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REAL-GAS THERMODYNAMIC ANALYSIS OF THE WAVE-ROTOR COMBUSTION TURBINE

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ABSTRACT

Cyclic or pulsed combustors have the potential to produce a rise in pressure through confined combustion as opposed to conventional gas turbine engines that use steady flow combustion accompanied by loss in pressure. The pressure changes allow an improved work output through such unsteady devices. The thermodynamic analysis indicates a significant improvement in engine performance by implementing constant-volume combustion, particularly using the wave rotor combustion. The work output through a wave-rotor combustion depends on pressure exchange between the fluid, unsteady waves within the rotor and expansion within the wave-rotor. The thermodynamic analysis uses real-gas thermodynamic properties for accurate prediction of the engine performance.

INTRODUCTION

The combustors in conventional gas turbine engines are steady flow devices which burn fuel, ideally at constant pressure. The performance of gas turbine engines has seen significant improvement over the years with advancements in compressor flow and turbine cooling technology. However, further evolutionary improvement in turbomachinery performance brings diminishing returns. A major remaining problem is that burning gas at constant pressure causes appreciable loss in available work potential. The thermal efficiency of the engine is a function of the compressor pressure ratio. However, increasing the pressure ratio would require increased turbine work to drive the compressor, resulting in a higher weight for a given power output. Also, increased pressure would result in higher temperatures at the turbine inlet, but material limitations dictate the maximum inlet temperature and thus limit the heat addition in the combustor. The use of cooling air required by the high temperatures also tends to reduce the thermal efficiency of the engine. Further, increasing the combustor temperature results in higher NOx emissions [1]. The use of inter-cooling and reheating in gas turbines has been shown to improve performance but adds more volume and weight to the propulsion system. Such disadvantages of current steady-flow gas turbine engines have inspired the investigation of new thermodynamic cycles.

NOMENCLATURE

P	[Pa]	Pressure
T	[K]	Temperature
h	[J/kg]	Specific enthalpy
s	[J/kg]	Specific entropy
u	[J/kg]	Specific internal energy
c_p	[J/kg-K]	Specific heat
Q	[J/kg]	Heat release during combustion
W	[J/kg]	Specific work
V	[m/s]	Velocity
SFC	[kg/HP-hr]	Specific fuel consumption
Special characters		
η	[-]	Efficiency
π	[-]	Pressure ratio
γ	[-]	Gamma
Subscripts		
c		Compressor
t		Turbine
numerals		Defined thermodynamic states
A,B		Defined thermodynamic states
GT		Gas turbine
PGC		Pressure gain combustor
WRCT		Wave rotor combustion turbine

Wave rotors were initially developed in the 1940s and 50s as dynamic pressure exchangers for gas turbines, superchargers for piston diesel engines, and compressors for hypersonic wind tunnels [2]. Recent work has shown that a wave rotor can be used as a work-producing combustor for air breathing engines and land-based gas turbine units [3-4]. Wave rotor technology offers a method of sequencing confined combustion in multiple chambers to generate pressure gain and relatively steady inflow and outflow, making it suitable to integrate with inlets, nozzles and turbomachinery.

A simple illustration of a wave rotor is shown in Figure 1. The wave rotor consists of multiple passages arranged around the circumference of a shaft and housed within a rotating drum [5-6]. The drum rotates between two stationary end plates, each of which has ports that control the fluid flow through the passages. Due to the rotation of the drum, the passages are periodically exposed to the upstream and downstream flows through the ports. The sudden opening and closing of the ports initiates gas-dynamic compression and expansion waves within the channels. The gas is ignited after pre-compression by the shock wave, and the combustion mode can be premixed turbulent deflagration or detonation [6]; deflagration is desirable for easier integration with turbomachinery.

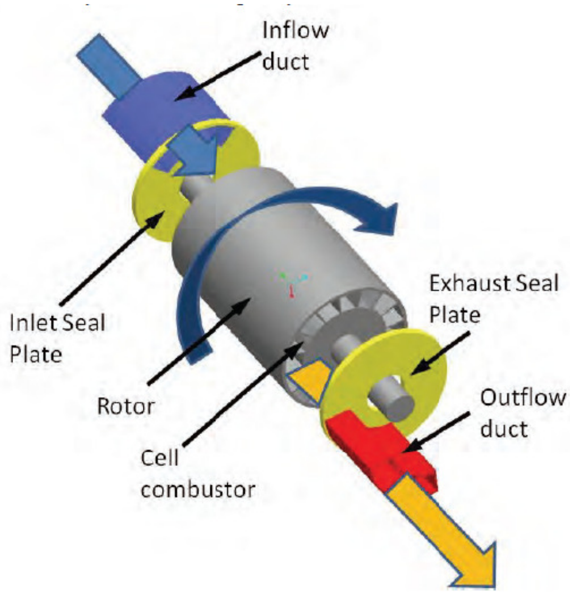

Figure 1 Illustration of a wave rotor [3]

The wave rotor can be used as a "wave turbine" by employing curved channels so that the high-speed flow turning and discharge results in a reaction torque and generation of shaft power output. With curved channels the wave rotor achieves shock compression, fast deflagration, gas expansion and shaft power, allowing for a lower pressure ratio compressor upstream and a turbine with fewer stages downstream. Additionally, the shaft output from the wave rotor can be used to drive the compressor or fan upstream.

WAVE ROTOR CYCLE

The wave rotor combustion cycle is illustrated in Figure 2, which shows an unwrapped view of the wave rotor with fluid flowing from left to right and rotation in the upward direction. As the air enters the wave rotor, the rotation of the channels causes a sudden closure of the exhaust port which creates a shock compression wave. The gas undergoes pre-compression by the shock and is then ignited by an igniter located opposite the inlet port. The igniter initiates continuous and fast confined combustion, reducing the residence time of the hot gases which decreases the amount of NO_x produced.

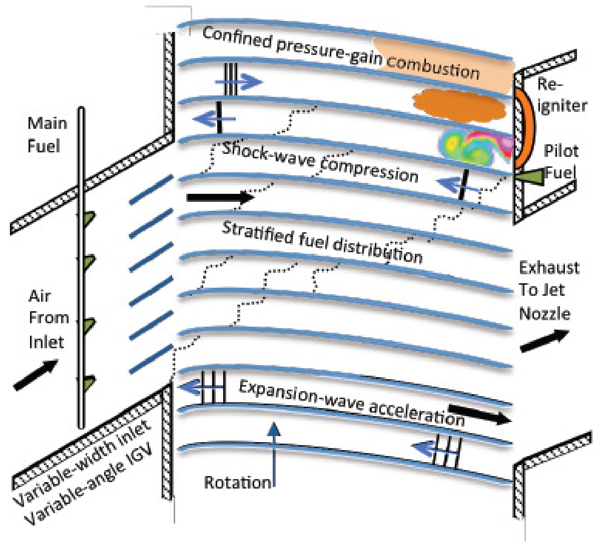

Figure 2 Internal processes in a wave rotor with curved channels

The channel then rotates and is opened to the exhaust port, which initiates an expansion wave and the burnt mixture flows out of the exhaust port. As the channel continues to rotate, it becomes aligned with the inlet port, accepting a fresh charge of air. A portion of the channel is still venting out the exhaust gas simultaneously to the filling of fresh air. The subsequent sudden closing of the exit port generates a hammer shock which propagates towards the inlet wall. The velocity profile of the curved channel shown in Figure 3 indicates that the work output from a turbomachine is dependent on the blade angle and turning of the flow from inlet to outlet. The steady inflow and outflow of fluid and the modest operating temperature allow for easy integration of the wave rotor with the turbine and compressor.

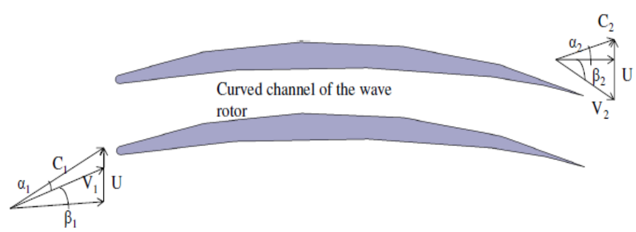

Figure 3 Velocity diagram inside a single channel of the wave rotor

V_1=Velocity of fluid relative to the blade at inlet
C_1=Velocity of fluid at inlet
U=Rotational velocity of the rotor
V_2=Velocity of fluid relative to the blade at outlet
C_2=Velocity of fluid at outlet
α, β= air angles

$$W = U * (\tan\beta_1 - \tan\beta_2)$$

THERMODYNAMIC ANALYSIS

The thermodynamic analysis follows an air standard cycle, using heat addition in place of fuel burn in the combustor to simplify the analysis. The air is considered as a thermally perfect gas but not a calorically perfect gas, so that realistic changes in specific heat are included. The analysis using constant gas properties was performed in previous work [7-8]. The reference case of a conventional gas turbine follows a Brayton cycle as shown in the *T-s* diagram and engine schematic in Figure 4. The cycle consists of compression from states 1 to 2, nearly constant pressure combustion from states 2 to 3_b and turbine expansion from states 3_b to 4_b. The wave rotor combustion turbine cycle follows a Humphrey cycle, with the conventional combustion process replaced with constant-volume wave rotor combustion from states A to B, and as explained later, the inclusion of internal wave compression from states 2 to A and wave expansion from states B to 3.

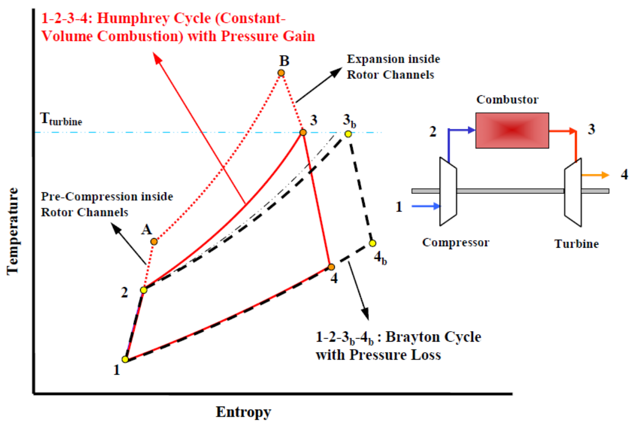

Figure 4 *T-s* diagram and schematic with station numbering [9]

Conventional gas turbine cycle

The air at the inlet of the compressor is assumed to be at ambient conditions (P_1, T_1), the adiabatic efficiency of the compressor is η_c, and the compressor pressure ratio (Π_c) is varied independently to study the performance of the cycle. The compressor exit temperature is calculated using Cantera thermodynamics software [10] by specifying enthalpy and pressure at the compressor exit (h_2, P_2). The exit pressure and enthalpy are calculated as:

$$P_2 = P_1 * \Pi_c \qquad (1)$$

$$h_2 = h_1 + \left(\frac{h_{2s} - h_1}{\eta_c}\right) \qquad (2)$$

The ideal enthalpy (h_{2s}) corresponding to a given pressure at state 2 is calculated using Cantera by equating the entropy at states 1 and 2. The work done in compressing the gas from P_1 to P_2 is calculated as:

$$W_c = h_2 - h_1 \qquad (3)$$

The next stage in the conventional gas turbine cycle is constant pressure combustion, with heat addition Q, determined by an assumed turbine inlet temperature T_3. Though the combustion is ideally at constant pressure, in reality there will be some pressure loss at the exit of the combustor. The heat addition Q and exit pressure P_3 from the combustor are calculated as:

$$P_{3b} = P_2 * (1 - \Delta p) \qquad (4)$$

$$Q = h_{3b} - h_2 \qquad (5)$$

The final stage in the gas turbine cycle is expansion through the turbine from state 3_b to 4_b (ambient conditions) with assumed turbine efficiency η_t. Cantera is used to compute the ideal enthalpy h_{4s} from entropy and pressure, and the exit temperature from the enthalpy and pressure (h_4, P_1), where the exit enthalpy is calculated as:

$$h_{4b} = h_{3b} + \left(\frac{h_{4bs} - h_{3b}}{\eta_t}\right) \qquad (6)$$

The turbine work, a portion of which is used to drive the compressor upstream, is calculated as:

$$W_t = h_{3b} - h_{4b} \qquad (7)$$

The net specific work and thermal efficiency for the ideal gas turbine cycle are given by:

$$W_{gt} = W_t - W_c \qquad (8)$$

$$\eta_{gt} = \frac{W_{gt}}{Q} \ . \qquad (9)$$

Pressure gain combustion cycle

In the pressure gain combustion cycle, the analysis for the compressor and turbine remains unchanged, but the analysis from state 2 to 3 differs due to the introduction of constant volume combustion instead of constant pressure combustion. The internal process for a wave rotor follows the path 2-A-B-3 on the *T-s* diagram in Figure 4, where the states A and B denote intermediate states. The gas undergoes shock compression from states 2 to A, constant-volume combustion from states A to B and finally expansion from states B to 3. In the case of pressure gain combustion without work output, the pressure at the exit of the wave rotor, P_3 is greater than pressure at inlet of the wave rotor, P_2, indicating a gain in pressure during the combustion process from state 2 to 3. The energy balance of the wave rotor with steady inflow and outflow shows that the enthalpy change from state 2 to state 3 equals the internal energy change from states A to B as the work output is equal to zero. The thermodynamic properties at state A are estimated from an assumed inflow Mach number and shock wave relations given by:

$$V_{2-r} = V_2 - V_A \qquad (10)$$

$$M_{inlet} = M_2 - \left(\frac{a_A}{a_2}\right) * M_A \qquad (11)$$

$$M_A = \frac{(\gamma - 1) * M_2^2 + 2}{2 * \gamma * M_2^2 - (\gamma - 1)} \qquad (12)$$

The value of γ for Equation 12 is calculated using Cantera by specifying the inlet thermodynamic state (P_2,T_2). The temperature and pressure at state A can be calculated from the Mach number obtained from Equation 11. The thermodynamic properties at state B are computed from the energy balance equation:

$$Q = U_B - U_A = h_3\,. \qquad (13)$$

The pressure at the exit of the wave rotor combustor P_3 is iterated until the turbine inlet temperature equals the assumed value. The net specific work output and thermal efficiency from the thermodynamic cycle involving a pressure gain combustor are given by:

$$W_{pgc} = W_t - W_c \qquad (14)$$

$$\eta_{pgc} = \frac{W_{pgc}}{Q} \qquad (15)$$

Wave rotor combustion turbine cycle

For the wave rotor combustion turbine cycle, the analysis for the compressor and turbine remains unchanged from the ideal gas turbine cycle, but the combustion cycle is different from both the ideal gas turbine cycle and pressure gain combustion cycle. The thermodynamic properties at state A are estimated from an assumed inflow Mach number and the shock wave relations (Equations 10-12). The energy balance for the work-producing wave rotor combustion turbine is not the same as pressure gain combustion as it involves work W_{comb} (Equation 17), so an iterative approach is used to calculate pressure P_B and temperature T_B. There will be no pressure gain for the work producing combustor, so the pressure at the exit of the wave rotor combustor, P_3 will be equal to the pressure at the inlet of the combustor, P_2. The expansion from state B to state 3 assumes a given expansion efficiency. The entropy at state 3, S_3, is calculated using Cantera by specifying the pressure and temperature (P_3, T_3). An initial guess is made for the pressure at state B, and since the combustion takes place at constant volume, Equation 16 is then used to calculate the temperature at state B:

$$\frac{P_A}{P_B} = \frac{T_A}{T_B}\,. \qquad (16)$$

The entropy at state B is calculated by specifying pressure and temperature (P_B, T_B). The pressure at B is iterated until the entropy at states B and 3 become equal. The internal energy at states A and B are estimated using Cantera by specifying the pressure and temperature at the respective states. The net work from the combustor can be calculated using Equation 17:

$$W_{comb} = (U_B - U_A) - (h_3 - h_2)\,. \qquad (17)$$

The net specific work and thermal efficiency for the ideal gas turbine cycle and work-producing wave rotor combustion turbine cycle are given by:

$$W_{wrc} = W_t + W_{comb} - W_c \qquad (18)$$

$$\eta_{wrc} = \frac{W_{wrc}}{Q}\,. \qquad (19)$$

RESULTS AND DISCUSSION

The design conditions considered for the thermodynamic cycle analysis are listed in Table 1. The thermal efficiency and specific power output from the thermodynamic cycle are calculated as a function of the compressor pressure ratio, which is varied from 5 to 70. The thermal efficiency as a function of compressor pressure ratio for both a real gas and an ideal gas with constant thermodynamics properties are illustrated in Figure 5.

Table 1 Design conditions for the thermodynamic cycle

Ambient pressure (Pa)	100000
Ambient temperature (K)	300
Turbine inlet temperature (K)	1600,1400
Compressor efficiency (%)	0.85
Turbine efficiency (%)	0.9
Pressure drop in combustor (%)	5
Mach number at wave rotor inlet	0.65
γ (constant gas properties)	1.31
Gas constant R for air (J/kg-K)	287

The performance parameters calculated using real gas properties were very close to the corresponding values calculated using constant gas properties. The difference in the results occurs because the specific heats of the gases do not remain constant over the range of temperatures in the cycle.

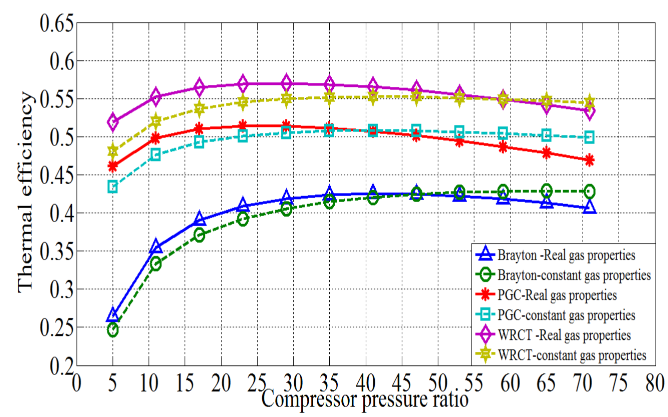

Figure 5 Thermal efficiency as a function of pressure ratio for a turbine inlet temperature of 1600K

The thermal efficiencies for the three thermodynamic cycles follow a similar trend but with one very important difference, in the case with real gas properties, the efficiency increases with increasing compressor pressure ratio and remains nearly constant after reaching its a maximum value. However, for the case using the constant gas properties listed in

Table 1, the efficiency decreases significantly after reaching its maximum value. Therefore, using real gas properties as the temperature changes throughout the cycle is critical for accurate prediction of the thermal efficiency. It is observed in Figure 5 that, when compared to the conventional Brayton cycle at high pressure ratios, the pressure gain combustion cycle achieves 17 percent reduction in fuel consumption rate and thus leading to reduction in carbon-dioxide emissions, and the wave rotor combustion turbine cycle shows an additional 10 percent improvement. At a low pressure ratio of 5, the pressure gain combustion cycle shows a 75% improvement over the Brayton cycle, while the wave rotor combustion turbine cycle shows a further 20 percent reduction in fuel consumption rate.

Thermal efficiency is an important factor during design of a gas turbine engine as it represents the effective conversion of chemical energy to useful work; however it should not be the only parameter used to assess performance. The thermal efficiency as a function of specific work output is illustrated in Figure 6. The constant volume combustion cycle achieves significant improvement in work output as compared to the conventional constant pressure combustion cycle. The compressor pressure ratio is a variable along each curve; its optimal value usually would lie between the points of maximal efficiency and the point of maximal specific work, depending on their relative importance in an application. It is observed that at the optimal point on the curve using real gas properties, there is a 25 percent improvement in the specific work output for the pressure gain combustion cycle and a 50 percent further improvement in work output for the wave rotor combustion turbine cycle.

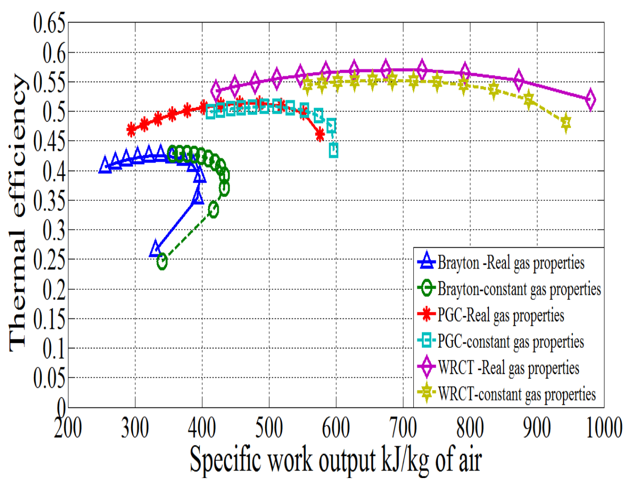

Figure 6 Specific work output vs Thermal efficiency for turbine inlet temperature of 1600k

In general practice, the pressure ratio of the compressor is increased in order to improve the thermal efficiency, as illustrated in Figure 5. However, at the same time the work output decreases with increasing pressure ratio as shown in Figure 7. For a higher pressure ratio, a larger fraction of the turbine work is used to drive the compressor and so the work output decreases; this means that a greater portion of chemical energy is not converted to propulsive work. So increasing the pressure ratio in order to increase thermal efficiency is negated by the decrease in work output.

The improvement in efficiency even at low pressure ratios and a wide operating range makes the wave rotor combustion technology a good prospect for the future. Also, because the wave rotor combustor can operate at lower compressor pressure ratios the turbine driving the compressor can have fewer stages. This would reduce the weight and size of the engine considerably, allowing for a higher thrust to weight ratio for the engine.

It is also important to consider the turbine inlet temperature while estimating the performance potential of a new engine design. To achieve higher work output, the turbine inlet temperature could be increased so that there is a higher degree of expansion through the turbine. However, increasing the temperature pushes the turbine material to its thermal limit and causes significant stress on the turbine blades. The work output as a function of the turbine inlet temperature is shown in Figure 8. When there is an increase of 200 K (from 1400 K to 1600 K) in the turbine inlet temperature, the work output improves considerably.

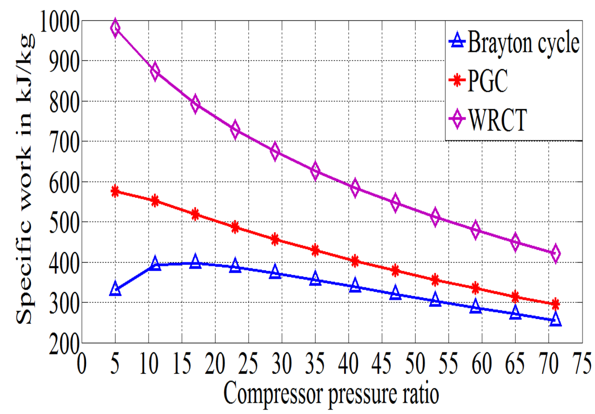

Figure 7 Specific work output vs Compressor pressure ratio for turbine inlet temperature of 1600k

For the Brayton cycle, at the higher turbine inlet temperature (1600 K) the specific work output increases by 150 kJ/kg as compared to the lower turbine inlet temperature (1400 K) at a pressure ratio of 30. For the pressure gain combustion cycle and the wave rotor combustion turbine cycle, there is also an increase in work output with increase in turbine inlet temperature, but even at lower turbine inlet temperatures, these devices tend to perform better than the ideal gas turbine engine following a Brayton cycle. Therefore, engines using pressure gain combustion or a wave rotor combustion turbine can achieve comparable work output with less stress on the turbine blades. At a pressure ratio of 30 the pressure gain combustion cycle increases the work output by about 130 kJ/kg at an elevated turbine inlet temperature, and the wave rotor combustion turbine cycle shows an improvement of 230 kJ/kg at an elevated turbine inlet temperature.

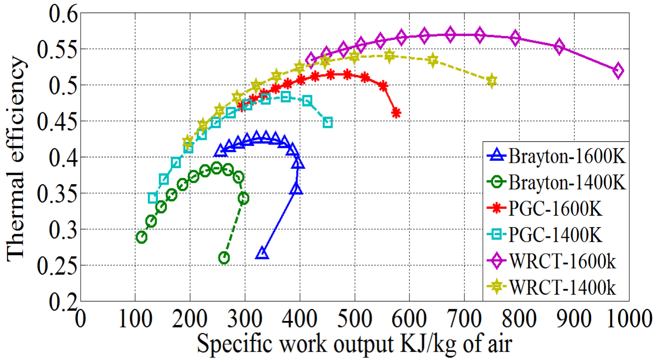

Figure 8 Illustration of performance as a function of turbine inlet temperature

However, increasing fuel prices and the environmental impact of greenhouse gases have pushed the designers to improve fuel efficiency of aircraft engines [11]. With the wave rotor technology we can achieve higher work output, lower specific fuel consumption, and better thermal efficiency while reducing the size and weight of the propulsion system. These advantages make wave rotor technology an attractive option for future engine development. Therefore, we performed a simple initial study to investigate the feasibility of integrating a gas turbine engine with wave rotor technology. A compressor pressure ratio of 30 was selected to study the performance, as most of the midsized aircrafts use an engine that operates between 30-40 overall pressure ratios. The specific fuel consumption as a function of specific work output is plotted in Figure 9 and is calculated as:

$$SFC = \frac{1}{LHV\ of\ fuel * thermal\ efficiency} \qquad (20)$$

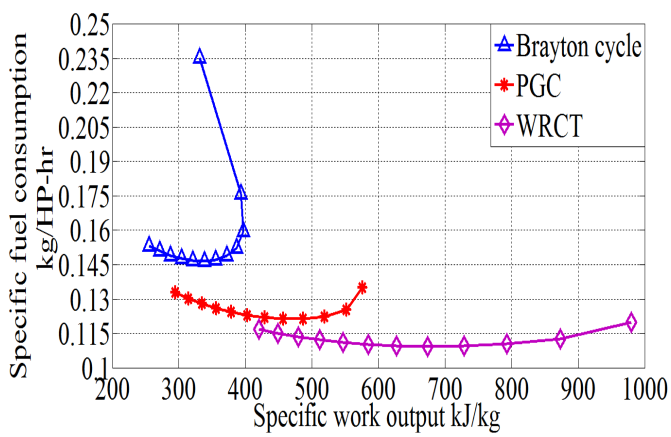

Figure9 Illustration of specific fuel consumption as a function of specific work (1600K)

Assuming that the shaft power from an ideal gas turbine engine cycle with overall pressure ratio of 30 is close to 10000 HP (typical for a standard engine), the same pressure ratio used with a wave rotor combustion turbine cycle would provide a shaft output of 18000 HP, an 80% increase. The specific fuel consumption for the ideal gas turbine and wave rotor combustion turbine cycle was calculated and the wave rotor combustor is found to be 30 percent more efficient in terms of specific fuel consumption as compared to the Brayton cycle (Table 2). From these results it can be observed that the air intake needed for a work-producing wave rotor combustion turbine is significantly less compared to the ideal gas turbine engine that follows a Brayton cycle.

Table 2 Performance comparison between Brayton and wave rotor combustor cycle at 30 pressure ratio

Performance parameter	Brayton engine	Wave rotor combustor
Shaft power HP	10000	18000
Specific fuel consumption kg/HP-hr	0.16	0.11

The predicted improvement in fuel efficiency suggests that wave rotor technology would be a good option for a midsized aircraft that runs on an overall pressure ratio between 30 and 40 such as the Boeing 737. Prior work [3] has shown the use of wave rotor technology in low pressure ratio scenario such as the Rolls-Royce Model MT5S engine. Future work will include analysis to investigate the use of a wave rotor combustion turbine in the engine for a midsized passenger aircraft and to predict the fuel burn benefit over the current gas turbine engines.

CONCLUSIONS AND FUTURE WORK

The performance of the wave rotor combustion turbine shows a significant potential improvement over the Brayton cycle in terms of thermal efficiency, specific work output and specific fuel consumption. The increase in thermal efficiency indicates improved conversion of chemical energy to useful work. The increase in the specific work output may allow for a reduction of propulsion system weight and a resulting increase in the thrust to weight ratio. Finally the improvement in specific fuel consumption will significantly decrease the amount of fuel burnt during an aircraft mission from takeoff to landing. The wave rotor combustion turbine performs better than a traditional gas turbine engine following a Brayton cycle at both low and high pressures. Because the wave rotor combustor has pre-compression and expansion of the gas during combustion, it allows the use of a compressor with a lower pressure ratio and a turbine with fewer stages. A lower pressure ratio compressor tends to be significantly lighter and occupies less volume, leading to a lower engine size and weight. The specific power from a wave rotor cycle is greater than that from the Brayton cycle, thus allowing the use of a smaller fan for the first stage of the propulsion system. The turbine can be designed for a lower inlet temperature and still achieve the same power as that of a Brayton cycle. The work output is not heavily dependent on the turbine inlet temperature, which is the case in a conventional gas turbine unit where a typical emphasis of the design is to attain a higher turbine inlet temperature to extract more work from the device. Prior experimental work has been conducted at the Zucrow Laboratories at Purdue University [12-14] to demonstrate the performance of a straight channel wave rotor combustor. The experimental rig, with rotor speed of 2100 rpm, air flow rate of 5 kg/s and premixed C_2H_4 as the fuel, is shown in Figure 10 [12]. It is emphasized that this thermodynamic study indicates the potential benefits, but detailed design of the wave rotor combustion turbine is needed to realize these benefits effectively. In particular, the rotation speed of the WRCT may impose limits on the amount of power that can be generated. Further, the outflow of a wave rotor combustor tends to be non-uniform. There may be additional losses associated with recovering useful work from the non-uniform outflow using conventional turbines. Indeed, a significant motivation to generate power within the wave rotor is that it mitigates the impact of non-uniform flow in downstream conventional turbines.

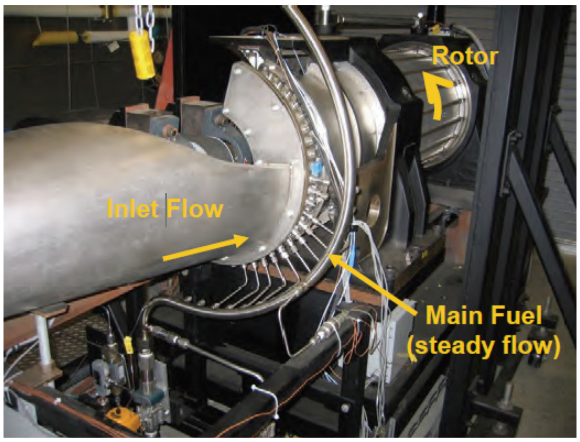

Figure 10 The wave rotor combustor rig used at Purdue Zucrow labs [12]

Future work will include both numerical and experimental studies. We will continue development of a computational fluid dynamics (CFD) code that incorporates curved channels in the wave rotor to predict the fluid dynamics and work output. Experimental work will focus on designing, building and testing a curved channel wave rotor combustor and studying the gas dynamics inside the wave rotor using advanced diagnostics such as particle image velocimetry (PIV). The performance of the wave rotor will be analyzed by integrating it with different aircraft configurations including midsized passenger aircraft, business jets, supersonic jets, unmanned aerial vehicles (UAVs) and hybrid electric aircrafts.

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