DESIGN, CONSTRUCTION, AND PRELIMINARY VALIDATION OF THE
TURBINE REACTING FLOW RIG

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ABSTRACT

This thesis presents the design, construction and partial operation of the Turbine Reacting Flow Rig (TuRFR), which is a high temperature turbine vane test facility at The Ohio State University’s Aeronautical and Astronautical Research Laboratory. It is capable of producing combustor temperatures up to 2200°F and is rated for normal operation at pressures of 30 psig. The facility matches real engine flow parameters such as Mach number, gas temperature, density ratio, pattern factor, and turbulence level. It is designed to test industry hardware and has a modular design to allow for various turbine vane shapes and sizes. It consists of a steel base (for flow conditioning and supporting of the burner), natural gas burner (for elevating the gas temperature), spool piece (for viewing burner during operation), cone (for accelerating the flow), equilibration tube (for allowing entrained seed particles to reach thermal and kinematic equilibrium), transition piece (for sealing with the equilibration tube and transitioning from a circular to rectangular cross section), view section (for optical access to the turbine vanes), and vane holder (for securely holding the turbine vane in place during experiments). It is capable of providing film cooling air at a density ratio of 2.8, with plans to upgrade the heating system for lower density ratios. To date, the facility has been tested at mass flowrates up
to 2.6 lbm/s, which is adequate mass flow to fill four 1st stage high pressure vane passages from a modern high bypass aero-engine at a representative inlet Mach number of 0.25 and gas temperature of 2200°F.
To my loving wife Amy and son Teagan
ACKNOWLEDGMENTS

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I thank Dr. Samimy for serving on my graduate committee and loaning his Chromalox heater, which is currently being used for elevating the temperature of the film cooling air. I appreciate the invaluable tips, guidance, and support from our friends at Brigham Young University (Weiguo Ai, Dr. Fletcher, Robert Laycock, and Spencer Harding). Many thanks go to Stuart Benton for his assistance in the many projects involved with the TuRFR. Ken Copley and Ken Fout from the shop were a huge help with machining, installation of the I-beams, and anything requiring a forklift. I would also like to thank all the staff at the Aeronautical and Astronautical Research Laboratory for their assistance. I also would like to note the support of Dan Kelley from Kelley Industrial Sales with his help in designing the sealing system for the TuRFR.

Finally and most importantly I would like to thank my wife Amy and son Teagan for their encouragement, support, and patience. They gave me the strength to keep going.
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## NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_s$</td>
<td>surface area</td>
</tr>
<tr>
<td>$Bi$</td>
<td>Biot number</td>
</tr>
<tr>
<td>$C_D$</td>
<td>coefficient of drag</td>
</tr>
<tr>
<td>$C_0$</td>
<td>orifice plate discharge coefficient</td>
</tr>
<tr>
<td>$D$</td>
<td>drag or pipe diameter</td>
</tr>
<tr>
<td>$DR$</td>
<td>density ratio</td>
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<tr>
<td>$E$</td>
<td>modulus of elasticity</td>
</tr>
<tr>
<td>$G$</td>
<td>specific gravity</td>
</tr>
<tr>
<td>$I$</td>
<td>second moment of inertia or momentum flux ratio</td>
</tr>
<tr>
<td>$L_c$</td>
<td>characteristic length</td>
</tr>
<tr>
<td>$M$</td>
<td>mass flux ratio</td>
</tr>
<tr>
<td>$\overline{Nu}$</td>
<td>diameter averaged Nusselt number</td>
</tr>
<tr>
<td>$P$</td>
<td>electrical power</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$R$</td>
<td>reaction load or electrical resistance</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number, based on diameter or chord</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$V$</td>
<td>velocity or voltage</td>
</tr>
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</table>
VR velocity ratio

c specific heat capacity

d diameter

\( \bar{h} \) diameter averaged convective heat transfer coefficient

k thermal conductivity

\( \dot{m} \) mass flowrate

p pressure

q specific weight

t time

y orifice plate flow function

\( \rho \) density

\( \mu \) kinematic viscosity

\( \eta \) efficiency

\( \theta \) nondimensional temperature

\( \delta \) deflection or boundary layer thickness

\( \phi \) equivalence ratio

\( \gamma \) ratio of specific heats

\( \varepsilon \) emissivity

\( \sigma \) Boltzmann’s constant

**Subscripts**

a air

ave spanwise average
<table>
<thead>
<tr>
<th>c</th>
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</tr>
</thead>
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<tr>
<td>cr</td>
<td>critical</td>
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<tr>
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<td>thermocouple bead</td>
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<tr>
<td>p</td>
<td>particle</td>
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<tr>
<td>s</td>
<td>surface or surroundings</td>
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<tr>
<td>stoich</td>
<td>stoichiometric</td>
</tr>
<tr>
<td>t</td>
<td>thermal</td>
</tr>
<tr>
<td>x</td>
<td>position on flat plate</td>
</tr>
<tr>
<td>1</td>
<td>location of compressor inlet</td>
</tr>
<tr>
<td>3</td>
<td>location of turbine inlet</td>
</tr>
<tr>
<td>∞</td>
<td>freestream</td>
</tr>
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</table>
CHAPTER 1

INTRODUCTION

1.1 Background

The idealized case of the simple cycle gas turbine engine is known as the Brayton cycle, illustrated in Figure 1. In the ideal Brayton cycle the working fluid undergoes isentropic compression (compressor), constant pressure heat addition (combustor), isentropic expansion (turbine) and constant pressure heat rejection. The thermal efficiency of this process is a function of the compressor inlet temperature \( T_1 \) and combustor exit temperature \( T_3 \), given in Equation 1. According to this idealized case an increase in the firing temperature of a gas turbine engine will increase its efficiency. However real engines exhibit component inefficiencies, pressure losses and heat losses, which reduce their overall thermal efficiency. For example, a GE F-class simple cycle gas turbine engine operated with a firing temperature of 2350°F in 1995 (today they operate in excess of 2400°F) [12, 28]. The ideal thermal efficiency for this cycle is 56% while the actual was 36%, which is a drop of 20% from its idealized case. This gap between the actual and idealized efficiency encourages engine manufacturers to continually design more efficient engines.
Some ways of increasing the overall efficiency of a real gas turbine engine is to increase the firing temperature, improve component efficiencies and make cycle modifications (land-based engines). Currently, F and H-class combined cycle gas turbine engines operate in the range of 55-60% efficiency [12, 28]. Over the years firing temperatures in gas turbine engines have been increasing at a steady rate, as shown in Figure 2. This figure illustrates firing temperatures approaching 2600°F, which is the
operating temperature of a GE H-class engine [12, 28]. The three enabling technologies for the increases in temperature are film cooling techniques, thermal barrier coatings (TBCs), and single crystal turbine materials.

![Historical trend of gas turbine firing temperatures](image)

**Figure 2 Historical trend of gas turbine firing temperatures [10]**

Film cooling air from the exit of the compressor is used to cool the turbine blades both internally and externally. The right image in Figure 3 illustrates the complexity of internal passages of a turbine blade, which may include fins, serpentine passages, and turbulence generators for increased internal cooling. The turbine blade surface is cooled by cooling air ejected at an angle from its surface, shown in Figure 3 on the left. This causes two effects; convective cooling as the air passes through the film cooling holes and decreasing the fluid temperature next to the blade surface by covering it with a thin
layer of cooler air. The coolant air along the surface of the metal mixes with the warmer freestream air as it travels downstream, shown in Figure 4, which necessitates film cooling holes over the majority of the turbine blade. The effectiveness of film cooling depends on several parameters such as density ratio, velocity ratio, mass flux ratio, momentum flux ratio, freestream pressure gradients, blade curvature, freestream turbulence, and more [4]. Film cooling allows for turbine blades to operate in an environment where the freestream temperature is higher than the blade’s material melting temperature.

Figure 3 Schematic of a typical film cooling configuration [4]

Figure 4 Thermal profiles (defined as $\theta = \frac{T_{\infty} - T}{T_{\infty} - T_c}$) of coolant jet exiting at an angle of 35° to the surface [4]
In addition to film cooling, TBCs can enable turbine blades to operate with engine freestream temperatures exceeding the temperature limitations of the turbine metal. TBCs typically consist of two layers, a ceramic top coat and a bond coat between the ceramic and metal. Figure 5 is an example of the structure of a TBC system in conjunction with film cooling air. A representative temperature gradient is drawn depicting the effects of each layer of the turbine and its cooling system. Typically this temperature gradient in the TBC is within the range of 126-270°F. In combination with film cooling, TBCs can allow combustion gas temperatures in engines up to 450°F above the melting temperature of the turbine metal [12]. At lower operating temperatures, TBCs are still advantageous since they decrease the turbine metal temperature, which increases the component’s service life. The top coat of TBCs is designed for low thermal conductivity and to be tolerant of strain due to the cyclic temperatures they operate under. The bond coat is rich in aluminum to create a thermally grown oxide scale as it reacts with oxygen [12]. This protects the blade metal from oxidation. TBCs are one source of protection for turbine blades from the harsh thermal environments of gas turbine engines.
Single crystal materials are typically used in the early stages of the turbine where temperatures are highest for the turbine. Single crystals are favored over multi-grain structures (equiaxed or columnar grain structures) due to their creep resistance at high temperatures. At about half the absolute melting temperature the grain boundaries become weaker than the grain bodies [29]. The single crystal structure does not have grain boundaries, so it does not have the weakening effects equiaxed and columnar grain structures exhibit at high temperatures due to their polycrystalline structure. The single crystals for turbine blades are mainly composed of nickel, which provides exceptional high temperature strength, toughness, and resistance to oxidation. Single crystal turbine blades can operate with average metal temperatures of 1922°F, with temperature spikes in localized zones as high as 2192°F [26]. It is important to note 2192°F is roughly 90%
of the melting temperature of nickel-based turbine blades. Advances in single crystal materials have allowed turbine blades to operate at extreme temperatures while maintaining a high level of strength.

The conditions a gas turbine engine operates under are quite severe. At normal operational temperatures (2400-2600°F) an increase of only 45°F in turbine metal temperature can reduce its part life by a factor of two [4]. The combustor in a gas turbine engine causes high levels of turbulence. Turbulence is desired in the combustor to enhance fuel-air mixing for higher intensity combustion [11]; however it is undesirable at the turbine inlet. High turbulence levels increase heat transfer to the turbine vane surface by increasing the local heat transfer coefficient, which potentially results in higher metal temperatures. To protect the combustor liner, cooler dilution air directly from the compressor exit is injected along its surface. This causes spatial temperature variations (called pattern factors) in the gas exiting the combustor. Barringer reports on four combustor exit temperature profiles, as shown in Figure 6 [2]. The temperature is non-dimensionalized according to Equation 2. This temperature variation can lead to spatially nonuniform heat transfer to the vane surface, which can result in local melting and cracking of the vane material. The combustor also creates nonuniform pressure profiles at the turbine inlet, which can cause spanwise variations in the aerodynamic loading of the turbine vanes [2].
Particulate ingestion is another issue for engines. Filtration systems for land-based gas turbine engines can filter up to 99% of the contaminants entering the engine, but poorly maintained filtration systems can allow significantly higher amounts of airborne contaminants to enter the turbine [19]. Small concentrations of contaminants are significant for high mass flow systems. For example, if 1 ppmw impurities pass through the filter for a large combined cycle gas turbine engine operating at approximately 1500 lbm/s would result in 5.4 lbm/hr of contaminant ingestion [34]. Over an operating year of
8000 hours the total ingestion would be about 21 tons. This is sufficient to cause significant turbine wear. Particulate ingestion at the turbine inlet can also be caused by combustion of dirty fuels. Some of the fuels used in modern gas turbine operation are derived from coal and biomass. Clean up of these fuels can reduce the level of their contaminants to less than 1 ppmw, but this level is 2 to 10 times higher than manufacturer recommended levels for natural gas engines [34]. At elevated temperatures fuel ash contaminants inside the engine can become molten and deposit on the turbine surface. Wenglarz et al [35] showed the level of deposition increases significantly as the temperature increases past the melting temperature of the particulate. Kim et al [21] exhibited accelerated deposition rates as contaminants adhere to the turbine surface. If the particles do not deposit they have the potential of causing erosion. As the TBCs erode, the turbine surface is exposed to higher temperatures. Also, the turbine hardware is more susceptible to corrosion without its protective bond coat of the TBC.

The objective of the facility described in this thesis is to replicate these severe operating conditions to better understand the nature of the effects of these conditions on turbine hardware.

1.2 Literature Review

Several facilities have been constructed and operated that match various turbine inlet conditions. Three typical types of turbine related test facilities include rotating transient test facilities, fixed vane test facilities, and deposition/erosion facilities.

1.2.1 Rotating Transient Test Facility

A rotating transient test facility is a short duration blowdown facility with typical runtimes of a few milliseconds to a few seconds. This type of facility can match engine
Reynolds number, Mach number, pressure ratio, gas to metal temperature ratio, corrected speed, and corrected mass flow. A typical layout consists of a large stagnation tank, combustor simulator, turbine test section, and a discharge tank, shown in Figure 7. For elevated temperature studies a facility can utilize a system of heaters or a shock tube. The Ohio State University’s Gas Turbine Laboratory Test Facility (TTF) is an example of such a facility that can be driven with high enthalpy fluid from a shock tube, shown in Figure 8. In this facility, the turbine test section is a full scale, fully annular turbine with one or more stages. A combustor simulator is used to provide typical spatial distributions of pressure, temperature, and turbulence intensity to the inlet of the turbine section. Pressure and temperature profiles of the inlet and exit are measured using pressure transducer and thermocouple rakes. Surface pressures on the turbine vane can be measured with Kulite pressure transducers. Surface temperature gauges provide heat flux measurements by assuming a one-dimensional conduction analysis. The data acquisition system typically uses sampling rates equal to or greater than 100 kHz to capture the transient effects. Rotating transient test facilities exist at the TTF [30], Turbine Research Laboratory at the Air Force Research Laboratory [1], Marshall Space Flight Center [6], and many other locations.
Rotating transient test facilities serve an important part of the experimental community for testing full scale real hardware with real flow profiles at the inlet of the turbine section. However, these tests are done at low temperatures relative to typical turbine inlet temperatures and are not done at steady state conditions.
1.2.2 Fixed Turbine Cascade Facility

The fixed turbine cascade rig is a steady state facility that utilizes scaled up turbine vanes, such as those shown in Figure 9. These are used to evaluate aerodynamics and heat transfer characteristics of turbine vanes [33]. They can match Reynolds number, Mach number, pressure ratio, and mass flow. They also can vary the turbulence intensity to the inlet of the turbine section, similar to the rotating transient test facility. They can be run at elevated pressures. The uniqueness of these facilities is their capability to test several vane profiles with high degrees of turning at varying incidence angles. Fixed turbine cascade rigs exist at the NASA Lewis Research Center [33], Ohio State University’s Aeronautical and Astronautical Research Laboratory, Heat Transfer Laboratory of the University of Minnesota [5], and many other locations. Limitations of these facilities are they use linear cascades, a single vane row, operate at low temperatures, and do not use real engine hardware.

Figure 9 Transonic Turbine Blade Cascade at NASA’s Lewis Research Center [33]
1.2.3 Accelerated Deposition/Erosion Facilities

The Turbine Accelerated Deposition Facility (TADF) at Brigham Young University studies the patterns and effects of deposition on high pressure turbine blade materials. It replicates the total mass throughput of particulate ingestion of a typical gas turbine engine over a year of normal operation (~10,000 hours) by increasing the concentration of airborne contaminants in a several hour test. The facility matches turbine inlet Mach number, gas temperature, blade temperature, particulate size and chemistry, turbine blade material, impingement angle, blowing ratio, momentum flux ratio, and density ratio of a typical high pressure turbine blade. A detailed description and validation study of the facility is given by Jensen et al [18].

The design of the TADF is illustrated in Figure 10. Air enters the base, is dispersed by a layer of marbles, is straightened by an aluminum honeycomb, is heated within a natural gas burner, accelerates through a cone to the desired Mach number, travels down an equilibration tube, and impinges on the test specimen (coupon). The particulate is added to the airflow immediately upstream of the natural gas burner. The equilibration tube allows the particles to reach thermal and kinematic equilibrium with the hot gas flow. The natural gas (methane) feeds into the base of the TADF through four lines; each branching off into two additional lines. Each line is then bent so that the fuel exits the line parallel and in the same direction as the freestream air; making a flameholder. There are a total of eight flameholders, as shown in Figure 11. The flameholders are 3/8 inch ID tubing with four notches at the fuel exit. The fuel is premixed with air upstream of the base to decrease sooting from the combustion process. The mixture inside the premixer is maintained at a fuel rich level to prevent flashbacks.
The base of the TADF is made of 1/4 inch thick steel with a 12 inch inner diameter. Downstream of the burner the material is Inconel 600 and 601. The equilibration tube has an inner diameter of 2.4 cm. The facility can operate at exit Mach numbers of 0.3 at 1200°C.

The test specimens used in the experiments are known as coupons, which are samples of turbine blade materials with thermal barrier coatings. The diameter of a typical coupon is on the order of 2.5 cm. Some coupons include holes to simulate film cooling holes. The coupons are held in a fixture that can vary the hot gas impingement angle between 0° and 90° to replicate the various approach angles of particles along the surface of a turbine blade.

The particulate injection system is created by using several loaded syringes installed in parallel along a manifold tube. The syringes are loaded with particles, which are injected into the manifold tube. A brush rotates inside the manifold tube to prevent clumping at the exit of the syringes and on the wall of the tube. Downstream of the manifold tube the particles enter a glass bulb where they mix with air. The mixture then flows through small diameter tubing into the TADF, as shown in Figure 10. The syringes and glass bulb are continually vibrated to reduce caking of the particulate.
Figure 10 Schematic and photo of the TADF [18]
University of Cincinnati operates an erosion facility similar to the TADF capable of temperatures up to 1093°C, shown in Figure 12 [14]. Deposition and erosion facilities are important tools in analyzing the structures and patterns of deposition, erosion and corrosion development. However this type of facility does not operate with real engine hardware or real engine flow patterns.
Figure 12 Schematic of the University of Cincinnati’s erosion wind tunnel facility [14]
1.3 Facility Objectives

Based on the background given of the severe conditions high pressure turbine vanes operate under, and the facilities currently in operation for turbine related testing, there is a need for a new facility. This facility would need to replicate engine flow conditions at the inlet to the nozzle guide vane: temperatures at or near 2200°F, Mach numbers of 0.1-0.25, and film cooling density ratios of approximately 2. This facility should be able to create combustor exit temperature profiles and turbulence levels. The facility would need to operate for several hours for steady state tests. It would need to utilize actual turbine hardware. It needs to be modular to allow testing of various turbine vane shapes and sizes. Finally, the facility would need to be capable of performing deposition studies by inserting particulate into the gas stream prior to heating. Based on these needs, The Ohio State University has developed the Turbine Reacting Flow Rig (TuRFR), which is discussed in the following chapters.
CHAPTER 2

FACILITY DESIGN

The Turbine Reacting Flow Rig (TuRFR) is a high temperature elevated pressure wind tunnel designed for testing high pressure turbine vanes. The parameters the facility can match include gas temperature, Mach number, film cooling density and velocity ratio (and consequently mass and momentum flux ratio), blade temperature, freestream turbulence, pattern factor, and seed particle size, impingement angle, chemistry, and loading. The parameters the TuRFR will not match are absolute pressure and Reynolds number. Table 1 includes several operating parameters of the TuRFR compared to industry engines. The JT9D and PW4000 are civilian aero engines. The F100 is a military engine. The GE F-Class is a land-based gas turbine engine. Depending on the study the facility will match the pertinent engine operating conditions.

For flow heat transfer performance the TuRFR will match gas temperature and Mach number. The gas temperature is important since the magnitude of heat transfer to the turbine hardware is a function of the gas temperature. The Mach number is the ratio of the local velocity to the local speed of sound and is an indication of compressibility effects. By matching the Mach number and turbine inlet temperature provides the correct temperature distribution over the blade.
<table>
<thead>
<tr>
<th>Turbine Inlet Conditions</th>
<th>TuRFR</th>
<th>JT9D</th>
<th>F100-PW-229</th>
<th>PW4000</th>
<th>GE F-Class</th>
</tr>
</thead>
<tbody>
<tr>
<td>TIT (K)</td>
<td>1478</td>
<td>1422</td>
<td>1755</td>
<td>1478</td>
<td>1600</td>
</tr>
<tr>
<td>PR</td>
<td>2.0</td>
<td>24.5</td>
<td>23.0</td>
<td>27.5</td>
<td>15.5</td>
</tr>
<tr>
<td>R (/J/kg/K)</td>
<td>290.4</td>
<td>290.4</td>
<td>290.4</td>
<td>290.4</td>
<td>290.4</td>
</tr>
<tr>
<td>M</td>
<td>0.25</td>
<td>0.25</td>
<td>0.25</td>
<td>0.25</td>
<td>0.25</td>
</tr>
<tr>
<td>γ</td>
<td>1.3</td>
<td>1.3</td>
<td>1.3</td>
<td>1.3</td>
<td>1.3</td>
</tr>
<tr>
<td>V (m/s)</td>
<td>187</td>
<td>183</td>
<td>203</td>
<td>187</td>
<td>194</td>
</tr>
<tr>
<td>Pamb @ Sea Level (40 kft)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>P (kPa)</td>
<td>202</td>
<td>2475</td>
<td>490</td>
<td>2778</td>
<td>550</td>
</tr>
<tr>
<td>ρ (kg/m³)</td>
<td>0.471</td>
<td>5.992</td>
<td>1.187</td>
<td>4.558</td>
<td>0.903</td>
</tr>
<tr>
<td>μ (N-s/m²)</td>
<td>5.4E-05</td>
<td>5.3E-05</td>
<td>5.3E-05</td>
<td>5.9E-05</td>
<td>5.9E-05</td>
</tr>
<tr>
<td>Rec/L @ Turbine Inlet (m⁻¹)</td>
<td>1.6E+06</td>
<td>2.1E+07</td>
<td>4.1E+06</td>
<td>1.6E+07</td>
<td>3.1E+06</td>
</tr>
<tr>
<td>Re (Normalized by TuRFR)</td>
<td></td>
<td>12.77</td>
<td>2.53</td>
<td>9.57</td>
<td>1.90</td>
</tr>
</tbody>
</table>

Table 1 Comparison of TuRFR operating parameters to real engines [10, 24]

For deposition studies the TuRFR will match gas temperature, Mach number, blade temperature, film cooling density and velocity ratio, and seed particle size, impingement angle, chemistry, and loading. The gas temperature will affect the seed particle’s absolute temperature and phase. Wenglarz and Wright [34] reported on the deposition due to use of coal derived liquid fuels for power generation. A significant increase in deposition rates were experienced for a gas temperature of 2300°F compared to deposition rates at a temperature of 2000°F. They concluded this was due to a higher percentage of molten particles present in the gas stream at the higher temperature. The velocity of the seed particles is dependent on the gas temperature and Mach number. The momentum of the particle is dependent on its velocity, size, and density. Wenglarz and Fox [35] studied the effects of burning coal-water fuels with respect to deposition. They observed larger particles, and consequently higher momentum particles, were noticed to
have a higher number of impacts on a turbine’s surface compared to smaller diameter particles. The particle’s impingement angle is important since it affects the normal force a particle impacts the turbine blade. The seed particle’s chemistry affects whether it is more susceptible to deposition effects [21]. Deposition in real engine hardware occurs over long intervals, but Jensen [18, 19] validated the theory of matching the particulate total loading by increasing its concentration in the gas stream and reducing the runtime. For example, a gas turbine operating for 10,000 hours with a particulate concentration of 0.02 ppmw has a total loading of 200 ppmw-hrs. This can be matched by increasing the concentration to 50 ppmw at 4 hours for a total loading of 200 ppmw-hrs. Tests done by Kim et al [21] concluded increased blade temperatures resulted in higher deposition rates.

For film cooling effectiveness studies the TuRFR will match gas temperature, Mach number, density and velocity ratio (and consequently mass and momentum flux ratio), and freestream turbulence. The arguments for matching the gas temperature and Mach number are the same as the flow heat transfer performance discussion. The various film cooling ratios (density, velocity, mass flux, and momentum flux) are critical parameters affecting film cooling performance. The convective heat flux to a surface is a function of the temperature gradient and the convective heat transfer coefficient. The film cooling ratios characterize the flow along the blades surface. Bogard et al [4] reports on the effects of varying these ratios. For example, a high momentum flux ratio can result in a detached cooling stream, which reduces the overall film cooling effectiveness. The freestream turbulence affects the mixing of the film cooling stream with the freestream, with higher turbulence levels causing more rapid decay of the film cooling effectiveness.
[20]. Other factors that affect film cooling include surface roughness, curvature, hole geometry, and angle of injection [4, 9], but these are dependent on the turbine hardware.

The parameters the facility will not be matching are pressure and Reynolds number, given in Table 1 relative to real engines. Pressure has not been mentioned in the reviewed literature to be a factor in the above mentioned studies. Many facilities perform tests at atmospheric pressure [18, 20, 32]. The Reynolds number effects will be evaluated on a case-by-case basis. Boundary layer effects due to the Reynolds number based on blade chord are diminished by the complex structure of cooling holes on the blade’s surface. Film cooling causes three dimensional flows and earlier transition of the boundary layer [8]. So how not matching the Reynolds number will affect the flow will depend on the film cooling, surface roughness, blade curvature, type of study, and others.

2.1 Rig Design

A portion of the rig is shown in Figure 13 and a full schematic shown in Figure 14. It is broken up into two main parts; the lower assembly (steel base, spool piece, cone, and equilibration tube) and the upper assembly (transition piece, view section, and vane holder piece), the materials of which are listed in Table 2. The main portion of air enters the base through four ports, is dispersed through a muffler and pebble bed, is straightened through an aluminum honeycomb, is heated within a natural gas burner to elevate its temperature to 2200°F, and accelerates inside the cone to Mach numbers of approximately 0.25 as the cross-sectional area contracts. Then the air passes through a dilution jet section, expands to fill the area of a turbine vane set, and finally exhausts
through the turbine vanes to ambient pressure. Typically, the pressure ratio across one high pressure turbine vane set is about 2, so the entire rig was designed to withstand a minimum pressure of 30 psig at 2200°F with a design safety factor of four.
Figure 13 TuRFR lower assembly (steel base, spool piece, cone, and equilibration tube)
Figure 14 Schematic of entire TuRFR
<table>
<thead>
<tr>
<th>Part / Component</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel Base</td>
<td>Mild Steel</td>
</tr>
<tr>
<td>Spool Piece</td>
<td>Inconel 617</td>
</tr>
<tr>
<td>Spool Piece – Viewports</td>
<td>Inconel 600</td>
</tr>
<tr>
<td>Cone</td>
<td>Inconel 617</td>
</tr>
<tr>
<td>Steel Half Flanges</td>
<td>Mild Steel</td>
</tr>
<tr>
<td>Equilibration Tube</td>
<td>Inconel 617</td>
</tr>
<tr>
<td>Upper Assembly</td>
<td>602CA</td>
</tr>
<tr>
<td>Upper Assembly – Film Cooling and Dilution Jet Pipes</td>
<td>RA601</td>
</tr>
</tbody>
</table>

**Table 2 Materials for each component in the TuRFR**

**2.1.1 Steel Base**

The first section of the rig is the steel base. Air enters the bottom of the steel base through four 1-1/2 inch NPTF half couplings that are circumferentially spaced 90° apart. The air then enters the muffler system, shown in Figure 15. The muffler system breaks down the four strong jets into tens of smaller jets and disperses them uniformly throughout the base. The muffler is designed from four equal length 2 inch PVC schedule 40 pipes and one standard 2 inch cross, all highly perforated. Each pipe extends radially inward from the wall of the steel base and connects with the other three pipes in the center by means of the cross. This design is favorable since the reaction forces from air entering one pipe are balanced by the reaction forces of the opposing side. The air exits through the perforated faces of the PVC pipe into an eight inch tall open section to allow
diffusion of the jets throughout the diameter of the steel base. The muffler system creates a coarsely dispersed flow that needs to be further diffused throughout the base.

![Figure 15 Top view schematic of muffler system inside steel base](image)

The next flow conditioning unit is a pebble bed system. This was constructed by laying down two steel mesh screens directly on top of each other that hold a thick layer of small rocks (approximately 0.5 to 2.0 inches in diameter). The steel meshes are supported by a 1/4 inch thick steel lip welded to the inside of the steel base eight inches from its bottom flange. The mesh screens serve two purposes; support the rocks and further
disperse the main airstream. This dispersion effect is very important since the air immediately downstream of the muffler system has a non-uniform flow distribution. High momentum flow zones are present which are strong enough to blow the small rocks out of the way causing a void in the pebble bed. The screens break up these high flow zones. Larger rocks would be heavy enough to oppose those high flow zones, but they would not disperse the flow as uniformly. So a thin layer, approximately three inches tall, of small rocks was spread across the top of the screen meshes. The air exiting the pebble bed is turbulent and three-dimensional, so an aluminum honeycomb 4 inches in height is used immediately downstream to straighten the flow and remove any swirl. A uniform flow is necessary for the burner section so flame stability can be maintained to provide uniform heating.

The flow conditioning system and burner reside inside the steel base, which is not a high temperature nickel-based alloy like the other sections. The amount of heat conducted from the downstream nickel-based alloy sections and the radiation from the burner do not heat the steel base enough to be of concern. The steel base remains relatively cold since the main airflow upstream of the burner is cold, less than 40°F. The bottom of the steel base was constructed with a 1 inch thick blind flange. The upper flange is made of 3/4 inch thick carbon steel. The walls of the steel base are 3/8 inch thick carbon steel. The thickness was chosen based on the pressure the steel base will operate under. The steel base is designed to operate at a maximum pressure of 70 psig, which includes a safety factor of four based on a hoop stress calculation. This is higher than the typical pressure rating of 30 psig because the flow conditioning system incurs a
large pressure drop. This causes the pressure upstream of the flow conditioning system to be higher.

The steel base was designed with a diameter of 39 inches so the airspeed would be relatively low immediately upstream of the burner. Low speed flows for combustion are desirable since it is easier to sustain a flame in a slow moving fluid. Gas turbine engine combustors utilize similar bluff bodies to cause a local low speed zone where the combustion occurs. Figure 16 illustrates the blockage of flow typical in an annular gas turbine combustor near the flameholder. Only a small percentage of the flow is allowed to enter the combustor near the flameholder. The flame is then diluted with the excess air. The expected average flow speed inside the steel base immediately upstream of the burner for an air mass flow of 2.6 lbm/s is 1.7 ft/s (32°F at 29.4 psia) and immediately upstream of the cone is 9.3 ft/s (2200°F at 29.4 psia).

Figure 16 Schematic of a typical mass flow distribution in a gas turbine combustor [16]
2.1.2 Natural Gas Burner Design

The burner’s design requirements were: sustain a stable flame, create good mixing of the fuel and air for a lean burn, and supply an intense enough flame for the required amount of heat release to provide a net outflow temperature of 2200°F. The design of the burner section occurred over several iterations. The first design was similar to a typical outdoor gas grill, shown in Figure 17. The fuel was supplied through four fuel lines. Each of these fuel lines broke into two additional lines inside the steel base, which were curved to uniformly cover the diameter of the steel base. The fuel line was made of 1/2 inch OD copper tubing. The tubing was initially perforated by a small drill bit (~1/16 inch) and crimped at the ends. The result of this design was flames that were too tall, unstable and non-uniform, as seen in Figure 17. As premixing was attempted many of the flames blew out while others continued tall and yellow. Yellow, orange or red flames are an indication of a rich mixture and produce undesirable flame sooting. Similar results were observed with larger diameter holes. This design suggested that it is necessary to have larger diameter flameholders for increased flame stability.
Gleaning the knowledge of the previous design, the next iteration used eight fuel supply lines, each breaking off into four flameholders, shown in Figure 18. The fuel lines entered near the top of the steel base through eight 3/8 inch NPTF holes spaced approximately 45° apart. The flameholders were made of 1/2 inch OD copper tubing bent upward. The tips were notched and flared to cause turbulence in the surrounding airflow to enhance mixing and increase the intensity of the flame. Without this turbulence generation system the mixing would rely on the shear layer effects of the two fluids, which is not a very effective mixing technique. Intense mixing of fuel and air is favorable for more complete combustion. The flames were denser with this design but not very stable. It seemed a larger diameter flame front would solve the issue of stability.
The third iteration added a 1/2 to 1-1/2 inch copper transition piece to the top of the 1/2 inch OD copper tubing, seen in Figure 19. 1/4 inch holes were drilled in the expansion section of the transition piece. As the fuel exited the 1/2 inch OD copper tubing into the transition piece its high velocity jet entrained the surrounding air through the holes, seen in Figure 19. This reduced the amount of premixing required further upstream in the fuel line, which reduced the likelihood of a flashback. The result of this design was a larger flame front area and denser flame with good stability, but the flames were still tall, approximately 18 inches. At high air mass flows the temperature of the air exiting the top of the equilibration tube was only about half or three-quarters of 2200°F, with flames occasionally extending beyond the top of the lower assembly. Also, the premixing had no visible effect on the flame structure. It was concluded a more intense, shorter flame, and, if possible, larger flame front area was necessary.
Figure 19 The flameholder for the third burner iteration (schematic with flow structure on left and actual flameholder on right)

The fourth iteration addressed the previously mentioned concerns by capping the previous flameholders and perforating the sides with twenty 5/16 inch diameter holes, seen in Figure 20. A washer was placed just upstream of the holes to increase local turbulence and help create a recirculation zone for the air to mix with the fuel. This allowed the fuel to eject further radially outward before being turned parallel to the main airflow. With this design the flames were more stable and had a larger flame front area than previous designs. With premixing the flame height was reduced from approximately 18 inches to approximately 6 inches and changed the color of the flame from yellow and red to blue. Significantly more heat release was witnessed both near and far downstream of the flameholders than the previous designs. The temperature exiting the top of the equilibration tube had a maximum of 2200°F with 0.089 lbm/s of fuel and 2.6 lbm/s of air flowing through the base section. A theoretical energy balance estimated the required mass flow of fuel to achieve a temperature of 2200°F would be 0.080 lbm/s (see Section
2.2.3). This flameholder design is the preferred design since the goal temperature of 2200°F was achieved and correlated very well with an energy balance.

![Figure 20 Final flameholder design (left) and burner arrangement (right)](image)

**2.1.2 Spool Piece, Cone and Equilibration Tube**

The components downstream of the steel base are made of nickel-based alloys. These are preferred in high temperature applications over standard steel materials due to their strength at high temperatures and excellent oxidation resistance [22]. The spool piece, cone, and equilibration tube are made of Inconel 617, which has a yield strength of 10 ksi at 2000°F [27].

The spool piece serves two purposes; allow visual access to the burner section of the TuRFR and provide a means of removing the cone. Similarly to the rest of the TuRFR
the spool piece was designed to withstand a pressure of 30 psig (which includes a safety factor of 4) at 2000°F. Autodesk Inventor was used to perform the stress analysis using an internal pressure of 125 psig, given in Figure 21. The plot shows some stress concentrations of 13.5 ksi. But the analysis was done without the viewports and steel flanges, which will reduce the overall stresses.

![Figure 21 Spool piece stress plot with an internal pressure of 125 psig](image)

The spool piece was composed entirely of 3/16 inch Inconel 617, except for the viewports that were made of Inconel 601. The flanges were not sufficiently thick to handle the temperature rise and pressure, so two sets of removable 3/4 inch thick steel flanges were used to sandwich the Inconel 617 flanges. The inside of the TuRFR can be seen through two 5/8 inch diameter viewports spaced 90° apart. The windows were made
of quartz to withstand the elevated temperature. The spool piece facilitates the removal of
the cone from the steel base by replacing four of its flange bolts with eyebolts. Straps,
with a 1000 lbm capacity, are connected to each of these eyebolts and the assembly is
then lifted by a 500 lbm capacity Skyhook hoist. The combined weight of the spool piece,
cone and equilibration tube is approximately 250 lbs.

The cone’s purpose is to accelerate the hot gas flow to Mach numbers in the range
of 0.2 to 0.3 (\(\bar{u} = 150 \text{ to } 225 \text{ m/s}\)). The cone has an inlet to exit area ratio of 68:1. The
cone was built out of 3/16 inch thick Inconel 617, including the flange. Similarly to the
spool piece, 3/4 inch steel flanges were used to increase the strength and rigidity of the
Inconel 617 flange. The exit cross-sectional area closely matches the inlet cross-sectional
area of two GE CFM56 vane cassettes (approximately 19 square inches). At an air mass
flowrate of 2.6 lbm/s the Mach number at the exit of the cone would be Mach 0.28
(2200°F at 29.4 psia). Initially the cone was not manufactured with ribs, which were later
added to increase the pressure rating of the cone to 30 psig (which includes a safety factor
of 4) at 2000°F. Autodesk Inventor was used to perform the stress analysis using an
internal pressure of 125 psig, given in Figure 22. The plot shows some stress
concentrations of 18.7 ksi, but these stresses are located at the bottom face of the cone.
This is where the flange is located, so those stress concentrations are not real. Again, with
the addition of the steel flange the cone maximum stresses will be about 10 ksi.
The equilibration tube is welded to the top of the cone. The purpose of this section is to provide sufficient length at the required Mach number for the particulate used in deposition studies to come to thermal and kinematic equilibrium. It is 31 inches long and made of 1/4 inch thick Inconel 617. This component is thicker than the rest to increase its part lifespan since it is exposed to higher temperatures than the upstream sections. The length was designed such that a typical particle with a diameter of 40 microns would reach thermal and kinematic equilibrium before exiting the pipe.

A lumped capacitance model was used to determine the time for a particle to reach thermal equilibrium with the hot gas flow, which assumes spatially uniform temperature in the solid [17]. This model is valid when conduction dominates over
convection. A numerical check of model’s validity is done by checking the Biot number, given by Equation 3. The characteristic length $L_c$ in this case is the particle’s volume divided by its surface area. The error incurred from using the lumped capacitance model for a solid with a Biot number less than 0.1 will be small. The time for a solid to reach a given temperature $T$ is given by Equation 4, where the average convective heat transfer coefficient $\bar{h}$ is found for a sphere by use of Equations 5 and 6 [17]. This is valid for Prandtl numbers in the range of $0.71 < Pr < 380$ and Reynolds numbers in the range of $3.5 < Re_{dp} < 7.6 \times 10^4$. The characteristic length for the Reynolds number is the particle’s diameter and the velocity is the difference between the freestream and particle velocity. Using silicone dioxide as an example with a diameter of 20 microns the particle’s temperature will reach 95% of the hot gas temperature in 1.3 ms, assuming the particle is at rest the entire time. It takes about 50% longer for the particle to reach kinematic equilibrium.

$$Bi = \frac{\bar{h}L_c}{k}$$

Equation 3

$$t = \frac{\rho V c}{\bar{h}A_s} \ln \left( \frac{T_i - T_\infty}{T - T_\infty} \right)$$

Equation 4

$$\bar{h} = Nu_{dp} \frac{k_{air}}{d_p}$$

Equation 5

$$Nu_{dp} = 2 + \left( 0.4Re_{dp}^{1/2} + 0.06Re_{dp}^{2/3} \right) Pr^{0.4} \left( \frac{\mu}{\mu_s} \right)^{1/4}$$

Equation 6
The time for the particle to reach the fluid velocity can be calculated based on the drag coefficient $C_D$, which is given in Equation 7. The drag coefficient can be found using a piecewise numerical scheme based on Reynolds numbers, given in Table 3 [7].

To calculate the time required for the particle to come up to speed with the freestream velocity Equation 7 must be integrated for $V_p$. Due to the complexity of analytically integrating Equation 7, a MATLAB script was used to integrate numerically, given in Appendix E. Using this theory it takes a particle approximately 2.2 ms to reach 95% of the fluid velocity. The range on the particle’s Reynolds number is 33.5 ($t = 0$ s) to 1.7 ($t = 2.2$ ms). This would require an equilibration tube length of about 12 inches.

\[
C_D = \frac{8D}{\pi \rho (V_{\infty} - V_p)^2 d_p^2} = \frac{4d_p \frac{dV_p}{dt}}{3(V_{\infty} - V_p)^2}
\]  

Equation 7
For $Re \leq 0.01$, 
\[ C_D = \frac{9}{2} + \frac{24}{Re} \]

$0.01 < Re \leq 20$, 
\[ C_D = \frac{24}{Re} \left[ 1 + 0.1315Re^{(0.82 - 0.05w)} \right] \]

$20 < Re \leq 260$, 
\[ C_D = \frac{24}{Re} \left[ 1 + 0.1935Re^{0.6305} \right] \]

$260 < Re \leq 1.5 \times 10^3$, 
\[ \log_{10} C_D = 1.6435 - 1.1242w + 0.1558w^2 \]

$1.5 \times 10^3 < Re \leq 1.2 \times 10^4$, 
\[ \log_{10} C_D = -2.4571 + 2.5558w - 0.9295w^2 + 0.1049w^3 \]

$1.2 \times 10^4 < Re \leq 4.4 \times 10^4$, 
\[ \log_{10} C_D = -1.9181 + 0.637w - 0.0636w^2 \]

$4.4 \times 10^4 < Re \leq 3.38 \times 10^5$, 
\[ \log_{10} C_D = -4.339 + 1.5809w - 0.1546w^2 \]

$3.38 \times 10^5 < Re \leq 4 \times 10^5$, 
\[ C_D = 29.78 - 5.3w \]

$4 \times 10^5 < Re \leq 10^6$, 
\[ C_D = 0.1w - 0.49 \]

$10^6 < Re$, 
\[ C_D = 0.19 - \frac{8 \times 10^4}{Re} \]

Where $w = \log_{10} Re$

| Table 3 Relationship for calculating the drag coefficient based on the Reynolds number [7] |
2.1.3 Upper Assembly

The purpose of the upper assembly is to hold the vane cassettes, allow optical access to the vane leading edge, provide infrastructure for film cooling and dilution jets, be modular in design to accept various sizes and shapes of vane cascades, provide a sealing mechanism with the lower assembly, and provide the transition from a cylindrical cross section to the shape of the entrance of the vane set. The upper assembly is shown in Figure 23. It was also necessary for the upper assembly to be supported independently of the lower assembly to allow for thermal growth independent of the lower sections.

Figure 23 Exploded view of the upper assembly
The upper assembly consists of the transition piece, view section, and the vane holder. The transition piece cross section begins round, with a diameter of 5.485 inches, and transitions into a 4 in by 10 in rectangular cross section. The transition piece will slide over the equilibration tube. There is a system of two sealing methods at the interface between the equilibration tube and transition piece to minimize hot air leakage, shown in Figure 24. The primary seal is created by seating four 1/4 inch square strips of flexible graphite inside the 1/4 inch slots on the end of the equilibration tube. The strips slightly intersect with the transition piece as it slides over the equilibration tube. This intersection causes the graphite strips to distort and create a snug seal between the two surfaces. The secondary seal is created by packing three 1/8 x 1/4 inch strips of flexible graphite inside the space between the equilibration tube and the transition piece near its flange. A separate flange with an axial extension on its inside radius will bolt into the flange at the base of the transition piece. As the two flanges tighten together the axial extension on the loose flange compresses the three graphite strips, shown in Figure 24. As they compress in the axial direction they expand in the radial direction and create a tight seal between the two faces. Flexible graphite was recommended by a sealing specialist due to its high resistance to temperature up to 6420°F and low coefficient of friction. When the lower and upper assemblies heat up they thermally grow in opposite directions, about 1.1 inches for the lower assembly and 0.65 inch for the upper assembly, so the least amount of resistance in movement between the two parts is favorable. Restricting the thermal growth of one of the pieces in the vertical direction could cause the part to buckle and fail.
The top of the transition piece is welded to the view section, seen in Figure 23. The purpose of the view section is to provide optical access to the vanes, create pattern factors by means of dilution jets, and support the entire upper assembly. The view section is an 8 x 11 x 8.625 inch box. The optical access is provided by extending two 3.5 inch ID pipes at an angle of 35° from the sides of the box, seen in Figure 23. The ends of the pipes are flanged so optical grade quartz windows may be sandwiched between the two flanges. With the flanges the pipes have an internal viewing diameter of 3.5 inches. This provides two views of the turbine vane set, shown representatively in Figure 25. The optical ports may also be used for velocity or temperature measurements using backscatter laser Doppler velocimetry (LDV), planar laser-induced fluorescence (PLIF), etc. The dilution jets are created by inserting two perforated plates at the hub and tip sides.
of the view section box, shown in Figure 26. Different hole patterns on the plates may be used to vary the pattern factors. This creates two thin chambers on each side of the box. Two 1 inch pipes extend out from each of these chambers, which provide the connections for the dilution jet air source. These pipes are made of RA601 since they will not experience the full temperature of the hot gas flow and piping is more readily available in this material. Two plates with slots extend out from the view section box. These plates are bolted into the I-beam supports, which support the entire upper assembly, shown in Figure 27. The view section is designed to be used for any small turbine vane set size (inlet area less than 28 square inches) and shape.

**Figure 25 Representative views through the optical access**
The vane holder section mounts to the top of the view section, shown in Figure 23. The purpose of the vane holder piece is to transition from a rectangular cross section to the actual shape of the turbine vane set, securely hold the vane set in place, and provide a means for supplying film cooling air to the vane set. The lower part of the vane holder section changes the cross section from a rectangle to the annular shape of a turbine vane ring. A shallow box is welded to the top of the transition section. Tolerances are kept very tight (on the order of 0.005 inch) in this piece so that the turbine vane set is securely held down. Ceramic gaskets are used to compress the turbine vane set, which will allow for a small thermal growth in the turbine vane and its holder. Inside this
shallow box are a set of plates, which separate the hub and tip sides of the turbine vane, shown in Figure 23. This is required since the amount of film cooling supplied to the hub can be different than the amount supplied to the tip. Three 1/2 inch pipes extend from each the hub and tip sides of the vane holder box for connecting up to the film cooling air supply. The material of these pipes is the same as the dilution jet pipe material since they experience a similar situation where they do not see the full temperature of the hot gas flow. The entire vane holder section must be redesigned for each type of vane set, which makes it possible to test various vane geometries with the TuRFR.

2.1.4 Structural Support

The upper assembly is mounted to a set of I-beams to support loads created by the exiting hot gas and to allow the upper assembly to thermally grow (downward) independent of the lower assembly. The strength of nickel-based alloys decreases as temperature increases, so it was favorable to minimize any loads on the upper and lower assemblies by use of separate support structures. The separate support structure is an I-beam system of four I-beams, shown in Figure 27. There are two 178 inch long beams (W8 x 40) spanning the north-south walls of the test chamber. At each end they rest on an I-beam (W8 x 40) stub, which is welded onto a 24 x 30 inch (1/2 inch thick) steel plate. Two 90 inch long I-beams (W8 x 40) span the distance between the two longer I-beams. The upper assembly is mounted to these cross I-beams.
Figure 27 Complete layout of the TuRFR, illustrating the positioning of the I-beams
The beams were designed to minimize deflection in the upper assembly caused from operation of the TuRFR. The beam deflection was calculated based on the free-body diagram given in Figure 28 and its governing equation in Equation 8. For the 90 inch long I-beams, $P$ is the weight of the upper assembly, $M$ is the moment created by the turning of the turbine vanes, and $R_A$ and $R_B$ are the reaction forces from the 178 inch long I-beams. For the 178 inch long I-beams $P$ is the largest reaction force from the cross beams, $M$ is zero, and $R_A$ and $R_B$ are the reactions from the wall mounts. Adding the deflections from the two sets of I-beams gives the total deflection at the upper assembly during operation. Based on an upper assembly weight of 325 lbs and a moment created by the turning of the hot fluid of 540 lb-in the deflection will be 0.04 inch.

**Figure 28 Simplified load model for the 90 and 178 inch long I-beams**

$$
\delta = -\frac{a}{6LEI} \left[ Pb(L^2 - b^2 - a^2) + \frac{q}{4}(L^3 - 2La^2 + a^3) + Ma(6a^2 - 4a^2 - 2L^2) \right]
$$

**Equation 8**

The wall mounts for the beams were designed to take the full load of the beams if one side were to fail. This assumes the bolted connection between the I-beams and the I-
beam stub welded on the wall mounts does not fail. The free-body diagram is given in Figure 29 and its solution in Equation 9. The wall plates were designed to have four bolts in the top and bottom each with a distance \( d \) apart, shown in Figure 30. Based on one wall mount failing the lower bolts will act as the pivot point for the moment \( M_w \). This will cause a tensile load in the top bolts equal to the moment divided by the distance \( d \). The loading on the bolts is 5550 lbs per bolt, which is below their rated loading of 20,000 lbs.

\[
R_w = R + qL, \quad M_w = Ra + \frac{qL^2}{2}
\]

Equation 9

![Figure 29 Simplified load model for the 178 inch long I-beams if one wall mount fails](image_url)
2.1.5 Instrumentation

Real-time measurements of the TuRFR will be done by a system of thermocouples, pressure transducers, mass flowmeters, an infrared camera and potentially laser-doppler velocimetry. Several K-type thermocouples are installed throughout the TuRFR for monitoring the gas and metal temperatures. The main air supply’s mass flow is measured by a set of choked flow orifice plates [28]. Two are used for verification of the mass flow measurements. When a flow is choked the mass flow can be calculated based on Equation 10, where \( d \) [in] is the orifice hole diameter, \( D \) [in] is the pipe diameter, \( C_0 \) is the discharge coefficient equal to 0.83932, \( y_{CR} \) is the critical flow function based on temperature and pressure and equal to 0.6887, \( p_1 \) [psia] and \( T_1 \) [°R] are the upstream static pressure and temperature, \( G \) is the specific gravity, and \( \gamma \) is the ratio of specific heats [25]. Equation 10 is valid as long as the flow is choked through the orifice. This can be verified by measuring the downstream static pressure, \( p_2 \) [psia], and testing it against the constraint given in Equation 11.
\[
\dot{m}[\text{lbm/hr}] = 2195.6C_0d^2y_{cr}p_1 \frac{1}{1 - \frac{\gamma}{2}(\frac{2}{\gamma + 1})^{(\gamma+1)/(\gamma-1)} \left(\frac{d}{D}\right)^4} \sqrt{\frac{G}{T_1}} \quad \text{Equation 10}
\]

\[
p_2 \leq \left(\frac{2}{\gamma + 1}\right)^{\gamma/(\gamma-1)} \frac{1}{1 - \frac{\gamma}{2}(\frac{2}{\gamma + 1})^{(\gamma+1)/(\gamma-1)} \left(\frac{d}{D}\right)^4} p_1 \quad \text{Equation 11}
\]

The fuel and premixer air mass flowrates are measured by a combination of a commercially available volumetric flowmeter (by Hedland), pressure gage, and thermocouple. The graduations on the flowmeters are designed for air, but the manufacturer provides a correction method for other fluids, shown in Figure 31. Multiplying the volumetric flowrate by the density of air at standard temperature and pressure (STP), provides the mass flow of each of these lines.
Figure 31 Volumetric flowmeter correction factor guidelines [15]

The turbine vane surface temperature will be measured by an Electrophysics Silver 420 shortwave infrared camera. It has a 20° x 16° field of view and a spectral response range of 3.6 to 5.1 microns. The camera has a sensitivity of 0.02°C due to cryogenic cooling of its detector to a temperature of 70K. The camera has a maximum frame rate of 100 Hz with a temperature measurement range of 5 to 1500°C.
2.2 Auxiliaries Design

2.2.1 Location

The TuRFR is located in the southern rocket pit (room 191C) of the Aeronautical and Astronautical Research Laboratory (AARL). The room has inner dimensions of 180 x 222 inches with 18 inch thick reinforced poured concrete walls on the north, west and south walls. The rocket pit has steel bay doors on the east side and the entire roof opens. There are 3 inch holes along the top and bottom of the north and south walls. There are two 13 x 19 inch windows on each the north and south walls, one of which looks into the control room to the south.

2.2.2 Air Supply

The TuRFR uses air fed from the high pressure system at the AARL. The high pressure system utilizes three Ingersol-Rand reciprocating compressors to boost the air pressure to 2350 psig at 0.73 lbm/s per compressor. The high pressure air is stored in two 750 cubic feet pressure vessels. The control room layout is shown in Figure 32. The TuRFR implements a high pressure regulator to drop the high pressure air to the appropriate level depending on the mass flowrate desired. After the regulator, the air line is split into two 2 inch lines to allow for metering using a set of choked flow orifice plates. The air is then fed into four 1 inch hoses connected to the base of the TuRFR. For safety, there is an automatic shutoff valve upstream of the regulator, which closes if the pressure immediately upstream of the orifice plates exceeds 350 psig. A relief valve downstream of the choked flow orifice plates opens if the pressure exceeds 190 psig. The
pressure losses in the airline of the TuRFR are significant, which requires a minimum pressure in the high pressure system based on the required mass flow. For example, a mass flowrate of 2.6 lbm/s requires a minimum tank pressure of 1000 psig.
Figure 3.2 Control room layout
2.2.3 Fuel Supply

Raising the temperature of such a large mass flow of air to 2200°F requires a large amount of fuel. The first step in determining the fuel requirements is to define the chemical reaction based on the equivalence ratio, assuming lean conditions, given in Equation 12. Next, an equivalence ratio is selected at which combustion is expected to occur. The fuel mass flowrate can be calculated based on the equivalence ratio and the air mass flowrate, given in Equation 13. The temperature of the products of combustion can be assumed to be the adiabatic flame temperature, which is based on the equivalence ratio, given in Figure 33. The final temperature of the entire flow can be calculated by using an energy balance, mass flow multiplied by enthalpy, between the unreacted air and the products of combustion, assuming an adiabatic process. Based on this method with an equivalence ratio of $\phi = 1.0$, air mass flow of 2.6 lbm/s, and a final mixture temperature of 2200°F the TuRFR was estimated to utilize 0.080 lbm/s of fuel or 288 lbm/hr.

$$CH_4 + \frac{2}{\phi}(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + \frac{7.52}{\phi} N_2 + \left(\frac{2}{\phi} - 2\right)O_2$$  \hspace{1cm} \text{Equation 12}

$$\phi = \frac{\left(\frac{m_f}{m_a}\right)_{\text{actual}}}{\left(\frac{m_f}{m_a}\right)_{\text{stoich}}} = \frac{\left(\frac{m_f}{m_a}\right)_{\text{actual}}}{\left(\frac{m_f}{m_a}\right)_{\text{stoich}}}$$  \hspace{1cm} \text{Equation 13}
Figure 33 Adiabatic flame temperature for methane (plot created by data from Chemkin, a commercially available combustion software package)

The TuRFR’s burner requires a significant amount of natural gas such that the supply from the city line is insufficient, so the fuel is supplied by several banks of 3500 psig K-bottles, shown in Figure 34. Each K-bottle can hold approximately 14 lbm of fuel at 3500 psig. The gas for the bottles is supplied from the city line through two FMQ 2-36 Fuelmaker compressors, which have a discharge pressure of 3600 psia and a volumetric flowrate of 1.9 scfm. Downstream of the bottles the fuel pressure drops through a pressure regulator, which allows control of the mass flowrate. The mass flowrate is measured by the combination of a volumetric flowmeter, pressure gage, and thermocouple immediately downstream of the needle valve. Following the flowmeter the
fuel enters a tee, which is connected with the air line. This serves as the premixer. Air can be supplied to the premixer via a needle valve downstream of the main air line orifice plates, seen in Figure 32. The mass flowrate of the air added to the premixer is measured similarly to the fuel, by the combination of a volumetric flowmeter, pressure gage, and thermocouple. After the premixer, the fuel-air mixture splits into eight lines, which feed into the burner. It is particularly important to be careful during premixing that the equivalence ratio in the fuel lines is well above the flammability limits. At atmospheric pressure these equivalence ratio limits are approximately 0.55 to 1.8. But at higher pressures the upper flammability limit increases, as seen in Figure 35. At a pressure of 30 psia the upper flammability equivalence ratio limit increases to 4.0.

Figure 34 Two gas cylinder banks of natural gas
Safety is especially important with the use of natural gas at high pressures in a confined space. Methane sensors were installed throughout the rocket pit as well as the rooms that surround it. All openings from the rocket pit to the surrounding rooms were sealed off as much as possible. Relief valves were placed in each bank of cylinders (eight K-bottles, set pressure of 3450 psig) and downstream of the fuel pressure regulator (set pressure of 500 psig). All valves and equipment are rated well over their respective operational pressures. During premixing the equivalence ratio in the premixer is closely monitored and maintained in the rich range above the flammability limits as noted earlier. All gas cylinders have current certifications, which are valid until May 2013. There is a
main fuel shutoff valve in the control room and the rocket pit should the fuel need to be cut off.

2.2.4 Film Cooling Air

The current design of the film cooling supply is to use a 97 kW Chromalox heater (Model #GCHB-2797XX) to elevate the temperature of the film cooling air and inject it into the film cooling ports on the vane holder section of the upper assembly. Film cooling parameters that are to be matched to that of real engines are the velocity ratio, density ratio, mass flux ratio, and momentum flux ratio, given in Equations 14, 15, 16, and 17, respectively. By matching the velocity and density ratios the mass and momentum flux ratios are matched. Typical gas turbine engines operate near a density ratio of 2.0 [4], which is the goal of this facility.

\[ VR = \frac{V_c}{V_\infty} \quad \text{Equation 14} \]

\[ DR = \frac{\rho_c}{\rho_\infty} \quad \text{Equation 15} \]

\[ M = DR \cdot VR \quad \text{Equation 16} \]

\[ I = DR \cdot VR^2 \quad \text{Equation 17} \]

The Chromalox heater is designed to operate on 480VAC, which would produce the 97 kW rated power. Unfortunately, only 208VAC was available. Since the heater is a
A resistive heater's resistance remains constant as the power supply is switched from 480V to 208V. Using Ohm’s law, Equation 18, the power output at 208V is 18.4 kW. If the heater is assumed to be 100% efficient then the expected temperature rise for a film cooling mass flow of 0.13 lbm/s (5% of 2.6 lbm/s) is 560°F, according to Equation 19. This would only provide a density ratio of 2.5 with a gas temperature of 2200°F. It will be necessary to install a 480V power supply if higher temperatures are desired from this heater.

\[ R = P_1 \frac{V_1^2}{V_2} = P_2 \frac{V_2^2}{V_2} \]  \hspace{1cm} \text{Equation 18}

\[ \Delta T = \frac{P}{mc} \]  \hspace{1cm} \text{Equation 19}

The air for the film cooling is supplied from the main air line immediately downstream of the choked flow orifice plate mass flow meters. The air then travels through the heater located in the northeast corner of the rocket pit. The exhaust flow from the heater will enter two manifolds, one for each side of the vane holder (hub and tip). Each manifold will have a metering valve and mass flowmeter. Each manifold will have three lines exiting them downstream of the mass flowmeter and metering valve. These lines will have a section of flexible hose to allow for thermal growth in the line. Further details of the downstream end of the heater are still being designed.
2.2.5 Particulate Feed System

The particle feed system is patterned after a commercial volumetric feeder, typical design given in Figure 36. The key elements of the feeder are the hopper, agitation system, and auger. The hopper stores the particulate to be ingested. The agitation system keeps the particulate from sticking and caking. The auger forces the particulate out of the feeder into the flow. The current design for the particulate feed system is a modified version of the volumetric feeder, shown in Figure 37. The hopper is a large diameter pipe, which connects to a small diameter pipe tee fitting. The auger sits inside the small diameter pipe tee and extends through a 3/8 inch diameter pipe. The end of the 3/8 inch pipe connects to a 1 inch tee. The airflow passes through the 1 inch tee normal to the auger to entrain the particles as they are ejected by the auger. This air-particle mixture is then fed into the TuRFR immediately upstream of the burner. No agitation system is currently designed. Based on testing and analysis of this setup further steps will be made to design an agitator, if necessary. The air supply for the particulate feed system will come from the main air line immediately downstream of the choked flow orifice plates.
Figure 36 Photo of typical volumetric feeders

Figure 37 Schematic (left) and photo (right) of particle feed system
2.2.6 Other Auxiliaries

The TuRFR has a large surface area, which can be a source of high heat losses. The heat losses to ambient would necessitate more burning of natural gas to maintain a hot flow at the turbine inlet. To minimize these losses a 1 inch thick ceramic blanket is wrapped around the spool piece, cone, equilibration tube, and the upper assembly, shown in Figure 13. The blanket is a low density flexible quilted blanket constructed of alumina fiber made by Zircar Ceramics. It has a maximum use temperature of 2912°F with a thermal conductivity of 0.23 W/m/K at 2200°F.

The TuRFR is monitored by a Labview program shown in Figure 38. The mass flow of the total air is monitored from the choked flow orifice plates. The fuel and premix air mass flowrates are monitored by entering the pressure and volumetric flowrate of each into the Labview screen. The program monitors the equivalence ratio of the fuel line and warns if there is a flashback danger in the line. Four temperature measurements are monitored from the K-type thermocouples, which more can be added as necessary. In the lower right-hand corner of the screen is a window for the display from a webcam. The webcam is positioned looking through one of the spool piece’s viewports to monitor the flames. All the white background cells are manual inputs and the gray background cells are updated real-time from the data acquisition system.
Figure 38 Screen capture of the Labview program
CHAPTER 3

VALIDATION TESTING

Tests used for validating the TuRFR include a flow visualization of the exit flow from the aluminum honeycomb, a Kiel probe test, an exit temperature profile from the equilibration tube, and a heater exit temperature test.

3.1 Flow Visualization for Flow Conditioning System

The qualitative flow visualization of the air flow downstream of the honeycomb was done with the cone and burner section removed. A square mesh screen was laid across the top of the steel base. Several tufts were attached to the screen, which were the means of the flow visualization. A visual inspection was made for consistency in the way the tufts stood while air was flowed through the steel base. The final test with the mesh screen proved to have fairly uniform flow. There were some high and low flow spots, but these were rare with uniformity dominating.

3.2 Kiel Probe Test

A Kiel probe was also used to accurately measure the flow uniformity at the exit of the equilibration tube. A Kiel probe is an excellent total pressure probe for turbulent flows where the direction of the flow varies widely and is unknown. It can accurately measure the total pressure at an angle up to about 45° from the flow direction [31]. The
velocity of the flow can be calculated from the Kiel probe’s pressure measurement according to Equation 20. This equation is only valid for incompressible flows since it’s based on Bernoulli’s equation. The Kiel probe was connected to a mercury manometer with 0.1 in Hg graduations. A total of 31 datapoints were taken in the radial direction along the exit plane of the equilibration tube, with r=0 corresponding to the center of the equilibration tube. The results are given in Figure 39, where the pressures have been converted to velocities and normalized by the freestream velocity. The last point on the right side of Figure 39 corresponds to the tube wall location at r=2.375 inches. These results confirm a fairly uniform velocity profile with lower velocities near the tube wall.

\[ v = \sqrt{\frac{2\Delta p}{\rho}} \]  

\textit{Equation 20}
The boundary layer can also be seen in Figure 39, where the velocity begins to reduce near the tube wall. The boundary layer thickness is defined by the portion of the flow that is less than 99% of the freestream velocity. The point in Figure 39 where the velocity is approximately 99% of the freestream is between $r=1.745$ inches and $r=1.824$ inches. An average of these two locations will be taken as the limit of the boundary layer, which is 1.785 inches. This corresponds to a boundary layer thickness of 0.590 inch. For this test the gas temperature was 41.5°F and the Reynolds number based on the equilibration tube inner diameter was 679,000. A Reynolds number greater than 10,000 for pipe flow indicates turbulent flow [17]. For turbulent pipe flow to be fully developed
a minimum tube length of 10D is required, which is 47.5 inches. The equilibration tube is 31 inches, so the flow is not fully developed at its exit. The theoretical boundary layer thickness for a developing turbulent flow, assuming a flat plate, is given in Equation 21 [13], where Reynolds number is based on the position on the flat plate. The theoretical boundary layer thickness is 0.539 inch, which is 8.6% less than the measured value. One factor that could be accountable for this difference between measured and theory is the tube wall is not a flat plate. The ratio of the boundary layer thickness to the tube radius is 0.25. This appears large to make the assumption of a flat plate. The flat plate would underestimate the boundary layer thickness for a curved surface. Another factor causing a difference between measured and theory is the boundary layer development upstream of the equilibration tube along the cone wall. This would cause an initial boundary layer thickness at the equilibration tube inlet, resulting in a thicker boundary layer at the equilibration tube exit.

$$\delta(x) = \frac{0.37x}{(\text{Re}_x)^{0.2}}$$  \hspace{1cm} \text{Equation 21}

When the TuRFR operates at full temperature, 2200°F, the Reynolds number based on the inner diameter of the equilibration tube is 98,000. This is also a turbulent flow that is not fully developed at the equilibration tube exit. Based on Equation 21, the theoretical boundary layer thickness is 0.787 inch. Assuming this underestimates the actual boundary layer thickness by the same factor as the cold test, the expected boundary layer thickness is approximately 0.861 inch.
3.3 Equilibration Tube Exhaust Temperature

The exit temperature of the equilibration tube is another condition of importance. This is measured by placing a thermocouple rake with five thermocouples across the exit plane of the equilibration tube and rotating it about the equilibration tube’s center axis, shown in Figure 40. The thermocouples were placed such that one is in the center, two at a radial distance of \( R_1 = 1 \) inch, and two at a radial distance of \( R_2 = 2 \) inches. The rake was rotated at angular increments of 15°F to provide a thermal map of the entire equilibration tube exit plane. This resulted in a total of 60 data points. The test was done at 2.6 lbm/s air mass flow and a fuel mass flow of 0.06 lbm/s. The air mass flow is the expected mass flowrate for testing a set of CFM56 vane cassettes. Initial testing demonstrated a very nonuniform temperature profile, which necessitated the use of multiple thermocouples. The temperature profile from the test is given in Figure 41, where the coordinates are in inches and the contours are in °F. The theoretical temperature for this flow is 1750°F, based on the theory given in section 2.2.3. The higher temperatures were near the center of the tube. The flow near the equilibration tube wall was measured to be approximately 1000°F. This suggests the wall temperature is considerably lower than the freestream gas temperature. During this test the insulation was removed for the last 12 inches of the equilibration tube to provide access for the thermocouple rake. An energy balance on the tube wall confirms without insulation the tube wall will be several hundred degrees Fahrenheit cooler than the freestream gas due to radiative losses. These losses due to radiation were the cause of the reduced temperatures near the tube wall. Figure 42 is a plot of the temperature profiles from the thermocouple rake rotated from 0° to 165°, with
15° oriented parallel to the west-east line. This provides clear visualization of the effects of the heat losses through the tube walls without the insulation with temperatures peaking near the center. The center probe (r=0) had lower temperature readings as the thermocouple rake was rotated greater than 90°. The rake was internally cooled with air, and some of this air would leak out through the holes drilled for the thermocouples. It was noticed after the test the center probe was bent upwards close to one of the holes. The cooling air leaking through the hole most likely was the cause of the reduced temperature readings. Figure 41 shows the temperature concentrations towards the southwest portion of the equilibration tube. This is most likely due to the higher concentration of flameholders in the southern portion of the combustor, shown in Figure 20 (south is oriented to the left in the photo). To provide a more uniform temperature profile, the ceramic insulation should be added to the rest of the equilibration tube and the fuel flowing through the south flameholders should be reduced.
Figure 40 Thermocouple positions at equilibration tube exit

Figure 41 Equilibration tube exit temperature profile for fuel mass flow of 0.06 lbm/s
The temperature measured by the thermocouples can underestimate the actual gas temperature due to heat losses. An energy balance on the thermocouple results in the governing equation given in Equation 22, where $T_\infty$ is the freestream gas temperature, $T_j$ is the thermocouple probe temperature, and $T_s$ is the temperature of the solid surfaces in the view of the thermocouple. The conduction term is assumed to be negligible since the Biot number is close to or less than 0.1, given in Table 4. Since the thermocouple views both the hot surface of the equilibration tube wall and the ambient temperature surface of the roof and walls of the rocket pit, the total radiation is assumed to be the average of these two views, given in Equation 23. Emissivities are approximately 1.0 for the various surfaces. For a spherical probe the convective heat transfer coefficient can be found from Equations 5 and 6. Four of the thermocouples were sheathed with a diameter of 1.52 mm, and the center thermocouple had a bead diameter of 0.13 mm. Table 4 includes the
radiation effects for these two diameter probes. If the probe for the sheathed thermocouple provides a temperature of 2200°F (1478 K), then the actual gas temperature can be as high as 2460°F (1622 K). The bare thermocouple is less sensitive to radiation effects due to its small diameter bead. It would only have a maximum error of 67°F (37 K). Two methods of reducing the effects of radiation are to reduce the thermocouple bead’s diameter and to immerse it in the flow such that it does not view any cold surfaces.

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Table 4 Thermocouple radiation effects on measured temperature
\[ \bar{h}(T_\infty - T_j) = k \frac{\partial^2 T}{\partial x^2} + \varepsilon\sigma(T_j^4 - T_s^4) \]  
\[ \bar{h}(T_\infty - T_j) = \frac{\sigma}{2} [ (T_j^4 - T_{tube}^4) + (T_j^4 - T_{ambient}^4) ] \]

**3.4 Heater Test**

The heater exit temperature was tested for a mass flowrate of 0.13 lbm/s. The heater has a built-in J-type thermocouple located immediately downstream of the exhaust. The heater was assumed to be at steady state conditions when the sheath temperature near the heating coils became constant, which is included in the heater’s overtemp display. The sheath temperature reached a maximum of 656°F. At this steady state condition the heat exhaust fluctuated between 474°F and 497°F. This is quite close to the theoretical exit temperature calculated in section 2.2.4 at 560°F. Discrepancies between the measured and theoretical case are a result of inefficiencies in the heater and heat losses. Based on this test the heater is capable of supplying film cooling air at a density ratio of 2.8 for a freestream temperature of 2200°F.
CHAPTER 4

CONCLUSIONS AND RECOMMENDATIONS

4.1 Anticipated Studies

It is anticipated that the facility will be used for several experimental studies. One of the current studies for this facility is turbine deposition studies, which is funded by the Department of Energy and National Energy Technology Laboratory. The purpose of this work is to verify turbine design features that prevent them from being susceptible to the effects of deposition, corrosion, and erosion. This would be accomplished by studying deposition effects on various real turbine vanes through an accelerated deposition test. The test would last several hours and in that time match the total mass throughput of particles a typical gas turbine engine would experience during normal operation over a given period, such as one year.

Many other studies can be done with the TuRFR. One type of study could be to vary pattern factors and analyze their effects on the turbine vane metal temperature. In conjunction, the facility could be used to study how different film cooling schemes would alleviate or exacerbate these effects. To increase the allowable metal temperature of a turbine vane, ceramic matrix composites are being developed for turbines. The TuRFR would be capable of testing this new material once hardware has been created. Also, the
TuRFR can be used to validate lower temperature tests from facilities like the rotating transient test facility or the fixed turbine vane cascade rigs. Possible research topics might also include inter-turbine burners. As the hot gas is expanded through the turbine it is possible to reheat that fluid as it cools from expansion. This is a relatively new area of research, which may be tested in the TuRFR. Also, validation of CFD data is an important aspect of the TuRFR, and is a significant portion of the work currently being funded by DOE and NETL.

4.2 Lessons Learned

Many valuable lessons were learned during the construction and testing of the TuRFR. These include the following:

- During lighting of the TuRFR it is imperative to vent residual fuel if lighting is unsuccessful after five seconds. The fuel remaining inside the TuRFR can reach a combustible mixture level, which would cause a deflagration wave out of any opening when trying to relight.

- Pipe losses and pressure drops through regulators are significant sources of pressure losses. These pressure losses increase the required source pressure to operate the facility. Since the TuRFR may require more air than can be supplied by the air compressors an increase in required source pressure will reduce the maximum runtime of the facility.

- Pipe unions make piping reworks significantly easier and less time consuming.

- Combustion is very complex. Much time was spent in the design of the burner. It is not a simple matter of adding fuel and air.
• When purchasing a hoist make sure it can support significantly more weight than is expected to be lifted with it. Deflection can be an issue even when the stress is within the tolerances of its design. The hoist purchased to lift the cone is designed to handle the weight but deflects significantly when loaded.

• The volumetric flowrates that volumetric flowmeters provide are at standard temperature and pressure (STP).

• When measured results do not match the accepted theory start with verifying that the instrumentation is functioning properly and correctly calibrated.

4.3 Recommendations

Upon examining the overall design and layout of the facility it might be a good idea to add or change a few things. This is a list of recommendations for the TuRFR:

• Add pressure transducers in the fuel and premix lines. This would help with maintaining real-time calculations of the line’s equivalence ratio.

• Install a switch to the compressors so that if the pressure in the tanks falls below a certain pressure the compressors will turn on automatically. It takes a considerable amount of time (days) to pressurize the gas tanks.

• Install more fire extinguishers in and around the rocket pit.

• Purchase two-way radio earmuffs to ease communication during operation of the TuRFR. The noise levels during operation are high enough to make communication difficult, especially when someone in the rocket pit is trying to communicate with someone in the control room.
• Increase the size and number of viewport holes on the spool piece. Larger holes would allow for better visual access to the flames.

• Add flashback arrestors to the fuel line in case a flammable mixture level is reached in the gas line.

• Install pressure transmitter and indicator lights for the air tank pressure and air compressor status. It is important to verify all three compressors are running the entire time during a high mass flow test, since compressors can shutoff automatically for a number of reasons.

• Install an ignition system to the burner.

• Remove all the steel half flanges from the cone and spool piece before removing the cone with the hoist. The hoist is rated for 500 lbs, which is more than the weight of the cone, spool piece, and four half steel flanges. But the deflection is significant when lifting this combined load. Removing the steel flanges reduces the weight by about 200 lbs and significantly reduces the deflection in the hoist.

4.4 Conclusions

The TuRFR is a new high temperature facility capable of running at temperatures of 2200°F, pressures of 30 psig, and mass flowrates as high as 2.6 lbm/s. It is located at the Aeronautical and Astronautical Research Laboratory at The Ohio State University. This is a state-of-the-art facility for testing real turbine vanes. This facility is particularly important since turbine hardware operates under such severe conditions. Trends show the conditions to be even more severe in the future. The design, construction, and preliminary validation testing of this facility is the subject of this thesis.
REFERENCES


[28] Smith, Chris, Contribution by Chris Smith.


APPENDIX A

EQUIPMENT DRAWINGS
Figure 43 TuRFR lower assembly

- EQUILIBRATION TUBE (INCO 617 1/4" THICK)
- CONE (INCO 617 3/16" THICK)
- INCONEL FLANGE x 3 (INCO 617 3/16" THICK)
- VIEW SECTION (INCO 617 3/16" THICK)
- STEEL FLANGE (MILD STEEL 3/4" THICK)
- BASE SECTION (MILD STEEL 3/8" THICK)
- WELD 4 X 1-1/2" NPT HALF COUPLINGS (NOTE: CAREFUL NOT TO POSITION HALF COUPLINGS OVER COMBUSTOR_FLOOR FLANGE HOLES)
- BLIND FLANGE (MILD STEEL 1" THICK)
Figure 44: Steel base assembly

STEEL FLANGE (MILD STEEL)

STEEL_RING

COMBUSTOR_FLOOR

HONEYCOMB_LIP

WELD 4 X 1-1/2" NPT HALF COUPLINGS
(NOTE: CAREFUL NOT TO POSITION HALF COUPLINGS OVER COMBUSTOR_FLOOR FLANGE HOLES)

SECTION A-A

ALL MACHINED SURFACES 32

TOLERANCE

XX = ±.01

XXX = ±.005

DIMENSIONS ARE IN INCHES

The Ohio State University
Figure 45 Steel base lower flange

88

WELD THIS COMBUSTOR_FLOOR TO THE BOTTOM OF THE STEEL_RING AS SHOWN IN THE COMBUSTOR_ASSEMBLY DRAWING.

SCALE .075

21.50

8 x Ø 0.563

21.50

1.00

88
Figure 46 Honeycomb support ring inside steel base
WELD ONE MILD STEEL FLANGE TO THE TOP OF THE STEEL RING AS SHOWN IN THE ASSEMBLY DRAWING.

1 EACH MILD STEEL

<table>
<thead>
<tr>
<th>TITLE</th>
<th>MATERIAL</th>
<th>SCALE</th>
<th>ALL MACHINED SURFACES ( \leq \frac{2}{3} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>FLANGE</td>
<td>MILD STEEL</td>
<td>0.75</td>
<td>DIMENSIONS ARE IN INCHES</td>
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<tr>
<td>REV</td>
<td>NO</td>
<td>2D</td>
<td></td>
</tr>
</tbody>
</table>

DIMENSIONS:

- \( \phi 39.00 \)
- \( \phi 47.00 \)
- 22.00
- 32 THRU HOLES \( \phi 0.563 \)

FIGURE 47 Steel base upper flange
Figure 48 Steel base cylinder

- Weld 4 x 1-1/2" NPT half couplings
- All machined surfaces ±0.01
- XXX = ±0.005
- Dimensions are in inches

The Ohio State University

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<tr>
<th>DRAWN</th>
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<tbody>
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<td>0.075</td>
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</table>
Figure 49 Spool piece assembly

NOTE: 3D DRAWING IS MISLEADING. VIEW PORTS ARE NOT TO BE LOCATED DIRECTLY UNDER THE FLANGE HOLES.

SECTION A-A

INCO_RING

FLANGE (INCONEL)

INCO_VIEW_PORT_TUBE
(NOTE: DO NOT POSITION THE VIEW PORTS DIRECTLY UNDER THE FLANGE HOLES. THEY MUST BE OFFSET.)

VIEW_PORT_CAP

NOTE: 3D DRAWING IS MISLEADING. VIEW PORTS ARE NOT TO BE LOCATED DIRECTLY UNDER THE FLANGE HOLES.

SCALE .075

ALL MACHINED SURFACES 32

DIMENSIONS ARE IN INCHES

The Ohio State University

DRAWN
Cramer 4/3/2008

CHECKED

QA

TOLERANCE

.XX = ±.01

.XXX = ±.005

MFG

PROD

TITLE

VIEW_SECTION.Assembly

SIZE A

MATERIAL INCO 617

REV

drawing sheet

DIMENSIONS ARE IN INCHES

SCALE 0.2

SHEET 1 OF 1
Figure 50 Spool piece cylinder

SECTION A-A

Ø39.00

90°

.188

6.00

3.00

2 THRU HOLES Ø.662

ALL MACHINED SURFACES 32

DIMENSIONS ARE IN INCHES

The Ohio State University

DRAWN

CHECKED

QA

TITLE

MPG

TOLERANCE

.AXX = ± .01

.AXXX = ± .005

SIZE

MATERIAL

DWG NO

REV

SCALE

0.1

3A

1

SHEET 1 OF 1
Figure 51 Spool piece viewport

DRAWN
Cramer 1/23/2008
CHECKED
The Ohio State University
QA
MFG

TOLERANCE
XX = ± .01
XXX = ± .005

DIMENSIONS ARE IN INCHES

SCALE 1 : 1
ALL MACHINED SURFACES 32 ∀
Figure 52 Spool piece viewport cap

THIS CAP IS FOR HOLDING A
\( \phi \).750 QUARTZ WINDOW (NOT OPTICAL GRADE)

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<th>REV.</th>
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The Ohio State University

TOLERANCE

\[ .XX = \pm .01 \]
\[ .XXX = \pm .005 \]

DIMENSIONS ARE IN INCHES

SCALE 1.5

SECTION A-A

INTERNAL THREAD
13/16-28 UN - 2B

\( \phi .625 \)

\( \phi .774 \)

1.000

SCALE 3.5

SHEET 1 OF 1
Figure 53 Cone assembly

**INCO_EQUILIBRATION (WELD THIS TO THE INCO_CONE)**

**INTERNAL WELDS TO BE GROUND SMOOTH**

**SECTION A-A**

**FLANGE (INCONEL)**

**INCO_CONE**

**SCALE .05**

**ALL MACHINED SURFACES ± .005**

**DIMENSIONS ARE IN INCHES**

---

**DRAWN**

**CHECKED**

**DRAWN**

**CHECKED**

The Ohio State University

**NOZZLE_ASSEMBLY**

**SIZE** A

**MATERIAL** INCO 617

**DRAWNO** 4

**REV** 0

**SCALE** 0.1

**SHEET 1 OF 1**
Rings are made with 3/16 in. plate to be 1 in. wide on top.

Comments and suggestions are welcome. The function of this cone is to be and acceleration metric that can withstand a pressure of 120 psig. This pressure is to ensure a safety factor of 3.
Figure 55 Equilibration tube

3 EACH (ONE (1) WELDED TO CONE AND TWO (2) SHIPPED LOOSE)

WELD THIS INCO_EQUILIBRATION TO THE INCO_CONE AS SHOWN IN THE NOZZLE_ASSEMBLY DRAWING.
Figure 56 Equilibration tube extension
WELD ONE INCONEL FLANGE TO THE INCO_CONE AS SHOWN IN THE NOZZLE_ASSEMBLY DRAWING, ONE TO THE TOP OF THE INCO_RING AND ONE TO THE BOTTOM OF THE INCO_RING AS SHOWN IN THE VIEW_SECTION DRAWING.

3 EACH INCONEL

32 THRU HOLES Ø.563

SCALE .075

.188

INCONEL FLANGE

THE OHIO STATE UNIVERSITY

DRAWN
Clامر
4/3/2008

CHECKED

QA

MRG

TOLERANCE

.XX = ± .01

.XXX = ± .005

DIMENSIONS ARE IN INCHES

ALL MACHINED SURFACES √
6 EACH MILD STEEL

TO BE SHIPPED LOOSE.

16 THRU HOLES Ø.56

R23.50

R19.75

22.00

.75

SCALE .075

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<td></td>
<td>.XXX = ±.005</td>
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ALL MACHINED SURFACES ±.005

DIMENSIONS ARE IN INCHES

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<th>SIZE</th>
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<tr>
<td>A</td>
<td>MILD STEEL</td>
<td>5</td>
<td>0</td>
</tr>
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SCALE 0.1
Figure 59 Upper assembly
Transition Piece

NOTE: The purpose of this piece is to make a round to rectangular transition.

The side walls are to be made from 3/8" plate.

This piece must be welded to the bottom of the view section (Sheet 3). Refer to assembly view (sheet 1) for reference.

Manufacturer will supply necessary bolts for assembly, will check fit with mating components, and will deliver assembled.
Figure 6.1 View section

NOTES: There are two elliptical holes, one on each side, of the same dimensions and locations.

There must be through holes to accommodate the external pipes on the side.

The holes on the flange and slots must be drilled for 1/8" bolts.

Manufacturer will supply bolts and nuts capable of handling temp. and assemble, including combustor simulation plates, prior to delivery to ensure proper fit.

This piece must be welded to the top of the transition piece, as shown in the entire assembly View (-sheet 1).
Combustor simulation plates

Qty: 4

NOTE: Give all corners a small round to prevent gouging slots upon assembly.

**Use nominal plate thickness. Thickness is not critical. Critical aspect is that these plates fit in the slots in the view section. Check fit before delivery and deliver assembled.
VIEWPORT PIPE

This piece connects to the elliptical cutouts on the view section.

NOTE: Manufacturer should provide bolts and nuts to assemble.

Manufacturer should also check with viewport cap to ensure proper assembly.

This piece welds onto View Section as shown in Entire Assembly Views.

---

b bolt circle for 1/2" bolts at 9" diameter.
viewport pipe cap
End Cap Bottom

NOTE: This entire piece is to be made with 3/8" thick material.

The purpose of this piece is to make a smooth transition from a rectangular section to an annular section.

All flange holes should be drilled to fit 1/2" bolt. Manufacturer will supply all necessary bolts to withstand 200°F and check fit with entire assembly before delivering assembled.

This piece should be welded to End Cap Midline (sheet 8) as shown in End Cap Assembly View (sheet 1).
NOTE: This entire piece is to be made from 2/3" material.

All Range holes are to be drilled to fit 1/2" bolts.

Manufacturer will supply bolts and nuts for assembly able to withstand 2200°F, and will check fit with mating pieces before delivering assembled.

***These dimensions are dimensions for the recess,

****The orientation of the Film Cooling Retainer is shown on sheet 9.
Figure 67 Film cooling retainers for end cap middle

Film Cooling Retainers
These pieces are to be welded as shown on End-Cap Middle (sheet 8).

NOTE:
All Film Cooling Retainers are to be made from same material as other components, 1/8" thick.
**End Cap Top**

**Figure 68** End cap top

---

**NOTE:** This entire piece is to be made from 3/16" material. All flange holes are to be drilled to fit 3/8" bolts. Manufacturer will supply bolts and nuts for assembly able to withstand 2200 psi, and will check fit with mating pieces before delivering assembled.

**** These dimensions are dimensions for the recess. The recess does not need to have square corners, but they need to be as square as possible with OEM.
Figure 69 Film cooling pipes

**Lower Film Cooling Pipe**

- Ø.50
- Put standard NPT threads on end of pipe.

**Upper Film Cooling Pipe**

- Ø.50

**Note:** These pipes are to be welded onto End Cap Hinge so that they protrude from the side. Refer to Entire View Assembly.

Use standard 1/2", sch. 40 pipe of same material as rest of assembly.
Film Cooling Walls

These plates are to come attached from the rest of the components.

NOTE: The thickness of these plates is NOT critical.
Figure 71 Seal flange
APPENDIX B

OPERATIONAL PROCEDURE
1. Turn on all three reciprocating air compressors following their startup procedure.
   a. Wait until tank air pressure is above 2000 psig.

2. Open rocket pit roof and east bay doors completely.

3. Before proceeding confirm the following:
   a. The gas cylinders are fully charged (~3500 psig).
   b. The fuel ball valve on the south wall of the rocket pit, the fuel ball valve on the north wall of the control room, and the fuel regulator in the control room are closed completely.
   c. The valves on the natural gas cylinders are open as well as the fuel valves on the north wall of the rocket pit.
   d. All fuel ball valves surrounding the combustor are closed except for the one under the spark plug hole.
   e. All persons utilizing the TuRFR have proper eye and ear protection

4. Open fuel ball valve on south side of rocket pit.

5. Start Labview data acquisition system
   a. Confirm power supply unit is on
   b. Confirm camera placement is good

6. Two persons will be required for the next several steps (Person 1 is inside the rocket pit and Person 2 is inside the control room):
   a. Person 2: Open main air ball valve inside control room to one-quarter open
   b. Person 2: Increase main air pressure regulator to an air mass flowrate of 0.1 lbm/s
c. Person 1: Remove spark plug on west side of combustor

d. Person 1: Ignite butane torch and insert into spark plug port ensuring that the butane torch does not get blown out by the air exiting the spark plug hole. If flame is blown out immediately relight the butane torch. Failure to do so could result in too much fuel entering the combustion section. Once ready signal to Person 2 to turn on the fuel.

e. Person 2: Once Person 1 gives the signal do the following to turn on the fuel:

   i. Open fuel ball valve on north wall of the control room
   
   ii. Increase pressure regulator till a flowrate of 20 SCFM registers on the fuel flowmeter

   iii. If lighting is unsuccessful after 5 seconds:

      1. Turn off the fuel immediately by closing the fuel ball valve.
      
      2. Open up the main air regulator to a mass flowrate of 2 lbm/s. Let air flow for one minute.

      3. Repeat procedure starting with step 5

         a. Remember the fuel regulator is already set, so once the fuel ball valve is opened fuel will begin to flow

f. Person 1: Thread spark plug or plug into the spark plug hole.

g. Person 1: Proceed opening two fuel valves counter-clockwise of the spark plug port, waiting five seconds between opening them
h. Person 2: Further open the fuel pressure regulator till a flowrate of 30 SCFM registers on the fuel flowmeter. Record mass flowrate on the Labview program.

i. Person 1: Proceed opening the rest of the fuel valves in a counter-clockwise pattern from the last fuel valve opened, waiting five seconds between each opening.

j. Person 1: Once all fuel valves around the combustor are open and all flameholders are confirmed lit exit the rocket pit and proceed to control room.

7. Open main air regulator till the mass flowrate becomes constant (the air flowrate should make a sudden jump before reaching a constant). If the sudden jump does not occur before reaching 2 lbm/s reduce the pressure on the high pressure regulator and close main air ball valve to less than ¼ turn without completely closing it. Then increase the pressure from the high pressure regulator.

8. Adjust main air ball valve to desired mass flowrate (2.6 lbm/s for two CFM56 vane cassettes).

9. Add air to the premixer by opening the needle valve till either a blue flame exits the flameholders or the equivalence ratio inside the premixer is no less than 10. A high equivalence ratio is necessary to insure no flashbacks occur.

10. Begin increasing the fuel flowrate and premixer air flowrate, following the direction of step 9, so that the temperature exiting the turbine vanes rises slowly. It is recommended to increase no more than 200°F every one minute.

11. While increasing the fuel and premixer air monitor the main air mass flowrate.
12. When done slowly bring down the exit temperature in increments of 500 °F per two minutes until about 200 °F is reached, then after two minutes close fuel valve and flush premixer with premix air only.

13. Allow TuRFR to run with air only for 5 minutes before shutting main air ball valve.
APPENDIX C

PROCEDURE FOR FILLING GAS CYLINDER BANKS
1. Close the ball valve on south wall of rocket pit.

2. Check the valve on top of each cylinder to ensure that it is in the open position and is ready for filling.

3. Check that each ball valve on each cylinder bank’s manifold is open and ready for filling.

4. Open ball valve just upstream of the cylinder bank, located on the north wall of the rocket pit.

5. For each compressor to be turned on open its upstream and downstream isolation valves. Also, open the ball valve immediately downstream of the tee union for the two compressors.

6. For each compressor, switch its breaker ON, located outside on the east wall of the control room. Note: it may be necessary to switch the breaker ON located inside Room 191.

7. Turn on compressor.

8. Walk around cylinder bank and listen for leaks at all connections.

9. If a leak is heard (an audible hissing noise indicates a leak) or a methane sensor indicates unsafe level immediately turn off compressor, locate the leak, and isolate it by closing cylinder and ball valves. Then follow procedure for replacing a cylinder/repairing a connection.

10. When the pressure in the cylinder bank reads 3400 psig turn compressor off and switch its breaker to OFF.

11. Close all valves located outside near the compressors.
APPENDIX D

PROCEDURE FOR REPLACING A CYLINDER / REPAIRING A CONNECTION
1. Make sure compressors are OFF.
2. Close the valve on top of every cylinder.
3. Recheck all cylinder valves to assure they are in the closed position.
4. Evacuate the entire gas line so all pressure gauges read 0 psig.
5. Close all ball valves in the gas line.
6. Remove cylinder/connection and make necessary repairs.
7. Reinstall cylinder/connection and check connection for integrity.
8. If any cylinders are pressurized begin the following one at a time:
   a. Open one pressurized cylinder’s valve.
   b. Check manifold pressure gauge.
   c. Wait several minutes and recheck manifold pressure gauge to verify pressure is constant (ie. No leaks).
   d. Use leak check liquid (ie: soapy water; leaks are indicated by bubbles) on cylinder/connection that was installed to assure no leaks are present.
9. If no cylinders are pressurized:
   a. Follow procedure for filling cylinder bank, but after the compressor is turned on check for leaks at the recently installed cylinder/connection with leak check liquid.
APPENDIX E

MATLAB SCRIPT FOR CALCULATING PARTICLE ENTRAINMENT TIME
% Code evaluates the time and distance for a particle to reach 95% of 
% the fluid velocity.

clear all
close all
clc
format short g

Mach = 0.25;  %Mach number at vane leading edge

%English Unit Inputs
dia = 4.733;  %Equilibration tube ID (in)
diamet = dia*.0254;  %Equilibration tube ID (m)

%Metric Unit Inputs
d = 20e-6;  %Particulate diameter (m)
mu = 530.0e-7;  %Dynamic viscosity (kg/s/m)
rho_p = 2900;  %Particulate density (kg/m3)
m = rho_p*pi*d^3/6;  %Particle mass (kg)
gamma = 1.3;
p = 202600;  %Absolute pressure at vane leading edge (Pa)
R = 287;  %Gas constant for air (J/kg/K)
T = 1473;  %Air temperature (K)
g = 9.81;  %Gravity effects (m/s2)

rho_f = p/(R*T);  %Fluid density at vane leading edge (kg/m3)
Area = pi*diamet^2/4;  %Equilibration tube cross-sectional area (m2)
mdot = rho_f*Area*Mach*sqrt(gamma*R*T);  %AARL required mass flowrate (kg/s)

vf = mdot/(rho_f*Area);  %Air velocity at vane leading edge (m/s)

vp(1) = 0;  %Particulate initial velocity at cone entrance (m/s)
accel(1) = 0;  %Initialize acceleration (m/s2)
dt = 0.0000001;  %Time increment for numerical solution (s)
t(1) = 0;  %Initialize time (s)
v_max = 0.95*vf;  %Maximum velocity of particles (m/s)
x(1) = 0;  %Initialize position (m)

n = 1;  %Array index
while vp(n) <= v_max;  %Stops when particles reach maximum specified velocity

Re = d*(vf-vp(n))*rho_f/mu;  %Reynolds number of particle in fluid
w = log10(Re);
if Re <= 0.01;
    Cd = 9/2+24/Re;
elseif Re > 0.01 && Re <= 20;
    Cd = 24/Re*(1+0.1315*Re^(0.82-0.05*w));
elseif Re > 20 && Re <= 260;
    Cd = 24/Re*(1+0.1935*Re^0.6305);
elseif Re > 260 && Re <= 1.5e3;
    Cd = 10^((1.6435-1.1242*w+0.1558*w^3);
elseif Re > 1.5e3 && Re <= 1.2e4;
    Cd = 10^(-2.4571+2.5558*w-0.9295*w^2+0.1049*w^3);
elseif Re >1.2e4 && Re <= 4.4e4;
end

125
Cd = 10^(-1.9181+0.6370*w-0.636*w^2);
elseif Re > 4.4e4 && Re <= 3.38e5;
    Cd = 10^(-4.3390+1.5809*w-0.1546*w^2);
elseif Re > 3.38e5 && Re <= 4e5;
    Cd = 29.78-5.3*w;
elseif Re > 4e5 && Re <= 1e6;
    Cd = 0.1*w-0.49;
else
    Cd = 0.19-8e4/Re;
    fprintf('Reynolds Number is very high...check inputs\n');
    Re
end;
Drag = Cd*pi*rho_f*d^2*(vf-vp(n))^2/8; %Drag on particle (N)
a = Drag/m-g; %Gravity effect is included (m/s^2)
accel(n+1) = a; %Acceleration array (m/s^2)
disp(['Particle Velocity (m/s) = ',num2str(vp(n))]); %Prints particle velocity (m/s)
vp(n+1) = vp(n)+a*dt; %Calculates new particle velocity (m/s)
t(n+1) = t(n) + dt*1000; %Time array (ms)
x(n+1) = x(n) + vp(n)*dt+a*dt^2/2; %Position array (m)
n = n+1;
end;
mach_p = vp/sqrt(gamma*R*T);
disp(['Equilibration Tube Length (in) = ',num2str(x(n-1)/.0254)]);
disp(['Time for particle to come to speed (ms) = ',num2str(t(end))]);

figure(1)
plot(t,mach_p)
title('Particle Mach Number vs Time')
xlabel('Time (ms)')
ylabel('Mach Number')
axis tight

figure(2)
plot(t,x)
title('Particle Position vs Time')
xlabel('Time (ms)')
ylabel('Position (m)')
axis tight

figure(3)
semilogy(t,accel)
title('Particle Acceleration vs Time')
xlabel('Time (ms)')
ylabel('Acceleration (m/s^2)')
axis tight

figure(4)
plot(x/.0254,mach_p)
title('Particle Mach Number vs Position')
xlabel('Position (in)')
ylabel('Mach Number')
axis tight