 10. Turbine stage in the Oxford rotor facility ILPT, Ainsworth et al.[1988], Allan et al.[2004]

[1988]. The specific turbine has 36 NGVs and 60 high-pressure turbine blades. The NGV exit Reynolds number is 2,700,000 and the pressure ratio of the turbine is 3.12. The rotational speed of the rotor is 8343 rpm and the blade tip clearance is about 2.25 % of the blade span. Other details of the facility operation and stage schematic are shown in Figure 10.

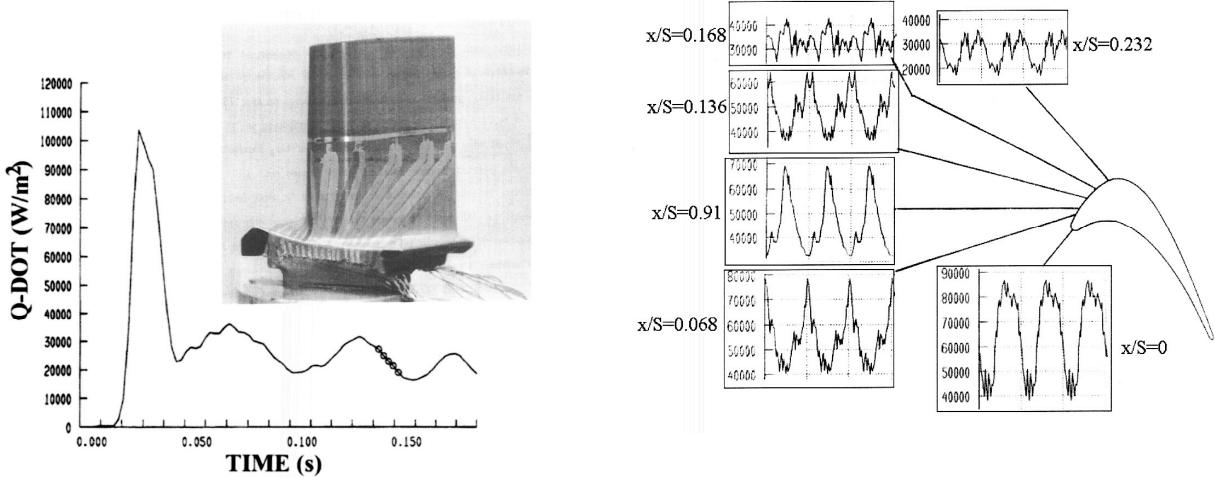


Figure 11. Unsteady heat flux sensors and heat flux measurements in turbine stage in the Oxford rotor facility ILPT, Allan et al., [2004]

Figure 11 shows the mid-span thin film heat flux sensors and a typical heat flux distribution from a pressure surface heat flux sensor. The oscillations in the heat transfer signal are due to the finite weight of the piston used in the compression tube facility. 180 ms of the short duration test is shown in the figure. Unsteady heat flux signals from the frontal part of the suction side are presented in Figure 11. The large fluctuations existing near the leading edge of the suction side start diminishing in magnitude at about $x/S=0.351$. High fluctuation levels are usually explained by the passage of the highly turbulent NGV wakes, Allan et al. [2004]. Figure 11 shows a double peak from all sensors between $x/S=0.068$ and 0.232. The spacing between the peaks varies due to upstream motion of the shock emanating from the trailing edge of the NGV and the downstream motion of the wake segment. Allan et al. also observed sharp heat flux spikes in heat transfer rate $x/S=0.479$ on the suction side. They attributed the spikes to the presence of reflected trailing edge shocks from the adjacent rotor blade.

The VKI Compression Tube Turbine Test Facility CT3: The VKI research turbine stage shown in Figure 12 is a representative of a wide class of aero-propulsion engines. The stage is driven by a

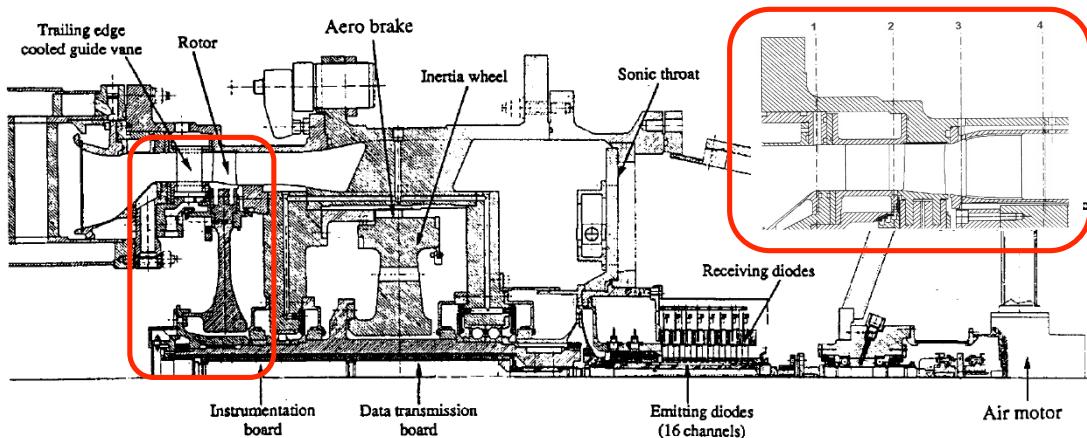


Figure 12. The VKI turbine research rig, Sieverding and Arts [1992], Dénos, R., [1996]

light piston compression tube facility CT3 that allows the adjustment of the Reynolds number and Mach number, the free-stream to wall and coolant to wall temperature ratios under realistic gas turbine conditions, Sieverding and Arts [1992], Denos [1996] and Sieverding et al. [1998]. Figure 12 shows the original facility configuration that consists of 43 trailing edge cooled guide vanes and a rotor system with 64 blades. The inset shows the modified turbine stage with the inclusion of a second stator system with 43 vanes. The second stator can be rotated in the azimuthal direction to investigate clocking effects. The tip clearance that has a weak variation from the leading edge to trailing edge is approximately 0.6 % of the rotor blade height. The operational conditions of the turbine rig are summarized in Figure 13.

1 & ½ STAGE FLOW CONDITIONS			
P_{01} [bar]		2.23	
T_{01} [K]		480	
P_{01}/P_{03}		2.63	
P_{s4}/P_{03}		0.72	
Rotational Speed [RPM]		6500	
M_{2is}	1.06	$Re_{2,Cs}^{(*)}$	1.2×10^6
$M_{3is,r}$	0.89	$Re_{3r,Cr}^{(*)}$	5.2×10^5
M_{3is}	0.42		
M_{4is}	0.68	$Re_{4,Cs}^{(*)}$	4.8×10^5

* Based on chord & outlet velocity (relative frame for rotor)

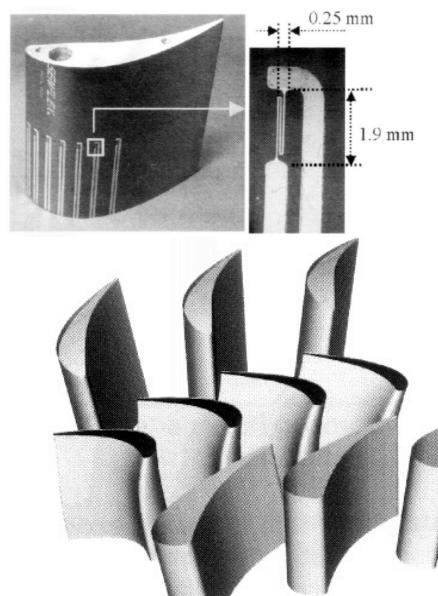


Figure 13. 1 & ½ stage VKI turbine research rig operational conditions, Sieverding et al. [1992,1998], and two-layered “Senflex” thin film heat transfer gauge array attached on the suction side of the second stator, Illiopoulos et al. [2004]

The VKI turbine research rig benefits from both single layered deposited thin film sensors and two-layered gauges for unsteady heat flux signal acquisition. A two-layered “Senflex” thin film heat transfer gauge array attached on the suction side of the second stator is shown in Figure 13. A successful implementation of this specific heat flux sensor array is explained in detail in Illiopoulos et al [2004]. Figure 14 presents unsteady Nusselt number traces in function of the rotor position from phase-locked heat transfer measurements performed on the second stator of the VKI turbine. The signals were obtained from two pressure side locations in the first 1/3 chord region. Four different repeats from the two sensors indicate excellent repeatability. The results are ensemble averaged for three complete rotor revolutions or 192 rotor passing events. Phase-locked measurement approach allows the researcher to extract the random unsteady part of the signal. Quantifying the wake passing and the potential interactions between the rotor and stator is of extreme importance when interpreting unsteady wall heating phenomena in transonic turbine stages. Figure 14 also shows the mean wall temperature, wall heat flux and heat transfer distribution on the pressure side and suction side of the second stator. The figure contains five different runs in the VKI turbine. Each run is obtained from a different initial wall temperature distribution in the rig. Since the turbine flow conditions are accurately repeated during each of the five individual experiments, the resulting Nusselt number distributions are almost the same as shown in Figure 14.

A three-dimensional computational simulation of the VKI transonic turbine stage is discussed in Michelassi et al. [2001]. Effect of the hub endwall cavity flow on the flow field of the VKI turbine

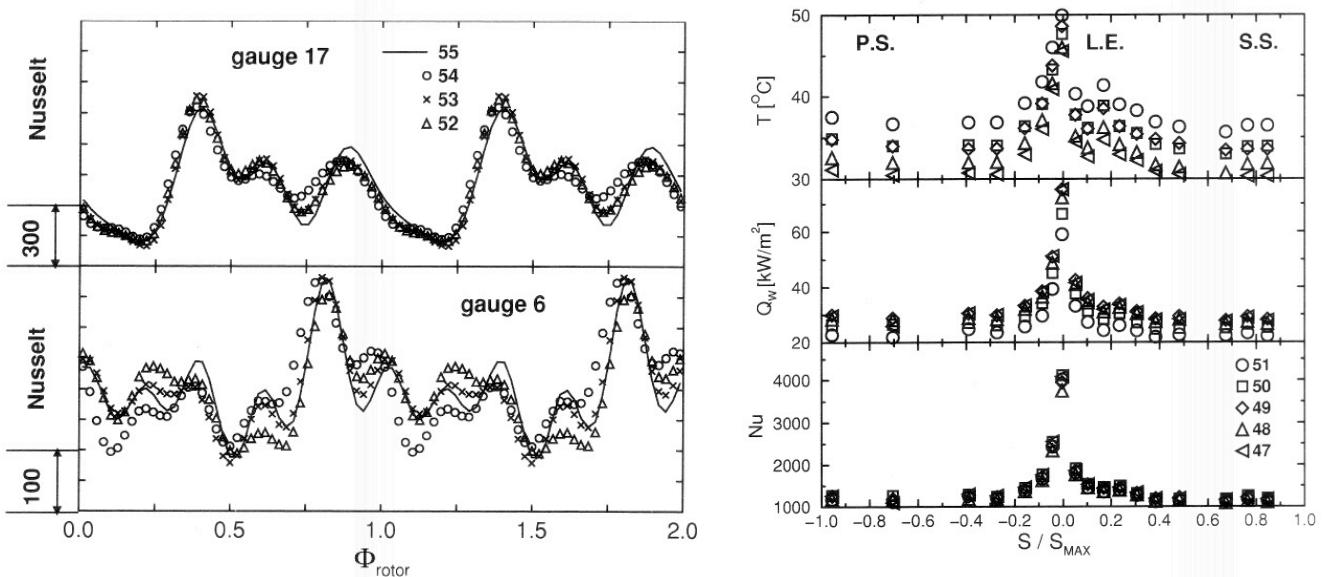


Figure 14. Phase-locked unsteady heat transfer measurements and time averaged heat transfer on the second stator of the VKI turbine research facility, Illiopoulos et al [2004]

stage is presented in Paniagua et al.[2004]. The effect of clocking and its influence on the aerodynamic and mechanical behaviour is investigated by Gadea et al. [2004].

Short Duration Turbine Research Facility (TRF) at Wright Patterson Air Force Base: Turbine Research Facility (TRF) at the Air Force Research Laboratory (Haldeman, et al, 1992) is presented in Figure 15. TRF is a transient “blow down” rig capable of operating for periods of several seconds at the correctly scaled Mach and Reynolds numbers of the turbines of actual engines. Before the start of a TRF test, the entire facility is evacuated and the fast acting valve between the supply tank and the test section is closed. Next, the supply tank can be filled with a

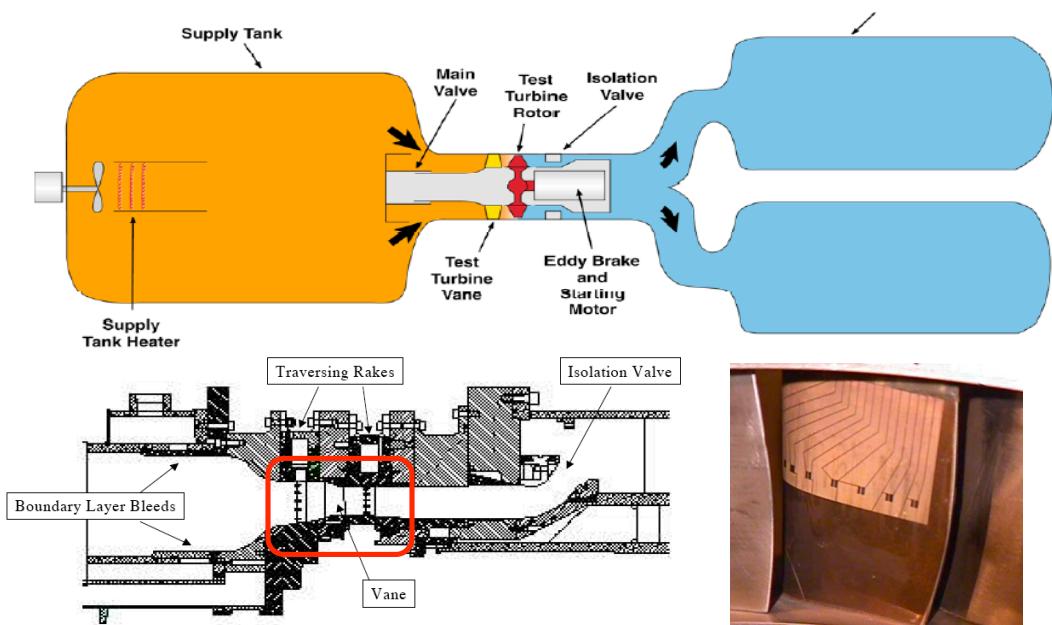


Figure 15. Short duration turbine research facility (TRF) stage and vane suction side heat transfer instrumentation at Wright Patterson Air Force Base, Haldeman et al. [1992]

mixture of N₂ and CO₂ whose composition is set to simulate the same specific heat ratio γ as that of combustion gases at the engine operating conditions. For most of the tests only nitrogen was used resulting in $\gamma = 1.4$ and a Pr = 0.71. This is reasonably close to the desired Pr of high temperature air, about 0.68. Figure 15 also shows a vane suction side surface that was wrapped with a 50 micron thick film sheet of Upilex similar to the flexible substrate used at the VKI turbine facility. This sheet was wrapped from the trailing edge to trailing edge of a single vane creating no surface discontinuities along the airfoil surface, Polanka and Meininger [2001]. Thirteen 0.04 micron thick platinum sensing elements were sputtered on top of the Upilex flexible substrate prior to installation on the vane suction side. The heat flux data from the gauges was determined by using the resistance change with time to a temperature trace via a calibration for resistance to temperature. The heat flux was then calculated based on the time rate of change of the surface temperature coupled with the thermal properties of the Pyrex or Upilex substrates.

Ohio State University Turbine Research Facility: This rotating turbine heat transfer facility utilized a shock-tunnel to produce a short-duration source of heated and pressurized gas that was passed through the turbine stage, Dunn and Haldeman [2000]. The experimental apparatus shown in Figure 16 consisted of an air driven shock tube, an expansion nozzle, a dump tank and a test system

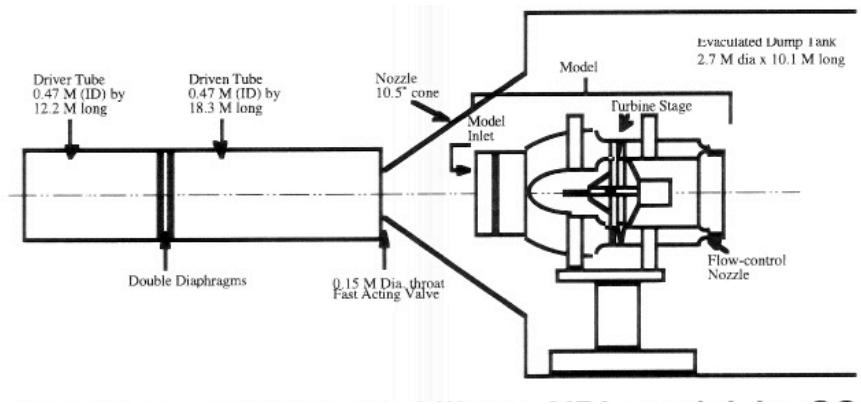


Figure 16. Shock tube driven turbine test facility at the Ohio State University,
Dunn and Haldeman [2000]

that housed the turbine stage. The shock tube has a 0.47-m diameter by 12.2-m long driver tube and 0.47-m diameter by 18.3-m long driven tube. The driver tube was designed to be long enough so that the wave system reflected from the driver endwall would not terminate the test time prematurely. The turbine is housed in a device located in the expansion nozzle of the shock-tunnel facility as illustrated in Figure 16. In order to start a heat transfer test sequence, the test section is evacuated while the driver and the driven tube are pressurized to predetermined values. Pressure values are selected to duplicate the design flow conditions. The turbine flow function, wall to-total temperature ratio, stage pressure ratios, and corrected speed are duplicated. The value of T_o can be set at almost any desired value in the range of 800°R (171 °C) to 3500°R (1671 °C). The pressure ratio across the turbine is established by altering the throat diameter of a flow control nozzle located near the exit end of the device housing the turbine, Haldeman et al. [2004]. The thin-film heat flux gauges were made of (~100 Å thick) platinum and were hand painted on an insulating Pyrex 7740 substrate. The response time of these thin films is in the order of 5×10^{-8} sec. The turbine stage shown in Figure 17 is a 1 & ½ stage (High-Pressure Turbine Vane-HPTV, High-Pressure Turbine Blade-HPTB, and Low-Pressure Turbine Vane-LPTV) turbine, configured in a manner that is representative of the highly 3D airfoil rows in a modern engine, Haldeman et al. [2004].

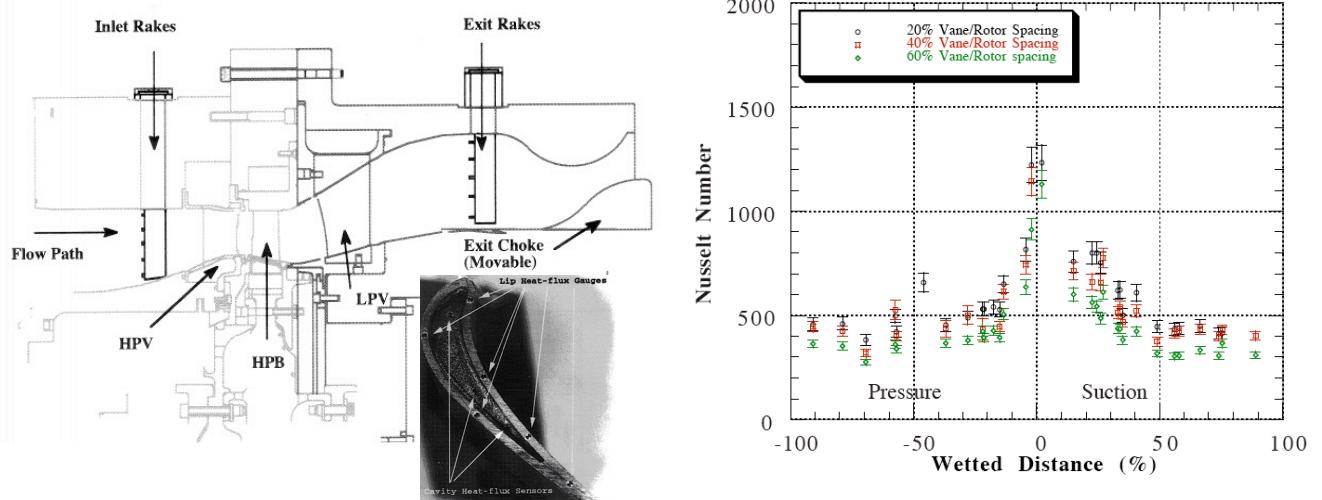


Figure 17. 1 & $\frac{1}{2}$ stage turbine stage at Ohio State University, Haldeman et al. [2004] and Nusselt number distribution, Dunn and Haldeman [2000]

The total pressure ratio across the rig is in excess of 5. The Ohio State turbine stage shown in Figure 17 has 38 vanes (both HP and LP) and 72 Blades. The exit choke is movable, so that a variety of pressure ratios are available. By controlling the pressure ratio across the stage, the inlet pressure, the corrected speed, and a wide variety of engine operating conditions can be replicated. The overall research effort performed in this rig was comprised of the aerodynamics of the turbine stage and unsteady heat transfer measurements performed around the rotor, rotor shroud and vane, Haldeman et al. [2004]. Heat-flux measurements and predictions from the blade tip region of this HP turbine was recently presented by Molter et al. [2006].

QinetiQ Light Piston Isentropic Compression Tube Driven Turbine Research Facility: The QinetiQ Isentropic Light Piston Facility (ILPF) as shown in Figure 18 is a short duration wind tunnel capable of “aero-heat transfer” testing of an engine size turbine at realistically simulated conditions, Chana et al. [2003]. This facility was originally used to test a single HP turbine stage. The facility was recently extended to 1.5 turbine stages with the installation of an IP stator. A unique feature of the facility is the aerodynamic turbo-brake that is on the same shaft as the turbine and is driven by the turbine exit-flow. At the design speed the turbo-brake power is matched with the turbine, and thus the turbo-brake maintains constant speed during a run as shown in Figure 18.

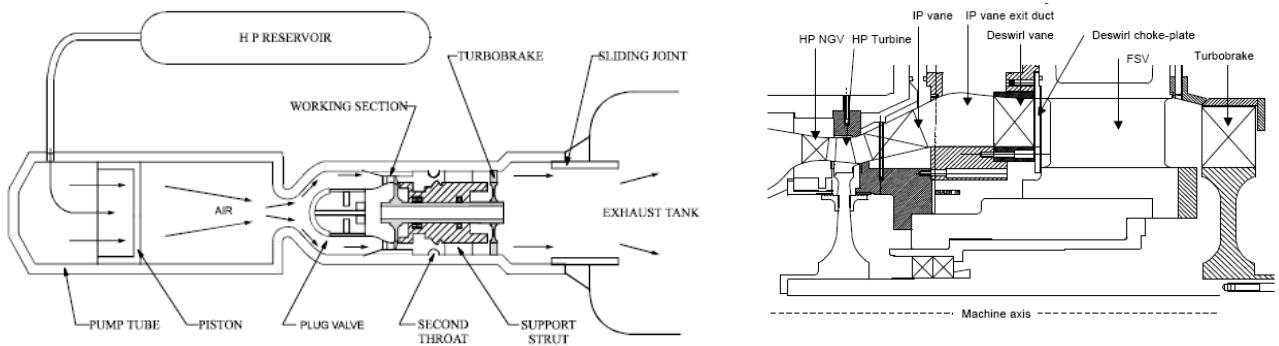


Figure 18. QinetiQ short duration turbine stage, Chana et al. [2003]

Air from a high pressure reservoir moves a piston down a piston tube, isentropically compressing and heating the working gas (air) inside the tube. When the desired pressure is reached, the

compressed air is suddenly released by means of a fast acting valve, into the working section. Steady-state turbine conditions are achieved for approximately 500 ms. The turbine stage is a modern high pressure Rolls-Royce aero-engine design with stage pressure ratio of 3.2 and nozzle guide vane Reynolds number of 2.54×10^6 . The stage is unshrouded and blade counts for the HP stator, HP rotor, and the IP stator are 32, 60 and 26 respectively, Chana et al. [2003].

Low Speed Large Scale Turbine Rig at Texas A&M University: The three stage low speed and large scale research turbine is the core component of the “Turbomachinery Performance and Flow Research Facility” TPFL at Texas A&M University. This facility, designed by Schobeiri [2000] is used for aerodynamics, efficiency, performance and heat transfer issues of high pressure, intermediate pressure, and low pressure turbine components. The overall layout of the test facility is shown in Figure 19, Suryanarayanan, [2006]. The rotational speed of this 0.685 m rotor can be adjusted between 1500 and 2550 rpm. The rotor rows have 46, 40 and 44 blades in a three-stage arrangement whereas the stator rows have 56, 52 and 48 airfoils. It consists of a 300 HP inverter controlled electric motor which drives the three-stage compressor in suction mode with a maximum pressure difference of 55 kPa and a volume flow rate of $4 \text{ m}^3/\text{s}$. The turbine inlet that has a heater

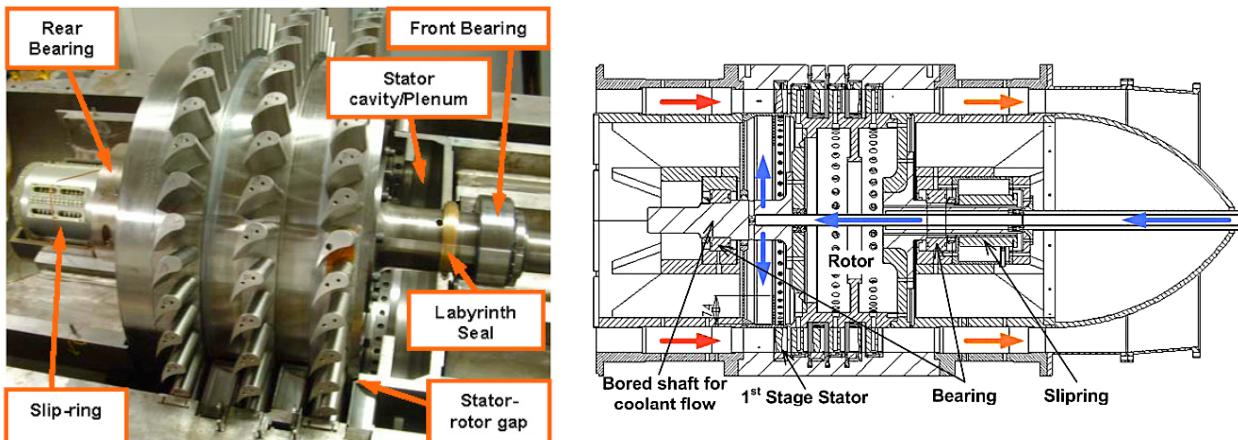
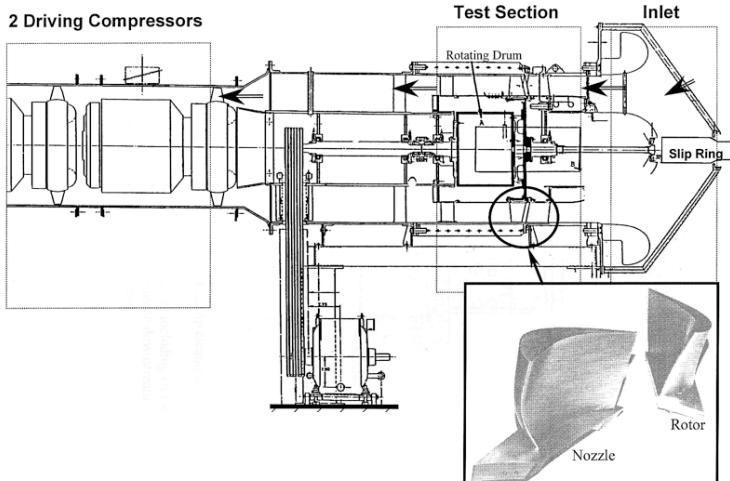


Figure 19 Three-stage, low-speed large-scale turbine research facility at Texas A&M University

section that is open to atmospheric conditions. The heater prevents condensation of water from humid air expanding through the turbine during the test. To determine the film cooling effectiveness under rotating conditions for leading edge film cooling, the turbine rotor was modified to integrate the coolant loop into the downstream section of the hollow turbine shaft and into the cylindrical hub cavity as described by Ahn et al. [2005]. This turbine rig is one of the first rotating facilities in which a pressure sensitive paint based film cooling effectiveness measurement technique is frequently and effectively employed.

Axial Flow Turbine Research Facility AFTRF at Pennsylvania State University: The Axial Flow Turbine Research Facility (AFTRF), at the Turbomachinery Aero-Heat Transfer Laboratory is shown in Figure 20. A recent description of the operational characteristics of this rotating rig designed mainly for aerodynamics and heat transfer investigations is available in Camci [2004]. The research facility is a large-scale, low speed, cold flow turbine stage depicting many characteristics of modern high-pressure turbine stages. The air flow through the facility is generated by a four stage axial fan located downstream of the turbine. The turbine rig has a removable casing segment for measurement convenience especially for casing and tip related aero-thermal studies. The relevant design performance data including reaction at blade hub and tip, Reynolds number at rotor exit and a few blade parameters. are also listed in Figure 20. Measured/design values of rotor

inlet flow conditions including radial, axial, tangential components and data acquisition details of the turbine rig are explained in detail by Camci [2004], Rao et al. [2004] and Lakshminarayana et al. [1996]. Instruments used for monitoring the performance parameters of AFTRF consist of total pressure probes, Kiel probes, pitot-static probes, thermocouples, and a precision in-line torquemeter. The turbine rotational speed that is adjustable to ± 1 rpm is kept constant around 1300 rpm by means of an eddy current brake.



Inlet Total Temperature: T_{01} (K)	289
Inlet Total Pressure: P_{01} (kPa)	101.36
Mass Flow Rate: Q (kg/sec)	11.05
Rotational Speed: N (rpm)	1300
Total Pressure Ratio: P_{01}/P_{03}	1.077
Total Temperature Ratio: T_{03}/T_{01}	0.981
Pressure Drop: $P_{01}-P_{03}$ (mm Hg)	56.04
Power: P (kW)	60.6
Rotor hub-tip ratio	0.7269
Blade Tip Radius; R_{tip} (m)	0.4582
Blade Height; h (m)	0.1229
Relative Mach Number	0.24
Number of Blades	29
Axial Tip Chord; (m)	0.085
Spacing; (m)	0.1028
Turning Angle; Tip / Hub	95.42° / 125.69°
Nominal Tip Clearance; (mm)	0.9
Reaction, Hub / Tip	0.197 / 0.519
Reynolds Number ($+10^5$) inlet / exit	(2.5~4.5) / (5~7)

Figure 20 Axial Flow Turbine Research Facility AFTRF at Pennsylvania State University, Turbomachinery Aero-Heat Transfer Laboratory, Camci [2004] and Lakshminarayana et al. [1996]

Figure 21 shows AFTRF rotor related aerodynamic details that can be highly influential in the determination of final wall heating patterns in axial turbines. A set of phase-locked LDA measurements clearly show the extent of tip vortical flow near the tip, wake pattern and rotor secondary flow structure. The middle figure presents the wall static pressure measurements performed near the blade tip in the rotating frame of reference. Although the measured suction side pressure coefficient agrees well with the RANS computations, the static pressure measurements near the pressure side corner where the tip leakage is significant show a discrepancy when

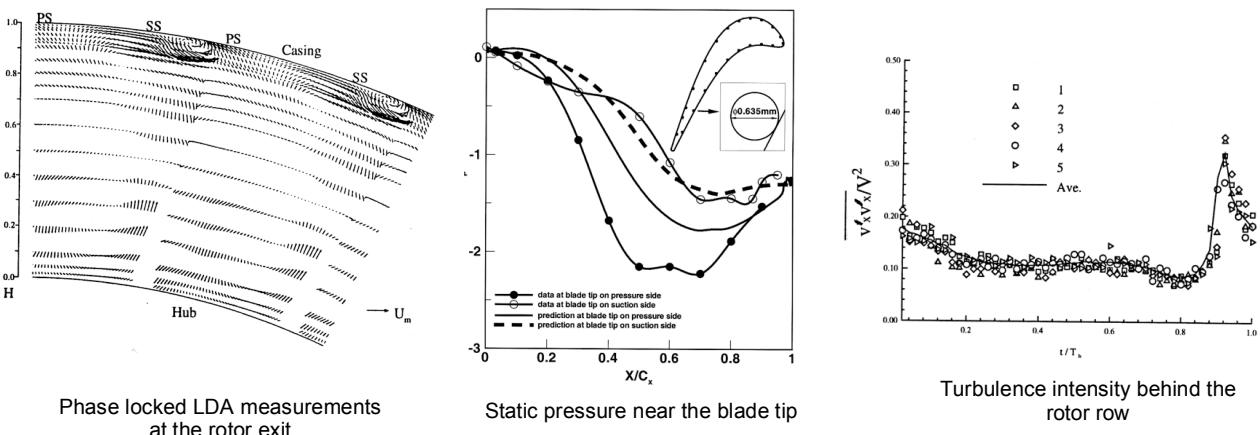


Figure 21 Rotor exit flow field, rotor tip static pressure and rotor exit axial turbulence intensity in AFTRF, Turbomachinery Aero-Heat Transfer Laboratory

compared to computational results. This may be attributed to the highly strained and separated relative flow pattern existing in this region where static holes may see leakage flow partially impinging on them. Finally, the rotor exit mid-span turbulence intensity in the circumferential direction shows that high levels of free stream turbulence ranging from 10 % to 35 percent in the wake are typical near the rotor exit plane.

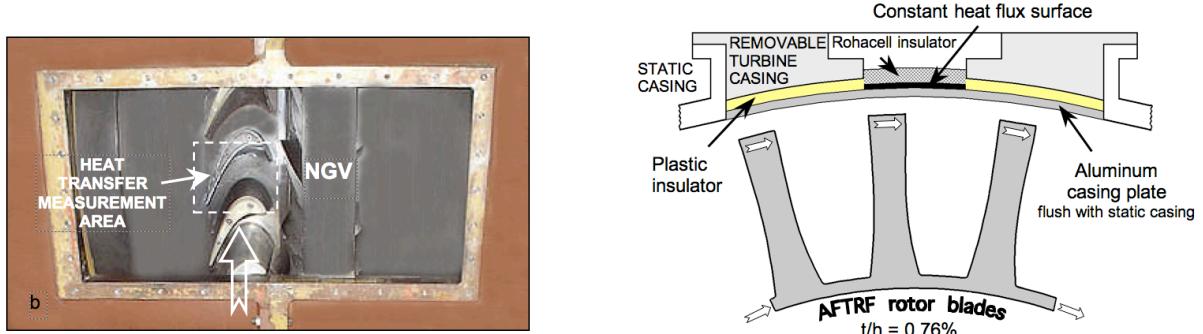


Figure 22 Heat transfer measurements using a constant heat transfer surface near the casing in AFTRF, Turbomachinery Aero-Heat Transfer Laboratory, Gumusel and Camci [2009]

The most recent aero-heat transfer research in the AFTRF is focused on the tip region and the casing region. Figure 22 presents the casing heat transfer measurement system which is mounted on the flow side of the “turbine access window” flush mounted in the research rig. This system was developed for measuring convective heat transfer coefficients on various “patterned” casing surfaces covering the rotating blade tip area. The flow side of the casing plate is flush-mounted with the rest of the inner part of the AFTRF casing. The axial distribution of the heat transfer coefficient is measured with an estimated uncertainty level between 5 % and 8 % of convective heat transfer coefficient h , Gumusel and Camci [2009].

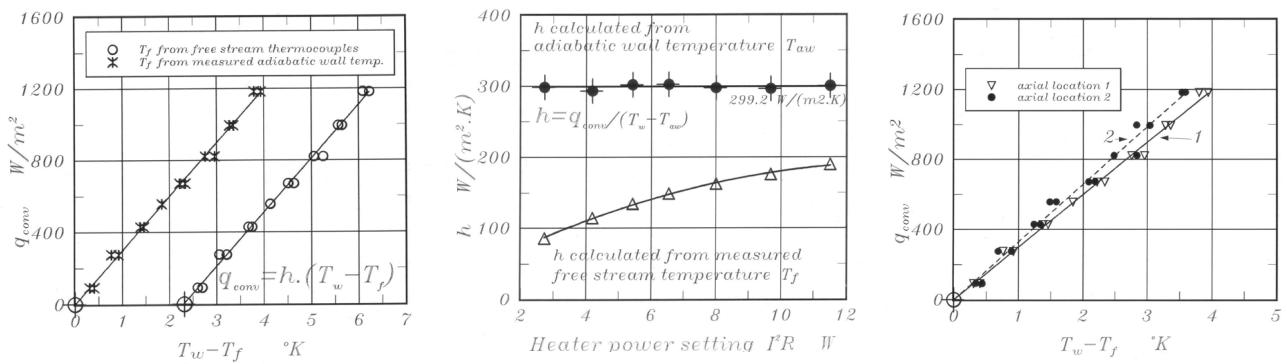


Figure 23 Determination of the adiabatic wall temperature and heat transfer coefficient on the casing surface of AFTRF, Gumusel and Camci [2009]

The heat transfer results shown in Figure 23 are obtained by a 3D heat loss analysis in order to reduce the measurement uncertainties. A 3D heat conduction analysis in the casing window and the casing plate is simultaneously conducted. The reference free stream temperature for this turbine convective heat transfer problem is the “adiabatic wall temperature” $T_f = T_{aw}$. Due to energy extraction from the working fluid in a turbine, the total temperature is continuously reduced in axial direction. Therefore, an accurate method of obtaining the true reference temperature (adiabatic wall temperature) is needed. Figure 23 shows the results of recent experiments when the power setting of the constant heat flux surface is continuously changed in order to obtain the local “adiabatic wall temperature”, at a selected axial measurement position. The dramatic influence of working with the

correct reference temperature in an axial turbine rotor environment is apparent in the middle part of Figure 23. The adiabatic wall temperature is a strong function of the axial position in the rotor frame of reference and it needs to be measured accurately in order to keep heat transfer coefficient uncertainties in the 5-8 % error band, Gumusel and Camci [2009].

CONCLUDING REMARKS

The present paper dealt with the experimental aero-heat transfer studies performed in rotating turbine research facilities. Since the full-scale operational conditions of modern gas turbines dictate high temperatures in excess of 3600 ° F and pressure ratios ranging from 20 to 50, experimental forced convection heat transfer research on the gas side of a rotating turbine is a technically challenging task. The current paper provided a brief review of turbine heat transfer research in various facilities that contained a rotor, including short-duration blow-down and large-scale/low-speed turbine systems. Since the final status of any forced convection heat transfer problem is closely related to the detailed structure of momentum transfer in highly unsteady, rotating, 3D and turbulent viscous flow environment, emphasis was also placed on supporting turbine aerodynamics research. After a discussion on the proper scaling of gas turbine convective heat transfer problem, the rotating facilities that produced heat transfer information were briefly described. Discussed heat transfer information was selected from the most recent publication in the open literature. Due to the excessive amount of information available in this field, only a few selected studies were discussed throughout the paper. The rotating turbine research rigs performing “only” stage aerodynamics/performance measurements, disk cavity flow experiments, rotating coolant passage experiments, turbine seal flow experiments were excluded from this heat transfer focused publication.

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