ABSTRACT
Results are presented predicting the significant performance enhancement of two small gas turbines (30 kW and 60 kW) by implementing various wave rotor topping cycles. Five different advantageous implementation cases for a four-port wave rotor into given baseline engines are considered. The compressor and turbine pressure ratios, and the turbine inlet temperatures vary in the thermodynamic calculations, according to the anticipated design objectives of the five cases. Advantages and disadvantages are outlined. Comparison between the theoretic performance (expressed by specific cycle work and overall thermal efficiency) of wave-rotor-topped and baseline engines shows a performance enhancement by up to 33%. The results obtained show that almost all the cases studied benefit from the wave-rotor-topping, but the highest gain is obtained for the case in which the topped engine operates with the same turbine inlet temperature and compressor pressure ratio as the baseline engine. General design maps are generated for the small gas turbines, showing the design space and optima for baseline and topped engines.

Keywords: Wave Rotor, gas turbine, efficiency, topping cycle

INTRODUCTION
Gas turbines are typical power sources used in a wide size range for stationary power plants and for propulsion. Recently, there has been considerable interest in the research, development, and application of small power generation systems. Small gas turbines are appropriate alternatives to internal combustion engines due to their long life time and robustness. However, their efficiency and specific power is mostly lower than those of the large scale systems. Hence, innovations are required to considerably enhance their performance.

At present, there are two major methods to enhance the performance of a small gas turbine: improving compressor and turbine efficiencies, or improving the thermodynamic process of the cycle by increasing turbine inlet temperatures. The aerodynamics of turbomachinery has already reached a very high level of component efficiencies up to around 90% [1]. Still improvement is possible, but further huge enhancements seem to be unlikely. From a thermodynamic point of view, increasing the turbine inlet temperature is the most efficient way to improve both the overall thermal efficiency and specific power. However, the maximum temperature of the gas entering the turbine is fixed by material considerations. Thus, a considerable jump in performance of small gas turbines can only be achieved by applying advanced thermodynamic processes that are not subjected to this limitation. Topping a gas turbine with a wave rotor is an appropriate solution, since the turbine inlet temperature may stay the same while the combustion takes place at a higher average temperature. Also a pressure gain additional to that provided by the compressor is obtained by the wave rotor. Therefore, the performance enhancement is achieved by increasing mostly both the overall thermal efficiency and specific power, hence reducing the specific fuel consumption considerably. This occurs to be especially effective in the range of smaller gas turbines often used for distributed power generation or propulsion of small vehicles.

The idea of using a wave-rotor topping cycle has been first proposed by Claude Seippel of Brown Boveri Company (BBC)
in Switzerland in 1942 [2, 3, 4]. Now BBC is Asea Brown Boveri (ABB) and its pressure wave supercharger termed as the Comprex® has been used commercially for passenger car and heavy diesel engines [5, 6, 7].

Since the early 1960s the General Electric Company (GE), General Power Corporation (GPC), and Rolls Royce were involved in the development of a prototype wave rotor for propulsion applications [8, 9]. Mathematical Science Northwest also studied various aspects of wave energy exchange and proposed a wave rotor design for aircraft turbofan engines. In 1974, GPC obtained its first contract from Ford Motor Company to develop a 175 hp wave engine for an automobile like the Torino [4]. In the 1980s different U.S. agencies like DARPA (Defense Advanced Research Program Projects Agency) and the U.S. Navy expressed interest and sponsored programs to develop an understanding of wave rotor science and technology. Many developments were presented in the 1985 ONR/NAVAIR Wave Rotor Research and Technology Workshop [9].

Interest in wave rotor technology has again increased recently. Since the 1990s, a large research program at NASA Glenn Research Center (GRC) collaborated by the U.S. Army Research Laboratory (ARL) and Rolls-Royce Allison has initiated to develop and demonstrate the benefits of the wave rotor technology, which will be useful in future aircraft propulsion designs [10-17]. An excellent overview is given by Welch [18].

Using a thermodynamic approach to calculate the thermal efficiency and specific power, in 1993 Wilson and Paxson [19] performed a feasibility study for topping jet engines with a wave rotor. Applied to the case of an aircraft flying at Mach 0.8, they showed that a wave-rotor topped engine may gain 1...2% in efficiency and 10...16% in specific power compared to a simple jet engine with the same overall pressure ratio and turbine inlet temperature. Paxson also developed a one-dimensional design model to calculate off-design wave rotor performance [20] and verified it using experimental data. The model solves the unsteady viscous flow field in an axial passage for time-constant inlet and outlet port conditions, while accounting for losses due to gradual passage opening and closing, viscous and heat transfer effects, leakage, and non-uniform port flow field mixing. Recent improvement and validations have completed it as a preliminary and general design tool. However, simpler computational tools would be beneficial for wide industrial use.

Using this model in 1995 Welch et al. [21] predicted in performance calculations for small (300 to 500 kW) and intermediate (2000 to 3000 kW) wave-rotor-enhanced turboshaft engines a 19...21% increase in specific power and a 16...17% decrease in specific fuel consumption compared with the baseline engines. Same calculations for a wave-rotor-enhanced large turbofan engine, equal in thrust to the baseline engine, show a 6...7% reduction in trust specific fuel consumption. Welch also established one-dimensional and a two-dimensional analysis models to estimate the performance enhancements of wave rotors [14, 15, 22, 23, 24].

In 1996, Snyder and Fish evaluated the Rolls-Royce Allison 250 turboshaft engine as a potential platform for a wave rotor demonstration, predicting a 18...20% increase in specific power and a 15...22% decrease in specific fuel consumption [18, 25, 26]. They used a detailed map of the wave rotor cycle performance accomplished by Wilson and Paxson [17, 19].

After numerical modeling [27, 28] in 1999, Fatisis and Ribaud from the French National Aerospace Research Establishment (ONERA) performed a basic investigation of the thermodynamic performance enhancement for various types of gas turbines topped with wave rotors, including auxiliary power units, turboshaft, turbojet, turbofan [29]. The variation of wave-rotor compression and expansion efficiency, as well as mixing and pressure losses in the ducting, are taken into account. The results show the largest gains in power and largest reduction of specific fuel consumption for engines with a low compressor pressure ratio and high turbine inlet temperature, such as turboshaft engines and auxiliary power units. These results are consistent with those obtained by Jones and Welch for the wave-rotor topping of turboshaft, high-bypass turbofan, auxiliary power, and ground based power engines [30].

The objective of the present work is a comprehensive and systematic performance analysis of two actual small gas turbines topped by a four-port wave rotor in various ways. While the performance evaluation of several gas turbine engines has been studied extensively [9, 18], to the knowledge of the authors, there exits no comprehensive work investigating potential benefits of various implementation cases of wave rotor topping cycles for small gas turbines. In contrast to using existing performance computational tools, the presented results were obtained using basic thermodynamic equations along with the wave-rotor characteristic equation [19]. The model can be employed to predict the performance improvement of various wave-rotor topping cycles without the need of knowing the details of the complex fluid mechanics within the wave rotor. The challenges and advantages associated with the different implementation cases are discussed.

WAVE ROTOR DESCRIPTION

In a conventional arrangement, a wave rotor is embedded between the compressor and turbine “parallel” to the combustion chamber. Figure 1 illustrates how a four-port wave rotor is used to top a gas turbine cycle. Detailed descriptions of such a wave rotor are provided in the references cited above.
the wave rotors utilize a high-pressure fluid to transfer its energy directly to a low-pressure fluid. After the additional compression of the air in the wave rotor, it is discharged into the combustion chamber. Then the pre-expanded burned gas is scavenged toward the turbine and the channels are re-connected to the compressor outlet, allowing fresh pre-compressed air to flow into the wave rotor channels. The pre-expanded gas entering the turbine from the wave rotor can have the same temperature as the gas would have in a conventional arrangement without the wave rotor. However, the gas pressure is higher than the compressor exit pressure by the pressure gain obtained in the wave rotor. This is in contrast to the untopped engine, where - by the pressure loss occurring in the combustion chamber - the turbine inlet pressure is lower than the compressor discharge pressure.

The general advantage of using a wave rotor becomes apparent when comparing the thermodynamic cycles of baseline and wave-rotor-enhanced engines. Figure 2 shows a schematic temperature-entropy diagram of a baseline and a corresponding wave-rotor-topped engine. The shown case is the one most commonly discussed in references and in this paper is referred to as Case 1. It is evident that both gas turbines are operating with the same turbine inlet temperature and compressor pressure ratio. All wave rotors considered in this report each have zero shaft work. Therefore, the wave rotor compression work is equal to the wave rotor expansion work. Thus the temperature increase from point “1b” to “4b” in the baseline engine and from point “2” to “3” in the wave-rotor-topped engine is the same. This leads to the same heat addition for both engines. However, the specific shaft work of the topped engine is higher than that of the baseline engine due to the pressure gain across the wave rotor \( p_i > p_{ib} \). Therefore, the overall thermal efficiency for the topped engine is higher than that of the baseline engine. The inherent gas dynamic design of the wave rotor compensates for the combustor pressure loss from point “2” to “3”, so the compressed air leaving the wave rotor is at higher pressure than the hot gas entering the wave rotor at point “3” [31].

THERMODYNAMIC CALCULATION

To evaluate the performance enhancement of topping small gas turbines with wave rotors, a thermodynamic approach is used calculating the theoretic performance (expressed by specific cycle work \( w_{net} \) and overall thermal efficiency \( \eta \)) of wave-rotor-topped and baseline engines. The methodology is similar to the one introduced by Wilson and Paxson [19] with some modifications.

There are several wave rotor topping cycles that may be suited for a gas turbine application. Knowing about possible design restrictions and preferences, when an existing gas turbine is to be enhanced, mainly five different advantageous implementation cases for a wave rotor into given baseline engines can be introduced as the following:

- **Case 1**: same compressor, same turbine inlet temperature
- **Case 2**: same overall pressure ratio, same turbine inlet temperature
- **Case 3**: same combustor
- **Case 4**: same turbine
- **Case 5**: same compressor, same combustion end temperature

Figure 3 visualizes all five cases in a schematic T-s diagram. Path 0-1b-4b-5b represents the baseline cycle and 0-1-2-3-4-5 indicates the wave-rotor-topped cycles. For the five wave-rotor-enhanced cycles, indices are used to indicate the Case number. One of the five cases might be preferable for the a practical design. However intermediate design cases are possible.
For each engine, it is assumed that the compressor inlet condition is known and is the same for both the baseline engine and the wave-rotor-enhanced engine. Considering the same “aerodynamic quality” of the wheels, the polytropic efficiencies are kept the same for the enhanced and baseline engine, for the compressor and turbine respectively. Incomplete combustion of the fuel is reflected by a combustor efficiency of 98%. Also a 2% pressure drop in the combustion chamber is considered. The fuel mass addition is ignored in the calculation. No pressure losses in intake air filter, exhaust silencer and additional piping, or heat losses or mechanical losses are taken into account. Such loses will reduce the predicted performance. The gases are treated as ideal gases with constant values for specific heat coefficients ($C_{p_{\text{air}}}=1,005\ \text{kJ/kgK}$, $C_{p_{\text{gas}}}=1,148\ \text{kJ/kgK}$) and the ratio of specific heats ($\gamma_{\text{air}}=1.4$, $\gamma_{\text{gas}}=1.33$). Air is entering the compressor at 101.3 kPa and 300 K. According to previous wave rotor investigations [19, 24, 29], the wave rotor compression and expansion efficiencies are assumed with $\eta_{W_{\text{c}}}=\eta_{W_{\text{e}}}=0.83$ and a wave rotor compression ratio of $PR_{W_{\text{c}}}=p_2/p_1=1.8$ appears to be conceivable for the envisioned application and is chosen for the discussion of representative values. In this work all performance plots are shown for various wave rotor pressure ratios indicating its effect on the performance enhancement.

In the following it is assessed how the wave-rotor-topping enhances the performance of two simple cycle baseline gas turbine engines (C-30 and C-60 in Table 1), each with single stage radial compressor and turbine.

**IMPLEMENTATION CASES**

**Case 1:** In Case 1 the pressure ratio of the compressor is kept unchanged, so the physical compressor of the baseline engine can also be used for the wave-rotor-enhanced engine, provided the mass flow is kept approximately the same. The pressure in the combustion chamber of the enhanced engine is increased by the compression ratio of the wave rotor. This may require modifications to the structure of the combustion chamber and to the fuel injection system. The heat addition in the combustor is the same as for the baseline engine, but it takes place after the energy exchange in the wave rotor, hence starting at higher temperature. Thus the combustion end temperature is higher than that of the baseline engine, possibly requiring additionally a thermal enhancement of the combustor structure. The turbine of the topped engine might need to be adapted to efficiently utilize the obtained higher pressure ratio. The turbine inlet temperature however is the same as it is for the baseline engine. This implementation case gives the highest performance increase for both baseline engines.

**Case 2:** In Case 2 the overall pressure ratio for the wave-rotor-enhanced engine is kept equal to that of the baseline engine, so that the combustor works under the same pressure. However, for the wave-rotor-topped engine, the heat addition in the combustor and the combustion end temperature are higher than those of the baseline engine. This may require some adaptation of the combustor, especially in the outlet region. The turbine and compressor work with a lower pressure ratio. Thus both may be adapted to benefit most. This might reduce the cost of the compressor and turbine due to reduction of stages in case multi-stage (mostly axial) types are used or due to reduction of the tip diameter in case radial (mostly single-stage) types are used. With a smaller tip diameter the wheels can be manufactured more economically in shorter time from cheaper materials with less strength and on smaller machines. Beside an attractive performance enhancement, this case additionally provides the highest turbine outlet temperature of all five cases investigated. The temperature of the leaving exhaust gas is much higher than that of the baseline engine. Therefore, this case is especially attractive for an external heat recovery application or for internal recuperation enhancing the performance even more.

**Case 3:** Case 3 assumes that it is desirable for the wave-rotor-enhanced engine to use the unmodified combustor of the baseline engine. So the overall pressure ratio for the wave-rotor-enhanced engine is kept equal to that of the baseline engine, as is the combustor inlet and outlet temperature. The heat addition in the combustor is consequently the same. The implementation of the wave rotor considerably reduces the pressure ratio of the turbine and compressor. The compressor pressure ratio is as low as in Case 2, and the turbine pressure ratio is even lower than in Case 2, as is the turbine inlet temperature. Thus, as discussed in Case 2, the turbine and compressor could be less expensive. Unfortunately the performance enhancement comes out to be nearly negligible for the smaller engine C-30 and negative for the C-60 engine.

**Case 4:** Case 4 employs the same physical turbine as the baseline engine instead of the same compressor in Case 1. The pressure in the combustion chamber is higher than that for the baseline engine but lower than it would be in Case 1. So less effort might be required to adapt the structure and fuel injection of the combustion chamber. Due to the wave-rotor-topping, the

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The wave rotor compression efficiency is greater than the compressor efficiency. Therefore the combustor inlet temperature is in fact negligibly less and hence the heat addition is negligibly more than that in the baseline engine. See Table 2 for values.

### Table 1: Baseline engine data, assuming $p_0=1\ \text{atm}$, $T_0=300\ \text{K}$, $C_{p_{\text{air}}}=1,005\ \text{kJ/kgK}$, $C_{p_{\text{gas}}}=1,148\ \text{kJ/kgK}$, $\gamma_{\text{air}}=1.4$, $\gamma_{\text{gas}}=1.33$

<table>
<thead>
<tr>
<th>Turbine inlet temperature</th>
<th>Baseline</th>
<th>C-30</th>
<th>C-60</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{ib}$</td>
<td>1116.5 K</td>
<td>1116.5 K</td>
<td>1227.6 K</td>
</tr>
<tr>
<td>($1550\ \text{°F}$)</td>
<td>380° F</td>
<td>380° F</td>
<td>450° F</td>
</tr>
<tr>
<td>Compressor outlet temperature</td>
<td>466.5 K</td>
<td>505.4 K</td>
<td></td>
</tr>
<tr>
<td>($390\ \text{°F}$)</td>
<td>450° F</td>
<td>505° F</td>
<td>576° F</td>
</tr>
<tr>
<td>Compressor pressure ratio</td>
<td>$P_{1b}/p_0$</td>
<td>3.6</td>
<td>4.8</td>
</tr>
<tr>
<td>Compressor isentropic efficiency $\eta_{C}$</td>
<td>79.6%</td>
<td>82.6%</td>
<td></td>
</tr>
<tr>
<td>Turbine isentropic efficiency $\eta_{T}$</td>
<td>84%</td>
<td>85%</td>
<td></td>
</tr>
<tr>
<td>Compressor polytropic efficiency $\eta_{pC}$</td>
<td>82.9%</td>
<td>85.9%</td>
<td></td>
</tr>
<tr>
<td>Turbine polytropic efficiency $\eta_{pT}$</td>
<td>81.7%</td>
<td>82.3%</td>
<td></td>
</tr>
</tbody>
</table>
compressor needs to produce less pressure ratio than that in the baseline engine. This allows for a less expensive compressor as discussed for Case 2 and 3. Resulting from the lower pressure ratio in the compressor, hence lower compressor discharge temperature, the heat addition in the combustor has to be more than that for the baseline engine to utilize the same allowed turbine inlet temperature. This case gives the second highest performance increase for both baseline engines.

**Case 5:** Case 5 is similar to Case 1 but the combustion end temperature - the maximum cycle temperature - is restricted to the turbine inlet temperature of the baseline engine, in order to not impose additional thermal requirements for the combustor. The overall pressure ratio is the same as in Case 1. It is greater than that of the baseline engine, by the wave rotor pressure ratio. The heat addition in the combustor is less than that for the baseline engine, due to the wave-rotor compression work added to the fluid before the combustion. The compressor is the same physical compressor as for the baseline engine. The turbine in the topped cycle works with a slightly greater pressure ratio than that of the turbine of the baseline engine, but the turbine inlet temperature is less than it is for the baseline engine. In fact, it is the lowest of all cases investigated. This may give the option to produce the turbine wheel at lower costs out of less thermal resistant material.

**PREDICTED PERFORMANCE RESULTS**

**Case 1:** Case 1 is the most common case discussed in the literature and it gives mostly the best performance enhancement. Therefore it is discussed here in more detail. Figure 4 shows the actual temperature-entropy diagrams for the baseline engines C-30 and C-60 as well as for the topped engines, simulated with a wave-rotor pressure ratio of 1.8. The overall pressure ratio of the enhanced engines is \(1.8 \times 3.6 = 6.48\) and \(1.8 \times 4.8 = 8.64\) respectively. The temperature-entropy diagram qualitatively shows that the topped engine has a higher efficiency compared to the baseline engine. This is because the turbine has higher specific work, while consumption of specific work by the compressor and specific heat addition to the cycle are the same as for the baseline engine.

Figure 5 illustrates the increase of cycle overall thermal

![Figure 4: Temperature-Entropy diagram for wave-rotor implementation Case 1](image)

![Figure 5: Overall thermal efficiency and specific work of the wave-rotor-topped engines versus wave rotor pressure ratio and overall pressure ratio for Case 1](image)
efficiency (green) and specific work (blue) with higher values of the wave rotor pressure ratio $PR_W = p_2/p_1$. The plot visualizes how the effect develops from the baseline engine with $PR_W = 1$ until $PR_W = 2$. The upper value is imagined as a practical limit for the investigated application. However, if the wave rotor pressure ratio increases beyond this limit the trend already shows that the increasing effect becomes less, while technical problems may increase.

With a conceivable wave rotor pressure ratio of 1.8, the overall thermal efficiency of the baseline cycle increases from 14.7% to 19.6% for the C-30 engine and from 19.1% to 23.9% for the C-60 engine. Simultaneously the specific work will increase from 122 kJ/kg to 163 kJ/kg for the C-30 engine and from 176 kJ/kg to 219 kJ/kg for the C-60 engine. This means an attractive relative performance improvement in overall thermal efficiency and specific work (red in Fig. 5) of about 33.6% for the C-30 engine and 25% for the C-60 engine.

**Case 2**: In Case 1, the same compressor was used for both the baseline and the wave-rotor-topped engine. Thus, the overall compression ratio of the wave-rotor combination was higher than that of the baseline engine, perhaps causing technical problems with the combustor. Now Case 2 considers another way to implement a wave rotor beneficially. While keeping the overall pressure ratio of the topped engine equal to that of the baseline engine, the compressor pressure ratio is reduced in the wave-rotor-enhanced engine. Lowering the compressor pressure ratio usually leads to higher isentropic compressor efficiency (for comparable aerodynamic impeller quality, here simulated by the same polytropic compressor efficiency), less mass of the compressor, and probably lower manufacturing costs.

Figure 6 depicts the actual temperature-entropy diagram of the baseline engines and the wave-rotor-topped engines for Case 2. The overall pressure ratio is fixed at 3.6 and 4.8 for the C-30 and C-60 engine, respectively. It is evident that the compressor work of the topped engine is less than that of the baseline engine. However, the turbine work is less too, but the heat addition for the topped engine is greater than that of the overall compression ratio of the wave-rotor combination was higher than that of the baseline engine, perhaps causing technical problems with the combustor. Now Case 2 considers another way to implement a wave rotor beneficially. While keeping the overall pressure ratio of the topped engine equal to that of the baseline engine, the compressor pressure ratio is reduced in the wave-rotor-enhanced engine. Lowering the compressor pressure ratio usually leads to higher isentropic compressor efficiency (for comparable aerodynamic impeller quality, here simulated by the same polytropic compressor efficiency), less mass of the compressor, and probably lower manufacturing costs.

Figure 7: Overall thermal efficiency and specific work of the wave-rotor-topped engines versus wave rotor pressure ratio and overall pressure ratio for Case 2.
baseline engine. So, without calculating the overall thermal efficiency and specific work it is problematical to determine whether the topped engine has a higher performance than the baseline engine. Clearly shown in Fig. 6 is that the turbine outlet temperature is considerably higher than that of the baseline engine. This makes this case especially attractive for internal recuperation or an external heat recovery application because additional heat at higher temperature is available from the exhaust gas.

Values of the overall thermal efficiency and specific work as well as their relative increases can be obtained from the plots in Fig. 7. Now, the relative increase of overall thermal efficiency (red in Fig. 7) is considerably less than the relative increase of specific work (cyan in Fig. 7), because the heat addition is greater in the wave-rotor-topped cycle.

Here, for the Case 2, the plots show that with a wave rotor pressure ratio of 1.8:
- the compressor pressure ratio reduces from 3.6 to 2 for the C-30 engine and from 4.8 to 2.67 for the C-60 engine.
- the overall thermal efficiency increases from 14.7% to 15.6% for the C-30 engine and from 19.1% to 19.6% for the C-60 engine.
- the specific work increases from 122 kJ/kg to 143 J/kg for the C-30 engine and from 176 J/kg to 200 J/kg for the C-60 engine.
- hence the overall thermal efficiency increases relatively by about 6.1% for the C-30 engine and 3.1% for the C-60 engine.
- and finally the specific work increases relatively by about 17.4% for the C-30 engine and 13.1% for the C-60 engine.

Table 2: Performance comparison between baseline engines and five cases of wave-rotor-topping with a wave rotor pressure ratio of 1.8

<table>
<thead>
<tr>
<th>Same as baseline engine</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
<th>Case 5</th>
<th>Baseline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engines</td>
<td>p₁, T₁, T₄</td>
<td>p₂ = p₉, T₄</td>
<td>p₂ = p₉, T₂, T₃</td>
<td>p₃ = p₂₉, p₄, T₄</td>
<td>p₁, T₁, T₃ = T₂b</td>
<td></td>
</tr>
<tr>
<td>T₄ [K]</td>
<td>C-30</td>
<td>C-60</td>
<td>C-30</td>
<td>C-60</td>
<td>C-30</td>
<td>C-60</td>
</tr>
<tr>
<td>q [kJ/kg]</td>
<td>829</td>
<td>920</td>
<td>1116</td>
<td>1228</td>
<td>1116</td>
<td>1228</td>
</tr>
<tr>
<td>Πc</td>
<td>3.60</td>
<td>4.80</td>
<td>2.00</td>
<td>2.67</td>
<td>2.00</td>
<td>2.67</td>
</tr>
<tr>
<td>Πr</td>
<td>4.34</td>
<td>5.82</td>
<td>2.58</td>
<td>3.45</td>
<td>2.53</td>
<td>3.38</td>
</tr>
<tr>
<td>w_c [kJ/kg]</td>
<td>167</td>
<td>206</td>
<td>81</td>
<td>116</td>
<td>81</td>
<td>116</td>
</tr>
<tr>
<td>w_f [kJ/kg]</td>
<td>330</td>
<td>426</td>
<td>224</td>
<td>315</td>
<td>205</td>
<td>290</td>
</tr>
<tr>
<td>w_{net} [kJ/kg]</td>
<td>163</td>
<td>219</td>
<td>143</td>
<td>199</td>
<td>124</td>
<td>174</td>
</tr>
<tr>
<td>η</td>
<td>0.196</td>
<td>0.239</td>
<td>0.156</td>
<td>0.196</td>
<td>0.149</td>
<td>0.189</td>
</tr>
<tr>
<td>(w_{net}) gain [%]</td>
<td>33.6</td>
<td>24.9</td>
<td>17.3</td>
<td>13.1</td>
<td>1.7</td>
<td>-1.1</td>
</tr>
<tr>
<td>(η) gain [%]</td>
<td>33.6</td>
<td>24.9</td>
<td>6.2</td>
<td>2.8</td>
<td>1.5</td>
<td>-0.9</td>
</tr>
</tbody>
</table>
All five wave-rotor topping cases: The more detailed documentation of Cases 3, 4 and 5 are not presented here. Instead, now follows a compilation of the predicted performance enhancement of all five investigated cases with a wave rotor pressure ratio of 1.8. Figure 8 shows the corresponding actual temperature-entropy diagrams for all five wave-rotor-topped cases and the baseline engines. Numerical values of relevant cycle data and the performance enhancement are summarized in Table 2.

In Table 2, \( T_4 \) is the turbine inlet temperature, \( q \) the specific heat addition, \( \Pi_C \) the compressor pressure ratio, \( \Pi_T \) the turbine pressure ratio, \( w_C \) the specific work consumed by the compressor, \( w_T \) the specific work provided by the turbine, \( w_{net}=w_T - w_C \) the specific work produced by the cycle, \( \eta \) the overall cycle efficiency, \( (\omega_{net})_{gain} \) the relative increase of the cycle specific work and \( (\eta)_{gain} \) the relative increase of the overall cycle efficiency.

Figure 9 shows maps of the relevant design space, which allow predicting the performance of wave-rotor-enhanced engines in terms of overall efficiency (green) and specific work (blue) for any combination of compressor pressure ratio (absissa) and wave rotor pressure ratio \( PR_W \) (parameter labeled in black). In these maps, the multiplication of compressor pressure ratio \( p_1/p_0 \) and wave rotor pressure ratio \( PR_W \) determines the overall cycle pressure ratio \( p_2/p_0 \) (red). The optimum compressor pressure ratio points for highest overall efficiency and specific work at each achievable wave rotor pressure ratio are connected by black solid lines.

These performance maps are general. The only specific parameters are indicated in the upper left corner of the map. They are turbine inlet temperature and polytropic efficiency of compressor and turbine, which correspond to the respective baseline engine. Such maps are not only very useful to explore the possible enhancement of already existing baseline engines, but they also serve very well for selecting a design point or region when designing a new wave-rotor-topped engine.

In both maps the performance points of the baseline engine and the wave-rotor-enhanced engines of Case 1, 2 and 4 with a wave rotor pressure ratio of \( PR_W=1.8 \) are indicated, separately for overall thermal efficiency and specific work. (In the map for the C-60 baseline engine, accidentally for Case 1 and 2 the performance points for specific work and efficiency fall together, due to the chosen scale of the ordinate axes) Starting from the performance point of the baseline engine, the performance values for Case 1 are found by moving vertically upwards (along a line for constant compressor pressure ratio \( p_1/p_0 \)) until the corresponding performance curve of the expected wave rotor pressure ratio is crossed. Case 2 is found by moving along a line for constant overall pressure ratio \( p_2/p_0 \). Case 4 actually lies on a constant turbine pressure ratio line (not shown). Case 3 and 5 can not be shown in the same map, since in both cases the turbine inlet temperature is less than indicated in the upper left corner of the map.

The results indicate that for every compressor pressure ratio in the shown design space, the performance of the topped engine is always higher than that of the corresponding baseline engine with the same compressor pressure ratio (Case 1 consideration). However, for higher compressor pressure ratios, less benefit can be obtained by using a wave rotor. In other words, the benefit is the greatest for lower compressor pressure
This clearly favors the wave-rotor-topping for small gas turbines with low compressor pressure ratios. This trend becomes even more obvious if for the same compressor pressure ratio, the differences between the values of the baseline engine and the enhanced engine are viewed. To illustrate this, Fig. 10 and 11 show the absolute gain and relative increase of the overall thermal efficiency and specific work when adding a wave-rotor to an existing compressor (Case 1 consideration).

Moreover, as expected and known for simple cycle engines ($PR_w=1$), for the wave-rotor-topped engines it is true too that the compressor pressure ratio for the maximum specific work is always less than for the maximum overall cycle efficiency. However, with increasing wave rotor pressure ratio, both

![Figure 10: Absolute gain in overall thermal efficiency and specific work versus compressor pressure ratio of the baseline engine with the same compressor pressure ratio (Case 1 consideration)](image1)

![Figure 11: Relative increase of overall thermal efficiency and specific work versus compressor pressure ratio of the baseline engine with the same compressor pressure ratio (Case 1 consideration)](image2)
optima come closer together, while moving towards lower compressor pressure ratio. This can be viewed as an additional advantage for applying wave rotors to small gas turbines with low compressor pressure ratios.

So as Fig. 9 shows, adding a wave-rotor with a 1.8 pressure ratio to the C-30 or C-60 baseline engines with a compressor pressure ratio \( \frac{p_1}{p_0} = 3.6 \) or \( \frac{p_1}{p_0} = 4.8 \) respectively, already brings the design point in the optimum range for highest specific work and nearly half way closer to the optimum for highest overall thermal efficiency.

CONCLUSION

For the implementation of a wave-rotor in the two simple cycle engines C-30 and C-60, the present investigation predicts an attractive performance enhancement for both considered baseline engines. The smaller C-30 engine would benefit even more from the wave-rotor-topping than the C-60 engine. The C-30 engine overall efficiency and specific work may increase more from the wave-rotor-topping than the C-60 engine. The smaller C-30 engine would benefit even more. Ideally only one component of the three main components, namely, compressor, combustor, and turbine, can remain unchanged when a wave-rotor is implemented. However, compromises may be considered for tests, prototypes, or cost effective development, keeping two or even all three main components unchanged.

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