International Refrigeration and Air Conditioning Conference
School of Mechanical Engineering

2014

# Exergy And Energy Analysis Of Waste Heat Recovery Options For Cooling Capacity Production

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# Exergy and energy analysis of waste heat recovery options for cooling capacity production

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## ABSTRACT

Electricity production in Lebanon stands at around 1.5 GW while the demand exceeds 2.5 GW at peak times and peak cooling demands, resulting in rationing cuts from between 3 to 20 hours a day which is the worst performance in the Middle East. Due to the country energy shortfall, a large number of small-scale backup generators is installed to address the electric and cooling needs. The proliferation of use of these backup generators presents an interesting potential for waste heat recovery in order to achieve additional power generation and cooling capacity production. This paper investigates two possible configurations for waste heat recovery: A first configuration studies the potential of combining an Organic Rankine Cycle (ORC) with a conventional vapor compression refrigeration cycle (VCRC) to meet the required cooling load. A second option evaluates the possibility of using a liquid desiccant cooling system (LDCS) to handle the latent part of the cooling load, in tandem with a conventional vapor compression refrigeration cycle handling the sensible part. The investigations are based on exergy and energy analysis carried out for both configurations to assess and compare their performance.

## 1. INTRODUCTION

Besides being massive drain on public resources with around 4% of the GDP for 2007, the Electricity sector in Lebanon has reached a critical stage in the last decade, and thus became unable to supply the reliable electricity needed for residences, commercial buildings, and industries. According to the World Bank, 2008, Lebanese citizens incur on average 220 interruptions of electricity per year, which is the worst performance in the Middle. Moving forward, solutions to Lebanon's electricity crisis are constrained by a limited government budget, a heavily subsidized electricity sector, low collection of electricity bills, an ageing infrastructure, human resources challenges and various interest groups resisting change.

For years now, private standalone power generators, backup generators, have been filling the gap all over the country despite the unbearable tariffs they impose on their clients. The electricity law of 1964 has granted Electricite du Liban (EDL) the monopoly of selling electricity to the public which makes these private generators technically illegal and as such, are not integrated into a wider regulated system.

The restructuring of the electricity sector has featured on the Lebanese government agenda since 1998. An electricity law emphasizing the privatization was first proposed, more than 60 consultant reports were prepared and different policies were adopted. But very little progress was made on implementing these initiatives due to disagreements across the political spectrum mainly around privatization. Demand for electricity is likely to reach over 4,000 MW by 2015. Unless EDL improves its ability to supply electricity and install new capacity and restore consumer confidence, back-up generation will increase its share of electricity supply in Lebanon to approximately 60% by 2015 (World Bank, 2008).

On the road to incorporating renewable energy sources into the generation mix, and working to improve energy efficiency on both load supply and demand, the backup generators will continue to be an important source to reduce the gap between supply and demand in Lebanon. A pragmatic approach would entail the Lebanese government leveraging the existing infrastructure of private generators across the country and adopting a policy of cooperation and coordination in the medium term.

In the face of unavoidable use of backup generators, one could think of enhancing their energy efficiency through the recovery of their waste heat. In the context of the electric energy shortfall on one hand, and the climate change, exhaustion of reserves of fossil fuels and ever increasing energy prices on the other hand, the waste heat recovery from these generators presents an interesting potential for additional power generation and cooling capacity production.

The interest in low-grade heat recovery has grown considerably lately. Numerous new solutions have been proposed to generate electricity from low temperature heat sources and are now applied to different fields such as solar energy, wind energy, industrial waste heat and engine exhaust gases (Quoilin *et al,* 2011). The potential for exploiting waste heat from engine exhaust gases or industrial processes is particularly interesting (Quoilin *et al*, 2011).Waste heat recovery technologies frequently reduce the operating costs for facilities by increasing their energy productivity. In addition to the economic benefits, waste heat recovery is a greenhouse gas free source of energy.

This paper investigates and compares two possible configurations for waste heat recovery. As one of the promising technologies of converting low grade heat into electricity, the low grade heat driven organic Rankine cycle (ORC) proved to be an attractive solution. The waste heat recovery from backup generators to drive an ORC is first investigated in this work. The ORC is coupled to a conventional vapor compression refrigeration cycle (VCRC) to produce cooling capacity.

A second configuration studies the recovery of waste heat for cooling capacity production via a liquid desiccant cooling system (LDCS). The (LDCS) handles the latent part of the cooling load, in tandem with a conventional VCRC handling the sensible part. Energy-driven LDCS are highly regarded by researchers and engineers due to the improved indoor air quality, their reduced energy consumption and low environmental impact. In this paper, exergy and energy analysis are carried out for both of the presented configurations in order to evaluate and compare their performances.

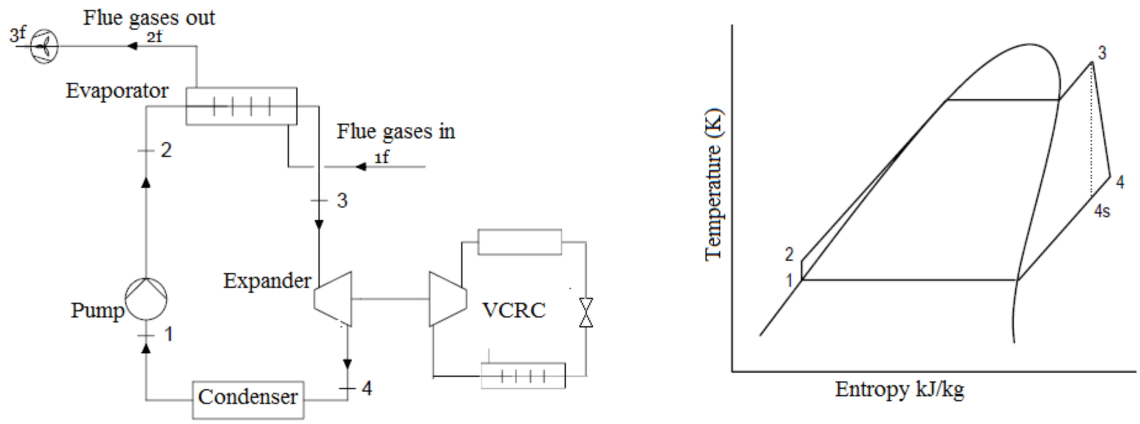
## 2. WASTE HEAT RECOVERY ORGANIC RANKINE CYCLE (WHR ORC)

### 2.1 System description

The ORC applies the principle of the steam Rankine cycle, but uses organic working fluids with low boiling points to recover heat from lower temperature heat sources instead of water. The simple ORC system integrates four basic components: an evaporator, an expander unit, a condenser and a working fluid pump.

The evolution of the working fluid into the investigated cycle can be summarized in four processes as shown in Figure 1. In the evaporator (process 2 to 3), the flue gases transfer their energy to the working fluid that evaporates and enters the expander. In the expander (process 3 to 4), the working fluid delivers mechanical work at the expense of pressure and temperature. The low pressure and temperature working fluid then enters the condenser where it condenses releasing heat at constant pressure and leaves as a saturated liquid (process 4 to 1), before entering the pump and completing a thermodynamic cycle (process 1 to 2). The blower (process 2f to 3f) is used to overcome the pressure drop in the evaporator. Both the evaporator and condenser operate at constant pressure. The compression process in both the pump and blower is considered adiabatic.

The cooling side is a standard VCRC. The compressor is directly coupled to the expander of the ORC. The conversion losses associated with electrical motors are eliminated. The high pressure vapor leaving the compressor undergoes standard condensing and evaporating processes to complete the loop. The working fluid HFC-134a is used as the refrigerant for its wide application.




(a) (b)

**Figure 1** Waste heat recovery Organic Rankine cycle. (a) Schematic diagram; (b) (T–s) diagram

The selection of the working fluid plays a critical role in exploiting the heat source efficiently to reach a high thermal efficiency and allow a high utilization of the available waste heat. According to Quoilin et al (2011), and Brasz and Bilbow (2004), refrigerants such as R1234yfa, R245fa, and R236fa are good candidates for moderate and low temperatures. It is desired to use fluids with high vapor density as they allow reducing the expander size and the heat exchangers areas. Additional working fluid characteristics to be taken into consideration are the toxicity, the flammability, the environmental impact, the cost and the chemical stability.

## 2.2 Mathematical model of the WHR ORC

In the present study, the temperature of the studied heat source varies between 200 and 400 °C, in agreement with the temperature range of exhaust gases from internal combustion engines, such as standalone backup generators (Sun and Li, 2011). An ambient temperature of 25 °C is considered for exergy calculations. When considering an ordinary recovery system, as shown in Figure 1a, the overall available resource is defined as bounded by the flue gas inlet 1f and the cooled flue gas down to ambient temperature 4f. Overall exergy resources are determined according to the cooling conditions of effluents and their molar compositions as expressed in equation (1):

$$\dot{E}x_{1f4f} = \dot{m}_f\left(ex_{1f} - ex_{4f}\right) \tag{1}$$

For each component, the thermal and/or mechanical power and exergy losses are evaluated from the energy and exergy balances, assuming stead state operation as presented in table 1. Combining the energy and exergy balance equations, the rate of exergy destruction is found as expressed in table 2 for the components of the ORC cycle.

**Table 1:** Energy and exergy balance for the components of the WHR ORC

| ***Component*** | ***Energy balance*** | ***Exergy balance*** |
|---|---|---|
| Evaporator | $\dot{m}_r(h_3 - h_2) + \dot{m}_f\left(h_{2f} - h_{1f}\right) = 0$ | $\dot{m}_r(ex_3 - ex_2) + \dot{m}_f\left(ex_{2f} - ex_{1f}\right) - \dot{E}x_{d,evap} = 0$ |
| Condenser | $\dot{m}_r(h_4 - h_1) + \dot{m}_{air}(h_{out} - h_{in}) = 0$ | $\dot{m}_r(ex_4 - ex_1) - \dot{E}x_{d,cond} = 0$ |
| Pump | $W_p = \dot{m}_r(h_2 - h_1)$ | $T_0.\dot{m}_r(s_1 - s_2) - \dot{E}x_{d,pump} = 0$ |
| Expander | $W_t = \dot{m}_r(h_3 - h_4)$ | $T_0.\dot{m}_r(s_3 - s_4) - \dot{E}x_{d,turbine} = 0$ |
| Blower | $W_b = \dot{m}_f\left(h_{3f} - h_{2f}\right)$ | $T_0.\dot{m}_f\left(s_{2f} - s_{3f}\right) - \dot{E}x_{d,blower}=0$ |

**Table 2** Exergy destruction for the components of the WHR ORC

| ***Component*** | ***Exergy destruction*** |
|---|---|
| Evaporator | $\dot{E}x_{d,evap} = T_o\left(\dot{m}_f\left(s_{2f} - s_{1f}\right) + \dot{m}_r(s_3 - s_2)\right)$ |
| Condenser | $\dot{E}x_{d,cond} = T_o\left(\dot{m}_r(s_4 - s_3)\right)$ |
| Pump | $\dot{E}x_{d,pump} = T_0(\dot{m}_r(s_2 - s_1))$ |
| Turbine | $\dot{E}x_{d,turbine} = T_0(\dot{m}_r(s_3 - s_4))$ |
| Blower | $\dot{E}x_{d,blower} = T_0.\dot{m}_f\left(s_{2f} - s_{3f}\right)$ |

Finally, total exergy losses for the ORC system are obtained by adding the contribution of each component as expressed in equation (2):

$$\dot{Ex}_{d,tot} = \dot{Ex}_{d,evap} + \dot{Ex}_{d,cond} + \dot{Ex}_{d,pump} + \dot{Ex}_{d,turbine} + \dot{Ex}_{d,blower} \quad (2)$$

The energy efficiency and the exergetic efficiency of the ORC are calculated as given in equations 3 and 4 respectively:

$$\eta_{en} = \frac{W_t - W_p - W_b}{\dot{m}_f\left(h_{1f} - h_{2f}\right)} \quad (3) \qquad \eta_{ex} = 1 - \frac{\dot{Ex}_{d,tot}}{\dot{Ex}_{1f4f}} \quad (4)$$

The VCRC is modeled by a coefficient of performance equal to 35 % that of a reversed Carnot cycle operating between a hot thermal reservoir at a temperature of 25°C (ambient temperature) and a cold thermal reservoir at a temperature of 12°C, equal to the temperature of chilled water in a refrigeration system.

## 3. LIQUID DESICCANT COOLING SYSTEM (LDCS)

### 3.1 System description

Desiccant cooling has long been adopted for both industrial and agricultural purposes, and is now playing a prominent role in the air-conditioning field. Desiccant dehumidification can be done by either liquid or solid desiccant. However, liquid desiccants have many advantages over solid ones: in terms of humidity control, energy consumption, performance, indoor air quality (Natural disinfectant of airborne particulates & microorganisms), ease of installation and maintenance. Moreover, liquid desiccants require lower regeneration temperature, around 50 to 60°C (Longo and Gasparella, 2005), (Qui *et al*, 2008), which can be easily obtained from solar energy and low to moderate temperature waste heat recovery (Gandhidasan, 2004).

In a LDCS, a concentrated liquid desiccant is distributed in a dehumidifier to absorb the moisture from the airstream passing through the dehumidifier. As a result, the liquid desiccant leaving the dehumidifier becomes diluted. Afterwards, using available and appropriate heat sources, the desiccant is reheated in the regenerator to a useful level of concentration. The present study investigates the potential of using waste heat recovered from exhaust gases of backup generators as the heat source needed for the regeneration of the liquid desiccant.

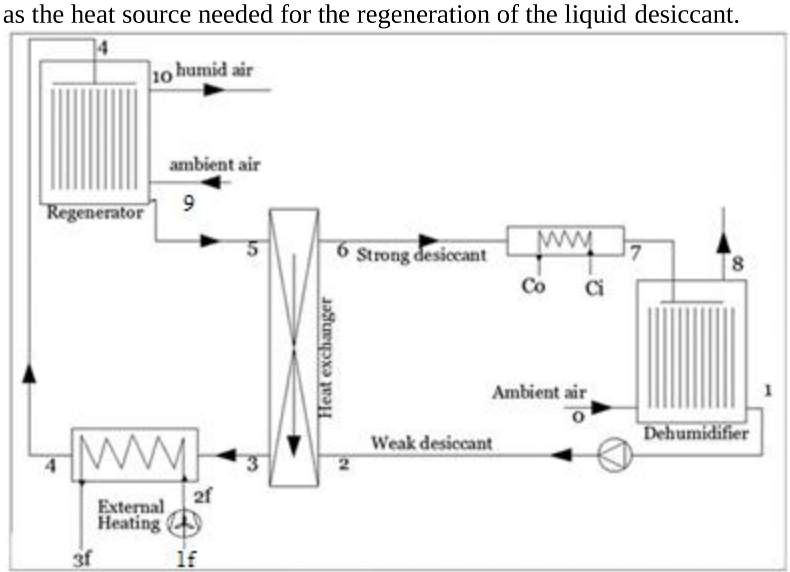



**Figure 2:** Schematic drawing of the LDCS

A LDCS is shown in figure 2. It is composed mainly of: a dehumidifier, a regenerator, and heat exchangers. In the dehumidifier, the inlet air is brought in contact with the strong liquid desiccant. The difference in surface vapor pressure between the liquid-desiccant film and the inlet air acts as the driving force for water vapor transfer from the air to the liquid water in the liquid desiccant. The latter absorbs the humidity and becomes diluted: it is then referred to as weak desiccant. In the regenerator, the waste heat recovered from backup generators is retrieved for the regeneration of the weak liquid desiccant. Heat exchangers are used to allow the heat transfer between the weak and strong desiccant, between the external heat source and the weak desiccant to be regenerated in the weak solution heat exchanger (process 3-4), and between the strong desiccant and the cooling water (process 6-7).

The choice of liquid desiccant depends on the temperature range of exhaust gases. For the range considered, aqueous solution of Lithium chloride, Calcium chloride, Ethylene glycol can be used (Patek et Klomfar, 2008). Lithium chloride is considered the most effective due to its low vapor pressure and hence will be considered in the present paper (Maatouk *et al*., 2012).

## 3.2 Mathematical model of the LDCS

Several models were developed to simulate the absorber in a liquid desiccant system. Models using lithium chloride were described by Khan and Martinez (1998), Ahmed et al. (1998) and Fumo and Goswani (2001). A model of packed bed liquid desiccant air dehumidifier using lithium chloride was developed and validated by Gandhidasan (2004), and will be adopted in the present study.

According to Gandhidasan, the mass flow rate of water condensed from the air to the desiccant solution is a function of the air flow in the dehumidifier, the outlet desiccant concentration, and the outlet desiccant temperature. Assuming steady state operation, the energy and exergy balances are developed for each component of the LDCS as illustrated in tables 3 and 4 respectively.

**Table 3:** Energy balance for the LDCS components

| ***Component*** | ***Energy balance*** |
|---|---|
| Regenerator | $\dot{m}_{ar}(h_9 - h_{10}) + \dot{m}_w h_4 - \dot{m}_s h_5 = 0$ |
| Dehumidifier | $\dot{m}_{ad}(h_0 - h_8) + \dot{m}_s h_7 - \dot{m}_w h_1 = 0$ |
| Cooling water heat exchanger (CWHX) | $\dot{m}_{cw}(h_{ci} - h_{co}) + \dot{m}_s(h_6 - h_7) = 0$ |
| Solution Heat exchanger | $\dot{m}_w(h_2 - h_3) + \dot{m}_s(h_5 - h_6) = 0$ |
| External Heating of the weak desiccant | $\dot{m}_f\left(h_{2f} - h_{3f}\right) + \dot{m}_w(h_3 - h_4) = 0$ |

**Table 4:** Exergy balance for the LDCS components

| ***Component*** | ***Exergy balance*** |
|---|---|
| Regenerator | $\dot{Ex}_{d,reg} = T_o(\dot{m}_{ar}(s_{10} - s_9) + \dot{m}_s s_5 - \dot{m}_w s_4)$ |
| Dehumidifier | $\dot{Ex}_{d,deh} = T_o(\dot{m}_{ad}(s_8 - s_0) + \dot{m}_w s_1 - \dot{m}_s s_7)$ |
| Cooling water Heat exchanger | $\dot{m}_{cw}(ex_{ci} - ex_{co}) + \dot{m}_s(ex_6 - ex_7) - \dot{Ex}_{d,CHWE} = 0$ |
| Solution heat exchanger | $\dot{m}_w(ex_2 - ex_3) + \dot{m}_s(ex_5 - ex_6) - \dot{Ex}_{d,SHX} = 0$ |
| External Heating of the weak desiccant | $\dot{m}_f\left(ex_{2f} - ex_{3f}\right) + \dot{m}_w(ex_3 - ex_4) - \dot{Ex}_{d,EHWD} = 0$ |

The rate of exergy destruction for the global LDCS system is the sum of the individual exergy destruction at each component as stated in equation (5):

$$\dot{Ex}_{d,tot} = \sum_i \dot{Ex}_{d,i} = \dot{Ex}_{reg} + \dot{Ex}_{deh} + \dot{Ex}_{CWHE} + \dot{Ex}_{SHX} + \dot{Ex}_{EHWD} \quad (5)$$

The overall available resource is defined as bounded by the flue gas inlet 1f and the cooled flue gas down to ambient temperature 4f. Hence, the exergetic efficiency of the LDCS is calculated as per equation (6):

$$\eta_{ex} = 1 - \frac{\dot{Ex}_{d,tot}}{\dot{Ex}_{1f4f}} \quad (6)$$

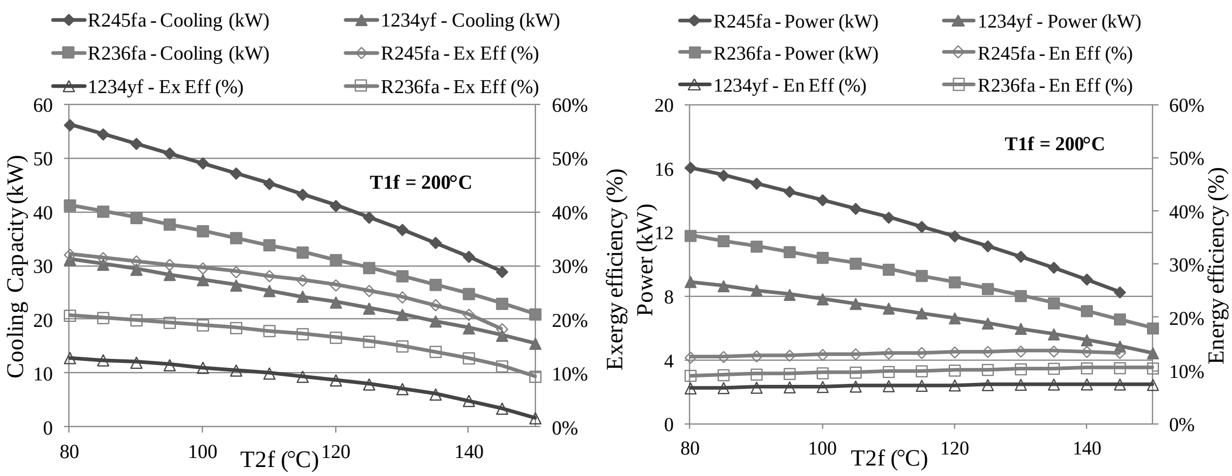
# 4. RESULTS AND DISCUSSIONS

The energy efficiency and exergetic efficiencies, the cooling capacity as well as the net power are evaluated based on a unit mass of flue gas for both waste heat recovery configurations as function of the flue gas inlet and exit temperature. For the simulations, an ambient temperature of 25°C is considered. Thermodynamic properties of the different states of the configurations are calculated using REFPROP for air, water, flue gas and refrigerant flows. Thermodynamic properties of the LiCl-$H_2O$ solution are calculated using the model developed by Patek and Klomfar (2008).

### 4.1 Results for the WHR ORC

To evaluate the performance of the WHR ORC, simulations are carried out for 3 working fluids: R1234yfa, R245fa, and R236fa. The high pressure level is set at 2200 kPa. The flue gas exit temperature varies between 150°C down to 80°C. An efficiency of 60% is assumed for the pump and of 75% for the expander. 3 flue gas inlet temperatures were tested: 200, 300 and 400°C corresponding to usual exhaust gases temperatures from backup generators.

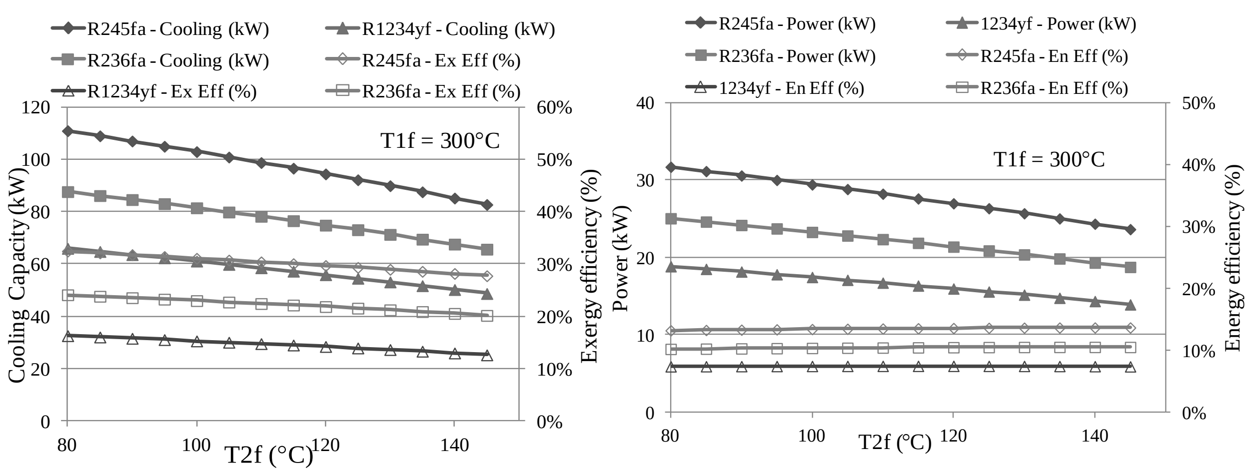
In order to choose the most adapted working fluid, simulations are performed for the 3 previously mentioned fluids at the flue gas inlet temperature previously set. The results are plotted against the flue gas exit temperature.
As can be seen from figures 3a and 3b corresponding to a flue gas inlet temperature of 200°C, R245fa exhibits the highest performance in terms of energy and exergy efficiencies, as well as net power delivered and cooling capacity produced. For a flue gas inlet temperature of 200 °C, a max cooling capacity of 60 kW is estimated and a net power of 16 kW is expected while the exergy and energy efficiencies reach 32% and 14% respectively.




(a) (b)

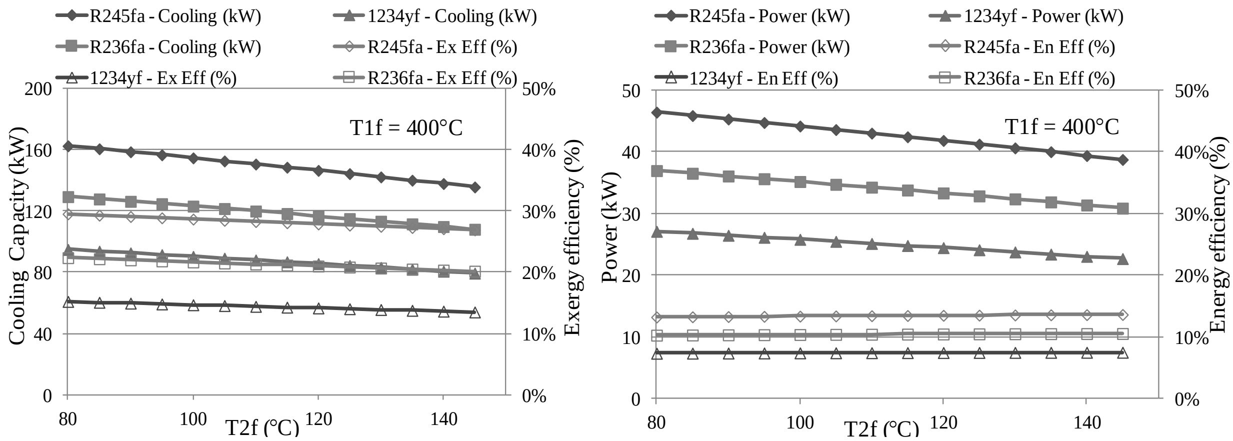
**Figure 3** Flue gas inlet temperature 200 ℃
a) Cooling capacity and exergy efficiency of WHR ORC with R1234yfa, R245fa, and R236fa
b) Net power and energy efficiency of WHR ORC with R1234yfa, R245fa, and R236fa

Calculations are also carried out for the 3 working fluids with a flue gas inlet temperature of 300 °C and 400 °C. The results came consistent with the previous ones: the highest performance of the WHR ORC is obtained with R245fa as can be seen in figures 4 and 5 respectively.




(a) (b)

**Figure 4** Flue gas inlet temperature 300 ℃
a) Cooling capacity and exergy efficiency of WHR ORC with R1234yfa, R245fa, and R236fa
b) Net power and energy efficiency of WHR ORC with R1234yfa, R245fa, and R236fa



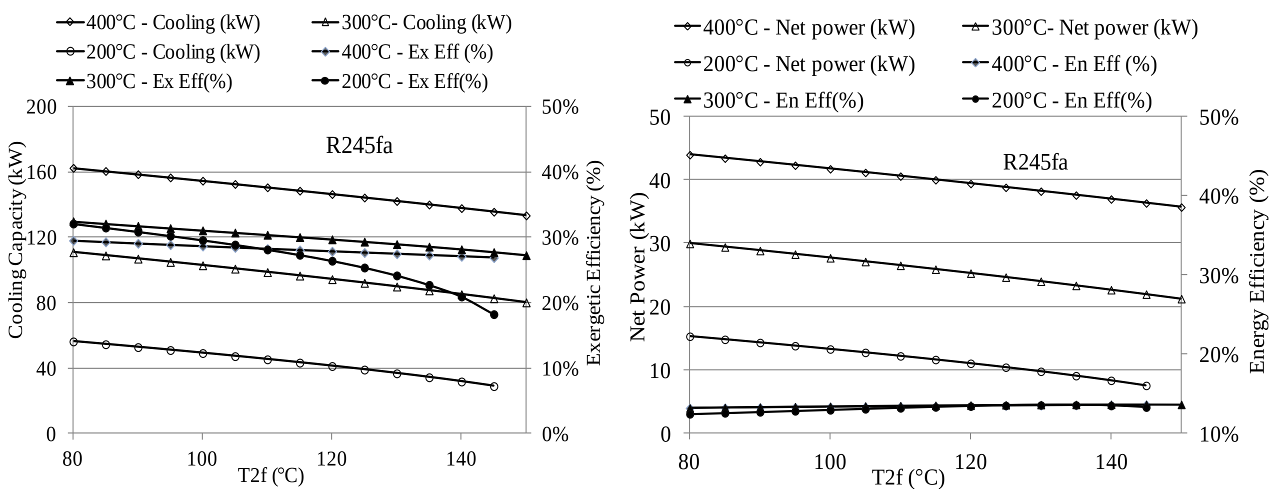

(a) (b)

**Figure 5** Flue gas inlet temperature 400 ℃

a) Cooling capacity and exergy efficiency of WHR ORC with R1234yfa, R245fa, and R236fa

b) Net power and energy efficiency of WHR ORC with R1234yfa, R245fa, and R236fa

Figure 6 illustrates the performance of the WHR ORC with R245fa as a function of the flue gas exit temperature and for three flue gas inlet temperatures: The performance is measured in terms the exergetic efficiency and cooling capacity (figure 6a) and in terms of net power delivered and energy efficiency (figure 6b). For a flue gas inlet temperature of 400°C and mass flow rate of 1kg/s, the WHR ORC using R245fa allows the production of 160 kW of cooling capacity and the delivery of 46 kW while having an exergy efficiency of 40% and an energy efficiency of 45%.




(a) (b)

**Figure 6:** a) Cooling capacity, exergetic efficiency of WHR ORC for R245fa as function of flue gas inlet and exit T

b) Net power and energy efficiency of WHR ORC for R245fa as function of flue gas inlet and exit T

The pie chart in Figure 7 shows the breakdown of the total exergy destruction of the WHR ORC. It can be seen that the major losses occur in the evaporator with a ratio of 43% of total exergy losses. This is primarily due to the temperature difference between the flue gas and working fluid. The losses are also significant in the condenser (26%) where the condensation takes place in the superheated region far from the saturation condition of the fluid.

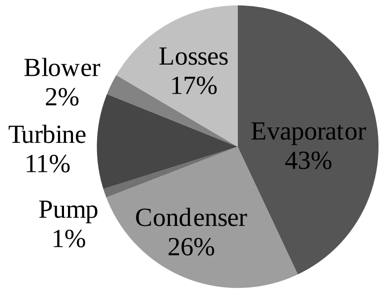



**Figure 6**: Distribution of Exergy destruction in the WHR ORC

**4.2 Results for the LDCS**

The effects of the desiccant concentration and flue gas inlet and exit temperatures on the global exergy efficiency of the system are studied. The inlet conditions of the model are stated in table 5. $T_{1f}$ varies between 200 and 400°C, and for a given $T_{1f}$, the exit temperature of the flue gas $T_{3f}$ is varied between 86 and 105°C.

**Table 5 LDCS model conditions**

| $T_0$ and $T_9$ | Mass flow rate of the absorber solution | Mass flow rate of air in the absorber | Efficiency of heat exchangers |
|---|---|---|---|
| 30°C, 80%Relative humidity | 6 kg/s | 1.5 kg/s | 60% |

Figure 7 shows the effect of the flue gas inlet temperature on the global exergy efficiency. It can be seen that the global exergy efficiency increases with the increase of $T_{1f}$. For a given flue gas inlet temperature, the global exergy efficiency increases as the flue gas exit temperature decreases.

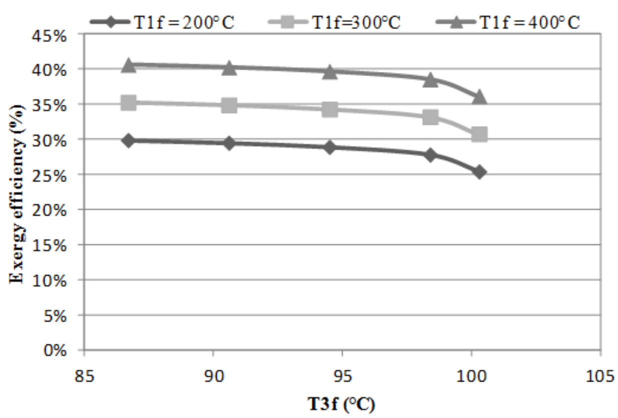



**Figure 7** Influence of flue gas inlet and exit temperature on the LDCS global exergy efficiency

For the conditions of air stated in table 5, the latent heat handled by the LDCS was found equal to 138 kW and the sensible to 15 kW. In the present work, this cooling capacity remained constant for the different flue gas conditions since the model fixes the temperature at the inlet of the absorber.

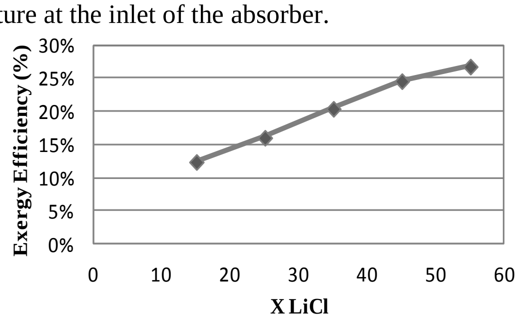



**Figure 8** Variation of the exergy efficiency of LDCS with LiCL concentration

The effect of the desiccant concentration at the dehumidifier inlet on the global exergy efficiency is illustrated in Figure 8. The global exergetic efficiency increases with the increase of the salt concentration by a slope of 4%/(kg/kg). This rate is consistent with the findings presented in Maatouk *et al*,2012.

The pie chart presented in figure 9 shows the contribution of the different components in the overall irreversibility rate according to the specific conditions listed in table 5. The major irreversibilities are due to the partial recovery of the energy in the hot flow, and in the external heating heat exchanger because of the important temperature difference between the hot and the cold flux.

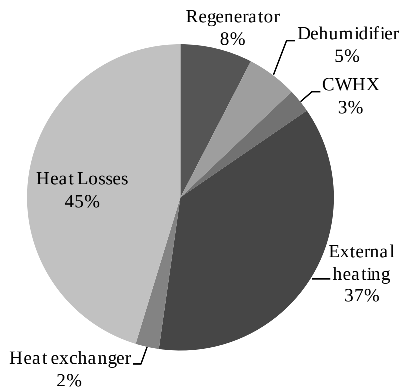



**Figure 9** Distribution of Exergy destruction in the LDCS

## 5. CONCLUSION

The recovery of waste heat by the production of cooling capacity is considered via two systems: an ORC and a LDCS. A parametric analysis was carried for both configurations to evaluate the performance of the system as a function of the flue gas inlet and exit temperatures and the working fluid properties. For The WHR ORC, results of simulations showed that an increase in the flue gas inlet temperature would enhance the overall exergy efficiency and the cooling capacity whereas a decrease of the flue gas exit temperature would result in better performance in terms of global exergy efficiency and cooling capacity production. For the LDCS, the calculations demonstrated that the global exergy efficiency increased with the increase of the exhaust gases temperature. For the WHR ORC, the highest performance is achieved with the R245fa while an increase of the salt concentration in LDCS would result in an increase of the system performance.

## NOMENCLATURE

| | | | | |
|---|---|---|---|---|
| ex | flow exergy | (kJ/kg) | **Subscripts** | |
| $Ex_d$ | exergy destruction | W | f | flue gas |
| h | specific enthalpy | (kJ/kg) | p | pump |
| m | mass flow rate | (kg/s) | r | refrigerant |
| Q | rate of heat transfer | (W) | cw | cooling water |
| s | specific entropy | (kJ/kg K) | b | blower |
| T | temperature | (K) or (°C) | w | weak |
| W | power | W | s | strong |
| **Abbreviations** | | | reg | regenerator |
| CHWX | Cooling water heat exhanger | | t | expander |
| EDL | Electricity of Lebanon | | deh | dehumidifier |
| ORC | Organic rankine cycle | | **Greek Symbols** | |
| WHR | waste heat recovery | | $\eta_{en}$ | thermal efficiency |
| LDCS | Liquid desiccant cooling system | | $\eta_{ex}$ | exergetic efficiency |

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