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THERMODYNAMIC BEHAVIOUR OF REGENERATIVE CYCLES AND ITS CONTRIBUTION TO PARABOLIC TROUGH POWER PLANTS PERFORMANCE

André Felippe Vieira da Cunha
Naum Fraidenraich
Universidade Federal de Pernambuco - UFPE, Departamento de Energia Nuclear - DEN, Fontes Alternativas de energia (FAE), Av. Prof. Luiz Freire, 1000, 50740-540, Recife (PE), Brasil , Tel./Fax: (+55) 81 2126-8252 / 2126-8251
andre.fvcunha@ufpe.br;

Abstract. *Heliothermal systems for power generation are being built in many countries all around the world, with the leadership of Spain and the United States. In those systems, water circulates and is heated in the absorber of the cylindrical-parabolic collectors' field and transform in superheating vapor. A thermodynamic Rankine cycle completes the process by converting the thermal energy of the steam into electricity. It is, therefore, a direct heat generating system. One possibility to increase the efficiency of the solar-electric plant is to improve the Rankine cycle performance. The regenerative cycle with multiple extractions in the turbine increases the efficiency making fully or partially adiabatic the preheating phase of the cycle. This paper provides an overview of how the thermodynamic variables of the regenerative system behave. It can be seen that the turbine extraction pressures tend to follow a polynomial function, as well as the enthalpy difference between two consecutive extraction exchangers is not constant, as predicted by simplified treatments reported in the literature. The efficiency of the conventional Rankine cycle, heat to electricity increases from 39% to 43%. Further gains can be obtained by introducing other improvements in the cycle.*

Keywords: *Thermal Cycle, Solar Energy, Regenerative Cycle*

1. INTRODUCTION

Solar plants of cylindrical parabolic collectors is a worldwide established technology. By the year 2013, around 3000 peak electrical MWs were already installed in Spain and the United States. Simulation of those power plants performance is a fundamental tool for the conceptual understanding of its operational behavior, experimental results and improvement possibilities. Developed by the National Renewable Energy (NREL) of the United States, the Solar Advisor Model (SAM) is available to obtain yearly simulations of those plants (NREL, Parabolic Trough Solar Power Network). The industry has also developed proprietary models to evaluate parabolic trough plants.

Numerical results obtained with an analytical model (Rolim *et al.* 2009) show that nominal overall efficiency (η_{ov}) of solar thermoelectric plants, given by the product of the collector's field and thermodynamic cycle efficiencies, are equal to 0.262, when built with evacuated absorber, 0.243, with non-evacuated absorber and 0.219 for bare absorber (without glass envelope). Those values depend on the evaporation temperature of the power cycle and are obtained for temperatures which maximize the overall efficiency (320, 310 and 300 ^oC, respectively).

The analysis leading to those figures considers optical and thermal losses. Other losses like those resulting from sun width increase, tracking errors and absorber misalignment, micro and macro irregularities of the reflector surface contribute to lower those efficiencies. Also, the power cycle is severely affected by the non-isentropic efficiency reducing moreover the overall efficiency of the system. With the opposite tendency, the quality of materials used in solar power plants, reflectors, absorbers, improved structures suggest that, if feasible, slight improvements will be obtained at a very slow time rate.

We analyze the thermodynamic cycle (Rankine cycle) associated to the collector's field, looking for a way of improving its efficiency (η_{el}). There are various possibilities to approximate the Rankine to a Carnot cycle. We explore the benefits of a regenerative cycle with various stages. Isentropic efficiency of the thermodynamic cycle obtained with commercial plants will also be evaluated and added to the model (Robert, 2001).

This work establishes a theoretical basis of the behavior of efficiency and turbine work of a regenerative cycle depending on the number of turbine extractions. The analysis considers: a) Variation of efficiency and work cycle depending on the number of stages, evaporator temperature and overheating; b) The relationship between the evaporation temperature and the optimum performance of the cycle. It is expected that this work can provide a broad understanding of the thermodynamic system behavior and yield useful information for thermoelectric solar systems design with direct steam generation in the absorber region (direct vapor generation-DGV).

2. REGENERATIVE CYCLES AND ITS CONTRIBUTION TO SOLAR POWER PLANTS PERFORMANCE

The regenerative cycle is a derivation of the Rankine cycle, providing the preheating of the fluid after leaving the condenser. It is obtained through heat exchange with small fractions of vapor released by the turbine along the expansion process, preventing the irreversibility associated with the exchange of energy with external sources of heat. As a result the average temperature of the heat source is increased, which in turn translates into an increase of the thermodynamic efficiency of the cycle. Fig. 1 illustrates a regenerative cycle with "*n*" partial extractions of steam along the turbine, connected to the feed water heaters of direct contact (mixture heaters). The system is composed by the field of solar collectors, turbine, condenser, mixture heaters of direct contact and pumps.

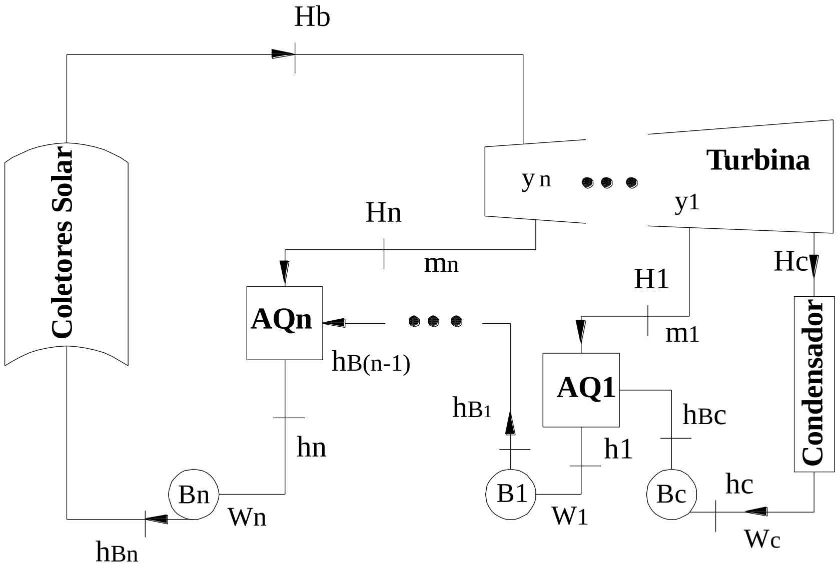

Figure 1. Regenerative cycle with multiple extractions.

Fig. 2 shows a (T-S) diagram of the regenerative thermodynamic cycle behavior. In this diagram it can be seen that the turbine exit has an enthalpy value (*Hc*) of the two phase fluid (liquid + vapor). It enters the condenser to come out at a completely saturated condition (h_c), which is the case in a conventional Rankine cycle. In the case of a regenerative cycle preheating with steam extraction increases the enthalpy of the condensed fluid from (h_c) up to (h_n), at the inlet of the collector field. The collector field will still preheat the fluid up to (h_b), enthalpy under saturated conditions.

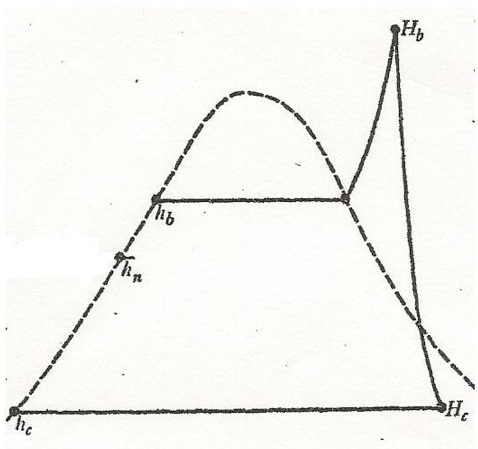

Figure 2. Representation of an Enthalpy versus Temperature diagram for the regenerative cycle.

These regenerative cycles can easily be used in solar thermal power plants with direct steam generation. The thermal energy of the steam which exits the field of solar collectors, is converted into mechanical energy (turbine) and power (generator). The regeneration process avoids the low temperature water entering the field collectors, improving its operating conditions and the overall efficiency of the cycle. The temperature of the water that enters the collector field is of the order of 200 °C.

In this work we carry out an analysis of a regenerative cycle with several extractions of turbine, describing what happens with the turbine work and cycle efficiency as a function of the number *(n)* of mixing heaters, evaporation temperature, superheating temperature in the collectors. At first a brief revision of Haywood's theory is made.

2.1 Review of Haywood's theory

In a regenerative cycle with superheated steam, the irreversibility of mixing heaters derives from the mixture of a subcooled liquid with a two-phase, saturated or superheated fluid. According to Bejan (1988), due to these irreversibilities, the efficiency of the regenerative cycle depends on the mixing heaters distribution, particularly the temperature difference between two adjoining heaters. Haywood (1949) states that the enthalpy difference at the output of a mixing heater and the adjacent one must be constant to obtain a maximum productivity. Haywood's theory, in spite of being only approximate, its simplicity and capacity to illustrate the main aspects of the problem makes it worth as a useful introduction to the subject.

The problem of determining the optimal allocation of a finite number of stages is an important topic on power plant projects with steam turbine. Souza (1980) conducted one graphical performance analysis of this regenerative cycle with several extractions aiming at a "Carnotization" of the Rankine cycle. In his analysis, he notes that after 3 mixture heaters a small increase in cycle efficiency is obtained.

Haywood (1949) conducts a general analysis of the regenerative cycle with infinite direct contact exchangers. The mass flow running through the condenser (*Wc*) is considered as unit (1.0) and thus the energy balance in the first exchanger can be written as

$$m_1(H_1 - h_1) = 1 \cdot (h_1 - h_c) \tag{1}$$

Where (m_1) is the mass flow in the first extraction and (h_c) is the enthalpy at the condenser outlet. Small letters (h_i) represent enthalpy of saturated liquid at contact exchanger output and (H_i) the enthalpy in each turbine extraction.

In his work Haywood (1949) shows that for the condition of maximum efficiency, the enthalpy difference at the output of two consecutive exchangers ($r_i = h_i - h_{i-1}$) and the difference between the incoming and output enthalpies ($t_i = H_i - h_i$) should be constant (with values r and t). Thus, from mass conservation in exchangers, similarly to Eq (1) for the first stage, the mass flow (W_i) at the outlet of each heat exchanger is given by:

$$W_i = \left(1 + \frac{R}{n \cdot t}\right)^i \tag{2}$$

where (R) is the enthalpy difference between the condenser outlet and the last heat exchanger, i.e. ($R = h_n - h_c$) and (n) is the number of heaters of the regenerative cycle. Since the value of r is constant it follows that ($r=R/n$). The fraction (x) of regenerative preheating per unit amount of total preheating is

$$x = \frac{h_n - h_c}{h_b - h_c} = \frac{R}{R_b} \tag{3}$$

A graphical notion of the fraction of enthalpy increase is given by Fig. 2, where it is noted that the difference between (h_c) and (h_n) is obtained by steam extraction at the turbine stages and the difference $(h_b$-$h_n)$ is completed inside the collector's field, close to saturation condition.

For Haywood (1949), the maximum efficiency of the cycle for a number *(n)* of heat exchangers is given by the preheating fraction

$$x_{\max} = \frac{n}{n+1} \tag{4}$$

In general, the cycle efficiency (η) can be calculated as the ratio of useful work and the heat output from the field of collectors, ie:

$$\eta = \frac{Q_{ther} - Q_{cond}}{Q_{ther}} = \frac{W_n(H_b - h_n) - (H_c - h_c)}{W_n(H_b - h_n)} = 1 - \frac{(H_c - h_c)}{W_n(H_b - h_n)} \tag{5}$$

Introducing in Eq. (5) the hypotheses that the enthalpy difference (t), between heat exchanger inlet and outlet, is constant and that the mass flow at the last stage (n) can be obtained from Eq. (2), substituting *(i)* by *(n)*, it is found

$$\eta = 1 - \frac{t}{\left(1 + \frac{R}{nt}\right)^{n} \left(H_b - h_n\right)} \tag{6}$$

Combining equation (3)-(4) and replacing the difference ($H_b - h_n$) by known values (t, n and R_b) it is found the following expression for the optimum efficiency of the regenerative cycle with n stages

$$\eta_n = 1 - \frac{1}{\left(1 + \frac{R_b}{(n+1)t}\right)^{n+1}} \tag{7}$$

2.2 Exact model of regenerative cycle performance with various extractions. Numerical results

Vieira da Cunha *et al.* (2014) described an exact system of equations and numerical calculations of regenerative cycles, having obtained: a)Turbine and pump work; b) Amount of heat required in the collectors' field; c) Heat rejected by the condenser and d) Cycle efficiency. The same methodology was used in this work for a wider analysis of the cycle with an arbitrary number of extractions. Behavior of some parameters of the regenerative cycle in function of evaporation temperature of the collectors' field (*Tev*) and the number of turbine extractions (n) have already been shown in Vieira da Cunha et al. (2014) and will only be reviewed in this article, including new temperature analysis at the entrance of the collectors and the parameters t ($t_i = H_i - h_i$) and r ($r_i = h_i - h_{i-1}$). In the model adopted by Vieira da Cunha *et al.* (2014), it is considered isentropic expansion turbine and pump work for the calculation of efficiency. The overheating temperature was considered 400 °C and the condensing pressure of 10 kPa. It is noteworthy that the fluid used is water that has a critical temperature of about 374 ^{o}C.

Figure 3 shows the results of the maximum efficiency of the cycle as a function of the evaporation temperature (*Tev*) and the number of turbine extractions (extraction up to 04), obtained with the simulated model, and compares those results with a conventional Rankine cycle. Unlike Vieira da Cunha *et al.* (2014), this work presents additional simulations. Note again that cycle efficiency is increased when the number of heaters (or turbine extractions) increases. This increased efficiency with increasing amount of heat exchangers (n) used is decreased considerably. The trend shows that the maximum cycle efficiency increases with the evaporation temperature. Vieira da Cunha *et al.* (2014) have demonstrated that the increase in maximum efficiency of the cycle also occurs with the superheating temperature at the exit of the collectors of the cycle. It is noteworthy that as you increase the number of heaters, simulation time performed with the software "Engineering Equation Solver (EES)" increases considerably, allowing to actual date the analysis up to 04 heaters and, in this case, evaporation temperature goes up to 250 ° C.

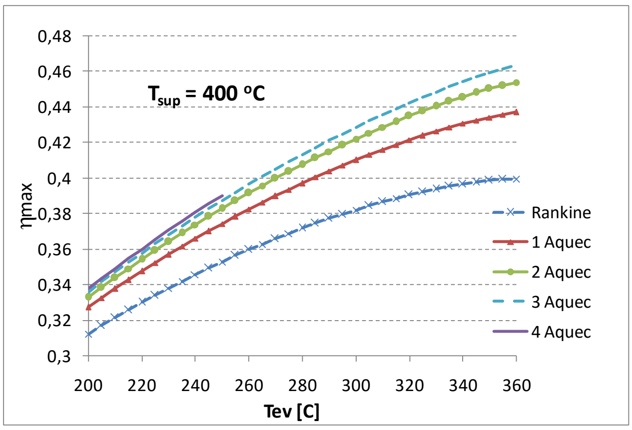

Figure 3. Behavior of maximum efficiency versus the evaporation temperature and the number of mixture heaters.

The maximum turbine work performance (w_{tmax}) as a function of the evaporation temperature and the number of heaters is shown in Fig. 4 (The regenerative cycle can have many values of work in the same evaporating temperature, but it is obtained the maximum value). These results are compared with the conventional Rankine cycle and, for each number of heaters, it is observed a maximum work at a given evaporation temperature. The same behavior was observed in Vieira da Cunha *et al.* (2011) and depends on the amount of heaters. The value of this work (w_{tmax}) also increases with superheating temperature, as shown in Vieira da Cunha *et al.* (2012).

The work for a given evaporation temperature and heat input lowers as the number of heaters increases. This occurs because heat is extracted from the expanding vapor in turbines, the more as the number of heater increases. Nonetheless efficiency improves as Fig. 3 shows, which in fact is the objective of the regenerative cycle.

The presence of a maximum work can be explained as follows. Work can be considered the product of evaporation heat H_{ev} multiplied by Carnot efficiency (η_C). For the equilibrium curve of water vapor, as the evaporation temperature T_{ev} increases the evaporation heat (H_{ev}) decreases and the Carnot cycle efficiency increases. At a given T_{ev} there is a maximum.

It is also noted that temperature of maximum work decreases as we increase the number of heaters, as observed in Fig. 4.

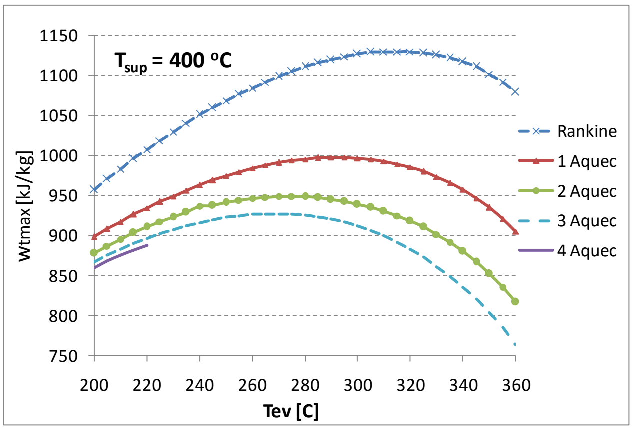

Figure 4. The turbine work behavior with the evaporation temperature and the number of mixture heaters.

Fig. 5 shows the behavior of the maximum efficiency achieved with a regenerative cycle with a certain number of heaters, using superheating temperature of 400 °C and the evaporation temperature of 310 ° C (close value of maximum work). It is observed, as in Fig. 3, that increasing the number of heaters increases the efficiency of the cycle, but this increase isn´t significant as reached four (4) heaters and thus economically impracticable for cycles with more than five heaters.

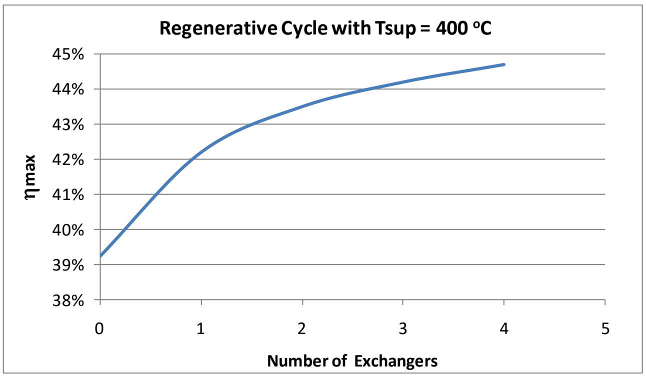

Figure 5. Behavior of (maximum) efficiency with the increase in the number of heaters for *Tev* = 310 ° C.

Considering that the mixtures occurring in the heaters are reversible and there isn't heat exchange with the environment, it can be determined which entry temperature of the collectors' field should be reach. Thus, assuming the second law for a volume control and the index *i* is input and the index *o* is out of this volume, it can be established the following equation (Van Wylen, 1993):

$$\sum \dot{m}_i s_i + \sum \dot{m}_o s_o = 0. \tag{8}$$

where *s* is specific entropy and $\dot{m}$ the mass flow.

Fig. 6 shows the temperature profile in the input field of the collectors (T_{B3}) according to the evaporation temperature (Tev) on a regenerative cycle with 3 mixture heaters. It is observed that the behavior of this T_{B3} when we increase temperature Tev is almost linear. The graph also shows the temperature at the condenser outlet, which is about 45 °C. For Tev = 300 ^{o}C, for example, the graph shows that the entrance temperature at the collector's field should be 225 ^{o}C, temperature that increases as evaporation temperature increases. The consequences of this behavior for the collector's field performance and the overall system efficiency will be analyzed in a future work.

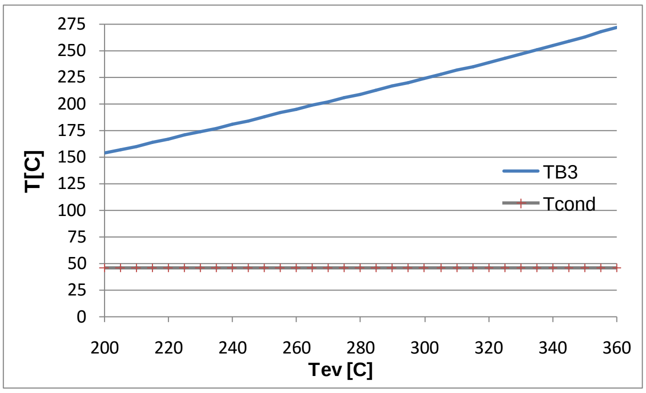

Figure 6. Temperature behavior at the entrance of collectors field as a function of the evaporation temperature.

3. CONCLUSIONS

This article presents the behavior of a regenerative cycle and its contribution in solar plants with parabolic cylindrical collectors system. It was observed that the efficiency of the regenerative cycle is greater than the Rankine cycle and this efficiency increases with the increase of the evaporation temperature, heater numbers and superheating temperature. It was observed that the efficiency does not increase significantly when the number of heaters exceed 04 (or 4 turbine extractions). The temperature entering the field of collectors shows a significant increase in regenerative cycles as compared with the Rankine cycle. The consequences of this behavior for the collector's field performance and the overall system efficiency will be analyzed in a future work.

4. ACKNOWLEDGEMENTS

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5. REFERENCES

Bejan, A., 1988, "Advanced Engineering Thermodynamics", Ed. John Wiley & Sons, United States of America.

Haywood, R. W. ,1949, "A Generalized Analysis of the Regenerative Steam Cycle for Finite Number of heaters", Proc. Inst. Mech. Eng., Vol. 161, pp. 157-162.

Robert C., 2001, Solar Trough Generation - The California Experience. Presented at ASES FORUM, Washington DC.

Rolim, Milton Matos; Fraidenraich, Naum; Tiba, Chigueru, 2009, Analytic modeling of solar power plant with parabolic linear collectors, *Solar Energy*, vol. 61, pp. 126-133.

Van Wylen, G. J. and Sonntag, R. E., 1993, "Fundamentos da Termodinâmica Clássica", 3^{o} edição, Ed. Edgard Blucher, São Paulo, SP.

Vieira da Cunha, A. F.; Fraidenraich, N. and Vilela, O. C., 2011, "Endoreversible Analysis of a Power Cycle – Comparison with Rankine Cycle", 21st Brazilian Congress of Mechanical Engineering, Natal, RN, Brazil.

Vieira da Cunha, A. F. and Fraidenraich, N., 2012, "Análise do Rendimento Ótimo de um Ciclo Regenerativo com Uma, Duas e Três Extrações da Turbina", Congresso Nacional de Engenharia Mecânica, São Luís, MA, Brasil.

Vieira da Cunha, A., Fraidenraich, N. and Souza Filho, R., 2014, Rendimento de Ciclos Regenerativos com Várias Extrações da Turbina, VIII Congresso Nacional de Engenharia Mecânica, 10 a 15 de agosto de 2014, Uberlândia - Minas Gerais.

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