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Assessment of parameters affecting the performance of Wave Rotor-Topped Industrial Turboshaft Engines

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***Abstract*—Implementation of wave rotor-topping techniques to gas turbines is intending to reduce specific fuel consumption and increase engine's specific work. In order to keep low the cost of a four-port wave rotor integration to an existing industrial gas turbine, the multistage compressor and turbine of the engine can remain unchanged. The wave rotor-topped engine performance can be calculated using non-adiabatic thermodynamic models. The influence of wave rotor flow characteristic parameters such as pressure ratio, expansion and compression efficiencies, along with engine's parameters such as compressor pressure ratio and turbine inlet temperature is studied. For this purpose, 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; turboshaft; performance; specific work; specific fuel consumption

I. INTRODUCTION

Wave rotor technology is a very promising 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 [2] and [3] described. 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, [4]. Thus, there is an unsteady energy exchange between high enthalpy hot gases and low enthalpy air by means of pressure waves. It was stated in [2] that 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. 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%.

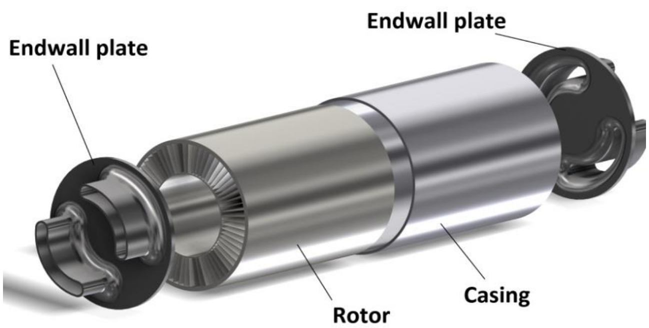


Fig. 1 *Four-port wave rotor schematic configuration*

The flow pattern inside the wave rotor 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 it was concluded in [10].

There are several possibilities to integrate a wave rotor in an existing gas turbine. An optimization study performed in [11] analyzing five different scenarios. The configuration assuming unchanged baseline compressor and turbine inlet temperature provides almost the best gas turbine performance enhancement of the wave rotor topped engine. However, the combustor could work under both higher pressure and temperature at the combustor exit, possibly requiring an enhanced structure as well as a fuel injection system, as stated in [11]. 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 it was concluded in [12].

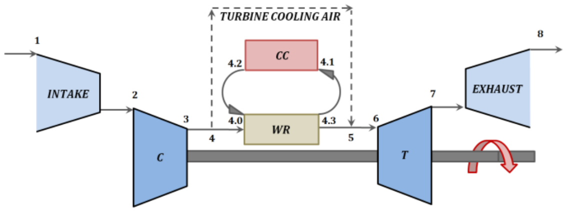
Performance maps of the topped engine were generated in [11] by varying the compressor pressure ratio, and the turbine inlet temperature. The above investigation assumed that the compression efficiency (η_C) and expansion efficiency (η_E) have constant values, namely $\eta_E = \eta_C = 0.83$. These values were also applied in [13] and in [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 reveals the gas turbine performance in terms of each individual wave rotor characteristic.

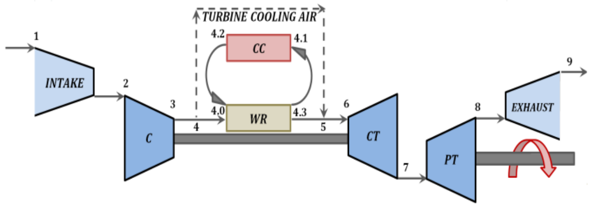
II. Gas Turbine Cycle Calculations

A. *Input data for one and two-sfaft turboshaft engines*

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 [14]. It is based on standard thermodynamic analysis of gas turbines, described in [16] and [17], adding the compression and expansion processes inside the wave rotor. Fig. 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.



(a)



(b)

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*

Constant values are assumed for the air and for the exhaust gases as: $C_{pc} = 1005\ J/KgK$, $\gamma_c = 1.4$ $C_{ph} = 1150\ J/KgK$, $\gamma_h = 1.333$. Table 1 summarizes typical values of input data used Inlet Temperature, (*TIT*) and the compressor pressure ratio (R_c) .

B. *Input data for wave rotor*

Typical input data for wave rotor thermodynamic calculations are summarized in Table 2. The wave rotor parameters chosen to be varied are the wave rotor pressure ratio, *PR*, the ducting and leakage losses (ΔP_{duct}) and compression and expansion efficiencies, η_C , η_E.

TABLE I Baseline Engine typical Input Data

Quantity	Symbol, Unit	Value
Ambient pressure	P_a , *KPa*	101.3
Ambient temperature	T_a , *K*	288
Intake pressure losses	$\Delta P_{in}(\%)$	1
Compressor pressure ratio	R_c	5 ÷ 30
Combustion chamber pressure losses	$\Delta P_{cc}(\%)$	5
Fuel Low Calorific Value	*FCV*, MJ/Kg	48.6
Turbine Inlet Temperature	*TIT*, K	900 ÷ 1600
Isentropic compressor efficiency	η_{isc}	0.85
Combustion chamber efficiency	η_{cc}	0.99
Isentropic turbine efficiency	η_{ist}	0.90

TABLE II WAVE ROTOR TYPICAL INPUT DATA

Quantity	Symbol, Unit	Value
Wave rotor pressure ratio	*PR*	1.4 ÷ 2.2
Ducting and leakage losses	$\Delta P_{duct}\,(\%)$	8 ÷ 16
Efficiency of compression processes inside the wave rotor	n_C	0.75 ÷ 0.92
Efficiency of expansion processes inside the wave rotor	n_E	0.75 ÷ 0.92

Fig. 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).

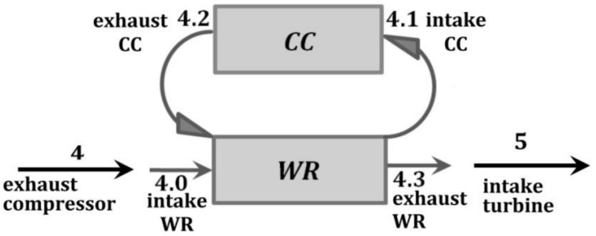


Fig. 3 *Symbols used for the four-port wave rotor thermodynamic calculations*

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. 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:

$$PR = \frac{P_{4.1}}{P_{4.0}} \tag{1}$$

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, $T_{4.0}$

$$T_{4.0} = T_{04} \tag{2}$$

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

$$T_{4.3} = TIT \tag{3}$$

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

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

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, $T_{4.1}$

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

where η_C is the compression efficiency inside the wave rotor

Stagnation pressure at the combustion chamber outlet $T_{4.2}$

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

where η_E is the expansion efficiency inside the wave rotor

Fig. 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 Fatsis (2016), 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*. 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.

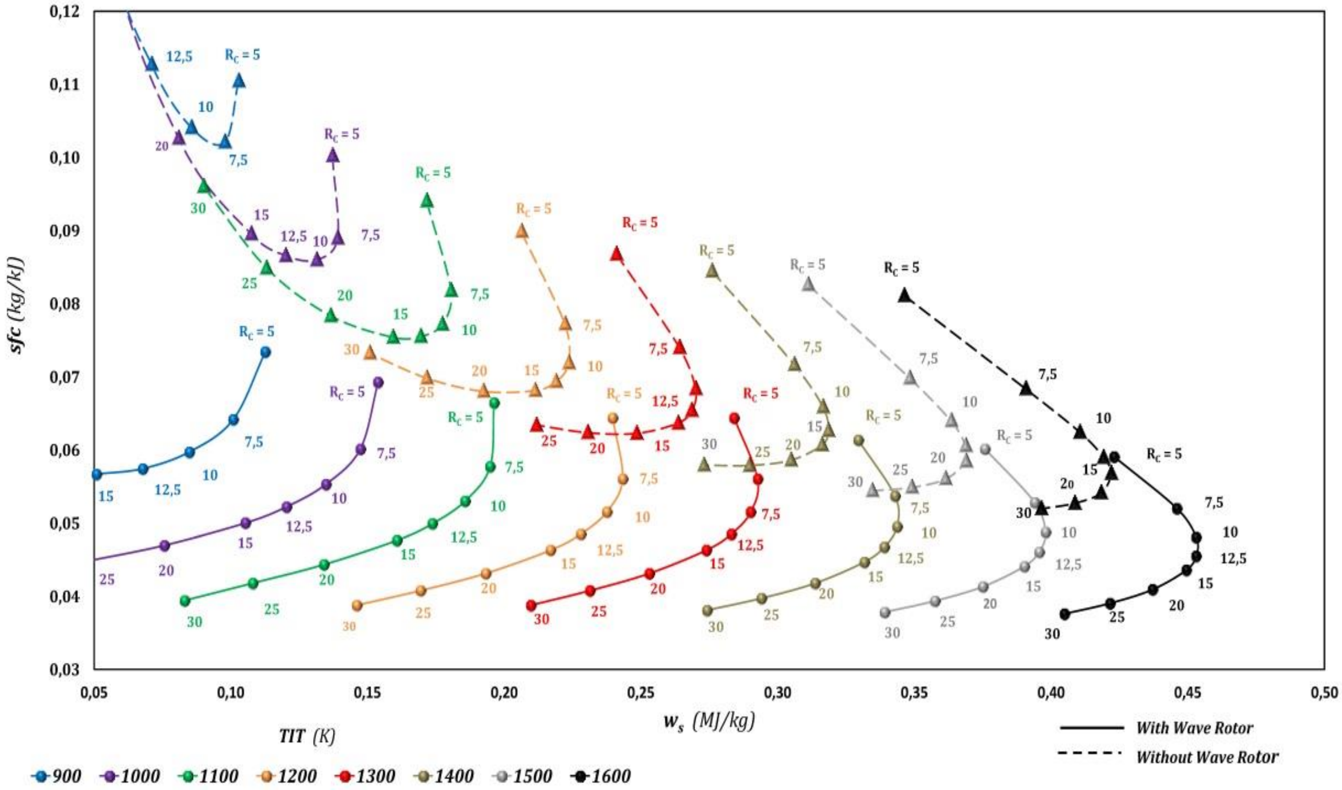


Fig. 4 *Performance of baseline and wave rotor topped one-shaft gas turbines at design point*

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.
3. Pressure losses variation at the ducts connecting the wave rotor to compressor, combustion chamber and turbine, $\left(\Delta P_{duct}\right)$. It is chosen between 8% and 16% while keeping the rest parameters of both gas turbine and wave rotor constant.

C. Effect of wave rotor pressure variation

Numerical and experimental studies carried out in the past in [5], [12] and [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.

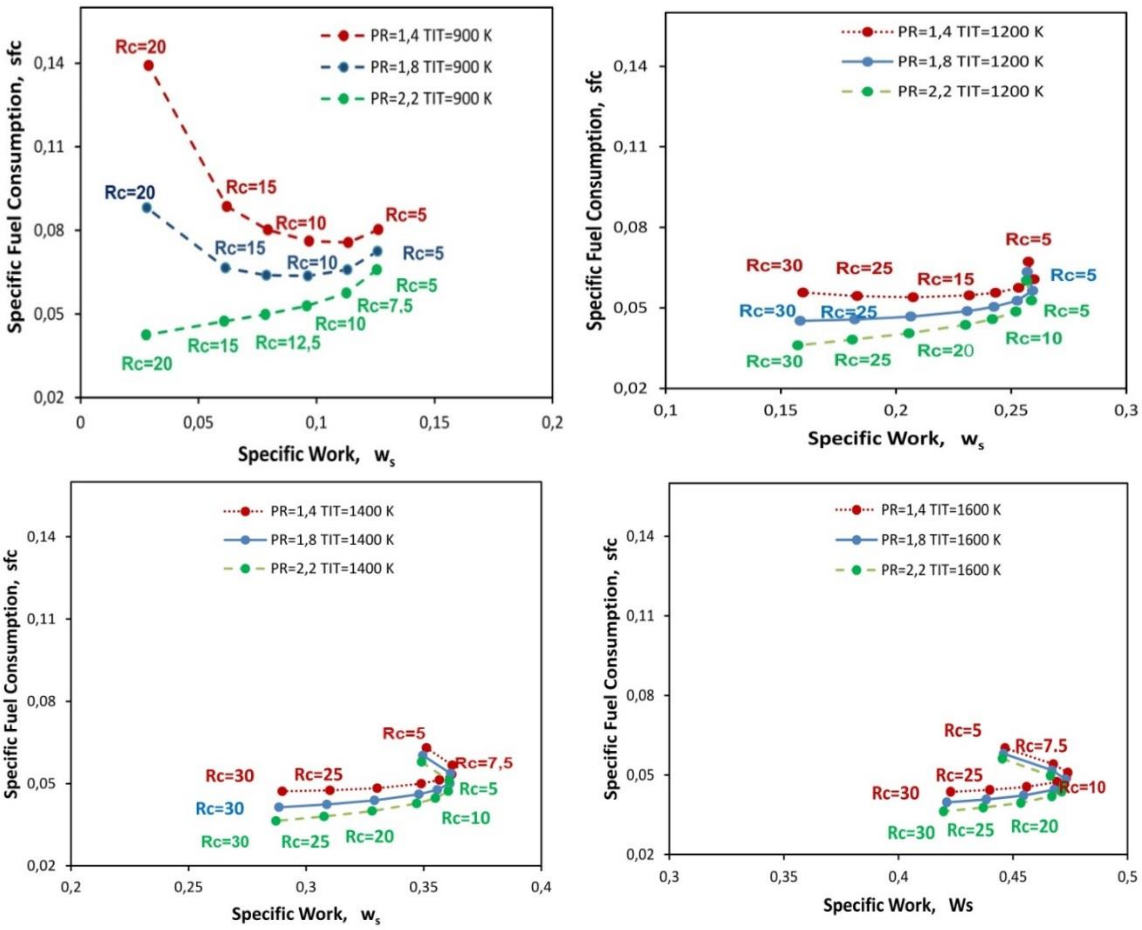


Fig. 5 *Performance of one-shaft gas turbines topped with four-port wave rotor with η_C=η_E=0.83 and variation of PR from 1.4 to 2.2 and TIT from 900 K to 1600 K*

D. *Effect of compression and epansion 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*

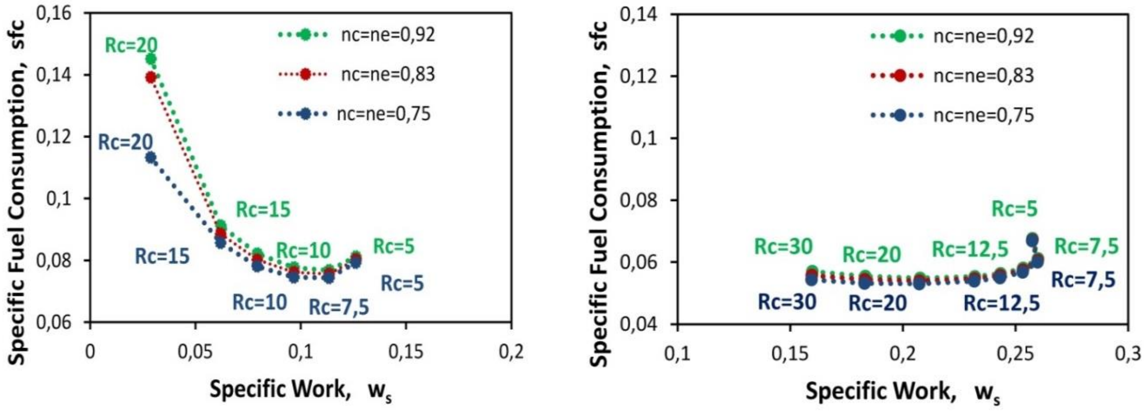


Fig. 6 *Performance of one-shaft gas turbines topped with four-port wave rotor with PR=1.4 and η_E ,η_C varying from 0.75 to 0.92 for TIT=900 K (left) and TIT=1200 K (right)*

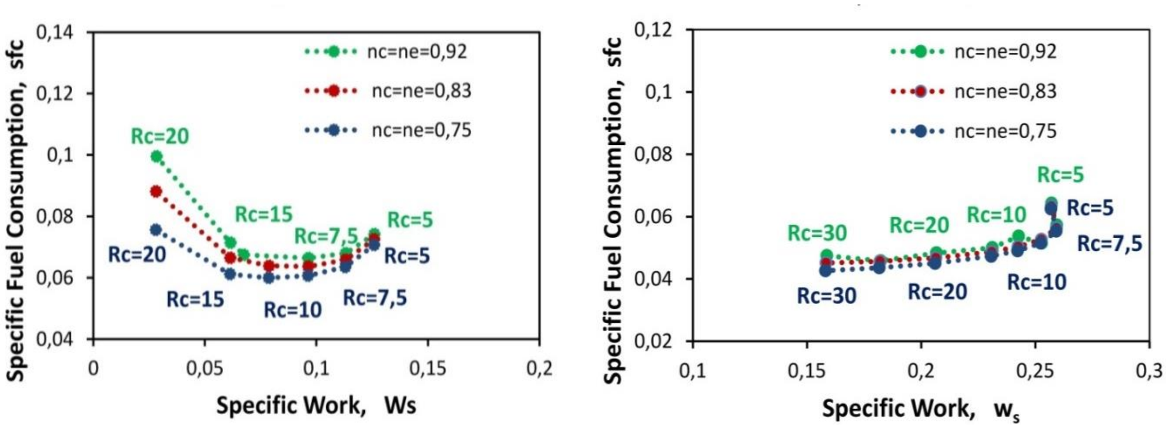


Fig. 7 *Performance of one-shaft gas turbines topped with four-port wave rotor with PR=1.8 and η_E ,η_C varying from 0.75 to 0.92 for TIT=900 K (left) and TIT=1200 K (right)*

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.

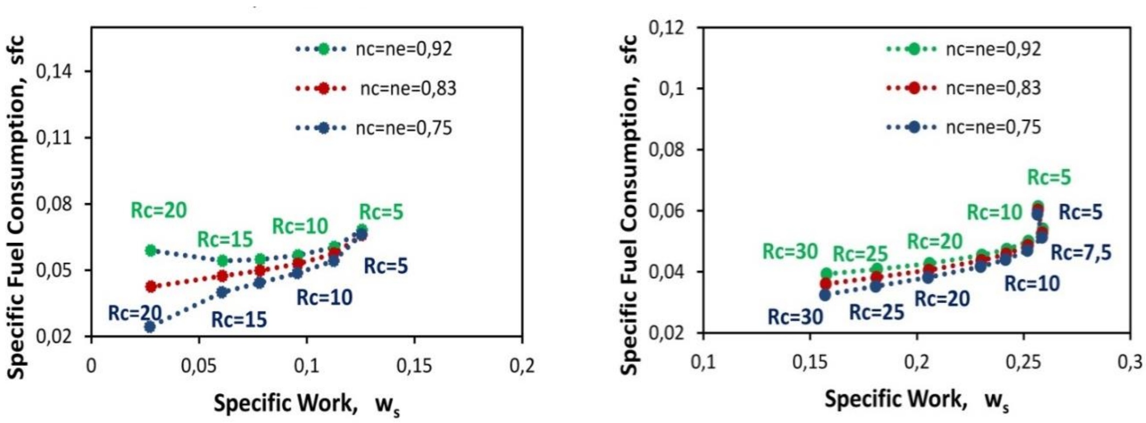


Fig. 8 *Performance of one-shaft gas turbines topped with four-port wave rotor with PR=2.2 and η_E ,η_C varying from 0.75 to 0.92 for TIT=900 K (left) and TIT=1200 K (right)*

E. Effect of 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 (ΔP_{duct}), was analyzed in [18] and in [13]. Fig. 9 illustrates the combined effect of both ΔP_{duct} and *PR* variation on engine's performance for a constant value of *TIT*=900 K. The cases of $\Delta P_{duct} = 8\%$ (dotted lines) and $\Delta P_{duct} = 16\%$ (continuous lines) for *PR*=1.4, 1.8, 2.2 are illustrated. It can be seen that when the pressure losses increase, the specific fuel consumption increases and the specific work decreases, at a constant values of *TIT and of* η_C, η_E

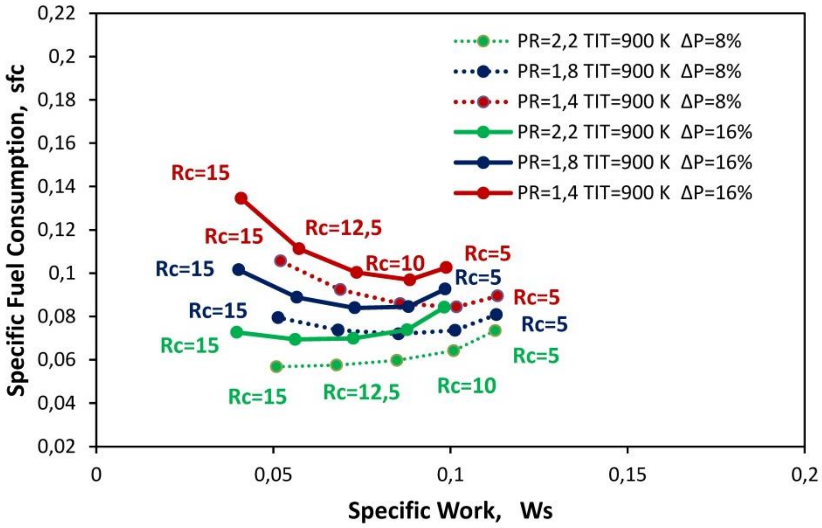


Fig. 9 *Performance of one-shaft gas turbines topped with four-port wave rotor with TIT=900 K,* $\eta_C = \eta_E = 0.83$ *and variation of PR and variation of ducting and leakage losses associated with the wave rotor*

III. CONCLUSIONS

In this article the influence of the wave rotor characteristic parameters on one and two-shaft industrial gas turbines is investigated. 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. 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 *R*c 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, 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.

Variation of the leakage and pressure losses affects also the engine's performance. The influence is noticeable at low values of *Rc* and low values of *TIT*. When ΔP_{duct} is increasing and *PR* is decreasing, the engine's *sfc* is increasing and *ws* is decreasing.

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