

Feasibility Study of Integrating Four-Port Wave Rotors into Ultra-Micro Gas Turbines (U μ GT)

Florin Iancu*, Pezhman Akbari* and Norbert Müller[§]
Michigan State University, East Lansing, MI, 48824

Ultra-micro gas turbines (U μ GT) have shown difficulties in obtaining high overall thermal efficiency and output power, resulting from miniaturization. Particularly, obtained compressor efficiencies have been as low as 40-50%, reducing optimum pressure ratios down to about 2. This work presents investigations of the feasibility and potential of integrating four-port wave rotors in microfabricated gas turbines to increase compression efficiency and optimum overall pressure ratio, hence increase overall cycle efficiency and power output. Practical implementation schemes and results of efficiency estimates are shown. The wave rotor efficiency is estimated first by simple extrapolation and then verified by a mathematical model. The model is based on gas dynamic equations for a moving normal shock wave in a channel and considers wall friction of the gas flowing through the channel. Knowing the inlet conditions and the pressure gain across the shock, the overall efficiency of the compression process in a wave rotor channel can be predicted. The results suggest that a compression efficiency in the range of 70-80% can be achieved in ultra-micro wave rotors. Based on thermodynamic cycle analyses a performance map was created that also gives optimum pressure ratios for a typical U μ GT application.

Nomenclature

$C_{p_{air}}$	=	specific heat for air
$C_{p_{gas}}$	=	specific heat for gas
f	=	friction coefficient
γ_{air}	=	air specific heat ratio
γ_{gas}	=	gas specific heat ratio
η_{PT}	=	turbine polytropic efficiency
η_{PC}	=	compressor polytropic efficiency
η_{WC}	=	wave rotor compression efficiency
η_{WT}	=	wave rotor expansion efficiency
L	=	half channel length
Π	=	pressure ratio
p	=	pressure
PR_w	=	wave rotor compression ratio
T	=	temperature
w, u	=	velocity

Subscripts

$comb$	=	combustor
s	=	shock
t	=	total (stagnation)

* Research Assistant, Department of Mechanical Engineering, 2500 Engineering Building, AIAA Student Member.

[§] Assistant Professor, Department of Mechanical Engineering, 2455 Engineering Building, AIAA Professional Member.

I. Introduction

The 21st century is an era dominated by mobility, gathering and exchanging of information, and a high demand of distributed power generation. Ultra-micro gas turbines (U μ GT) are seen as appropriate solutions for propelling mini unmanned air-vehicles (UAV) and powering miniaturized wireless sensors and equipment on board. This vision has been inspired by the improvements in Micro Electrical Mechanical Systems (MEMS) technology. Furthermore, these microfabricated gas turbines are suitable for high-density, distributed and redundant power generation onboard aircrafts, and other vehicles. U μ GT can achieve higher power densities compared to larger gas turbines as indicated by the so called Cube-Square Law. Because the output power is proportional to the mass flow rate of working fluid through the engine and the mass or volume of the engine is proportional to the cube of the characteristic length, the power density ($Power/Mass=L^{-1}$) is increased as the device is miniaturized. Fundamentally, the specific work output (or thrust) of thermodynamic cycles is not a function of scale, but only of the flow thermodynamic properties at each state of the cycle.

Inspired by the microfabricated gas-turbine project at MIT in 1995¹, it has been envisioned that such a turbine is able to develop 10-20 W electrical energy as a gas turbine generator, or 0.1 N of thrust as a turbo jet engine. A typical size of an U μ GT is 20mm \times 20mm \times 4mm. Beside the micro gas turbine engine, the MIT research group has also focused on development of micro-scale high-speed compressor impellers², new fabrication techniques³, combustion systems^{4,5}, igniters⁶, and heat exchangers.⁷ The previous work on ultra-micro thermal systems includes the research conducted by Frechette on steam turbines for power generation.⁸ An important aspect of these systems are heat transfer and heat loss problems.⁹ Ribaud has also developed a numerical model to investigate the combustion chamber principles.¹⁰

Investigations of such U μ GT have shown difficulties in obtaining high efficiency and output power. The reasons are additional losses due to miniaturization and the dominant 2D geometries typical for MEMS-fabrication technology. The flow paths encounter often 90 degrees bends, which are known to decrease efficiency and mass flow rate. The compressor isentropic efficiency typically drops $\leq 60\%$. For instance, for an optimum pressure ratio and compressor characteristic of such engines, Müller and Frechette¹¹ have found a compressor isentropic efficiency of around 40%. For a slightly larger scale compressor (12 mm in diameter), but not using microfabrication technology, Johnson and Kang¹² have found an efficiency of 65% using experimental and numerical methods.

From the above it can be concluded that innovations are desired to attractively enhance the performance of U μ GT. Wave rotor technology has shown a huge potential for enhancing efficiency and output power of gas turbines especially in the range of smaller sizes, where the pressure ratio is typically low^{13,14}. To enhance the low performance of U μ GT, the idea of integrating a wave rotor into an U μ GT is proposed. This paper aims to investigate the feasibility of integrating wave rotor devices in such small engines.

II. Wave Rotor Technology

The potential of non-steady machines for performance enhancement of thermodynamic cycles has been recognized, but rarely exploited. Shock tubes, shock tunnels, pressure exchangers, pulse combustors, pulse detonation engines, and wave rotors are among the best-known wave devices developed so far. The basic concept underlying these devices is the transfer of energy between different fluids with shock and expansion waves. By generating compression and expansion waves in appropriate geometries, wave machines can transfer the energy directly between fluids without using mechanical components such as pistons or vaned impellers. In fact, these devices properly represent applications of classical non-steady one-dimensional compressible flow theory. The inherent non-linearity of large-amplitude wave phenomena in compressible flow fields and unusual geometry of non-steady devices has impeded the wide application of these machines in the gas turbine community. An innovative technology involving wave rotors that are state-of-the-art non-steady flow devices has provided new opportunities for further significant performance improvements of today's gas turbines.

Wave rotors do not use mechanical components such as pistons or vaned impellers to compress the fluid. Instead, the pressure rise is obtained by generating compression waves in appropriate geometries. It has been proved that for the same inlet and outlet Mach numbers the pressure gain in time-dependent flow devices can be much greater than in steady flow devices.¹⁵ As schematically shown in Fig. 1, the essential feature of these devices is an array of several channels arranged around the axis of a cylindrical drum. The drum rotates between two end plates each of which has a few ports or manifolds, controlling the fluid flow through the channels. The number of ports and their positions vary for different applications. Through rotation, the channel ends are periodically exposed to the ports located on the stationary end plates initiating compression and expansion waves within the wave rotor channels. Therefore, unlike a steady-flow turbomachine, which either compresses or expands the gas, both

compression and expansion are accomplished within a single component. To minimize leakage, the gap between the end plates and the rotor has to be very small or the end plates with sealing material could contact the rotor.

The rotor may be gear or belt driven or preferably direct driven by an electrical motor (not shown in the picture). The power required to keep the rotor at a correctly designed speed is negligible.^{16,17} It only needs to be sufficient to overcome rotor windage and the friction in the bearing and contact sealing if they are used. Alternatively, rotors can be made self-driving. This configuration called the “free running rotor” can drive itself by using the momentum of the inflow or outflow to rotate the rotor.¹⁸

In a conventional arrangement, a wave rotor is embedded between the compressor and turbine “parallel” to the combustion chamber. Figure 2 illustrates how a four-port wave rotor is used to top a gas turbine cycle. Air from the compressor enters the wave rotor (state 1) and is further compressed inside the wave rotor channels. After the additional compression of the air in the wave rotor, it discharges into the combustion chamber (state 2). Here, combustion takes place at a higher pressure and temperature than in the baseline engine. The hot gas leaving the combustion chamber (state 3) enters the wave rotor and compresses the air received from the compressor (state 1). Because of the energy transfer, the burned gas expands during the compression of the air and is afterward scavenged toward the turbine (state 4). Due to the pre-expansion in the wave rotor, the burned gas enters the turbine with a lower temperature than that of the combustor exit. However, the gas pressure is higher than the compressor exit pressure by the pressure gain obtained in the wave rotor. The turbine inlet total pressure is typically 15 to 20% higher than the air pressure delivered by the compressor. This is in contrast to the untopped engine where the turbine inlet pressure is always lower than the compressor discharge pressure due to the pressure loss across the combustion chamber. As a result, more work can be extracted from the turbine. Finally, the channels are re-connected to the compressor outlet, allowing fresh pre-compressed air to flow into the wave rotor channels and the cycle repeats.

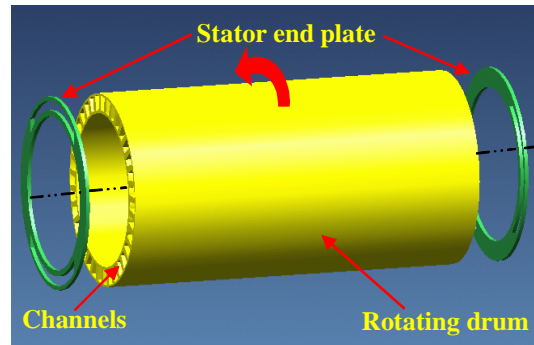


Figure 1: Schematic configuration of a typical wave rotor machine.

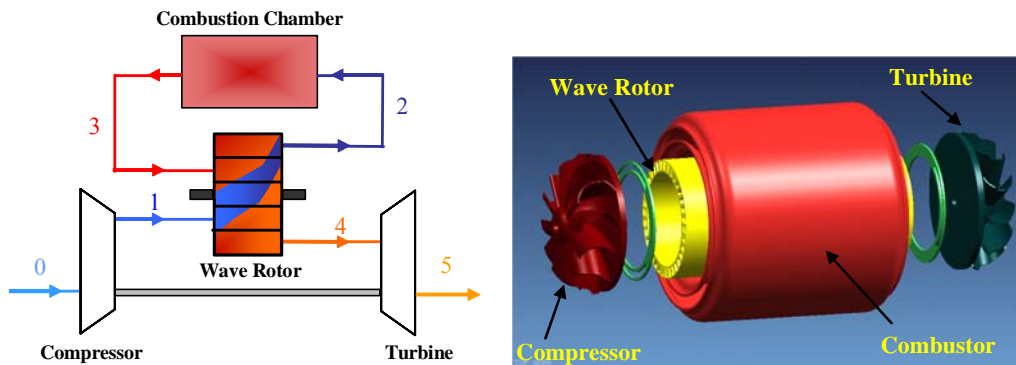


Figure 2: A schematic of a gas turbine topped by a four-port wave rotor.

The general advantage of using a wave rotor becomes obvious when comparing the thermodynamic cycles of baseline and wave-rotor-enhanced engines. Figure 3 shows a schematic temperature-entropy diagram of a turbine baseline engine and the corresponding wave-rotor-topped engine. Other advantageous implementation cases for the wave rotor into the given baseline engine are also possible.^{13,14,19} It is evident that both gas turbines are operating with the same turbine inlet temperature and compressor pressure ratio. However, the output work of the topped engine is higher than that of the baseline engine due to the pressure gain across the wave rotor. Because the amount of heat addition is the same for both cycles, the overall efficiency for the topped engine becomes higher than that of the baseline engine.

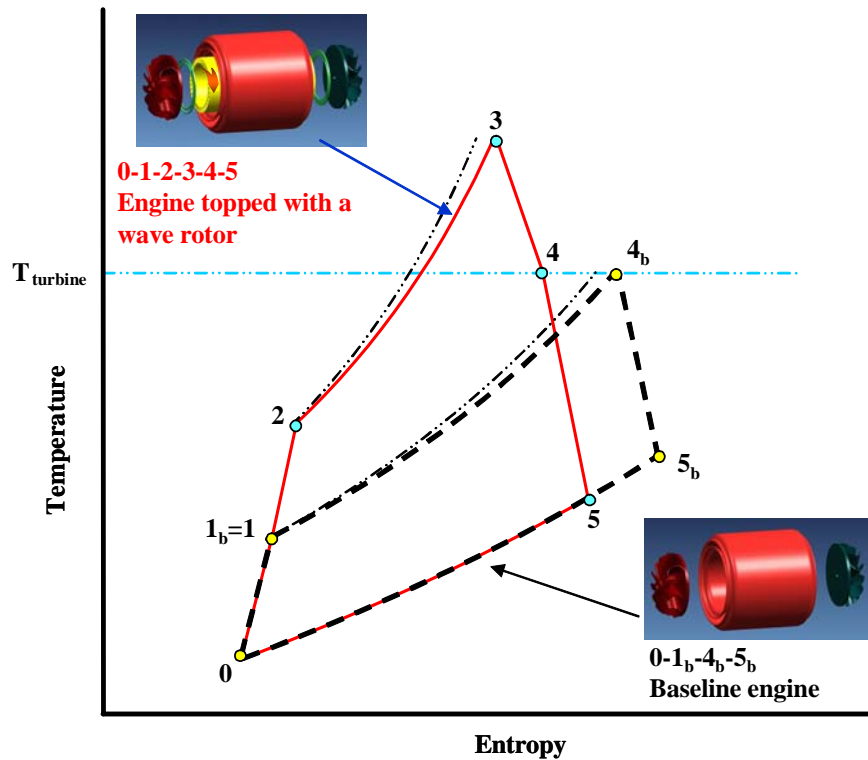


Figure 3: Schematic Temperature-Entropy diagram for a gas turbine with and without a wave-rotor.

There are several important advantages of wave rotor machines. Their rotational speed is low compared with turbomachines, which results in low material stresses. From a mechanical point of view, their geometries can be simpler than those of turbomachines. Therefore, they can be manufactured relatively inexpensive. Also, the rotor channels are less prone to erosion damage than the blades of turbomachines. This is mainly due to the lower velocity of the working fluid in the channels, which is about one-third of what is typical within turbomachines.¹⁶ Another important advantage of wave rotors is their self-cooling capabilities. They are naturally cooled by the fresh cold fluid ingested by the rotor. Therefore, applied to a heat engine, the rotor channels pass through both cool air and hot gas flow in the cycle at least once per rotor revolution. As a result, the rotor material temperature is always maintained between the temperature of the cool air, which is being compressed and the hot gas, which is being expanded.

III. Performance Enhancement of an U μ GT Using Wave Rotor Technology

The way in which the wave-rotor topping enhances the cycle at micro scale often differs from that at larger scales. At a large scale, mostly the goal is either to increase the cycle overall pressure ratio or to substitute the wave rotor for costly high pressure turbomachinery stages. The most significant performance gain has been found for engines with low compressor pressure ratio and high turbine inlet temperature.^{20,21} At the ultra-micro scale, the optimum cycle pressure ratio is very small, e.g. around 2, due to the low component efficiencies.¹¹ Thus, mostly a single-stage centrifugal compressor can easily generate the low optimum overall pressure ratio and a further increase with the same efficiency actually decreases the desired performance. Therefore, the wave rotor integration is most effective if its compression and expansion efficiencies are greater than those of the turbomachinery components of the baseline engine. This enhances not only the overall compression and expansion efficiencies, but it also increases the optimum cycle pressure ratio to a greater value allowing for additional performance improvements. In such a case in which the wave rotor compression efficiency is higher than that of the spool compressor, a wave rotor can enhance the performance of an U μ GT that was already designed for an optimum pressure ratio. While then the optimum overall compression ratio increases with the wave rotor integration, usually the pressure ratio of the spool compressor decreases. This can additionally enhance the isentropic efficiency of the spool compressor, provided its polytropic efficiency (aerodynamic quality) stays the same. These effects are schematically shown in Fig. 4.

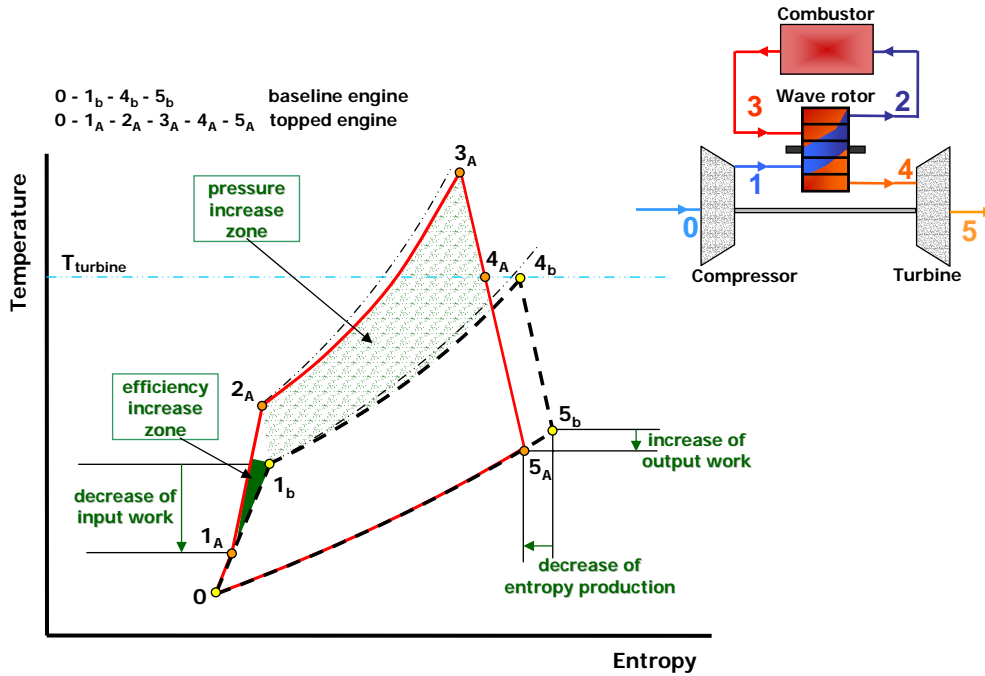


Figure 4: Temperature – Entropy diagram for gas turbines, with and without wave rotor (left). Block diagram of the thermodynamic cycle of the wave-rotor-topped gas turbine (right).

IV. Conceptual Designs for Wave Rotor Implementation into U μ GTs

Based on possible design restrictions and preferences, mainly three different advantageous conceptual designs for a four-port (or multiple of that) wave rotor integrated into a baseline U μ GT are suggested. Figure 5 shows a wave rotor added at outer diameter of the disk of the “classical” MIT baseline design.¹ This innovative design has several advantages. The wave rotor rotates with the compressor/turbine disk i.e. there is still only one single rotating disk in the engine. Additionally, because the wave rotor is a self-cooled device, it isolates the compressor disk from the combustion chamber while in the wave rotor the heat is given to the compressed air adding a recuperative effect. The end plates with the ports at either side of the wave rotor can be etched in the same wafer as the stationary guide vanes.

Ultra-Micro Turbine Design (U μ GT) - Design 1

Classical Design

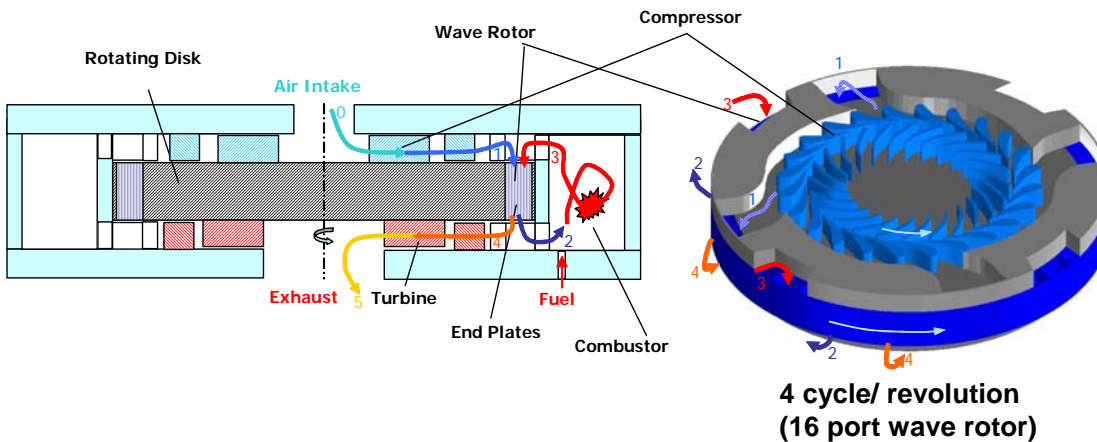


Figure 5: Conceptual designs of U μ GT – “Classic” design.

The second possible design implies using additional wafers allowing a multi layer rotor, as shown in Fig. 6. However, the diameter of the rotor is smaller than in the first design. This results in a smaller frontal area, which is favorable for propulsion of air vehicles and may generate less stress in the disks. Further, this design allows for better separation of the combustion chamber from the compressor, reducing the heat introduced to the compressor. The major challenge with this design is the perfect axial alignment of the compressor/turbine unit with the wave rotor, which may be achieved with the common laser aligning method. The flow connection from the compressor to the wave rotor may be viewed as a challenge in respect of keeping the pressure loss small. However, the equivalent diameter of this connection may be designed sufficient large. Further, this may aid in isolating the compressor case from the combustor heat, especially with the counter flow of the compressed air, where a regenerative effect is seen again.

The third design concept introduces a new idea in respect of having multiple wave rotors arranged circumferentially around the compressor/turbine unit, as shown in Fig. 7. The advantage of this design is that no additional stresses occur in the main rotor, which is the compressor/turbine disk. The stresses in the separate small diameter wave rotors are negligible since they can rotate at relatively low speed. Similar to the first classic design, this design requires less wafers than the second design, which translates into lower fabrication cost. The challenge associated with this design is driving all wave rotors at appropriate speed which may be achieved by arranging the wave rotor ports in proper angles, so that the impulse of the fluid streams can be utilized.

Ultra-Micro Turbine Design (U μ GT) - Design 2

Two-Layer Design

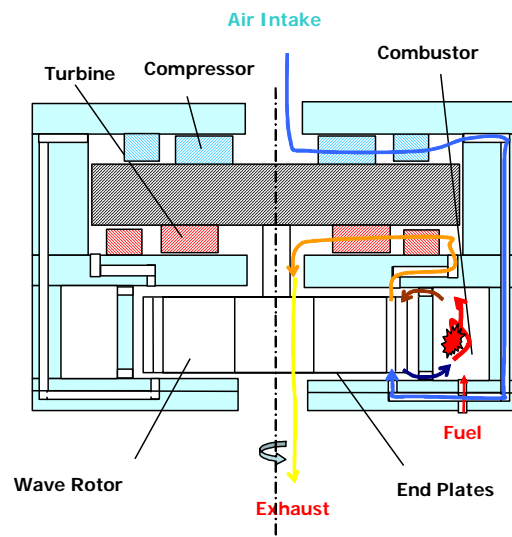


Figure 6: Conceptual designs of U μ GT – “Two-layer” design.

Ultra-Micro Turbine Design (U μ GT) - Design 3

Exterior Wave Rotor Design

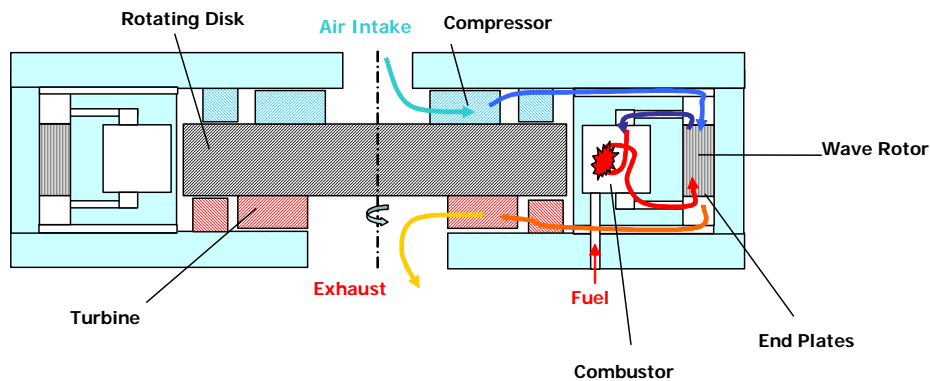


Figure 7: Conceptual designs of U μ GT – “External Wave Rotor” design.

V. Calculations of the Wave Rotor Efficiency

As explained above, the implementation of a wave rotor at ultra-micro scale appears most effective if its compression efficiency is greater than that of the baseline spool compressor. Whereas the latter ranges low around 50% at ultra-micro scale compared to about 70-90% at large scale, the compression efficiency of wave rotors (η_{WC}) have been found to be in the range of 70-86%, mostly assuming that the wave rotor expansion efficiency is equal to its compression efficiency ($\eta_{WT} = \eta_{WC}$). This may be considered as matching the efficiency of large scale compressors or turbines and as almost double of that achieved with ultra-micro scale compressors. The favorable wave rotor efficiencies are subsidized by the following. Taussig²² has reported $\eta_{WC} \cdot \eta_{WT} = 0.70-0.74$, measuring the overall efficiency and not distinguishing between compression and expansion efficiency. In experiments at Rolls-Royce, Moritz²³ has found $\eta_{WC} \cdot \eta_{WT} = 0.6$. Kollbrunner²⁴ has measured η_{WC} alone as 60-68%. Wilson and Paxson²¹ in their calculations have used $\eta_{WC} = \eta_{WT} = 0.83$, resulting in $\eta_{WC} \cdot \eta_{WT} = (0.83)^2 = 0.69$. Fatsis and Ribaud²⁰ have varied both compression and expansion isentropic efficiency between 80-86% in their performance calculations. Thus, it appears that $\eta_{WC} = \eta_{WT} = 0.83$ is a reasonable assumption for a large scale wave rotor with a cell length of about 200-300 mm. For the smallest documented wave rotor with a channel length of 69 mm²⁵, using the wave-rotor characteristic equation (Eq. (1) below) introduced by Wilson and Paxson²¹ and $\eta_{WC} = \eta_{WT}$, the authors calculated an isentropic compression efficiency of about 79%.

$$\frac{P_{14}}{P_{11}} = \Pi_{comb} \cdot PR_W \cdot \left\{ 1 - \frac{\frac{Cp_{air}}{Cp_{gas}} \frac{1}{\eta_{WT} \cdot \eta_{WC}} \frac{T_{11}}{T_{14}} \left[PR_W^{(\gamma_{air}-1)/\gamma_{air}} - 1 \right]}{1 + \frac{Cp_{air}}{Cp_{gas}} \frac{1}{\eta_{WC}} \frac{T_{11}}{T_{14}} \left[PR_W^{(\gamma_{air}-1)/\gamma_{air}} - 1 \right]} \right\}^{\frac{\gamma_{gas}}{\gamma_{gas}-1}} \quad (1)$$

Exploring the application of even smaller wave rotors suitable for ultra-micro gas turbines, the question arises if such wave rotor efficiencies can be maintained at ultra-micro scale. No values are available at this scale. However, the above results show no or only a small decrease in efficiency with reduced size, which encourages investigations at the ultra-micro scale.

Using the available wave rotor efficiencies above versus the corresponding wave rotor channel length, a trend can be generated as shown as a solid green line in Fig. 8. The simple linear extrapolation predicts a wave rotor efficiency at ultra-micro scale (about 1mm channel length) that is greater than 70%. Such a compression efficiency of a microfabricated wave rotor is much better than the obtained efficiencies of around 50% for microfabricated compressors.¹¹ Furthermore, efficiency values of compressors suitable for or corresponding to the reported wave rotor topped cycle are shown red in Fig. 8. This allows to show the red broken trend line for the compressor

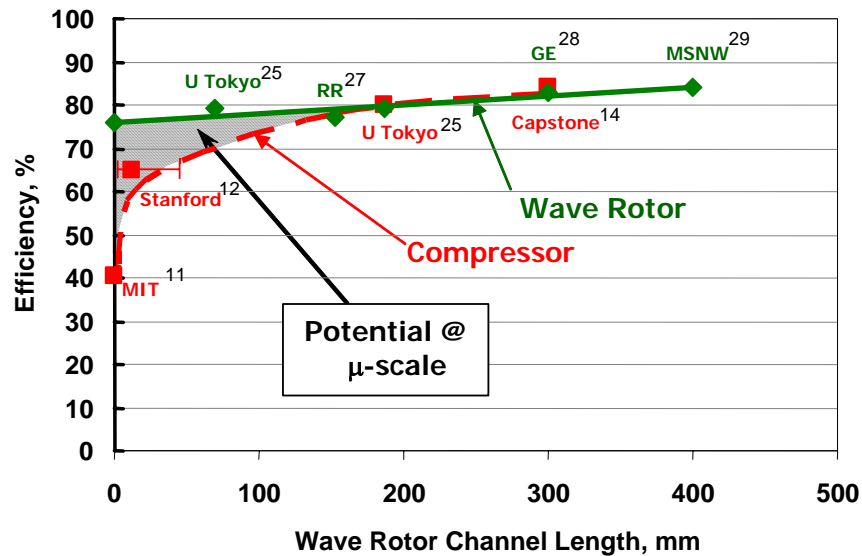


Figure 8: Efficiency trend of compression process. Green: wave rotor efficiency; Red: compressor efficiency.

efficiency corresponding to the wave rotor length. Both trends for the wave rotor efficiency and for the compressor efficiency coincide at larger scale. However, towards smaller wave rotor size for $U\mu GT$, the compressor efficiency trend falls far below the wave rotor efficiency trend. This clearly suggests an advantage of using a wave rotor for $U\mu GT$.

Since the trend of the wave rotor compression efficiency can only be considered as a very first and crude estimate, a mathematical model was created that better reflects the physical background. The model considers the entropy production by a single normal shock that runs through the wave rotor channel and the wall friction generated by the gas flowing through the channel. Heat transfer effects are neglected and losses in exterior ducting and during the channel opening and closing are not considered.

Focusing on the phenomenon occurring inside a single channel, the one-dimensional mathematical model is based on the gas dynamics of normal shock waves for one-dimensional flow as described by Anderson.²⁶ The model assumes a constant friction coefficient along the channel. The wave rotor channel is simulated as a tube with an equivalent diameter as shown in Fig. 9. The model relates the efficiency of the compression process to the velocity, pressure, and temperature of the gas at the entrance of the channel (u_1, p_{t1}, T_{t1}), the pressure ratio across the shock (Π_s), the friction coefficient, and tube dimensions. In Fig. 9, a shock wave is shown that moves in opposite direction to the flow. Friction is considered along the lengths L before and after the shock, assuming that the distance between points 2 and 3 is negligible. The shock wave position could be averaged to the middle of the channel, due to the fact that for low friction coefficients the resulting efficiency varies almost linearly with the position of the shock through out the channel as shown for the below example in Fig. 10 .

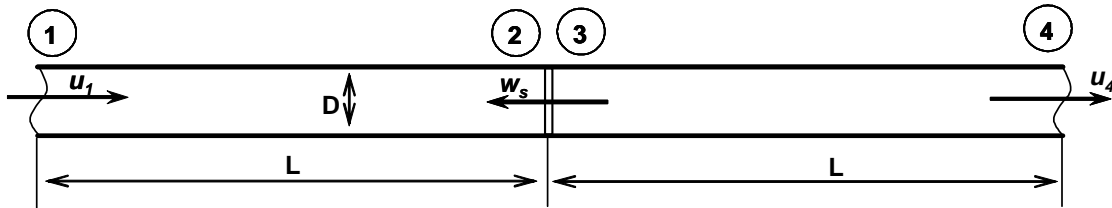


Figure 9: Model of wave rotor channel.

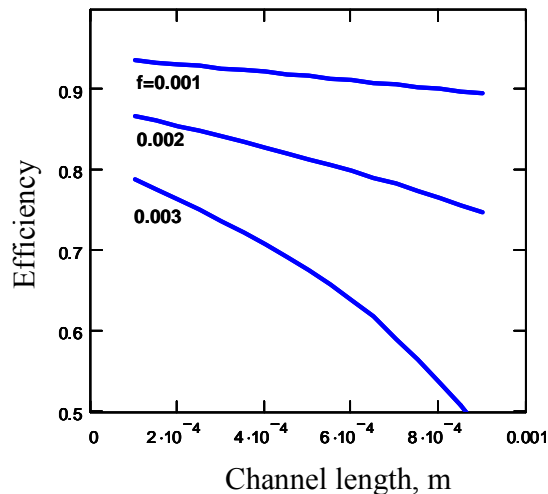


Figure 10: Efficiency of shock wave compression in micro wave rotor channel depending on shock wave position and channel friction coefficients for a ratio length/diameter of 10.

Assuming a friction coefficient of 0.003 is representative for air/silicon typical in microfabricated gas turbines, for a channel length of 1 mm and a cross-sectional area of 0.0078 mm^2 with inlet conditions: $p_{t1} = 194.5 \text{ kPa}$, $T_{t1} = 443.3 \text{ K}$, and conservative high $u_1 = 300 \text{ m/s}$, the efficiency of the compression process averages at 80% for the

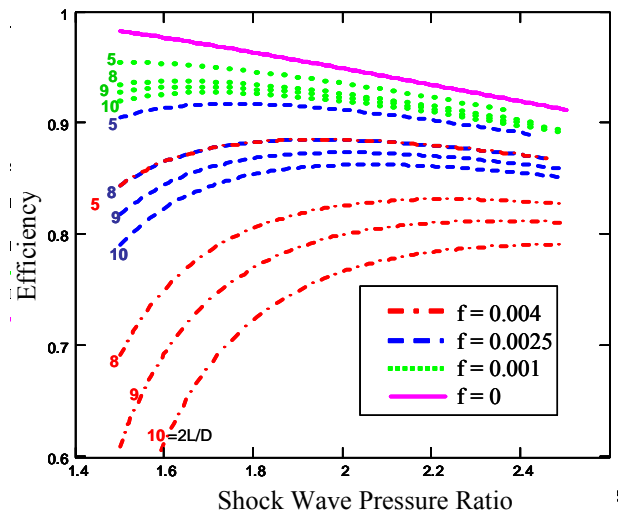


Figure 11: Efficiency of the compression process in micro wave rotor channel for different friction coefficients and length/diameter ratios.

relevant shock pressure ratios as shown in Fig. 11. Figure 11 shows the isentropic shock wave compression efficiencies versus the shock strength for various friction coefficients and channel length to diameter ratios. For a friction coefficient of zero, the upper most graph is independent of the channel geometry. When friction is considered, it is seen that smaller length to diameter ratios yield higher efficiencies, which is encouraging at the ultra-micro scale of microfabricated gas turbines where the channel length mostly turns out to be very short and channel diameters preferably should not be too small. Additionally, the efficiency of the shock compression process increases rapidly to around 80% with increasing shock strength up to approximately $I\Gamma_s = 1.7 \dots 2$. After this it stays almost constant, especially for greater length/diameter ratios and friction coefficients. This can be viewed as a significant advantage compared to compressors where a decrease in isentropic efficiency is expected at higher pressure ratios. Finally, Fig. 11 shows that the same efficiency is obtained if the loss coefficient (the product of friction coefficient and length/diameter ratio) is the same. While the efficiency curves approach the zero friction line with smaller loss coefficient, their best efficiency points move towards greater shock strength with greater loss coefficients. The model itself is applicable for both large scale and micro scale, since the gas dynamics of the shock waves are assumed to be the same and the friction losses are proportional to the dimensionless loss coefficient.

VI. Thermodynamic Cycle Investigations

After determining possible wave rotor efficiencies with the above approach, a thermodynamic calculation was performed to predict the performance enhancement of an U μ GT. The analytical thermodynamic model documented by Akbari and Müller¹³ was used to calculate the performance of both baseline and wave-rotor-topped cycles. The compressor inlet and outlet conditions are assumed to be the same for both engines. The baseline engine has a radial compressor with a pressure ratio of 1.94 and polytropic efficiencies of the compressor and turbine are 46% and 68%, respectively. Compression and expansion efficiencies of the wave rotor are both the same and equal conservatively assumed with 70%. The gases are treated as ideal gases with constant specific heat values ($C_{p,air} = 1.005 \text{ kJ/kgK}$ and $C_{p,gas} = 1.148 \text{ kJ/kgK}$) and constant ratios of specific heats ($\gamma_{air} = 1.4$ and $\gamma_{gas} = 1.33$). No losses are considered in the ducting, intake and exhaust. The intake conditions of the compressor are 101.3 kPa and 293 K, while the temperature at turbine inlet is fixed to 1465K. For a wave rotor pressure ratio of 1.8 and optimum overall pressure ratio, the thermodynamic calculation shows that the thermal efficiency is increased from 2.5% for the baseline engine to 7.5% for the wave-rotor-topped engine. This is equivalent to a relative increase of thermal efficiency of nearly 200% which is a significant value. Simultaneously, the compressor ratio reduces from 1.94 for the baseline engine to 1.44 for the enhanced engine.

Figure 12 shows a performance map in which both the baseline and the topped engines can be found. Variations of the specific work (blue), the overall thermal efficiency (green), and the specific fuel consumption (red) as functions of the wave rotor pressure ratio (PR_w) and compressor pressure ratio (abscissa) are shown. The main fixed parameters are the turbine inlet temperature (T_{t4}) and the polytropic efficiencies of the compressor (η_{PC}) and turbine (η_{PT}) indicated in the box in the middle of the map. The black solid optimum lines connect the optimum compressor pressure ratio points at each achievable wave rotor pressure ratio for highest overall efficiency, for highest specific work, and least specific fuel consumption (SFC) respectively. It can be seen how the optimum shifts towards lower spool compressor ratio and higher overall pressure ratio with increasing wave rotor pressure ratio as discussed above.

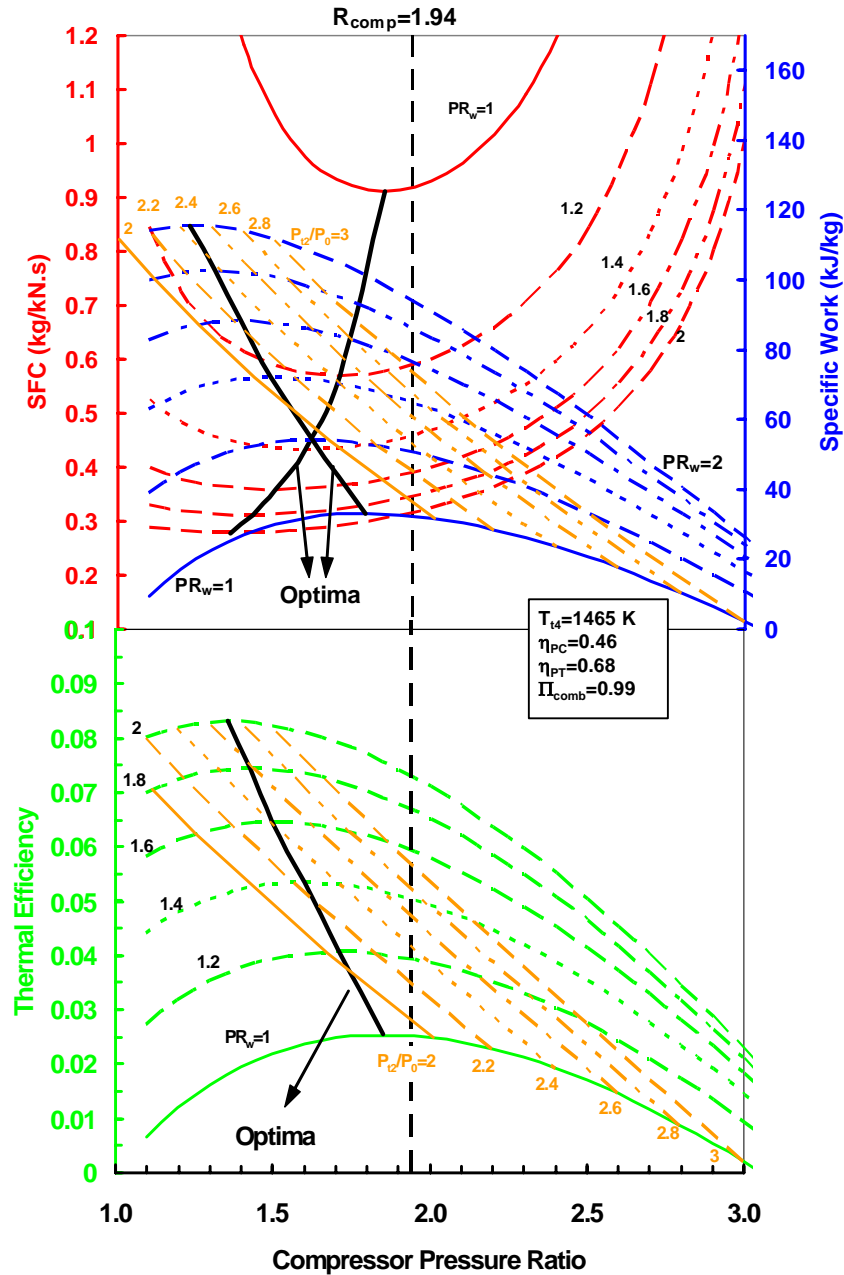


Figure 12: Performance Map of baseline and wave-rotor-topped ultra-micro gas turbines.

VII. Conclusion

Utilizing a wave rotor to improve the performance of an U μ GT appears to be a promising solution. Even if pressure ratio of the baseline engine is already optimized, the wave rotor can still enhance both the overall thermal efficiency and cycle specific work output if the wave rotor compression efficiency is higher than that of the baseline engine compressor. Adding a wave rotor also reduces the baseline compressor pressure ratio and the exit temperature of the compressor. Furthermore, this may reduce the compressor diameter and rotational speed which results in reduced mechanical and thermal stresses and relaxed design constraints. From the manufacturing point of view, adding a wave rotor is much easier at micro scale than at macro scale because the wave rotor can easily be etched in silicon due to its common extruded 2D shape. Additionally, in a regenerative way the wave rotor allows to harvest some of the significant amount of heat conducted away from the combustor through the structure, which is a severe problem for microfabricated gas turbines and also reduces the efficiency of the spool compressor severely. Three possible designs for integrating a wave rotor in microfabricated gas turbines are introduced. Based on documented wave rotor efficiencies at larger scale and subsidized by a gasdynamic model that includes wall friction, the wave rotor compression efficiency at microfabrication scale could be estimated with about 70%, which is much higher than the obtained efficiency of compressors in a microfabricated gas turbine. It is shown that at such ultra-micro scale, the wave rotor can have the highest efficiency for shock wave pressure ratios in the range of 1.7-2, assuming that the microfabrication can generate a smooth enough surface with a low friction coefficient. The results show that the efficiency depends not only on pressure gain across the shock wave traveling through the wave rotor channel, but also depends highly on the loss coefficient for the channel geometry. According to the here employed model that is applicable for all wave rotor sizes, shorter wave rotor channels with larger diameter let expect a higher compression efficiency of the wave rotor.

Acknowledgments

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