Gas Turbine Governing Dynamics and Control Systems

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Kyle Smith

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AUTHOR: Kyle Austin Smith

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Dr. Shaun Kelly, Project Advisor

Dr. Richard Cavaletto, Department Head
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ABSTRACT

This project includes discussion of the governing dynamics of gas turbines employed for aviation propulsion and electrical generation systems. A small gas turbine in a ground power unit configuration was acquired from the United States Air Force via Avon Aero Supply Inc. in Danville, Indiana. The turbine was overhauled and reconfigured with a more modern control system which allowed for throttling of the turbine and real time measurement of critical operational parameters.

Primitive testing was conducted to ensure proper operation of the throttling and data gathering functions.
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LIST OF FIGURES

Figure 1. Boundary Layer Properties (Chen, 2010) ................................................................. 4

Figure 2. Velocity profile of boundary layer. Left: Normal profile; Center: Optimally deflated profile at maximum area ratio; Right: Area ratio too great, flow reversal occurs. (Wilson, 1998) ....................................................................................................................................................... 5

Figure 3. C-17A engine 3 experiences a compressor stall during reverse thrust (Mallinson, 2006) ...................................................................................................................................................... 8

Figure 4. GPU unit as received from Avon Aero. ...................................................................... 10

Figure 5. Compressor wheel with carbon buildup. ................................................................. 11

Figure 6. Gearbox after priming. ........................................................................................ 11

Figure 7. Bolt shanks being removed from turbine housing. .................................................. 12

Figure 8. Turbine housing and gearbox during secondary assembly ........................................ 12

Figure 9. Completed PLC installation .................................................................................. 13

Figure 10. Turbine on completed test stand .......................................................................... 14

Figure 11. Turbine performance at start up using data located in Appendix D ...................... 16

Figure 12. NI Lookout screenshot during steady-state operation .......................................... 17
LIST OF TABLES

Table 1. Excel test run data. .............................................................................................................31
INTRODUCTION

Turbomachinery is classified as those devices which produce changes in enthalpy via fluid-dynamic lift (Korakianitis, 1998). Within this broad definition, there are two categories of turbomachinery, work producing and enthalpy producing devices. In all applications mechanical energy is recovered via a turbine, whereas some applications require an enthalpy increase prior to combustion, and thus include a various number of compressor stages. It should be noted that turbine may refer to either the turbomachine as a whole, a section of the turbomachine, or a blade component within a section (Aungier, 2006). This text will only concern itself with the design and governing dynamics of open-cycle gas turbines as utilized by the aerospace industry for thrust production and power generation.

The modern-day gas turbine operates on a thermodynamic cycle known as the Brayton Cycle. This cycle is composed of four processes applied in a specific order: compression, combustion, expansion, and evacuation (Bailey, 2010). Ideally this would be an isentropic and isobaric cycle; however, losses due to viscous flow and thermal radiation generate entropy which makes the isentropic case an impossibility in even the most efficient turbines.

Operational demands in the aerospace industry require a level of design and evaluation which is unmatched in most industries. In the course of a normal commercial flight, the turbine will experience a working fluid temperature variation of 150 degrees Fahrenheit or more coupled with a 400 percent change in density. Temperature variations of this magnitude will produce dimensional variations which may promote unsatisfactory flow conditions. Additionally, changes in fluid density may instigate local flow disruptions which can result in decreased service life or even catastrophic failure of the power plant; case-in-point axisymmetric stall (Kerrebrock, 1992).

Recently the aerospace industry has seen implementation of new departure procedures which save fuel. Pioneering a new departure throttle program, Airbus and Singapore Airlines have developed a procedure for departing London’s Heathrow International which reduces stress on turbine components, reduces noise for nearby residents, and saves the airline an estimated 300kg of fuel per departure (Kingsley-Jones, 2010). The turbine control module (TCM) program alteration now allows pilots to select a FLEX (TCM controlled throttle setting) throttle setting which allows the computer to control turbine throttle as well as aircraft orientation to achieve maximum climb rate in the most fuel efficient manner.

Today gas turbine design requires complex fluid analysis which can only hope to estimate machine component and working fluid interaction at best. With a better understanding of the interaction between fluid and machine components one can more adequately design a control system which will allow for more efficient operation and increased service life. The intent of the this project is to more fully understand the governing dynamics of turbines, and using the knowledge, program a custom TCM for a small gas-turbine.
LITERATURE REVIEW

Fundamentals

A sufficient understanding of the movement of working fluid in the rotating frame of reference is requisite for discerning any thermal or mechanical process in the gas turbine cycle. Particulate flow will be described in four dimension Euclidean space with the fourth parameter being defined as time. The relation between the absolute velocity \( \vec{v} \) and the relative velocity \( \vec{w} \times \Omega \times \vec{r} \) is as follows:

\[
\vec{v} = \vec{w} \times \Omega \times \vec{r} \tag{1.0}
\]

\[
\vec{w} = \sum_{n=1}^{3} w^n \hat{e}_n \tag{1.1}
\]

The angular velocity \( \vec{\Omega} \) and position vector \( \vec{r} \) parameters express the movement of the rotating frame of reference (Wilson, 1998). For flow fields which are irrotational in the absolute frame of reference it follows that the streamlines cannot be irrotational in the rotational reference frame as shown in Equations 1.2 and 1.3 (Kruyt, 2009).

\[
\nabla \cdot (\vec{\Omega} \times \vec{r}) = 0 \quad \nabla \times (\vec{\Omega} \times \vec{r}) = 2\Omega \tag{1.2}
\]

\[
\nabla \cdot \vec{w} = 0 \quad \nabla \times \vec{w} = -2\Omega \tag{1.3}
\]

When assessing flows between two blade cascades, or even in a single volume between two blades in the same cascade, it becomes necessary to calculate the divergence and curl of the vector field in order to understand the complete interaction between the fluid and control surface. Given the field \( \vec{v} \) defined by Equation 1.4, the divergence 1.5, and curl 1.6, can be calculated accordingly at any given point within field boundaries.

\[
\vec{v} = [v_x(x, y, z, t), v_y(x, y, z, t), v_z(x, y, z, t)] \tag{1.4}
\]

\[
\nabla \cdot \vec{v} = \frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} + \frac{\partial v_z}{\partial z} \tag{1.5}
\]

\[
\nabla \times \vec{v} = \begin{vmatrix}
\frac{\partial v_z}{\partial y} - \frac{\partial v_y}{\partial z} \\
\frac{\partial v_x}{\partial z} - \frac{\partial v_z}{\partial x} \\
\frac{\partial v_y}{\partial x} - \frac{\partial v_x}{\partial y}
\end{vmatrix} \tag{1.6}
\]

Aerothermodynamics

The operating principles of any open-cycle gas turbine can be described by the Brayton Cycle which contains four processes operating at a steady state. Initially a working fluid is isentropically compressed to increase internal energy (Chen, 2010). Immediately following compression, the fluid experiences isobaric expansion allowing all kinetic energy to be
utilized as static pressure. The high pressure fluid is mixed with a combustible liquid and ignited. Post-combustion, the fluid contains both high kinetic and pressure energies which are exploited to impose angular acceleration to the turbine blades. Passing through multiple stages of blades, much of the fluids kinetic and pressure energies have been lost. The final process of isentropic expansion, will allow the fluid to approach atmospheric conditions with respect to velocity and pressure only (Chen, 2009).

This thermodynamic cycle in its ideal form is isentropic, however due to the nature of viscous flow and entropy generated by the interaction between the working fluid and control surfaces, this ideal case becomes an impossibility.

Rothalpy ($I$) is a thermodynamic value derived from Bernoulli’s Equation (1.7), and describes the energy of the working fluid relative stagnation pressure, stagnation enthalpy, and blade tip speed (Kruyt, 2009).

\[
\frac{\partial \phi}{\partial t} + \frac{1}{2} \hat{\mathbf{w}} \cdot \hat{\mathbf{w}} + \frac{p}{\rho} - \frac{1}{2} (\hat{\mathbf{\Omega}} \times \hat{\mathbf{r}}) \cdot (\hat{\mathbf{\Omega}} \times \hat{\mathbf{r}}) = c(t) \tag{1.7}
\]

\[
\phi = \frac{\hat{\mathbf{v}}}{\hat{\mathbf{V}}} \tag{1.8}
\]

The subscript $R$ denotes the velocity potential $\phi$ is calculated with respect to the rotating frame. In cases where a vaneless diffuser is present, the “free impeller assumption” can be made, where a lack of influence of static components on dynamic components will result in a uniform field for the velocity potential.

\[
I = \frac{p}{\rho} + \frac{1}{2} \hat{\mathbf{w}} \cdot \hat{\mathbf{w}} - \frac{1}{2} (\hat{\mathbf{\Omega}} \times \hat{\mathbf{r}}) \cdot (\hat{\mathbf{\Omega}} \times \hat{\mathbf{r}}) \tag{1.9}
\]

In cases where a steady flow exists, rothalpy will be a constant value between the leading and trailing blade edges; however, this is not the case for incompressible and unsteady flows (Kruyt, 2009).

Fluid Mechanics

Typically, flow over a blade surface is viscous, and is associated with a Reynolds number in excess of $10^5$. Due to the high Reynolds number, the boundary layer is assumed to be thin. However, the velocity profile from inviscid flow to control surface has a large gradient, and therefore, friction stresses induced in the working fluid due to friction cannot be neglected. In order to quantify the inefficiency of flow, the principles of the boundary layer theorem must be applied (Wilson, 1998). Figure 1.0a illustrates a typical velocity profile.
Using the notation convention of Figure 1.0, boundary layer properties such as displacement thickness ($\delta^*$), momentum thickness ($\delta^{**}$), and energy change ($\delta^{***}$) can be derived as follows in Equations 1.10, 1.11, and 1.12 respectively. Those variables associated with a subscript ‘e’ are fluid properties taken at atmospheric conditions (Wilson, 1998).

$$
\delta^* = \int_0^\delta \left(1 - \frac{\rho W}{\rho_0 W_e}\right) \, dy \tag{1.10}
$$

$$
\delta^{**} = \int_0^\delta \frac{\rho W}{\rho_0 W_e} \left[1 - \frac{\rho W}{\rho_0 W_e}\right] \, dy \tag{1.11}
$$

$$
\delta^{***} = \int_0^\delta \frac{\rho W}{\rho_0 W_e} \left[1 - \left(\frac{\rho W}{\rho_0 W_e}\right)^2\right] \, dy \tag{1.12}
$$

Often designers strive to minimize pressure loss across a cascade so as to avoid a flow regime change. The shape factor $H$ is defined by Equation 1.7.

$$
H = \frac{\delta^*}{\delta^{**}} \tag{1.13}
$$

As $H$ nears unity, the presence of an increasingly adverse pressure gradient diminishes Reynolds number, triggering a flow transition into the turbulent regime. A shape factor of 1.3 is characteristic of turbulent flow, and conversely a factor of 2.3 is typical of laminar flow conditions (Wilson, 1998).

**Diffusion**

Diffusers play an important role in any open-cycle gas turbine, manipulating the internal energy of the working fluid in order to achieve a more efficient cycle. Present day design convention employs diffusers immediately following the final compression stage, as well as after the final turbine cascade.
Diffusers utilize a change in cross sectional area, known as the area ratio, to deflate the velocity profile as seen in Figure 2. In an ideal case the working fluid would be allowed to expand at the maximum possible rate, and would reach stagnation pressure (Wilson, 1998). Using equation 1.14, the gradient of the velocity profile near the wall boundary can be calculated. It follows that in a steady-state system the maximum rate of expansion can be arrived at by assuming the angle is normal to the wall face and solving for the required gradient. However, in reality the flow field is unstable, and so the most efficient pressure recovery occurs while the gradient is nearly normal to the shroud face. Momentary boundary layer breaks will occur with flow reversals traversing the length of the diffuser wall (Wilson, 1998).

\[
\tan \alpha_w = \left( \frac{\partial c}{\partial y} \right)_w
\]

\[
\tau_w = \mu \left( \frac{\partial c}{\partial y} \right)_w
\]  

(1.14)  

(1.15)

The shear in the working fluid at any point in the velocity profile can be solved by multiplying the velocity gradient by the dynamic viscosity term as seen in Equation 1.15.

There are two main configurations of diffusers utilized to stabilize the compressed fluid flow prior to combustion, vaned and vaneless. Although the lower solidity of the vaneless configuration allows for a greater flow range, vaned diffusers generally generate a higher static pressure and more stable flow field (Mansoux, 1994). The vaned diffuser delivers a higher static pressure to the combustion section by utilizing a radial array of aerodynamically shaped vanes. These vanes are oriented nearly normal to the blade lean at the compressor's circumference. By redirecting flow, kinetic energy is remanifested as static pressure. However, the additional surface area and redirection of flow induces and efficiency loss due to friction and rothalpy. Conversely, the lower solidity of the vaneless configuration also allows for the free impeller assumption to be made. Consequently rothalpy, as given by equation 1.9, will remain constant as it passes through the diffuser, and fluid rotation with respect to the rotating frame of reference will be zero (Chen, 2010).
Additionally, for the design of a high-efficiency vaned diffuser, research indicates the innermost radii of the vanes should be located no closer than five percent of the compressor radius from the compressor’s circumference to allow for wake mixing and flow stabilization (Wilson, 1998). This result is further supported by Pinarbasi (2008), where hot-wire anemometer measurements in a vaned diffuser produced quantitative results as to the sensitivity of diffuser performance relative blade space ratio. In diffusers whose vanes leading edges were nearest to the impeller vibrations, noise levels, and Mach numbers at the vanes leading edge all increase.

**Blade Structure and Composition**

The blades of both the turbine and compressor sections consist of highly refined airfoil cross sections which allow for the efficient transfer of energy between its mechanical and fluid states. As with any non-payload item in the aerospace industry, the turbine as a whole must comply with weight restrictions in order to increase the operational payload of the aircraft (Carter, 2005). Additionally the cross sections must be structurally sound to cope with mechanical, aerodynamic, and thermal loading.

Turbine blades regularly see temperature variations of a thousand degrees or more in a single operational cycle (start up to shut down), whereas relatively little temperature variation is seen during steady state operation. However, compressor blades experience a much wider range of operational temperatures determined by which blade cascade they are installed. The turbofan and initial stages of the compressor will often see temperatures similar to ambient temperature which can range from moderate to warm on the ground, and as little as -50°C at altitude (Carter, 2005). However, the ladder parts of the compressor will experience much higher operational temperatures relative to ambient temperature due to heat added to the working fluid during the compression process (Carter, 2005).

Turbines are designed with tight tolerances between the blade tips and shroud in order to help each blade cascade achieve a higher efficiency. This tight tolerance limits the amount of elongation a blade may experience due to centrifugal forces during operation in order to prevent contact with the shroud. These forces are greatest during takeoff when a turbofan blade may see as much as 90 metric tons of centrifugal loading (BBC, 2010). This cyclic loading scheme results in a condition known as creep. Although the applied centrifugal force is generally stable under normal operational conditions, the magnitude of the force and time duration leads to permanent deformation of the element. This deformation will eventually lead to blade tip contact with the shroud known as rubbing (Carter, 2005). When this occurs, the blade may be trimmed or replaced, and shroud inspected for permanent structural damage.

Aerodynamic loading also drives the structural design of both turbine and compressor blades. The loading imposed on any given blade is a function of the pressure gradient across the blade cascade either as increase in pressure in the compressor, or a decrease in pressure in the turbine (Carter, 2005). These forces can be decomposed into lift and drag force components. The lift force generated by an airfoil is given by the equation $1.16$ where: $\rho$ is air density, $v$ is the true airspeed, $A$ is the profile area of the wing as seen from above or below, $C_L$ is the coefficient of lift. This coefficient varies with the angle of attack, Mach number, and Reynolds number (Anderson, 2004).
\[ L = \frac{1}{2} \rho v^2 A \; C_L \]  
(1.16)

\[ C_L = \frac{l}{\rho v^2 c} \]  
(1.17)

The lift force will act along an axis perpendicular to the direction of rotation, whereas the drag force will act in a direction inversely collinear to the direction of rotation and is given by the following equation.

\[ F_d = \frac{1}{2} \rho v^2 A \; C_D \]  
(1.18)

The density of the fluid is denoted as \( \rho \), the velocity as \( v \), and the cross sectional area as seen normal to the direction of fluid flow as \( A \). The coefficient of drag \( C_D \) is a dimensionless number and a function of fluid properties, Reynolds number, and Mach number (McCormick, 1979).

While understanding physical forces acting on a blade during operation determine its structural properties, the operational environment plays a major role in determining the material from which the blade will be manufactured. Foreign object damage (FOD) is a principle reason for component replacement during turbine maintenance (Carter, 2005). At altitude a turbofan or compressor blade can come in contact with moisture which is at a sub-freezing temperature. This moisture freezes on contact with any surface, and leads to an abrasive erosion of the blade material if operation is continued in these conditions (Carter, 2005). The ingestion of larger FOD such as foul or tools left near a turbine intake have also lead to the failure of turbine components. Ingestion of any FOD in large enough quantities can stall the compressor and lead to a “flame-out” of the turbine. As such modern turbines are tested as a part of certification for water, ice, and foul ingestion. Modern day turbofans are designed to accelerate larger FOD outwards through the bypass duct so as to minimize flow disturbance into the compressor, and damage to any components (Carter, 2005). FOD damage is primarily seen in compressor and turbofan cascades. However, turbine blades are also damaged by carbon buildup on combustion injectors which is released into the cascade during operation, or by ceramic thermal coatings which erode from the walls of the combustor (Carter, 2005).

Many turbine components are composed of high Nickel super alloys due to their superior mechanical strength, and ability to form chromium based coatings through chemical reactions which act as protective films (Carter, 2005). Compressor blades may see corrosion due to oxidizing agents contained in the atmosphere either introduced by industrial exhausts, or naturally occurring processes. Turbine blades are also highly susceptible to corrosion by tetra-ethyl lead which is contained in some fuels used in aviation. As a result, these fuels are used sparingly (Carter, 2005). Manufactures strive to protect components against nickel corroding elements by treating turbine components with exotic coatings which contain Yttrium and Platinum (Carter, 2005).
Aerodynamic Instabilities

During initial design, general solutions assume steady state operation with consistent flow fields. In reality, flow fields are irregular requiring more refined solutions for compressor and turbine operating ranges. Instabilities such as rotating and axisymmetric stall arise from these flow perturbations (Burguburu, 2010). These two conditions, previously believed to be unrelated, have now been shown to be mathematically linked. Research produced by the Massachusetts Institute of Technology and French researchers has been able to model these conditions relatively accurately.

With any airfoil design, certain conditions arise where stall is inevitable. Compressor and turbofan blades are no different, and often exhibit flow behavior based on both up and downstream conditions. Rotating stall is an nth order limit cycle oscillation which is initialized by flow disruption which stalls an airfoil(s) of the compressor (Mansoux, 1994). Localized regions of stall are known as a stall cells, and one or multiple cells may be present. Cells may transverse the face of the compressor cascade in a radial fashion at frequencies varying from the rotational frequency of the blades (Burguburu, 2010). This region of stall destabilizes the compressor greatly degrading compressor performance and efficiency; propagating non-uniform flow to downstream cascades and or diffusers.

If flow conditions are such that the limit cycle oscillation becomes divergent, the stall will propagate across the compressor face inducing an axisymmetric stall (Mansoux, 2010). When an axisymmetric stall condition occurs, flow through the cascade is momentarily suspended allowing flow reversal to occur through the compressor and combustor sections (Kerrebrock, 1992). This form of compressor stall may also occur when the compressors pressure rise capabilities are exceeded. Flame cans in the combustor section contain small perforations which allow a controlled flow of air for combustion. However, they also act as throttling devices downstream of the compressor (Mansoux, 1994). If the combustion rate is decreased rapidly, as is the case during a rapid decrease in turbine throttle, pressure immediately upstream of the flame cans will exceed the compressors operational abilities and induce a flow reversal (Kerrebrock, 1992).

![Figure 3. C-17A engine 3 experiences a compressor stall during reverse thrust (Mallinson, 2006).](image-url)
It should be noted that these stall conditions induce varied cyclic loading on the cascade(s), and in the case of axisymmetric stall induce violent flow conditions which can cause component failure and “flame out” due to a momentary vacuum in the combustor section. Once a favorable pressure gradient is restored the compressor may continue to operate normally. However, if the flow conditions which initially induced the stall persist, the stall will become self-reproducing, and can produce violent harmonic oscillations which will lead to a catastrophic failure of the turbine (Kerrebrock, 1992).
Acquisition and Initial Condition

In the spring of 2011 a Garrett AiResearch GTP30-51 turbine was sourced from Avon Aero Supply Inc. in ground power unit (GPU) configuration. Although little was known about the specific condition of the unit, the turbine wheel did spin freely and the unit had been operated for 2700 hours since last overhaul.

Debris had collected in the fuel lines and consequently obstructed fuel flow completely at the main fuel cutoff valve. The fuel tank was drained, and all steel fuel lines removed and checked thoroughly for foreign debris. The fuel booster pump located in front of the fuel filter assembly was removed and disassembled to be cleaned. As well the main fuel filter was drained and checked for particulate buildup.

After the fuel system was cleared of any particulate, the oil level and electric connections were checked prior to the first start. The gear case contained the proper amount of oil which was free of metal shavings and any contaminants. Although there was evidence of minor rusting on some of the larger electrical connections, all connections were solid and backing nuts were tight so the decision was made to attempt a system start.

The electrical system was connected to a larger Caterpillar articulated loader utilizing a set of heavy duty jumper cables, and the fuel tank was filled with one gallon of 87 octane gasoline. The primer pump took approximately 30 seconds to boost pre-start fuel pressure as observed by an audible clicking from the pump. When the starter relay was closed the turbine spun up as expected, and exhaust gas temperature (EGT) started to climb. After
approximately 45 seconds EGT reached 600°F and subsequently 900°F after two minutes. The starter audibly disengaged from the gear set, and turbine throttled up to 100% at 2:41. At full throttle the engine appeared to be running smoothly and maintaining a constant EGT. When the generator contact was closed gauges indicated an output of 120V at 400 Hz.

Turbine Overhaul

Although there were no indications of adverse operating conditions during start, steady state operation, or shutdown, the prolonged start time was cause for concern. After consulting USAF T.O. 35C2-3-366-2 the decision was made to disassemble the turbine for overhaul. The manual indicated that prolonged starting periods were significant of carbon buildup on control surfaces and injector nozzles, or a misconfigured fuel control assembly.

The turbine was removed from its housing, and overhaul began in fall of 2011 at California Polytechnic State University. After complete disassembly each part was inspected for defects in order to avoid catastrophic failure in high speed components during further operations. Gearbox components were free of defects and spun freely.

The compressor wheel was coated with a relatively thin film of carbon. No defects of any kind were noted, and the wheel was decarbonized with an appropriate non-chlorinated chemical. The turbine wheel was free of any defects as well, and contained no particulate buildup of any kind, and as a result was not unchanged.

Significant amounts of corrosion coating were missing from the compressor inducer housing as well as the gearbox. As a result, the gearbox halves were reassembled, and covers machined for any orifices. The gearbox and compressor housing were media blasted with #XX glass beads to remove all coatings and expose the aluminum and titanium surfaces. Both assemblies were wiped down with a moist rag to remove and loose particulate from the media blasting process. The gearbox was then treated with surface conversion chemical, Alodine 1201, to prepare the aluminum for painting. The gearbox and compressor housing received three initial coats of zinc chromate as a base primer as seen in Figure 6. This was followed by two coats of Temp A150 epoxy based corrosion coating, each coat
separated by an hour to allow proper drying.

During disassembly several of the air plenum assembly sheared off due to their threads seizing in the turbine housing. Although no official overhaul was on record, it was clear a copper based anti-seize had been applied to these threads as some point which led to the thread corrosion. As a result a total of 6 bolt shanks had to be precision drilled out of the air plenum and turbine housings and holes rethreaded as seen in Figure 7.

Prior to reassembly all gear surfaces and bearings were flushed with mineral spirits to ensure no debris had contaminated them during their exposure. Additionally the oil filter and pump were inspected for particulate intrusion and cleaned thoroughly although no irregularities were noted.

Initial Reassembly

The turbine was reassembled as per USAF manual instructions. New AN-style stainless steel fasteners were purchased from a local aerospace components distributor, and torqued to proper specifications. The original control equipment was affixed to the turbine, and the power plant was reinstalled in the original GU housing. Once all electrical connections were made and gearbox filled with two quarts of Aeroshell 308, which meets MIL-PRF-7808L specifications, the turbine was tested.

The electrical system was connected to two 12V batteries in series which offered 223 cold cranking amps. The turbine spooled up without issue and reached full throttle in 1:30 on kerosene. However, a consistent stream of white smoke from the exhaust and fluttering indicated severe problems. Upon further investigation in subsequent starts the cause of the flutter was determined to be a stream of oil from the main shaft bearing at the face of the compressor; oil was freely flowing into the intake producing a self-reproducing compressor stall.
Secondary Disassembly and Reassembly

The turbine was once again removed from its housing and completely disassembled to determine the cause of the leak. A mechanical shaft seal which is located between one of the two main shaft bearings and the compressor face was found to be excessively worn. Additionally, the carbon seal material contained a hairline fracture which compromised the seal. This fracture was most likely the result of debris intrusion during the initial disassembly. A new seal was sourced from Alamo Aircraft Ltd. The new seal was installed in relatively clean conditions using latex gloves. Special care was taken during the secondary reassembly to ensure proper seal seating so as to avoid reproducing a seal failure.

The remainder of the reassembly and turbine installation was unremarkable. Four complete start and shut-down cycles were completed without incident and the issue was considered resolved.

Control System Modifications

The original control system was primitive in that it allowed for turbine operation at full throttle, and altering of the output electrical signals. Although sufficient for a GPU, this project required throttling of the turbine and real-time measurement of various parameters.

A new control system was designed around a Koyo Direct Logic 06 series programmable logic controller (PLC) donated by Automation Direct. Four modules were added for additional measurement capabilities. A high speed counter module capable of counting pulses up to 100 kHz was utilized to track turbine speed. An analog input module was used to measure signals from three separate pressure transducers as well as a potentiometer utilized for throttle control. An analog output module was tied to the analog input channel which read potentiometer position. The module would increase or decrease current output accordingly in order to change the displacement of the cartridge valve utilized for fuel flow control. A thermocouple module was also utilized to measure resistance of the thermocouple circuit. The thermocouple circuit is most critical since exhaust gas temperature load is a good measure of turbine load and combustion conditions.

All instrument measurements would be read in real time via National Instruments Lookout. However, several inputs were programmed into the PLC unit as fail op criteria. If the thermocouple measured any temperatures exceeding 300°C, regardless of the status of the burnout indication bit, the fuel circuit would be opened causing

Figure 9. Completed PLC installation.
the normally-closed style cartridge valve to close immediately. Additionally, if the oil pressure ever fell below 30 PSI or the emergency stop circuit was activated the same series of events would occur.

Initially the PLC was connected to 24V across two automotive batteries wired in series using 18AWG wiring. However, during testing it was discovered that the starter would draw enough power from the circuit to de-energize the PLC. Two more 12V 7Ah batteries were added in series on a dedicated PLC circuit so as to isolate the PLC power supply from heavy loads such as the starter and igniter which have larger current draws.

Test Stand Construction

The original housing was in poor condition so a simple test stand was constructed to mount the turbine, new control system, and provide a volume to store fuel and batteries. The entire structure was composed of 10 gauge mild steel. The axles were machined from 1” round stock to accommodate the prefabricated wheel hubs. After initial welding was completed, the tank was pressurized to 4 PSI and welded joints were tested for leaks using soapy water.

The turbine was mounted on the cart using brackets modeled after the original mounting hardware. New fuel intake lines were fabricated from #6 steel tubing. The fuel booster pump was relocated to mount horizontally on the chassis so some bolts were easier to access for future maintenance. Fuel return lines from the cartridge valve body and high pressure fuel pump were routed back to tank with #4 steel tubing.

Figure 10. Turbine on completed test stand.
RESULTS

Issues Prior to Starting

Before the starter was engaged live data was read from the PLC unit using National Instruments Lookout 6.1 and a serial connection. Despite having an onboard calibration transistor installed on the thermocouple unit, the thermocouple which measured EGT was not within an allowable tolerance for operation. Using a Fluke meter with thermocouple attachment and industrial heat gun, the thermocouple output was calibrated manually using a scalar value of 1.32. Additionally, all pressure and thermocouple values had to be divided by a factor of 10 so a correct value would be displayed in the HMI. These corrections were made as mathematical formulas in the HMI, and the raw values remained unchanged in the PLC logic.

When the HMI displayed what seemed to be satisfactory values for all parameters a starting cycle was attempted. The starter would engage and immediately disengage as confirmed both audibly and visually. The clutch engagement lasted a few milliseconds and was cyclical as long as the starter circuit was closed. After further investigation it was discovered the PLC would require an isolated power circuit for proper operation due to a high power drain by the turbines starter motor. An isolation circuit was implemented consisting of a 12VDC sealed lead acid (SLA) battery for full-time PLC operation, and the main batteries continued to power the starter, igniter, pressure transducers, fuel boost pump.

Initial Start

The first three starts were conducted utilizing kerosene as a fuel. The physical observation and data collected via Lookout indicated smooth starts in 20-30 seconds. There was however a lot of noise in the tachometer reading. The throttle percentage (%N1) regularly varied between 85% and 95%, although audibly there was no change in operation. Despite the 12VDC isolation circuit, the power draw from the 2 24VDC main relays which operated the igniter and starter were significant enough to once again deenergize the PLC during turbine operation. The power isolation circuit was expanded to include a second 12VDC SLA battery in series to provide the PLC with a full 24VDC of full-time power. This solved all issues during starting and run-up so additional testing could be performed.

Adjustments and Subsequent Testing

Due to the high cost of readily available kerosene, 87 octane gasoline was used in subsequent starts. Starting cycles with the new fuel would regularly occur in 20-30 seconds. Exhaust temperatures would hold steady at approximately 150°C, a value slightly less than operation on kerosene (200°C). Gasoline also produces smoke from the exhaust on shutdown, a condition not observed with kerosene. Even after the turbine wheel had come to a complete stop black smoke would be emitted from the exhaust diffuser and compressor in some cases. Briefly engaging the starter would eliminate the smoke, however this smoke would indicate excess fuel in the combustion chamber and an incomplete combustion cycle during operation.
Operational Data

The following data was collected during the second test start of the turbine operating on 87 octane gasoline.

![Turbine performance at start up using data located in Appendix D](image)

Figure 11. Turbine performance at start up using data located in Appendix D.

A table of corresponding values can be found in Appendix D. The tachometer data contains a significant amount of noise due to the absence of a data smoothing algorithm. According to the technical operations manual of this turbine the main shaft speed should be 56,000 ± 2,800 RPM depending on environmental conditions. So although the data indicates a wide range of variance (6,000 RPM), it is more likely the actual shaft speed was a more steady value lying somewhere between the indicated values of 47,000 and 53,000 RPM at full throttle.

The EGT values can be explained using simple dynamic principles. As the turbine spins up more fuel is added by the fuel control unit to provide the energy required to overcome the angular momentum as well as rotational acceleration. Once the turbine reaches full throttle and achieves steady-state operation the turbine wheel no longer requires acceleration, only enough energy to overcome angular momentum. It follows that the EGT values will drop directly correlating to a decrease in energy added to the system. After a minute of run-time a steady-state value without load of 150°C is reached.
The gear-driven oil pump, gear-driven fuel control unit, and N1 pressures are directly correlated to main shaft speed so it is logical that their data patterns mimic that of the RPM value.

**HMI**

This HMI simply displays what is necessary to operate the turbine in its simplistic test-stand configuration. Although in its current configuration the turbine can be operated without Lookout, the real-time data helps the operator ensure no parameters are out of range. In addition, the HMI allows the operator to shut-down the turbine if conditions which will lead to a failure of the power plant are imminent.

After using the CTRIO workbench to configure the high-speed counter module in DirectSoft 5, the RPM value did not require any further scaling or manipulation. The various pressure values seen are divided by a factor of 10 as previously discussed to represent actual values. The blue “%N1” value was a simple percentage calculation which would divide the RPM value by 56,000 then multiply by 100. The run time was a counter expression which would begin an up-count using the computer’s clock on an RPM value greater than 0, and reset to 0 on an RPM value of 0.

![Figure 12. NI Lookout screenshot during steady-state operation.](image-url)
DISCUSSION

Future Operational Recommendations

Although startup cycles on low octane gasoline were nominal, shutdown presented many problems. Prolonged spin down times and visible smoke from the compressor and turbine were cause for concern about the conditions in the combustion chamber even after the turbine wheel had ceased rotation. Due to time restrictions there are no clear conclusions as of yet as to why this problem persisted other than the chemistry of the fuel is not optimal for operation. As such, operation only on kerosene, JP-4, or diesel #2 is recommended as the aforementioned problems were no observed when operated on these fuels.

With regards to the tachometer housing, there was evidence of erosion in the housing due to a chemical reaction between the housing polymer and turbine oil. This erosion will undoubtedly lead to a total breakdown of the structure which holds the tachometer sensor in place. If this failure were to occur during normal operation the results could be devastating for both the operator and equipment. Therefore, the housing needs to be refabricated from a metal alloy which does not react with synthetic ester oil. The original mechanical housing was made from an aluminum derivative. Aluminum would most likely render the best results since it is non-ferrous and the potentiometer utilizes a magnetic field for operation. Tolerances in this housing are particularly tight however, so the dimensions listed in Appendix C for manufactured parts must be consulted prior to machining.

For more accurate readings of the rotational speed of the main shaft, the high speed counter module would require reconfiguration to incorporate a data smoothing algorithm. The more this value is smoothed in the CTRIO Workbench, the more the reading resolution is degraded. Since turbine speed is a critical reading, it is important to carefully balance resolution and noise to arrive at a solution which provides the operator with a reliable, readable number.

CONCLUSION

Reliable starting and operation of the turbine was achieved as well as real-time data logging via PC. Future works will hopefully allow for throttling capability as well as a more sophisticated HMI which will provide operational comparisons of different fuels. Alternative fuels could then be tested for efficiency and potential use in turbine based industrial power generation. Testing equipment could also be affixed to the unit to provide flow measurement values for the various fuels to help further understand power plant efficiency.
REFERENCES


APPENDIX A
HOW PROJECT MEETS BRAE MAJOR REQUIREMENTS
Major Design Experience

The BRAE senior project must incorporate a major design experience. Design is the process of devising a system, component, or process to meet specific needs. The design process typically includes fundamental elements as outlined below. This project addresses these issues as follows.

Establishment of Objectives and Criteria

Project objectives and criteria are established to meet the criteria outlined in the project contract. This project required participants to remove the antiquated turbine control system and replace it with a modern control system based on a programmable logic controller which would provide real-time data collection and statistics which were previously unavailable to the operator.

Synthesis and Analysis

Although this particular project required very few engineering calculations, understanding of engineering mathematics and electrical systems was requisite in order to create a control system which would provide a steady operational platform for the turbine.

Construction, Testing and Evaluation

The turbine was overhauled as per USAF manual procedures, the control system was designed and wired, and preliminary testing was conducted to ensure steady-state operation and repeatability could be achieved.

Incorporation of Applicable Engineering Standards

Although some manuals and data consulted when designing the system utilized IEEE and ISO hydraulic standards, no such standards are contained within this report.

Capstone Design Experience

The BRAE project is an engineering design project based on the knowledge and skills acquired in earlier coursework (Major, Support and/or GE courses). This project incorporates knowledge/skills from these key courses.

CE 204 – Mechanics of Materials I
CE 207 – Mechanics of Materials II
ME 211 – Engineering Statics
ME 212 – Engineering Dynamics
ME 302 – Engineering Thermodynamics
BRAE 216 – Fundamentals of Electricity
EE 321 – Electronics
EE 361 – Electronics Laboratory
BRAE 328 – Measurements and Computer Interfacing
BRAE 421 – Equipment Engineering I
BRAE 422 – Equipment Engineering II
Design Parameter Constraints

This project addresses a significant number of the categories of constraints listed below.

**Physical.** The completed unit must fit within the confines of a standard 8' truck bed.

**Health and Safety.** There is potential hazard in event of catastrophic turbine failure, however under normal running conditions the user is in no real danger.

**Political.** Potential for validation of new component designs which will lead to greater efficiency in general aviation and power generation applications.

**Other.** The turbine must be able to maintain steady-state combustion for at least one hour.
APPENDIX B
LADDER LOGIC
// LOAD TACHOMETER

## CTRIO Config

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// LOAD PRESSURE AND THROTTLE INPUTS

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// LOAD THROTTLE OUT

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LDA O3000
OUT V731
LD  K1
OUT V733
LD  K1
OUT V734
LD  K0
OUT V735
LD  K1
OUT V736
```

// WRITE THROTTLE VALUE TO THROTTLE OUTPUT

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LD  V435
DIV  K5
OUT V4035
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APPENDIX C
MANUFACTURED PARTS
16 SPLINES

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Table 1. Excel test run data.