Gas turbine performance enhancement for naval ship propulsion using wave rotors

Antonios Fatsis

General Department, Kapodistrian University of Athens, Psachna, Evoias, Greece

ABSTRACT
The propulsion demands of high-speed naval vessels often rely on gas turbines, producing significant amount of power achieving thus high performance requirements. Gas turbines can be used either to provide purely mechanical propulsion, or alternatively to generate electricity, which is subsequently used by electric drives to propel the ship. However, the thermal efficiencies of gas turbines are lower than those of Diesel engines of similar power. In the context of improving the performance of existing marine gas turbines, it is proposed to enhance engine’s performance by integrating a pressure wave supercharger (or wave rotor), while keeping the basic components of the baseline engine unchanged. This article is an application of a method developed by the author for marine gas turbines with and without recuperator, [Fatsis 2019]. It is found that important benefits are obtained for the topped engines in comparison to the self-standing baseline engines. The wave rotor integration is more appropriate for engines with low compression ratios and high turbine inlet temperatures. Application of wave rotor technology to recuperated marine gas turbines shows efficiency gain exceeding 19% for all the compression ratio and turbine inlet temperatures examined.

Abbreviations: COGAS, Combined gas turbine and steam turbine; COGES, Combined gas turbine electric and steam; GTE, Gas turbine engines

Nomenclature

| Symbol | Definition |
|--------|------------|
| FA     | Fuel to air ratio |
| FCV    | Fuel calorific value, MJ/kg |
| Cpa    | Specific heat of air, J/kg K |
| Cpg    | Specific heat of exhaust gases, J/kg K |
| m      | Mass flow rate, kg/s |
| n      | Efficiency |
| P      | Pressure, Pa |
| PR     | Wave rotor pressure ratio |
| rc     | Compressor pressure ratio |
| sfc    | Specific fuel consumption, kg/J |
| T      | Temperature, K |
| TIT    | Turbine inlet temperature, K |
| W      | Power, W |
| ws     | Specific power, MJ/kg |

| Symbol | Definition |
|--------|------------|
| 7      | Compressor turbine exit |
| 8      | Power turbine exit |
| 9      | Exhaust |
| α      | Ambient |
| C      | Compression processes inside the wave rotor |
| E      | Expansion processes inside the wave rotor |
| f      | Fuel |
| g      | Exhaust gases |
| cc     | Combustion chamber |
| isc    | Isentropic compressor |
| ist    | Isentropic turbine |
| HE     | Heat exchanger |

Greek Symbols

| Symbol | Definition |
|--------|------------|
| γ      | Ideal gas constant |
| ΔP     | Pressure losses, Pa |

Subscripts

| Subscript | Definition |
|-----------|------------|
| 0         | Stagnation conditions |
| 1         | Intake |
| 2         | Compressor inlet |
| 3         | Compressor exit |
| 4         | Wave rotor inlet |
| 5         | Mixer |
| 6         | Compressor turbine inlet |

Introduction
During the last 40 years most Navies have begun to utilise aero-derivative gas turbine engines as prime movers for surface combatants due to enhanced performances that could not be attained with diesel engines, Brady (1988). Although the naval community had attempted the use of this type of engines for ship propulsion earlier, it was only after a successful commercial campaign during the 70s that gas turbines were used for naval propulsion systems. Development in gas turbine technology during the last years made possible reduced manning, maintenance and weight, short warm-up times, ease of control, low NOX and negligible SOX emissions due to higher grades of fuel and therefore reduced cost, Kayadelen and Üst (2013). Simple gas turbine cycles attain levels of thermal efficiency of approximately 37%. When compared to diesel engines, gas turbines require less space for the main engine and the auxiliaries; create reduced vibrations and have faster reaction to variations of
The main disadvantages of gas turbines are related to high fuel consumption which combined to the high price of the fuel for aero-derivative gas turbines, makes their operation costly. Additionally, the efficiency of gas turbines drops as the ambient temperature rises, and thermal efficiencies of gas turbines are lower than those of diesel engines of similar power. Therefore several methods have been adopted to improve the efficiency of gas turbines for naval use (RAENG 2013). The most popular of those applied in naval gas turbine engines (GTEs) is the ICR (Intercooled Recuperated) cycle where a recuperator (or regenerator) is added after the low-pressure compressor, prior to the combustion chamber of the intercooled cycle to recover waste exhaust heat and heat up the compressed air.

A feasibility study of intercooling and regenerating the marine RR-Spey gas turbine engine was conducted by Thomson et al. (1987). It was demonstrated that the RR-Spey SM1A engine was modified by adding intercooler and regenerator attaining a gain in fuel consumption of more than 20%. The ICR engine based on the RB211 aero-engine is projected to save up to 30% annual fuel savings according to Cristall and Parker (1993). Shepard et al. (1994) presented in a comprehensive way the requirements, technical specifications and project implementation details. English (2003) concluded that the WR-21 engine offers significant benefits in terms of efficiency, reliability, reduced maintenance cost, as well as improved naval characteristics for existing and new generation Frigates and Destroyers. Part-load performance of two-shaft gas turbines can be improved by introducing variable guide vanes (VGV) as suggested by Haglind (2010).

An alternative to overcome the part-load performance is to install and configure the gas turbine to operate in combination with other propulsion prime movers such as the diesel engine either as ‘Combined Diesel and Gas turbine’ (CODAG) or as ‘Combined Diesel or Gas Turbine’ (CODOG) configuration in which boost power for top speed operations is to be provided by the bigger engine, while low-speed fuel economy at low power settings is to be provided by the smaller one whenever necessary as Parker et al. (1998) recommend.

Gas turbine emissions satisfy the MARPOL Tier III requirements (IMO 2008). NOx and SOx emissions of gas turbine engines are less than Internal Combustion Engines equipped with SCR with urea-injection and closed-loop scrubbers, offering a significant reduction in NOx emissions, easily matching nitrogen regulations for current marine engines, (Barsi et al. 2019).

Haglind (2008a, 2008b) underlines that the aim of combining different propulsion configurations for large ships is the optimisation of system design in order to minimise fuel consumption and maximise operating flexibility and reliability. The ‘Combined Gas and Steam Turbine’ (COGAS) where the shafts of the turbines drive directly the propeller constitutes one type of configuration, whereas the ‘Combined Gas Turbine and Steam Turbine integrated Electric Generator’ (COGES) is another possible configuration.

Barsi et al. (2020) showed that small gas-steam combined plants can be used in various naval applications, both as a prime mover, or combined with other engines or in modular configurations for application on larger vessels.

At full load the COGAS configuration offers higher efficiency as the transmission efficiency is higher. With a COGES configuration, it is easier to achieve rational part-load performance, because one or several power units can be shut off as the power is reduced while the remaining power units are operated at their high efficient power condition, (Cristall and Parker 1993).

A novel method for efficiency improvement and power increment, originally proposed for aeronautical applications, is the integration of a pressure wave exchanger or wave rotor to marine gas turbines. A wave rotor consists of a purely cylindrical rotor inside a casing. The rotor is composed of two coaxial cylinders. Circumferentially equidistant axial straight thin blades are formed between these cylinders. Two stationary endwall plates with perforated circumferential openings are mounted at the rotor extremities, allowing only partial inflow and outflow through the rotor blade channels, as described by Weber (1995) and Povinelli et al. (2000). Depending on the number of openings (or ports), wave rotors can be classified as three-port, four-port or five-port configurations. In Figure 1, one can see the four-port wave rotor assembly, which is the configuration best suited as a gas turbine topping device, Wilson and Paxson (1993).

The rotor is connected via a ducting system to the compressor, turbine and combustion chamber of the baseline engine. Unlike conventional turbomachinery components, the principle of operation of wave rotors is based on propagation of unsteady pressure waves inside the various rotor channels. These moving pressure waves are formed inside the wave rotor channels when a high enthalpy gas stream (e.g. hot combustion gas) is coming in contact for a short time — so that mixing is avoided— with low enthalpy gas (e.g. compressed air) being inside the rotor. According to the basic theory of gas dynamics described by Weber (1995) and verified analytically and numerically by Iancu and Müller (2005), the propagation of a compression wave inside each of the rotor channels results in the formation of an expansion wave and its propagation at the opposite direction. The contact discontinuity between the high and the low enthalpy gas streams guarantees that no mixing between the two streams will occur. When a wave rotor is integrated into a gas turbine, extra compression in the air flow is achieved by means of compression waves formed inside the wave rotor channels when hot exhaust gases coming out of the combustion chamber come in contact with air from the compressor. Simultaneously expansion is achieved when expansion waves are directed at the outflow port towards the turbine. An efficient design of wave rotors is attained when the pressure waves inside the rotor result in uniform and steady flows at the outflow ports, so that the air flow towards the combustion chamber and the gas flow towards the turbine are uniform. For the case of the four-port through flow wave rotor, the rotor blades are self-cooled because hot gas and compressed air are traversing the rotor, so no extra cooling is needed. Feasibility studies of integrating a wave rotor to aircraft gas turbines showed the reduction of the specific fuel consumption and an increase of the specific thrust delivered by the engine, as stated by Jones and Welch (1996). A recent publication by Fatsis (2019) explored the possibility of integrating a four-port wave
rotor also to industrial gas turbines. The wave rotor constant volume combustor utilises unsteady pressure waves and controlled burning to provide pressure increase during combustion, unlike the conventional chambers of gas turbine engines (Elharis et al. 2010). Akbari and Nalim (2009) explore the benefits as well as the challenges of this concept. Complex phenomena such as unsteady combustion and flame–pressure wave interactions have not yet studied adequately and require further research effort to be fully understood.

This article is an original study on wave rotor technology applied to marine gas turbines used as prime mover for naval propulsion. It is applying the thermodynamics model developed by Fatsis (2019) to marine gas turbines. Performance assessment is accomplished for two-shaft marine gas turbines with and without recuperator at design point conditions. The most important parameters of the wave rotor topped engines are identified and their variations around standard values listed in the literature indicate the effect on engine’s performance. It is concluded that the integration of wave rotor in a marine gas turbine improves the thermal efficiency of the engine and at the same time increases its specific power. The improvement is more remarkable for engines operating with low compressor pressure ratios and high turbine inlet temperatures, for the range of compressor pressure ratios examined.

Gas turbine thermodynamic calculations

Input data for two-shaft gas turbines

The procedure of the thermodynamic calculations of two-shaft gas turbine cycles with the integration of a four-port wave rotor is described in detail by Fatsis (2018). It is based on standard thermodynamic analysis of gas turbines, as presented by Horlock (2003) and by Razak (2007), supplemented by modelling the processes inside the wave rotor. Figure 2 illustrates the configurations for the wave rotor-topped two-shaft gas turbines used in this article.

Two-shaft gas turbines are well-suited to marine applications because their operation covers a wide range, behaving almost as a constant power machine (Stapersma and Woud 2005). Various numerical models are available in the literature for the steady state, as well as for the transient simulation of gas turbines, as the ones by Benvenuto and Campora (2003 and 2005), using thermodynamics analysis of gas turbines for marine propulsion. A monitoring and diagnostic system based on artificial neural network for the case of a gas turbine-controllable pitch propeller plant has been developed by Campora et al. (2017) allowing the generation of a large amount of operational data, in order to overcome the lack of experimental data.

The present method takes into account the effect of temperature variation for the calculation of specific heats of air and combustion gases at various stages throughout the gas turbine cycle according to Ebaid and Al-hamdan (2015) who derived polynomial curve fitting relations using data from tables. In the equations below $T_a$ and $T_g$ are the average temperatures during the compression and expansion processes in the compressor and turbine, respectively.

For air at low-temperature range of 200–800 K

$$C_{Pa} = 1.0189 \times 10^3 - 0.13784T_a + 1.9843 \times 10^{-4}T_a^2$$
$$+ 4.2399 \times 10^{-7}T_a^3 - 3.7632 \times 10^{-10}T_a^4$$

(1)

For air at high-temperature range of 800–2200 K

$$C_{Pa} = 7.9865 \times 10^2 + 0.53397T_a - 2.2882 \times 10^{-4}T_a^2$$
$$+ 3.7421 \times 10^{-8}T_a^3$$

(2)

For specific heats of products of combustion

$$C_{Pg} = C_{Pa} + \frac{FA}{1 + FA}B_T$$

(3)

where $B_T$ at low-temperature range of 200–800 K is given by:

$$B_T = -3.59494 \times 10^5 + 4.51647T_g + 2.8116 \times 10^{-3}T_g^2 - 2.1709$$
$$\times 10^{-5}T_g^3 + 2.8689 \times 10^{-8}T_g^4 - 1.226 \times 10^{-11}T_g^5$$

(4)

and $B_T$ at high-temperature range of 800–2200 K is given by:

$$B_T = 1.0888 \times 10^3 - 0.14167T_g + 1.916 \times 10^{-3}T_g^2 - 1.2401$$
$$\times 10^{-6}T_g^3 + 3.0669 \times 10^{-10}T_g^4 - 206117 \times 10^{-14}T_g^5$$

(5)

The fuel to air ratio, $FA$ in the combustion chamber is given by:

$$FA = \frac{C_{ph}T_05 - C_{pc}T_{4.1}}{n_{CC} \cdot FCV}$$

Constant values are assumed for the air and for the exhaust gases as: $\gamma_c = 1.4$ and $\gamma_h = 1.333$.

Typical values of input data used for marine gas turbines are summarised in Table 1 according to available data by Woodyard (2004). This data does not correspond to a specific engine, but to a series of engines. This is why the compression ratio varies from 5 to 30 and the Turbine Inlet Temperature (TIT) from 1000 K to 1600 K.

Input data for wave rotor

Typical input data for wave rotor thermodynamic calculations are shown in Table 2. The values indicated are in agreement with data used by Wilson and Paxson (1993) in early studies of wave rotor-topped gas turbine engine performance evaluation, as well as with data used in more recent publications on wave rotors (Akbari and Müller 2003a; Dempsey et al. 2006).

Parameters that affect the efficiency and specific power of the topped engine chosen to be investigated are the wave rotor pressure

![Figure 2. Two-shaft gas turbine configuration, C: compressor, T: turbine, CC: combustion chamber, WR: wave rotor, CT: compressor turbine, PT: power turbine.](image-url)
The wave rotor pressure ratio is a critical parameter that characterises the performance of the wave rotor and accordingly the performance of the topped gas turbine engine. It is defined as:

\[ PR = \frac{P_{4.1}}{P_{4.0}} \]  

(1)

This parameter delivers the extra compression inside the wave rotor of the air flow stream.

Stagnation temperature at the cold air port of the wave rotor, \( T_{4.0} \)

\[ T_{4.0} = T_{04} \]  

(2)

Stagnation temperature at the port towards the turbine, \( T_{4.3} \)

\[ T_{4.3} = TIT \]  

(3)

Stagnation pressure at the cold air port of the wave rotor \( P_{4.0} \)

\[ P_{4.0} = P_{04} \left( 1 - \frac{\Delta P_{duct}}{100} \right) \]  

(4)

where the term \( \Delta P_{duct} \) includes the pressure losses at the ducts connecting the wave rotor to compressor, combustion chamber and turbine and the leakage losses occurring between the rotating part (rotor) and the stationary openings (ports) of the wave rotor.

Stagnation temperature at the wave rotor exit towards the combustion chamber, \( T_{4.1} \), is obtained by using isentropic efficiency of the compression process inside the wave rotor.

\[ T_{4.1} = T_{4.0} \cdot \left( \frac{PR^{(\eta_{C}^{-1})/\gamma_{C}} - 1}{\eta_{C}} + 1 \right) \]  

(5)

where \( \eta_{C} \) is the compression efficiency inside the wave rotor.

Stagnation pressure at the combustion chamber outlet \( P_{4.2} \)

\[ P_{4.2} = P_{4.1} \left( 1 - \frac{\Delta P_{cc}}{100} \right) \]  

(6)

where the term \( \Delta P_{cc} \) represents the pressure losses in the combustion chamber.

Stagnation temperature at the combustion chamber exit, \( T_{4.2} \), is obtained by using isentropic efficiency of the expansion process inside the wave rotor.

\[ T_{4.2} = \frac{T_{4.3}}{1 - \left( \frac{P_{4.3}}{P_{4.2}} \right)^{(\gamma_{E}^{-1})/\gamma_{E}}} \cdot n_{E} \]  

(7)

where \( n_{E} \) is the expansion efficiency inside the wave rotor.

The pressure at the wave rotor outflow towards the turbine is given according to Wilson and Paxson (1993) by:

\[ P_{4.3} = P_{4.0} \cdot PR \left\{ \frac{1 - \frac{C_{pa}}{C_{ph} \eta_{E} TIT}}{1 + \frac{C_{pa}}{C_{ph} \eta_{E} TIT}} \cdot \left( T_{04} \left( PR^{\frac{\gamma_{C} - 1}{\gamma_{C}}} - 1 \right) \right)^{\frac{1}{\gamma_{C}}} \right\}^{\frac{1}{\gamma_{C} - 1}} \]  

The above relation is known in the literature (Wilson and Paxson 1993) as the characteristic of the wave rotor. Akbari and Müller (2003b) have also used the above equation for a preliminary design of wave rotor.
Performance

Power consumed by the compressor, \( W_C \)

\[
W_C = \dot{m}_{\text{air}} \cdot C_{pc} \cdot (T_{03} - T_{02}) \tag{8}
\]

Power produced by the turbine \( W_T \) is used to drive the compressor

\[
W_T = W_C \tag{9}
\]

Heat added by the fuel \( \dot{Q}_{in} \)

\[
\dot{Q}_{in} = m_f \cdot FCV \tag{10}
\]

Power produced by the power turbine, \( W_{PT} \)

\[
W_{PT} = (\dot{m}_{\text{air}} + m_f) \cdot C_{ph} \cdot (T_{07} - T_{08}) \tag{11}
\]

Net power delivered by the engine, \( W_N \)

\[
W_N = W_{PT} \tag{12}
\]

Specific power \( ws \)

\[
ws = \frac{W_N}{\dot{m}_{\text{air}}} \tag{13}
\]

Thermal efficiency \( \eta_{th} \)

\[
\eta_{th} = \frac{W_N}{\dot{Q}_{in}} \tag{14}
\]

Specific fuel consumption, \( sfc \)

\[
sfc = \frac{1}{\eta_{th} \cdot FCV} \tag{15}
\]

The temperature of the metal used to manufacture turbine blades goes up to approximately 800–900 °C, but the maximum surface temperature of blades has been increased up to 1000 °C due to the recent application of thermal barrier coatings on blade surfaces, Moon et al. (2018). The present thermodynamic model accounts for cooling the high pressure turbine in case \( T_{IT} \geq 1300 \text{K} \). Two possible solutions can be proposed for air cooling the high pressure turbine of the wave-rotor topped engine: If it is to be done by external cooling, it can only be achieved by subtracting air flow from the high pressure outflow port of the wave rotor having higher pressure and lower temperature than the exhaust gases directed to the turbine. This guarantees that no backflow is occurring; however the presence of exhaust gases at the outflow of port 4.1 according to Snyder (1996) due to recirculation increases the temperature of the cooling fluid which is further mixed with the high temperature exhaust gases exiting the port 4.3 of the wave rotor and directed towards the turbine nozzles. Air at the compressor exit cannot be used for cooling purposes since its pressure is lower than the pressure of the exhaust gases in port 4.3 of the wave rotor.

If the turbine is to be cooled by internal (convective) cooling, air from the compressor can be extracted and directed inside the turbine nozzles and rotor blades, absorbing thus heat from the blade (Razak 2007).

The unsteady flow nature created by pressure wave propagation in the blade channels, does not allow mixing of air with exhaust gases inside the rotor, Weber (1995). Exhaust gases from the combustion chamber (after expansion in the wave rotor) are directed towards the turbine through the port 4.3, Paxson et al. (2007). Depending on the design of the baseline engine, alternative methods for the turbine cooling can be examined, however, the investigation of alternate cooling methods was not investigated as such was outside the scope of this study.

The quantity of the air mass flow rate to be used in the mixer in order to moderate the exhaust gas temperature can be calculated using an energy balance equation based on the inlet air flow rate, power output, efficiency, gas turbine outlet temperature and \( T_{IT} \) according to the method proposed by Prasad et al. (2016) and by Jonsson et al. (2005). Turbine blades are normally made of nickel or rhenium alloys capable of withstanding high heat without distortion. The High Pressure Turbine airfoils as well as the Power Turbine airfoils can also be cooled. They can be made of INCO 738 coated with a silicon aluminate coating (Shepard et al. 1994).

Wave rotor two-shaft gas turbine analysis

Figure 4 presents the performance curves of wave rotor topped two-shaft gas turbines at design point for various values of \( r_c \) and \( T_{IT} \), illustrated with continuous lines in comparison to the base line (without wave rotor) two-shaft gas turbines illustrated with dashed lines. These results correspond to \( PR = 1.8 \) and \( n_c = n_c = 0.83 \). The choice of these values is in agreement with data used by Wilson and Paxson (1993) in early studies of wave rotor-topped gas turbine engine performance evaluation, as well as with data used in the

![Figure 4. Thermal efficiency percentage against specific power for baseline and wave rotor-topped two-shaft gas turbines (\( r_c \): compressor pressure ratio).](image-url)
most recent publications on wave rotors (Akbari and Müller (2003a), Akbari et al. (2006), Dempsey et al. (2006), Fatsis (2019)) and are considered as well established values in the literature.

Dashed lines with triangular symbols illustrate the performance of the baseline engines, while continuous lines with spherical symbols of the same colour illustrate the performance of the wave rotor-topped two-shaft gas turbines. Each symbol in the diagram corresponds to the design point conditions of a specific two-shaft gas turbine. From this figure, it can be seen that for a given value of $TIT$, both thermal efficiency and specific power of wave-rotor topped engines are increasing with respect to their values of the base line engines. It is observed that the performance curves of the wave rotor-topped engines are shifted to the upper right part of the diagram. Similarly, Figure 5 presents the $sfc - ws$ distribution of wave rotor topped two-shaft gas turbines at design point conditions for various values of $r_c$ and $TIT$. One can observe that the performance curves of the wave rotor-topped engines are shifted to the lower right part of the diagram compared to the corresponding baseline gas turbines without wave rotor. Therefore the wave rotor integration, decreases baseline engine’s $sfc$ and simultaneously increases its $ws$.

Figures 4 and 5 indicate that two-shaft gas turbines having low values of the compressor pressure ratio, $r_c$, are more favourable to the integration of a four port wave rotor in terms of thermal efficiency, specific fuel consumption and specific power, especially at high values of $TIT$.

Figure 6 shows qualitatively the effect of integrating a four-port wave rotor on two-shaft engines. In this figure, a typical case with $PR = 1.8$, $n_C = n_E = 0.83$ for $TIT = 1500K$ is presented. In the same figure, the percentage increase in $ws$ and in $nth$ are illustrated.

A clear benefit in terms of both $ws$ and $nth$ increase for all values of $r_c$ considered can be seen. More specifically, the prevalent increase in $nth$ reaches 19.3% and in $ws$ reaches 18% for compressor ratio $r_c = 5$. This increase decays as $r_c$ increases, ending up to a minimum value of 4.8% and 2.6% for $nth$ and $ws$, respectively, for engines of compression ratio $r_c = 30$. This means that for all range of gas turbines there is a clear performance enhancement.

Figure 7 shows qualitatively the effect on specific fuel consumption of two-shaft engines when integrating a four-port wave rotor. A typical case with $PR = 1.8$, $n_C = n_E = 0.83$ for $TIT = 1500K$ is presented. The major reduction in $sfc$ is 23.9% for $r_c = 5$, descending to 5% for $r_c = 30$. Conclusively from Figures 6 and 7, performance
enhancement of two-shaft wave rotor-topped engines operating at design point conditions having $TIT = 1500$ K is maximised for low values of $r_c$. Referring to Figures 4 and 5, we conclude that the higher the $TIT$ (especially for low values of $r_c$) the more the benefits are for the engine’s performance.

### Parameters influencing performance of two-shaft gas turbines

Various studies carried out in the past, such as Jones and Welch (1996), Welch et al. (1999), Fatsis and Ribaud (1999), Povinelli et al. (2000) indicated that the main parameters influencing the performance of wave rotor topped two-shaft engines are:

(i) Wave rotor pressure ratio ($PR$).

(ii) Pressure losses variation at the ducts connecting the wave rotor to compressor, combustion chamber and turbine ($\Delta P_{duct}$), confirmed in previous studies by Fatsis and Ribaud (1999) and by Akbari and Müller (2003a).

Numerical studies done by Fatsis (2017) showed that the variation of the compression and expansion efficiencies inside the rotor ($\eta_C, \eta_E$) have a noticeable influence on $sfc$ at values of $TIT$ less than 1200 K. As $TIT$ increases, the influence of $\eta_C$ and $\eta_E$ variation is becoming negligible. The influence of these parameters on the topped engine performance in combination with the typical baseline engine parameters shown in Table 1, is examined setting a typical value of the $TIT$ for marine gas turbines, namely $TIT = 1500$ K, and applying compressor pressure ratios $r_c$ from 5 up to 25 or 30.

### Effect of wave rotor pressure ratio variation on performance of two-shaft gas turbines

Numerical and experimental studies carried out by Jones and Welch (1996), by Povinelli et al. (2000) and by Okamoto and Araki (2008) concluded that the operation of a four-port wave rotor is effective when its pressure ratio $PR$ attains the value of 1.8.

Figure 8 presents the $sfc$ – $ws$ distribution for two-shaft gas turbines with $TIT = 1500$ K for three different values of $PR = 1.4, 1.8$ and $2.2$. The $PR$ variation has a slight effect on $sfc$ for low compressor pressure ratio $r_c$ values, but no effect on $ws$. For $r_c > 10$, the gas turbine performance is independent of the wave rotor pressure ratio $PR$. It is interesting to notice that for $r_c < 10$, the case of $PR = 1.4$ results to slightly lower $sfc$ and slightly higher $ws$ than the one corresponding to the $PR = 2.2$ case. Therefore the influence of the wave rotor

![Figure 7. Reduction in specific fuel consumption ($sfc$) for two-shaft gas turbines topped with four-port wave rotor with $\eta_C = \eta_E = 0.83$, $PR = 1.8$ for $TIT = 1500$ K ($r_c$: compressor pressure ratio).](image-url)

![Figure 8. Performance of two-shaft gas turbines topped with four-port wave rotor with $\eta_C = \eta_E = 0.83$, $TIT = 1500$ K and variation of $PR$ from 1.4 to 2.2 ($r_c$: compressor pressure ratio).](image-url)
pressure ratio $PR$, has negligible effect on engine’s performance for $TIT = 1500$ K, when the rest of the parameters are kept unchanged.

**Effect of leakage and pressure losses variation**

The effect of the pressure losses in ducts connecting the wave rotor to compressor, combustion chamber and turbine, as well as leakage losses at the extremities of the wave rotor ($\Delta P_{duct}$), was investigated experimentally by Wilson et al. (2007). It was concluded that leakage deteriorates the performance of the wave rotor. Experiments revealed that reduction in mass flow of the order of 5% due to leakage, results to an equal reduction of the wave rotor pressure ratio. Measurements carried out by Hendricks et al. (1997) using bidirectional brush seals between the rotor and the endwall plates indicate that the use of brush seals improve the wave rotor efficiency from 36 to 45 percent at low leakage (i.e. small rotor endwall gap spacing) and from 15 to 33 percent at high leakage (i.e. larger endwall gap spacing). Details on the aerodynamic design of the ducts connecting the wave rotor to the compressor, combustion chamber and turbine are discussed by Slater and Welch (2005).

Leakage studies are incorporated to 1-Dimensional CFD codes by adding source terms only to the first and last cells of the computational domain formed inside the rotor channels. These terms affect only the continuity and energy equations, as is done by Paxson and Wilson (1993), Fatsis et al. (1998), Okamoto and Nagashima (2007). Akbari et al. (2008) presented a leakage quasi 1-Dimensional model taking into account the variations in pressure and temperature determining the local leakage circumferentially around the rotor.

Figure 9 presents the $sfc – ws$ distribution for the case where $PR = 1.8$, $TIT = 1500$ K, $\eta_C = \eta_E = 0.83$, for leakage and pressure losses $\Delta P_{duct} = 4\%$ and $8\%$. From this figure it can be seen that for $TIT = 1500$ K, when $\Delta P_{duct}$ is increasing, $ws$ is decreasing and at the same time $sfc$ is slightly increasing. The effect of pressure losses is more apparent in low-pressure ratios. The higher the pressure ratio, the less important is the effect of the losses on engine’s performance.

**Wave rotor two-shaft recuperated gas turbines analysis**

In marine gas turbines, recent advancements with the WR-21 ICR engine showed that significant fuel savings can be achieved by introducing a recuperator in the baseline engine, English (2003). This device recovers waste energy from the gas turbine exhaust, preheating the air entering the combustion chamber, improving cycle efficiency and reducing fuel consumption, Shepard et al. (1994). Wang et al. (2013), Chen et al. (2013) proposed an analytical method for thermodynamic optimisation of a triple-shaft open ICR gas turbine. Ji et al. (2017) proposed a comprehensive methodology to analyse the off-design performance of marine intercooled gas turbines.

Under this perspective, a through-flow four-port wave rotor can be introduced to the baseline gas turbine – recuperated cycle, as Figure 10 illustrates.

The thermodynamic calculation of the recuperator is based on an iterative procedure. Initially, the temperature of the ‘cold’ exit 5.2 of the heat exchanger is assumed ($T_{05,hyp}$). Then fuel mass flow through the combustion chamber is:

$$n_f = \frac{m \left( C_{ph} T_{06} - C_{pc} T_{05,hyp} \right)}{n_C \cdot FCV} \quad (16)$$

The ‘hot’ exit temperature of the recuperator ($T_{10}$) is calculated as:

$$T_{10} = T_{09} - n_{HE} (T_{09} - T_{5,1}) \quad (17)$$

where $n_{HE}$ is the efficiency of the recuperator (its value lies between 0.84 and 0.92).

The updated value of the ‘cold’ exit of the recuperator is calculated as:

$$T_{05,real} = T_{5,2} + n_{HE} (T_{09} - T_{5,1}) \quad (18)$$

If the quantity $|T_{05,real} - T_{05,hyp}|$ is less than a prescribed error (e.g. 0.001), then the calculation is converged, otherwise a new value $T_{05,hyp} = T_{05,real}$ is assumed and a new iteration begins.

The pressure loss in the recuperator is expressed by means of the recuperator pressure loss $\Delta P_{HE,loss}$ that takes values between 1% and 4%.

The performance map of two-shaft wave rotor recuperated engines with respect to the corresponding two-shaft baseline recuperated engines at design point conditions is shown in Figure 11. Dashed lines with triangular symbols illustrate the performance of the baseline recuperated engines, while continuous lines with spherical symbols of the same colour illustrate the performance of the wave rotor-topped recuperated two-shaft gas turbines.

From Figure 11, it can be observed that for the case of two-shaft recuperated wave rotor-topped engines the integration of the wave rotor increases significantly the engine’s efficiency for both low and high values of $rc$. At higher $TIT$ values, the performance curves of the topped engines recover their expected fish-hook shape.

Figure 12 shows qualitatively the effect of integrating a four-port wave rotor on two-shaft recuperated engines. A typical case with $PR = 1.8$, $n_C = n_E = 0.83$ for $TIT = 1500$ K is illustrated. In the same figure, the percentage increase in $n_{th}$ is shown (the specific power, $ws$ is not affected by the presence of the recuperator) and a net benefit in terms of $n_{th}$ increase for all values of $rc$ is achieved. More specifically, the prevalent increase in $n_{th}$ reaches 20.6% for compressor ratio $rc = 5$, whereas the minimum increase is always higher than 9%. This increase is kept almost constant as $rc$ increases, even for engines with $rc = 25$ or 30. Most of the aero-derivative gas turbines used for marine applications have compressor pressure ratios between 15 and 20. For all these engines, the gain in $n_{th}$ is greater than $10\%$. So the application of wave rotors to recuperated engines leads to important efficiency improvements.

Figure 13 shows qualitatively the effect on specific fuel consumption when integrating a four-port wave rotor on two-shaft recuperated engines. A typical case with $PR = 1.8$, $n_C = n_E = 0.83$ for $TIT = 1500$ K is presented. The decrease in $sfc$ is 29.2% for engines with $rc = 5$, and for engines with $rc = 20$ it is 24%. It can be seen from Figures 12 and 13 that performance enhancement of two-shaft...
wave rotor-topped recuperated engines operating at design point conditions with $T_{IT} = 1500\text{ K}$ is maximised for low values of $r_c$, but there is a net benefit which is kept almost constant independent of the value of compressor pressure ratio $r_c$. For the typical value of $T_{IT}$ examined, the net benefit in terms of thermal efficiency is more than 19% (from Figure 12) and the reduction in $sfc$ is close to 24% for all compressor pressure ratio values examined, fact that makes the integration of a four-port wave rotor to a marine gas turbine recuperated engine even more attractive.

A comparison between the wave rotor-topped two-shaft recuperated gas turbines and the wave rotor-topped two-shaft gas turbines without recuperator is illustrated in Figure 14.

One can observe that the thermal efficiency of wave rotor-topped two-shaft recuperated gas turbines, surpasses 50% for values of
Figure 13. Reduction in specific fuel consumption (sfc) for two-shaft recuperated gas turbines topped with four-port wave rotor with $\eta_C = \eta_E = 0.83$, PR = 1.8 for $TIT = 1500\,K$. ($r_c$: compressor pressure ratio).

Figure 14. Comparison of performances between wave rotor-topped two-shaft recuperated (continuous lines) and wave rotor-topped two-shaft gas turbines (dashed lines) ($r_c$: compressor pressure ratio).

$TIT \geq 1400\,K$, for small to moderate compression ratios and is comparable to combined cycle efficiencies, whereas the specific power attains almost the same value as the one calculated for the wave rotor-topped engines without recuperator. Benefits for wave rotor recuperated engines are observed for all values of $r_c$ and $TIT$. Taking as an example the case where $TIT = 1500\,K$, a marine wave rotor-topped and recuperated gas turbine with $r_c = 15$ have thermal efficiency of 50%. Similar thermal efficiency, of 47%, is attained for another wave rotor-topped recuperated gas turbine with $r_c = 20$ having the same $TIT$. These two examples clearly indicate that wave rotor topping enhances considerably the efficiency of typical marine recuperated gas turbines.

Integration of wave rotor in gas turbines

The concept of a prototype wave rotor topped demonstrator engine using as baseline engine the Rolls-Royce Alison 250 turboprop was presented by Snyder (1996) and by Welch et al. (1999). The modification of the original engine to include a four-port wave rotor and the associated ducting, while keeping the original compressor, turbine and combustor was evaluated. The rotor diameter and length found to be approximately equal to the tip diameter of the high pressure turbine of the original engine. The wave rotor was planned to be mounted on a separate shaft between the turbine and combustor, spinning coaxially with the gas turbine shaft at approximately one-third the speed of the gas generator spool through its operating range. The topped engine was predicted to produce 11.4% more shaft power (+20% specific power) with a 22% decrease in engine’s sfc at design point conditions. According to Snyder (1996), the greatest challenges of the topped engine are related to the re-design of the combustion chamber design and the design of the ducts connecting it to the wave rotor due to high temperatures.

Preliminary design estimates of marine gas turbines topped with four-port through-flow wave rotors

Based on the four-port wave rotor demonstrator engine designed by Snyder (1996), a reasonable estimation of the ratio of the external to internal rotor diameter, $d_e/d_i$ (see Figure 15) is $d_e/d_i = 1.5$. Assuming 60 rotor blades and having as input: (i) the mass flow rate of the baseline engine, (ii) two (2) cycles of operation per rotation and (iii) the external diameter of the high pressure turbine, an approximation
of the internal and external rotor diameter can be obtained. Exact dimensions of the circumferential extension of port openings can be acquired by calculating the wave diagram inside the rotor. This can be done either by: (a) numerical CFD calculations of the 1-Dimensional unsteady flow pattern inside the rotor considering various types of losses (e.g. Paxson and Wilson 1993; Fatsis et al. 1998; Okamoto and Nagashima 2007 and Akbari and Müller 2003b) or (b): analytical calculations of isentropic flow using the characteristics theory (Resler et al. 1994, 2001).

The integration of a four-port wave rotor to an existing aero-derivative gas turbine is expected to increase the overall length and width of the engine accounting for the rotor and the ducting connections with compressor, combustion chamber and high pressure turbine. Based on the mass flow rate of 70kg/s of a typical 25-MW engine, the rotor outside diameter could be $d_o = 0.81$ m, the rotor internal diameter $d_i = 0.54$ m and rotor length $L = 0.8$ m. Figure 15 shows a simplified cross section of the rotor geometry.

Typical dimensions of the GE LM2500 (GE 2018) are: length: 8.00 m, width: 2.74 m, height: 3.05 m and of the RR MT30 (RR 2020) are: length: 8.7 m, width: 2.7 m, height: 3.5 m. An increase of the engine’s length by 1.5m (including the coupling of the rotor shaft) and its width by 0.8 m (due to the diameter of ducts required to transfer compressed air and exhaust gases to the rotor) is expected. The height of the topped engine is not likely to exceed its original value. Figure 16 presents schematically the foreseen increase in dimensions of a typical 25–36 MW wave rotor-topped marine gas turbine.

Detailed investigation of the wave rotor-enhanced engine weight depends on the technical specifications of the engine. Welch et al. (1995) suggest that an assessment of the weight added due to the wave rotor and associated ducting for the case of small and intermediate turboshift engines (mass flow rates between 5 and 30 kg/s) is approximately 20% of the baseline engine core weight. The choice of materials used for the rotor, endwall plates, rotor casing and ducting, plays a key point to the overall weight estimation. Wilson et al. (2007) used titanium for the rotor designed for a four-port wave rotor experiment at NASA Lewis. Typical alloys that can also be used are: Ti 6Al-4 V (alpha-beta alloy, Grade 5) with density of 4430 kg/m$^3$ (Neonickel 2021) and alloy IN-738 with density of 8110 kg/m$^3$ (Nickel Institute 2021). A rough estimation of the mass of the rotor depicted in Figure 15 including the upstream and downstream endwall plates and the ducting, ranges between 300 and 400 kg depending on the material used. Taking as reference engines the GE LM2500 and the RR MT30, the extra weight due to the wave rotor is about 6% the weight of the unpacked baseline engine. Therefore, the overall mass of the baseline engine is not significantly differentiated by the addition of the rotor’s extra mass.

**Performance benefits of existing marine gas turbines**

The expected merits of wave rotors for marine propulsion systems consist of:

- Increase of thermal efficiency.
- Increase of specific power.
- Reduction of specific fuel consumption.
- Minor to medium increment of two out of three engine’s dimensions (length – width) with insignificant changes in height and mass.
- Possibility of in-rotor constant volume combustion (Elharis et al. 2010), replacing the conventional combustion chamber of gas turbines.

Possible demerits are related to the fact that the wave rotor technology being novel and under development, needs investments to be conducted, namely:

- Design and manufacturing of necessary ducting to connect the wave rotor to existing components of the gas turbine.
- Cooling requirements of the ducts connecting combustion chamber exit to wave rotor ‘hot’ inlet port.
- Fabrication of prototypes for experimental performance validation and wave rotor-topped engines design optimisation.

Applying the thermodynamic model described previously, Table 3 summarises the performance benefits in terms of power and efficiency.

| Engine     | Power (MW) | Efficiency % |
|------------|------------|--------------|
| GE LM 2500 | Baseline   | 25           | 26.7          | 37 | 39.5 |
|            | Topped     | 36           | 38.1          | 40 | 42.4 |
| Alstom GT35| Baseline   | 17           | 18.3          | 33 | 36   |
|            | Topped     | 36           | 38.1          | 40 | 42.4 |
| Solar Taurus 70 | Baseline | 8.2           | 8.8           | 34 | 37.7 |
|            | Topped     | 36           | 38.1          | 40 | 42.4 |
| MT30       | Baseline   | 36           | 38.1          | 40 | 42.4 |
|            | Topped     | 25           | 26.7          | 37 | 39.5 |
| WR-21      | Baseline   | 25           | 26.7          | 37 | 39.5 |
|            | Topped     | 36           | 38.1          | 40 | 42.4 |
efficiency enhancement of some of gas turbines used widely in marine propulsion applications.

**Conclusion**

In this article, the performance assessment of two-shaft gas turbine engines topped with a four-port through-flow wave rotor as a prime mover for naval ships was performed. Integration of wave rotor technology in marine gas turbines can moderate the fuel consumption and increase the specific power of the engine. Rough estimation of the extra weight of the wave rotor and associated ducting was found to increase the weight of typical marine aero-derivative engines by 6%. Previous references (Welch et al. 1995) discussing the wave rotor topping of small and medium turboshaft engines estimate the extra weight increase to 20%. This difference can be attributed to the fact that in small and medium-range engines the extra weight of the topped engine compared to the weight of the baseline engine is significant. For the typical marine aero-derivative engines examined, the topped engine weight increase is a small percentage of the naval ship deadweight and the extra dimensioning due to the wave rotor and the associated ducting is not expected to impose major changes in the machine room. In the thermodynamic model developed, the compressor, gas turbine and compressor turbine of the baseline engine are kept unchanged to keep the wave rotor’s integration cost low.

However, Akbari and Müller (2003a) investigated other integration scenarios requiring the re-design of some of the baseline components that lead to similar performance benefits. In rotor constant volume pressure gain combustion as new promising technology, however, presents some unique challenges (Akbari and Nalim 2009). Fundamental combustion patterns, ignition and flame control remain poorly understood so far. Unsteady combustion in the wave rotor channels involves complex flame–wave interactions fundamentally different from familiar internal combustion engines and steady-flow gas turbine combustion and they need extensive investigation (Nalim et al. 2017).

Ambient pressure and temperature, thermodynamic constants for the air and hot gases, thermal efficiencies for compressor, compressor turbine and power turbine as well as compression and expansion efficiencies for the processes inside the wave rotor, are the input data required for the calculations of the present model. Performance maps at design point illustrate the benefits of wave rotor-topped engines with respect to the corresponding baseline engines. Depending on the design requirements concerning specific power and specific fuel consumption, the topped engines maps help to select the most favourable engine and its precise operating conditions (compressor pressure ratio, TIT).

For two-shaft engines working at a given compressor pressure ratio, the higher the TIT is, the more the benefit gain of the wave rotor topped engine is attained in terms of thermal efficiency, specific fuel consumption and specific power. Assuming compression and expansion efficiencies inside the wave rotor, as well as ducting and leakage pressure losses specified by previous researchers, it was demonstrated that for typical marine aero-derivative engines listed in Table 3 the increase in thermal efficiency remains higher than 6% and the increase in specific power remains higher than 5% at \( TIT = 1500 \text{ K} \).

The parameters selected as important for the performance of the wave rotor and of the topped engine are: The wave rotor pressure ratio and the ducting and pressure losses associated with the wave rotor. Each of these parameters is varied around a mean value which is well established in the literature and used by other researchers in the past.

The influence of wave rotor pressure ratio (PR) on specific fuel consumption is negligible for \( r_c > 10 \) for \( TIT = 1500 \text{ K} \). For all the cases examined no influence of PR variation on specific power was observed.

Leakage and pressure losses are mainly influencing specific power, while specific fuel consumption remains almost unchanged. Results showed that when the pressure losses increase, specific fuel consumption also increases whereas specific power decreases, for all values of compressor pressure ratios examined. The effect of pressure losses is more apparent for low-pressure ratios. The higher the pressure ratio, the minor the effect of losses on engine’s performance is.

For the case of wave rotor-topped recuperated marine gas turbine engines, the thermal efficiency increases by at least 19% and the specific fuel consumption decreases by at least 24% for all pressure ratios examined for the typical value of \( TIT = 1500 \text{ K} \) with respect to the baseline recuperated engines. Peak thermal efficiency can exceed 50% for \( TIT > 1400 \text{ K} \) at low values of the compressor pressure ratio. Efficiency and specific fuel consumption enhancement of marine recuperated wave rotor gas turbines attain significant levels even for large compressor pressure ratios, fact that makes attractive the integration of wave rotor to this type of engines.

The integration of four-port wave rotors to marine gas turbines has the potential to improve their performance, although there are challenges to be successfully surpassed.

**Disclosure statement**

No potential conflict of interest was reported by the author(s).

**Notes on contributor**

Professor Antonios Fatsis is a Mechanical Engineer, graduated in 1989 from the University of Patras, Greece. He pursued post-graduate studies in the Turbomachinery Department of the “von Karman Institute for Fluid Dynamics” (VKI) in 1990 and obtained his Ph.D. degree in Applied Sciences in 1995. He worked as a research engineer at the Energetics Department of the “Office National d’Etudes et de Recherches Aéropatiales” (ONERA), France, until 1997. He acquired experience in the maritime industry working at Registro Italiano Navale SpA until 2005. In October 2005, he joined the Mechanical Engineering Department of the Technological University of Central Greece where he was appointed as the Head of the Internal Combustion Engine and Gas Turbine Laboratory. Since 2018 he is full Professor and faculty member of the General Department of the Kapodistrian University of Athens. He has published several journal and conference articles on topics related to internal combustion engines, turbomachinery, pumps and Computational Fluid Dynamics.

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**I n d e x t e r m s:**

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- Turbomachinery
- Recuperator
- Hydrogen
- Wave rotors
- Marine propulsion
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