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Energy and Exergy Analysis of an Organic Rankine Cycle used for Ylang-Ylang Essential Oil Distillery Waste Heat Recovery for Power Production in Anjouan Island

Malik El'Houyoun Ahamadi, and Hery T. Rakotondramiarana

ABSTRACT

In the ylang-ylang essential oil distillers in Anjouan Island, the used energy is 100% firewood biomass. A large amount of this energy is dissipated in the environment just in the combustion chamber itself. As it turns out, the flue gases in this process take away the most part of it. Thus, in a process of energy efficiency of stills, the present work aims at assessing the possibility to convert the residual heat from the process into electricity. For that purpose, energy and exergy modeling of an organic Rankine cycle was implemented. It was found that a large amount of exergy is destroyed in the evaporator. Similarly, it emerges that the exergy efficiency of the cycle depends on the inlet temperatures of the exhaust gases in the evaporator and on the inlet pressure of the working fluid in the turbine, and that it is much better for low exhaust gas temperatures. At these low values of gas temperatures, it appears that the improvement in exergy efficiency and energy efficiency are linked to the increase in the inlet pressure of the working fluid in the turbine. It follows from the obtained results that the discharged hot water and the residual heat of gases having temperatures ranging from 180°C to 300 °C, could be used for power production which can reach electrical powers between 1.4kW and 4.5kW.

Keywords: residual heat recovery, modeling, ylang-ylang essential oil distiller, biomass energy conversion

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I. Introduction

In the Comoros, access to electricity still remains a luxury in the 21st century while the world is talking about energy transition, energy efficiency etc. This problem affects all households in general and rural households in particular. In the industrial sector, each industry tries to adapt in its own way, but small businesses which constitute the majority of this island industries are the most affected by power production insufficiency. The ylang-ylang distillation sector is no exception. It is even one of the most affected. In a previous study on the energy and exergy balance of the essential oil of ylang-ylang traditional distillation in the island of Anjouan in the Comoros, Malik et al. [1] provided a global overview of the entire energy chain from the combustion chamber to the final stage of obtaining the essential oil. The most energy used in the distillation process is biomass, particularly firewood. The study showed that the energy chain of the process is very deficient in energy. Another study conducted by Zamyn [2] raised serious environmental problems linked to this distillation technique. Indeed, the hot exhaust gases injected into the ambient air at high temperatures and the hot water coming from the condenser being directly discharged into nature without any treatment or cooling process, are major environmental issues. Zamyn [2], showed that at high temperatures, these hot fluids unbalance the ecosystem as they not only destroy the surrounding vegetation but also kill and constrain the animals living in this ecosystem to migrate.

Several studies on the recovery of waste heat by Rankine cycle exist in the literature. In their work, Saidur et al. [3], carried out energy, exergy and economic analysis of an industrial boiler. The authors showed that the studied boiler had significant energy losses through the flue gases. The obtained results showed that the recovery of waste heat from the flue gases is one of the most efficient methods to save energy in the boiler. Other authors show that the recovered energy can be converted into electricity to be useful. Maatouk et al. [4] studied waste heat recovery using an organic Rankine cycle (ORC) and a liquid desiccant cooling system (LDCS). It found that for the case of ORC, a decrease in the inlet temperature of the exhaust gases implied an increase in the overall exergy efficiency while a decrease in the outlet temperature of the exhaust gases could result better performance in terms of overall exergy efficiency and cooling production capacity. The study conducted by Leonardo et al.

[5] showed that it would be interesting to control the input temperatures of the source and the turbine to reduce the exergy loss in the cycle and increase its exergy efficiency for Rankine cycles with regeneration. One of the advantages of ORC is that organic fluids have the ability to be evaporated at low temperatures. The performance of ORC varies depending on the technology used in the cycle. The study presented in [6], uses a simple Rankine cycle without a recuperator to recover heat released from the exhaust gases of an automobile engine. As results, it was shown that the maximum rate of the evaporator is 38% for the case studied with an isentropic efficiency of the expansion machine of 60%. The authors proposed an increase of the exchange area of the exchangers and heat transfer coefficients or an improvement in the efficiency of the expansion machine to have an increase in cycle performance. In [7], several thermodynamic cycles were presented for the study of the recovery of waste heat from the exhaust gases of internal combustion engines; it was concluded that, the Rankine cycle has the best performance for heat recovery from exhaust gases. Syed et al. [8] showed that ORC cycles are well suited to heat recovery for low and medium temperatures as they offer significant advantages over conventional steam cycles [9] in [8]. Other studies compare an ORC without a recuperator and an ORC with a recuperator [10, 11]. More precisely, Bouhamady et al. [10] made such comparison for the case of an ORC for a 3kW power solar center (CSP) micro solar power plant. It was found that ORCs with recuperators have a better performance compared to simple ORCs (without recuperator). Indeed, increases of 26.7% and 25% were respectively found for the thermal efficiency and the exergy efficiency of the cycle. On the other hand, while making such comparison for an ORC cycle applied to a biomass power generation system, Ependi and Nur [11] found that adding a recuperator to the cycle increases cycle efficiency and reduces fuel consumption. Collings and Yu [12] showed the interest of using a recuperator in ORCs, to recover the energy that could be rejected in the condenser during condensation and reintroduce it into the cycle. While studying the recovery of waste heat and water by drying the exhaust gases from lignite combustion, Han et al. [13] concluded that ORC is a good solution for the recovery of waste heat as well as water at low temperatures and high humidity concentrations. Dibazar et al. [14] carried out an exergy analysis of three types of ORC for waste heat recovery: basic ORC (BORC), ORC with single regeneration (SRORC) and an ORC with double regeneration (DRORC). It follows from the results of their advanced exergy study of the three ORCs that the ones with regenerators have a strong potential to reduce the irreversibilities of the cycle compared to the BORC. Moreover, cycle enhancements should be given for turbines, evaporators, condensers and hot water supply respectively. Baral [15] made a techno-economic study of a solar-geothermal hybrid ORC cycle by using R134a and R245fa as organic fluids for system performance, and found that the thermal efficiency of the system changes with the temperature of the heat source : the higher the temperature of the hot source, the greater the thermal efficiency of the cycle.

The interest of ORC cycles on power production was highlighted. That result is encouraging for solar-geothermal hybridization. By comparing a steam Rankine cycle (SRC) with an ORC, Satheeshkumar and Lim [16] showed that, for a better efficiency of the waste heat recovery, the ORC must be implemented in the chimney of the studied system. However, the cost analysis of such a project shows that the economic viability of the implementation of an ORC in a gas power plant becomes complicated as it requires a change of the initial thermodynamic assumptions. Lecompte et al. [17] discussed the interest of ORC for its similarities to the good stability of a SRC, with regards to its high efficiency compared to that of SRC for energy conversion with low temperatures and abundance accompanying the operating and maintenance experience. The use of ORC cycle with an electric arc furnace that the aforementioned authors made, allowed them to draw the following conclusions: it is crucial to take into account the variation of the heat profile for a typical batch process in the furnace electric arc. A sub-critical optimization of the ORC system is capable of a net electrical power of 752kWe with steam operating at 25 bar. Eveloy et al. [18] investigated the use of an ORC cycle in an industrial gas turbine to improve plant efficiency and found that the use of the organic fluid R245fa allowed having a surplus of 5.2MW of electricity generated by the installation. The use of this energy recovery system has saved primary energy which could amount to approximately 1.3 million standard cubic feet per day (MMSCFD). Another study [19] made a more detailed analysis of ORC cycles for heat recovery by presenting very detailed mathematical models and optimization models.

It is clear from the analysis of several studies available in the literature on ORC cycles that the ones with recuperators are much more efficient.

Additionally, several works on the choice of working fluids in the case of ORCs are available in the literature. Bouhamady et al. [10], asserted that it is very difficult in practice to find a working fluid which satisfies all environmental conditions. However, one can always find compromises for each application. These authors cited, for example, the studies carried out by Tchanche et al. [20] and Walng et al. [21] who worked on the selection of working fluids for low temperature Rankine cycles and concluded that the organic fluids R134a and R152a, are the best candidates for ORCs operating at low temperature. Besides, another investigation made a detailed review [22] with regard to the choice of the working fluid for the ORCs: Chen et al. [23] studied 35 fluids and pinpointed propane and R134a as good choices for hot cycles at temperatures around 380K. Similarly, Tchanche [24] studied propane as a working fluid. As for Gu et al. [25], they recommend the organic fluid R134a as a good candidate for binary supercritical ORCs in geothermal power plants. Walraven et al. [26] asserted that isobutane, propane and R134a appear as good candidates from the point of view of exergy efficiency and the best from the point of view of energy efficiency for hot springs of a temperature from 398K. Schuster et al. [27] recommended R134a and propane due to their low critical temperatures, for supercritical cycles. Mago et al. [28] recommended propane and isobutane for their energy efficiency.

The cycle that we present in this study differs from most of ORCs encountered in the literature as it includes a preheating exchanger rather than a recuperator. For cycles with recuperators, the recuperator is used to recover the thermal energy remaining in the working fluid after being expanded in the turbine. More exactly, the present study aims to contribute to the search for passive solutions to the energy efficiency problems raised in [1] on the one hand and the environmental issues raised in [2] on the other hand by recovering the waste heat due to the traditional distillation process of the essential oil of ylang-ylang for power production using ORC. The recovery takes place at the level of the hot water leaving the distillation condenser and at the level of the fumes. Indeed, recovering the waste heat from the exhaust gases on the one hand will allow us cooling these combustion gases, and on the other hand, the recovery of heat contained in the hot water of the still condenser contributes to its cooling. It was found from our on-field investigations that the combustion gases are much hotter than the hot water coming from the condenser. This justifies our choice, to place the preheating exchanger in hot water and the evaporator in the combustion gases. Moreover, our choice of working fluids to be used in the studied system is based on results obtained by the aforementioned previous works [20-28] reported in the literature. In the present study, the organic fluids R134a and propane (R290) were retained.

II. MATERIALS AND METHODS

A. *System description*

The studied system is shown in Figure 1. The working fluid recovers the energy contained in the combustion gases by means of a heat exchanger. It vaporizes and circulates towards the turbine where, a mechanical energy is produced then converted into electrical energy by a generator. The working fluid leaves the turbine and passes into a condenser where it is condensed and delivered to a regenerator by a pump before being reintroduced into a preheating exchanger and then into the evaporator, and so the cycle continues. The hot sources of this cycle are respectively the flue gases and the coolant (water) of the distillation condenser which leaves the condenser hot.

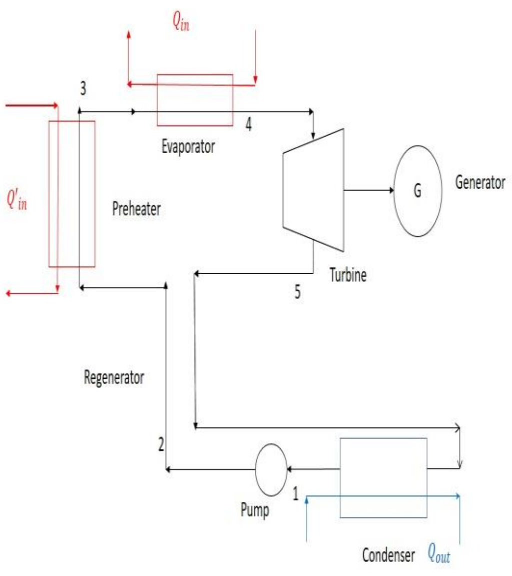

Fig 1. Schematic Diagram for Rankine cycle

B. *Mathematical modelling*

This section presents the mathematical model of the waste heat recovery system during the traditional distillation of ylang-ylang essential oil in Anjouan Comoros.

The model equations are established by taking into account the following assumptions:

1) Heat losses in the various components of the system are neglected
2) The kinetic energy of the various fluids involved in the cycle is conserved. Similarly, the potential energy of gravity is conserved
3) The cycle operates in steady state*Figures*

C. *Energetic study of the ORC cycle*

In steady state, the mass and energy conservation equations are written for each component of the system:

– Mass balance

$$\sum_e \dot{m}_e - \sum_s \dot{m}_s = 0 \quad (1)$$

– Energy balance

It is obtained by using the first principle of thermodynamics [29]:

$$0 = \dot{Q} + \dot{W} + \sum_e \dot{m}_e \left(h_e + \frac{1}{2}v_e^2 + gz_e\right) - \sum_s \dot{m}_s \left(h_s + \frac{1}{2}v_s^2 + gz_s\right) \quad (2)$$

The indices e and s respectively indicate the inflows and outflows in a component i of the complex system.

Respectively, $\dot{W}$and $\dot{Q}$ are the thermal and mechanical powers exchanged with the external environment.

– Power supplied by the pump (compression of the working fluid)

It is defined by:

$$\dot{W}_p = \dot{m}_{fw}(h_1 - h_2).\eta_p \quad (3)$$

Where, $\eta_p$is the isentropic efficiency of the pump, and $\dot{m}_{fw}$ is the mas flow rate of working fluid, h_1 *and* h_2 are respectively the upstream and downstream enthalpies of the pump, where the relation between the both enthalpies is:

$$h_2 = h_1 + v(p_2 - p_1) \quad (4)$$

p_2 *and* p_1 respectively the outlet pump pressure and the inlet pump pressure, v is the specific volume of the working fluid at the pressure p_1.

– Energy recovered in the preheating exchanger

$$\dot{Q}'_{in} = \dot{m}_{fp}(h_{p3} - h_{p2}) = \dot{m}_{fw}(h_3 - h_2) \quad (5)$$

$\dot{m}_{fp}$is the mass flow rate of the preheating fluid, which is the hot water recovered at the outlet of the distillation condenser.

– Heating of the working liquid under pressure from the liquid state to the vapor state

$$\dot{Q}_{evap} = \dot{Q}_{in} = \dot{m}_g(h_{g4} - h_{g3}) = \dot{m}_{fw}(h_4 - h_3) \quad (6)$$

– Mechanical power Supplied by the turbine

$$\dot{W}_t = \dot{m}_{fw}(h_4 - h_{5is}).\eta_t = \dot{m}_{fw}(h_4 - h_5) \quad (7)$$

η_t is the isentropic efficiency of the turbine

– Energy supplied by the condensation of the working fluid in the condenser

$$\dot{Q}_{cond} = \dot{m}_{fw}(h_5 - h_1) = \dot{m}_{fc}(h_{fc1} - h_{fc5}) \quad (8)$$

In equations (2) to (8), $h_i$is the enthalpy of component i expressed by:

$$h_i = C_{pi}\Delta T_i \quad (9)$$

Or got by the thermodynamics table of the working fluids used.

The net power recovered by the cycle is the difference between the power supplied by the turbine and that consumed by the pump:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \tag{10}$$

The thermal efficiency of the ORC cycle is defined by:

$$\eta_C = \frac{\dot{W}_{net}}{\dot{m}_{fw}(h_4 - h_2)} \tag{11}$$

The combination of equations (5) and (6), gives us the expression of the working fluid flow rate us written in equation (12):

$$\dot{m}_{fw} = \frac{\dot{Q'}_{in} + \dot{Q}_{evap}}{h_4 - h_2} \tag{12}$$

D. Exergetic analysis of the ORC cycle

The whole energy contained on the combustion flue gas is not fully converted to useful work. Thus, an exergy analysis enables to assess the potentiality of the cycle to convert the exhaust gas waste heat to an useful energy. An exergy balance of the exhaust gas gives the exergy destroyed as given by equation (13) [1]:

$$\sum_i \dot{m}_{ei}\varepsilon_{ei} - \sum_j \dot{m}_{sj}\varepsilon_{sj} - \sum_k Q_k\left(1 - \frac{T_0}{T}\right) - I = 0 \tag{13}$$

In which,I is the destroyed exergy $\varepsilon_x$the exergy of x component and Q the thermal energy. e and s index represent respectively the inlet and outlet parameters. The exergy is defined by equation (14):

$$\varepsilon = h - h_0 - T_0(S - S_0) + \frac{v^2}{2} + gz \tag{14}$$

Where, the chemical exergy is neglected. h_0, T_0 and S_0 are respectively, the enthalpy, the temperature and the entropy of a reference environment.

In the following equation, we consider that, the whole of the system components are adiabatic. Hence, the term $Q(1 - \frac{T_0}{T})$, is neglected. The kinetic exergy is neglected too.

Destroyed exergy in pump

It is defined by [30]:

$$I_{Pump} = \dot{m}_{fw}T_0(S_2 - S_1) \tag{15}$$

– Destroyed exergy in the evaporator

$$I_{evap} = \dot{m}_g(\varepsilon_{gin} - \varepsilon_{gout}) - \dot{m}_{fw}[(h_4 - h_3) - T_0(S_4 - S_3)] \tag{16}$$

Then: $\varepsilon_{gin} - \varepsilon_{gout} = (h_{g4} - h_{g3}) - T_0(S_{g4} - S_{g3})$ (17)

– Destroyed exergy in preheater

$$I_p = \dot{m}_{fp}\left[\left(h_{p3} - h_{p2}\right) - T_0\left(S_{p3} - S_{p2}\right)\right] - \dot{m}_{fw}[(h_3 - h_2) - T_0(S_3 - S_2)] \tag{18}$$

According, to equations (16) and (18), we obtain the whole exergy destroyed in the preheater exchanger and the evaporator heat exchanger:

$$I_{ev} = \dot{m}_{fw}T_0(S_4 - S_2) - T_0[\dot{m}_{fp}\left(S_{p3} - S_{p2}\right) + \dot{m}_g(S_{g4} - S_{g3})] \tag{19}$$

– Destroyed exergy in turbine

$$I_t = \dot{m}_{fw}T_0(S_4 - S_5) \tag{20}$$

– Destroyed exergy in the condenser.

$$I_C = \dot{m}_{fw}[(h_5 - h_1) - T_0(S_5 - S_1)] - \dot{m}_{fc}[(h_{fc1} - h_{fc5}) - T_0(S_{fc1} - S_{fc5})] \tag{21}$$

The whole exergy destroyed of the ORC cycle is given by:

$$I = \sum_i I_i \tag{22}$$

Where I_i is the irreversibility of the component i

The evaporator extracted exergy is defined, according to [31] and [32], by equation (23):

$$\dot{E}_{evap} = \dot{Q}_{evap}\left(1 - \frac{T_0}{T_{mevap}}\right) \tag{23}$$

Where $T_{mevap} = \frac{T_{g4} - T_{g3}}{\ln\left(\frac{T_{g4}}{T_{g3}}\right)}$

The total exergetic efficiency is given by equation (24):

$$\eta_{ex} = \frac{\dot{W}_{net}}{\dot{E}_{evap}} \tag{24}$$

The proportion of energy destroyed is:

$$\psi = \frac{I}{\dot{E}_{evap}} \tag{25}$$

The power contained in the flue gases is evaluated by:

$$\dot{Q}_g = \rho_g c_{pg} q_g (T_g - T_0) \tag{26}$$

in which, ρ_g, c_{pg}, q_g, T_g are respectively the density, the heat capacity, the volume flow the temperature of the exhaust gas andT_0 the ambient air temperature.

E. Working fluids

As mentioned in section 1, the working fluids retained for this study are the organic fluids R134a and propane (R290) of which thermodynamic properties can be found in [33].

III. RESULTS AND DISCUSSIONS

To understand how the cycle works, we observed the influence of the inlet pressure of the fluid in the turbine for a fixed flue gas temperature. The results are shown in Figures 2 and 3 for propane and R134a respectively. It can be noted that for a fixed gas temperature, the flow rate of the working fluid decreases while the pressure increases for both fluids. This is justified by the fact that for a fixed gas temperature, an increase in the inlet pressure in the turbine (therefore the inlet temperature in the turbine according to the thermodynamic properties of the fluids), requires a large amount of energy, that is, thermal energy from the hot source. This led us to observe what would happen if the inlet temperature of the gases discharged from the combustion chamber were variable for an inlet pressure in the fixed turbine. The results are shown in Figure4.

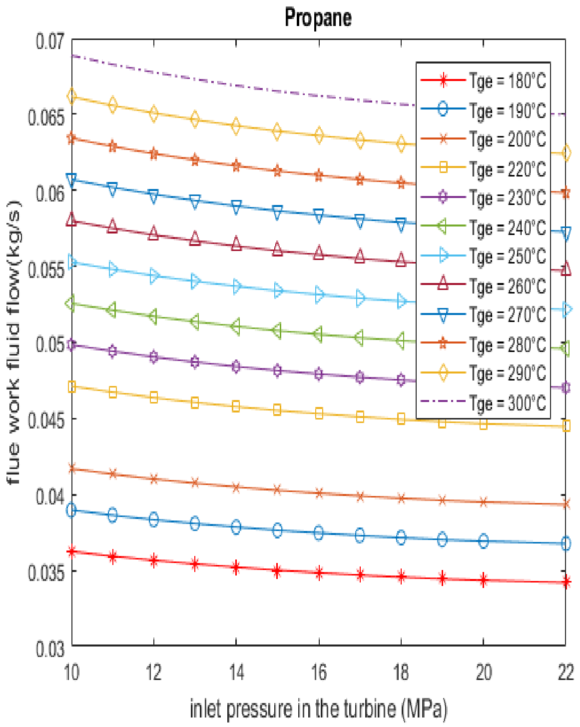

Fig. 2. Work fluid flow for a fixed flue gas temperature (Propane)

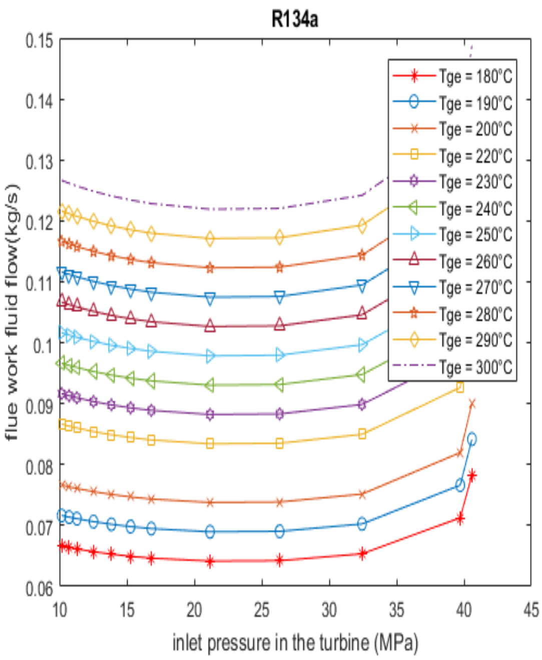

Fig. 3. Work fluid flow for a fixed flue gas temperature (R134a)

The observed results (Fig. 4) show that for an increasing temperature of the rejected gases, the flow rate of the working fluid is also for an inlet pressure in a fixed turbine. Indeed, for a pressure fixed at the inlet of the turbine, there must be a compensation between the mechanical power that the turbine will provide and the energy that the working fluid will provide. Thus, for an increasing temperature of the gases, this compensation will be obtained by increasing the flow rate of the working fluid to an inlet pressure in the fixed turbine. The results of figures 2, 3 and 4, then led us to observe the influence of these two parameters (inlet temperature of the rejected gases and inlet pressure in the turbine), on the flow rate of the working fluid if these two parameters varied simultaneously. The results are shown in Figure 5.

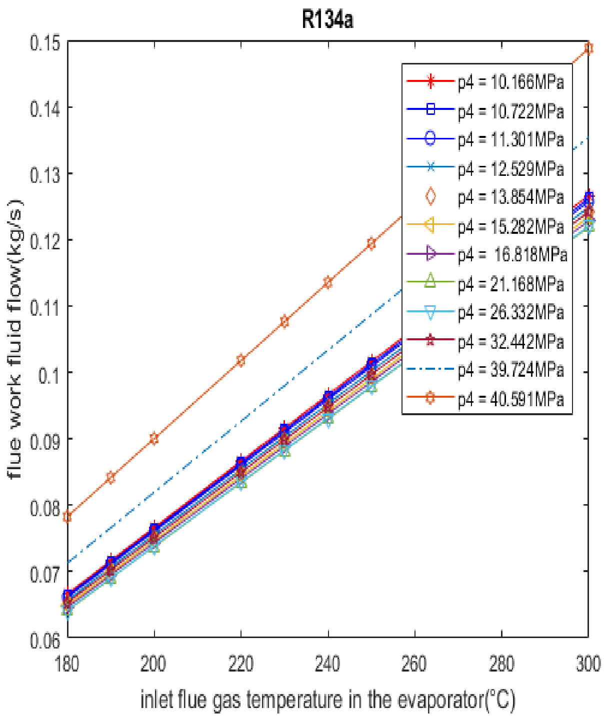

Fig. 4. Work fluid flow for a fixed flue gas pressure (R134a)

These results clearly show that the influence of the inlet temperature of the gases discharged from the combustion chamber is much greater than that of the inlet pressure in the turbine. The thermodynamic properties of the working fluids used in this work show that an increase in pressure follows with an increase in the temperature of the fluid [33]. Thus, the observations related to a change in the inlet pressure of the working fluid in the turbine would be the same for the inlet temperature in the turbine. Through these results, it can be noticed that they are in a good agreement with the results obtained by Bouhamady et al. [10] who studied the influence of the inlet temperature of the working fluid in the turbine on the mass flow of the working fluid, in the case of an ORC for a micro solar power plant, using R152a as working fluid. Indeed, they observed that the mass flow rate of the working fluid decreased while the inlet temperature in the turbine increased, which is the case in this work, for an inlet pressure in the turbine which increases (therefore for the inlet temperature in the turbine).

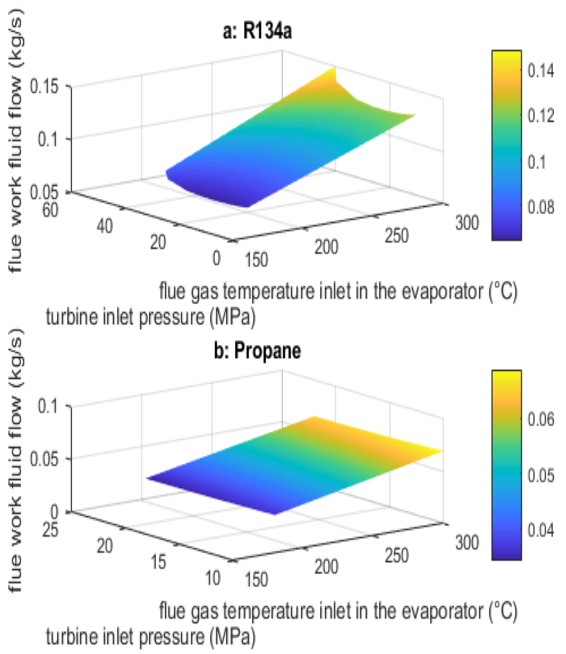

Fig. 5. Work fluid flow in function of inlet turbine pressure and inlet gas temperature

Figures 6, 7 and 8 show the influence of the inlet temperature of the gases discharged by the combustion chamber and the inlet pressure of the working fluid in the turbine on the supplied electric power, the net power cycle and mechanical power of the turbine. These results are obtained for an isentropic efficiency of the cycle pump of 80%, an isentropic efficiency of the turbine of 80% and an efficiency of the alternator of 85%. It follows from the obtained results that the inlet pressure in the turbine and the inlet temperature of the gases discharged by the combustion chamber have a great influence on the produced electrical power, the net power of the cycle and the mechanical power of the turbine. Indeed, an increase in these input parameters also implies an increase in the observed powers. For gas temperatures between 150 ° C and 300 ° C, we can recover up to 5kW of mechanical power in the turbine, which allows us obtaining an electrical power slightly higher than 4kW. Maximum power is obtained for high temperature and pressure values.

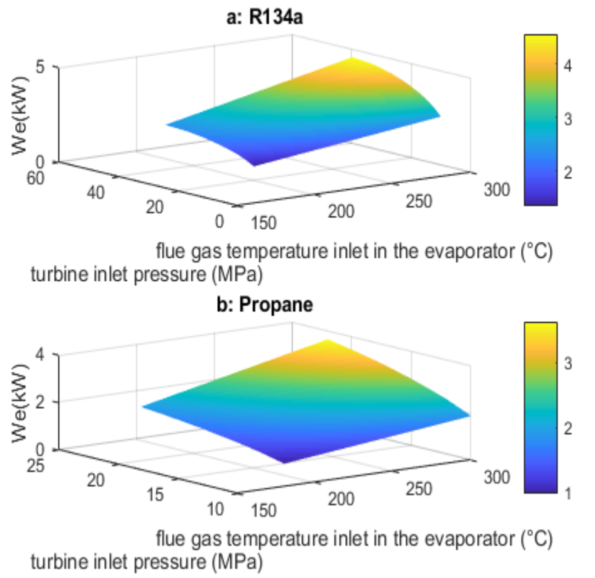

Fig. 6. Electric power in function of turbine pressure inlet and flue gas temperature inlet

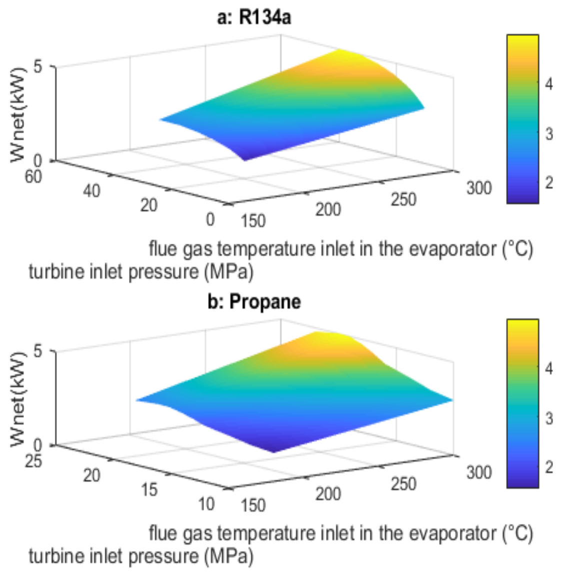

Fig. 7. Net power of the cycle

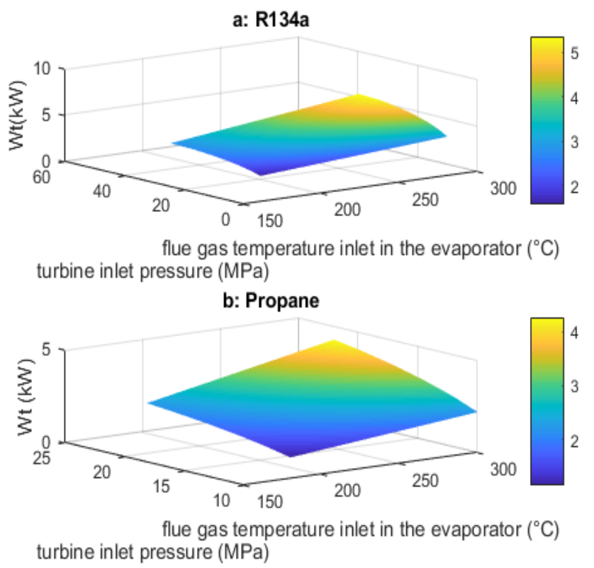

Fig. 8. Turbine power

The thermal efficiencies of the two working fluids are shown in Figure 9. The results show that the higher the inlet pressure in the turbine, the greater the thermal efficiency of the cycle. An important remark on the results of Figure 9 is that the thermal efficiency of the working fluid R134a is noticebly better than that of the working fluid R290. For example, it can be seen that for a pressure of 20MPa, the thermal efficiency for the R290 fluid is less than 15%, while it is greater than 15% for R134a. One can also observe the results with regard to the exergy cycle efficiencies for different values of the gas temperature and for different values of the inlet pressure in the turbine. The results are shown in Figures 10, 11, 12 and 13.

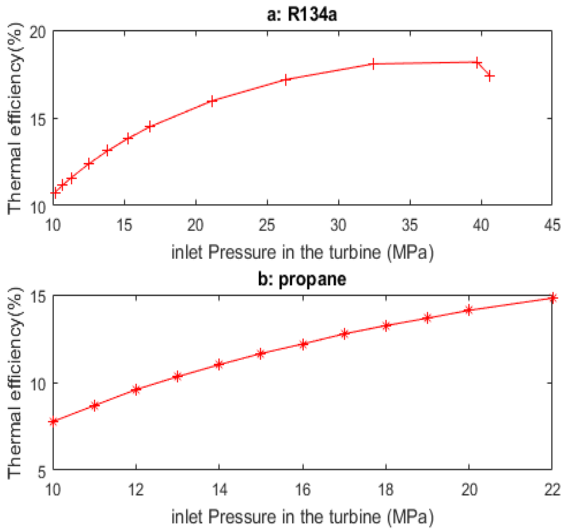

Fig. 5. Thermal efficiency

For different values of the inlet pressure in the turbine, the change of the exergy efficiency in the cycle for the two fluids can be analyzed. Figures 10 and 11 respectively show the results for R134a and propane (R290). It can be seen that the increase in the gas inlet temperature in the evaporator implies a decrease in the exergy efficiency of the cycle for a fixed inlet pressure in the turbine. However, large values of the inlet pressure to the turbine increase the exergy efficiency.

The same phenomenon was found by [4]. Another observation through these results is that the exergy efficiency is much greater for the working fluid R134a than for the fluid R290. It can be seen, for example, that for an inlet pressure in the turbine of 10.72 MPa the exergy efficiency of R134a reaches 50% at 180 ° C of the gas inlet temperature in the evaporator, while to reach 50% at the same inlet temperature in the evaporator, a pressure of 15MPa is required at the inlet of the turbine for the R290. These results can be confirmed by observing the exergy performance for both working fluids at different gas inlet temperatures in the evaporator, depending on the inlet pressure of the working fluid in the turbine. The results are presented in Figure 12. It follows that for an increasing inlet pressure in the turbine, the exergy efficiency increases for a fixed gas inlet temperature in the evaporator.

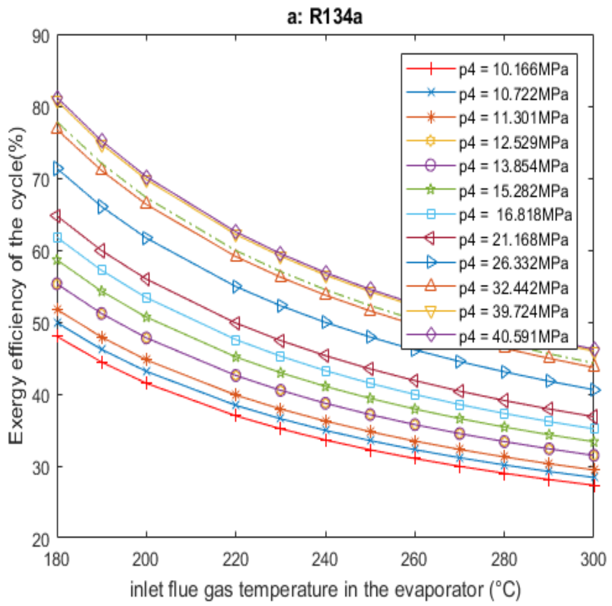

Fig. 60 Exergetic efficiency in function of flue gas inlet temperature

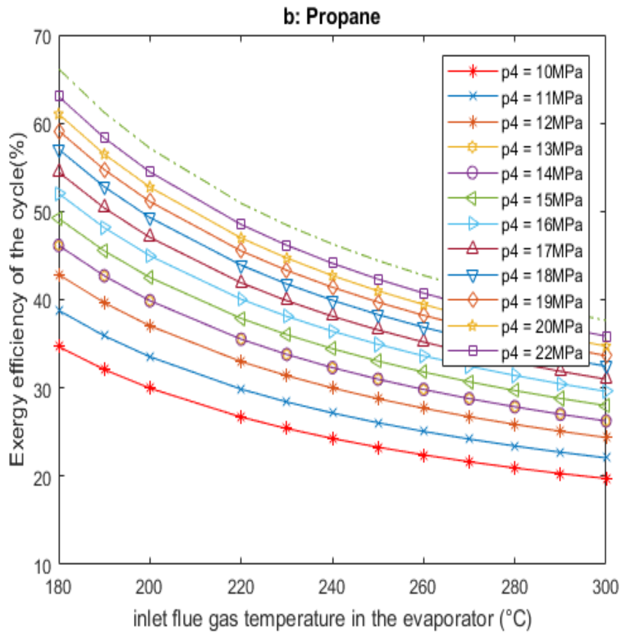

Fig. 71. Exergetic efficiency in function of inlet flue gas temperature

It is obvious from these results that a better exergy efficiency is obtained for low values of gas temperatures in the evaporator and for high values of inlet pressures in the turbine. These results are in agreement with the results presented in the literature, for example in [10], the authors observed the influence of the inlet temperature in the turbine on the exergy efficiency of the organic cycle, and the results show that an increase in the inlet temperature in the turbine implied an increase in the exergy efficiency of the cycle, which can be noticed in the present study, when the inlet pressure in the turbine of the working fluid increases.

Likewise, the results obtained in Wang et al. [34], shows that the exergy yield for R134a increases with the evaporation temperature. These changes in exergy can be explained by the fact that, for a fixed pressure, and a variable temperature of the gases, a large flow of the working fluid is needed, which also implies a significant destruction of the energy. On the other hand, for a fixed gas temperature, a slight decrease in the flow rate of the working fluid is observed, which limits the irreversibilities. These results can be confirmed from Figures 13, 14 and 15, which show the results obtained on irreversibilities during evaporation of the working fluid.

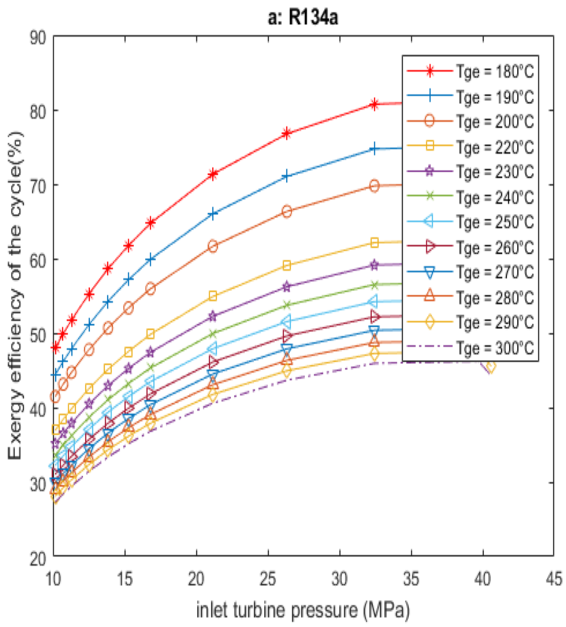

Fig. 82. Exergetic efficiency in function of turbine inlet pressure

One can see in Figure13 that for a gas inlet temperature fixed in the evaporator, the irreversibilities decrease during the evaporation of the working fluid. This justifies the fact that a decrease in the flow rate of the working fluid when increasing the inlet pressure in the turbine limits irreversibilities. As for figures 14 and 15, we see that for a fixed inlet pressure in the turbine, the irreversibilities become too great when the temperature of the gas entering the evaporator increases. This justifies the fact that increasing the flow of the working fluid also amounts to increasing the amount of energy destroyed in the cycle. Another important remark at the end of these results is that, increase the inlet temperature of the gases in the evaporator on the one hand, by increasing the inlet pressure of the working fluid in the turbine on the other hand, does not limit the destruction of exergy either.

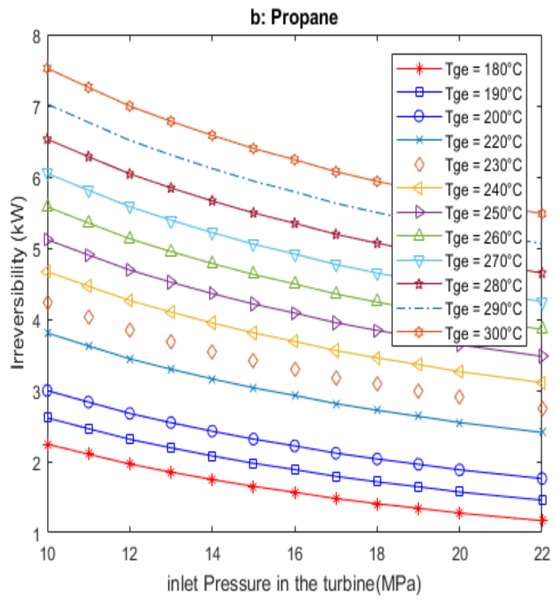

Fig. 13. Irreversibility in the evaporator in function of turbine inlet pressure

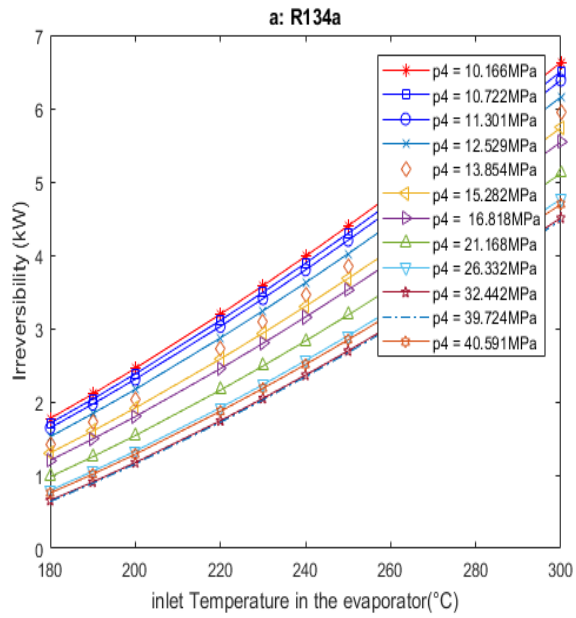

Fig. 95. Irreversibility in the evaporator in function of flue gas inlet temperature

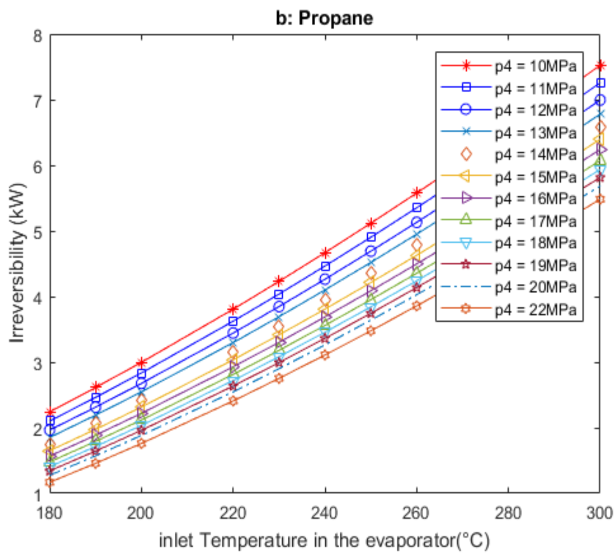

Fig. 106. Irreversibility in the evaporator in function of flue gas inlet temperature

IV. CONCLUSION

In this work, an analysis on the recovery of waste heat from ylang-ylang essential oil distilleries in Anjouan, Comoros was carried out. The possibility of converting this thermal energy transferred to the environment, into electrical energy was analyzed. For that purpose, an energy and exergy analysis of the cycle is made. The results show that with the types of distilleries encountered in the field (traditional type), we can recover up to 5kW mechanical (supplied by the turbine), and convert it into electrical energy up to 4.5 kW electric.

The results of the exergy analysis show that the irreversibilities in the evaporator become much greater when the temperature of the flue gases increases at the inlet of the exchanger. This is confirmed by the results of the bibliographic analysis which shows that the ORC cycles have a better performance for low temperatures of hot springs. The traditional Comorian stills reject gases with temperatures lower than 200 ° C in general, which allows us recommending the use of the organic Rankine cycle for the recovery and valorization of the residual heat of the distilleries based on the results of this study and those of the literature on ORC cycles.

It should also be noted that, for better energy and exergy cycle efficiencies at low temperatures, it is preferable to impose high pressures on the inlet of the turbine. Thus, the sizing of the heat exchangers (evaporators, preheating exchanger and condenser) must be carried out in this direction in order to have better optimization of the system. As an extension of the present work, it would be interesting to do a very detailed study on the optimization of such a cycle, by modeling and optimizing the exchangers and the other components of the system (pump and turbine) as they can provide outlet temperatures and pressures (therefore inlet in the turbine), of the working fluid, adequate for the correct operation and the best energy efficiency and exegetic cycle for the low temperatures of the exhaust gases. The goal would be to seek to optimize the system in such a way that the exchangers can provide sufficiently high temperatures and pressures at the inlet of the turbine, while having the best exchange surface so that we can recover the maximum heat.

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