ANALYSIS OF A COMBINED BRAYTON/RANKINE CYCLE WITH TWO REGENERATORS IN PARALLEL

J. T. dos Santos[a],
T. M. Fagundes[a],
E. D. dos Santos[b],
L. A. Isoldi[b],
and L. A. O. Rocha[c]

[a] Universidade Federal do Rio Grande do Sul,
Escola de Engenharia
Porto Alegre, RS, Brasil
[b] Universidade Federal de Rio Grande,
Escola de Engenharia
Rio Grande, RS, Brasil
[c] Universidade do Vale do Rio dos Sinos,
Programa de Pós-Graduação em Engenharia
Mecânica, São Leopoldo, RS, Brasil
luizor@unisinos.br

Received: May 25, 2017
Revised: June 19, 2017
Accepted: July 20, 2017

ABSTRACT

This work presents a configuration of two regenerators in parallel for a power generation Brayton/Rankine cycle where the output power is 10 MW. The working fluids considered for the Brayton and Rankine cycles are air and water, respectively. The addition of a regenerator with the previous existing cycle of this kind resulted in the addition of a second-stage turbine in the Rankine cycle of reheat. The objective of this modification is to increase the thermal efficiency of the combined cycle. In order to examine the efficiency of the new configuration, it is performed a thermodynamic modelling and numerical simulations for both cases: a regular Brayton/Rankine cycle and the one with the proposed changes. At the end of the simulations, the two cycles are compared, and it is seen that the new configuration reaches a 0.9% higher efficiency. In addition, the vapor quality at the exit of the higher turbine is higher, reducing the required mass flow rate in 14%.

Keywords: Brayton/Rankine cycle, thermal efficiency, parallel configuration, regenerators

NOMENCLATURE

$\dot{m}$	mass flow rate, kg/s
T	temperature, K
p	pressure, kPa
x	vapor quality
k	polytrophic coefficient
W	work, kJ
Q	heat, kJ
h	specific enthalpy, kJ/kg

Greek symbols

Η	efficiency

Subscripts

c	cycle
t	thermal
s	ideal
w	water
a	air
b	pump
n-m	process n-m
X	state X

INTRODUCTION

In the current Brazilian energetic matrix, power-generating cycles, which transform the heat resultant of fossil fuels into electric energy, play a relevant part, because, even though the society claims for cleaner energy production methods, these cycles present a strategic alternative supply for the drought season. In addition, this method provides a low initial cost and relatively low implementation costs.

Within this context, seeking for alternative to improve the efficiency of these cycles is extremely important, not only in an economic point of view, reducing the fuel consumption, but also in an ambient-friendly part, reducing the emission of pollutants to the atmosphere.

Considering this necessity of a more efficient use of the energetic resources available, this study aims to model and numerically simulate a combined Brayton/Rankine power generation cycle with an alternative configuration of the heat regenerators, adding a regenerator in parallel and a second-stage turbine in the combine cycle.

The study of combined Brayton/Rankine cycle with the use of regenerators has been a point of focus for some researchers. Bejan et al. (2012) showed that the path to improve the efficiency of regenerator, aiming the maximum heat transfer and minimum head loss, was to turn the current parallel channel structure into more complex structures in the form of dendritic channels, with the tree shaped configuration being a tendency of evolution for those systems.

Can Gülen e Raub W. Smith (2009) described a simple, but very precise method, to estimate how much power can be extracted from a lower Rankine cycle given an exergy output by a gas turbine (higher Brayton cycle). The method is based in the Second Law of Thermodynamics and, according to the authors, considering the technical and economic restrictions, the model can deliver the maximum power that can be extracted by the lower cycle to any gas turbine by just considering the output exergy.

Habib and Zubair (1992) examined the performance of regenerative/reheat Rankine power generating plants through an exergetic analysis. His results indicate that most losses occurred in the boiler and that these losses could be minimized by heating the feeding water, reducing the irreversibility of the cycle in 18%.

Akiba et al. (1993) created three types of computer software to calculate the thermodynamic properties of working fluids and evaluate the performance of the combined Brayton/Rankine cycle. Several parameters were varied in order to analyze their effects over the performance of the combine cycle in power plants. According to the authors, the numerical results were very compatible to the data of active power plants.

Khaliq and Kaushik (2004) applied the Second Law of Thermodynamics in a combined Brayton/Rankine cycle with reheat of the gas turbine. It was investigated the effects of pressure, temperature, number of reheats and pressure loss. It was noticed that the combustion chamber accounted for more than 50% of the total exergy loss. In addition, the efficiency of the cycle and the maximum output power were found in a medium pressure and were improved significantly when two reheat stages were present. However, for more than two reheat stages, the efficiency and output power did not raise significantly.

Franco and Casarosa (2004) evaluated the feasibility of high efficiency combined cycles. The results showed that it is possible to obtain power plants with efficiency higher than 62%. In order to do that, it would be necessary to make appropriate use of the existing technologies, without the need of waiting for new technological development of turbines.

Alabdoadaim et al. (2004) investigated the performance of a big cycle composed of a Brayton, two reverse parallel Brayton and a Rankine cycle. The expansion pressure relation between the two reverse Brayton cycles was varied, considering values above the atmospheric pressure. Results showed that the best thermal efficiencies occurred for higher expansion pressures, reaching values of approximately 54%.

Zhang et al. (2012) proposed a regenerative Brayton combined with two reverse Brayton cycles in parallel, with the regeneration occurring before the reverse cycles. Through the analysis of the First Law of Thermodynamics, it was found that the cycle could reach a higher thermal efficiency when compared to that of a common cycle, but with a lower specific work. The optimal efficiency found for this system was of 51.2%.

Gomez et al. (2014) presented a power generation plant in which a closed Brayton cycled was combined in parallel with a vapor Rankine cycle, exploiting the cold exergy available due to the regasification process of liquid natural gas. The cold exergy was used to bring the helium used in the Brayton cycle to cryogenic temperatures in the compressor entrance and to generate electrical power through direct expansion. Energetic and exergetic analysis showed that it is possible to obtain high power efficiency in the power plant, given the right conditions.

Considering this necessity of a more efficient usage of the available energetic resources, this study proposed an alternative configuration of heat regenerators in a combined Brayton/Rankine cycle, where a regenerator in parallel is added to the system along with a second stage turbine.

In order to analyze the performance of this new configuration, a thermodynamic modelling and numerical simulations were performed for two cases: a regular Brayton/Rankine cycle and another one with the proposed modifications. At the end of the simulations, the results obtained for both cycles are compared.

COMBINED BRAYTON/RANKINE CYCLE WITH REGENERATORS IN PARALLEL

In this work, a new configuration for the Brayton/Rankine cycle was proposed. The objective was to increase the efficiency by inserting an extra regenerator and a second stage turbine, using the concept of reheat for the Rankine cycle.

It was proposed that this extra regenerator would be in parallel with the previous existing, as illustrated in Fig. 1. In this case, the hot air output from the gas turbine (in red in Fig. 1) would be divided in two streams at the same temperature: the first passing through regenerator 1 and being used to heat the vapor input for the first stage turbine. This flow was called $\dot{m}_{4a}$. The second flow was used to reheat the vapor output from the first stage turbine through regenerator 2. This flow was called $\dot{m}_{4b}$.

In order to equate the cycle mathematically, the following hypothesis were considered: Steady-state system; Air modelled as an ideal gas with polytrophic coefficient k = 1.4; no head and heat losses; no changes in kinetic and potential energies; Pumps, compressors, turbines and regenerators are adiabatic.

In the formulation of the mathematical model of the different components of the cycle, it was used the approach similar to Moran and Shapiro (2002). In this approach, the volumes of control are defined as well as the energy and mass balance for each process of the cycle, which are:

Process 1-2: Isentropic compression through the compressor.

Process 2-3: Heat input by the heating/combustor heat transfer.

Process 3-4: Isentropic expansion through the turbine.

Process 4-5: Heat rejection through the cooling heat exchanger.

Process 6-7: Heat input through regenerator 1.

Process 7-10: Isentropic expansion through the first stage turbine.

Process 10-11: Heat input through regenerator 2.

Process 11-8: Isentropic expansion through the second stage turbine.

Process 8-9: Heat rejection through the condenser.

Process 9-6: Isentropic compression through the pump.

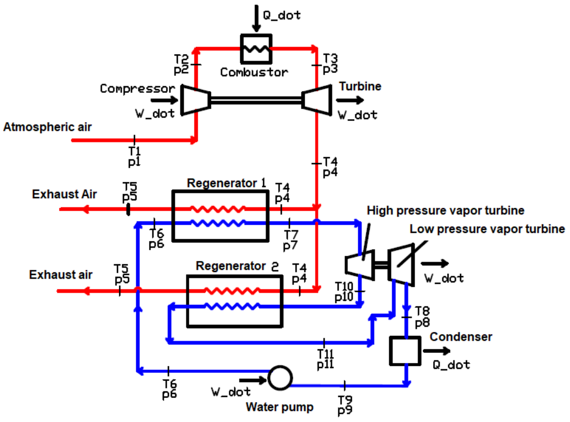

Figure 1. Illustration of the proposed Brayton/Rankine cycle with regenerators in parallel.

The work generated in the compressor in the process 1-2 is presented as

$$\frac{W_c}{\dot{m}_a} = h_2 - h_1 \tag{1}$$

and a correction to a real case, where irreversibility is considered, can be obtained as

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{2}$$

where $\dot{W}c/\dot{m}a$ the power per unit mass consumed by the compressor h_1 and h_2 the enthalpies of air at the inlet and outlet of the compressor and h_{2s} the specific enthalpy at the compressor outlet to the ideal case. In addition, η_c is the isentropic efficiency of the process. It is important to mention at this time that specific power, enthalpy and heat per unit mass are given in kJ/kg, for all the times these properties are mentioned.

For process 2-3, the heat per unit mass delivered to the air

$$\frac{\dot{Q}_{2-3}}{\dot{m}_a} = h_3 - h_2 \tag{3}$$

where h_3 is the specific enthalpy of the standard air leaving the combustor.

For the gas turbine, process 3-4, the power per unit mass generated in the turbine, given, h_{4s} and h_4 the enthalpies of air in the exit of the turbine, for ideal and real cases, respectively, given in and η_{ta} the isentropic efficiency of the turbine.

$$\frac{\dot{W}_{ta}}{\dot{m}_a} = h_3 - h_4 \tag{4}$$

$$\eta_a = \frac{h_3 - h_4}{h_3 - h_{4s}} \tag{5}$$

In the cooling heat exchanger, represented by the process 4-5, the heat per unit mass that is removed from the fluid, in kJ/kg, is given by

$$\frac{\dot{Q}_{sai}}{\dot{m}_a} = h_4 - h_5 \tag{6}$$

where h_5 is the specific enthalpy that leaves the combustor, in kJ/kg.

The volumes of control of the regenerators of the combined cycle were established, with process 4-5 in the air part and processes 6-7 and 10-11 for the water part. By applying mass and energy balances, the air mass flow rates that pass through regenerators 1 and 2, respectively, are obtained as

$$\frac{\dot{m}_w}{\dot{m}_{4a}} = \frac{h_4 - h_5}{h_7 - h_6} \tag{7}$$

$$\frac{\dot{m}_w}{\dot{m}_{4b}} = \frac{h_4 - h_5}{h_{11} - h_{10}} \tag{8}$$

$$\dot{m}_a = \dot{m}_{4a} + \dot{m}_{4b} \tag{9}$$

where $\dot{m}_w$ is the mass flow rate of water, h_5 is the specific enthalpy that leaves the combustor, h_7 and h_{11} are the specific enthalpies of saturated vapor that leave each of the regenerators.

For the process 6-9, Eq. (7) the power per unit mass consumed by the pump, in kJ/kg for an ideal case is given as

$$\frac{\dot{W}_b}{\dot{m}_w} = h_6 - h_9 \tag{10}$$

and the correction to a real case is

$$\eta_b = \frac{h_{6s} - h_9}{h_6 - h_9} \tag{11}$$

Here, h_9 is the specific enthalpy of the saturated liquid in the inlet of the pump and h_{6s} and h_6 are the specific enthalpies of the compressed liquid that leaves the pump, in kJ/kg, for the ideal and real cases, respectively.

For the regenerators, processes 6-7 and 10-11, the heat per unit mass delivered to the fluid is, respectively, as

$$\frac{\dot{Q}_{6-7}}{\dot{m}_w} = h_7 - h_6 \tag{12}$$

$$\frac{\dot{Q}_{10-11}}{\dot{m}_w} = h_{11} - h_{10} \tag{13}$$

where h_7 and h_{11} are the specific enthalpies of saturated vapor that leaves each of the regenerators. For equating purposes, it is considered that the vapor generator is inside the control volume of the regenerator.

In the first stage vapor turbine, disregarding the losses, it is possible to obtain the power per unit mass generated by the turbine as

$$\frac{\dot{W}_{wa}}{\dot{m}_w} = h_7 - h_{10} \tag{14}$$

The real enthalpy of the outlet flow can be obtained if the isentropic efficiency of the turbine is known, through

$$\eta_{twa} = \frac{h_7 - h_{10}}{h_7 - h_{10s}} \tag{15}$$

Here, $\dot{W}$ h_{10} and h_{10s} are the specific enthalpies of the vapor at the outlet of the turbine, for real and ideal cases, respectively. The same can be considered for the second stage turbine,

$$\frac{\dot{W}_{wb}}{\dot{m}_w} = h_8 - h_{11} \tag{16}$$

$$\eta_{twb} = \frac{h_8 - h_{11}}{h_8 - h_{11}} \tag{17}$$

For the condenser (process 8-9), the heat per unit mass absorbed by the condenser can be derived as

$$\frac{\dot{Q}_{8-9}}{\dot{m}_w} = h_8 - h_9 \tag{18}$$

where h_9 is the specific enthalpy of the vapor at the exit of the condenser.

Through the power balance in the three cycles, Brayton, Rankine and Combined cycle, the work from the Brayton cycle (W_a), the Rankine cycle *(W_w)* and the total work (W_t) can be obtained, respectively, by

$$\dot{W}_a = \dot{W}_{ta} - \dot{W}_c \tag{19}$$

$$\dot{W}_w = \dot{W}_{wa} + \dot{W}_{wb} - \dot{W}_b \tag{20}$$

$$\dot{W}_t = \dot{W}_w + \dot{W}_a \tag{21}$$

The ideal thermal efficiency, η_{tcc}, is defined as the total liquid work produced by a determined power cycle divided by the external heat that is delivered to it. Applying this definition to the combined cycle, the objective function, is obtained:

$$\eta_{tcc} = \frac{\dot{W}_t}{\dot{Q}_{2-3}} \tag{22}$$

METHODOLOGY

The methodology used in this work was a thermodynamic modelling coupled with numerical simulation and comparing the results obtained for two systems: a benchmark case, where a regular combined Brayton/Rankine cycle and a second case based in the cycle illustrated in Figure 1.

The equations for both cases are practically the same, apart from the benchmark case only having one regenerator and turbine. Thus, Eqs. (10), (13), (14), (16), (17) and (18) are suppressed.

In order to have a strict comparison between both cases, the input data for both have to be the same, given that a change in the thermodynamic properties or the efficiencies of the cycles would lead to disparities in the comparison, thus nullifying the results.

The algebraic system of non-linear equations were solved using the software Interactive Thermodynamics (IT) and the thermodynamic properties of the fluids involved in the process, standard air and water, were obtained with the software's database.

In the present work, all equations were solved with residual error below 10^{-13}, thus it is possible to affirm that the numerical error did not influence the results obtained.

Opposite to the numerical results, the hypothesis used in the modelling for this work might have more influence over eventual deviation of the results obtained here. However, this is not going to be analyzed in this work.

The thermodynamic properties, as well as the efficiency of the cycle components are shown in Table 1. As previously mentioned, the same values were considered for both systems in such a way that the only points with a degree of freedom in the cycle with two regenerators were in points 10 and 11, the exact points where the cycles are different.

Table 1. Thermodynamic properties, efficiencies and total power for the benchmark case (regular cycle).

Parameter	Value
T_1	300
p_1	100
T_3	1,400
p_3	1,200
p_4	100
T_5	480
p_5	100
T_7	673
p_7	8,000
p_8	8
p_9	8
$\dot{W}_t$	100,000
η_c	0.84
η_{ta}	0.88
η_{tw}	0.9
η_b	0.8

However, the post-reheat temperature, T_{11}, was considered to be the same 673 K, the same temperature as T_7 (the temperature of the fluid leaving the regenerator for the benchmark case). This was necessary in order to consider that the regenerator from both cases would have the same capacity.

After these considerations, the only thermodynamic property from the system that remained free to analyze its influence in the proposed system was the pressure from point 10, the intermediate pressure between the two stages of the turbine. It is important to reiterate that the pressure from points 10 and 11 are considered the same.

By modeling the cycle with regenerators in parallel, the pressure in point 10 was changed from 400 to 1500 kPa, with intervals of 100 kPa. With this, the responses of the cycle for this range of pressure were obtained. These values were compared to the one obtained in the benchmark case. In addition, it is important to restate that this pressure range was chosen because the vapor quality of the fluid at the first stage turbine of the low cycle always remained in the saturation region.

Pressure values above 1500 kPa (superheated vapor) were also analyzed. However, since the efficiency of the system did not increase considerably and the vapor quality of the fluid at the exit of the second stage decrease abruptly, with values below 90%, these results were suppressed from this text.

ANALYSIS AND RESULTS

Using the previously described methodology and model, all interest parameters were obtained for the studied cycles. These parameters are: efficiency, mass flow rates, turbines powers, outlet vapor turbine vapor qualities and external heat delivered to the system.

The results obtained for the relevant parameters to the benchmark case are shown in Tab. 2.

Table 2. Relevant parameters obtained for the benchmark case.

Parameter	Value
$\dot{m}_a$	239.8
$\dot{m}_w$	30.44
η_{tcc}	0.4939
$\dot{Q}_{23}$	202,500
$\dot{W}_a$	6.89E+04
$\dot{W}_w$	3.11E+04
x_8	0.8034

The results obtained for the cycle with regenerators in parallel can be seen through the graphs of Figs. 2-6.

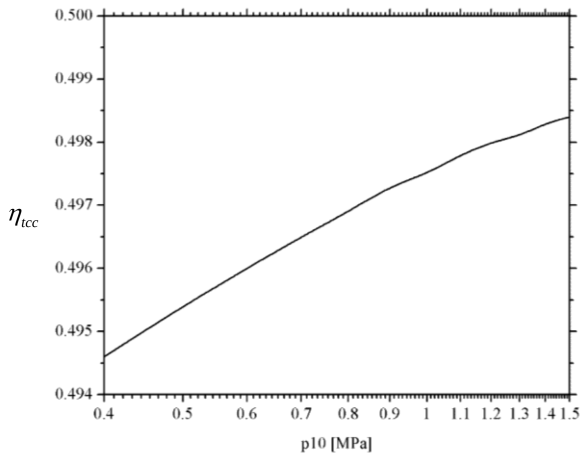

Figure 2. Effect of the pressure of point 10 over the ideal thermal efficiency of the cycle with two regenerators in parallel.

Looking at Fig. 2, it is seen that as the pressure of point 10 increases, there is an increase of thermal efficiency in the combined cycle, starting with 49.46% for a 400 kPa pressure and reaching 49.84% for 1500 kPa. In addition, by comparing the results from Fig. 2 to the benchmark case, it is seen that the proposed cycle reaches higher efficiencies, with a thermal efficiency increasing up to 0.45%. Looking in other perspective, this means that if the benchmark case is turned into a cycle with regenerators in parallel, its efficiency would increase by roughly 1%.

The results shown in Figure 3 reveals that, for the proposed cycle, a lower mass flow rate of air was obtained, especially for pressure values around 1500 kPa for point 10.

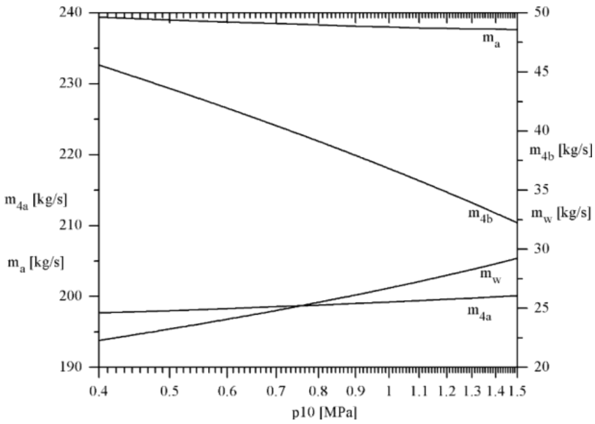

Figure 3. Variation of the mass flow rates of the working fluids regarding the pressure of point 10, for the proposed cycle.

By looking at Figure 3, it is possible to notice that, as pressure p_{10} increases, the mass flow rate of air that goes through regenerator 1 also increased, while reducing the mass flow rate that goes through regenerator 2. This result is comprehensible, since as the intermediate pressure between the two stages turbine (p_{10}) increases, less energy is needed in the second regenerator in order to change the working fluid up to the properties considered in point 11.

It is possible to notice that the exceeding energy that did not pass through the second regenerator is utilized mainly in the first regenerator, where a mass flow rate of compressed liquid (increasing with the increase of p_{10}) has to be transformed into superheated vapor.

Still regarding the results in Figure 3, it is seen that the mass flow rate of vapor increases steadily with the increase of p_{10}. However, when compared to the benchmark case, even for the highest value of p_{10} studied, this mass flow rate was reduced significantly for the new cycle, being, at the worst case, 14.3% lower than the mass flow rate for the benchmark case. This reduction brings a benefit of having more compact installations.

By analyzing the results Figure 3, combined with Fig 4, it is seen that the reduction of the total mass flow rate was consequence of the reduction of the external heat given to the cycle, as expected. This result is shown in Figure 4.

If it is considered that the rate of mixture for combustion (ar and fuel) are the same for both cases, this decrease of 0.9% found for the total air flow and for the heat delivered to the cycle would account for an equal reduction for the fuel consumption. Furthermore, for a power plant active for more than 20 years, a great amount of fuel could be saved, thus the importance of this result has a big impact economically.

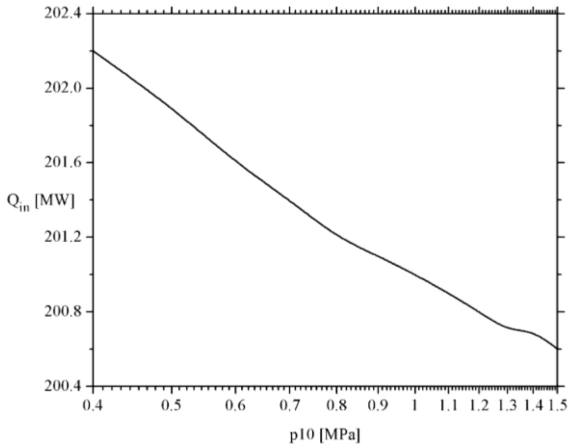

Figure 4. Change in the external heat given to the cycle in function of the pressure on point 10 for the studied cycle.

By looking at Figure 5, it is seen that the resulting vapor qualities for the proposed cycle were more adequate than the benchmark case. This improvement was expected, since it was added an extra regenerator, and consequently a new reheat process to the system.

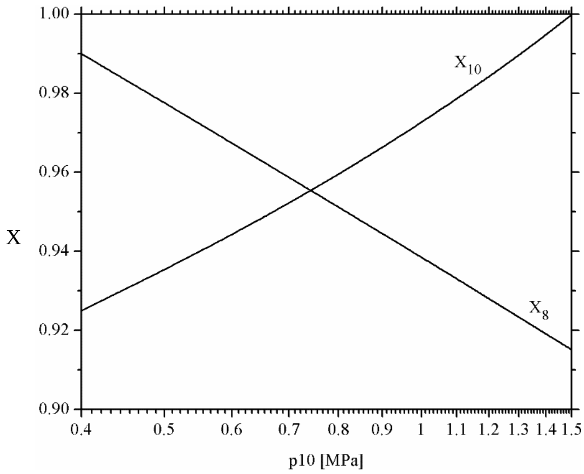

Figure 5. Results for the vapor quality at the exit of the turbines for different values of pressure on point 10.

There is an increase in the vapor quality, going from 80.34% of the benchmark case, for a minimum of 91.51% for the proposed cycle. This would mean a lower wear of the vapor turbines, since they are projected to work mainly with vapor, being damaged by liquid fluids.

Finally, the net amount of work produced for the turbines of the system were analyzed. Results are shown in Figure 6.

By analyzing Figure 6, it was noticed that the power generated by the gas turbine was reduced. But, since the system efficiency increased, these results was already expected, because all the heat coming into the system from external sources do so through the higher cycle. In addition, since the thermodynamic properties of the air were fixed, a smaller amount of air and, thus, a lower power is generated by it.

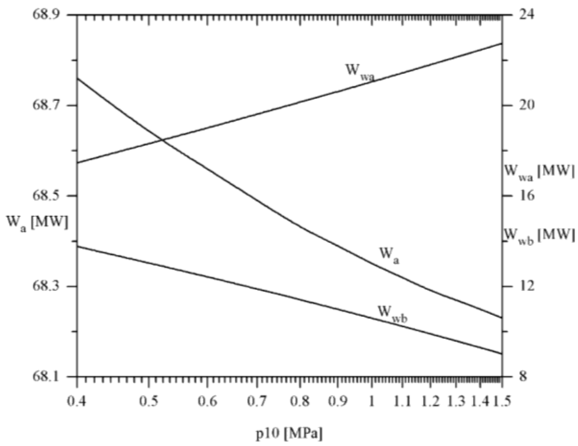

Figure 6. Influence of pressure p_{10} over the liquid power of the turbines for the proposed cycle.

It is also noticed that the turbines of the Rankine cycle made up for the power reduction in the gas turbine. This is mainly due to the second stage turbine, where the power produced increased as the pressure level increased. This result was also expected, since by increasing p_{10} it is also reduced the energy removed from the working fluid in the first stage turbine and, consequently, increased the amount of energy available to be extracted by the second stage turbine.

CONCLUSIONS

Through the present study, it is concluded that the application of regenerators in parallel in combined Brayton/Rankine cycles can account for several benefits. The main advantage is the improvement of the cycle efficiency, which would reduce significantly the energy consumption, resulting in less fuel consumption while also reducing the pollution that comes from the combustion of fossil fuels.

In addition, the vapor mass flow rate is reduced in 14.3%. This opens the possibility of having more compact and cheaper systems, since every component in the cycle could be reduced, for a given power demand. On the other hand, for the same installation, more power could be extracted from the cycle.

The utilization of an extra regenerator benefited the reheating process of the Rankine cycle. This proposition showed to be interesting in the aspect of reducing the wear of the vapor turbines, because the vapor quality at the exit of the turbines was higher. In this case, the turbines could work in a system with less liquid.

As the main disadvantages of applying the cycle proposed in this work in a power generation plant are: the introduction of a second stage turbine, coupled with automation systems that would control the air flows and a possible increase in the air and vapor piping.

It is also shown in this work that the usage of regenerators in parallel is a good alternative to increase the thermal efficiency of power plants. However, an exergetic and economic analysis would be required to analyze, in a wider form, if the application of the proposed system is advantageous. These analyses are going to be addressed in future works.

ACKNOWLEDGEMENTS

The authors thank the Conselho Nacional de Desenvolvimento Científico e Tecnológico (CNPq) and the Comissão de Aperfeiçoamento de Pessoal de Nível Superior (CAPES) for the financial support, as well as the universities UFRGS and FURG that helped in the making of this work.

REFERENCES

Alabdoadaim M. A., Agnew B., and Alaktiwi A, 2004, Examination of the Performance Envelope of Combined Rankine, Brayton and Two Parallel Inverse Brayton Cycles, Journal of Power and Energy, Vol. 218, pp. 377-385.

Bejan, A., Lorente, S., and Kang, D., 2012, Constructal Design of Regenerators, International Journal of Energy Research, Vol. 37, pp 1509-1518.

Can Gülen S., and Smith, R. W., 2009, Second Law Efficiency of the Rankine Bottoming Cycle of a Combined Cycle Power Plant, Journal of Engineering for Gas Turbines and Power, Vol. 132, pp. 1017-1-27.

Franco, A., and Casarosa, C., 2004, Thermoeconomic Evaluation of the Feasibility of Highly Efficient Combined Cycle Power Plants, Energy, Vol. 29, pp. 1963-1982.

Gómez, M. R., Garcia, R. F., Gómez, R. R., and Carila, J. C., 2014, Thermodynamic Analysis of a Brayton Cycle and Rankine Cycle Arranged in Series Exploiting the Cold Exergy of LNG (Liquefied Natural Gas), Energy, Vol. 66, pp. 927-937.

Habib M. A., and Zubair S. M., 1992, Second-Law-Based Thermodynamic Analysis of Regenerative-Reheat Rankine-Cycle Power Plants, Energy, Vol. 17, pp. 295-301.

Khaliq A., and Kaushik S. C., 2004, Second-Law Based Thermodynamis Analysis of Brayton/Rankine Combined Power Cycle with Reheat, Applied Energy, Vol. 78, pp. 179-197.

Masashi, A., Emad A., and Masao, T., 1993, A Study of High-Accuracy Calculations of Combined Brayton/Rankine Cycles for Power Generation, JSME International Journal, Serie B, Vol. 36, No. 1, pp. 178-183.

Moran, M. J., and Shapiro, H., 2002, *Fundamentals of Engineering Thermodynamics*, 4th ed., John Wiley & Sons.

Zhang Z., Chen L., and Sun F., 2012, Energy Performance Optimization of Combined Brayton and Two Parallel Inverse Brayton Cycles with Regeneration before the Inverse Cycles, Scientia Iranica, Transactions B: Mechanical Engineering 19, pp. 1279-1287.