



DESIGN AND TESTING OF A MICRO-SCALE WAVE ROTOR SYSTEM

THESIS

Kevin P. Lapp, Ctr.

AFIT-ENY-MS-17-S-067

**DEPARTMENT OF THE AIR FORCE
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THESIS

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Kevin P. Lapp, BS

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Abstract

The drone or micro-air vehicle industry has rapidly expanded in recent years. This thesis focuses on finding an alternative to power a Group 1 UAV with a maximum weight between 0 – 20 lbs. Nearly all of these vehicles are currently powered using battery-operated systems. While electrical power is currently the norm, the Air Force would like to pursue hydrocarbon-fueled aircraft for the higher energy to weight content. Normal internal combustion and turbine turbomachinery performs poorly at low mass flow rates required due to losses from a combination of factors. Wave rotors are being pursued as another option for powering these small-scale devices. At these small flow rates, wave rotors have been shown to suffer less losses while requiring a less complex geometry. Previous work has examined the performance of a through flow wave rotor producing around 1.8 kW. This effort is investigating a smaller design around 500 W. Currently it utilizes a wave rotor with an external burner to complete a Brayton-like cycle. The previously validated NASA Glenn Research Center quasi-one-dimensional Computational Fluid Dynamics code was used to explore a range of design options for the new 500 Watt wave rotor. Analysis and computer aided design work was completed to ensure rotor operation at the elevated temperatures and high rotational speed. Testing was completed to find overall compressor efficiencies for comparison with typical turbomachinery.

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Kevin P. Lapp

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Nomenclature

A	Area
a	Local Sonic Velocity
C	Discharge Coefficient
D	Outer Diameter
d	Throat Diameter
d^*	Hydraulic Diameter
h_c	Channel Height
ke	Kinetic Energy
L	Length
M	Mach Number
$\dot{M}$	Mass Flow Rate
N	Number of Reflected Pressure Waves
n	Number of Channels
P	Pressure
P_0	Stagnation Pressure
p	Goal Power
P_t	Total Pressure
R	Specific Gas Constant
Re_e	Reynolds Number
Re^*	Reynolds Number Based on CI's Sonic Velocity
T_s	Static Temperature
T_t	Total Temperature
t_p	Time for a Pressure Wave to the Channel Length
t_r	Residence Time
t_s	Total Time for One Rotor Revolution
V	Volume
v	Flow Velocity
Y	Expansion Factor
Υ	Ratio of Specific Heats
η_c	Compressor Efficiency
Ω	Angular Velocity
Θ_{0t}	Angular Distance to Fully Open a Port
ρ	Density
μ	Dynamic Viscosity
ω	Rotational Speed

List of Abbreviations

AFIT	Air Force Institute of Technology
AFRL	Air Force Research Lab
AI	Air In
AO	Air Out
ASME	American Society of Mechanical Engineers
BBC	Brown Boveri Company
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
CI	Compressor In
CO	Compressor Out
DERF	Detonation Engine Research Facility
DOD	Department of Defense
DMLS	Direct Metal Laser Sintering
EGR	Exhaust Gas Recirculation
EI	Exhaust In
EO	Exhaust Out
ESC	Electronic Speed Controller
GRC	Glenn Research Center
IC	Internal Combustion
MAP	Manifold Absolute Pressure
MAV	Micro Air Vehicle
NASA	National Aeronautics and Space Administration
PWS	Pressure Wave Supercharger
Q1D	Quasi-One Dimensional
RPA	Remotely Piloted Aircraft
RPM	Revolution Per Minute
TI	Turbine In
TO	Turbine Out
UAV	Unmanned Aerial Vehicle

DESIGN AND TESTING OF A MICRO-SCALE WAVE ROTOR SYSTEM

I. Introduction

I.1 General Issue

The drone or micro-air vehicle industry has rapidly expanded in recent years. This industry has expanded from the Remotely Piloted Aircraft (RPA) to both the commercial industry and the Department of Defense (DOD). The commercial industry utilizes these aircraft for aerial photography and for the delivery of packages. Similarly, the DOD employs these aircraft for high-endurance passive observation and ordinance delivery. This project explored the systems used to power the smallest classes of drones, Class 1 and Class 2. These classes consist of vehicles that weigh less than 25 kg. Vehicles of this size have been powered using electric motors, internal combustion (IC) engines, and miniature gas turbine jet engines. The electric motors suffer from low power density compared to the hydrocarbon burning engines. Both the IC engines and the miniature gas turbine engines experience losses as they are scaled down to power the small aerial vehicles. The IC engines are often times modified engines from civilian applications such as yard equipment. These small single cylinder engines encounter issues when operating at higher altitudes where the intake air has reduced pressure, temperature, and density. The miniature gas turbine engines can deal with the altitude changes, but encounter issues with the small complex geometries and high spin rates required. Overall, all three engine types face issues that require new technical developments or other drone power plant options.

The small IC engines' main area of concern is the drop in Manifold Absolute Pressure (MAP). This pressure reduction has a direct loss in crankshaft horsepower and torque. To combat these losses, a variety of options have been tried including aircraft specific designs and turbo-normalization with turbochargers or superchargers. Specific designs for each aircraft are costly and not worth the investment based on the wide range of micro-air vehicles. Furthermore, specific designs add complexity where an engineer would be needed for each unique system. The use of turbo chargers or superchargers does increase the MAP, but adds weight and complexity to the small systems. The weight and complexity reduces the overall efficiency of the power system.

The small gas turbine engines can handle the reduction in pressure, temperature, and density, but encounter issues as the geometries are scaled down. Most of these issues stem from the complexity and geometric issues faced by the turbomachinery. This turbomachinery has been developed to its efficiency limits and has few options for any improvements. The low mass flow rates associated with the small sizes further reduces the engines performance as the fluids properties change. As the fluid mass flow rate declines, compressor efficiency plummets. The combination of turbomachinery issues and fluid properties limits the operating range and applications of small gas turbine engines.

Small engines can also be use in combination with each other to power larger vehicles. The use of distributed power has many benefits for the range of micro-air vehicles pursued by the DOD. With proper integration, a small engine could power multiple classes of vehicles based on the number of engines used. This method reduces

the amount of engine designs required and simplifies the repair and maintenance of the aircraft. Distributed power is not limited to aircraft and has potential across all industries.

An alternative engine type called a wave rotor could be the solution for replacing the low power density electric motors, poor altitude performing IC engines, and complex geometries of small gas turbine engines. Wave rotors or pressure wave superchargers are a unique method of energy exchange. Instead of using rotors and stators (gas turbine engine) or a piston (IC engine), a cylindrical drum rotates between four separate ports. This drum serves as both a conventional turbine and compressor and uses oscillating pressure waves to do and extract work from the fluid. Wave rotors are opportune for small mass flows as they suffer less from the losses that are associated with low mass flow rates. Furthermore, wave rotors have less complex geometries that do not encounter issues when reduced in size. Power can be extracted from the wave rotors exhaust by converting the exhaust to thrust via a nozzle or shaft-work to spin a propeller. Based on the promises associated with using a wave rotor, this project researched the creation of a small 500-watt wave rotor.

I.2 Objectives

An overall object for this project was to design and test a 500-watt wave rotor for use as a Brayton cycle engine. This engine has multiple applications, but this research specifically looks at powering a micro-air vehicle the size of a small bird. While many parameters are investigated, the primary area of interest is the compression efficiency, η_c . To accomplish the primary goals and all secondary goals, the process was separated into multiple steps:

1. Use the NASA code developed by Dr. Dan Paxson to create a wave rotor cycle that will produce 500 Watts of power [1,2].
2. Design and create the wave rotor and all necessary hardware for testing of the wave cycle created in step one.
3. Test the wave rotor and complementary hardware for capability to spin at the design point.
4. Test the previously built combustor by McClearn for use at the new design point [3].
5. Couple the wave rotor hardware and combustor to test the design point for comparison to computational results as well as computation of efficiencies.

I.3 Methodology

All of the steps are intertwined, but required their own particular set of actions to be completed. In general, the tasks were accomplished in a serial order with some overlap between certain parts. The different steps are slightly discussed below and covered more in depth in the following chapters.

To understand the computational code, the first step was recreating the cycle used by McClearn. The code was then used with some previously calculated parameters to create a wave rotor cycle that will produce the goal power output. This power is based on the final exhaust properties of the wave rotor. The code work provided port conditions, rotational rate, and the physical parameters needed to create the wave rotor. The code and the process of using it is discussed further in Chapter 2 and 3.

The final outputs of the code allowed the wave rotor and accessory hardware to be designed using computer aided design (CAD) software. These items included the wave rotor, shroud, endwalls, and motor mount. Other software was utilized to complete a static stress analysis and a rotordynamic analysis on the wave rotor before any designs were finalized. These analyses ensured the wave rotor could withstand the rotational and thermal forces and would not encounter any detrimental modes during spin up. A bearing arrangement as well as a sealing method was devised for coupling the wave rotor to the endwalls. A commercial electric motor was chosen to spin the wave rotor as the rotational design point specified by the computational code. The process to design and create these different parts is detailed further in Chapter 3.

The aforementioned hardware was built and assembled. With the system built, the wave rotor was spun to the design point. During this process, starter air was flowed through the system to help spin the rotor. This testing was completed at the Detonation Research Facility (DERF) as a collaboration with The Air Force Research Lab (AFRL). This processes is discussed in further detail Chapter 3 and 4.

Testing of the McClearn can type combustor was completed at this projects design point. With a stable flame acquired, the combustor was cleared for coupling with the wave rotor hardware. This process is discussed in Chapter 3 and 4 with more information regarding the combustor in Chapter 2 and the McClearn thesis [3].

The coupling of the combustor and the wave rotor was completed in multiple steps. First a cold flow test was done to check for leaks as well as ensure all the instrumentation was working. The hot testing was completed with data being taken

during spin up and at the design point. This process and the results are discussed further in Chapter 3 and 4.

Overall, this research aimed to investigate a unique engine type to see how it compares to common engines of similar size and mass flow. The wave rotor was designed, built, and tested to examine how this technology performs when sized for small turbomachinery.

II. Background and Previous Research

II.1 Introduction

The drone or unmanned aerial vehicle (UAV) industry has been growing substantially in the commercial, industrial, and defense sectors. Estimations show defense spending reaching twelve billion dollars within the next decade [4]. These drones have wide scale applications from armament delivery, to passive surveillance. The growth of the drone industry will bring technological advances as the different aircraft look to outperform the competition. The power plant requirements for these drones vary widely based on size and mission of the aircraft. For the smaller scale UAVs or micro air vehicles (MAV), battery powered systems or small combustion engines have become commonplace. These small combustion engines are normally commercially made and designed for other applications. At altitude, these engines normally face large degradation in efficiency as the air pressure and density decreases compared to sea level. Other issues plague the smaller engines compared to their larger counterpart. The loss in efficiency creates a need for a new power plant for the small-scale drones of the future. Batteries have been utilized to power a variety of MAVs, but their low energy density limits their energy capacity per weight. Aircraft like the RQ-11 Raven and the AeroVironment Wasp III utilize battery powered electric engines [4,6]. These aircraft provide surveillance for the warfighter, but are limited by range and service ceiling. Figure 1 demonstrates the difference in energy density for a range of energy supply units [7]. The low energy

capacities per weight for batteries will either limit the aircraft performance or will be too heavy to be carried by the MAV.

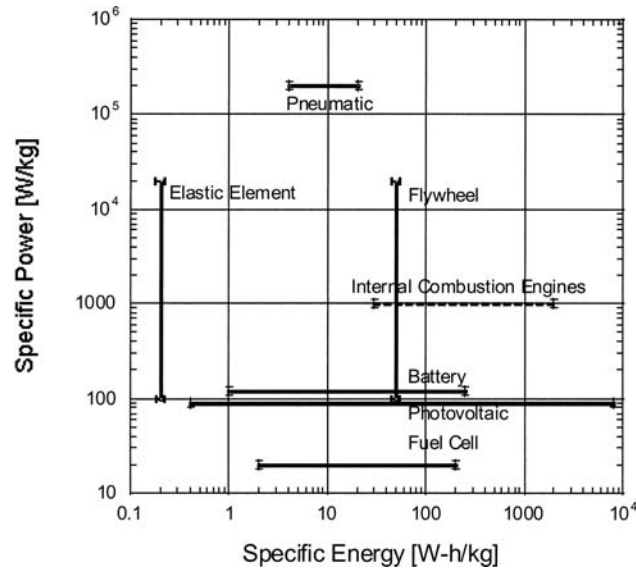


Figure 1 - Power and Energy Comparison [7]

One possible solution to power the small-scale drones is the utilization of a wave rotor. Wave rotors or pressure wave superchargers are a unique method of energy exchange. Instead of using rotors and stators, a cylindrical drum rotates between four separate ports. This drum serves as both a conventional turbine and compressor. Wave rotors are opportune for small mass flows, as they do not suffer from the same losses that plague typical turbo machinery.

Both typical turbo machinery and wave rotor operations are discussed in further detail later in this chapter. Furthermore, the efficiency issues that plague turbo machinery are detailed next in Section II.2. The benefits and operations of wave rotors as well as the different types are discussed in Section II.3 and Section II.4 respectively. Also, an overview of wave rotor history is provided in Section II.5 as well as a discussion on the

computational code used for this research in Section II.6. Lastly, a summary of current wave rotor research is presented in Section II.7.

Wave rotors have been around for over half a century and have limited commercial success. Modern advances in both analysis and machining have allowed an overall improvement on wave rotor design. The rise of the small drone industry may be the niche that allows the wave rotor industry to expand and become prevalent. This research looks to design and create a micro-scale wave rotor that could be used as a stand-alone propulsion system or be a part of a distributed power system.

II.2 Small Engine Issues

Engines producing a few kilowatts or lower are currently used in home equipment such as a pushing lawnmower or a leaf blower. These engines normally operate at one specific design point and can be bulky based on their operating market. With the rise of the drone industry, similarly sized engines have been researched to reduce size and increase efficiency. This research has investigated both internal combustion piston engines and gas turbines. The following sections will detail the limiting factors involved with shrinking each engine type and the effects engine size has on overall efficiency.

II.2.1 Internal Combustion Engines

Internal combustion engines are a critical part of our society today and have been for over a century. From powering automobiles to handheld garden equipment, IC engines range in size and purpose. IC engines have been used to power small drones for

the US military. Among these is the RQ-16 T-Hawk created by Honeywell for the United States Army. The T-Hawk uses a twin piston engine that generates around six kW of power [8]. The propulsion system for the T-hawk allows it to be airborne for around forty minutes and has an operating range of six nautical miles or eleven kilometers. This system is one of the smaller drones being utilized with an IC engine and is still magnitudes larger than the MAV size the US military is pursuing.

Internal combustion engines operate on the premise of compressing and expanding gasses at elevated temperatures and pressures. As mentioned with the T-hawk, IC engines normally have a piston that is used for the energy transfer. These engines are better known as a two-stroke or four-stroke engine. The general engine cycle process incorporates the movement of the piston with the introduction of fuel and oxidizer and the extraction of the combustion products. Timing, the combustion reaction, and the exchanging of gasses are all critical areas to the IC engine performance. Though this engine type has been researched and worked with for over a century, a range of parameters can plague the engine's performance. The factors affecting the use of these engines with small UAV's are described below.

Exchanging gasses is a critical part of IC engine performance with most engines containing some feature to induce flow swirl or turbulence inside the combustion chamber. This turbulence or swirl helps mix the fuel and oxidizer for high combustion efficiencies. The rate of mixing depends on the type of ports used for the fuel and oxidizer along with the Reynolds number of the flows [9]. As IC engines decrease in size, the overall flow speed decreases. This decrease in velocity occurs because of an increase in relative fluid friction as well as the smaller turn angles inside the engine [7,10].

Reynolds number of the flow will fall as the overall flow velocity decreases. Equation 1 demonstrates the relationship between Reynolds number (R_e) and flow velocity (v). For flow in a pipe, Reynolds number uses a hydraulic diameter term, d^* , instead of the length term. The reduced size in small IC engine piping lowers the associated Reynolds number.

$$R_e = \frac{\rho v L}{\mu} \quad (1)$$

With the lower Reynolds number having a negative effect on mixing, the combustion process is slower and less efficient. Furthermore, combustion efficiency decreases as size decreases based on the higher percentage of quenching volume [4]. Quenching occurs when the flame inside the combustion chamber is extinguished by coming into contact with the chamber walls. This process creates unburned hydrocarbons that leave the engine with the exhaust. The unburned hydrocarbons directly reduce the combustion efficiency and can cause knock [9].

Menon et al. investigated combustion efficiencies for small IC engines. They found that between 50%-60% of fuel energy was being lost through incomplete combustion. This energy loss gives a combustion efficiency less than 50% while a normal sized combustion engine will have an efficiency greater than 95% [11]. Even with infinite flame speed and complete mixing, the effective volume of the combustion chamber dictates the amount of energy released by the fuel [7].

When shrinking down internal combustion engines, physical processes like heat transfer, exchanging of gasses, and overall combustion vary from their larger counterparts. Heat transfer issues rise as the surface area-to-volume ratio increases. The overall heat transfer to the walls surges as the combustion chamber size decreases. This increase in heat transfer creates both efficiency and material issues. At micro-scale sizing,

the heat transfer becomes too great for self-sustained combustion for the respective fuel type [7]. For small IC engines there comes a point where the device is no longer thermodynamically practical [10].

Along with combustion efficiency, the mechanical efficiency of an IC engine suffers as the engine size is reduced. The decrease in mechanical efficiency occurs because of two intertwined reasons. As discussed before, with decreasing engine size comes an increase in fluid friction. Furthermore, all of the engine components decrease in size. Some of these components are moving parts including the piston, crankshaft, camshaft, valves, and others. The increase of friction on these small components can create a situation where more power must be added to the system than the total power produced. Menon, et al. also investigated the loss in mechanical efficiency with small-scale IC engines. A range of efficiencies from 50%-70% was found for engines ranging from 8 watts to 656 watts. While a modern automobile IC engine would operate at 90% during normal operations [9,11]. Note the mechanical efficiency does decrease for all IC engines as the engine is throttled back. The decrease when throttled corresponds to the decrease in flow velocity found inside the engine [9].

In general, remote control (RC) airplanes have stayed at lower altitudes and within eyesight of the pilot. The lower altitudes allowed the engines to operate with efficient air densities and pressures. With the goal of achieving higher altitudes with the next generation of MAVs, the aircraft engine must perform well with an increase in altitude. Air pressure decreases 11% while air density decreases by 9% for altitude increments of 1000 m. With most internal combustion engines being designed for ground use, the decrease in pressure and density has detrimental effects on performance.

Research done on a one-kilowatt class IC engine has shown a 3.5% decrease in overall power for every 300m increase in altitude [12].

Overall, a variety of reasons cause IC engines to be inefficient at small scales and at high altitudes. These issues include flow velocity, thermal losses, mechanical losses, fluid friction, and combustion efficiency. Though great for a plethora of jobs and situations, internal combustion engines lack the performance parameters to power the next generation of MAVs.

II.2.2 Gas Turbines

Similarly to internal combustion engines, gas turbine engines suffer from efficiency issues as they decrease in size. The decrease in efficiency comes from a combination of factors including blade size, machining limitations, and heat transfer. These factors and other turbomachinery issues will be discussed in the following paragraphs.

Decreasing engine size and power requires a size reduction in all components as well as a decrease in mass flow. For a gas turbine engine, work is done and extracted using a collection of compressor and turbine blades. Compressor performance is based on the ratio of pressure and temperature entering and leaving the compressor. Equation 2 illustrates this relationship in the calculation of the compressor efficiency or η_c .

$$\eta_c = \frac{\left(\frac{P_{tCO}}{P_{tCI}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\left(\frac{T_{tCO}}{T_{tCI}}\right) - 1} \quad (2)$$

The subscript CI stands for flow entering the compressor while the subscript CO signifies flow leaving the compressor. For gas turbine engines, the level of compressor efficiency is critical for overall performance. An increase in compressor efficiency allows the respective engine to operate at higher altitudes, consume less fuel, and overall outperform the previous engine version.

The compressor blades experience an increase in viscous losses as their size decreases. The increase in viscous losses comes from the viscous boundary layer becoming significant because of the relatively large tip clearance. In other words, when the blades become small the necessary tip clearance is relative large, a viscous boundary layer is formed. The viscous boundary layer also increases, based on the Reynolds number being lower for small gas turbines. Therefore, laminar or viscous flow is dominant inside these small engines compared to turbulent flow being dominant in large-scale engines [13]. When this occurs, the engine components require more energy to produce the same amount of work. On a mass flow basis, compressor efficiency plummets when the flow goes below 0.14 kg/s [14]. The mass flow for this project is around 0.02 kg/s. Furthermore; nearly 2/3 of the work produced by the turbine is used to spin the compressor in a modern gas turbine [13]. Therefore any decrease in compressor efficiency will increase the power required to a higher percent of the work created by the turbine. This increase in power required is taken directly from the goal power out. The combination of viscous boundary layer forces and the amount of power required for the compressor make small-scale gas turbines inefficient.

As turbomachinery decreases in size, the process of machining the small geometries becomes exceptionally difficult. Advances in additive manufacturing have

helped, but the industry is new and growing [15]. With smaller geometries, the machining tolerances become more of an issue. These tolerances may dictate how large a gap is between a rotating part and a stationary part. For smaller gas turbines, the gap or machining tolerance issue has more of an effect than the same gap on a larger gas turbine.

Another engine component that has issue with the decrease in size is the bearing system. The smaller diameter of the blades requires a higher revolution per minute (RPM) to create the same blade tip speed. Some small gas turbines have been designed to spin at 250,000 RPM [14]. This high spin rate causes issues across the board and must have specially designed bearing systems. Some of these issues include a larger range for detrimental modes to occur and an increase stresses involved with spinning. Furthermore, the speed ramp up process and sustaining the high RPMs create vibrational issues that can propagate to other engine components. Overall, shrinking the size of gas turbines requires an increase in shaft and blade speed, which causes a range of component issues.

As with internal combustion engines, heat transfer parameter and combustion suffer as gas turbines decrease in size. The increase in surface area to volume raises the amount of heat transfer from the flow into the gas turbine walls. If too much heat transfer occurs, the combustion flame can be extinguished [16]. Furthermore, the heat transferred into the walls is a loss and reduces the overall efficiency. Material limits must always be a concern with any heat transfer increase. To achieve high combustor efficiency, adequate residence time, t_r , for a complete reaction must be present by having a large enough combustor. In other words, the combustor must be large enough to allow for the combustion to take place. If the residence time and size are not adequate, the engine will be releasing unburned hydrocarbons along with the exhaust gas. When incomplete

combustion occurs, the engine is not extracting all of the energy created by the compressor and the reaction process. The combustor size normally required for efficient combustion is not possible when designing a small-scale gas turbine [13].

This section has shown the issues that face internal combustion piston engines and gas turbine engines as they decrease in size and power output. Both engine types suffer in regards to thermal issues and machining limitations. The main issue facing IC engines is the reduction in ability to exchange gasses. Gas turbines main issue is the viscous forces found at the end of the compressor blades. With both small engine performances lacking, there is an operational area for a new system to be commonplace in. Through this research and other projects, wave rotors may be able to replace normal small-scale engines.

II.3 Wave Rotor Operation and Benefits

Wave rotors operate on different premises compared to normal engines. Because of the differences, a range of benefits and issues are associated with wave rotors. The way wave rotors work as well as the pros and cons with this method are discussed in this section.

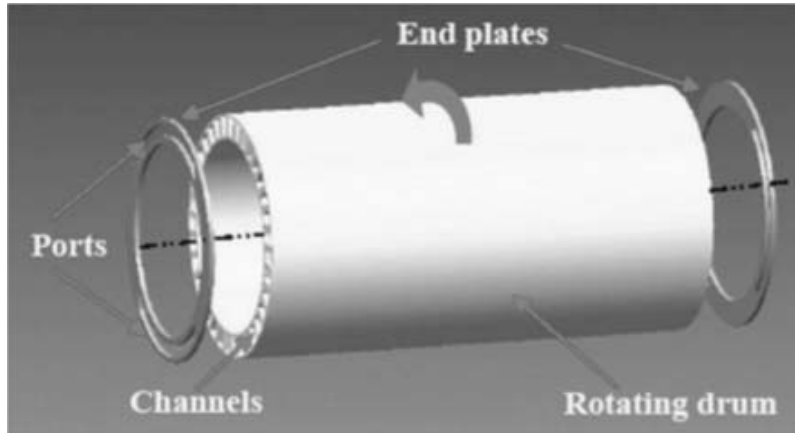


Figure 2 - General Wave Rotor Schematic

Wave rotors consist of a cylindrical drum containing through channels aligned along the outer diameter. These channels are then rotated at high speeds in between four different openings. The rotating drum, channels, and different ports are illustrated in Figure 2

The labels for these port openings, used in this document, are compressor in (CI), compressor out (CO), turbine in (TI), and turbine out (TO). Other wave rotor research has used air in and out (AI, AO) to represent the compressor ports and exhaust in and out (EI,

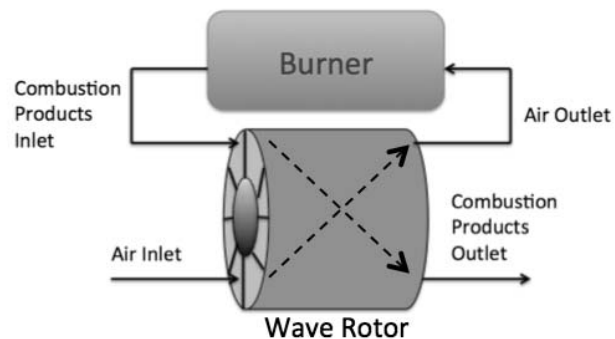


Figure 3 - General Process of a Through-Flow Wave Rotor

EO) to represent the turbine ports. While inside the wave rotor, work is either done to the fluid or extracted from the fluid using oscillating pressure and expansion waves. In general, the wave rotor has two parts to it. The first part (CI to CO) replicates a compressor where fresh air enters the wave rotor and exits with higher density, pressure, and temperature from the work done on the fluid. The second part (TI to TO) mimics a turbine where exhaust from the burner enters one of the channels, is expanded, and work is extracted out. Each channel does the job of a compressor and a turbine as it rotates between ports. The compression and expansion are dependent on each other and rely on the timing of reflecting shock waves. The path the fluid takes from being fresh air to becoming exhaust is shown in Figure 3.

Understanding the wave propagation and the internal physics of a wave rotor is complex and difficult. This paragraph will follow one channel as it rotates around the annulus while explaining the internal workings. To help the understanding, Figure 4 is labeled and colored accordingly. The dashed orange lines are expansion waves while the solid blue lines are compression waves. The illustration has an x-axis of rotor length and a y-axis of theta. The y-axis can be thought of as the circumferential distance the channel covers as the rotor spins. This description is for a through-flow wave rotor. Starting the bottom right, the TO port opens (Labeled as 4 in Figure 4) to ambient air to reduce the total and static pressure inside the channel and send an expansion fan across the channel. Second the CI port opens (Labeled as 1 in Figure 4) as the expansion fan reaches it and fresh non-pressurized air fills the rotor channel. As the fresh air enters the wave rotor, the TO port will close creating a weak hammer shock that will compress the fresh air. Next, as the annulus rotates, the CI port is closed and the TI port opens (Labeled as 3 in Figure

4) allowing exhaust gas from the burner or engine to enter the channel. The timing between the CI port closing and the TI port opening is minimized to reduce leakage. As the fluid from the TI port enters, the main compression shock is created. This shock travels down the annulus and reaches the opposite wall and reflects back right as the CO port opens (Labeled as 2 in Figure 4). The reflection allows for further compression of the fresh air and forces the fresh air out of the CO port. The TI port closes before the CI port and the pressurized air is routed to the burner to be utilized. The closing of TI creates an expansion wave from the TI port to the CO port. Finally, the TO port opens and the exhaust gas is routed out of the wave rotor which then restarts the whole cycle.

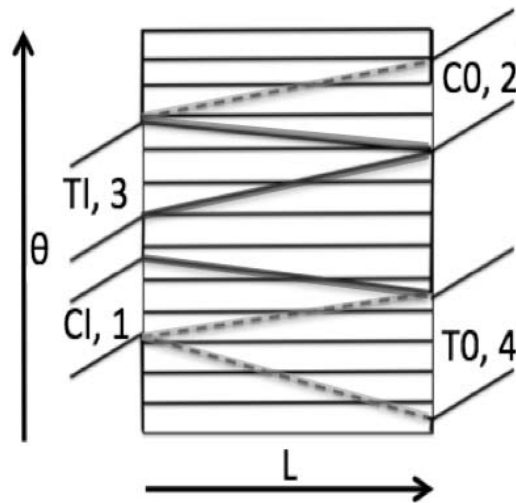


Figure 4 - Ideal Wave Propagation with Ports

The above cycle happens twice in each channel per completed revolution. In other words, the wave pattern occurs every 180° . To ensure the wave pattern is created, rotational speed, port locations, rotor length, and internal temperature are critical. The rotational speed and port locations determine port timing in terms of opening and closing.

The rotor length and internal temperature determine wave propagation speed. The distance traveled is the rotor length and the waves travel at the local sonic velocity based on internal temperature. Because of the importance of wave timing, wave rotors perform poorly in off-design conditions. This detriment, other difficulties, and wave rotor benefits are detailed in the following paragraph.

As discussed in Section II.2, typical engines suffer when sized for smaller purposes. Wave rotors have benefits that allow them to perform better at these small sizes. The first of these benefits is the lack of complex geometries incorporated in a wave rotor design. Unlike the critical curves of a compressor blade, a wave rotor can do and extract work with straight channels. Wave rotors reduce the amount of stages needed in a system since the compressor is also the turbine. The reduction in number of stages needed helps the size required and reduces the overall weight. Furthermore, wave rotors use shock waves to compress the incoming air. Because of this, wave rotors do not experience the viscous forces that plague small turbines. These shock waves have a higher compression efficiency compared to a compressor blade. The relationship between isentropic efficiency and pressure increase for shock waves and compressors is shown in Figure 5.

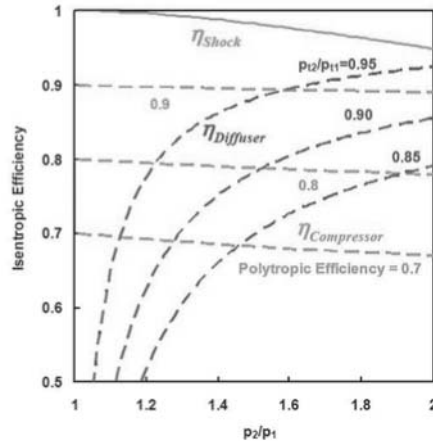


Figure 5 - Isentropic Efficiency of Shock Waves Vs. Compressor [17]

Another benefit to using shock waves for compression over a compressor is the response time. Since shock waves travel faster than the speed of sound, the response time is notably faster compared to other forms of compression like a turbocharger [17]. Figure 6 displays this difference by comparing the response time of the commercially produced Comprex® to a general turbocharger.

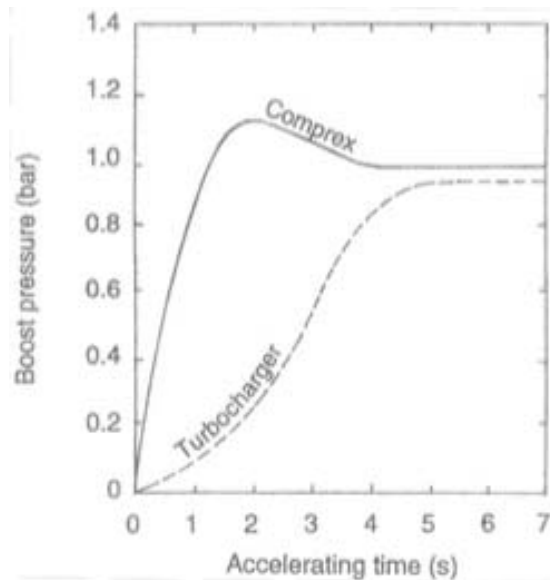


Figure 6 - Response Time of a Wave Rotor Vs. a Turbocharger [17]

Wave rotors have their fair share of disadvantages as well. As mentioned earlier, wave rotors perform best at one design point. Because of this, wave rotors are non-ideal for engines requiring frequent throttle changes. The design point requirement does help if the system is designed for an aircraft engine with a specified cruising speed and altitude. Another disadvantage stems from the method of opening and closing ports. Unlike typical engines that use valves for fluid timing, wave rotors use the rotating cylinder and the port locations for timing. Because of the moving parts, the flow paths can never be completely open or closed [18]. This issue causes internal wave rotor leakage. Further wave rotor complexities include difficulties with determining compression and expansion efficiencies. Since the compressor is also the turbine, it is impossible to dictate when the flow in the wave rotor is compressed intake air or re-circulated exhaust gas. Computational work has been completed to compute these efficiencies, but no clear method is currently available.

II.4 Types of Wave Rotors

Two types of wave rotors have been created throughout the research and development of these devices. The differences stem from the application and usage of the system. The two types studied are reverse-flow and through-flow wave rotors. The operation for the through-flow wave rotor was discussed previously in Section II.3. The wave diagrams for each wave rotor type are characterized in Figure 7. This section will clarify the general differences before briefly explaining the reverse-flow wave operation and general benefits of each type.

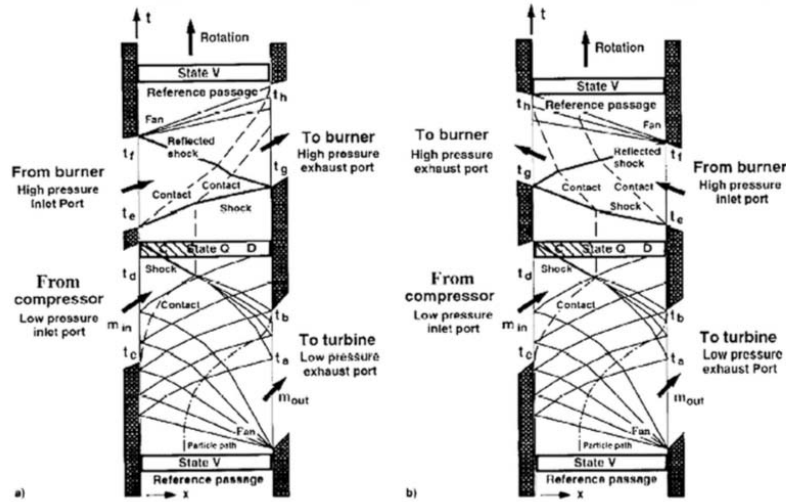


Figure 7 - Wave Diagrams for a Through-flow (Left) and Reverse-flow (Right) Wave Rotors [19]

Through-flow wave rotors have a variety of uses from turbine topping cycle to cooling flow for a refrigerator. A through-flow wave rotor has the CI and TI ports on one side with the CO and TO ports on the opposite side. Because of this, the fresh intake air travels through the wave rotor length from one side to the other. The path of the fresh air allows the wave rotor to be fairly self-cooled and will maintain a constant rotor temperature throughout. The self-cooling allows through-flow wave rotors to operate in conditions that would be too hot for other types of wave rotors.

Reverse-flow wave rotors are frequently used as a Pressure Wave Supercharger (PWS). In a reverse flow, the CI and CO port are on the same side and the TI and TO port are on the same side. Therefore, the cool intake air that is being compressed stays on one side while the hot combusted air stays on the opposite side. In other words, either flow enters the channel and reverses direction to exit on the same side it entered. This port setup creates a relatively hot and a relatively cold side of the wave rotor. Based on the

absence of the self-cooling, compared to the through-flow, reverse-flow wave rotors can only be utilized in a lower temperature situation. The lower temperature is required from a material and structural standpoint. The overall results of a reverse-flow wave rotor are the same as the through-flow with the work being done by a series of shock and expansion waves. The wave patterns are similar with the TO port opening to create an expansion fan and closing to create a hammer shock. Comparably, the CI port opens to bring fresh air into the wave rotor. Since the exhaust gas (TI port) enters on the opposite side compared to a through-flow, reverse-flow wave rotors do not have the reflected shock across the fresh intake air.

Reverse-flow wave rotors do have some benefits compared to the through-flow system. The first advantage deals with the rotor bearing locations. These bearings are used to support the rotor as it spins inside the shroud. If the design only requires one bearing location, the bearings can be put on the cool side. This location reduces the thermal stress encountered by the bearings. Another benefit comes the amount of exhaust gas recirculation (EGR). Since the fresh intake air and exhaust products do not cross the entire wave rotor, the amount of re-circulated exhaust is lower than a through-flow wave rotor.

II.5 Wave Rotor History

Early patents by both Knauff, in 1906, and Burghard, in 1913, described similar devices to exchange pressure [20, 21]. These devices consisted of long narrow channels revolving on a cylindrical drum between ports and were called “static pressure

exchangers”. The ports had different pressures and the channels allowed pressure exchange to happen. Burghard, in 1928, created the first description of a “dynamic pressure exchanger” using pressure waves to both compress and expand the fluid [22]. The utilization of pressure waves helped coin the term “Wave Rotor”. At this time, the knowledge on unsteady fluid mechanics hampered scientific progress.

Further work was not started until 1940 when the Brown Boveri Company (BBC) from Switzerland designed and tested a wave rotor as a topping cycle for a locomotive gas turbine [23]. This wave rotor consisted of thirty channels rotating at 6,000 RPM. Early tests had promising results including pressure ratios of 3:1 and an efficiency of 69%, but poor integration and mismatched design work led to dismal results. The success and failures from this work helped spread the interest in using wave rotors.

A joint effort between the U.S Bureau of Aeronautics and the ITE Circuit Breaker Company investigated using wave rotors as a pressure wave supercharger for diesel engines [23]. The two groups tested designs from 1947 to 1955. Success from these tests stimulated BBC to once again work with wave rotors. Starting in 1955, BBC along with the Swiss Federal Institute of Technology started research on supercharging commercial vehicles. The first prototype was used on a diesel truck in 1971 and in 1978 work began to supercharge a passenger vehicle. One design, called the Comprex®, was utilized to supercharge an Opel 2.1 L diesel engine. An assembled and labeled Comprex® is shown in Figure 8.

The fast reaction time, condensed size, and reduced weight compared to normal turbochargers made the Comprex® successful. This wave rotor was successfully tested on vehicles created by Ferrari, Peugeot, and Mercedes-Benz. Mazda used the Comprex®

in over 150,000 Mazda 626 Capella passenger vehicles by the year 1987. Other companies that researched wave rotors early on included Power Jets Ltd (1949-1967), Ruston-Hornsby Turbine Company (1950s-1960), General Electric Company (1956-1963), General Power Corporation (1960s-1984), and Rolls-Royce (1965-1972). The academia sector has also researched wave rotors with work coming from Cornell University from 1948 to 2001 and the Naval Postgraduate School from 1981 to 1986.

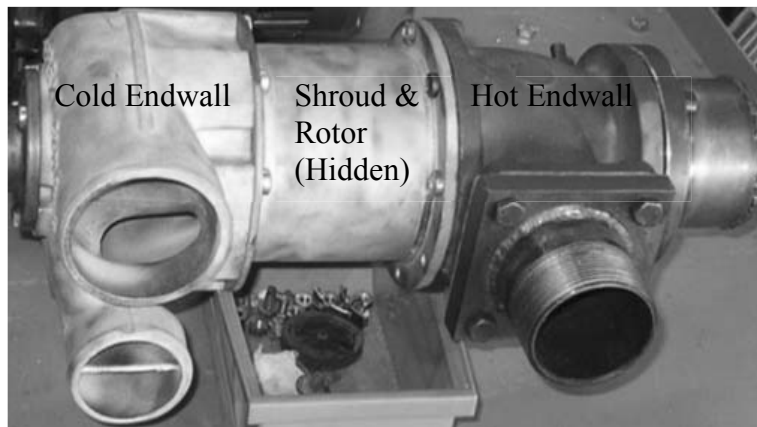


Figure 8 - Assembled Comprex® PWS [24]

More recent research has been done at Michigan State University, Allison/Rolls-Royce Liberty Works alongside Indiana University-Purdue University at Indianapolis, National Aeronautics and Space Administration John H. Glenn Research Center, and The Air Force Institute of Technology (AFIT). Michigan State University started their work in 2002 and focused on ultra-micro wave rotors [25]. These small wave rotors were being pursued to boost an ultra-micro gas turbine. The wave rotors are on the scale of 1mm in length with a radius of 0.42mm. This small scale brings issues in regards to losses and the manufacturing processes. Advances in 3-D printing may allow for higher quality ultra-

micro wave rotors. Other wave rotor research at Michigan State University looked at using a wave rotor as a component in a water refrigeration system [26].

Work between IUPUI and Allison/Rolls-Royce Liberty Works deals with another type of wave rotor called an Internal Combustion Wave Rotor. Like the through wave rotor detailed here, each annulus was charged and discharged as it rotates past different ports. Instead of just being charged with air, internal combustion wave rotors are charged with a fuel air mixture and the mixture is ignited [27]. Some design advantages compared to a normal wave rotor include not needing an external burner, low NO_x formation, and fewer connections where losses can occur. In 2009, a design was tested at Zucrow Labs at Purdue University. These tests spanned over 30 days with the longest run being a 3 second firing. A total of 34 successful tests were completed consisting of approximately 2,000 combustion events. Sealing the individual combustion events was found to be more of a performance issue compared to an operability issue. Equivalence ratios from 0.39 to 1.03 were tested and the experimental data helped validate the modeling approach.

Starting in the early 1990's, research has been conducted at NASA Glenn Research Center focusing on using a wave rotor as part of a Brayton Cycle [28]. With an overall goal of increasing the combustion pressure of a gas turbine engine, a wave rotor was used in between the high-pressure turbomachinery and the combustor. This experiment consisted of a four-port rig where each port could be monitored and controlled. The rotor from this experiment is shown in Figure 9.

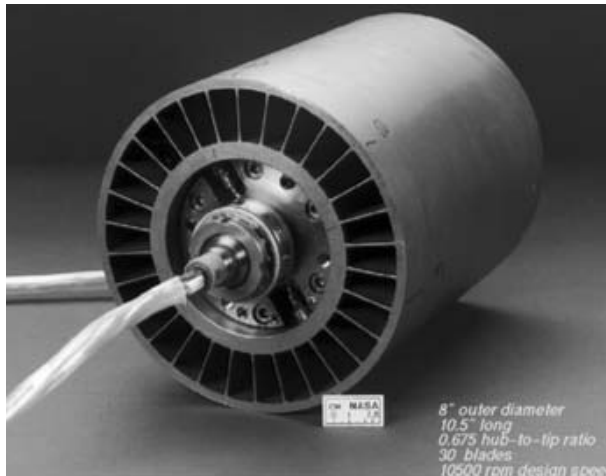


Figure 9 - Rotor from NASA GRC Four-Port Experiment [28]

Low pressure cooled air would first enter the wave rotor and complete the compression part of the cycle. Once exited, this air would be measured for temperature, pressure, and mass flow. A second supply of air was passed through the expansion part of the wave rotor. This stream of air was at an elevated pressure and heated using an electric heater. Both the temperature and pressure were determined based on the air exiting the compressor stage. The second air supply made this an open loop, but using the correct temperature and pressure from the compressor output allowed for the experiment to behave as a closed loop.

II.6 Computational Fluid Dynamics Code

A computational fluid dynamics (CFD) code was utilized for this project and has been used for multiple decades to model wave rotor wave patterns. Dr. Dan Paxson of NASA Glenn Research Center (GRC) developed this code starting in 1992. This quasi-one-dimensional code is a finite volume technique based on the method of Roe [1]. The method of Roe is used with a second order Lax-Wendroff scheme to model one wave rotor channel as it rotates between the different port openings. The code was initially

used by GRC to simulate the wave rotors designed and tested there for use as a turbine topping stage. Through the years Dr. Paxson has created edits to the code and multiple studies have demonstrated the accuracy.

The CFD code created by Dr. Paxson was not the first created in order to simulate wave patterns inside a wave rotor [29]. In order to improve the computations, a variety of new assumptions were used to create the quasi-one-dimensional code. Previous wave rotor models assumed a channel that was sealed on both ends and had instantaneous port openings and closings. Furthermore, previous codes utilized inviscid flow and did not include a loss model for the leakage between the wave rotor channel ends and the closest face of each endwall [1]. To account for the opening and closing of the ports, a temporal feature was utilized to replicate the effects created [2]. The code uses the size and location of each port along with the rotational speed to calculate these effects. The leakage model added is based on the minuscule gap between the rotor and the endwalls. The gap must be large enough to allow the wave rotor to spin freely without hitting the endwall and must account for any thermal growth. To model this gap and the associated leakage, the discharge coefficient of the channel was added to the code. [1]. Lastly, frictional effects were found to be more substantial than previously thought and a viscosity term based on Reynolds number was added to account for the viscous and momentum forces. The combination of new assumptions and additional parts created a more accurate model for predicting wave rotor performance.

As mentioned before, the CFD code utilizes the method of Roe to complete an explicit finite volume analysis. The code views the channels as shock tubes and uses the comparison to simulate the shock wave compression as well as the expansion fan process.

Rotational velocity is negligible compared to the axial velocity of the shocks bouncing back and forth inside the wave rotor channels. Euler equations are applied with the assumptions that the gas flow is adiabatic, calorically perfect, inviscid, and one-dimensional. The second order Lax-Wendroff scheme is used to solve the Euler equations. This scheme is required to create a valid solution because of the discontinuities involved with the shock waves and expansion waves and to calculate both the differential and integral forms of the discrete conservation equations [1, 30]. The utilization of the method of Roe does approximations to create a less computationally expensive model while still dealing with the shock discontinuities. Paxson has demonstrated that this method allows for high resolution near the discontinuities in the shock waves while keeping an overall solution that is thoroughly close to the actual solution [1].

The code models the environment inside one channel as it completes one cycle rotating between the four different ports. For the design created here, a channel completes two cycles for every complete rotation for the rotor. The channel is divided up into discrete cells and has the conservation of mass, momentum, and energy applied. The conservation equations are then used to calculate the flux of each variable throughout each cell. Through this process, the code computes pressure, temperature, density, and velocity as a ratio of inlet port conditions. There are three modules that combine to create the CFD code. The first part, WRITR, uses a temperature ratio and other inputs to create an ideal wave rotor cycle. The second module, WR, adds in relations and real effects to create a non-dimensional rotor and wave cycle. The final module, SIMBC, adds in loss modules and real physics in order to create the results that will be seen in the experiment.

When completed the code provides a wave rotor cycle, inlet/outlet conditions, and port locations/angles. Based on a rotational speed and a Reynolds number relationship, the code has the user create the physical size of the rotor.

The CFD code has been utilized and tested multiple times since it was first created. The first round of validation occurred during the mid 1990s while NASA was conducting wave rotor experiments [31]. The initial code work was done to analyze mass flow rates between 0.23 kg/s and 2.27 kg/s. A NASA experiment using a Rolls-Royce Allison Model 250 engine along with a wave rotor was conducted at a flow rate of 1.36 kg/s. During this time, a heat transfer model was added into the code to create a relationship between the CO and TI ports with the inclusion of an external burner loop for a through flow wave rotor. The addition of the CO and TI relationship was used in this paper's analysis as the wave rotor designed here is a through flow and utilizes an external burner [31].

The Roll-Royce Allison model 250 engine experiment allowed for the comparison between the experimental and computational results. Initial comparisons demonstrated similar trends for both temperature and pressure, but the actual values were off. This difference primarily occurred because of the measurement locations in the experimental setup. The experimental setup was edited to move the various sensors as close as possible to the locations specified and analyzed in the code. These locations are where the endwalls and rotor meet or the gap discussed earlier. Further work on the experimental set up included a moveable endwall to better handle the thermal expansion and limit the endwall to rotor gap. These edits allowed for fewer losses and more accuracy between the experimental and computational results [30].

The wave rotor research conducted at AFRL involves small-scale wave rotors that are below the mass flow rate area used to create the CFD code. In order to ensure the code is still valid, work was completed by Smith et al. This work was done by comparing the results from a commercially made Compres® pressure wave supercharger, (0.099 kg/s), to the computational results. The comparison showed a maximum difference of 5.96% with an average difference of 2.77% and demonstrated the capability of using the code to design small-scale wave rotors. To compare the results, the code required edits to have it compute a reverse flow wave rotor. Paxson modified the code to alter the intake air path by having the CI and CO ports on the left side of the rotor while having the TI and TO ports on the right side [23]. Further work was completed by Mataczynski to create a scaled PWS and compare it to the code results. This wave rotor operated at less than 1.36 kg/min [32]. On-design condition results had little discrepancy between the computational and experimental results. The code demonstrated correct trends for the off-design analysis, but underestimated the actual performance values. The underestimation on the off-design computational analysis occurred because of endwall pockets that were added to the experimental design. These endwall pockets are used to dampen unwanted waves and are not included in the code. The code has been used since this validation to create a wave rotor designed by McClearn [3].

II.7 Associated Wave Rotor Research

The Air Force Institute of Technology and the Air Force Research Laboratory have worked together on wave rotors since 2012 [23]. Two recent small scale wave rotor or pressure wave supercharger projects have led to the development of the current project.

II.7.1 PWS to Supercharge a Diesel Engine

Work began in 2013 as part of Mataczynski's master's thesis [32]. This work looked at testing the commercially made Comprex®, simulating the Comprex® wave pattern with the NASA CFD code, using the same code to create a reverse-flow wave pattern for use with a 95 cc Brison engine, and finally the creation and testing of the newly designed PWS [32].

Improvements in the test setup allowed a higher inlet air temperature for testing the Comprex®. This improvement increased the inlet air temperature from 883 R, tested by Smith et al., to 1035 R [24, 32]. The elevated temperature allowed for balanced mass flows that were not achieved in previous testing. Mataczynski was able to test the Comprex® at operating conditions similar to what the device was originally designed to. Once the Comprex® had been tested, the NASA CFD code was used to simulate the wave pattern created. The results from this simulation demonstrated the codes ability to correctly replicate and design the internal workings of wave rotors.

With the code validated, work was completed to design a scaled down reverse-flow wave rotor for use with a 95 cc Brison engine. This wave rotor design had a length of 4.1 cm and an outer diameter of 4.3 cm. Since it was a scaled down version of the Comprex®, this design had a higher spin rate of 34,500 RPM compared to 15,000 RPM

[32]. The Mataczynski design utilized a cantilevered shaft that was press fitted in the cold side endwall and was spun using an electronic brushless motor. This design never hit the design rotational speed and topped out at 22,000 RPM because of insufficient power to the motor [33]. Mataczynski worked on other PWS designs and made improvements along the way. The current design couples with a Kubota 722 diesel engine and is detailed next.

The present system is a reverse flow wave rotor designed to spin at 26,500 RPM [34]. To account for previous spin up issues, the rotor is attached to a shaft on the cold side of the PWS. This shaft is then supported inside the left endwall with two sets of bearings. Dyrobes[®] was used to analyze this rotor and shaft setup. The software found the first critical rotational mode to occur at 41,169 RPM or 142% of the maximum operating speed.

This design improves on the leakage issue encountered in previous designs. The leakage led to overheating of the bearings and errors in the air in mass flow calculations. To account for these issues, a number of design changes were made. These design changes include separating the leakage path from the air inlet steam, a radial shaft seal was added in between the rotor cavity and the first bearing, and a water-cooling passage was added to cool the first bearing.

Currently this wave rotor system has been used with an electric heater for the exhaust gas inlet flow as shown in Figure 10. The electric heater was used for the initial tests to ensure the system could spin at the required rate and to test out all of the instrumentation and the required test set up. This testing showed the system's ability to spin at the design point and the ability to acquire all the required flow data. With the

successful electric heater test, the project members are working on coupling the PWS with the Kubota engine.

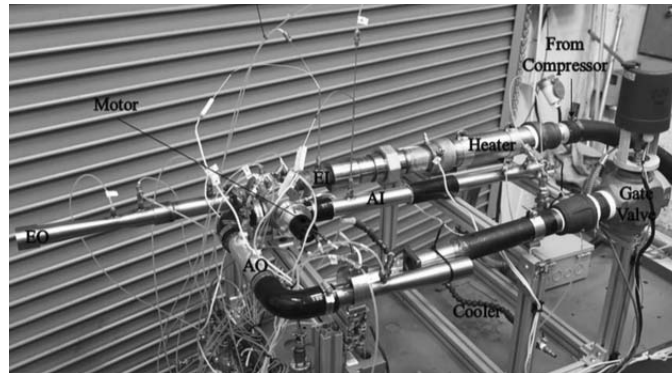


Figure 10 – Pressure Wave Supercharger Test Setup [33]

II.7.2 1.5 kW Through-Flow Wave Rotor

The second relevant project of concern to this thesis was the development of a through-flow wave rotor designed and created by McClearn [3]. This project had a goal of creating a Brayton cycle engine by using a wave rotor with an external combustion loop. The goal power out for this project was 1.5 kW to power a Class 2 UAV system. This project also included the design and testing of an external can type combustor.

The wave rotor designed by McClearn had a length of 6.4 cm and an outer diameter of 4.2 cm [3]. This rotor was designed to spin at 38,500 RPM and utilized two concentric shafts on either side of the rotor. The rotor had a hollow center to account for any uneven heating throughout wave rotor operation. The rotor and shaft were supported using an air bearing around the rotor and two unshielded deep groove ball bearings on one side of the rotor. The other side of the rotor was used to attach the electric motor. This system encountered some detrimental modes during spin up, but was able to throttle past them.

During a later test, the shaft was damaged when the wave rotor was spinning near 39,000 RPM.

The McClearn thesis also designed a combustor loop and had venturis made for the low mass flows associated. In order to supply the required temperature and mass flow from in between the CO and TI ports, a custom can type combustor was created. The custom combustor was necessary because the flow requirements could not be met by anything available for purchase commercially. The can type combustor was designed by McClearn with the help of Mr. Dave Burrus of ISSI. The cross-section view of the design is shown in Figure 11.

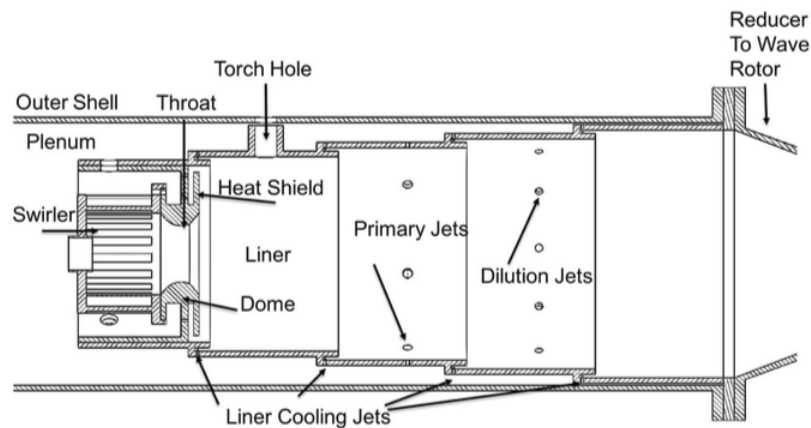


Figure 11 – Combustor Drawing Cross-section [3]

Looking at the cross section view, flow from the CO Port was directed to the left hand side of the combustor. Ethylene, as the chosen fuel, was added to the flow before the swirler was reached. The fuel lean mixture entered the combustor and was ignited by a hydrogen torch. The torch was used because conventional spark plugs would not be able to survive in the high temperature environment. The hydrogen torch consisted of a ceramic tube with two through holes. One of these passages contained a wire to create a

spark while the other passage contained the hydrogen. At the end of the ceramic tube, a supply of air was given to help ignite the hydrogen.

The combustor was tested to find the operability zones. During this testing, the inlet air provided was heated using a T63 combustor. Minimum ignition temperature was found to be 560 K at an air mass flow rate of 1.59 kg/min. Any inlet temperatures lower than this, the combustor would not sustain the ethylene flame. With the minimum ignition conditions known, the combustor was tested at the design point and a range of conditions. Low flow rate testing was started with an inlet air mass flow rate of 1.54 kg/min. This inlet airflow rate was used with an ethylene flow rate of 0.045 kg/min. The sustained outlet temperature for this condition was 1,150 K. The ethylene flow rate was then increased to 0.050 kg/min and produced an outlet temperature of 1,183 K.

Two venturis were designed and used for the McClearn experimentation rig are shown in Figure 12. One of the venturis monitors the intake air going into the CI Port while the other monitors the final exhaust gas going out of the TO Port. The two venturis were designed and built by Fox Venturi of Fox Valve Development Corporation. The designs were based on pressure, temperature, mass flow rate, fluid type, and an approximate ratio of specific heats. The CI venturi had a throat diameter of 20.07 mm while the TO venturi had a throat diameter of 23.88 mm. The venturis are instrumented to capture static pressure, differential pressure, total pressure, and temperature. The CI venturi has been previously calibrated for mass flows between 0.79 kg/min to 2.27 kg/min while the TO venturi was calibrated for mass flows between 1.25 kg/min to 3.40 kg/min [3]. The CI venturi had discharge coefficients from 0.937 to 0.963 with a discharge coefficient of 0.956 at the design point of 1.59 kg/min. A maximum Y

uncertainty value of 0.36% was found giving the venturi a discharge coefficient of 0.951 ± 0.019 . The Y uncertainty is used to account for measurement uncertainty and is part of the ASME standards. The TO venturi had discharge coefficients from 0.957 to 0.974 with a discharge coefficient of 0.958 at the design point of 1.59 kg/min. For the TO venturi, a maximum Y uncertainty value of 0.56% was found giving the venturi a discharge coefficient of 0.967 ± 0.009 .

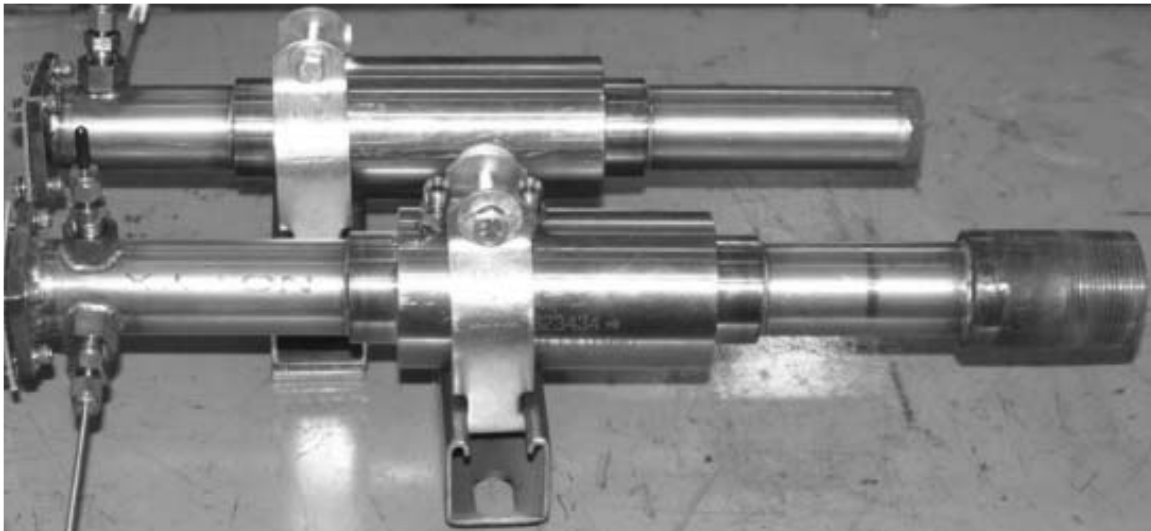


Figure 12 – Venturis with CI on Top and TO on Bottom [3]

Because of the similarity between the projects, the Lapp wave rotor was designed to utilize some of the previously built equipment. Among this equipment was the can type combustor and the venturis. Design work was done when creating the endwalls to ensure the new wave rotor could be coupled to the previously built hardware. The venturis were calibrated for the new design point and the combustor was tested to find the correct output.

III. Experimental Setup

The overarching goal of this research was to design and develop a through-flow wave rotor that produces around 500W of power as part of a Brayton-cycle engine. To accomplish this task, an iterative process was completed from defining goals to testing the system. With the goal power output known, the quasi-one-dimensional code was utilized to perform the computational analysis and create a wave rotor design. The design obtained from the computational code dictated the channel location, rotational speed, and rotor length. With these measurements identified, the rotor and the supporting hardware were designed. Supporting hardware included the endwalls, shroud, and motor mount as well as internal components like the bearing and sealing systems.

Once an overall design had been created in CAD, the rotor was analyzed for structural and thermal stresses. This examination created an iterative process to ensure the components are to industry safety regulations. When the design met all criteria for both project goals and structural safety, the design components were created. The final step included the assembly of all hardware pieces and the integration with a previously built and tested experimental rig by McClearn [3]. Components of this rig were tested to ensure the compatibility of the two systems. This chapter illustrates the steps and processes that went into creating an experimental wave rotor system. These steps include the computational work in Section III.1, hardware design in Section III.2, combustor testing in Section III.3, venturi calibration in Section III.4, rotor spin test in Section III.5, data acquisition in Section III.6, and conclusions in Section III.7.

III.1 Wave Pattern Creation

A large part of this design work included the use of the quasi-one-dimensional code to design the pressure wave pattern that the wave rotor will create. This pattern is a function of many parameters, but is most affected by rotor length and spin rate. First pre-code work determined pressure and temperature ratios as well as mass flow for the goal power output. Next, a combination of three codes was utilized to develop the pattern. These interdependent codes were WRITR, WR, and SIMBC. At the completion of SIMBC, a Matlab script was used to graphically illustrate the produced wave pattern. A final run of the WR code along with the wave pattern illustration was used to determine wave pattern quality. The following parts of this section detail the work that went into creating the wave pattern.

III.1.1 Pre-Code Work

Using project goals, some calculations were accomplished by hand before any major computational work begun. With the goal of producing 500 W, a desired mass flow was calculated from the kinetic energy of the exhaust flow. To complete this computation, first a goal T_4/T_1 was selected. Looking at previous designs and knowing the goal power was lower, a value of 2.5 was decided for T_4/T_1 . This value was used inside WRITR, which produced a P_4/P_1 of 1.15. Both of these ratios consist of the final output port TO, 4, over intake port CI, 1. The locations of these ports along with the TI port, 3, and CO port, 2, are shown in Figure 13. The temperatures and pressures used in the calculations were in the rotor frame of reference and neglect the free stream velocity.

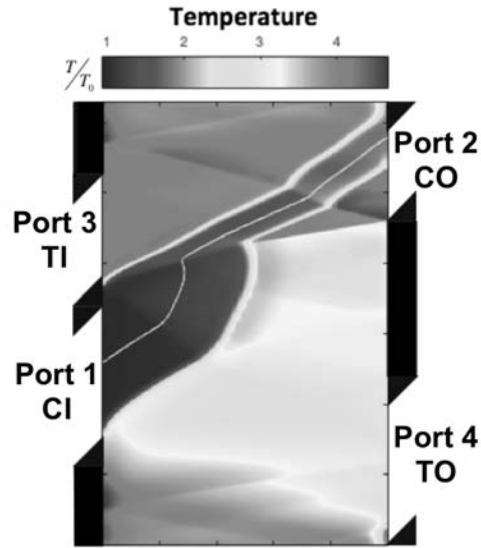


Figure 13 - On Design McClearn x-t Diagram [3]

WRITR is used here to acquire an ideal cycle pressure ratio based on the temperature ratio and other factors. WRITR is part of the quasi-one-dimensional code and will be explained further in the Section III.1.2. This pressure ratio along with the ratio of specific heats (γ), between the constant pressure and constant volume values, allowed the calculation of Mach number shown in Equation 3. Next, using the calculated Mach number (M), specific heat ratio, and an initial T_0 (calculated using the T_4/T_1 ratio) the static exit temperature of TI was calculated as seen in Equation 4. With static exit temperature known, Equation 5 was used with R and γ to calculate a , local sonic velocity. Exit velocity was then calculated using Equation 6. Next, specific kinetic energy was calculated using Equation 7. Lastly, Equation 8 was used to calculate the required mass flow out of Port 4, TO, to reach the design point. This equation used the specific kinetic

energy and the goal power. These equations created a goal mass flow out of Port 4 of 1.06 kg/min.

$$M = \sqrt{\frac{2}{\gamma-1} \left(\left(\frac{P_4}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right)} \quad (3)$$

$$T_s = \frac{T_4}{1 + \frac{\gamma-1}{2} M^2} \quad (4)$$

$$a = \sqrt{\gamma R T_s} \quad (5)$$

$$v = Ma \quad (6)$$

$$ke = \frac{v^2}{2} \quad (7)$$

$$\dot{M} = \frac{P}{ke} * 60 \quad (8)$$

With a goal mass flow and temperature ratio, the quasi-one dimensional (Q1D) code was utilized to create a wave pattern with these values. The following sections describe the different parts of the code and how they were used.

III.1.2 WRITR Script

WRITR is the first of three computational scripts used to create the pressure wave cycle. WRITR is used to approximate ideal port exit conditions and port timing based on thirteen input parameters. Four of these parameters were utilized and altered for this study while the others were held constant. The four parameters this study employed were the non-dimensional opening time (THETOT), expansion ratio (EXPRAT), cycle temperature ratio (T4T1), and a toggle deciding how much of the cycle was analyzed (UPCYC). These four parameters are discussed further in the following paragraphs. The nine parameters held constant consisted of two parameters determined previously by

McClearn and seven parameters that were used in previous NASA studies. The two inputs dictated by McClearn included the spacing between the CI and TI ports (IDLAND) and the ratio of specific heats (GAM). The ratio of specific heats was calculated using a weighted average between the fresh intake flow and the products of the combustor. A γ value of 1.348 was the outcome of the weighted average. The spacing parameter has a minimum value of 1.0 and relates the number of non-dimensional opening times, THETOT, which can fit between the CI and TI ports. As with the McClearn study, the value for IDLAND was kept at the minimum of 1.0.

The primary input parameter altered inside WRITR was the T4T1 value. This term or the cycle temperature ratio was based on the a priori conditions decided previously in Section III.1.1 to calculate required mass flow. The ratio is based on the final exhaust gas, TO, over the initial intake air, CI. The ratio of 2.5 was utilized and not altered.

The input parameter UPCYC was used as a toggle between doing a full or partial WRITR run. This difference allows for timesaving by allowing the user to determine approximate opening times and expansion ratios before analyzing the whole cycle. The half cycle produces values for Port 1, CI, and Port 4, TO compared to values for all four ports.

Two other parameters were altered in order to achieve a successful result from the WRITR script. These parameters were manipulated to allow for better results in the following WR script. The first modified factor was the non-dimensional opening time, THETOT. This opening time was how long the initial CI port was open on a time or distance scale. The opening time was used to create a reference scale for calculating the

other port timings. Because of high mass flows calculated initially, the non-dimensional opening time was lowered from an original value of 0.574. Values tried ranged from 0.450 to 0.574 with a final value of 0.500 being utilized.

The other manipulated value was the expansion ratio, EXPRAT. This parameter characterizes the gas expansion between the TI and TO ports. This expansion represents the “turbine” part of the wave rotor cycle and powers the compression done on the intake air. An initial EXPRAT value of 0.300 was initially used when running WRITR. Other values ranging from 0.280 to 0.450 were attempted with a final value of 0.300 being used. These values were tried till a high enough port four-stagnation pressure (P_{04}) along with a low enough port four-stagnation temperature (T_{04}) was acquired. It is important to note that the range of THETOT and EXPRAT values stems from the overall code requiring many iterations to create a successful wave pattern.

Figure 14 displays the final inputs used for this project. The overarching path for the code goes from WRITR to WR to SIMBC, but many times there would be multiple iterations between WRITR and WR before SIMBC was attempted. A SIMBC result scan send the user back to either the WR or WRITR. The repetitive process occurs based on what each part of the code adds to the physics and analysis inside the code. One more item to note is the difference in the friction factor, FSSC, from the McClearn study to this project. The previous wave rotor design used 0.298 as the friction factor while this project used 0.483. This difference comes from later discussions between Dr. Dan Paxson and the AFIT wave rotor group [35].

EXPRAT=	0.30000	RATIO OF SPECIFIC HEATS ~1.3-1.35
GAM=	1.34800	SPACIAL STEP*
DX=	0.02000	TIME STEP*
DTI=	0.01000	NON-DIMENSIONAL OPENING TIME
THETOT=	0.50000	FRICTION FACTOR
FSSC=	0.48260	# OF INTERIOR CELLS PLUS ONE*
PNTS=	51	INLET SHOCK PRESSURE RATIO*
PRI=	1.00000	EXHAUST CLOSING TIME 1.0:U=0.0 0.0:U=U_UNIFORM*
NCT=	1.00000	EXHAUST TEMP / INLET TEMP
T4T1=	2.50000	COOLING BLEED- INLET FLOW / EXHAUST FLOW*
BLEED=	1.00000	1=FULL CYCLE 0=ONLY BOTTOM
UPCYC=	1	# OF O.T.'S BETWEEN INLET CLOSE AND DELIVERY OPEN
IDLAND=	1.00000	

Figure 14 - Final WRITR Script Inputs

The completion of WRITR produced an ideal, non-dimensional wave rotor cycle. Information created includes temperature ratios, pressure ratios, locations, and sizes for each port. The output file generated required minor modifications before being utilized in the next computational script, WR.

III.1.3 WR Script

WR is the second of three computational scripts used to create the pressure wave cycle. WR is used to calculate the ideal wave rotor cycle with the inclusion of physical geometries and real world effects. In order to implement physical geometries, the first step inside WR was the referencing of port one, CI, conditions. The conditions referenced in this step consisted of the temperature and pressure values. These ratios allow for physical geometries to be employed with the addition of other values as well as an easier to comprehend code output. The ease of comprehension stems from the CI conditions being atmospheric; therefore all the values are ratios of normal atmospheric conditions. For this work, these values are 289 K and 101.33 kPa.

With the WR inputs referenced correctly, the script was now used to add in physical geometries and real world effects. Rotor geometries were altered inside a separate excel file that calculated the code inputs. The rotor dimensions manipulated were length (L), channel height (h_c), outer diameter (D), number of channels (n), and rotational speed (ω) in revolutions per minute. The code inputs calculated were rotor angular velocity (Ω), angular distance to fully open channel to a port (θ_{ot}), Reynolds number based on CI's sonic velocity (Re^*), hydraulic diameter (d^*) over length (DONL), and channel height over length (HONL). The relations between the rotor dimension and the code inputs can be seen in Equations 9 – 13.

$$\Omega = \frac{\omega \pi L}{360 a} \quad (9)$$

$$\theta_{ot} = \frac{2\pi}{n} \quad (10)$$

$$Re^* = \frac{\rho a L}{\mu} \quad (11)$$

$$DONL = \frac{d^*}{L} \quad (12)$$

$$HONL = \frac{h_c}{L} \quad (13)$$

With the values referenced to Port 1 and then relations between the physical geometries and code inputs created, the next step consisted of running WR with different sized rotors. This part of the code required the most time based on there being few evident trends. During this process, the main parameters were rotor length and spin rate. These two parameters are inversely related where a shorter rotor will require a faster spin rate. Other factors such as channel height, number of channels, and overall diameter had relatively less effects on the code outputs. The goal when running the WR script was the matching of the port mass flows as well as acquiring correct temperature and pressure

ratios. Matching of mass flows consisted of changing the inputs till mass flow at Port 1 (CI) and Port 4 (TO) were close while also getting mass flow at Port 2 (TI) and Port 5 (CO) to be close. These port numbers are different then what has been used before and are strictly for the code. The process of matching mass flows was a juggling match where getting one to match was easy, but getting both to match was a long endeavor. Some leeway is present as the next script, SIMBC, has the ability to match the flows as long as the solution is viable. Methods to match the mass flows as well as criteria that determine a successful code output are described below in Section III.1.6. The final output of the WR script is shown in Figure 15 with the main results highlighted. The main rotor parameters used in this output were a length of 3.73 cm and a spin rate of 60,000 RPM.

	MF	EF	MO	UB	PB	TB	P0	T0	ALPH
1	0.656	0.650	1.232	0.564 0.513	0.802 0.802	0.944 0.944	0.999	0.999	0.002
2	0.873	2.730	3.474	0.724 0.686	2.968 2.967	3.047 3.047	3.327	3.138	-0.001
3	0.000	0.000	0.000	0.000 0.000	0.000 0.000	0.000 0.000	0.000	0.000	0.000
4	0.713	1.837	1.962	1.056 1.031	0.979 0.979	2.394 2.394	1.324	2.588	-0.001
5	0.940	1.917	2.966	0.614 0.569	3.150 3.150	1.983 1.983	3.573	2.048	-0.001
6	0.000	0.000	0.000	0.000 0.000	0.000 0.000	0.000 0.000	0.000	0.000	0.000

Figure 15 – Final WR Script Outputs

Beta values were altered inside the WR code in order to keep the endwall port incidence angles correct. These angles are similar to inlet guide vanes in a turbine engine where they direct the flow to help spin the rotor and have the least amount of impingement. The angles are detailed further in Section III.2.3. The UB values in the

script output (boxed red in Figure 15) are converted to an angle in radians and fed back into the input file as beta values for each port. The far right column of the script output dictates how well the angle is performing with an alpha value of zero being the ideal port incidence angle.

With the correct pressure ratios (boxed blue in Figure 15), temperature ratios (boxed orange), alphas (boxed green), and matching mass flows (boxed black) the code can move to the next step. The process of analyzing the WR output and why these design values work is discussed in Section III.1.7. WR is used one last time after SIMBC to create a file that is used to create the x-t diagrams.

III.1.4 SIMBC Script

SIMBC is the third and final computational script used to create the pressure wave cycle. The final script added in loss mechanisms as well as coupled the CO and TI ports with an external heater model. To run SIMBC, the input file from WR was used and had an additional ten lines of inputs added near the end. These supplementary inputs allowed the SIMBC script to add in the losses and heater model.

The different loss mechanisms were added in one type at a time. This process was done to allow the code to converge from each previous solution. The first loss type consisted of two gap values. These values were PGAP and WGAP and both had values of 0.016. Both values were calculated based on the ratio of wall spacing over channel height [35]. The wall spacing is the distance given between the end of the rotor and the endwall face. For this design, a value of 0.127 mm was used for the wall spacing and was based

on minimizing the distance while ensuring the two parts will not make contact. The channel height is the radial distance that each passage uses. This distance was altered during the WR script and has a final value of 7.62 mm. The second loss type added was the discharge coefficient, CD. A value of 0.8 was used for CD, which coincides with what McClearn used and what Dr. Paxson recommends [3].

With the loss mechanisms added in, QCORR and BLOSS were utilized in the code. These two parameters were altered until the temperature outputs were back to their previous values from WR. The temperatures must be relatively close to the previous values to ensure the associated temperature ratios are correct. QCORR is a non-dimensional heat addition term that is used to model the burner loop between the CO and TI ports. This input models the burner by being a non-dimensional enthalpy term based on the temperature ratios of both CO and TI. BLOSS was used to account for the incomplete combustion and losses that impede typical combustors. The combination of QCORR and BLOSS dictates properties of the external burner to accommodate for the rotor losses.

Other parameters added in during SIMBC included PRANTL, WALONL, RHLVC, RHLVB, RHLVV, and AV. The addition of PRANTL allowed for heat transfer to the rotor walls by including a Prandtl number. WALONL improved this calculation by being the ratio of rim thickness over the length of the rotor. The other four inputs were increased slowly until the solution could no longer converge. These four inputs represent non-dimensional fluidic capacitance terms as ratios to the volume of different regions inside the wave rotor system [35]. Increasing these terms in the code till failure provided accelerated convergence. The final values for these inputs are shown in Table 1. During

the addition of all six inputs, QCORR and BLOSS were manipulated to sustain the desired code outputs. The desired outputs are the temperature and pressure ratios generated from WR. Final values for QCORR and BLOSS were 2.5 and 1.3 respectively. The final code output from SIMBC can be seen in Figure 16.

Table 1 - Final SIMBC Inputs

Input	Value
PRANTL	0.70
WALONL	0.03
RHLVC	0.50
RHLVB	0.05
RHLVV	0.009
AV	1.5

				P0		T0			
	1			1.00000		1.00000			
	4			0.86656		2.71875			
	2			2.96071		3.83837			
	5			2.79546		3.83837			
	CAV			1.70346		3.11535			
	MFFRAC			1.42466					
	WDOTIN			-0.00601					
	EPSILON			0.41441					
	KLOSS			0.66662					
	QCORR			2.50000					
			MFCORR	P4/P1	T4/T1		OMEGA		
			0.37259	1.13275	2.71875		0.70168		
	MF	EF	M0	UB	PB	TB	P0	T0	ALPH
1	0.427	0.419	1.014	0.439	0.876	0.960	1.000	0.994	-0.000
				0.352	0.876	0.960			
2	0.608	2.325	2.725	0.784	2.658	3.729	2.966	3.836	-0.000
				0.739	2.658	3.729			
4	0.427	1.157	1.390	0.823	0.954	2.601	1.133	2.719	0.000
				0.780	0.954	2.601			
5	0.608	1.587	2.282	0.654	2.813	2.548	3.145	2.622	-0.001
				0.599	2.813	2.548			

Figure 16 - Final SIMBC Script Outputs

With the final values being calculated with the SIMBC script, two steps were completed to analyze the numbers. The first of these steps was transferring the outputs to an Excel file for ease of understanding. Since the temperatures and pressures are ratios,

this allowed the user to see actual values based off the atmospheric conditions at CI. These outputs were also used with Equations three through eight to calculate the overall power output of the system. The next process consisted of taking the SIMBC input file and making it compatible with the WR script. Once compatible, WR was run to create the output file used to illustrate the wave pattern inside Matlab. This process and results are shown in Section II.1.5. The two final outputs of the code, the x-t diagram and the Excel document, determine whether the solution is valid or if more design iterations are needed.

Completion of the CFD required a final check of the code outputs. These outputs and the physical geometries associated are shown in Table 2 and compared to the McClearn design. T_4/T_1 , P_4/P_1 , and mass flow rates were computed to be 2.72, 1.13, and 1.09 kg/min respectively. Though not at the original goal ratios, these values and the additional code parameters created a working wave pattern. These values were used with Equations three through eight to calculate the kinetic energy of the exhaust. These calculations result in a kinetic energy of approximately 500 Watts. When comparing the ratios to the preliminary calculations, the higher temperature ratio raises the local sonic velocity inside the channels and the exhaust velocity. The lower pressure ratio reduces the calculated Mach number. The reduced Mach number and the increased local sonic velocity are used to calculate the specific kinetic energy for a given mass flow. Equation 2 was used to calculate the compression efficiency, which is included in Table 2. Overall these results satisfied the project requirements and allowed the design to move to the following step of checking the x-t diagrams.

Table 2 - Wave Rotor Design Comparison

Parameter	Current Design	McClearn Design [3]
Rotor Length	3.73 cm	6.35 cm
Rotor Diameter	3.56 cm	4.19 cm
Channel Height	0.76 cm	0.64 cm
Rotational Speed (ω)	60,000 RPM	38,500 RPM
Rotor Channels	23	29
T_4/T_1	2.718	3.063
P_4/P_1	1.132	1.315
Exit Mass Flow Rate	1.09 kg/min	1.59 kg/min
Compression Efficiency	21.0%	23.9%

III.1.5 Matlab Visualization and Analysis

With the three computational scripts complete, WR was utilized one more time to create an output file. This file was read using a Matlab script created by Dr. Paxson and edited by McClearn. The Matlab script creates the x-t diagrams shown previously in Section II.1.1. These x-t diagrams are useful for understanding the physical processes inside a wave rotor as well as determining the quality of a particular wave pattern. Six total x-t diagrams were created with the primary illustrations being static pressure and static temperature. The static pressure and static temperature x-t diagrams for the on design wave rotor are shown in Figure 17. On design entails spinning at the correct rotational speed and having the correct heat addition as dictated by the code. The other parameters depicted in x-t diagrams include density, velocity, total/stagnation pressure, and total/stagnation temperature.

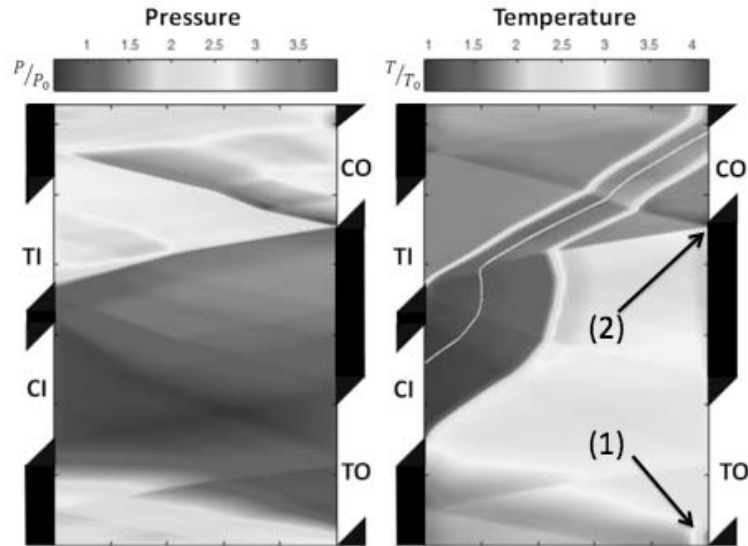


Figure 17 - x-t Diagrams - Static Pressure and Static Temperature

The pressure contour plot in Figure 17 (left) clearly demonstrates the increase in pressure as the cycle begins (bottom left) at the CI port and exits out the CO port (top right). This diagram also enables visualization of additional pressure wave propagation between the two endwalls. Each additional wave reflection boosts the pressure and temperature of the incoming air. The temperature contour plot (right) includes a streamline demonstrating the flow of the fresh inlet air from CI to CO. A few key aspects of the contour plots reveal if the overall wave rotor has been designed successfully. The first is the expectation for a small amount of fresh air in the bottom right of the contour plots (Shown as (1) in Figure 17). With an overall goal of exiting all the fresh air out through the CO port, it is critical to keep this spot to a minimum size. The other key aspect concerns the location of the main hammer shock. The desire is for this shock to reflect just as the CO port opens (Shown as (2) in Figure 17). Rotational speed and port locations are critical for correct shock reflection locations.

As mentioned before, further x-t diagrams were created from the Matlab script. Density and velocity are shown in Figure 18 while stagnation or total pressure and temperature are shown in Figure 19. Similar to the stagnation pressure and temperature x-t diagrams, the density and velocity plots illustrate the motion of the internal shock waves. The density plot shows how impactful the main hammer shock is as it crosses the fresh incoming air (Shown as (1) in Figure 18). The velocity diagram is in the rotor frame of reference and illustrates the lower density and higher velocity exhaust gas entering the rotor (Shown as (2) in Figure 18) and accelerating as it is expanded further while exiting the rotor (Shown as (3) in Figure 18).

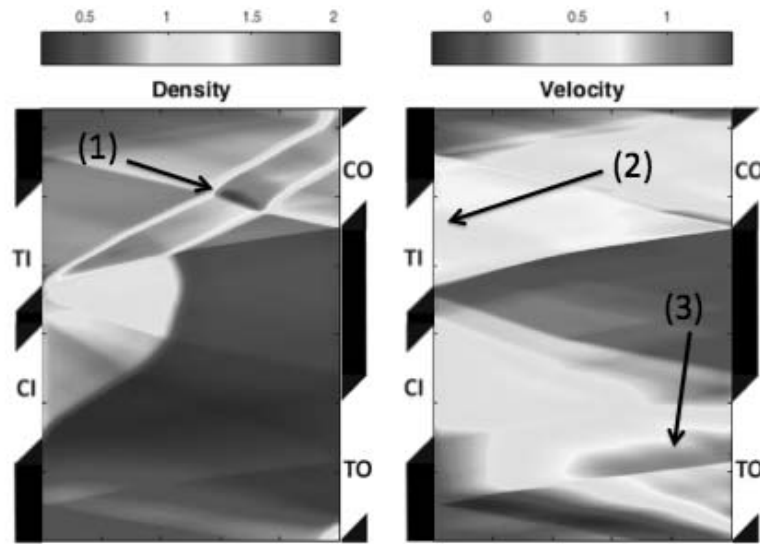


Figure 18 - x-t Diagrams - Density and Velocity

The x-t diagrams for stagnation pressure and stagnation temperature in Figure 19 are thoroughly similar to the static pressure and static temperature plots. The one main difference is found in the pressure plot where the incoming exhaust gas (Shown as (1) in Figure 19) dominates over the stream of fresh air.

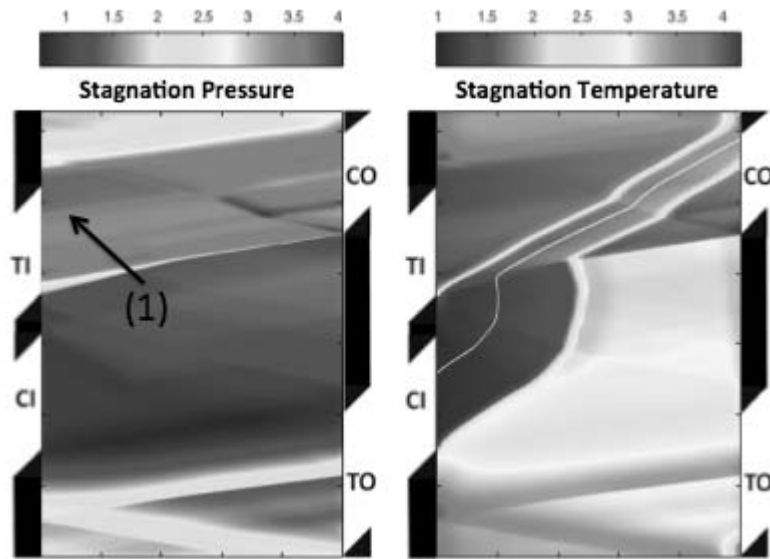


Figure 19 - x-t Diagrams - Stagnation Pressure and Stagnation Temperature

Analysis of the different x-t diagrams was the last step of the code process to ensure a successful wave rotor design point. As mentioned previously, the code was an iterative process that required countless attempts before a design was chosen. The following section discusses some of the methods tried as well as different design criteria that were used.

III.1.6 Code Methods Used and Design Criteria

While running the WR script inside the code, multiple methods were attempted to create a successful wave sequence. The first of these methods involved scaling the design of McClearn [3]. This attempt took all of the physical geometries and scaled them accordingly. The parameters used by McClearn are shown previously in Table 2 in Section II.1.4. The length, chamber height, and diameter were all cut in half while the rotational speed was doubled. The rotational speed was increased based on the inverse

relation speed shares with length. Unfortunately this process did not return any viable results. The second method attempted used wave timing to calculate the physical geometries. First, total time (t_s) for one full revolution of the rotor was calculated using Equation 14.

$$t_s = \frac{60}{\omega} \quad (14)$$

From here the time (t_p) for a pressure wave to traverse the channel length was calculated using Equation 15 with N being the number of back and forth waves. During the iterative part of this calculation, a range of N values from six to nine was tried.

$$t_p = \frac{t_s}{2*N} \quad (15)$$

The two in the denominator accounts for each rotation having two complete wave cycles for every one revolution. With t_p known, the local sonic velocity of the air, a , was used to calculate overall length needed. Equation 16 shows this calculation. The local sonic velocity came from previous calculations that were used in Section III.1.1 to find the kinetic energy of the exhaust.

$$L = t_p * a \quad (16)$$

With length and rotational speed being the main parameters for designing a successful wave rotor, these calculations were used to create a test matrix. The independent and changing variables for this test matrix were n and ω , while the output was overall channel length. These calculations and test matrix allowed for a starting spot when running the code. While results were closer than just starting from an arbitrary point, countless iterations were still completed until a design was chosen.

The WR results from Figure 15 in Section III.1.3 and the physical geometry results from Table 2 in Section III.1.4 were used to describe what makes this design applicable. The first and easiest check to confirm the design will work was the location of the highest temperature and highest pressure. P05 must have the highest pressure while T02 must be the highest temperature. P05 referenced the pressure out of the CO port while T02 referenced the temperature going into the TI port. The temperature and pressures as well as the port locations are shown in Figure 17 and Figure 19. Both of these requirements were met during WR as P05 was 3.573 and T02 was 3.138. Furthermore, the temperature values can be used to make sure they are in a materials operating range. For this project, there was an underlying goal of keeping the temperature ratio between T02 and T01 under 4.0. For this design, the final temperature ratio out of SIMBC had a value of 3.14 resulting in a max temperature of 1108 K. Based on the maximum temperature, Inconel was chosen as the material for creating the rotor, endwalls, and the containment shroud.

A second design criterion was the pressure differential between Port 5 and Port 2. In general, pressure drop of 5% is needed between these ports. This differential is required to prevent back flow through the wave rotor. The pressure drop for this design had a value of 5.7%. The last check using the temperatures and pressures comes from the T4/T1 ratio and the P4/P1 ratio. As mentioned before in Section III.1.4, these two values came from design criteria that were decided pre-code work. The goal temperature ratio was 2.5 and a final value of 2.72 was acquired. The target pressure ratio was 1.15 and a final value of 1.13 was obtained. Though not quite on target, the new ratios fashioned a successful wave pattern.

Other design criteria came from looking at the relative geometry of the design. The first criterion is the ratio between length and outer diameter. Though not an exact rule, a ratio of around one is desired. This ratio stems from other successful wave rotor designs and discussions with Dr. Paxson [35]. Other options were tested out during the design process of this rotor including long narrow rotors and short wide rotors. None of these designs worked when it came to the respective design performance in terms of temperatures and pressures.

Another geometry design criteria came from the inner and outer radius of the channels. Equation 17 was used to create this ratio with the code inputs where D is outer diameter and h_c is channel height. A ratio of 0.6 or higher is ideal for a successful design. This stipulation comes from discussions with Dr. Paxson [35] and previous successful wave rotor designs [3, 32]. The radius ratio for this project is 0.57.

$$Radius\ Ratio = \frac{\frac{D}{2} - h_c}{\frac{D}{2}} \quad (17)$$

Some designs have gone near 0.5, but overall this is ill advised. The reason for needing this ratio comes from magnification of centripetal forces as the hub to tip ratio declines. The combination of different centrifugal forces along side with cold and hot interfaces can create fluidic disparity. This can create a situation where substantial radial pressure gradients would encounter substantial axial density gradients. The combination of the two flows will create an area of baroclinic velocity. This baroclinic velocity comes from adverse pressure gradients found in the flow. The results from these fluid interactions include unstable interfaces that smear massively and the fluid not going out of the correct ports [35]. Furthermore, incident losses vary largely from the hub to the

tip based on the different circumferential speeds and as such it could be that the flow is not 1-D for a given passage.

The last design criterion is the overall goal of the WR code. This overall goal is the matching of the mass flows. For the cycle to converge inside SIMBC, MF5 needs be close to MF2 and MF4 needs be close to MF1. This process is long and can take countless iterations of the geometry and spin rate. Though they do not have to be spot on, mass flow matching is the driving point of this design step. For this design, the MF4 and MF1 values were off by 0.057 where MF4 was 0.713 and MF1 was 0.656. The MF5 and MF2 values were off by 0.067 where MF5 was 0.940 and MF2 was 0.837. The values here are unit less as they are ratios of the inlet port conditions.

III.1.7 Further Matlab Analysis

McClearn used the Matlab script discussed previously to further analyze the flow inside the wave rotor. This extra analysis creates plots that compared pressure, temperature, velocity, and density across each port as well as comparing Mach number, pressure, and temperature as one cycle is completed. These plots can be viewed in Appendix A.

III.2 CAD Design

With the computational code process complete, a rotor and the supporting parts were designed using CAD software. For this project, Solidworks 2013 was used to create these designs. Other software, NX9 and Dyrobes, was used to do structural analysis and rotor

dynamic analysis respectively. The following sections will illustrate the process used to create the rotor in Section III.2.1, containment shroud in Section III.2.2, endwalls in Section III.2.3, and motor mount in Section III.2.4. The motor used for spinning the wave rotor is also discussed in Section III.2.4.

III.2.1 Rotor Design

The completion of the computational code dictated physical geometries for the rotor channels. These geometries included length, number of channels, inner radius, and outer radius. These characteristics are shown in Table 2 in Section III.1.4. The number of channels was kept prime in order to avoid acoustic resonance. For this design, the channels are straight through the rotor.

Other design parameters for the rotor were based on computational analysis and lessons learned from previous designs. Rotor design parameters that were left to manipulation included the channel webbing thickness and outer diameter that were based on static stress analysis, and shaft integration that was determined from rotordynamics. Static stress analysis was done using NX9 while the rotordynamics analysis was done through a shaft model created in Dyrobes. The static stress model was developed from the CAD model created in Solidworks. Inside NX9, the model was given rotational speed, temperature, two fixed translational constraints at the bearing locations, and material properties. The rotational speed was 70,000 RPM, which was roughly 115% of the design point and the highest speed that the rotor may encounter during testing. The temperature was 900K across the entire model corresponding to the cavity temperature

calculated from the computational code. Figure 20 shows the NX9 model before the analysis was started. The blue lines are the translations constraints that fix the rotor in position so it can only rotate along the centerline. Not shown in this image are a plethora of added temperature inside the rotor channels.

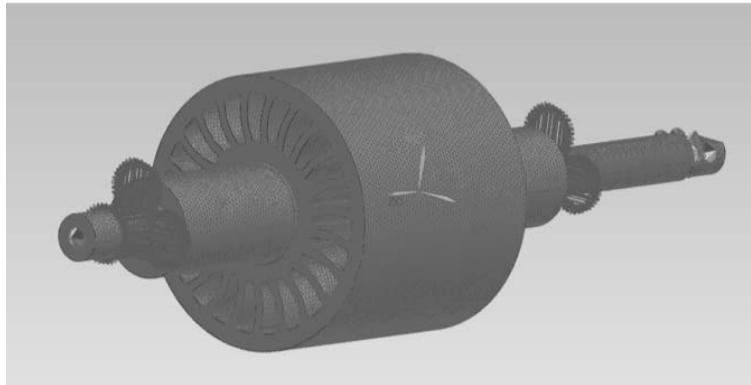


Figure 20 - NX9 Model Setup

The values of interest during the testing were displacement, max principle stresses, and max shear stress. A maximum displacement of 0.00329 mm was found at the far end of the right shaft extension. Figure 21 shows the results from the max principle stresses, 485 MPa, and max shear stresses, 326 MPa. Both stress ceilings were

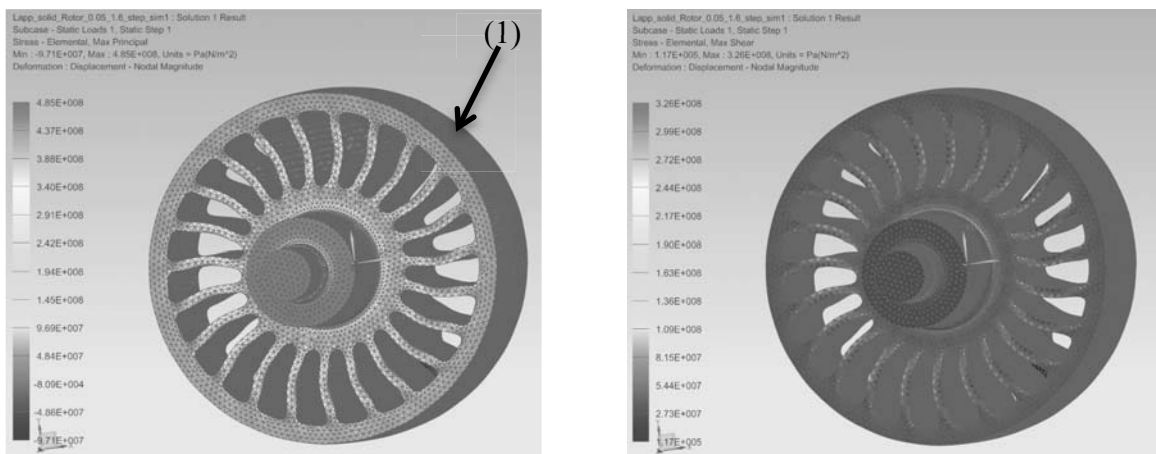


Figure 21 – Max Principle Stress (Left) and Max Shear Stress (Right)

found inside the rotor where the webbings connect to the rotor. Based on these stresses and the material properties of Inconel 625 at temperature, the rotor had a safety factor of 1.52.

To achieve these values, both the channel web thickness and the outer diameter thickness were manipulated. The channel web thickness changed from 0.254 mm to 1.27 mm and the outer diameter thickness decreased from 5.1 mm to 2.54 mm. The largest outer diameter was minimized while allowing enough thickness for material to be removed for balancing purposes. The location of material removal (Shown as (1) in Figure 21) and was completed by The Balancing Company, Inc out of Vandalia, Ohio. The removable thickness gave the rotor a final outer diameter of 40.64 mm. The minimizing of the outer diameter thickness helped reduce the forces that the rotor will encounter when spinning at high RPM values. Furthermore, web thickness was reduced while making sure the rotor could withstand the rotational and thermal stresses. Further design modifications were done to the channels to ensure the rotor will survive the high rotational speeds and the high temperatures. The first of these was to curve the channel webbing in between the channels. The curve has a radius of 2.54 cm was added in the direction of rotation to allow for thermal growth and not impinge on the incoming flow. Lastly, the channel corners were rounded with a radius of 0.635 mm to reduce stress concentrations and allow for easier machining.

The other parameter open to manipulation was the shaft integration. The final shaft design utilized lessons learned from previous wave rotors as well as a Dyrobes, rotordynamic software. Past designs helped decide the type of shaft, while the software dictated the shaft geometries and bearing locations. Previous wave rotor designs by

Mataczynski and McClearn encountered difficulties during spin up [3, 32]. These issues occurred from exciting detrimental modes while trying to reach the spin rate design point. The first design, created by Mataczynski, utilized a press fit shaft with a cantilevered rotor for a rotor spinning at 34,500 RPM. The next design, by McClearn, included an integrated through shaft with the rotor supported at both ends. This shaft was hollow to help with uneven growth and thermal stresses. Furthermore, this design utilized one bearing system along with an air bearing and had a rotational design point of 38,500 RPM. Lessons from both of these designs are incorporated into the current projects design.

The current shaft was a solid through shaft supported at both ends by bearings as seen in Figure 22. The bearing locations (Shown as (1) in Figure 22) were positioned away from hot gasses while minimizing rotational issues dynamically. The solid through shaft (Shown as (2) in Figure 22) has an outer diameter of 35.56 mm. The first solid shaft extensions (Shown as (3) in Figure 22) have an outer diameter of 14.73 mm and extend 25.40 mm based on the rotor dynamic analysis discussed in this section. The right hand

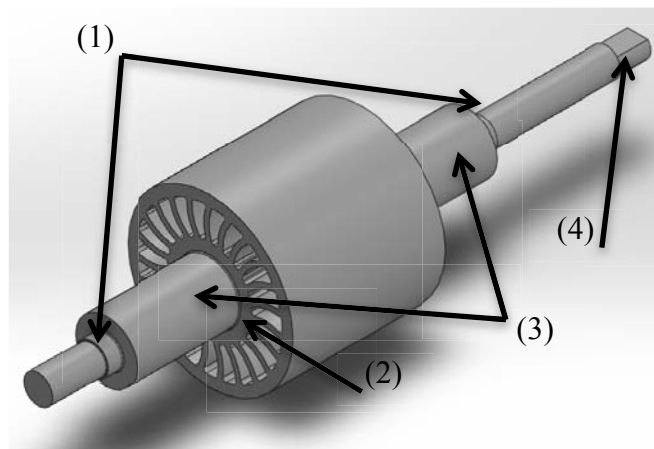


Figure 22 - Wave Rotor CAD

side extension was longer to allow for the motor connection and has a flat for the Love Joy coupling setscrew (Shown as (4) in Figure 22).

The diameters of the shafts as well as the solid through shaft were all determined with the rotordynamic software Dyrobes. A shaft model was created inside Dyrobes that included shaft geometries, bearing locations, and bearing properties as shown in Figure 23. The bearing properties were used inside the program to calculate the bearing stiffness. This stiffness was found to be roughly 160,000 lbf/in.

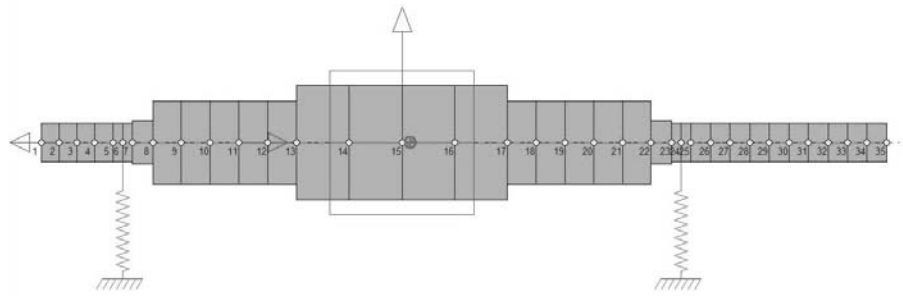


Figure 23 - Dyrobes Shaft Model

Following the advice of Dan Gillaugh of AFRL/RQTI, three different analyses were completed. These three different analyses consist of a critical speed map, forced response analysis, and a peak-to-peak deflection calculation. The critical speed map, shown in Figure 24, illustrates where the three main detrimental modes would be excited based on rotational speed and bearing stiffness. The black circle represents the operating design point of the rotor. Mode 1 occurs at 83,404 RPM, which is 139% of the design point. For this design, the bearings used are 618/7 Deep groove ball bearings, single row created by SKF. These bearings were chosen for their high operating speeds and small size.

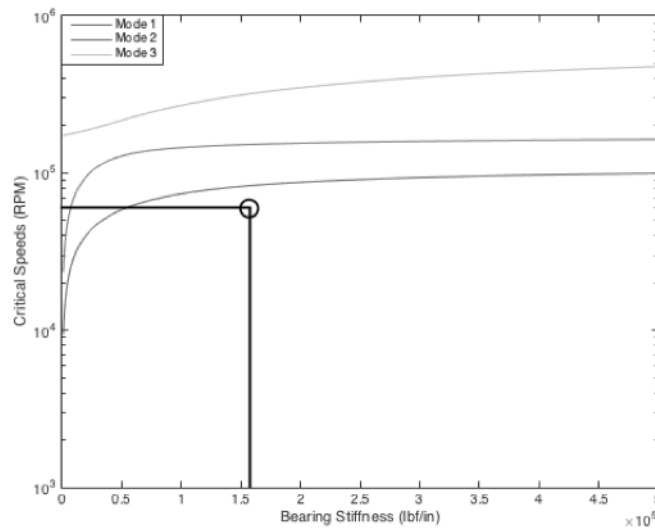


Figure 24 - Critical Speed Map

The second rotor dynamic analysis done was a forced response analyses that calculated the axial force each bearing will feel at different rotational speeds. This analysis required the addition of an eccentricity to the input file. Eccentricity is an estimated distance that the rotor center of gravity is away from the center of the rotor. Based on the size of the rotor, an eccentricity of a $\frac{1}{4}$ millimeter was used. The off centered distance along with the rotational speed and shaft properties produce an axial force. As shown in Figure 25, the bearings will encounter a force of approximately 90 N at the operating design point of the rotor, 60,000 RPM as indicated by the black circle. The forced response analysis was used to ensure the bearings would withstand the axial force and to decide what wave washers to utilize. The bearings will survive the axial force as the bearings have a dynamic load capability of 780 N and a static load capability of 260 N. The wave washers will be discussed later in this section.

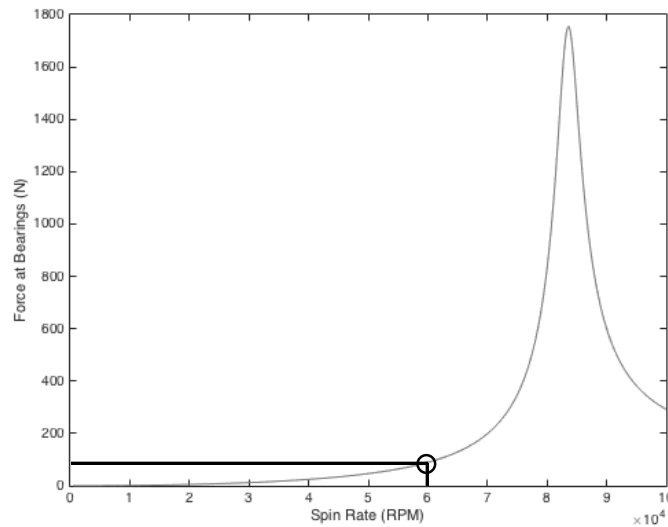


Figure 25 - Forced Response Analysis

The next analysis was a peak-to-peak deflection calculation. This computation required the eccentricity as well as the weight of the rotor. Figure 26 illustrates the peak-to-peak deflection found at each bearing location. The calculation found that each bearing would have a peak-to-peak deflection of 6.3 mm or a deflection of 3.15 mm in one direction.

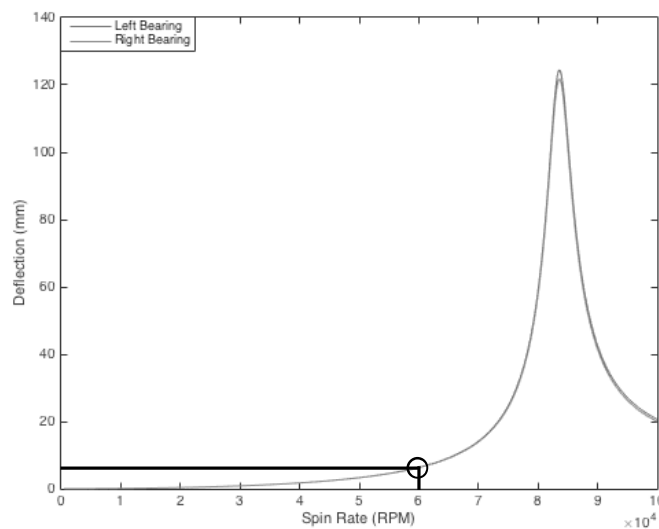


Figure 26 - Peak-to-Peak Deflection

The three results from Dyrobes were used to alter the solid through shaft as well as the first shaft extensions. Criteria for a successful design were based on the standards of RQTI. These benchmarks included no detrimental modes in the operating range and no detrimental modes within 20% of the max steady state speed (60,000 RPM). Both of these conditions are met as the first detrimental mode was outside the operating range and was at 139% of the max steady state design point (83,400 RPM).

This shaft requires a bearing system that will maintain the required spacing while allowing the system to flex during the rotor's thermal growth. To accomplish these goals, the bearing system utilizes shaft and endwall abutments, multiple bearings, and multiple types of washers. Figure 27 illustrates the relative locations of the bearing system of the wave rotor hardware.

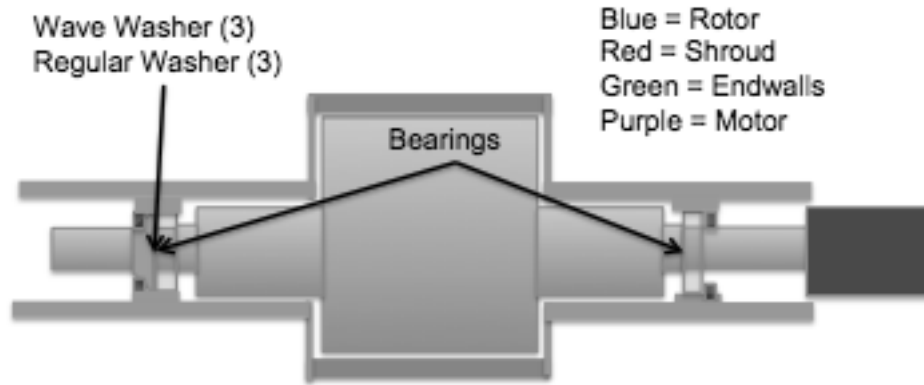


Figure 27 - Shaft Setup

The design employed two bearings located 28.91 mm from the endwall to rotor interface on the left and right side. This distance was maximized to keep the bearings from the hot gasses found inside the wave rotor. These bearings have a maximum speed 100,000 RPM and are small enough to work with the tight geometries of the endwalls.

The bearings were kept in place using abutments on the wave rotor and on either of the endwalls. The wave rotor shaft extensions have the required abutment diameters for the inner bearing race and utilized interference fits to keep the bearings in place. The inner shaft of each endwall has the required geometries for the outer bearing race abutment. Locational clearance fits inside each endwall contain the bearings. The bearings are held in place between the abutments on the rotor shaft and the abutments inside the endwall shafts. Based on the different type of fits, the bearings were pressed onto the rotor shaft before being fitted inside each endwall.

To account for axial growth and tolerance build up, a combination of wave washers and regular washers were utilized. These washers were used inside the left endwall to allow the rotor to grow to the left and to provide initial preload on the bearings. The right endwall contains just a bearing to fix the rotor positioning on this side. As shown in Figure 27, the left hand side contains three wave washers and three normal washers. Seastrom Manufacturing Co., Inc created the wave washers and regular washers. The regular washers are used to separate the wave washers and ensure the correct amount of preload. The regular washers product number was 5710-566-30 and had a thickness of 0.762 mm. Two types of wave washers were required for this design to ensure the correct bearing and endwall abutment, allowing enough flex, and withstanding the forces during operation. The first wave washer used was model 5805-25 and had the correct inner diameter to sit on the bearings outer-race. This first wave washer had a max deflection of 0.356 mm and could withstand 4 N of force. The second and third wave washers used were model 5806-116-2 and had the correct outer diameter to sit on the endwall abutment. The second and third wave washers had a max deflection of 0.305 mm

and could withstand 110 N of force. The three wave washers provided 0.966 mm of flex. As shown in Figure 25, the bearings will have 90 N of force at the design rotational speed. This force and the additional bearing preload means the first wave washer was flexed before the other two wave washers. The stack up of these washers is shown later in Figure 36.

III.2.2 Shroud Design

The shroud was designed after the rotor and was used to encompass the spinning rotor as well as be an attachment point for both endwalls. An isometric view of the shroud CAD is shown in Figure 28. The shroud must allow for thermal growth in both the radial and axial direction as well as ensuring the gap distance (discussed in Section III.1.5) was adequate. To account for radial growth, the inner diameter of the shroud was 41.91 mm allowing a gap of 0.635 mm between the rotor outer diameter. This distance was adequate as the maximum thermal radial growth of the rotor was 0.381 mm. The endwall has an extra 0.762 mm of length where the rotor is situated. This extra length consists of the space for the gaps, thermal growth, and leeway for tolerance buildup. Based on these parameters, the gap on the left hand side can range from 0.279 mm to 0.432 mm while the right hand side gap can range from 0.127 mm to 0.229 mm. The difference in gap sizes stems from how the wave rotor is held in place. As shown in Figure 27, the right hand side of the wave rotor is held in place with the bearing abutments while the left hand side consists of bearings, washers, and wave washers. The combination of washers and wave washers allowed flex for thermal growth. The extra

spacing on the left hand side of the shroud accounted for the thermal growth that the flex allowed. The design here was to have the right hand side be stationary while the left hand side can move with thermal growth by compressing the wave washers.

In addition to housing the rotor, the shroud must be the connection point for the two endwalls. The mating between the shroud and the endwalls was accomplished using the five-hole bolt pattern (Shown as (1) in Figure 28) on each side of the shroud. The holes were sized to freely fit a size 8-32 screw. As mentioned previously, the port locations are critical in creating a successful wave rotor. To accomplish this alignment, the shroud and endwalls have corresponding dowel-pin holes (Shown as (2) in Figure 28). The shroud must also contain the pressure from the cavity and create a seal between the endwalls. The ring and step (Shown as (3) in Figure 28) were used to create this seal. The touching interface can withstand the cavity pressure of 172 kPa. The step adds to the seal with geometry as well as centering each endwall. The shroud was machined in house out of the same material as the rotor, Inconel 625.

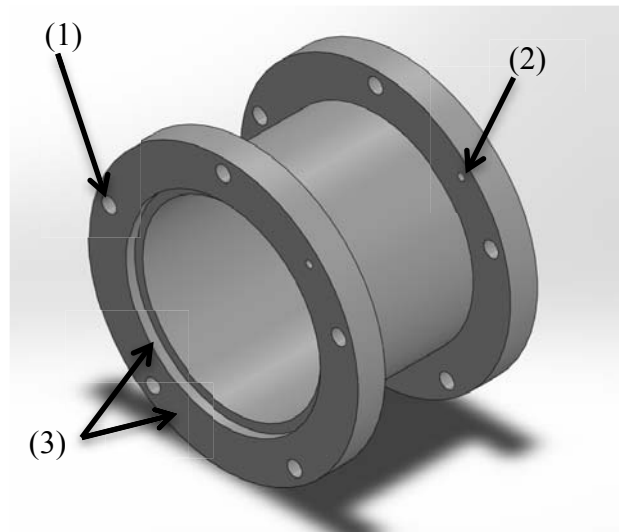


Figure 28 - Isometric View of Shroud CAD

III.2.3 Endwall Designs

The endwalls were the most intricate part of the wave rotor design. Because of the complexity, these two parts were 3-D printed using Direct Metal Laser Sintering (DMLS). The two parts are situated on opposite sides of the rotor and shroud and have five primary responsibilities. These obligations include port timing (based on port locations), routing of the four different flow paths, sizing to match up with the previously built burner loop (discussed in Section II.7.2), containment of the bearing and washer system (discussed in Section III.2.1), and being a connection for the motor mount.

The printing was completed by C&A Tool Engineering, Inc. out of Churubusco, Indiana. C&A was used for their chosen orientation of printing. The printing process was completed with the angled ports faced down. This method allowed for the highest order of accuracy for the port locations and angles. C&A used Inconel 718 for the endwalls. For maximum accuracy, the parts were over grown in certain areas and later machined to the correct tolerances and surface finishes. Areas overgrown included bolt-holes, the center shaft, and the steps for the connections to the shroud. The steps were extruded an extra 5 mm to provide a base before starting the curved passages.

Port timing creates the shock waves that allow the wave rotor to do work as well as extract work from the intake and exhaust gas. The computational code provides port locations as well as entry/exit angles for all four ports on a 0 to Π radian basis. Therefore, each port was created twice and the wave rotor completes two cycles for each revolution. Table 3 demonstrates the locations of each port in radians as well as the entry/exit angles associated.

Table 3 - Radial Port Locations and Angles

Port	Open/Close Cycle 1	Open/Close Cycle 2	Entry/Exit Angle (Degrees)
CI	0.72/1.65	3.86/4.79	36.67
CO	2.29/2.95	5.43/6.09	23.49
TI	1.84/2.65	4.98/5.79	19.48
TO	0.00/1.05	3.14/4.19	18.33

To achieve these design constraints CAD models for both endwalls were generated. Figure 30 shows these CAD diagrams for the left and right endwalls with different parts numbered. The left endwall contains the CI Ports (1) and the TI Ports (2). Also highlighted on the left 5-hole bolt pattern (4). Both the ring and step seal and the bolt pattern are used to mate with the shroud and were discussed in Section III.2.2. The instrumentation taps are used to measure static temperature or static pressure and are discussed further in Section III.6. The right endwall contains the CO Ports (5) and the TO Ports (6). Also shown on the right endwall are an instrumentation tap (7), a dowel pinhole (8), and the connection interfaces (9). The instrumentation taps are used to measure either static temperature or static pressure and are discussed further in Section III.6. The dowel pinhole is used to ensure the two endwalls are aligned correctly with each other to guarantee correct port location and timing. The connection interfaces are used and sized to connect to the previously used burner loop and venturis.

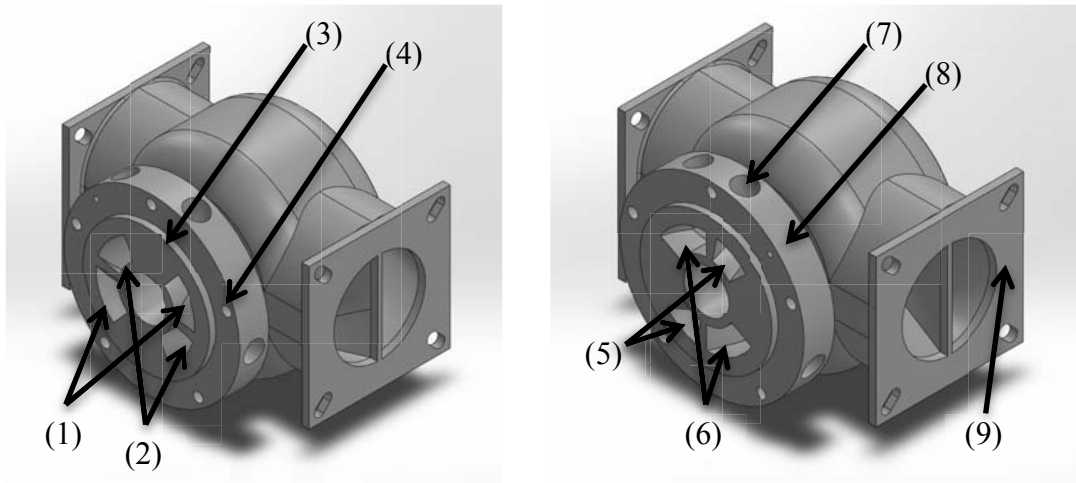


Figure 30 - Left Endwall CAD (Left), Right Endwall CAD (Right)

The entry/exit angles help the wave rotor spin by directing the air at the correct angles to reduce impingement. This design utilizes the first 12.7 mm of the endwalls to contain the correct port angles. The 3D printing process was done in a way to ensure the angles would be correct as well as the port locations. Figure 29 demonstrates how each port angle was included in the CAD models where β is the angle provided in Table 3. The angles labeled 1. (Left side) are used for the CI and TI ports while the angles labeled 2. (Right side) are used for the CO and TO ports.

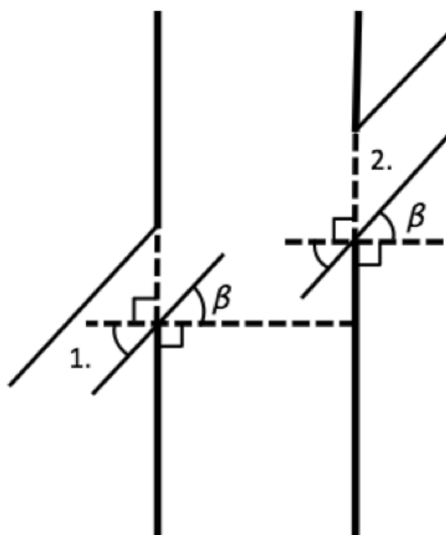


Figure 29 – Endwall Port Angle Diagram

The second responsibility was the routing of the eight different flow paths. This routing added another level of intricacy to the designs. The difficulty comes from the required final locations, the location of the center shaft, the desire to minimize flow path losses, and the requirement to have two locations for each port type. The two locations come from the wave patterns occurring twice per revolution. The two paths for each port are separated until the flow path leaves each respective endwall. Figure 31 demonstrates the different flow paths for both the left endwall and right endwall. The routes were designed to have the least sharp angles while maintaining the general path size. Both of these tasks were difficult since the paths cross each other and the center is taken up by the bearing support system. For the left endwall, the green passages are the CI Port flow paths while the orange passages are the TI Port flow paths. For the right endwall, the red passages are the CO Port flow paths while the blue passages are the TO Port flow paths.

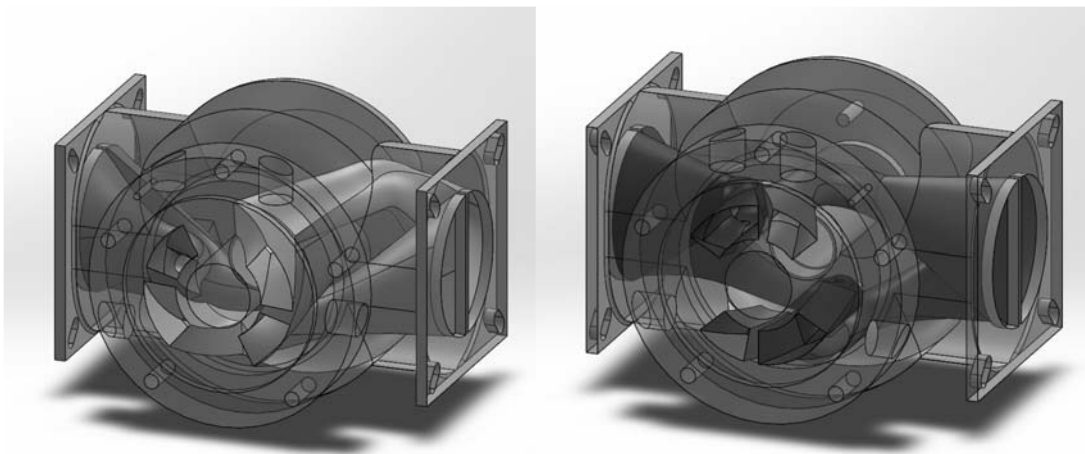


Figure 31 - Left and Right Endwall Flow Paths

Size matching with the previously built burner loop was critical for successful testing. In order to utilize the combustor, the connecting ports were required to be in the same geometric location. This stipulation dictated the overall length of the endwalls and

the shroud. The endwall lengths were manipulated for this purpose instead of the shroud, as endwall length did not dictate wave timing or gap spacing.

The final two responsibilities are containment of the bearing system and being a connection for the motor mount. The center shaft going through each endwall served as

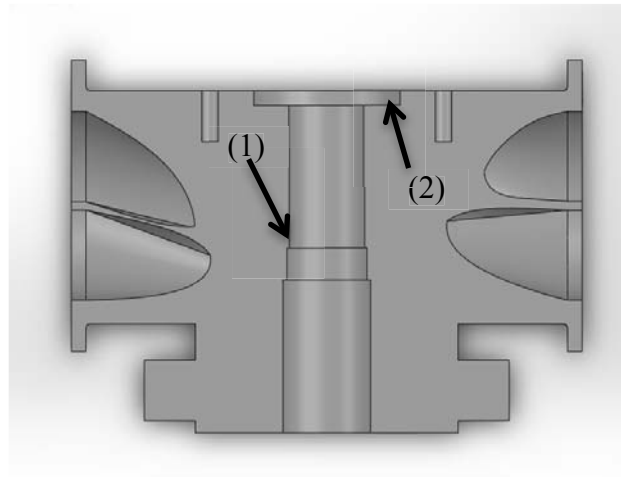


Figure 32 - Cut View of the Right Endwall

the bearing system containment. The endwall contribution to the containment consisted of a smaller diameter shaft area (Shown as (1) in Figure 32) that provided abutment for the bearing. The connection for the motor mount was on the far side of the right endwall. This connection used a ring and step interface (Shown as (2) in Figure 32) similar to the endwalls to shroud connections.

III.2.4 Motor and Motor Mount Design

In order to spin the wave rotor, a sensed brushless electronic motor and an electronic speed controller (ESC) were purchased. The motor and the ESC were high-grade remote control car parts created by Tekin. The motor model chosen was the

Redline Gen3 Motor with a 10.5 turn option and was powered by a 14.8V lithium polymer battery. This battery was already in use by the AFIT wave rotor group. An approximate RPM can be calculated by multiplying the kV of the motor times the voltage

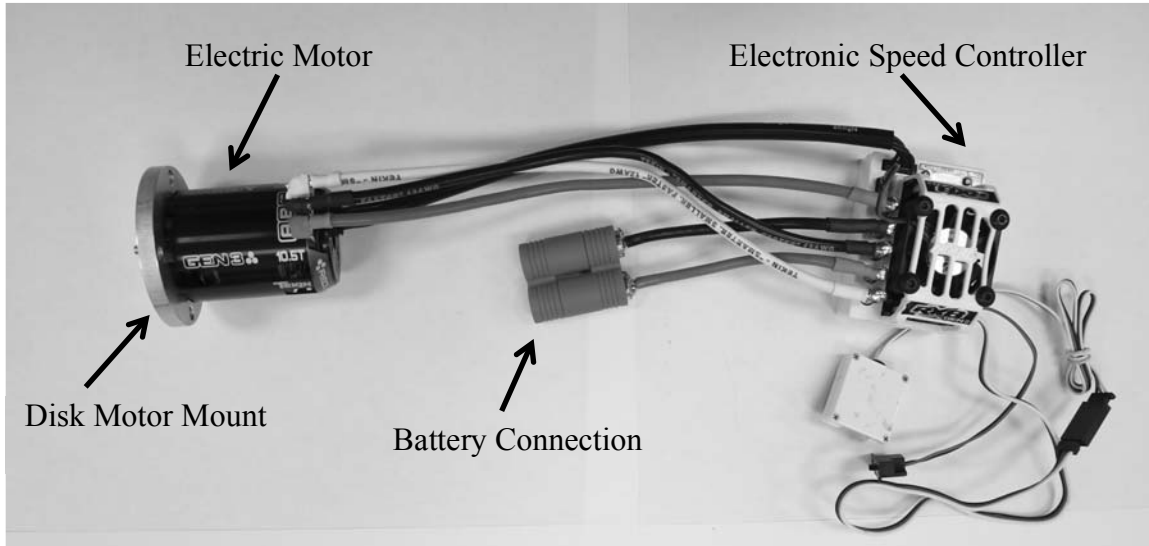


Figure 33 - Electric Motor and Electronic Speed Controller

supplied. The term kV is used in the remote control industry and is the ratio of RPM over voltage for a specific motor. The motor used for this research had a kV value of about 3500-4000 kV. Therefore roughly 15.0 – 17.2 volts are required to reach the design speed of 60,000 RPM. Both the motor and speed controller could be altered or programmed to reach higher spin rates. The electronic speed controller was the RX8 Gen2. This ESC was designed for higher than normal voltages and was programmable using Tekin software. The chosen electric motor and electric speed controlled are shown in Figure 33. The motor was attached to the disk motor mount. The reasoning behind selecting this motor came from discussions with an engineer from Tekin. Parameters used to decide the motor and speed controller included spin rate, rotor weight, and what size of battery would power the motor.

The final CAD piece designed was a two part motor mount as shown in Figure 34. This motor mount consisted of a thin disk that attached directly to the Tekin motor and a longer piece that connected the disk to the right endwall. Both pieces were created out of

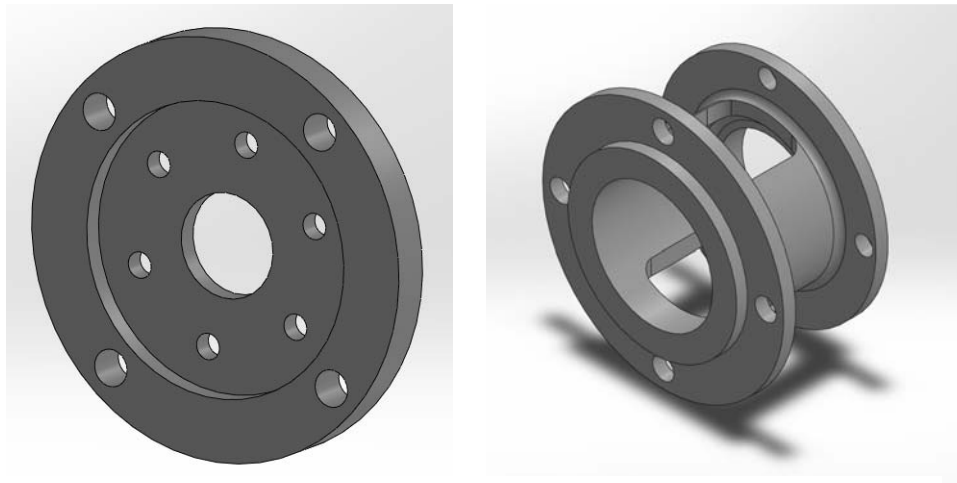


Figure 34 - Disk Motor Mount (Left) and Endwall Mount (Right)

Aluminum 6061 for ease of machining and lack of thermal requirements. The connection from the disk to the motor utilized a five-bolt pattern of size M5. This size was stipulated by the tapped connections on the front face of the motor. The second connection from the disk to the longer piece consisted of a concentric raised ridge that fit inside a depressed annulus similar to the connections between the endwalls and shroud. The four-bolt hole pattern for securing the two mount pieces together was a free-fit for a size 8 screw. The endwall mount also had a concentric raised ridge for matching up with the right endwall as discussed in Section III.2.3. This ring and step interface was secured down using the bolt pattern on the endwall mount and the 6-40 tapped holes on the right endwall. The diameter of the Love Joy Couplings and the length of the wave rotor shaft as well as the

motor shaft dictated the interior size of the endwall mount. Two windows were added into the endwall mount design to help the assembly process.

III.2.5 Assembly

Six machined parts fit together to create the wave rotor designed for this research. Combining the parts is done in a specific order to ensure the correct fits. The different parts in an exploded view are shown and labeled in Figure 35.

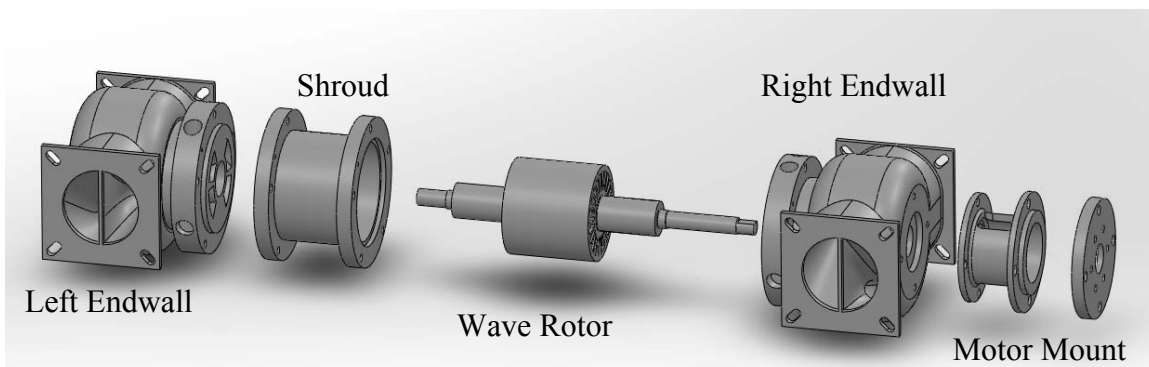


Figure 35 – Expanded Assembly Isometric View

The first step in this process was adding the SKF bearings to the rotor. Since an interference fit was used, the bearings were pressed on. The amount of force added to the bearings was kept low as previous attempts encountered bearing roughness if too much force was used. Here a high speed thermal grease created by Kluber Lubrication was added to both sets of bearings. The next step consisted of combining the right endwall and the shroud. The right endwall was used first as it is the rigid side of the design. The endwall and shroud were lined up and had the dowel pin pushed through before tightening the bolts.

With the right endwall and shroud combined, the wave rotor was added next. The wave rotor was pushed through the shroud and right endwall until the interior right

endwall abutment caught the outer bearing race. The next step was an iterative process with adding the left endwall, wave washers, and regular washers. The stacking of regular washers and wave washers on the rotor shaft is shown in Figure 36. Different amounts of regular washers were tried to obtain the correct amount of flex from the wave washers. As mentioned in Section III.2.1, a total of three regular washers were used to separate the wave washers and preload the bearings. The combination of washers created a flex amount of 0.254 mm when the left endwall was tightened down.



Figure 36 - Assembly Process with Washers and Bearing

To connect the motor to the rotor shaft, first the motor was connected to the disk motor mount. Next, the Love Joy couplings were added to both the motor shaft and the wave rotor shaft. The endwall mount was then attached to the right endwall with the windows being positioned at the top and bottom or the 12 and 6 o'clock positions. The

window positions helped the next step of mating the Love Joy couplings to the inside of the endwall mount. The couplings sometimes required movement along either shaft in order to get the connection correct. When the spacing is right, high strength Loctite 246 was used on the coupling set screws. At this point, the wave rotor should still be easily spun by hand. If the wave rotor was hard to spin, either the coupling or the washer combination required edits.

The actual wave rotor cannot be seen when assembled. A colored cross section is shown in Figure 37 that illustrates how all of the parts interact with each other. The washer and bearing locations are pointed out and are contained in the spacing between the wave rotor shaft and the endwall cavities.

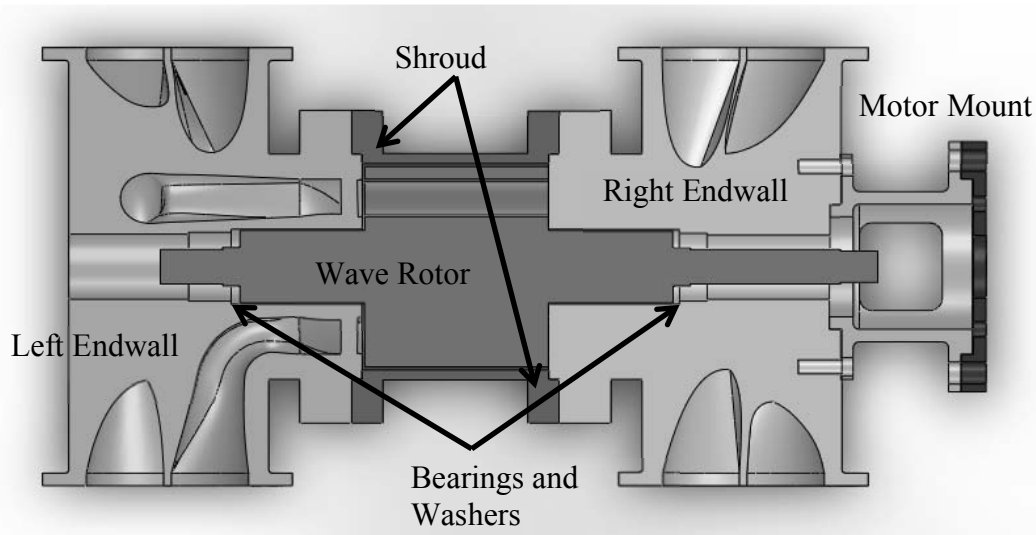


Figure 37 - Cross Section View

III.3 Rotor Spin Test

Initial experimentation tested aspect of the three main components separately before combining them together. The three main initial tests consisted of spinning the wave

rotor, calibration of the venturis, and initial checkout of the combustor loop. The venturi calibration is discussed in Section III.4 and the combustor loop testing is detailed in Section III.5.

Preliminary testing of the wave rotor consisted of doing spin tests to reach the design point of 60,000 RPM. To reach this spin rate, the sensored brushless electric motor was first run by itself from neutral to the max throttle value. Having the motor spin at different throttle values was accomplished using a Berkley box or a signal generator. This method was different from normal operation where the motor and ESC would be controlled using a remote control car controller. The signal generator was attached to the ESC via the white signal wire. The signal generator gave a square wave from 0 to 5 Volts at different pulse lengths. The pulse length determined the throttle value where 1.0 ms was reverse/break, 1.5 ms was neutral, and 2.0 ms was full throttle. For all testing done here pulse lengths between 1.5 ms and 2.0 ms were used. The frequency used by the signal generator was 100 Hz. After a spin test was completed, the ESC was connected to a laptop where the Tekin provided software, Hotwire, downloaded information on spin rate, voltage, and temperatures of the ESC and motor. This software was also used to program the ESC.

For the initial spin test, all motor and ESC options were kept to the factory standard and the wave rotor was not attached to the motor. These precautions were taken to ensure the motor and ESC would not be harmed as well as ensuring the rotational direction of the motor. No issues were encountered during this test and a spin rate of 56,713 RPM was reached. A final spin test of just the motor was performed where the timing on the motor was increased from 20° to 30°. This test did not show any large change in the

maximum RPM value. Changing the timing on the motor changed when the electro magnets are pulsed. This excitation was relative to the incoming poles based on degrees away. Increasing the timing too much has detrimental effects on the motor based on the pulse timing.

The second group of spin tests had the wave rotor attached and had an air assist. The air assist was used to help the wave rotor spin and cool the brushless motor. The air assist was shop air routed to the TI port and to a tube directed at the motor. The TI port attachment helps spin the rotor because of the endwall entry and exit angles discussed in Section III.2.3. The setup for these air assisted spin tests is shown in Figure 38.

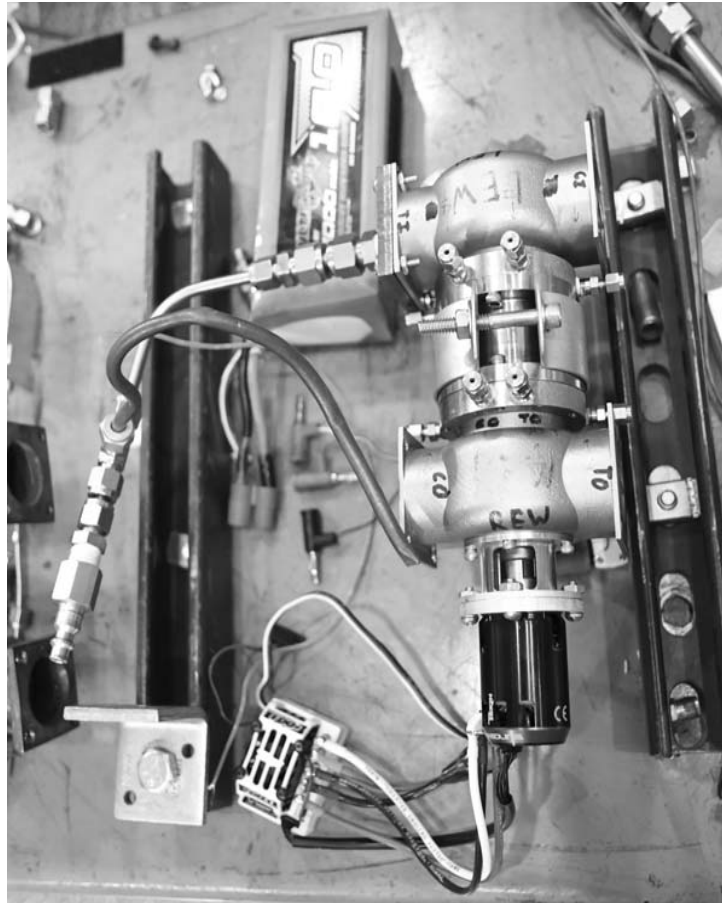


Figure 38 - Air Assisted Rotor Spin Test Setup

The ESC did not require the cooling air as a fan inside the ESC starts automatically when being used. The air routed to the TI port helps the wave rotor spin because of the port angles discussed in Section III.2.3. During these tests, motor timings of 30° and 40° were tried. The motor reached a spin rate of 55,631 RPM or 92.7% of the design point with the air assist and a 30° motor timing. The information about this run is illustrated in Figure 39. Note that the highlighted location is not quite at the highest RPM value. The temperature values for the motor and ESC are in degrees Celsius. Some coupling issues were encountered during spin down of the wave rotor. When the rotational speed was reduced too fast, one or more of the Love Joy couplings may come loose. This happened to some of the air-assisted runs and was combated using the high strength Loctite 246 on the coupling set screws.

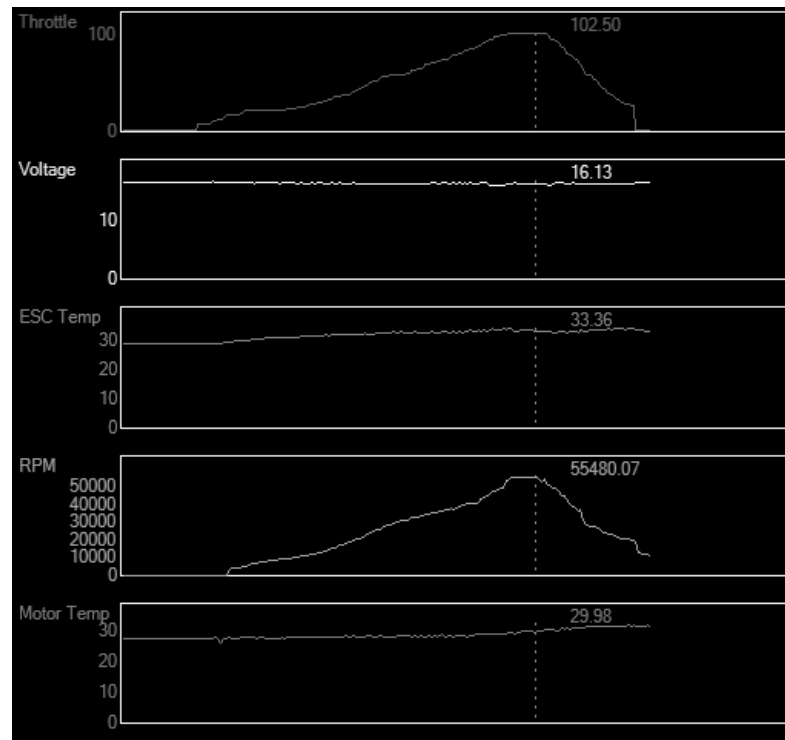


Figure 39 - Air Assisted Spin Test Results

A few final spin tests were done without the air assist. These tests reached over 50,000 RPM, but did not reach the high speeds that the previous tests hit. The motor and ESC did not encounter any issues during these tests besides some coupling movement. With the wave rotor and electronic motor demonstrating a high spin rate capability, the spin tests were ceased. Though the wave rotor did not hit the design spin rate of 60,000 RPM, the spin tests were stopped. The tests were discontinued to not risk damaging any of the hardware before coupled tests could occur. Conversations with the Tekin president provided insight on practices to try later to reach the final design point. These practices involved adding timing to the motor and ESC. Initial spin tests tried motor timing from 20° to 30°, but a significant change was not found. The timing in the ESC was not altered during the spin tests.

III.4 Venturi Calibration

As discussed briefly in Section II.7.2, two venturis were required to monitor the mass flow going in and out of the system. These two venturis, shown in Figure 12, were borrowed from the previous work done by McClearn [3]. Before being used, the venturis were calibrated using known mass flows. This process calculated the venturi discharge coefficient as well as the uncertainty. The discharge coefficient allows the mass flow to be calculated from the pressure and temperature measurements on the venturis [36]. The following equations were used to calculate the mass flow as well as the uncertainty values. The equations are consistent with the American Society of Mechanical Engineers

(ASME) definition. Equation 18 illustrated how to calculate the mass flow with the following Equations (19 – 21) explaining different terms in the mass flow equation.

$$\dot{M} = \rho_2 V_2 A_2 = \frac{CYA_2}{\sqrt{1-(A_2/A_1)^2}} \sqrt{2\rho_1(P_1 - P_2)} \quad (18)$$

$$Y = \left(r^{2/\gamma} \frac{\gamma}{\gamma-1} \frac{1-r^{(\gamma-1)/\gamma}}{1-r} \frac{1-\beta^4}{1-\beta^4 r^{2/\gamma}} \right)^{1/2} \quad (19)$$

$$\beta = d/D \quad (20)$$

$$r = P_2/P_1 \quad (21)$$

In the above equations, the subscript 1 refers to inlet values and the subscript 2 refers to throat values. In Equation 19, the lowercase d is the throat diameter while uppercase D is the inlet diameter. Other nomenclature included is C for the discharge coefficient, P for pressure, T for temperature, V for volume, and ρ for density. To calculate the venturi Y uncertainty, Equation 22 was used.

$$Y_{uncertainty} = \pm \left((4 + 100 * \beta^8) \frac{P_2 - P_1}{P_1} \right) \% \quad (22)$$

To complete the calibration, a known range of mass flows was run through the venturis. This calibration was done in English units, so both English and metric will be used below. The air was compressed by a Sullair TS32 compressor and temporarily stored in a pressure vessel before being routed to the venturis. This mass flow was held accurate to within 0.01 lbm/min or 0.045 kg/min. With both venturis having a design point mass flow of 2.26 lbm/min or 1.03 kg/min, the venturis were calibrated for the same range. This array of mass flows spanned 1 lbm/min (0.45 kg/min) to 3.5 lbm/min (1.59 kg/min) in increments of 0.25 lbm/min (0.11 kg/min). The known mass flow was

passed through a sonic nozzle to choke the flow before entering the venturi. For the lower mass flow rates (1 lbm/min to 1.75 lbm/min or 0.45 kg/min to 0.79 kg/min), both venturis used a 0.110” or 0.279 cm throat diameter sonic nozzle. For the higher mass flow rates (2.0 lbm/min to 3.5 lbm/min or 0.91 kg/min to 1.59 kg/min), the CI venturi used a 0.315” or 0.800 cm sonic nozzle while the TO venturi used a 0.201” or 0.511 cm sonic nozzle. The difference here was required based on how the venturis were connected to the air supply. Because the TO venturi had to be connected using a 3/8” Swagelock fitting, the flow was restricted to the internal diameter of 0.280” or 0.711 cm. This meant that the sonic nozzle for the TO had to have a throat diameter smaller than the flow restriction to successfully choke the flow at the sonic nozzle.

During the testing, two pressures were recorded as well as a venturi temperature. The pressures recorded included the static inlet pressure and the differential pressure between the venturi inlet and throat. The temperature was measured using a K-type thermocouple. The static pressure was recorded using an Omegadyne PX-329 pressure transducer while the differential was calculated using a Honeywell Sensotec differential transducer.

Post processing of the recorded data was done using Equations 18 – 22 and the known mass flow from the compressors. This process was done for each mass flow of interest. The throat pressure, P_2 , was found using the static inlet pressure and subtracting the differential pressure. The density of the flow was found using the static inlet pressure, the flow temperature, and the specific gas constant, R . After calculating values for r and β , Equation 19 was used to calculate Y . Next Equation 18 and the known mass flow was

used to solve the discharge coefficient. The final calculation used Equation 22 to add in an uncertainty value to Y .

The CI venturi had a range of discharge coefficients going from 1.07 at the lowest mass flow to 1.29 at the highest mass flow with a discharge coefficient of 1.20 at the design point. The TO venturi had a range of discharge coefficients going from 0.96 at the lowest mass flow to 1.28 at the highest mass flow with a discharge coefficient of 1.25 at the design point. Machined venturis should have discharge coefficients lower than 0.98 and ASME dictates the coefficient should be around $0.995 \pm 1.0\%$ [32, 36]. The high discharge coefficient found during the venturi calibration means an error occurred during the experiment. This error could have come from an instrumentation issue or a leak in the plumbing. The issue with the high discharge coefficients was not realized until after other testing was completed and the coupled set up was built. Because of this timing and the high demand for time inside the test bay, it was decided to use previous calculated discharge coefficients for the initial tests. As stated previously in Section II.7.2 the CI venturi had an overall discharge coefficient of 0.951 ± 0.019 while the value for the TO venturi was 0.967 ± 0.009 [3].

III.5 Can Type Combustor Testing

As discussed in Section II.7.2, McClearn made a can type combustor that was used to supply the heated air into the TI port. Before coupling the wave rotor and combustor, the combustor was tested to find the exit conditions and flame stability with different fuel and air mass flows. This section details the setup used for the ethylene fuel

line, hydrogen torch, and supplementary air supply as well as the process used in testing.

Figure 40 shows the setup used by this testing and is labeled to provide additional insight.

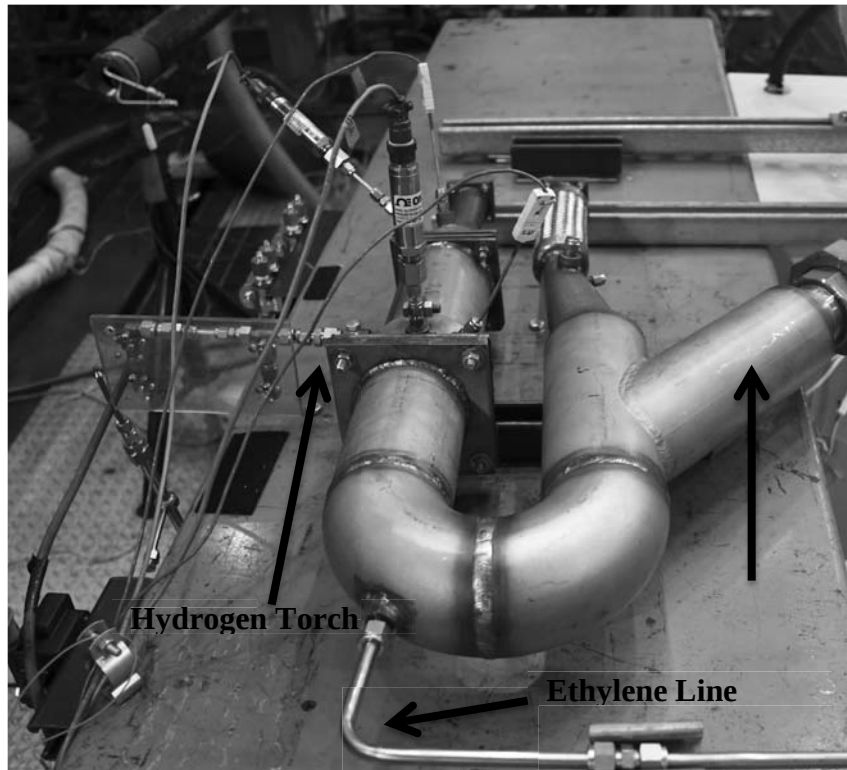


Figure 40 - Can Type Combustor Test Setup

The ethylene fuel line provided gaseous ethylene to the can type combustor at the start of the swirler, as shown in Figure 11. This line entered the combustor loop at a different location compared to when it was used by McClearn [3]. The reason for the movement of the line came from a hot recirculation zone found previously through testing. Instead of coming in from the side of the can type combustor like the hydrogen torch does, the new ethylene line entered near the back of the U-bend in the heat addition loop as seen in Figure 40. At the end of the ethylene line was a Delavan peanut nozzle # 22. This nozzle provided a concentric cone spray with a half angle of 45.7° [3]. Upstream

of the Delavan nozzle was a last chance pneumatic valve, a sonic spool piece, a dome loader for providing different pressures to the spool piece, a pneumatic valve near the ethylene bottle, and the regulator attached to the ethylene bottle. The two pneumatic valves were a safety feature stipulated for all testing at the Detonation Engine Research Facility. The sonic spool piece was used to monitor the amount of ethylene being provided to the can type combustor. Using Equation 23, the ethylene mass flow was calculated based on the upstream gas pressure, P_0 , and the upstream gas temperature, T . Other terms used in the equation are throat diameter, d , specific gas constant, R , sonic nozzle discharge coefficient, C , and the ratio of specific heats, γ . This method and equation were also used for the hydrogen torch.

$$\dot{m} = C \frac{\pi d^2}{4} P_0 \sqrt{\frac{\gamma}{RT}} \left(1 + \frac{\gamma-1}{2}\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}} \quad (23)$$

The sonic nozzle used for the ethylene line had a diameter of 0.69 mm or 0.027". Based on previous combustor testing completed by McClearn, an upstream pressure range of 145 psig to 155 psig was tested corresponding to an ethylene mass flow of 0.049 kg/min to 0.052 kg/min. The dome loader used to alter the upstream pressure was a TESCOM Venting Regulator Model No. 26-2065D24A370 controlled by a Control Air Inc Type 500 I/P pressure regulator. The pressure regulator required a 100 psig nitrogen line and was electrically controlled with an input voltage from 0-10 volts.

The hydrogen torch, shown in Figure 41, was used to ignite the can type combustor. As mentioned in Section II.7.2, the hydrogen torch was used instead of a typical spark plug because of the high temperatures inside the combustor. The hydrogen torch has three inputs (hydrogen, air, and a voltage), but only two were used for this

project. The air line was not used as previous research found a steady flame could be accomplished without the air addition [3]. Previous experiments using the hydrogen torch

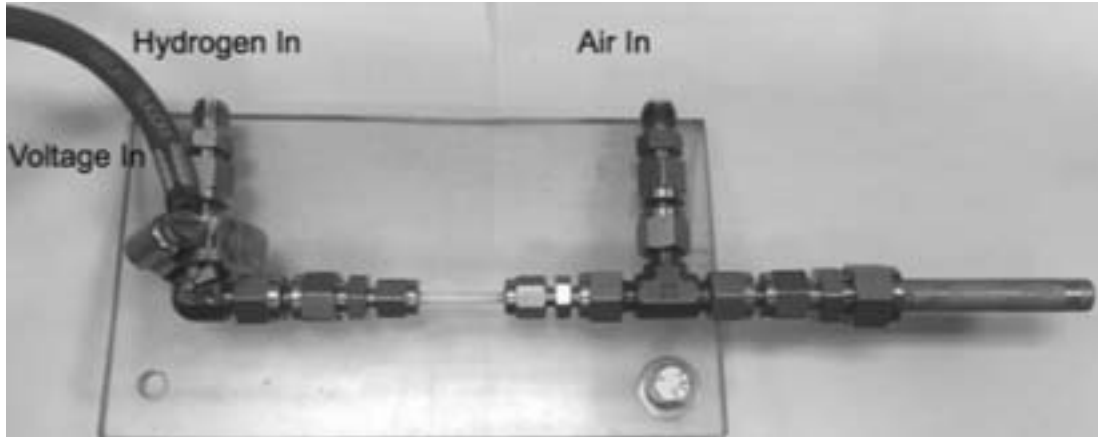


Figure 41 - Hydrogen Torch Used for Ignition [3]

provided the required mass flow for the hydrogen line. Using a sonic spool piece with a nozzle diameter of 0.04 mm or 0.016", 60 psig of hydrogen was supplied from the regulator giving a hydrogen mass flow of 0.0018 kg/min. The voltage was provided using a high output smart coil created by AEM Performance Electronics. It was controlled using a signal generator providing 4-9 ms of 0-5V TTL signal.

The supplementary air used to test the combustor was the same supply used to calibrate the venturis. Like before, a sonic nozzle and the lab control systems were used to provide different mass flows. Previous combustor testing by McClearn heated the supplementary air line using a T63 combustor [3]. Because the T63 combustor was not available, an electric heater was added to heat the air in case flame stability could not be obtained. One last item added to the test rig was a square flange on the exit of the

combustor. This flange had an 11.5 mm diameter hole in it to provide adequate backpressure.

The process for testing the can type combustor consisted of providing different ethylene mass flows to find the outlet temperature at given air mass flow rates. The overall goal for the combustor was to provide it with 1.46 kg/min of air at 757.5 K and have it provide an output temperature of 1108 K. These values come from the code results based on flow properties entering and exiting the wave rotor channels. Because of losses throughout the system, these values are only used as initial goals and were altered as coupled testing progressed.

III.6 Data Acquisition

A range of pressure transducers and thermocouples were used in the venturi calibrations, the can type combustor testing, and the full coupled testing. These sensors allowed for real time monitoring of the systems as well as data for post processing. Furthermore, using the two types of sensors along with the CI and TO venturis allowed mass flow into the system and mass flow out of the system to be known. This section will detail the different sensors used and their corresponding locations.

The same DAQ chassis and DAQ cards were used throughout all of the different tests. The DAQ chassis was a cDAQ-9184 created by National Instruments. This chassis contained four different DAQ cards. The first DAQ card was a National Instruments 9214 thermocouple card. This card has sixteen channels for thermocouples of different types. For this project, twelve K-type thermocouples were used. The K-type thermocouples

have a reported error of 0.75% above 273 K and are rated to a continuous temperature of 1373 K. All of these thermocouples were used as probes inside a flow besides Number 11 which was stripped to the wires and latched down to the outside of the TO venturi. This was done to know the material temperature of the venturi so thermal expansion calculations could be done.

The second DAQ card was a National Instruments 9201 pressure transducer card. This card had eight channels that were all used for Omega pressure transducers with two of the eight being differential gauges. These transducers were all located in either the venturis or in the endwalls. The third DAQ card was a National Instruments 9215 pressure transducer card containing four channels. Like the second card, all pressure transducers used were Omega pressure transducers. These sensors were used in the hydrogen fuel line, the ethylene fuel line, and inside the can type combustor. Figure 42 details the locations of the instrumentation used for this project. Not shown in this figure are the thermocouples and pressure transducers used in the hydrogen and ethylene line.

The pressure transducers used for this research consisted of different Omega types with varying ranges and reported error. The Omega PX-429's had a reported error of 0.08% at full scale and was the most common pressure transducer used. One Omega PX-329's were used and had a reported error of 0.25% at full scale. The differential pressure transducers, Omega PX-409, had a reported error of 0.08% at full scale. All of these pressure transducers were powered with 28 V from a BK Precision DC Regulated Power Supply with variable output.

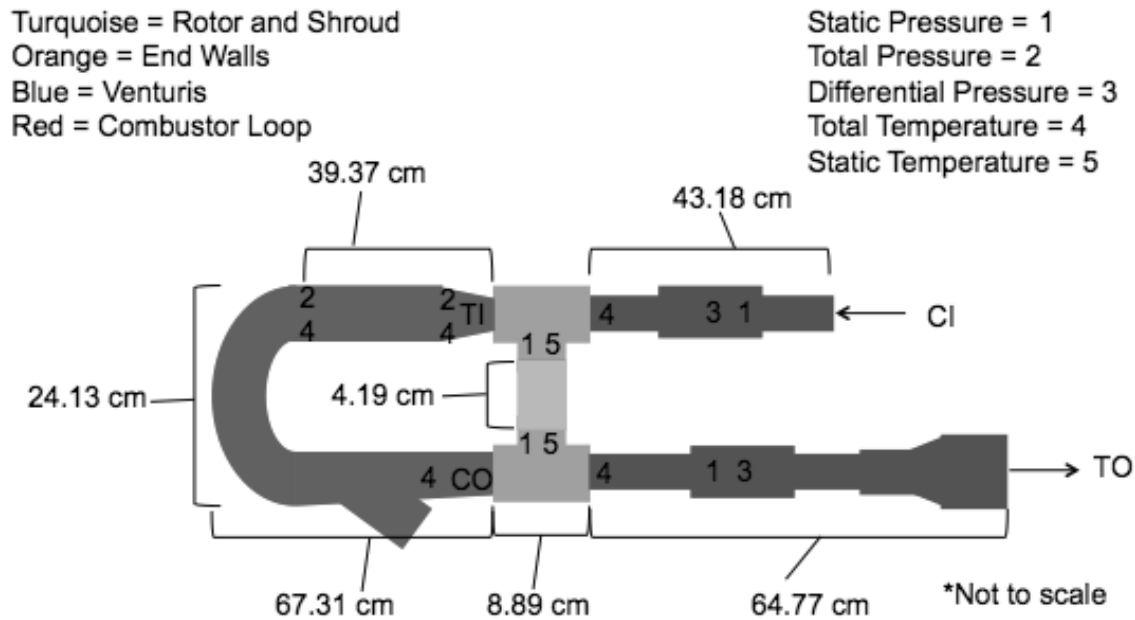


Figure 42 - Wave Rotor Instrumentation

The fourth DAQ card was a NI-9402 and was not used during this research. McClearn used this card to measure the digital signal from the electric motor [3]. Doing this allowed the research to know real time RPM data while testing. This process was not incorporated with this research, but can and should be for future testing.

The program used to monitor the sensors and take data was the same that McClearn used for his work. Some items were edited including the addition of thermocouple channels, changing the second DAQ card to monitor more pressures, and a real time mass flow calculator using the CI venturi.

III.7 Summary

An overall goal of creating a 500 W through flow wave rotor guided this thesis project. Tied into this overall goal were multiple step-by-step objectives. This chapter has detailed the building blocks that have enabled the design of a through flow wave rotor to

the uncoupled tests of the associated hardware. The main parts of this process were the code work, models in CAD, machining the components, and work to test the uncoupled systems. The code work consisted of three iterative parts, which started with goal temperature and pressure ratios and ended with the required geometries, spin rate, and port conditions of a wave rotor. The CAD work started with the outputs from the code and created a wave rotor system that can withstand the forces encountered during operation while matching up to previously made hardware. The wave rotor system created included two endwalls, the wave rotor, a containing shroud, and the motor mount pieces. Uncoupled testing demonstrated the ability for each part of the system to perform at the respective design point.

IV. Analysis and Results

This thesis project had five over-arching goals guiding the work. The first of these goals was to use the NASA code to create a through-flow wave rotor cycle to produce 500 watts of exhaust power [1,2]. Following the completion of the first objective, the second goal was to use the design to manufacture the wave rotor and accessory hardware for testing the wave rotor cycle. With the parts created, the third task was to test the wave rotor and complementary hardware at the design spin rate dictated by code output from the first objective. The fourth goal was to test the previously built can type combustor by McClearn for use at the new design point [3]. The final objective was to couple the wave rotor hardware to the combustor and run the wave rotor at the design point. The data collected at the design point will be compared to the computational results from the first goal. The results from creating the wave rotor cycle inside the CFD Q1D and the following hardware designs are discussed in Section IV.1. The supplementary tests of the wave rotor system parts are detailed in Section IV.2. Results from the test plan and the coupled test are described in Section IV.3.

IV.1 Simulation and Production of the 500 Watt Through Flow Wave Rotor

In order to design the wave rotor, the three-module CFD code was used. When the code was completed, the physical geometries and flow properties at each port were known. The physical parameters from the code allowed the system to be modeled and manufactured. Section III.1 discussed the code process, the criteria of a successful wave

pattern, and some of the results. Section III.2 went over the process of designing the wave rotor hardware. This section will elaborate more on the code outputs as well as the final manufactured hardware.

The overall outputs of the code include mass flows, pressures, and temperatures as ratios of the ambient conditions. The wave rotor physical parameters are known from the code as they are the inputs being altered through the second code module. In order to understand all of the outputs, the results are reformed to illustrate the physical conditions. The temperature, pressure, and mass flow at each port are shown in Table 4. These values enable comparing the coupled test results as well as calculating the power output. The total power output comes from the final exhaust or the TO port. The equations used to calculate power out were shown in Equation 3 through Equation 8. The values and results used for the power out calculations are shown in Table 5.

Table 4 - Port Flow Parameters at Design Point

Port	Total Pressure (kPa)	Total Temperature (K)	Mass Flow (kg/min)
CI	101.3	287.2	1.029
CO	318.7	757.5	1.464
TI	300.5	1108	1.464
TO	114.8	785.5	1.029

Table 5 - Power Out at Design Point

Mach	0.434	Web Thickness	1.27 mm
Sonic Velocity	541.6 m/s	Static Temperature	758.2 K
Exit Velocity	234.8 m/s	Power	~475-500 Watts

After the CFD code had produced the wave cycle, CAD work was completed to model the wave rotor, endwalls, shroud, and motor mount pieces. The shroud and wave rotor were machined out of Inconel 625 while the motor mount pieces were machined out

of Aluminum 6061. The endwalls were printed out of Inconel 718 at C&A Tool Engineering, Inc. before being machined to their final dimensions. The assembled wave rotor is shown in Figure 43 with the left endwall labeled LEW and the right endwall labeled REW. Note the motor is attached, but is not secured down.



Figure 43 - Assembled Wave Rotor System with Motor

IV.2 Supplementary Hardware Tests

Before the complete wave rotor system was tested, three individual systems were evaluated. These three tests included calibrating the venturis, spin testing the wave rotor, and characterizing the can type combustor. The venturi calibration and results were discussed in Section III.4. The spin tests and results were detailed in Section III.3. The setup and process for testing the can type combustor were detailed in Section III.5 and the results from that test are detailed next.

As discussed previously, the purpose of testing the can type combustor was to find the correct amount of ethylene mass flow to reach the goal output and determine flame stability. The test was completed by slowly increasing the pressure of the ethylene

fuel line, which by using the sonic spool piece increased the ethylene mass flow rate. The original plan was to use the dome loader to increase the pressure going to the sonic spool piece. Because of issues with the dome loader, it was bypassed and a pressure was set on the ethylene bottle regulator. Based on previous combustor characterization by McClearn and the goal temperature out of 1108 K, the pressure regulator was set to 150 psig [3]. The pressure supplied to the sonic spool piece had a slow climb from 140 psig to 150 psig. This climb in pressure allowed the test to observe the increase in combustor out temperature as the ethylene mass flow rate was increased. Initially ambient temperature air was provided to the combustor loop at 1.54 kg/min. At this flow rate the combustor experienced flame instability and had to be relit multiple times. Because of the flame instability, the air mass flow rate was reduced to 1.36 kg/min where a steady flame was obtained. The temperature out of the combustor and the associated ethylene line pressure are shown in Figure 44.

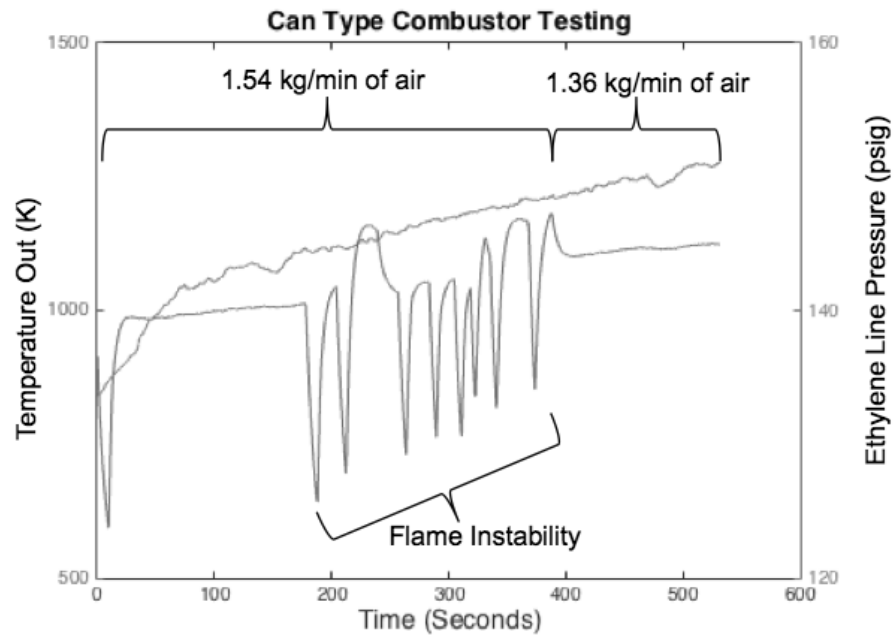


Figure 44 - Combustor Characterization Test

The results from this test demonstrated a steady temperature out of $1108\text{ K} \pm 15\text{ K}$ when the ethylene line was at 150 psig. This ethylene pressure line corresponded to fuel mass flow rate of 0.05 kg/min. Because of this test, a steady ethylene pressure of 150 psig was implemented for subsequent coupled tests. Through the combustor testing, it was found that the combustor could produce a steady flame with the correct output temperature. Though a more thorough equivalence sweep was desired, the results from this test allowed the testing to move forward to the coupled tests.

IV.3 Coupled Tests

The last goal of this research was to test the entire wave rotor system for comparison to the computational results as well as calculating the efficiency and power out. All prior objectives were stepping-stones to the final tests. The coupled setup is shown in Figure 44 with the main items labeled. Starting at roughly 11 o'clock, the power supply was used to power the pressure transducers detailed in Section III.6. The hydrogen torch was used to ignite the ethylene and air mixture. Two venturis measured the flow in CI and TO where the arrows indicate the direction of the flow. The actual wave rotor is shown next. Several wires for the thermocouples and pressure transducers are visible in the figure. The brushless motor is labeled next with the ESC being connected via the red, black, and white cables that are below and to the right of the motor. The large plumbing that joins the combustor loop is the supplementary air used to start the combustor. The final item highlighted is the fuel ethylene line.

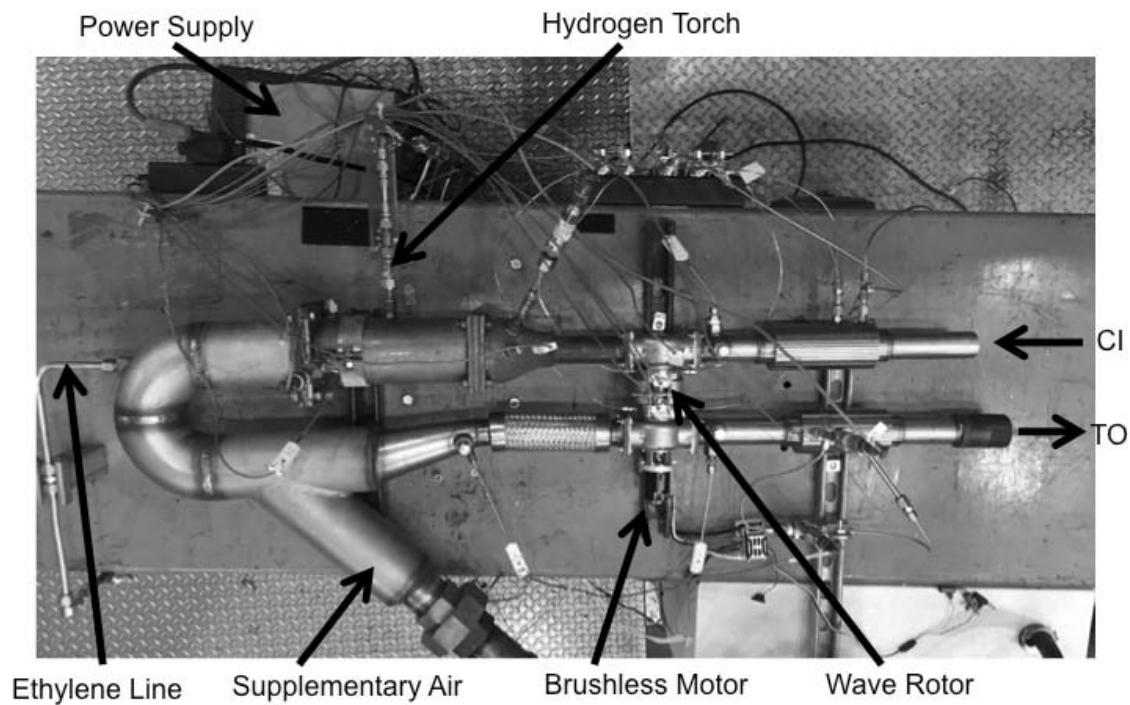


Figure 45 - Coupled Test Rig

IV.3.1 Start Up

A start up test plan was devised in order to get the wave rotor operating at the design point. Start up was an issue with this system because of the co-dependence between the combustor loop and the wave rotor. In order to successfully ignite the ethylene and fuel mixture, a mass flow of air must be provided from the wave rotor. In order for the wave rotor to provide this mass flow, it must be spinning at the design point and receiving the combustion products from the combustor loop. The supplementary air line was used to provide the initial air flow to the combustor loop. This air flow was used to start the can type combustor which then feeds the wave rotor the required exhaust products. The first step in this process was providing air to the combustor loop through

the supplementary air line. The air was provided to obtain a steady flame in the combustor and to ensure the flow proceeded in the proper direction through the combustor loop. Note, at this point there was just air flowing going the combustor loop and the ignition process had not taken place.

Next, the wave rotor was spun at 20% throttle or roughly 20,000 + RPM. The wave rotor was spun before the combustor was ignited to ensure that there would be even heating throughout the rotor. With the wave rotor spinning, ethylene was flowed to the swirler and the hydrogen torch ignited the combustor. With the combustor lit, the temperatures were monitored until a steady temperature out had been reached. The fourth step was to throttle the wave rotor to 100% or 50,000 + RPM. At this point, the wave rotor should be operating at its design point with the correct temperature in from the combustor and the correct spin rate. With the wave rotor operating, the supplementary air to the combustor loop was reduced based on the amount of air coming in the CI port as measured by the CI venturi, instrumentation, and a real time calculation. This calculation was from Equation 18. This part was a balancing act between providing air to the combustor loop and having the wave rotor bring in ambient air. The final step of the test plan was to throttle the wave rotor to different speeds to determine the rotational speed operating map. This overall test plan was used for both coupled tests and is described in further detail in Appendix B.

IV.3.2 Coupled Test One

The first coupled test started with providing 0.91 kg/min of air to the combustor loop via the supplementary air line. The supplementary air to the combustor was reduced from the 1.36 kg/min used in the combustor test based on the useful range of the sonic nozzle. Since the test plan called for the supplementary air to be reduced once the wave rotor was operating, it was decided to have an air flow operating range from 0.91 kg/min to 0.45 kg/min of airflow. This range of air flow required a sonic nozzle with a throat diameter of 4.06 mm. Once the air flow was steady, the wave rotor was spun to 20% throttle. This process occurred before any data was taken and is not shown in the following figures. With the wave rotor spinning, data acquisition was started and the ignition process began.

The main three parts of the first coupled test were the ignition process, RPM ramp up, and full throttle. These three segments are discussed in further detail in the following

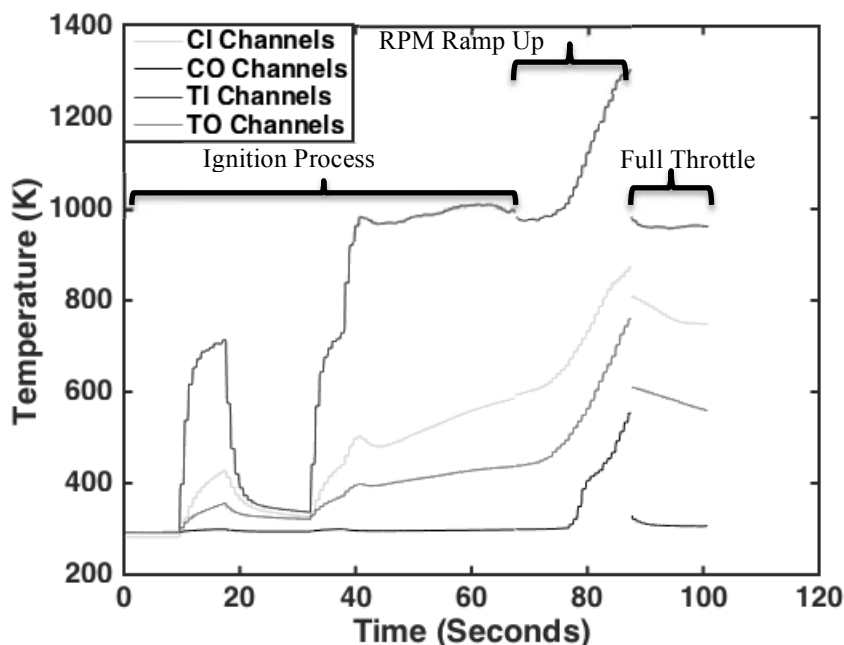


Figure 46 - Endwall Channel Temperatures During Entire Test

paragraphs. A plot of the endwall channel temperatures with the major sections labeled is shown in Figure 46. The disconnects between the different sections was where data acquisition was stopped and then restarted. This process was done to ensure it would be possible to tell the different sections apart during post processing.

The ignition process is the first 67 seconds of the test. First the ethylene line was opened to start the fuel flow and the mixture of fuel and air was ignited using the hydrogen torch. Ignition was accomplished with just the spark and no hydrogen flow was required. The temperatures inside the combustor quickly began to rise for ten seconds before the flame was extinguished at the 20-second point as seen in Figure 47. The flame went out because of backpressure from the spinning wave rotor. At this point the pressure inside the wave rotor was on average 0.42 psia higher than the pressure leaving the combustor. The higher pressure in the rotor came from the supplementary air line. The spark was then triggered again at 32 seconds and a temperature rise occurred. This

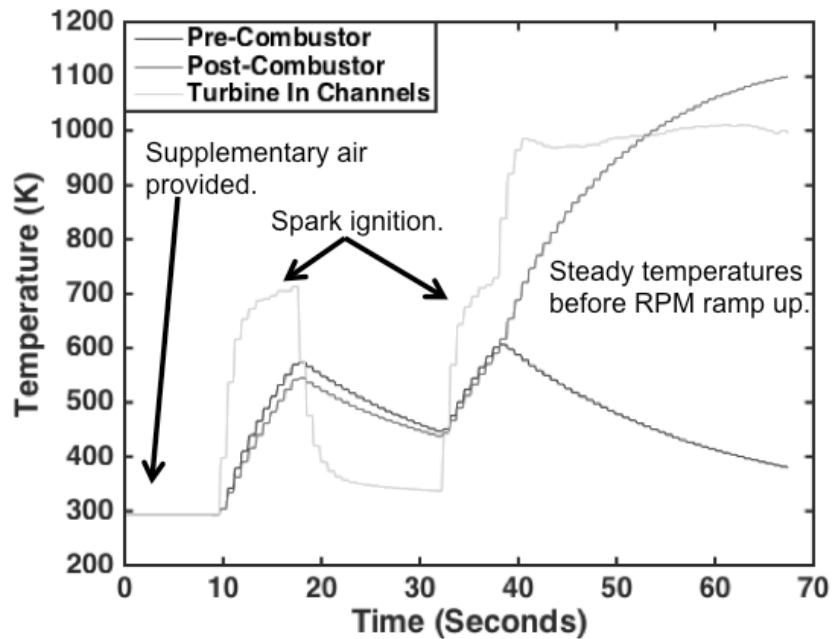


Figure 47 - Temperatures During Ignition

ignition sequence created a steady flame and the post combustor thermocouple showed the correct temperature output value of $1106\text{ K} \pm 15\text{ K}$ as shown in Figure 47. The locations of these sensors are shown in Figure 42.

During the ignition sequence, the most important sensors to watch were the temperatures before the combustor, after the combustor, and inside the TI port. A plot of these temperatures is shown in Figure 47 demonstrating the ignition process for this test. The temperatures in the TI channels jump above the temperatures inside the combustor during the spark ignition process. One possibility for this sharp temperature rise was fuel and air reaching the TI channels before the ignition occurred and burning in the channels. The second possibility is an exothermic reaction from the burnt graphite gaskets discussed later. The pre-combustor temperature increased during the ignition process from the back pressure created by the coupled wave rotor as shown in Figure 48. This

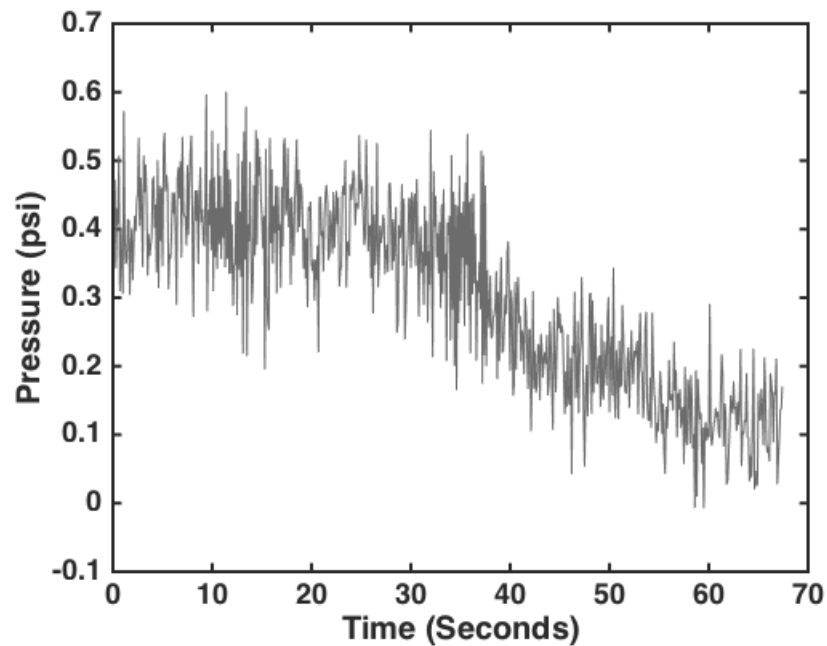


Figure 48 – Back Pressure Reduction During Ignition

back pressure was overcome around 40 seconds into the ignition process. At this point the post-combustor temperature continued to increase while the pre-combustor temperature decreased. The pressure difference from the combustor exit to the wave rotor was reduced from an average of 0.42 psig during the first ten seconds to an average of 0.17 psig during the last 27 seconds of the ignition process. The reduction in back pressure allowed for the flame to stay ignited and the temperature to reduce before the combustor. The reduced back pressure occurred because the flow bypassed the wave rotor and went to ambient conditions via the CI and TO venturi.

The flow path of the exhaust out of the CI venturi was known from the increase in temperature in the CI ports and the CI venturi as well as the black smoke that exited the CI venturi. Instead of bringing in fresh ambient air, the CI venturi began emitting exhaust gas from the combustor. The CI venturi had an increase in temperature from 351 K to 461 K between 40 seconds and 67 seconds into the ignition process when the steady flame was achieved. The CI ports in the left endwall experienced a temperature increase from 495 K to 584 K during this time frame. The increase in temperatures inside the CI venturi, the TO venturi, and the temperatures inside the CI channels is shown in Figure 49. The temperature increase is evident as the two venturis experience similar temperatures until around the 40 second mark. The temperature in the CI channels had similar peaks as the TI channel temperature data shown in Figure 47. Though not noticed during the test, this increase in temperature in the CI channels and CI venturi is a clear sign that the wave rotor is not operating correctly. The CI channels and CI venturi temperatures should read near ambient as the wave rotor is supposed to bring fresh air in through the CI passages.

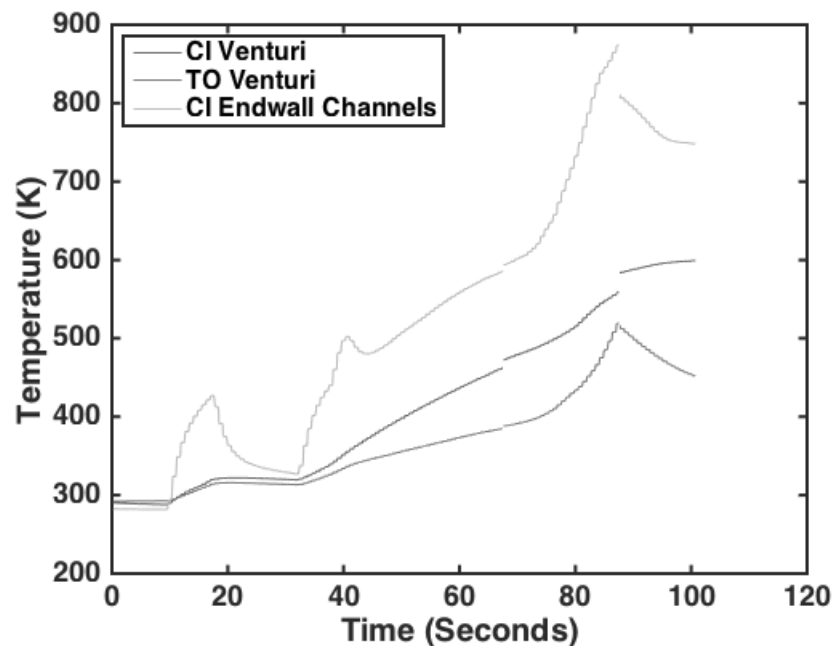


Figure 49 – Venturi and CI Channel Temperatures During Test One

Another reason the exhaust gas did not travel through the wave rotor correctly was the pressure from the combustor supplementary air. This supplementary air made the CO and TI channels experience pressures between 19.5 psia and 24.9 psia. The TO and CI channels had pressures ranging from ambient to 14.6 psia during the same time frame. These pressures and the corresponding temperatures during the test are shown in Figure 50. Because of the pressure differences, the exhaust gasses escaped through both TO and CI channels instead of just the TO channels. The wave rotor needs to be operating correctly for the CI channels to bring in ambient air. The wave pattern detailed in Section II.3 will bring in the ambient air at and around the wave rotor design point. The exhaust exiting the CI venturi made it impossible for the wave rotor to bring in fresh ambient air. The noise seen in the pressure data for the CO and TI channels around 15 and 35 seconds was caused by the spark ignition.

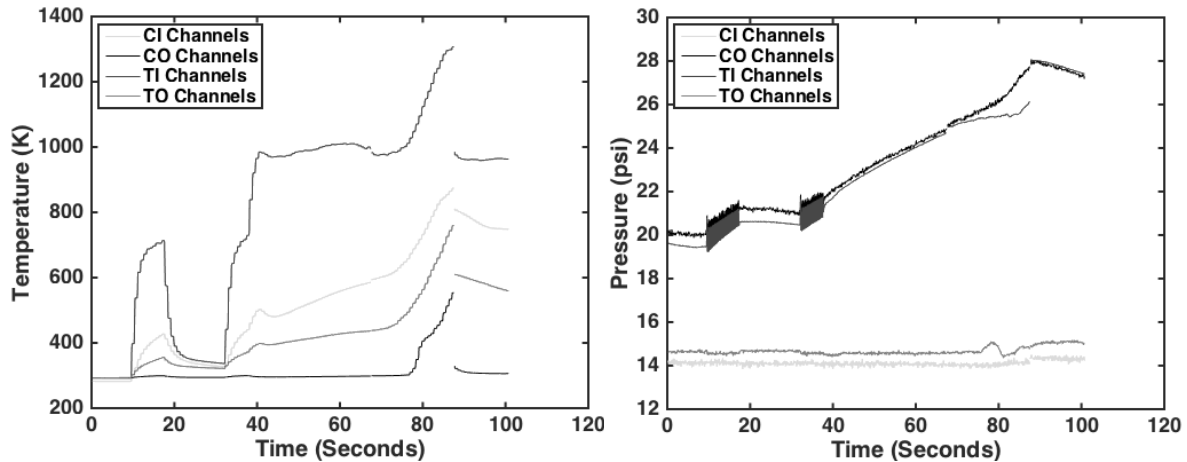


Figure 50 - Temperatures (Left) and Pressures (Right) in the Endwall Channels During Test One

One lesson learned from the data in Figure 47 was the amount of temperature loss between the outlet of the combustor to the flow entering the wave rotor through the TI channels. The complex channel geometries inside the endwalls lead to losses that were unknown until the entire system was tested. In this case, roughly 100 K was lost during the flow routing from the combustor outlet to the TI wave rotor channels. This system loss is the difference in temperatures between the green and red lines in Figure 47 at the end of the ignition sequence. The temperatures found in the TI channels steadied around 40 seconds when the wave back pressure was overcome and the exhaust flow started exiting out the CI venturi.

Since the temperatures reached the goal temperature out, the wave rotor was throttled from 20% to 100% to bring the wave rotor up to full speed over roughly twenty seconds. This speed increase took the wave rotor from the 20,000 RPM range to the 50,000+ RPM range based on a multitude of previous spin tests. The actual spin rates are not known because of speed controller issues that are discussed later.

The pressures during the ramp up phase, 67 to 87 seconds, showed promise with the CO channels seeing an increase as the throttle was increased. Furthermore the final exhaust, TO, demonstrated the pressure ratio compared to the ambient for the kinetic energy power out of the exhaust flow. The pressure ratio between the exhaust flow and the intake flow, P_4/P_1 , was 1.07 where the goal pressure ratio from the code was 1.13. The temperature ratio out, T_4/T_1 , during this time was 0.82 because of the exhaust gas exiting through the CI venturi. The goal temperature ratio between intake and final exhaust gas was 2.72. When compared to actual ambient air, the temperature ratio for the TO flow was 2.60. Using the equations described in Section III.4, the mass flow out of the TO venturi was calculated to be 0.061 kg/min. The design point mass flow out of the TO venturi was 1.028 kg/min. Based on the TO venturi mass flow, the temperature ratio, and the pressure ratio, the wave rotor produced 15.5 Watts. This calculation was based on Equations 3 through 8 and calculated the kinetic energy of the flow. The compression efficiency calculation, Equation 2, calculates a negative value because of the temperature rise from the exhaust gas leaving the CI passages.

The temperature plots during the ramp up phase illustrated the need to decrease the amount of fuel or air going to the burner. As the burner was supplied with heated air from the wave rotor, the temperatures entering the wave rotor, T_I , increased past the design point. The supply of heated air was part of the wave rotor operation. Reducing the amount of supplementary air could have helped this issue, but that was not tried during this coupled test. The temperature plot illustrates another sign that the system was not working properly. As the throttle was increased, all temperatures increased including the

fresh intake air channels, CI. This temperature further shows that the fresh air was not being brought into the wave rotor system.

After the RPM ramp up, the wave rotor was held at max throttle for thirteen seconds. The temperatures and pressures inside the different endwall channels during this phase are shown in Figure 50 from 87 to 100 seconds. After this short amount of time, the decision was made to shut down the system because of multiple reasons. For shut down, the wave rotor was spun down and the combustor was extinguished by stopping the fuel flow. Both of these processes occurred quickly and near the same time. Reducing the throttle to the neutral position spun the wave rotor down. This was done by reducing the pulse length from the signal generator to 1.5 ms. The fuel flow was stopped by shutting the pneumatic valve closest to the combustor loop. The supplementary air was continued to help cool down the system. The first reason to shut down was the change in pitch heard from the wave rotor and electronic motor. The change in pitch was a sign that the motor and rotor spin rate had been changed. The second was the intake air venturi, CI, was releasing dark exhaust instead of bringing in fresh air. The mass flow calculations during this phase and the RPM ramp up phase show that no air was entering the wave rotor system through the CI venturi. Lastly, the temperatures on the DAQ showed that the exhaust flow was not going in the correct direction by exiting out of the CI venturi instead of the TO venturi. The wrong direction was known from the increase in temperature inside the CI channels and the CI venturi.

Once the system was at a safe temperature, the system was investigated. The first issue was that the wave rotor and electric motor had become uncoupled. This uncoupling occurred because the wave rotor and motor wanted to spin at different speeds. The wave

rotor slowed down because of bearing damage and internal contamination. This required more power than normal to spin at max throttle. The additional power required by the motor increased the amperage inside the speed controller. This increase in amps had a detrimental effect on the ESC to where the data saved for this test was only flat lines at zero. Because of this the actual spin rates, motor temperature, ESC temperature, and voltage during the first coupled test were unknown. When initially investigated, the wave rotor could not be freely spun and would stop after a slight turn in either direction. The more the wave rotor was spun by hand, the more it could turn indicating that something had lodged itself inside the system. Taking the system apart allowed further investigation of the wave rotor. The internal components had carbon deposits throughout with a greater amount on the left endwall side of the rotor. The carbon deposits on the left endwall port face is shown in Figure 51.



Figure 51 - Left Endwall Face with Carbon Deposits

Based on the black soot throughout the system, it was believed that the wave rotor stopped spinning because of a pieces of burnt material became lodged inside the shroud, rotor, and endwall gaps. The material that burnt came from the graphite gaskets used to seal the flanges together. When the gaskets were cut, too much material was in the flow path and the material did not have the thermal properties required for the internal conditions. Because of these two issues, the gaskets were burnt and deposited carbon inside the wave rotor which stopped the system from spinning, soot spread throughout the internal components, and created the dark smoke that came out the CI port. One of the burnt gaskets is shown in Figure 52.



Figure 52 - Burnt Gasket

Though this test was short lived, a couple lessons came from it. The first was the need for new gaskets. Though the graphite gaskets worked great for the pre-combustor flanges, anything after the combustor needed to be cut correctly and have the correct thermal properties. Because of this, the rebuild of the wave rotor system used metal-faced high temperature gaskets. The second lesson came from the temperatures recorded in the endwalls during the RPM ramp up process, shown in Figure 50. As the combustor was

provided with heated air from the wave rotor, the combustion process produced air with higher temperatures than the goal TI temperature. Because of this, the fuel flow rate of ethylene needed to be reduced to reduce the temperature out. The last lesson came from the flow leaving the CI port. Since the CI port is open to ambient and is the shortest path, the exhaust from the combustor was leaving through the CI port. To fix this issue, later tests had a direct couple on the CI venturi to provide air flow into the wave rotor.

This test did not have successful results, but the temperatures and pressures recorded inside the wave rotor help understand what happened. Furthermore, the pressure and temperature ratios found can be compared to the ratios calculated by the CFD code shown in Figure 16. The temperature and pressure ratios recorded during the RPM ramp up and the post ramp up phases are shown in Table 6. The RPM ramp up phase occurred from 67 to 87 seconds into the test while the full throttle phase started after and ended 100 seconds into the test.

Table 6 - Temperature and Pressure Ratios During Coupled Test One

Phase	T_4/T_1	T_3/T_1	T_2/T_1	P_4/P_1	P_3/P_1	P_2/P_1
RPM Ramp Up	0.815	1.701	0.483	1.074	1.858	1.973
Full Throttle	0.705	1.448	0.147	1.060	1.960	1.959
Computational	2.719	3.836	2.622	1.133	2.966	3.142

IV.3.3 Coupled Test Two

The second coupled test included multiple lessons learned. The first adjustment made for the second test included changing out the gaskets used. The metal-faced high temperature gaskets were used to seal the flanges between the wave rotor, the combustor loop, and the venturis. The second modification was reducing the pressure on the ethylene line to 130 psig. The first coupled test revealed TI temperatures above the goal

1108 K goal once the rotor reached near the design speed. Reducing the ethylene mass flow to 0.044 kg/min was expected to lower the temperature out. The last change for the second coupled test was the decision to slowly ramp up to full throttle. This test plan change included taking data at each throttle increase of 10% and starting at a 20% throttle. Slowly increasing the throttle was desired to understand the ramp up process and alleviating overstressing the system.

Shortly into the second coupled test issues were encountered and the test was stopped. This test ended with a broken motor and speed controller. The motor encountered more resistance than normal when spinning the rotor. This analysis comes from the change in pitch from the motor system when the ignition sequence occurred. The RPM records from this test showed the motor was spinning at half of the normal RPM values. This additional force slowing the motor overloaded the motor and speed controller and burnt out the internal wiring. Another motor and speed controller were ordered to continue testing that is not detailed in this document.

IV.3.4 Coupled Test Three

A third coupled test was attempted with changes from the previous two tests. The third test did not use the electronic brushless motor and it had an air line directly coupled to the CI venturi inlet. The air line to the CI venturi was split off from the supplementary air line. The air mass flow coming into the CI venturi along with the air mass flow from the supplementary air line allowed the wave rotor to free spin. The free spin was accomplished because of the angled endwalls discussed in Section III.2.3. Initially 1.58

kg/min of air was supplied from the compressors and split between the CI venturi and the supplementary air line. Once the flow was steady, the wave rotor was spun by hand to get over the initial resistance. At this point, the incoming air mass flows significantly increased and sustained the wave rotor rotational speed. The air mass flow was reduced to 1.36 kg/min and the ignition process began. The wave rotor continued to free spin during the temperature increase before the rotation stopped. The wave rotor stopped spinning when the TI channels had temperature readings around 900 K. At this point, the decision was made to stop the test. During the test, the combustor exhaust did travel through the wave rotor and exited out the TO venturi. A temperature ratio, T_4/T_1 , of 1.14 was achieved before the wave rotor stopped spinning. The post test examination did not show any signs that the wave rotor made contact with the shroud or the endwalls.

IV.4 Summary

Chapter IV has described the results from this research. Section IV.1 detailed the simulation results of the wave rotor cycle and showed the final created hardware. Information about the port temperatures, pressures, and mass flows were shown as well as the values that led to the overall power out. The assembled wave rotor was shown with the motor attached. Section IV.2 explained in detail about the supplementary hardware tests with a focus on the can type combustor testing. The chosen fuel line pressure and the corresponding temperature out were detailed along with the overall process. Section IV.3 discussed the start up procedure and the two attempted coupled tests. Although the

tests were unsuccessful, they provided insight on the work that is required to successfully test this through flow wave rotor.

V. Conclusions and Recommendations

Drones are becoming more prevalent across all industries as the technology is expanded and more uses are found. Improvements in small surveillance systems and computing power make the drone the ideal method of battlefield surveillance. Currently the methods used to power these small UAVs are electric motors, gas turbines, and piston engines. The electric motors encounter issues because of their low power density. The gas turbines and piston engines perform poorly at the required low mass flows.

With the desire for the drones to fly higher, longer duration, and farther other power plant options should be considered. One method that has the high power density of burning hydrocarbons without some of the efficiency issues is a wave rotor system. This research has designed, manufactured, and tested a 500 W through flow wave rotor. The wave rotor system detailed here has the opportunity to prove if a small scale wave rotor is applicable for the growing drone industry.

V.1 Research Objectives

This research had an overall goal designing, producing, and testing a 500 W through flow wave rotor. The objectives below helped guide the work to completing the overall goal.

1. Use the NASA code developed by Dr. Dan Paxson to create a wave rotor cycle that will produce 500 Watts of power [1,2].

2. Design and create the wave rotor and all necessary hardware for testing of the wave cycle.
3. Test the wave rotor and complementary hardware for capability to spin at the design point.
4. Test the previously built combustor by McClearn for use at the new design point [3].
5. Couple the wave rotor hardware and combustor to test the design point for comparison to computational results as well as computation of efficiencies.

The process of creating a wave rotor cycle was started by first learning how to use the code and how the code worked. This step was accomplished by recreating the wave rotor pattern used by McClearn in his thesis work [3]. With a general understanding of the CFD code, preliminary calculations were performed to determine a starting temperature ratio and pressure ratio. The three modules of the code were iteratively worked through until a wave pattern satisfying all of the requirements was created.

The outputs of the code and geometries of previously created hardware were used to complete the second task. Parts designed during this step included the endwalls, shroud, wave rotor, and motor mount pieces. Furthermore, work was done to ensure the parts would be able to withstand the operating forces and the system would couple to the combustor loop and the venturis. Lastly, all of the connecting pieces, items for the bearing system, and the brushless electronic motor were decided during this step.

Once all of the hardware had been printed, machined, or ordered, the wave rotor system was assembled. Three types of spin tests were completed to ensure the rotor

would not encounter any detrimental modes. None of the tests encountered any issues and all tests found the wave rotor system to easily spin faster than 50,000 RPM. The motor and speed controller were not pushed to their operating limits to ensure the items would not be damaged.

With the spin tests complete, the can type combustor was assembled for testing. To test the combustor, methods were used to supply it with specific air and fuel flow rates. The combustor was tested to find the exit temperature given the air and fuel flow rate while ensuring a steady flame. In the end, an ethylene line pressure of 150 psig was found that provided a steady flame with the correct temperature out of $1108\text{ K} \pm 15\text{ K}$.

The final step was coupling the venturis, wave rotor, and combustor loop for a complete system test. This coupled test was done a couple times with many lessons learned. Among these lessons is the need to provide initial air into the CI port, reducing the ethylene mass flow, not needing the hydrogen to ignite the combustor, and signs to watch out for that would indicate if the system should be shut down. This information will help testing as these experiments continue on. Areas of trouble like the gaskets and the additional pressure from the supplementary air must be figured out to achieve successful results.

Although a successful wave pattern has not been created through the coupled testing, the wave rotor system and combustor loop have demonstrated their ability to operate at the design conditions. A couple additions to the test rig could be all that is required to obtain the desired results. These additions are detailed in Section V.2. When the desired results are achieved, this system will be a major milestone in through flow wave rotor technology. Efficiency comparisons to normal turbomachinery and the overall

power output could show the promise of this technology as a future power plant for the drone industry or other applications.

V.2 Recommendations for Future Research

In order to properly obtain the wave cycle, additions should be made to the test system. The first of these is a direct connection of fresh air to the CI port. This direct coupling will ensure that during start up the wave rotor system will be bringing in fresh air through this port. The system used here just had a venturi connected and counted on the wave rotor cycle to bring in the fresh air. As shown in the first coupled test, the CI port can become the shortest distance for the combustor exhaust air to travel. The addition to the CI port will be used initially to help the wave rotor start the cycle before bringing the pressure down to ambient. This CI coupling could be accomplished using a pressure sensor and a sonic nozzle similar to the current ethylene and hydrogen fuel lines.

Other items that could help the research include a way to cool the system, real time RPM data, and programming of the motor and ESC. Cooling the system will increase the life of the bearings and ensure that nothing is burnt or hurt during testing. Flowing nitrogen over the wave rotor system or designing new endwalls that incorporate water passages for cooling can accomplish this. The real time RPM data will be beneficial during testing where the signal generator pulse length can be altered. These changes to the pulse length have varied RPM results. Real time RPM data can be achieved by adding in an Eagle Tree data logger or a Hall effect sensor. The Hall effect sensor will provide signal every time the two magnetic poles in the motor are triggered.

Programming the motor and ESC may allow the wave rotor system to steadily spin at the design point or other desirable spin rates. Overall this system is quite close to being a complete Brayton cycle through flow wave rotor.

V.3 Summary

The work completed and the results found during this thesis have shown the system's ability to become a working small scale through flow wave rotor. Initial work used the CFD code developed at NASA to design a through flow wave rotor cycle that produces roughly 500 Watts of power. CAD and other analysis software were used with the code outputs to create a functional wave rotor system. Testing of the wave rotor system and a can type combustor demonstrated their ability to steadily operate at the design point. Coupled testing of the venturis, wave rotor system, and combustor loop illustrated the areas of the system that require further work to obtain successful results.

Further testing should be done to determine the best operating practices for the wave rotor system. With the operating procedures determined, this wave rotor system will help further the understanding of the wave rotor physics and whether this technology can serve a purpose in a plethora of industries requiring a small scale power system.

Appendices

Appendix A: Additional Computational Analysis

The following figures were created by McClearn and used the CFD results to further examine the design point. These figures were mentioned in Section III.1.7. Figures 53 through 55 illustrate the Mach number, temperature, and pressure on either side of the wave rotor as it rotates one full revolution. Figures 56 through 61 show the pressures, temperatures, density, and velocity at each port.

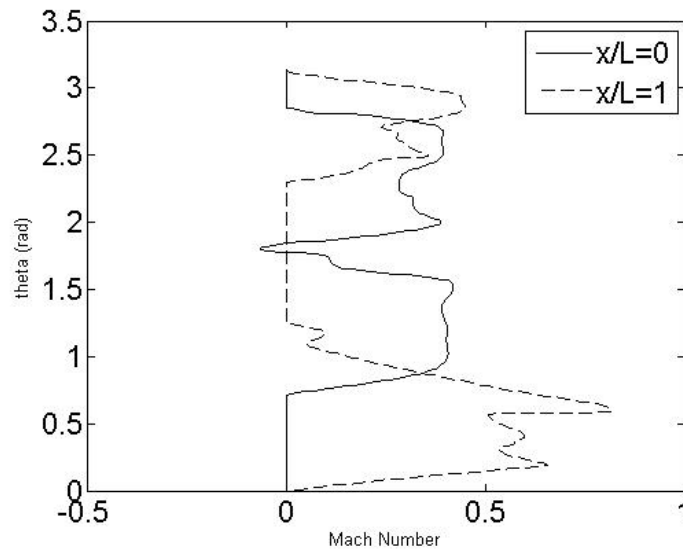


Figure 53 - Mach Number In and Out of Wave Rotor

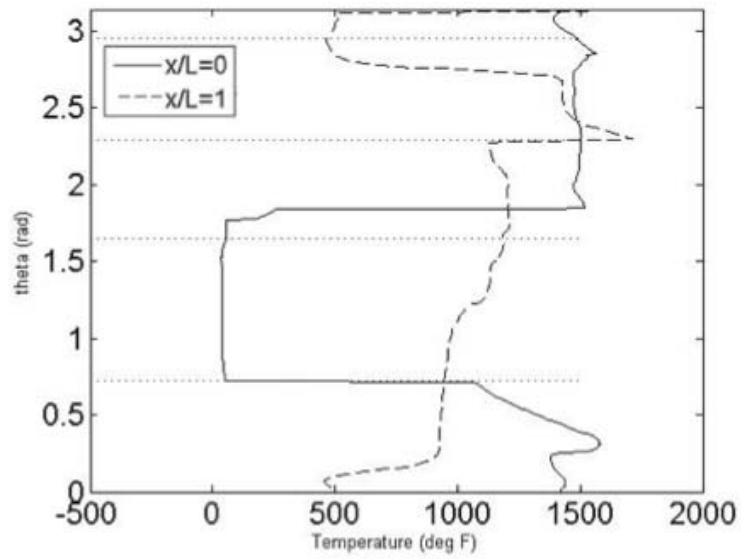


Figure 54 - Temperatures In and Out of Wave Rotor

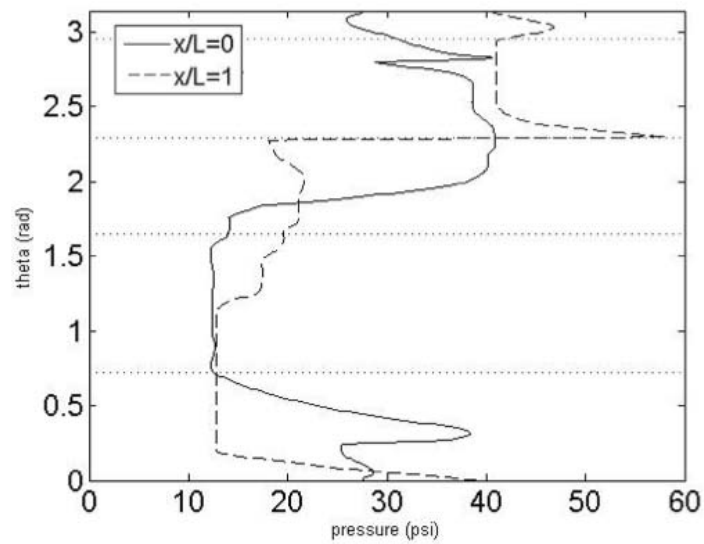


Figure 55 - Pressure In and Out of Wave Rotor

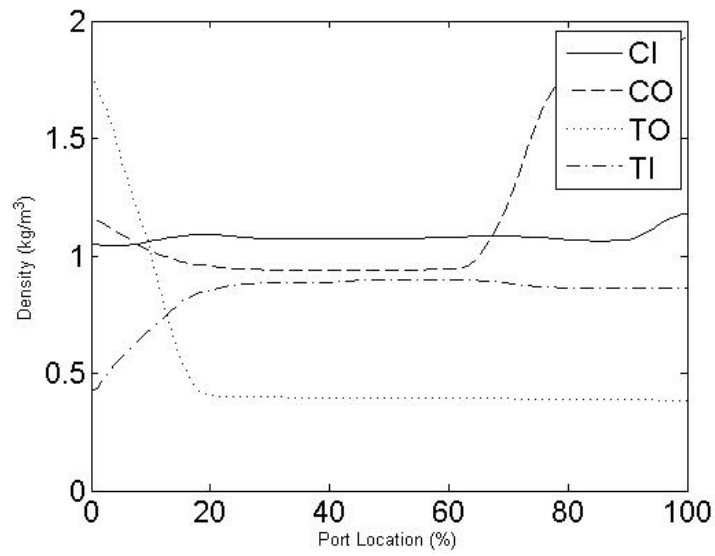


Figure 56 - Port Density

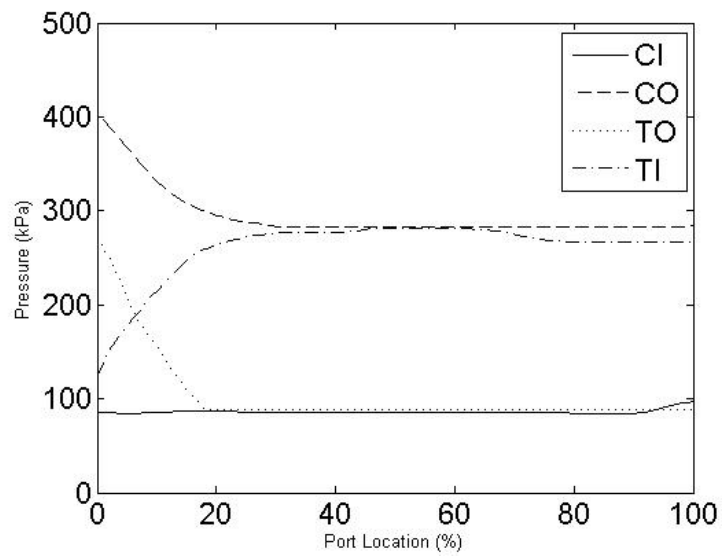


Figure 57 - Port Static Pressure

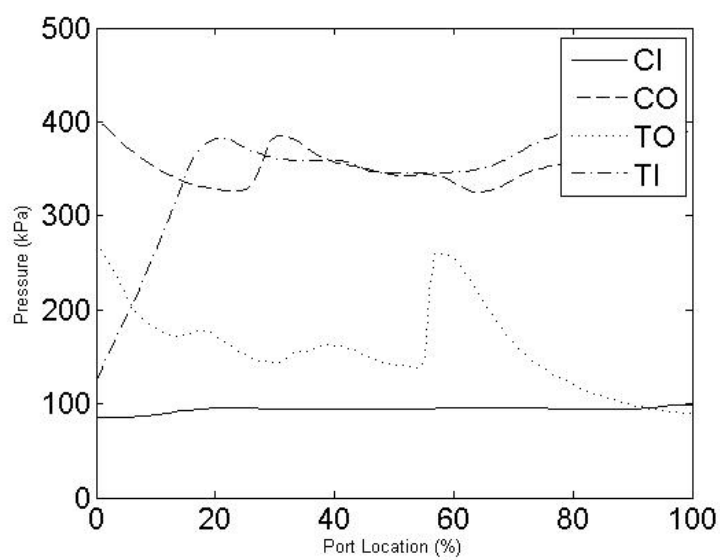


Figure 58 - Port Total Pressure

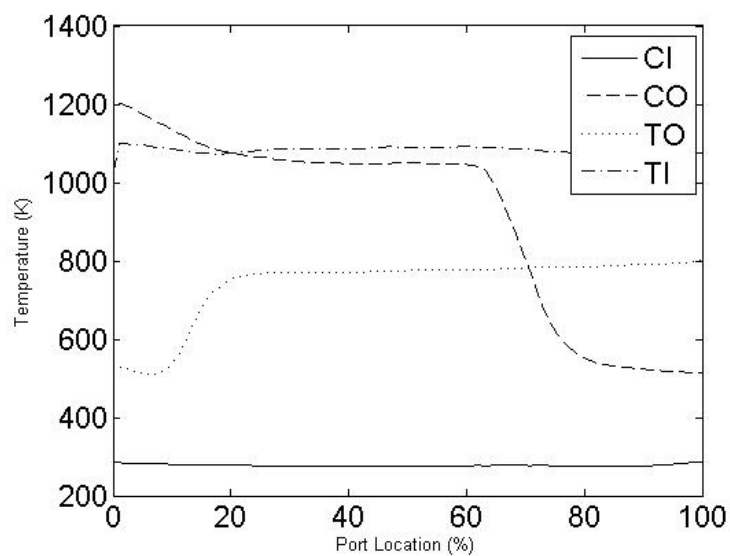


Figure 59 - Port Static Temperature

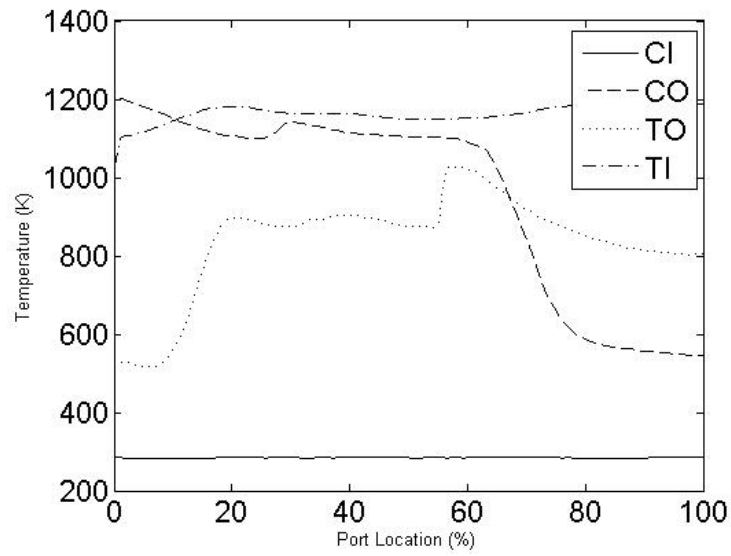


Figure 60 - Port Total Temperature

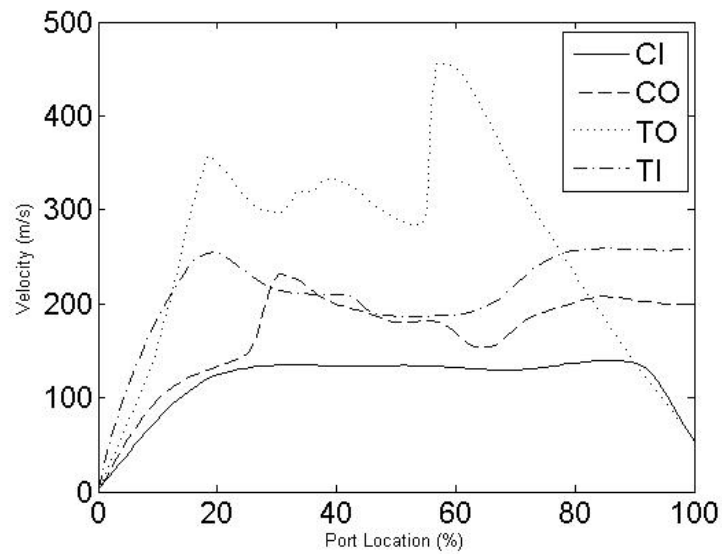


Figure 61 - Port Velocity

Appendix B: Operating Procedure for Coupled Tests

INTRODUCTION: This Standard Operating Procedure (SOP) applies to all personnel, government, and contractor that will be operating the micro scale wave rotor with the small can combustor in Bldg 71A, D-Bay. This document establishes step-by-step procedures for conducting the operation to protect workers from physical harm. It addresses the emergencies action to follow. All personnel involved in the operation must understand the hazards, familiar and understanding prior to starting this operation.

Note: The OP needs to be submitted for review when making significant changes to currently approved experiments.

DOCUMENT REFERENCES:

- AFRL/RZ OI 91-202, Environmental, Safety and Occupational Health (ESOH) Programs
- AFRLI 91-101, Laboratory System Safety Program
- RQ Chemical Hygiene Plan
- PDRF Safety Hazard Analysis

Personnel Protective Equipment (PPE) Requirements:

- Safety Glasses/Goggles required for handling fluids, high pressure systems, and machining operations
 - Ties/id badges/loose items shall be appropriately secured when working in the lab.
 - Hearing protection required for exposure to elevated noise levels
 - Lab coats, coveralls, or suitable fire resistant clothing required for handling fluids.
- When specific tasks are performed, additional appropriate PPE will be mandated.

PRETEST

GENERAL FACILITY PRETEST OPERATIONS

1. Turn on test cell lights (2 switches).
2. Turn on doorbell.
3. Turn on closed-circuit television systems and select appropriate channels.
4. Verify that vapor detectors are on and indicating a normal condition.
5. Turn on 5 and 24 VDC power supply switch on control console.
6. Pull emergency stop button on control console.
7. Energize reset switch.

TEST CELL SETUP AND INSPECTION

1. Verify test cell door safety switch is operational.
2. Verify that fire extinguishers are in their proper location.
3. Verify caps and covers for open ports, lines, hoses, electrical connections, ducts, tubes, and cavities are properly configured.
4. Inspect area for objects or materials that might adversely affect testing and/or personnel safety.

WAVE ROTOR SETUP AND INSPECTION

1. Ensure the motor, motor mount, and wave rotor are secured together and to the table.
2. Ensure the motor and ESC have the correct wiring.
3. From control room, ensure laptop is talking to the DAQ and pressure/temperature measurements for the endwall sensors and venturi sensors.
4. Ensure that the ESC is connected to the correct BNC panel location.
5. Ensure that the signal generator is set to 1500 us, 100 hz, TTL.
6. Ensure that the ESC toggle is in the off position.
7. Connect the battery to the ESC.
8. Turn on the ESC.

CAN BURNER SETUP AND INSPECTION

1. Ensure all thermocouples (4) and pressure transducers (4) are functional and powered with 28V.
2. From control room, ensure laptop is talking to the DAQ and pressure/temperature Measurements for the combustor, ethylene spool piece, and hydrogen spool piece match ambient conditions.
3. Check all sonic nozzles for proper size.
4. Ensure that the ignition torch and dome loader are connected to the correct BNC panel location.
5. Ensure the hydrogen and ethylene bottle regulators are at the correct pressure.

TEST STARTUP

1. Turn on Red Warning Light using switch on console.
2. Secure the fuel room and test cell.
3. Sound horn and verify all personnel have left the test cell.
4. Close inner test cell door and display "Test in Progress" sign.

TEST CELL MUST BE SEALED FROM NOW ON

CAN BURNER AND WAVE ROTOR START AND RUN PROCEDURE

1. Begin flowing air from the compressed air lines.
2. Supply 1500 us to wave rotor and increase to around 1600 us or when the rotor begins spinning.
3. Ensure valves to igniter are closed. Begin ethylene fuel flow and spark the torch igniter.
Ignition will be known as combustor outlet temperature will rise dramatically.

4. If no temperature rise is observed, open valve to the hydrogen line while continuing the spark ignition. Once a flame is established, close off the hydrogen line and stop the torch igniter.
5. Once a steady flame is created, increase the wave rotor speed to the desired rate.
6. Monitor both the combustor outlet temperature for fluctuations and re-light the combustor if needed.
7. Monitor the mass flow coming into the CI venturi and decrease the compressed air line flow accordingly. Note: Do not decrease this line to zero unless a closing valve has been added. Decreasing the flow to much could cause harmful backflow.

CAN BURNER SHUTDOWN

1. Turn off fuel flow
2. Continue flowing compressed air to cool down the wave rotor and the can burner.

Wave Rotor SHUTDOWN

1. Decrease the rotational speed to a lower throttle setting.
2. Continue to flow air while monitoring temperatures inside the wave rotor.
3. Reduce air flow once the temperature has been reduced by half.
4. Stop providing signal to the wave rotor when the temperatures are adequate.

TEST CELL MAY BE UNSEALED AT THIS POINT

1. If necessary, purge the fuel line(s) between the bottles and rig.
1. De-energize red warning light switch located on console.
2. Push emergency stop (panic button).
3. De-energize 5/24 VDC power supply.

FUEL SHUTDOWN

1. Close valve(s) on the fuel supply bottle(s).
2. Check for empty fuel bottles--replace or note for future reference.

3. Close exhaust door.

CAN BURNER AND WAVE ROTOR SHUTDOWN AND INSPECTION

1. Unplug power.
2. Check all parts for heat stress when cool.

FINAL SHUTDOWN

1. Shut down facility software program.
2. De-energize Berkeley box(es).
3. Turn off closed-circuit televisions and system controller.

EMERGENCY PROCEDURES

1. Conduct emergency shut down procedures.
2. Make sure alarm is ringing, fire department is called (see step 4 for the latter).
3. Evacuate according to procedures.
4. Call the fire department: indicate exactly where to respond and the severity (ALPHA is false alarm/no rush, BRAVO is minor issue/please respond, CHARLIE is need help, DELTA is lives are at stake, ECHO is the base is endangered). Phone #:937-257-3033
5. Send a runner to Bldg 71a, A-stand to direct appropriate emergency response. Having a runner show up at the spot approved by procedures will indicate legitimacy.

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14. ABSTRACT The drone or micro-air vehicle industry has rapidly expanded in recent years. This thesis focuses on finding an alternative to power a Group 1 UAV with a maximum weight between 0 – 20 lbs. Nearly all of these vehicles are currently powered using battery-operated systems. While electrical power is currently the norm, the Air Force would like to pursue hydrocarbon-fueled aircraft for the higher energy to weight content. Normal internal combustion and turbine turbomachinery performs poorly at low mass flow rates required due to losses from a combination of factors. Wave rotors are being pursued another option for powering these small-scale devices. At these small flow rates, wave rotors have been shown to suffer less losses while requiring a less complex geometry. Previous work has examined the performance of a through flow wave rotor producing around 1.8 kW. This effort is investigating a smaller design around 500 W. Currently it utilizes a wave rotor with an external burner to complete a Brayton-like cycle. The previously validated NASA Glenn Research Center quasi-one-dimensional Computational Fluid Dynamics code was used to explore a range of design options for the new 500 Watt wave rotor. Analysis and computer aided design work was completed to ensure rotor operation at the elevated temperatures and high rotational speed. Testing was completed to find overall compressor efficiencies for comparison with typical turbomachinery.					
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