

Parametric Study on Performance Characteristics of Wave Rotor Topped Gas Turbines

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Abstract—Implementation of wave rotor-topping techniques to gas turbines intends to reduce specific fuel consumption and increase engine's specific work. In order to keep low the cost of wave rotor integration to existing gas turbines, the original engine's compressor and turbine can remain unchanged. The performance of the wave rotor-topped engine is calculated using non-adiabatic thermodynamic models. The parameters chosen to be investigated are the wave rotor pressure ratio, wave rotor expansion and compression efficiencies, along with engine's parameters such as compressor pressure ratio and turbine inlet temperature. To accomplish this study, each of these characteristics is varied, while all the rest remain unchanged. It is concluded that variation of wave rotor characteristic parameters influences primarily the engine's specific fuel consumption for high values of compressor pressure ratio and low values of turbine inlet temperature, while the engine's specific work remains almost unchanged.

Keywords — Wave rotor, gas turbine, regeneration, performance, specific work, specific fuel consumption.

I. INTRODUCTION

Wave rotor technology is a very advantageous tool for gas turbine performance improvement. Integration of a wave rotor to gas turbines may reduce the specific fuel consumption and increase the specific power delivered by the engine, as stated in [1]. A wave rotor is composed of a rotor consisting of a number of axial straight blades between two coaxial cylinders inside a stationary casing. Two stationary endwall plates are located at the rotor extremities. These plates have circumferential openings allowing partial inflow and outflow through the rotor blade channels, as described in [2], [3]. In Fig. 1, one can see the four-port wave rotor configuration, which is best suited for gas turbine applications. When a wave rotor is integrated in 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. These waves are propagated inside the channels and depending on the relative circumferential position of the openings at the endwall plates of the rotor, either they reflect back inside the rotor channels as stronger compression waves or are directed towards the combustion chamber, creating simultaneously expansion waves moving in the opposite direction towards the turbine for further expansion, as described in [4]. Thus, there is an unsteady energy exchange between high enthalpy hot gases and low enthalpy air by means of pressure waves. The unsteady nature of the flow pattern inside the wave rotor

(or pressure wave supercharger) ensures the fact that there is no mixing between the air and the hot gases inside the rotor, [2]. The compression efficiency by means of unsteady pressure waves may supersede the efficiency of conventional steady flow devices. It has been shown in [5] that for pressure gain up to 2.2, the shock wave efficiency is larger than the efficiency of a steady flow diffuser. In such a case, the frictionless shock wave efficiency is estimated by the same authors as 93%

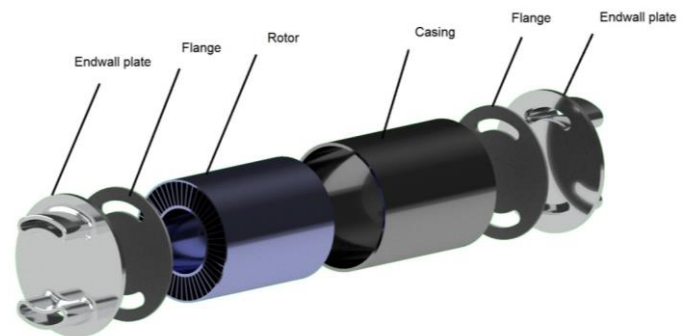


Fig. 1. Four-port wave rotor schematic configuration

The flow pattern inside the rotor channels is fully unsteady, dominated by the propagation of compression and expansion waves. Various numerical models have been proposed in [6], [7], [8] and [9] to predict the unsteady flow patterns inside the rotor passages. The flows entering and exiting the rotor in the ports of the stationary manifolds are almost steady, apart from the fluctuations due to the gradual opening and closing of the rotor channels as they enter and exit the ports, as concluded in [10].

There are several possibilities to integrate a wave rotor in an existing gas turbine. An optimization study performed in [11] analyzes five different configuration possibilities. The configuration assuming unchanged baseline compressor and turbine inlet temperature provides almost the best gas turbine performance enhancement of the wave rotor topped engine. Results indicate that the performance of the topped engine is always higher than that of the corresponding baseline engine with the same compressor pressure ratio value. The benefit is greater for lower compressor pressure ratios, whereas for higher ones the benefit diminishes. In fact, for compressor pressure ratios greater than 20, almost no benefit can be obtained. This clearly favors the wave rotor-topping for small gas turbines with low compressor pressure ratios, as concluded in [12].

Performance maps of the topped engine were generated in [11] by varying the compressor pressure ratio, and the turbine inlet temperature. The above investigation assumed that the compression efficiency (η_C) and expansion efficiency (η_E) have constant values, namely $\eta_E = \eta_C = 0.83$. These values were also applied in [13] and [14] in previous wave rotor studies. Even though wave rotor topped gas turbine engines seem to be beneficial throughout a wide range of gas turbine sizes, efficiencies, and operating conditions, the ones to be mostly favored are the ones with low component efficiencies, as it was concluded in [15].

The majority of the above research work has investigated the performance benefits of wave rotor-topped engines for industrial as well as for aeronautical applications, keeping the wave rotor operating parameters constant. To begin with, this article identifies the most important wave rotor parameters affecting the performance of the whole gas turbine. As a second step, each of these parameters is varied while all the rest remain constant, resulting to a parametric study that depicts the gas turbine performance in terms of each individual wave rotor characteristic.

II. ONE-SHAFT AND TWO-SHAFT GAS TURBINE CYCLE CALCULATIONS

A. Input data for one and two-shaft gas turbines

The procedure of the thermodynamic calculations of one and two-shaft gas turbine cycles with the integration of a four-port wave rotor is described in detailed in [14] and in [16]. Figure 2 illustrates the configurations for the one and two-shaft gas turbines used in this article. The calculations of the baseline (without the wave rotor) as well as the topped (with the wave rotor) engines are carried out assuming irreversible processes in all the gas turbine components. The thermodynamic model accounts also for cooling the turbine in case $TIT \geq 1300$ K by subtracting air flow from the compressor.

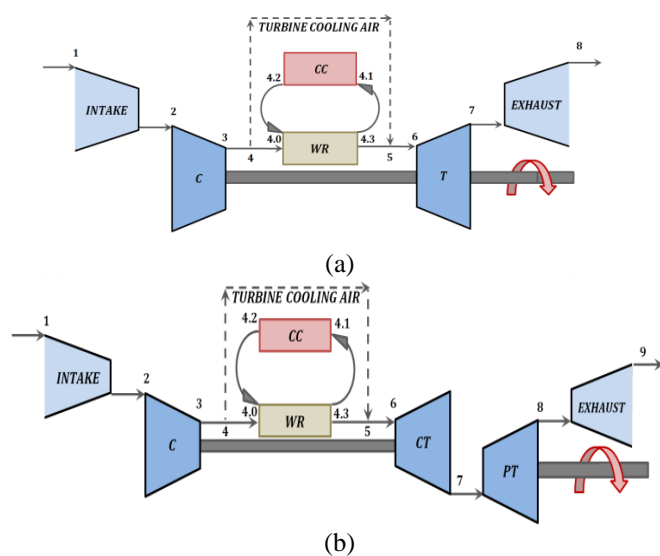


Fig. 2 (a) One-shaft and (b) Two-shaft gas turbine configurations, C: compressor, T: turbine, CC: combustion chamber, WR: wave rotor, CT: compressor turbine, PT: power turbine

B. Wave Rotor thermodynamic calculations

Figure 3 illustrates the model developed in this article to calculate the thermodynamic properties of air and hot gases when a four-port wave rotor is integrated to a one-shaft gas turbine. The “cold” entry of the wave rotor inflow port (point 4.0) receives compressed air from the compressor exit and releases it after being further compressed inside the rotor to the combustion chamber intake (point 4.1). The hot gases from the combustion chamber (point 4.2) enter the wave rotor “hot” inflow port, come in contact with the cold air being already inside the rotor channels, create compression waves to the air flow (compressing thus further the air from the compressor), get expanded and finally get directed to the outflow port towards the turbine (point 4.3) for further expansion.

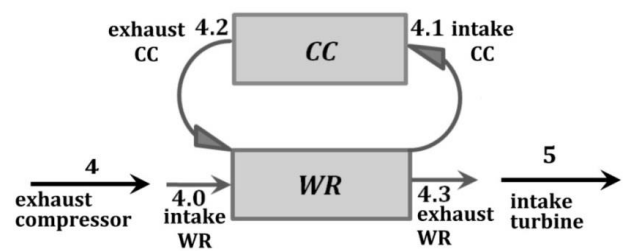


Fig. 3. Symbols used for the four-port wave rotor thermodynamic calculations

The wave rotor pressure ratio is a very important parameter that characterizes the performance of the wave rotor and accordingly the performance of the whole gas turbine. It is defined as:

$$PR = \frac{P_{4.1}}{P_{4.0}} \quad (1)$$

This parameter gives the extra compression inside the wave rotor of the air flow stream exiting the gas turbine compressor.

Stagnation temperature at the cold air port of the wave rotor, $T_{4.0}$

$$T_{4.0} = T_{04} \quad (2)$$

Stagnation temperature at the port towards the turbine, $T_{4.3}$

$$T_{4.3} = TIT \quad (3)$$

Stagnation pressure at the cold air port of the wave rotor $P_{4.0}$

$$P_{4.0} = P_{04} \left(1 - \frac{\Delta P_{duct}}{100} \right) \quad (4)$$

where ΔP_{duct} are the pressure losses at the ducts connecting the wave rotor to compressor, combustion chamber and turbine.

Stagnation temperature at the wave rotor exit towards the combustion chamber, $T_{4.1}$

$$T_{4.1} = T_{4.0} \cdot \left(\frac{PR^{(\gamma_c-1)/\gamma_c} - 1}{\eta_c} + 1 \right) \quad (5)$$

where η_c is the compression efficiency inside the wave rotor Stagnation pressure at the combustion chamber outlet $T_{4.2}$

$$T_{4.2} = \frac{T_{4.3}}{1 - \left[1 - \left(\frac{P_{4.3}}{P_{4.2}} \right)^{(\gamma_h-1)/\gamma_h} \right] \cdot \eta_E} \quad (6)$$

where η_E is the expansion efficiency inside the wave rotor.

III. PARAMETRIC STUDY FOR ONE-SHAFT AND TWO-SHAFT GAS TURBINES

Figure 4 presents the performance curves of wave rotor topped one-shaft gas turbines at design point illustrated with continuous lines in comparison to the base line (without wave rotor) one-shaft gas turbines illustrated with dotted lines. As a general conclusion, it can be stated that the performance curves of the wave rotor-topped engines are shifted to the lower right part of the diagram. According to [14], for low values of TIT , the integration of the wave rotor reduces significantly the engine's specific fuel consumption especially at high values of Rc . It increases the engine's specific work mainly at high values of TIT and low values of Rc . For $TIT=900$ K, $TIT=1000$ K and $TIT=1100$ K, the fish-hook shapes of the base line engine curves get smoothed for the topped engines. At higher TIT values, the performance curves of the topped engines recover their expected fish-hook shape.

Having identified the main flow parameters of the wave rotor, a parametric study investigates the effects on the engine's performance according to:

1. Wave rotor pressure ratio variation (PR). It is chosen in the range between 1.4 and 2.2 while keeping the rest parameters of both gas turbine and wave rotor constant.

2. Wave rotor compression and expansion efficiencies variation, (η_c, η_E). Both of them are chosen in the range between 0.75 and 0.92 while keeping the rest parameters of both gas turbine and wave rotor constant.

A. Effect of wave rotor pressure ratio variation

Numerical and experimental studies carried out in the past in [11], [12], [15], concluded that an adequate value of the wave rotor pressure ratio is $PR=1.8$.

In order to study the effect of the wave rotor pressure ratio, PR , on the performance of the gas turbine, thermodynamic calculations were done for three values of PR , namely, $PR=1.4, 1.8, 2.2$, while keeping the compression and expansion efficiencies inside the wave rotor η_c, η_E constant and equal to 0.83, a value recommended in the literature. Fig. 5 presents the distribution of specific fuel consumption in terms of specific work, for different values of the turbine inlet temperature TIT starting from 900 K to 1600 K. At $TIT=900$ K and $Rc=5$ one can observe small variations in specific fuel consumption (sfc) when PR is varied from 1.4 to 2.2. At high values of Rc , the sfc is strongly influenced by the value of PR . Specifically, for $Rc=20$ and $PR=2.2$, sfc is three times less than sfc corresponding to $PR=1.4$. From Fig. 5 it can be seen that the wave rotor pressure ratio plays an important role to the performance of one and two-shaft gas turbines, especially at large values of compressor pressure ratios (Rc) and low turbine inlet temperatures (TIT). As the wave rotor pressure ratio increases and for $TIT=900$ and $TIT=1000$ K, significant reduction in sfc can be noticed, while specific work (ws) remains almost unchanged. For values of $TIT=1200$ K and $TIT=1400$ K the gain in terms of sfc is less, although a slight reduction in ws is observed.

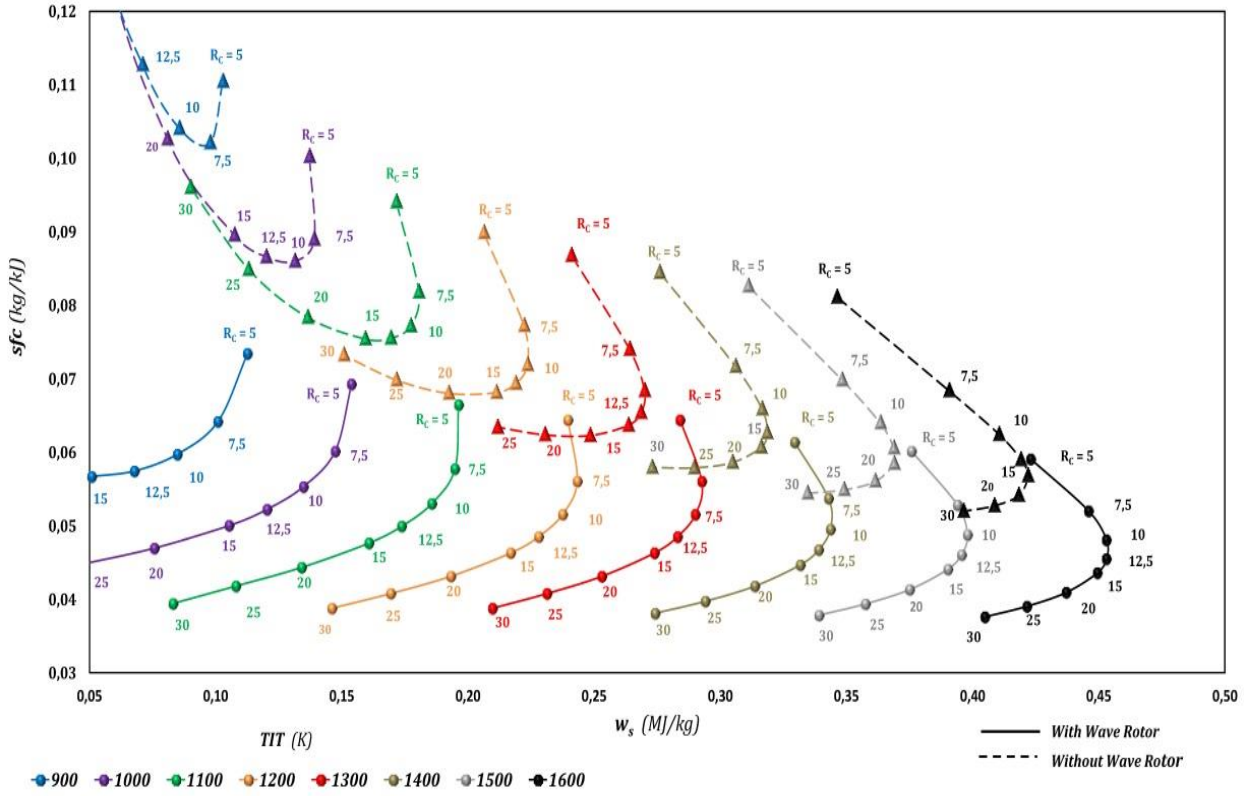
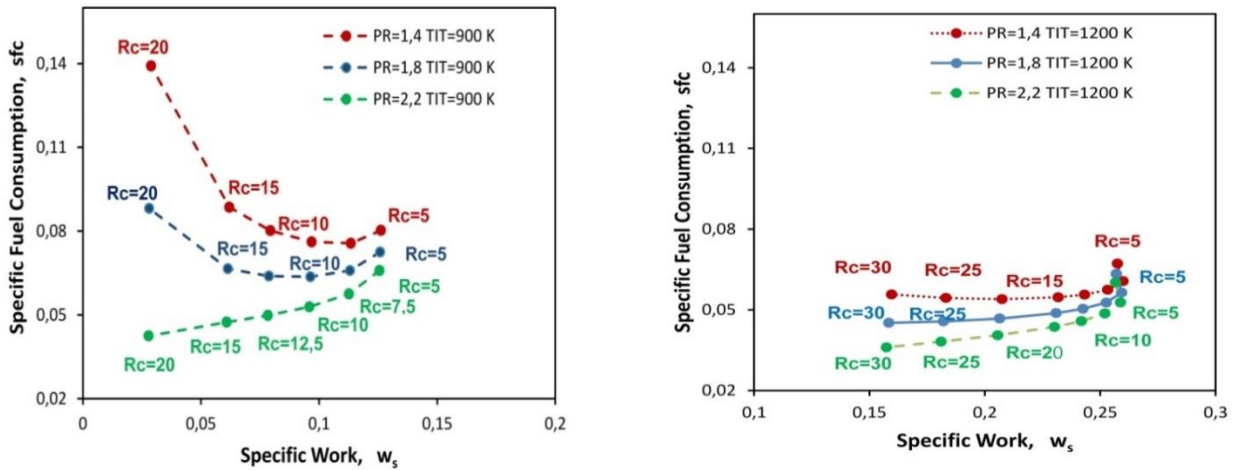


Fig.4. Performance of baseline and wave rotor topped one-shaft gas turbines at design point



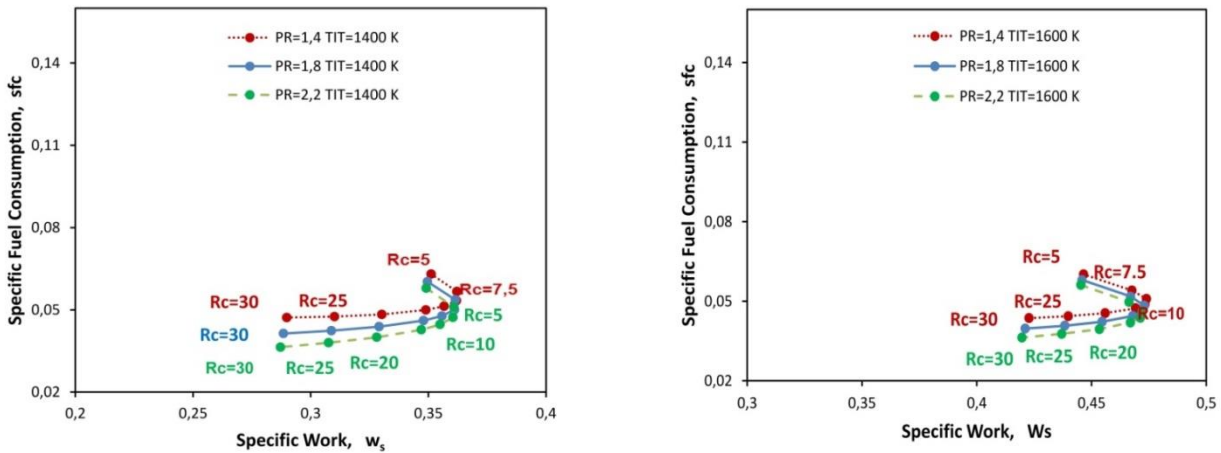


Fig. 5 Performance of one-shaft gas turbines topped with four-port wave rotor with $\eta_c = \eta_e = 0.83$ and variation of PR from 1.4 to 2.2 and TIT from 900 K to 1600 K

B. Effect of compression and expansion efficiency variation

One and two-shaft engine performance curves are obtained by keeping constant the value of PR , variation of η_e, η_c falls in the range of 0.75 up to 0.92 and TIT varies from 900 K to 1600 K.

In the left part of Fig. 6 one can see the performance of one-shaft gas turbines when the compression and expansion efficiencies inside the rotor are varied from 0.75 to 0.92, for $PR=1.4$ and $TIT=900$ K. Variations in sfc for different values of η_e and η_c can be observed only for the case of $Rc=20$, whereas for lower pressure ratios the values of sfc almost coincide. For $Rc=20$, important improvement can be observed between the values of 0.75 - 0.83 for η_e and η_c , but no significant variation is observed between this value up to 0.92.

For $TIT \geq 1200$ K the effect of variation of η_e and η_c is negligible and the performance curves coincide. This is why only the case for $TIT=1200$ K is illustrated in the right part of Fig. 6.

Similar conclusions are drawn from the results illustrated in Fig. 7, where $PR=1.8$. Slight variations between performance curves can be seen only for the case of $Rc=20$ and $TIT=900$ K between the curves corresponding to the three different values of η_e and η_c . At the right part of the same figure for $TIT=1200$ K the curves corresponding to the different values of η_e and η_c almost coincide, especially for low values of Rc . For the case that $PR=2.2$ one can observe slight variations for the three curves of different values of η_c and η_e at high values of Rc , in Fig. 8. At lower values or Rc the discrepancies are becoming less and finally the three curves coincide. The differences for $TIT=1200$ K are less than for $TIT=900$ K.

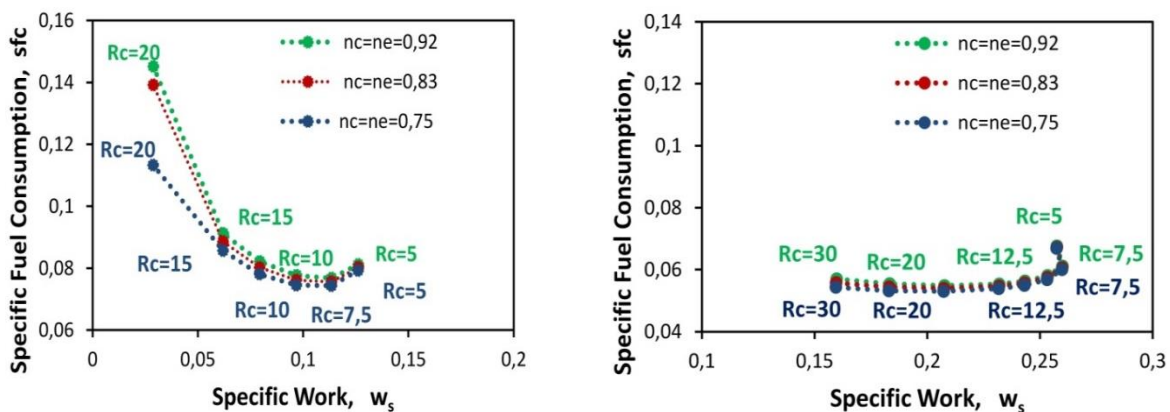


Fig. 6 Performance of one-shaft gas turbines topped with four-port wave rotor with $PR=1.4$ and η_e, η_c varying from 0.75 to 0.92 for $TIT=900$ K (left) and $TIT=1200$ K (right)

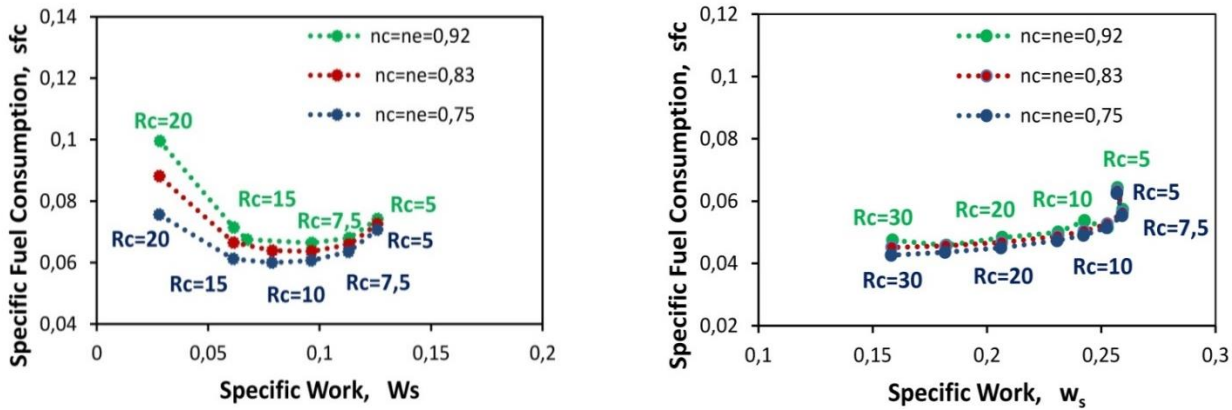


Fig. 7 Performance of one-shaft gas turbines topped with four-port wave rotor with $PR=1.8$ and η_E, η_C varying from 0.75 to 0.92 for $TIT=900$ K (left) and $TIT=1200$ K (right)

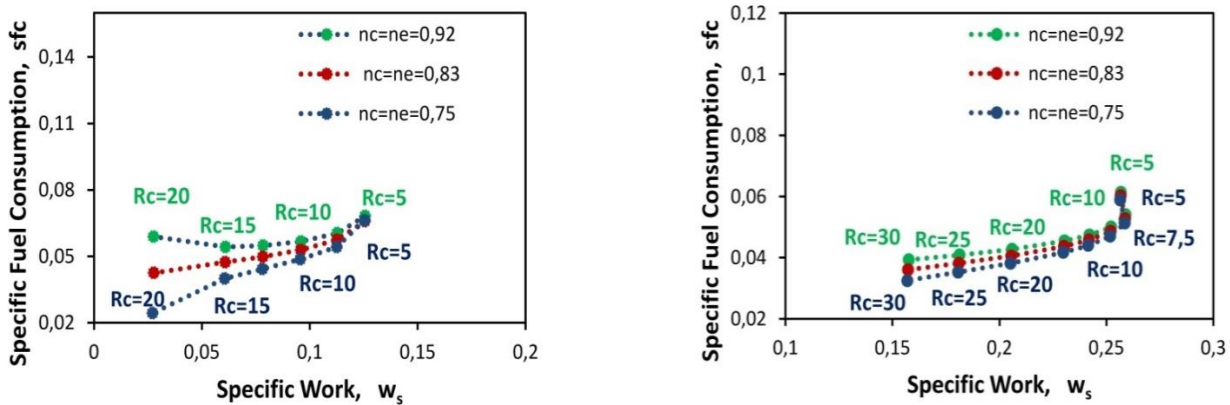


Fig. 8 Performance of one-shaft gas turbines topped with four-port wave rotor with $PR=2.2$ and η_E, η_C varying from 0.75 to 0.92 for $TIT=900$ K (left) and $TIT=1200$ K (right)

IV. PARAMETRIC STUDY FOR REGENERATIVE GAS TURBINES

The addition of heat exchanger in one or two shaft gas turbines is implemented mainly in ground based power plants where the engine's weight and dimension are not issues. It was shown in [16] that the incorporation of a heat exchanger leads to reduced specific fuel consumption and increased specific work, especially at low compressor pressure ratios. Figure 9 illustrates configurations of one and two shaft wave rotor-topped gas turbines including heat exchanger which are analyzed in the present study.

The main flow parameters for a regenerated wave rotor topped gas turbine were identified in [16] and are the same as the one and two shaft wave rotor topped engines without heat exchanger discussed in a previous paragraph.

Parametric study was carried out to determine the specific fuel consumption in terms (sfc) of the specific work (ws) by varying Rc from 5 to 30 and TIT from 900 K to 1600 K and η_C, η_E from 0.75 to 0.92. for different values of PR and different values of TIT .

Figure 10 illustrates $sfc - ws$ distributions for $\eta_C=\eta_E=0.92$ and variation of PR from 1.4 to 2.2 and $TIT = 1200$ K, 1400 K. It can be observed that for given values of Rc and TIT , ws

is not significantly affected by the variation of PR . On the contrary, sfc is increasing for engines with given values of PR, TIT, η_C, η_E , and high compressor pressure ratios, Rc .

A. Effect of wave rotor pressure ratio variation

In order to study the effect of the wave rotor pressure ratio, PR , on the performance of the gas turbine, thermodynamic calculations were done for three values of PR , namely, $PR=1.4, 1.8, 2.2$, while keeping the compression and expansion efficiencies inside the wave rotor η_C, η_E constant and equal to 0.83, a value recommended in the literature. Figure 10 presents the distribution of specific fuel consumption in terms of specific work at turbine inlet temperatures $TIT = 1200$ K and $TIT = 1600$ K.

It can be seen that the variations in terms of ws are less distinct with respect to the wave rotor topped engines without heat exchanger, Fig. 5. When comparing the left and the right part of Fig. 10, it is observed that the sfc variation is slightly more apparent for the case of $TIT=1200$ K, whereas the ws variation is more apparent for the case of $TIT=1400$ K.

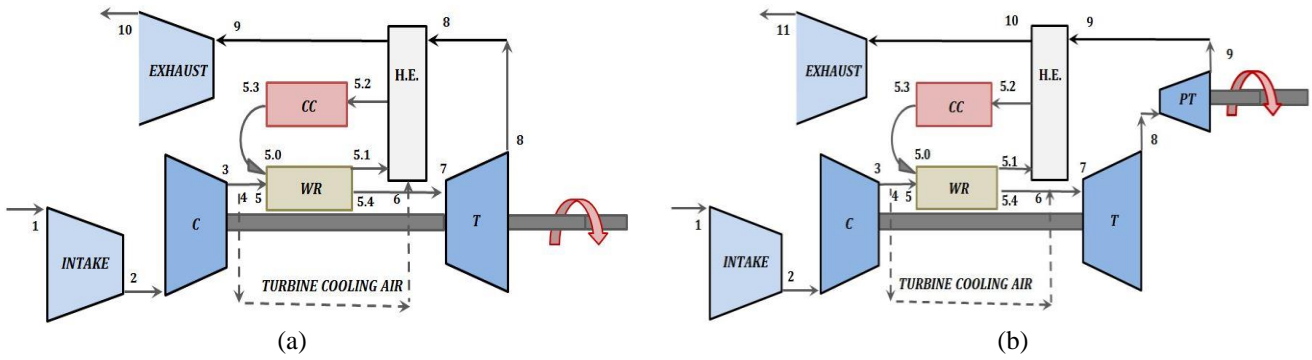


Fig. 9 (a) One-shaft and (b) Two-shaft gas turbine with heat exchanger configurations, C: compressor, T: turbine, CC: combustion chamber, WR: wave rotor, T: compressor turbine, PT: power turbine

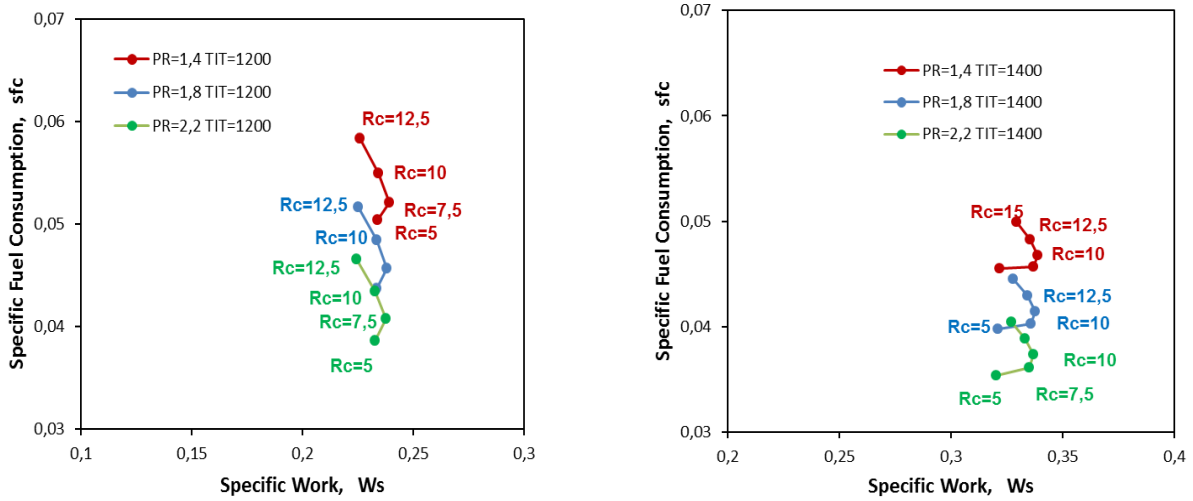


Fig. 10 Performance of one-shaft gas turbines topped with four-port wave rotor with $\eta_c=\eta_e=0.92$ and variation of PR from 1.4 to 2.2 and $TIT=1200$ K (left), $TIT=1400$ K (right)

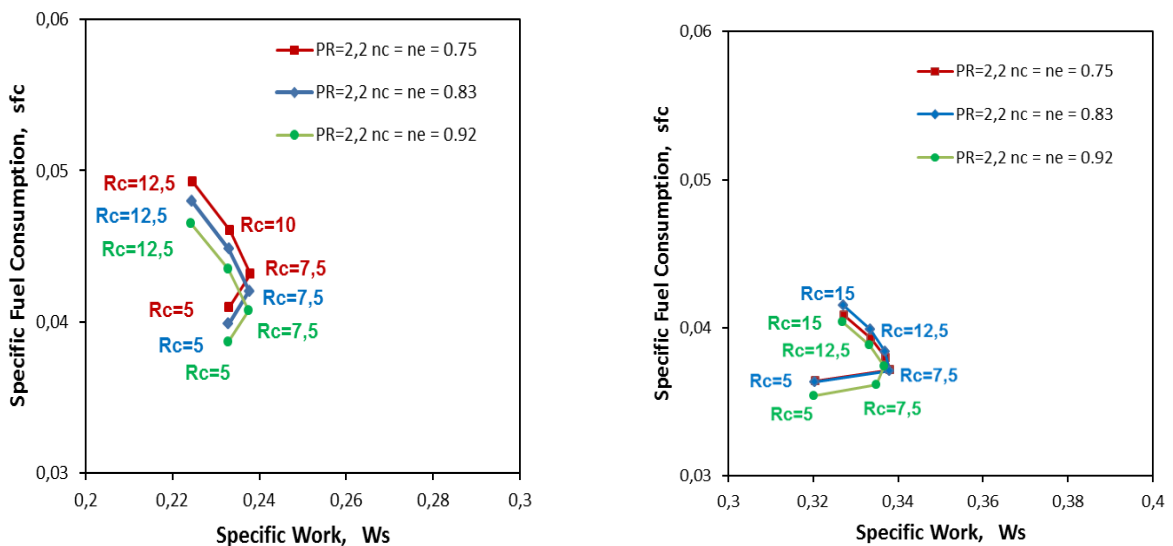


Fig. 11 Performance of one-shaft gas turbines topped with four-port wave rotor with $PR=2.2$ and variation of $\eta_c=\eta_e$ from 0.75 to 0.92 and $TIT=1200$ K (left), $TIT=1400$ K (right)

B. Effect of compression and expansion efficiency variation.
In the left part of Fig. 11 one can see the performance of one-shaft gas turbines with heat exchanger for $PR=2.2$ when the compression and expansion efficiencies inside the rotor are varied from 0.75 to 0.92, for $TIT=1200$ K and $TIT=1400$ K. For the three values of η_E and η_C examined, more important variations in terms of sfc are observed for the case $TIT=1200$ K, whereas more important variations in terms of ws are observed for the case $TIT=1400$ K. It can be seen that for both cases $TIT=1200$ K and 1400 K, the effect of η_E , η_C variation on performance is not important.

V. CONCLUSIONS

A parametric study was carried out to investigate the effect of flow parameters associated to the wave rotor for one and two-shaft industrial gas turbines with and without heat exchanger. The parameters chosen as important for the performance of the wave rotor and of the whole engine are: The wave rotor pressure ratio, the efficiency of the compression and expansion processes inside the rotor 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 researches in the past. For the wave rotor topped engines without heat exchanger, it is found that the wave rotor pressure ratio (PR) affects mostly the engine's performance, especially the specific fuel consumption (sfc) at values of TIT less than 1200 K and at high compressor pressure ratios (Rc). The more the PR increases, the more the sfc decreases at a given value of TIT . At high values of Rc the influence of PR is more important, whereas for low values of Rc , the PR affects less the engine's performance. The variation in PR gives almost no effect in the specific work (ws) of the engine. The variation of the compression and expansion inside the rotor (η_C , η_E) influences the sfc at values of TIT less than 1200 K and at high values of PR . At $PR=1.4$, the influence of η_C , η_E is negligible at values of Rc less than 15 . For higher values of PR , the influence of η_C , η_E can be seen only at high values of Rc . As TIT increases, the influence of η_C and η_E variation is becoming negligible. Engines with high TIT values are also not affected by the variations of η_C , η_E . This means that engines with low compressor pressure ratios seem to be affected less than engines with high pressure ratios. For the wave rotor topped engines with heat exchanger, the parametric study showed that the wave rotor pressure ratio (PR) has less distinct effect on the engine's performance; it is

limited mainly at low values of TIT . The variation of the compression and expansion inside the rotor (η_C , η_E) influences the engine at low values of TIT , whereas for high values of TIT the performance curves almost coincide.

REFERENCES

- [1] Jones, S.M. and Welch, G.E., "Performance Benefits for Wave Rotor-topped Gas Turbine Engines", *International Gas Turbine and Aeroengine Congress & Exhibition, Paper No. 96-GT-75*, 1996.
- [2] Weber H.E., "Shock Wave Engine Design", John Wiley and Sons, New York, 1995.
- [3] Povinelli, L.A., Welch, G.E., Bakhle, M.A. and Brown, G.V., "Potential Application of NASA Aerospace Technology to Ground-Based Power Systems", *NASA TM 2000-209652*.
- [4] Akbari P, Nalim R, Müller, N, "Performance Enhancement of Microturbine Engines Topped With Wave Rotors". *ASME Journal of Engineering for Gas Turbines and Power*, vol.128, 2006, pp.190-202.
- [5] Akbari, P., Kharazi, A.A. and Müller N., "Utilizing Wave Rotor Technology to enhance the turbo compression in power and refrigeration cycles", *Proceedings of International Mechanical Engineering Congress & Exposition, Paper No.2003-44222*.
- [6] Paxson, D., "An improved numerical model for Wave Rotor Design and Analysis", *NASA TM 105915*, 1993.
- [7] Welch, G.E. and Chima R.V., "Two-Dimensional CFD Modeling of Wave Rotor Flow Dynamics", *NASA TM 106261*, 1993.
- [8] Fatsis A, Orfanoudakis N, Pavlou D, Panoutsopoulou A, Vlachakis N "Unsteady flow modelling of a pressure wave supercharger", *Proc. IMechE Vol. 220 Part D: J. Automobile Engineering*, vol. 220, 2006, pp. 209-218.
- [9] Okamoto, K. and Nagashima, T., "Simple Numerical Modeling for Gasdynamic Design of Wave Rotors", *Journal of Propulsion and Power*, vol. 23(1), 2007, pp. 99-107.
- [10] Slater JW, Welch G.E., "Design of a Wave-Rotor Transition Duct", *AIAA Paper No. 2005-5143*.
- [11] Akbari P, Müller N, "Performance investigation of small gas turbine engines topped with wave rotors", *AIAA Paper No. 2003-4414*.
- [12] Fatsis, A., Ribaud, Y., "Thermodynamic analysis of gas turbines topped with wave rotors". *Aerospace, Science and Technology*, No. 5, 1999, pp. 293-299.
- [13] Akbari P., Nalim R., Donovan, E.S. and Snyder, P.H. (2008), "Leakage Assessment of Pressure-Exchange Wave Rotors", *Journal of Propulsion and Power*, vol. 24 (4), 732-740.
- [14] Fatsis A., "Performance Enhancement of One and Two-Shaft Industrial Turbohaft Engines Topped With Wave Rotors", *Int. Journal of Turbo and Jet Engines*, ahead of print, 2016.
- [15] Dempsey E, Müller N., "Performance Optimization of Gas Turbines Utilizing Four-Port Wave Rotors", *AIAA Paper No. 2006-4152*.
- [16] Fatsis, A., "Regenerated Turbohaft Engines for Ground Power Plants Topped with Four-Port Wave Rotors", *International Journal of Engineering Research and Technology (IJERT)*, vol 5(9), 2016.
- [17] Wilson, J., "Design of the NASA Lewis 4-Port Wave Rotor Experiment". *AIAA Paper 97-313*.