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EXPERIMENTAL STUDY AND SIMULATION OF ORGANIC RANKINE CYCLE WITH R123 AS THE WORKING FLUID

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Abstract. *The consumption from all energy sources in the last 55 years has increased by 275% worldwide, 494% in South and Central America and 1150% in Brazil. From the total amount of the energy consumed around the world, 80% is of fossil origin (oil, natural gas and coal). In Brazil, on the other hand, 80% of the total energy consumed is renewable whereas 61% comes from hydroelectric source, 9% from the wind, other 9% from biomass and 1% from centralized solar source. The wide spread use of fossil fuels is directly related to the emissions of carbon dioxide, the main gas causing the global warming, which gradually increases the average temperature of the planet, as it prevents the rejection of heat accumulated in the earth originated from the incidence of solar radiation. The decrease in the use of fossil energy sources, the increase in the use of renewable energy sources and increase in energy efficiency, from generation to consumption, contributes to the reduction of* CO_2 *emissions and its undesirable effects. Systems that operate according to the Rankine cycle allow the production of electrical energy from thermal sources. Traditionally, water is used as a working fluid and high temperature energy sources, above 370°C, mainly steam boilers. The Organic Rankine Cycle uses organic fluids that have temperatures and evaporation pressures lower than those of water, allowing the use of heat from sources with lower temperatures (between 60 and 250 °C) to generate electricity. In this cycle, the fluid is pressurized by the pump to the evaporator, where it heats at nearly constant pressure and turns into saturated or superheated steam. The steam expands in the expander (turbine, scroll, etc.) producing shaft work and, as a consequence, has its enthalpy decreased. Finally, it is condensed in the condenser at practically constant pressure, returning to the pump. The main objective of this paper is the thermodynamic modeling and the accomplishment of experimental measurements of a system operating according to an organic Rankine cycle (ORC) with R123. The thermodynamic behavior of the cycle was computationally simulated through the Ideal Rankine Cycle model, recording the points between each cycle process. Results were obtained for operating pressures from 171.5kPa to 361.6kPa, vapor temperatures from 68°C to 75°C and mass flow rates from 11.25kg/h to 30.7kg/h.*
Palavras-chave: *Organic Rankine Cycle, Renewable Energy, Alternative Fuels, R123, Experimental.*

1. INTRODUCTION

The world population and world production of goods is growing and will continue to grow for decades to come (International Energy Agency, 2016). This will lead to a greater need for energy and produce greater emissions of CO2 into the atmosphere, which is considered to be the biggest cause of global warming. Given this situation, it is desirable to develop technologies and actions that can minimize the use of fossil fuels and increase the share of renewable sources in the world energy matrix.

The Organic Rankine Cycle is a promising technology in this scenario. The Rankine Cycle commonly uses water as the working fluid in steam boilers at temperatures of 370°C or more. The particular case of the Organic Rankine Cycle uses organic fluids, which have lower evaporation temperatures than water, making it possible to generate electricity from thermal energy sources with temperatures from 60°C to 250°C (Viklund and Johansson, 2014) as the geothermal and the solar sources.

The Rankine Cycle works by producing work from the heat transferred between a hotter and colder heat source, using four main components: the pump, the condenser, the evaporator and the expander, which directly influence the performance of the cycle and, consequently, in the generation of energy. The first definition in the operation of the cycle is the type of heat source to be used, which determines the maximum evaporation temperature of the cycle. The second definition is the working fluid that will best adapt to the temperature range of the cycle. The choice of fluid will affect some variables of the cycle, such as evaporation pressure, condensation pressure, equipment dimensions that depend on the specific volume, etc.

Cycle performance also depends on the equipment, with the expander being the most significant. The expander can be of several types: scroll, turbine, various types of compressors connected in reverse. The turbine is usually the most common type.

Depending on the application of the cycle, the objective may be thermal optimization, or exergetic efficiency (Maraver et al, 2014), or even the net power produced. Cycle parameters can be simulated numerically (Wang et al, 2012) and analytically (Li et al, 2016) to achieve these goals, as well as experimentally (Hijriawan et al, 2021).

Generally, numerical simulation methods are related to the numerical solution of conservation of mass and energy thermodynamically from the control volumes. In this method, control volumes are used to represent the various components of the cycle and uniformity in the thermodynamic properties and steady state is assumed. The objective of this approach is to calculate the properties in the thermodynamic states at the various entry and exit points of the components of the cycle and to calculate, from these properties, the performance parameters (heat exchange rates, consumed and generated power, component efficiencies) . This approach has been widely used in ORC research in thermal energy recovery (Zhou et al, 2013), solar energy (Wang et al, 2010) and cogeneration (Anvari et al, 2016) applications.

Analytical models seek to calculate various parameters such as thermal efficiency, heat transfer rates and power consumed and generated in a direct and simplified way, avoiding complex simulation codes and calculating performance curves. These models use control volumes but assume some conditions to simplify the calculations. Such simplifications are usually considering constant thermodynamic properties, or ideal processes, or even a combination of ideal processes, using data obtained in real cycles (Li et al, 2016). Even though they are simpler than numerical models, analytical models are excellent methods for predicting cycle performance.

This research consists of operating a complete experimental bench built to simulate various working conditions of the Rankine cycle varying the temperatures, pressures, flows at the various measurement points and calculate the performance parameters.

2. NUMERICAL MODELING OF THE CYCLE

The equations used in the modeling of the cycle are essencialy applied to the processes occurring in the equipments of the cycle, as follows:

In the pump, it is calculated the work introduced in the cycle:

$$\dot{W} = \dot{m}\,.(i_2 - i_1) = \frac{\dot{m}\,.(i_{2s} - i_1)}{\eta_b} \tag{1}$$

In the evaporator, it is calculated the heat exchanged by the fluid:

$$\dot{Q}_e = \dot{m}\,.(i_3 - i_2) \tag{2}$$

In the expander, it is calculated the mechanical work produced by the cycle:

$$\dot{W}_t = \dot{m}\,.(i_3 - i_{4s}).\eta_t \tag{3}$$

In the condenser, it is calculated the heat exchanged by the fluid with the cold source:

$$\dot{Q}_c = \dot{m}\,.(i_4 - i_1) \tag{4}$$

To calculate the performance of the cycle, it is mainly used the first and second laws of thermodynamics (by exergy). (BRAIMAKIS *et al*., 2015).

The ratio between the net power generated and the power supplied to the cycle, defines the efficiency by the first law of thermodynamics:

$$\eta_{thermal} = (\dot{W}_{turbine} - \dot{W}_{pump})/\dot{Q}_e \tag{5}$$

The Carnot efficiency for a cycle operating between two thermal reservoirs, one of high temperature, T_{hot}, and one of low temperature, T_{cold}, is given by:

$$\eta_{Carnot} = 1 - \frac{T_{cold}}{T_{hot}} \tag{6}$$

LECOMPTE et al. (2015), states that the previous equation is not representative for the calculation of the efficiency in the case of a thermal source of finite capacity, shown in Figure 1, so that the efficiency of the Carnot cycle coupled to this source, recovers a fraction of heat, given by Eq. (7).

$$\eta_{frac,Carnot} = \frac{T_{hot} - T_2}{T_{hot} - T_{cold}} \tag{7}$$

Therefore, the efficiency of the reversible Carnot cycle is:

$$\eta^*_{Carnot} = \eta_{frac,Carnot} . \eta_{Carnot} = \frac{T_{hot} - T_2}{T_{hot} - T_{cold}} . \left(1 - \frac{T_{cold}}{T_2}\right) \tag{8}$$

The maximum theoretical efficiency of the cycle can be obtained by deriving the previous equation with respect to T2 and equating to zero is:

$$\eta^*_{Carnot.máx} = 1 - \sqrt{\frac{T_{cold}}{T_{hot}}} \tag{9}$$

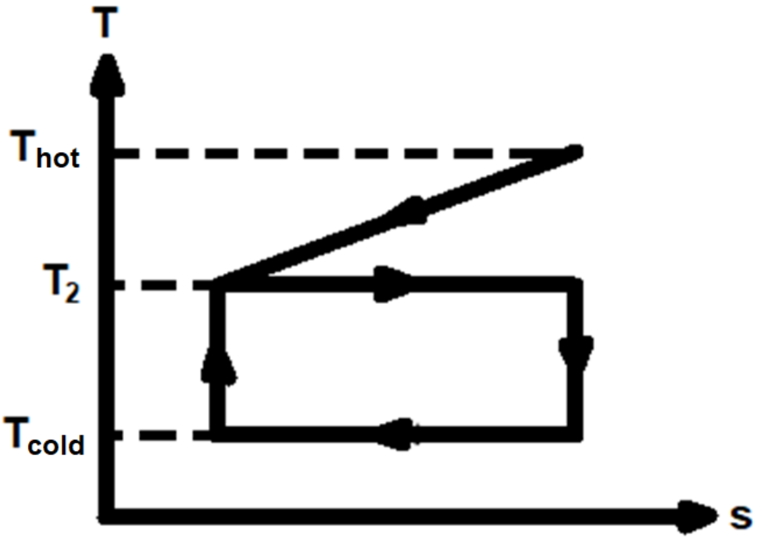

Figure 1. C**arnot cycle with finite capacity source.**

However, thermal efficiency considers only the first law of thermodynamics. Considering the energy balance and not considering the quality of energy and its potential to generate work. Thus, one can adopt the efficiency based on the second law of thermodynamics. The exergetic efficiency is the most suitable parameter to compare the thermodynamic performance of thermal power systems.

The thermal efficiency of the cycle if, then:

$$\epsilon_{exergy} = \frac{recovered\ energy}{supplied\ energy} = \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{\dot{E}_s} \tag{10}$$

$$\dot{E}_s = \left(1 - \frac{T_0}{T_s}\right) . \dot{Q}_e \tag{11}$$

Equation 2.12 combined with equations 2.10 e 2.11 shows that the exergetic efficiency can be treated as proportional to the carnot efficiency, which means that a exergetic efficiency if 100% would occur if the thermal efficiency were equal to the carnot efficiency.

$$\epsilon_{exergy} = \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{\left(1 - \frac{T_0}{T_s}\right) . \dot{Q}_e} = \frac{\eta_{thermal}}{\eta_{Carnot}} \tag{12}$$

Likewise, equation 13 shows a modified Carnot efficiency, so the measure of how close the cycle efficiency is to the modified Carnot efficiency is:

$$\epsilon^*_{exergy} = \frac{\eta_{thermal}}{\eta^*_{Carnot,max}} \tag{13}$$

3. ANALYTICAL MODELLING OF THE CYCLE

The modeling was implemented using the equations of Scagnolatto et al (2021).
The Brayton Cycle efficiency is:

$$\eta_{th,BC,s} = 1 - \frac{T_{cond} - \Delta T_H}{T_{evap} + \Delta T_H} \tag{14}$$

The Saturated Organic Rankine Cycle efficiency is:

$$\eta_{th,STORC,s} = 1 - \frac{\frac{T_{cond}}{T_{evap}} . (1 + Ja_{pre})}{1 + \frac{T_{evap} + T_{cond}}{2T_{evap}} . Ja_{pre}} \tag{15}$$

The Superheated Organic Rankine Cycle efficiency is:

$$\eta_{th,SHORC,s} = \eta_{th,STORC,s} + Ja_{sup} . (\eta_{th,BC,s} - \eta_{th,STORC,s}) \tag{16}$$

Where:

$$Ja_{pre} = \frac{\bar{c}_{p,l} . \left(T_{evap} - T_{cond}\right)}{q_{2-3}} \tag{17}$$

And:

$$Ja_{sup} = \frac{\bar{c}_{p,v,H} . \Delta T_H}{\bar{c}_{p,l} . \left(T_{evap} - T_{cond}\right) + q_{2-3} + \bar{c}_{p,v,H}} . \Delta T_H \tag{18}$$

4. EXPERIMENTAL APPARATUS

Currently, the experimental apparatus is complete and in operating conditions for experimental simulation of points in the process flowchart. One can determine the inlet temperature, inlet pressure and flow, and obtain output values for these variables. The apparatus is comprised by the equipments in Table 1.

Figure 2 shows a Picture of the experimental apparatus and Figure 3 shows a flowchart of the cycle operated by the experimental apparatus.

Figure 2. **Experimental Apparatus.**

Table 1. **Equipments of apparatus**

Equipment	Function
Lung tank	Saturated fluid reservoir that minimizes the effects of volume and pressure variation
Condenser	Ensures that the fluid returning from the expander is condensed to a saturated liquid/vapor state
Sub-cooler	Ensures that the fluid is further subcooled when it enters the pump
Mass flow meter	Measures the mass flow rate of the fluid and records the inlet conditions
Pump	Controls the mass flow rate and fluid pressure
Preheater	Optional feature to increase the cycle's thermal efficiency due to the use of energy from the heat exchange of the fluid itself that returns from the expander
Evaporator	Transmits thermal energy from the hot source to the system, capable of generating saturated steam or superheated steam
Expander	Expands the organic fluid
Cold water tank	Tank with connections to be connected to the sub-cooler and condenser, it has temperature control
Hot water tank	Tank with connections to be connected to the evaporator, having temperature control
Cold water pump	Circulates cold water from the tank to the condenser and sub-cooler
Hot water pump	Circulates hot water from the tank to the evaporator
Valves	Used for system operation and maintenance
Instruments	Connected at various points, monitored and controlled via a computer for system operation

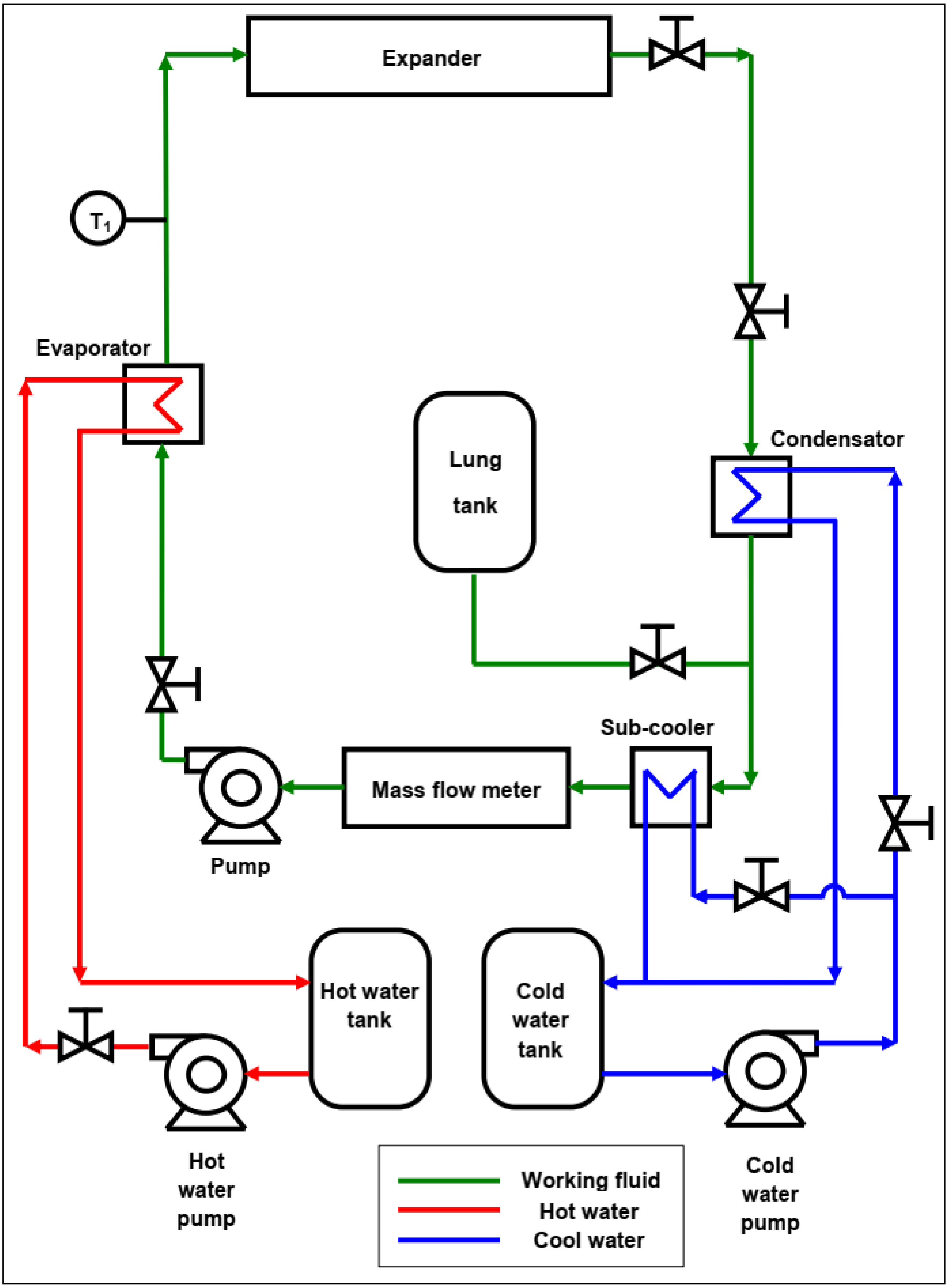

Figure 3. **Experimental apparatus process flowchart.**

5. EXPERIMENTAL RESULTS

The experimental results obtained on the experimental apparatus are showed in Table 2, the experimental data of experiments were used with numerical equations and, after with analytical equations. So, the results were compared.

Table 2. **Tests Results**

Informations	Experiment 1	Experiment 2	Experiment 3
Temp. in the inlet of the expander T_4 [°C]	68	71	75
Pressure in the inlet of the expander P_4 [kPa]	211.7	171.5	361.6
Mass flow rate $\dot{m}$ [kg/h]	11.25	15.0	30.7
Temp. of the hot source T_{hot} [°C]	73	76	80
Temp. of the cold source T_{cold} [°C]	16	17	19
Thermal Efficiency $\eta_{thermal}$	3.45%	2.44%	5.59%
Analytical Thermal Efficiency	3.67%	2.56%	5.89%
Thermal Efficiency Difference	+6.4%	+4.9%	+5.4%
Exergetic Efficiency ϵ^{*}_{exergy}	43.9%	29.6%	70.4%
Analytical Exergetic Efficiency	46.6%	31.2%	74.1%
Exergetic Efficiency Difference	+2.4%	+3.6%	+5.3%

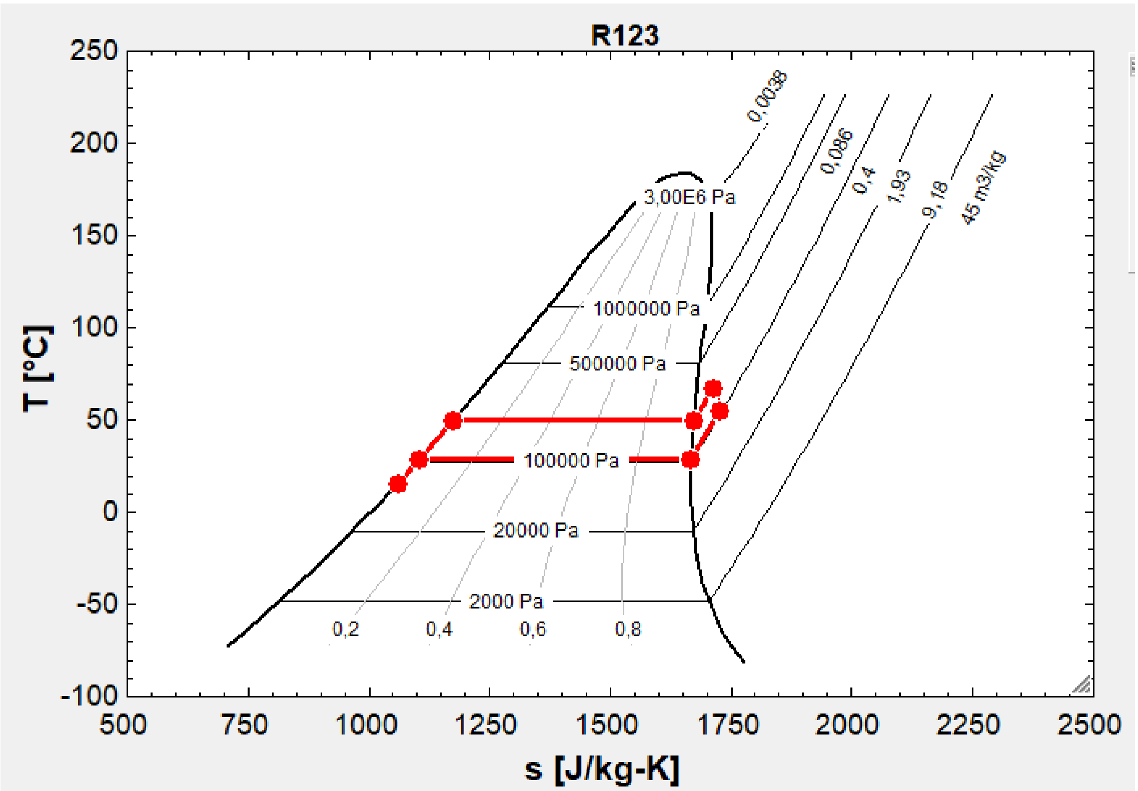

Figure 4. **Thermodynamic Cycle – Experiment 1.**

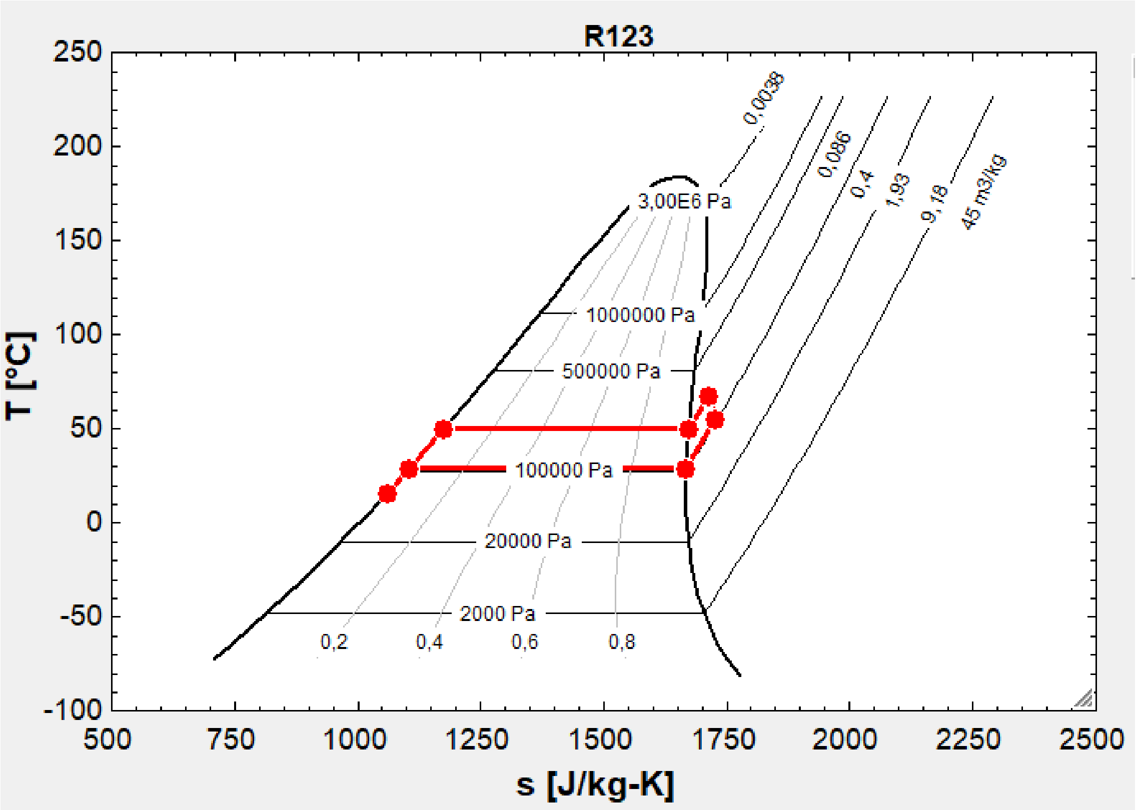

Figure 5. **Thermodynamic Cycle – Experiment 2.**

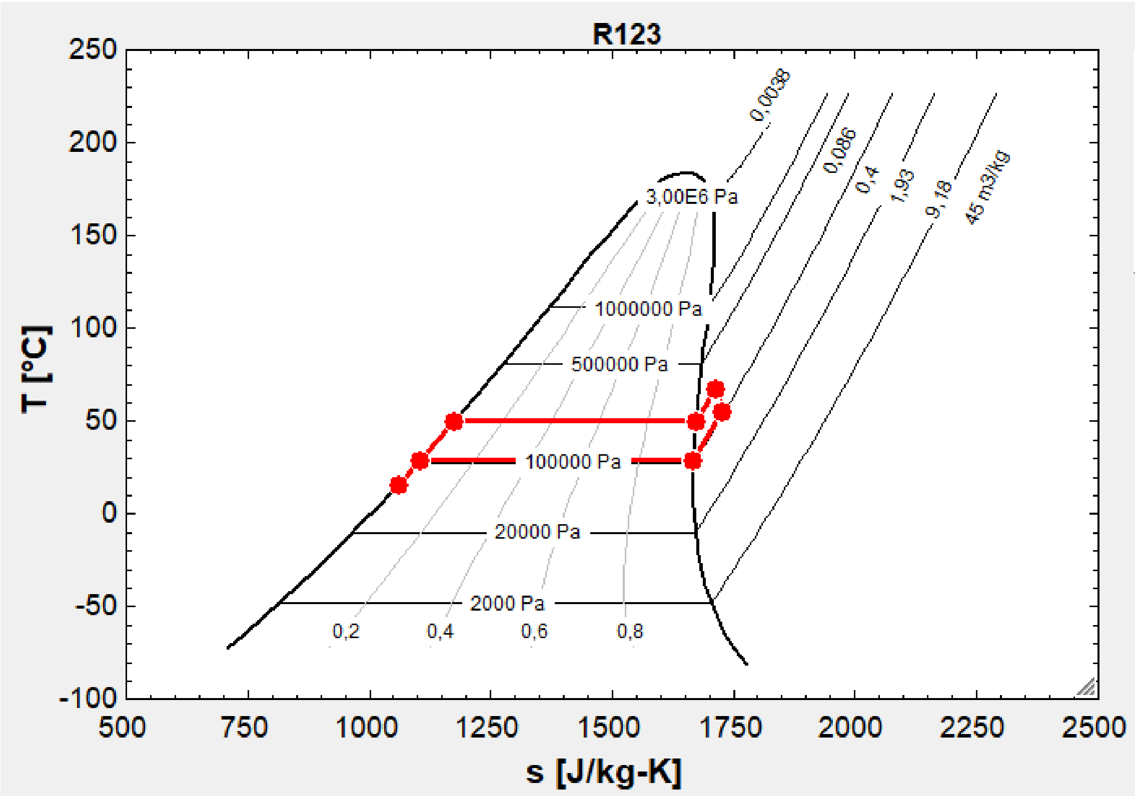

Figure 6. **Thermodynamic Cycle – Experiment 3.**

6. DISCUSSIONS AND CONCLUSIONS

Results were obtained for operating pressures from 171.5kPa to 361.6kPa, vapor temperatures from 68°C to 75°C and mass flow rates from 11.25kg/h to 30.7kg/h. Thermal cycle efficiencies were obtained from 3.45% to 5.59%. The exergetic efficiencies obtained ranged from 41.9% to 70.4%.

The best result was obtained in experiment 3, which has the highest evaporation pressure 361.6kPa. This is due to the larger net work area. On the other hand, the worst result was obtained in experiment 2, where the evaporation pressure 171.5kPa is the lowest and, consequently, the net work area is also smaller.

The evaporation pressure was more significant than the temperature of the superheated steam, since experiment 2 has a higher evaporation temperature than experiment 1, but the efficiency obtained in experiment 1 is higher.

When comparing the thermal efficiency with the analytical thermal efficiency, there was a small difference, with the analytical thermal efficiency being slightly higher in all three experiments, ranging from 4.9% to 6.4%.

The experimental results show that the bench is capable of simulating a good operating range of conditions. It is also noticed that, when analyzing the efficiencies through exergy, that experiment 3 has the best thermal efficiency of 5.59% and also the best exergetic efficiency of 70.4%.

In the case of exergetic efficiency, the analytical model also presented slightly higher results. In any case, the difference was small, from 2.4% to 5.3%.

One factor observed was that the exergetic efficiency is higher when the thermal efficiency is higher and they increase in a similar proportion. It is possible to notice this comparing experiment 1 in relation to experiment 2, where the thermal efficiency is 41.4% higher and the exergetic efficiency is 48.3% higher. The same occurs comparing experiment 3 in relation to experiment 1, where the thermal efficiency is 62.0% higher and the exergetic efficiency is 60.4% higher.

7. ACKNOWLEDGMENTS

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