TECHNICAL AND ECONOMIC ANALYSIS OF ORGANIC FLASH REGENERATIVE CYCLES (OFRCs) FOR LOW TEMPERATURE WASTE HEAT RECOVERY

A. Baccioli, M. Antonelli, U. Desideri*

Department of Energy, System, Construction and Territory Engineering

University of Pisa

Largo Lucio Lazzarino, 56122 Pisa, Italy

* Corresponding Author: umberto.desideri@unipi.it, Tel: +39 050-2217375; Fax: 050-2217333

Abstract

Organic Flash Cycles (OFCs) can improve the overall efficiency of waste heat recovery or geothermal systems due to a better match of the hot and cold heat transfer curves. However, the lower mean temperature difference between the heat transfer curves implies larger exchanger areas and therefore higher heat exchanger costs.

In order to reduce the exchanger size, a new cycle configurations is introduced in this paper, consisting in a new type of organic flash regenerative cycle (OFRC) for heat source temperatures in the range 80-170°C. The regeneration allows to recover part of the enthalpy of the liquid phase from the flash evaporator increasing the temperature of the liquid at the exchanger inlet, thus reducing the exchanger size. The thermodynamic performance of OFRCs are practically the same as of the OFC, but the specific cost of the system can be 20% lower. A variety of working fluids was tested and results have shown that long molecular chain alkanes provide the best thermodynamic efficiency, but those fluids have the main drawback of a low vapor density, resulting in very large expansion devices and condensers. R601a is the working fluid featuring the best tradeoff between thermodynamic efficiency and components size in the heat source temperature range between 80°C and 170°C. The comparison of the OFRC with conventional ORCs has shown the thermodynamic superiority of the OFRC with every tested fluid. Finally the

cost analysis has highlighted that OFRCs specific cost has the same magnitude as ORCs for mini and micro scale plants.

Keywords: Organic Flash Cycle, Organic Flash Regenerative Cycle, Organic Rankine Cycle

1. Introduction

Recovering energy from heat at low temperature is necessary if we wish to increase energy saving and exploit renewable sources which are currently scarcely used. In those applications ORCs play a major role because of the proprieties of organic fluids, which allow to exploit low temperature and variable temperature heat sources with compact and simple components allowing to build them with a small size [1-10]. However the presence of a phase change zone, while heating and superheating the fluid, implies that a fraction of the heat is transferred from a variable temperature heat source to a constant temperature fluid, and that the highest temperature the working fluid can reach is quite far from the source temperature [11], as well as the lowest temperature reached by the heat source is normally much higher than the lowest temperature of the cycle. These facts lead to exergy destruction in the heat transfer process and exergy loss in the release of high temperature heat to the environment, especially in the cases of geothermal or waste heat recovery systems.

Several different solutions have been presented in the literature to have a better match between the heat transfer curves of the heat source and the heat recovery system.

Kalina cycles introduced in the 1980s and using a mixture of water and ammonia as working fluid, tried to mitigate this problem by using a fluid featuring a a non-isothermal evaporation thereby reducing the average temperature difference in the heat recovery system. The layout of Kalina cycle is much more complex than the one of ORCs, because of additional separators and heat exchangers. These cycles have been widely studied in literature: one of the first analysis of Kalina cycles was carried by Stecco et al. in [12]: a model was developed to analyze advantage and disadvantage of these type of cycles with respect to other cycles. In further papers [13-14] sizing criteria for heat recovery boiler design and for geothermal exploitation with water-ammonia mixture were defined.

Nag et al. [15] showed that mass concentration at the turbine inlet has a strong influence on the cycle efficiency and they found an optimum value to maximize the cycle second law efficiency.

Bombarda et al. [16] compared the performance of a Kalina cycle with the one of an ORC with MM (hexamethyldisiloxane) as working fluid which recovered heat from a Diesel Engine exhaust. Despite the small advantage of the Kalina cycle respect to the ORC in terms of net mechanical power, the high working pressure of water-ammonia mixture demonstrated that this cycle was not applicable in the considered temperature range because of the high equipment costs.

Other authors analyzed the performance of ORCs with zeotropic mixtures. The mixture of different fluids results in a fluid which presents a temperature change during evaporation and condensation with a better matching of the exchange curves both in the evaporator and in the condenser. Differently from the Kalina cycle the layout is the same of classic ORCs, without the need of separators or additional heat exchanger. Wang et al. [17] analyzed a mixture of R245fa and R152a searching for the optimum mass fraction composition to maximize the efficiency of a solar ORC cycle. They stated that with zeotropic mixtures the use of a superheater coupled with an internal heat exchanger increases the efficiency of the cycle. Victor et al. [18] carried out an analysis with respect to the optimization of cycle efficiency among pure organic fluids, mixed organic fluids, water-ammonia mixtures and water-alcohols mixtures. They demonstrated that, using pure fluids, cycle efficiency increases in the temperature range 100-150°C, while the Kalina cycle provided the best results for temperature between 150°C and 250°C, despite the higher required pressure which would result in higher plant costs, while mixed organic fluids provided a lower cycle efficiency than pure fluids. The improvement of geothermal and waste heat recovery systems using zeotropic organic mixtures was largely studied in the literature [19-21], however few plant have been built.

The most beneficial effect of using zeotropic mixtures is the temperature glide at the condenser as demonstrated by Liu et al. in [22]. For this reason many authors have tried to improve the cycles efficiency by adopting supercritical solutions with zeotropic mixture fluids [23]. Supercritical cycles allow a better match of the heat transfer curves than fluid zeotropic mixtures because of the lack of the two phase zone in the heating process. When using supercritical cycles both the cycle efficiency and the heat transfer process are improved [24]: many authors analyzed this technology using CO_2 as working fluid [25-30]. Although the encouraging theoretical results, the high pressure and the problem concerning the design of a proper cooling system, require further studies on this technology and a further search of an optimal fluid [24].

The trilateral flash cycle is another technology designed to reduce entropy generation during the heat transfer process [31- 33], but nowadays no efficient two phase expander is available.

A modification to the trilateral flash cycle is represented by the Organic Flash Cycle, presented by Ho et al. in [34-36]. Differently from trilateral cycles, the OFC separates the vapor from the liquid after the throttling process and only the vapor is sent to the turbine, without the need of a two phase expander. The presence of a throttling process, however, introduced irreversibilities, which reduce the benefits of the close match of the heat transfer curves, and a second flash stage is needed to increase power output, above all when the temperature of the heat source is low (<200°C) [35,36]. Another problem of the flash cycle is that the working fluid has to be heated from condensing temperature up to the maximum temperature of the cycle, and since exchange curves must be as close as possible, heat exchangers cost is significant.

In this paper, some modifications to the Organic Flash Cycle, presented in [36] for low temperature waste heat recovery or geothermal system in the temperature range between 80 and 170°C, have been considered in order to lower system costs. In addition, exergy analysis has been carried out to compare the solutions proposed in the literature and in ORC systems, highlighting the advantages and the disadvantages. This paper presents an extensive analysis of OFRCs aiming at highlighting all benefits and drawbacks in comparison with ORCs both from a technical and an economic point of view. Positive displacement expanders, which were widely studied for organic fluids [37-43], are considered in this paper because of their relatively low cost [42,43] and in order to reduce the complexity and the cost of systems using turbo expanders as much as possible. The analysis has been carried out with both liquid and air cooled condenser. Various fluids have been employed as working fluid in order to evaluate the most appropriate fluid for OFRCs. Finally a comparison with conventional ORC system has been carried out.

Nomenclature		*Subscripts*	
$\dot{Q}$	Thermal Power [kW]	$exch$	exchanged
$\dot{m}$	Mass Flow rate [kg/s]	HTF	Heat Transfer Fluid
h	Specific Enthalpy [kJ/kg]	org	Organic Fluid
$\bar{C}_p$	Average Specific Heat [kJ/kgK]	av	Available
T	Temperature [K]	in	Inlet
$\dot{W}$	Mechanical Power [kW]	out	Outlet
$\dot{E}x$	Exergy [kW]	0	Ambient Reference State
s	Specific Entropy [kJ/kgK]	c	Cycle
$\dot{I}$	Exergy destruction and loss [kW]	N	Net
x	Vapor quality	$H.P.$	High Pressure Expander
P	Pressure [Pa]	$L.P.$	Low Pressure Expander
BWR	Back-Work Ratio	liq	Liquid phase
r	Volume expansion ratio	P	Pressure

		is	Isentropic
		$cond$	Condenser
		exp	Expander
Greeks		*Superscripts*	
ε	Recovery Efficiency	I	First Law
η	Efficiency	II	Second Law
ρ	Density [kg/ m^3]	‘	First Flash Evaporator
		“	Second Flash Evaporator

2. Cycles description

The Organic Flash Cycle (OFC) reported by [36] for low temperature heat recovery is a double flash cycle in order to deliver extra power with respect to the single flash cycle and it is shown in fig. 1. The liquid is pumped into the heat exchanger, where the fluid is heated up to the saturation temperature, in liquid phase (1-2), by the Heat Transfer Fluid (HTF), flashed in a throttling valve (2-3) and then introduced into a flash evaporator where liquid and vapor are separated. The vapor is then expanded in the high pressure expander (4-5), while the liquid is throttled in a second valve (6-7) and then mixed with the vapor coming from the first expander outlet (8). The two phase mixture is then separated in a second flash evaporator and the vapor is sent to the low pressure turbine (9-10), while the liquid is throttled (11-12), mixed with the vapor coming from the low pressure expander outlet and sent to the condenser (13-14). The temperature entropy (T-s) diagram of this cycle is shown in figure 2. The advantage of the cycle is that the temperature difference between the HTF and the working fluid is quite small and the outlet temperature of the HTF can be very close to the lowest temperature of the cycle, reducing the exergy loss associated to the HTF flow at the outlet of the system. However, the presence of three throttling valves causes reduction in the cycle efficiency and, in order to maximize the overall efficiency of the system, the pinch point temperature difference in the heat exchanger must be as small as possible [35]. This fact leads to the adoption of heat exchangers with larger surface area and higher system cost.

The modified flash cycle studied in this paper is an Organic Flash Regenerative Cycle (OFRC) and it is shown in fig. 3. The layout is very similar to the OFC: the fluid is heated up, in liquid phase, to the saturation temperature by the HTF fluid in the principal heat exchanger (1-2), laminated by a throttling valve (2-3) and flashed. The vapor, from the flash separator is sent to the high-pressure expander (4-5), while the liquid is laminated in a second throttling valve (6-7) and then re-mixed with the vapor from the expander (8). The two phase mixture is then separated in a second separator and the vapor driven to the low pressure expander (9-10) and therefore to the condenser (10-12). The liquid of the second flash evaporator is used to recuperate heat at the inlet of the heat exchanger (14), thus eliminating a throttling process. The T-s diagram of the modified cycle is plotted in fig.4.

FIGURE 1 ABOUT HERE

FIGURE 2 ABOUT HERE

FIGURE 3 ABOUT HERE

FIGURE 4 ABOUT HERE

As a result of regeneration, the outlet temperature of the HTF in the regenerative cycle is higher than in the simple cycle and both the heat transferred and the heat recovery efficiency are lower. However, in this case, since the amount of heat introduced in the regenerative cycle is smaller than in the simple cycle, and the enthalpy drop available for the expansion is almost the same, for a given fluid, the cycle efficiency results to be higher. For this reason, the overall efficiency (recovery and thermal cycle) will not be so different between the two solutions.

Moreover, without the regeneration, the HTF has to warm up the working fluid from the condensing temperature up to the higher temperature of the cycle, requiring larger heat transfer surface areas and increasing the system costs.

In the case of OFRC, regeneration is completely different from ORCs. In fact, in this last the heat of the superheated vapor, at the expander outlet, is transferred to the liquid at the outlet of the pump by means of a surface heat exchanger; in OFRCs, the enthalpy stream of the liquid from the low pressure flash evaporator is mixed with the liquid from the condenser to regenerate the cycle. In the case of ORCs therefore a surface heat exchanger is needed to regenerate. This device increases the complexity of the system and introduces two further pressure drops, on the liquid side and on the vapor side, reducing the net power output of the cycle, but increasing the cycle efficiency. In many cases, the regenerator can increase the global efficiency of the system, but in the case of WHR applications, if there are no constraints on the discharge temperature of the HTF, the regenerator reduces the system power output and therefore the efficiency of the system due to the pressure drops it causes. The use of this device is not recommended, due to the increase of system complexity, cost and to the performance reduction [44,45].

In the case of OFRC, instead the mixer and therefore the pressure drop caused by it, is just moved from the vapor side (typical of OFC) to the liquid side to operate the regeneration. The absence of a surface heat exchanger does not add any further cost to the system, with the exception of an extra pump and aims to reduce the main exchanger surface.

3. Methodology

The main equations, used to calculate the thermodynamic processes occurring in the OFRC, are reported in this section. Hot water in the range 80-170°C is assumed as HTF, in order to simulate a low temperature geothermal or waste heat recovery system. Similar considerations could be done for a heat recovery from exhaust gases or any other sensible heat source. The working fluids considered in this study allow a dry expansion and fluids properties were calculated by using the CoolProp library [46]. Exchangers heat loss to the surroundings was considered negligible.

For both the OFC and OFRC cycles, according to figs. 2 and 4, the heat exchanged between the HTF and the working fluid is calculated as:

$$\dot{Q}_{exch} = \dot{m}_{HTF}(h_{in} - h_{out}) = \dot{m}_{HTF}\bar{C}_{p_{HTF}}(T_{in} - T_{out}) = \dot{m}_{org}(h_2 - h_1) \quad (1)$$

Where h_{in} is the specific enthalpy of the HTF at the system inlet and h_{out} the specific enthalpy at the plant outlet.

The available heat is defined as:

$$\dot{Q}_{av} = \dot{m}_{HTF}(h_{in} - h_0) \quad (2)$$

Where h_0 is the specific enthalpy of the HTF fluid at ambient conditions.

The recovery efficiency was defined as the ratio between the exchanged heat and the available heat:

$$\varepsilon = \frac{\dot{Q}_{exch}}{\dot{Q}_{av}} \quad (3)$$

The global first law efficiency of the recovery system was obtained multiplying the cycle thermodynamic efficiency by the recovery efficiency:

$$\eta^I = \eta_c \cdot \varepsilon = \frac{\dot{W}_N}{\dot{Q}_{av}} \quad (4)$$

where η_c is the cycle efficiency and $\dot{W}_N$ is the net-power output of the system.

The exergy entering the system was computed as:

$$\dot{E}x_{av} = \dot{m}_{HTF}[(h_{in} - h_0) - T_0(s_{in} - s_0)] = \dot{m}_{HTF}\bar{C}_{p_{HTF}}\left[(T_{in} - T_{out}) - T_0 ln\frac{T_{in}}{T_0}\right] \quad (5)$$

Where s_{in} is the specific entropy of the HTF at the exchanger inlet and s_0 is the specific entropy of the HTF at ambient condition.

The exergydestruction in the generic n component of the systems can be computed as:

$$\dot{I}_n = T_0\left(\sum_i \dot{m}_i s_i - \sum_j \dot{m}_j s_j\right) \tag{6}$$

where the index i refers to the component inlets and the index j to the component outlets.

The exergy loss due to the release of the HTF at the exchanger outlet is computed as:

$$\dot{I}_{out} = \dot{m}_{HTF}[(h_{out} - h_0) - T_0(s_{out} - s_0)] = \dot{m}_{HTF}\bar{C}_{p_{HTF}}\left[(T_{out} - T_0) - T_0 ln\frac{T_{out}}{T_0}\right] \tag{7}$$

The second law recovery efficiency ε^{II}, i.e. the ratio between the exchanged exergy and the available exergy is:

$$\varepsilon^{II} = \frac{\dot{E}x_{exch}}{\dot{E}x_{av}} = \frac{\dot{m}_{org}[(h_2-h_1)-T_0(s_2-s_1)]}{\dot{m}_{HTF}[(h_{in}-h_0)-T_0(s_{in}-s_0)]} \tag{8}$$

The second law efficiency of the systems was then calculated according to the following expression:

$$\eta^{II} = \frac{\dot{W}_N}{\dot{E}x_{av}} = \frac{\dot{E}x_{av}-\sum_n \dot{I}_n-\dot{I}_{out}}{\dot{E}x_{av}} = \eta_C^{II}\cdot\varepsilon^{II} \tag{9}$$

3.1. Thermodynamic relations for OFCs

According to figs. 1 and 2, the main equations used in the thermodynamic analysis of the OFC are reported in this section.

The throttling processes were supposed to be isenthalpic: i.e. the enthalpy of point 3 is the same of point 2. The working fluid mass flow rate across the high pressure expander is:

$$\dot{m}_{H.P.} = \dot{m}_{org}\cdot x_3 \tag{10}$$

Where x_3 is the vapor quality at point 3 at the end of the throttling process. The range of this value is strongly dependent on the shape of the two-phase zone and the first flash pressure. The liquid mass flow rate separated inside the flash evaporator is:

$$\dot{m}'_{liq} = \dot{m}_{org} \cdot (1 - x_3) \tag{11}$$

At the outlet of the flash evaporator a throttling valve reduces the pressure of the liquid to the pressure of the vapor at the HP expander outlet, keeping the specific enthalpy constant. The two streams coming from the flash evaporator and from the HP expander are then mixed together in a mixing chamber according to the enthalpy balance:

$$\dot{m}_{org} h_8 = \dot{m}_{H.P.} h_5 + \dot{m}'_{liq} h_6 \tag{12}$$

The two-phase mixture resulting from the mixing process is then separated in a second flash drum. This separation is mandatory when operating with low temperature heat sources (<170-180°C) because of the low quality of the vapor at the mixer outlet, which does not allow an efficient expansion, neither when a positive displacement expander is used.

The vapor mass flow rate to the LP expander is:

$$\dot{m}_{L.P.} = \dot{m}_{org} \cdot x_8 \tag{13}$$

The calculation of the vapor quality at point 8 was carried out in a similar way as for the point 3, i.e. assuming the conservation of enthalpy during the throttling process (6-7).

The liquid fraction at the outlet of the second flash evaporator is:

$$\dot{m}''_{liq} = \dot{m}_{org} \cdot (1 - x_8) \tag{14}$$

The liquid is then throttled in a third valve and then mixed with the vapor coming from the LP expander according to the following enthalpy balance:

$$\dot{m}_{org} h_{13} = \dot{m}_{L.P.} h_{10} + \dot{m}''_{liq} h_{11} \tag{15}$$

The fluid at point 13 is then sent to the condenser. For low temperature applications, the point 13 is always located inside the two-phase region. From the condenser, the fluid is pumped into the heat exchanger. The feed pump work is calculated as:

$$\dot{W}_P = \dot{m}_{org} \frac{\Delta P}{\rho_{14} \cdot \eta_P} \quad (16)$$

Where η_P is the pump efficiency, ρ_{14} is the density of saturated liquid at point 14 and ΔP is the pressure difference between the condensing pressure and the cycle maximum pressure.

The work output of the HP expander was calculated as:

$$\dot{W}_{H.P.} = \dot{m}_{H.P.} \cdot (h_4 - h_{5is}) \cdot \eta_{is} \quad (17)$$

And in the LP expander:

$$\dot{W}_{L.P.} = \dot{m}_{L.P.} \cdot (h_9 - h_{10is}) \cdot \eta_{is} \quad (18)$$

Where h_{5is} and h_{10is} are the specific enthalpies along an isentropic process and η_{is} is the isentropic efficiency of the expanders. The value of the isentropic efficiency depends on the expander type. As stated above, since the Organic Flash Cycle provides good efficiencies only if two or more flash stages are used when recovering heat from low temperature heat sources, the cost of the expander should be the lowest possible, in order to have low overall costs. Positive displacement expanders can be the solution which allows to keep costs at low values. For this reason, the value of the isentropic efficiency was set constant at 0.7, typical value at design point, for positive displacement expanders with a built-in ratio under 5, as reported in [47]. The choice of this value is plausible for double flash systems since the volume ratio is very low and similar to the value of the maximum built in ratio.

The heat rejected at the condenser was calculated as:

$$\dot{Q}_{cond} = \dot{Q}_{exch} - \left(\dot{W}_{H.P.} + \dot{W}_{L.P.}\right) + \dot{W}_P \quad (19)$$

3.2. Thermodynamic Relations for OFRCs

The equations used for the high-pressure section of the OFRCs are the same used for OFCs. Differences are found downstream the second flash evaporator. In fact, the liquid from the second flash evaporator is not throttled in a valve but it is sent to a mixer to regenerate the liquid after the condenser, in order to increase the heat exchanger inlet temperature.

Referring to fig. 3 and 4 the mixer enthalpy balance is represented by the equation:

$$\dot{m}_{org}h_{14} = \dot{m}_{L.P.}h_{13} + \dot{m}''_{liq}h_{11} \tag{20}$$

A further pump is needed to raise the fluid pressure from the value at the condenser to the one at the second flash evaporator.

The global pumping work is calculated as:

$$\dot{W}_P = \dot{m}_{L.P.}\frac{(P_{11}-P_{cond})}{\rho_{12}\cdot\eta_P} + \dot{m}_{org}\frac{(P_2-P_1)}{\rho_{14}\cdot\eta_P} \tag{21}$$

Where ρ_{12} is the density of the saturated liquid at the condensing temperature and ρ_{14} is the density at the mixer outlet, evaluated at the second flash evaporator pressure and at the specific enthalpy resulting from the mixing process.

4. Thermodynamic analysis

In this section a thermodynamic comparison between the two cycles is carried out, in order to evaluate pros e cons of each cycle. Eight different organic fluids were tested on both cycles to define the optimum fluid to use in these cycles. A constant thermal input power of 900 kW was considered in all the analyzed cases. From the thermodynamic point of view, the performance of the system is independent from this last variable, being the isentropic efficiency of the device considered constant, as well as the exchanger inlet and outlet conditions. The choice of a constant thermal power input was due to the need of comparing the exchange surface of OFCs and OFRCs.

4.1. The maximum temperature of the cycles was set 10°C below the temperature of the HTF. For the non-regenerative cycle, the outlet temperature of the HTF was set at 35°C, i.e. at 8°C above the condensing temperature. The HTF outlet temperature of the regenerative cycle was also set at 8°C above the mixing temperature. Those values were chosen to ensure an optimal overall efficiency while keeping the costs of the heat exchangers at reasonable levels. A sensitivity analysis on the performances of both the two cycles related to different exchange conditions was reported at the end of the first section of this paragraph.Cycles comparison

The two cycles were compared, using N-Heptane as working fluid.

The second law efficiency of the two cycles was compared, by assuming 10 and 8°C for the approach and the pinch point, respectively. The two cycles showed the same trend (fig. 5) since the decrease in heat recovery efficiency (fig. 6) was practically counterbalanced by the increase in cycle efficiency (fig. 7). As reported in eq. 9, the global second law efficiency is given by the product of the recovery efficiency to the cycle efficiency. The first parameter is defined as the ratio between the actual exergy transferred to the working fluid and the available exergy from the heat source. The regeneration causes an increase of the HTF discharge temperature and therefore an increase of the exergy loss which reduces the recovery efficiency. Conversely, the cycle efficiency increases when operating the regeneration, due to the reduction of the effect of the multiplicity of the heat sources. The global effect is that the product of the recovery efficiency to the cycle efficiency is very similar. At temperatures lower than 110°C the OFRC presents a better efficiency than the OFC: in fact, due to the small temperature difference between the heat source and the lowest cycle temperature, the benefic effect of the reduction of the multiplicity of the heat sources on the cycle efficiency is dominant respect to the decrease of the recovery efficiency. At higher temperatures, instead, due to the larger temperature between the heat source and the lowest cycle temperature, the effect of the reduction of the recovery efficiency has a larger influence than the reduction of the multiplicity of the heat sources and therefore of the cycle efficiency. Due to this consideration, the OFC has a slight advantage with respect to the OFRC.

FIGURE 5 ABOUT HERE

FIGURE 6 ABOUT HERE

FIGURE 7 ABOUT HERE

A comparison between the exergy loss and destructions for the two cycles is shown in fig. 8 at the HTF inlet temperature of 80, 120 and 170°C. Exergy loss and destruction increase with the exergy content of the heat source, even if not proportionally: in fact, the second law efficiency increases with the temperature of the heat source.

FIGURE 8 ABOUT HERE

As expected the larger exergy loss in the regenerative system occurs when the HTF discharge temperature at the outlet of the heat exchanger is higher in comparison with the solution without regeneration.

All the other losses were smaller or comparable with those of the solution without regeneration.

In particular, the OFCs has the major losses in the condenser, in the throttling valves and in the heat exchangers:

- Condenser: the condenser of the OFCs has to exchange a larger thermal power than in the regenerative solution (fig. 9) and the whole mass flow rate of the cycle circulates in it;
- Throttling valves: the OFCs present three throttling process, one more than the regenerative cycle; the higher is the HTF inlet temperature the larger are these losses because of the higher maximum pressure of the cycle and of the larger pressure loss during the throttling processes;

- Heat exchanger: the temperature difference between the inlet and the outlet of the heat exchanger is larger in the OFCs than in the OFRCs and therefore the entropy production is larger.

The exergy analysis confirms that the two cycles have almost the same performance in the temperature range 80-170°C in terms of global efficiency and the larger loss in heat recovery efficiency of the regenerative cycle are compensated by the smaller losses in the cycles component. The regenerative cycle, however, provides the same power output but it requires smaller heat exchangers. The smaller size of the principal heat exchanger results from the lower maximum temperature difference between the inlet and the outlet of the exchanger (fig. 10), or, in other words, from the smaller amount of exchanged heat; The smaller size of the condenser, instead, is determined by the smaller amount of vapor to condensate because of the OFRC higher cycle efficiency. Table 1-4 reports the exchanger sizes and costs for the two considered cycles: exchangers were designed with the software Aspen Exchanger Design and Rating by minimizing the device cost, compatibly with the process requirements and avoiding dangerous working conditions (as an example tubes vibration). Shell and tube heat exchangers were considered.

FIGURE 9 ABOUT HERE

TABLE 1 ABOUT HERE

TABLE 2 ABOUT HERE

TABLE 3 ABOUT HERE

TABLE 4 ABOUT HERE

The specific cost of heat exchangers for unit power output is reported in fig. 10 for both OFCs and OFRCs, referring to an available thermal power of 900 kW, for each temperature.

FIGURE 10 ABOUT HERE

FIGURE 11 ABOUT HERE

Both cycles exhibit a high pump power consumption, since the maximum pressure is much higher than the expander inlet pressure.

Similarly to [48], the "Back-Work Ratio" (BWR) parameter, which accounts for the pump power consumption of the cycle, was introduced:

$$BWR = \frac{\sum_i W_{p_i}}{\sum_j W_{exp_j}} \tag{22}$$

where $\sum_i W_{p_i}$ is the sum of all pumps power consumption and $\sum_j W_{exp_j}$ is the sum of all expanders power output. This parameter strongly depends on the fluid properties. In this section n-Heptane was considered as working fluid. This fluid has a high critical temperature and is characterized by small saturation pressure variation at low temperature. Therefore, as long as the HTF temperature is low, the variation of the maximum pressure of the cycle is negligible and the BWR decreases, as a consequence of the increase of the optimal flash pressure. Increasing the HTF temperature, the saturation pressure of n-Heptane quickly increase, leading to a larger pump power consumption and to an increase of the cack-work ratio.

The BWR of the regenerative cycle is lower than the non-regenerative one at the maximum thermodynamic efficiency point (fig. 11). In the first case in fact, the compression is divided in two stages:

- the circulating pump increases the pressure of the cycle from the condensing pressure to the second flash evaporator pressure, and the mass flow rate in the pump is the same as the vapor fraction separated in the low-pressure flash evaporator;
- the feed pump increases the pressure of the whole mass flow rate of the cycle from the pressure of the second flash evaporator to the maximum pressure of the cycle.
- at the design point the first expander inlet pressure is higher in the OFRC than in the OFC: in fact the expander outlet temperature is higher in the OFRC than in the OFC, in order to increase the enthalpy of the second flash evaporator.

- In the OFCs the feed pump has to increase the whole mass flow rate of the cycle from the condensing pressure to the maximum pressure of the cycle.

Even though the OFRCs requires two pumps, the lower values of the Back Work Ratio can be achieved in smaller units, keeping costs similar to the ones of OFCs, as reported further in the work.

A sensitivity analysis on the effect of the temperature difference between the working fluid and the HTF at the heat exchanger outlet was carried out to evaluate the effect of the heat transfer curve distance on the system performances. The temperature difference between the HTF and the working fluid at the exchanger outlet modifies the exchanged thermal power and consequently the system efficiency, as reported in fig. 12 for both the OFRC and OFC cycles.

The results showed that the trend is almost the same for both cycles, however the efficiency decrease is larger for the OFRC at low temperature because of the higher temperature of the HTF at exchanger outlet.

These results, obtained with n-Heptane, can be considered a benchmark also for other fluids: in fact, the increase of the logarithmic mean temperature difference always causes a reduction of the system performance, due to the increase of the exergy destruction in the thermal exchange process and exergy loss in the higher HTF discharge temperature.

FIGURE 12 ABOUT HERE

4.2. Fluid comparison.

In this work, eight different fluids were tested as working fluid both for OFRCs and OFCs. The main properties of these fluids are reported in table 5. This analysis was performed assuming an approach temperature of 10 °C and a pinch point temperature of 8°C.

INSERT TABLE 5 ABOUT HERE

The differences in terms of second law efficiency are negligible for all the tested fluids (fig. 13), due to the reason discussed in the previous paragraphs regarding the trend of the recovery efficiency and the cycle efficiency. At higher temperatures, the second law efficiency with R245fa has a different trend with respect to that of all other fluids in both the analyzed cycles. This behavior is due to the critical point of R245fa, which is 154.1°C: increasing the temperature of the heat source above this value, the approach increases to avoid supercritical operation. In this way the exergy destruction in the heat transfer process increase due to the largest distance between the heat transfer curves. Alkane hydrocarbons with high molecular weight are the fluids which gave the best values of the second law efficiency. This fact is due to the close distance between the isobaric curves typical of these fluids: for a given pressure loss during the throttling process, the quality of the vapor is higher and it is demonstrated that the higher the quality of the vapor after the first throttling, the lower the exergy destruction in the throttling process, and the higher the efficiency of the flash cycle [35]. It is worth to notice that the fluids which present a high second law efficiency are those with a low value of the BWR.

In fact, exergy destruction during the first throttling process is proportional to the BWR: fluids with the highest efficiency values presents low values of pump power consumption. The value of BWR is much higher for OFCs with every used fluid (fig. 14), and from fluid to fluid the trend is different due to the different properties.

FIGURE 13 ABOUT HERE

FIGURE 14 ABOUT HERE

From the above analysis, high molecular weight alkanes are the fluids providing the best efficiency with both cycles. However, with the Double Flash Cycle, the presence of two expanders requires high accuracy in fluid selection in order to keep the size of the expanders as small as possible. Positive displacement expanders are the best choice for small heat recovery system with flash cycles. In fact, their cost is much lower than that of turbines. The size of these devices is influenced by the volume flow rate. The lower is the volume flow rate the lower is the cost of the expander [48]. For this sake, the volume flow rate for the high pressure and low pressure OFRC expanders was calculated, as shown in Fig. 15.

FIGURE 15 ABOUT HERE

As shown in fig. 15, the volume flow rate through the expander reaches very high values for high molecular weight alkanes, which makes their use practically impossible for this type of cycle. R601a, R245fa and R365mfc, in this order, are the fluids which present the lowest values of volume flow rate through the expanders. The trend in the cycles without regeneration is similar, but values are larger for each fluid. The bump in the figure at 90°C is due to the small variation of optimal flash temperature, and therefore of density, when the HTF varied from 80 to 90°C.

Another important device which can increase the cost of these cycles is the condenser. In fact, flash cycles introduce a larger amount of heat than traditional ORCs since they generally present a lower cycle efficiency. This results in a larger condenser than in ORC systems and therefore higher costs. For this reason, the working fluid should be selected to minimize the condenser size. In order to take into account the condenser size the ratio between the condensing latent heat and the vapor specific volume was used as a figure of merit, according to [49]. This figure of merit indicates approximately the amount of heat transferred by the condenser per unit hardware cost. The higher is its value, the larger is the amount of heat transferred per unit of volume, and the smaller is the condenser. The value for the various fluids is reported in table 6.

TABLE 6 ABOUT HERE

Alkanes with high molecular weight have a very low value of the figure of merit and therefore will require a large condenser. R245fa, i-Pentane and R365mfc are the fluids which minimize the condenser size.

In the considered temperature range, i-Pentane presents a low volume flow rate, requires a small condenser and does not require a too high pump power input, allowing smaller exchangers and expansion devices, although it has a lower cycle efficiency than other fluids: for these reasons this fluid might be the optimal working fluid for double OFRCs coupled with waste heat recovery or geothermal heat source in the temperature range between 80-170°C.

4.3. Cost Analysis

In this section the cost analysis of the main components of OFCs and OFRCs, operating with R601a is described, for various HTF temperature and for available amounts of heat.

Tables 7-10 reports the exchangers and condensers sizes for both OFCs and OFRCs when R601a is used, with 900 kW_{th} as input heat. Also in this case, the sizing of heat exchangers was carried out by using the software Aspen Exchanger Design and Rating. Both main exchanger and condenser required smaller exchanger area and therefore lower costs in the case of OFRCs than in the case of OFCs, and as demonstrated above, with this fluid, their sizes were smaller than the ones evaluated with N-Heptane.

TABLE 7 ABOUT HERE

TABLE 8 ABOUT HERE

TABLE 9 ABOUT HERE

TABLE 10 ABOUT HERE

As reported in the above tables, despite the small external diameter, heat exchangers for flash cycles can be very long, because of the large temperature difference between inlet and outlet. This results in transportation and layout problems which must be evaluated before the installation. A simpler and more compact plate heat exchanger however may be the solution to this problem.

The evaluation of pump costs was performed according the following relation, presented in [50]:

$$C_p = 900 \cdot \dot{W}^n \tag{23}$$

where W is the pump power consumption and n has the value of 0.25 if W<0.3 kW or 0.45 if W >0.3 kW.

Pumps costs are reported in table 11 for both cycle configurations.

TABLE 11 ABOUT HERE

Screw expanders were chosen because of their capacity of elaborating a high volume flow rate and for their limited costs. The costs of these expanders were evaluated from screw compressors in [51], referring to the Kore compressor manufacturer, as reported in fig. 16. To take into account of the different utilization of the device, i.e. the operation as an expander is the reverse of the operation of the compressor, the volume flow rate at the expander input was multiplied by the volume expansion ratio:

$$C_{exp} = 21.57 \cdot \left(V \dot{\cdot} r\right) + 3479 \tag{24}$$

where $\dot{V}$ is the expander flow rate.

FIGURE 16 ABOUT HERE

TABLE 12 ABOUT HERE

Expanders costs for OFCs and OFRCs are reported in tab. 12, for an available thermal power of 900kW$_{th}$. OFRCs requires smaller units than OFCs since the design flash pressure are normally higher than the ones of OFCs, because of the presence of the regenerator.

The comparison of the specific costs of the main component for both cycles is reported in fig17.

FIGURE 17 ABOUT HERE

4.4. Comparison with ORC systems

In this paragraph the regenerative flash cycle is compared with a conventional ORC for heat recovery or geothermal system. In the analysis, neither superheater nor recuperator was used in the ORC system.

FIGURE 18 ABOUT HERE

FIGURE 19 ABOUT HERE

Second law heat recovery efficiency is calculated according to the (3.9) always assuming a pinch point of 8 °C(fig. 18).

If we compare the efficiency of OFRC and ORC, the first cycle exhibited a better capability in the heat recovering process (fig. 19), thus entailing a lower HTF temperature at the outlet of the heat exchanger. Moreover, the exergy destruction during the heat transfer phase were lower than the ORC, because both fluids were at the liquid state, so that the heat transfer curves of both the fluids had a similar slope, on the contrary of an ORC plant in which the working fluid receives the heat partly at variable and partly at constant temperature.

Conversely, the ORC cycle presents smaller losses at the condenser because of the smaller amount of heat rejected, at the expander (ORCs just need a single expansion device), at the pump, because of the lower maximum pressure of the cycle, and because they do not require any throttling valve.

These facts are clearly counter-acting and the balance between advantages and disadvantages of both cycles depends on many factors, the main of which are the operating temperature of the cycle. In the analyzed temperature range, OFRCs provided better thermodynamic performance than ORCs.

The economic comparison between OFRCs and ORCs in this temperature range and for different available heat amounts was carried out using R601a for OFRCs and R245fa for ORCs, since this latter gave the best results in the thermodynamic analysis.

The ORC evaporators and condensers details are reported in table 13 and 14, for an available thermal power input of 900 kW_{th}. Similarly to OFRCs, a pinch point of 8 °C was used in the evaporator design.

TABLE 13 ABOUT HERE

TABLE 14 ABOUT HERE

ORC evaporators are more compact than the OFRCs ones, but the presence of vapor requires higher crossing areas and therefore exchangers have larger diameters.

Pump and expander costs are evaluated through the 5.1 and 5.2 respectively and are reported in table 15.

TABLE 15 ABOUT HERE

Specific costs, at optimal thermodynamic conditions, are reported in fig. 20 for OFRCs and for ORCs for various available heat.

FIGURE 20 ABOUT HERE

The two cycles presented almost the same main components specific costs in the heat source temperature range between 120 and 170°C and for available thermal power lower than 600 kW. For larger available thermal power OFRC cost increased more than the ORC cost, because of the need of an extra expander unit in the second stage, because of the increase of the volume flow rate which goes outside the range of the screw expanders costs.

This result is highly influenced by the future behavior of screw expanders cost. This analysis was carried out, in the favorable case of a growth of the positive displacement expanders market, in order that their cost will be similar to compressor ones.

Moreover, in this work shell and tube heat exchangers were analyzed: for flash cycle these devices are very long, and they can produce layout and transportation problems. Plate heat exchangers can be more compact and economic, and therefore more suitable for these applications and allowing a further costs reduction.

5. Conclusions

In this work, a new type of Organic Flash Regenerative Cycle (OFRC) has been studied for low temperature waste heat recovery and geothermal applications. Since two flash stages are needed with low temperature heat sources, positive displacement expanders were assumed to be used because of their cost lower than that of turbines. With respect to the conventional Organic Flash Cycle (OFC), the OFRC presented almost the same thermodynamic performances for heat sources in the temperature range between 80°C and 170°C: in fact, if from one hand the recovery efficiency of the OFRC was lower than the OFC, from the other the higher cycle efficiency compensated the recovery efficiency, resulting in the same cycle efficiency. The regenerator increased the working fluid temperature at the exchanger inlet, reducing the size and the cost of the heat exchangers. Moreover, due to the increase in cycle efficiency and to the smaller amount of exchanged heat, OFRCs condenser is smaller and cheaper than OFCs.

Different organic fluids have been tested both in the OFRC and the OFC. With the same fluid, the two cycles provided the same thermodynamic performance. High molecular weight alkanes gave the best results in terms of second law efficiency both in the OFRC and the OFC, due to the small distance between the isobaric lines and therefore to the low exergy destruction during the throttling process. However, the high volume flow rate through the two stage expanders made the use of these fluids unfeasible if positive displacement expanders are adopted. Among the tested fluids, R601a was the best option in terms of thermo-economic performances. Finally, a comparison with ORC cycles has been carried out and OFRCs demonstrated a better thermodynamic performance for all the tested fluids, because of the lower losses across the heat exchanger and to the lower HTF discharge temperature. The economic analysis highlighted the equality between those two technologies, for mini and micro scale plants, in the case of a future growth of the positive displacement market.

6. References

[1] G. Angelino, E. Macchi, M. Gaia, A review of Italian activities in the field of Organic Rankine Cycles, VDI Berichte. (1984) 465-482.

[2] B. Liu, K. Chien, C. Wang, Effect of working fluids on organic Rankine cycle for waste heat recovery, Energy. 29 (2004) 1207-1217.

[3] L. Jing, P. Gang, J. Jie, Optimization of low temperature solar thermal electric generation with Organic Rankine Cycle in different areas, Applied Energy. 87 (2010) 3355-3365.

[4] G. Pei, J. Li, J. Ji, Analysis of low temperature solar thermal electric generation using regenerative Organic Rankine Cycle, Applied Thermal Engineering. 30 (2010) 998-1004.

[5] J. Roy, M. Mishra, A. Misra, Parametric optimization and performance analysis of a waste heat recovery system using Organic Rankine Cycle, Energy. 35 (2010) 5049-5062.

[6] A. Lakew, O. Bolland, Working fluids for low-temperature heat source, Applied Thermal Engineering. 30 (2010) 1262-1268.

[7] A. Delgado-Torres, L. García-Rodríguez, Analysis and optimization of the low-temperature solar organic Rankine cycle (ORC), Energy Conversion And Management. 51 (2010) 2846-2856.

[8] S. Quoilin, R. Aumann, A. Grill, A. Schuster, V. Lemort, H. Spliethoff, Dynamic modeling and optimal control strategy of waste heat recovery Organic Rankine Cycles, Applied Energy. 88 (2011) 2183-2190.

[9] R. Rayegan, Y. Tao, A procedure to select working fluids for Solar Organic Rankine Cycles (ORCs), Renewable Energy. 36 (2011) 659-670.

[10] D. Hu, Y. Zheng, Y. Wu, S. Li, Y. Dai, Off-design performance comparison of an organic Rankine cycle under different control strategies, Applied Energy. 156 (2015) 268-279.

[11] T. Nguyen, J. Slawnwhite, K. Boulama, Power generation from residual industrial heat, Energy Conversion And Management. 51 (2010) 2220-2229.

[12] S. Stecco, U. Desideri, Thermodynamic analysis of the Kalina cycles: Comparisons, problems and perspectives, (1989) American Society of Mechanical Engineers (ASME).

[13] S. Stecco, U. Desideri, Considerations on the Design Principles for a Binary Mixture Heat Recovery Boiler, J. Eng. Gas Turbines Power. 114 (1992) 701.

[14] U. Desideri, G. Bidini, Study of possible optimisation criteria for geothermal power plants, Energy Conversion And Management. 38 (1997) 1681-1691.

[15] P. Nag, A. Gupta, Exergy analysis of the Kalina cycle, Applied Thermal Engineering. 18 (1998) 427-439.

[16] P. Bombarda, C. Invernizzi, C. Pietra, Heat recovery from Diesel engines: A thermodynamic comparison between Kalina and ORC cycles, Applied Thermal Engineering. 30 (2010) 212-219.

[17] X. Wang, L. Zhao, Analysis of zeotropic mixtures used in low-temperature solar Rankine cycles for power generation, Solar Energy. 83 (2009) 605-613.

[18] R. Victor, J. Kim, R. Smith, Composition optimisation of working fluids for Organic Rankine Cycles and Kalina cycles, Energy. 55 (2013) 114-126.

[19] H.C. Jung, S. Krumdieck, Analysis of zeotropic mixture in a Geothermal Organic Rankine Cycle Power Plant with an air-cooled condenser, Proceeding of the 35th New Zealand Geothermal Workshop (2013).

[20] F. Heberle, M. Preißinger, D. Brüggemann, Zeotropic mixtures as working fluids in Organic Rankine Cycles for low-enthalpy geothermal resources, Renewable Energy. 37 (2012) 364-370.

[21] G. Angelino, P. Colonna Dipaliano, Multicomponent Working Fluids For Organic Rankine Cycles (ORCs), Energy. 23 (1998) 449-463.

[22] Q. Liu, Y. Duan, Z. Yang, Effect of condensation temperature glide on the performance of organic Rankine cycles with zeotropic mixture working fluids, Applied Energy. 115 (2014) 394-404.

[23] H. Chen, D. Goswami, M. Rahman, E. Stefanakos, A supercritical Rankine cycle using zeotropic mixture working fluids for the conversion of low-grade heat into power, Energy. 36 (2011) 549-555.

[24] H. Chen, D. Goswami, E. Stefanakos, A review of thermodynamic cycles and working fluids for the conversion of low-grade heat, Renewable And Sustainable Energy Reviews. 14 (2010) 3059-3067.

[25] Y. Baik, M. Kim, K. Chang, S. Kim, Power-based performance comparison between carbon dioxide and R125 transcritical cycles for a low-grade heat source, Applied Energy. 88 (2011) 892-898.

[26] Y. Chen, P. Lundqvist, P. Platell, Theoretical research of carbon dioxide power cycle application in automobile industry to reduce vehicle’s fuel consumption, Applied Thermal Engineering. 25 (2005) 2041-2053.

[27] Y. Chen, P. Lundqvist, A. Johansson, P. Platell, A comparative study of the carbon dioxide transcritical power cycle compared with an organic Rankine cycle with R123 as working fluid in waste heat recovery, Applied Thermal Engineering. 26 (2006) 2142-2147.

[28] X. Zhang, H. Yamaguchi, K. Fujima, M. Enomoto, N. Sawada, Study of solar energy powered transcritical cycle using supercritical carbon dioxide, International Journal Of Energy Research. 30 (2006) 1117-1129.

[29] X. Zhang, H. Yamaguchi, K. Fujima, M. Enomoto, N. Sawada, Theoretical analysis of a thermodynamic cycle for power and heat production using supercritical carbon dioxide, Energy. 32 (2007) 591-599.

[30] X. Zhang, H. Yamaguchi, D. Uneno, K. Fujima, M. Enomoto, N. Sawada, Analysis of a novel solar energy-powered Rankine cycle for combined power and heat generation using supercritical carbon dioxide, Renewable Energy. 31 (2006) 1839-1854.

[31] I. Smith, Development of the trilateral flash cycle system. part 1: fundamental considerations, ARCHIVE: Proceedings Of The Institution Of Mechanical Engineers, Part A: Journal Of Power And Energy 1990-1996 (Vols 204-210). 207 (1993) 179-194.

[32] I. Smith, R. da Silva, Development of the trilateral flash cycle system Part 2: increasing power output with working fluid mixtures, ARCHIVE: Proceedings Of The Institution Of Mechanical Engineers, Part A: Journal Of Power And Energy 1990-1996 (Vols 204-210). 208 (1994) 135-144.

[33] C. Zamfirescu, I. Dincer, Thermodynamic analysis of a novel ammonia–water trilateral Rankine cycle, Thermochimica Acta. 477 (2008) 7-15.

[34] T. Ho, S. Mao, R. Greif, Comparison of the Organic Flash Cycle (OFC) to other advanced vapor cycles for intermediate and high temperature waste heat reclamation and solar thermal energy, Energy. 42 (2012) 213-223.

[35] T. Ho, S. Mao, R. Greif, Increased power production through enhancements to the Organic Flash Cycle (OFC), Energy. 45 (2012) 686-695.

[36] T. Ho, Advanced Organic Vapor Cycles for Improving Thermal Conversion Efficiency in Renewable Energy Systems, PhD Thesis, University of California, Berkeley, 2012.

[37] M. Antonelli, A. Baccioli, M. Francesconi, U. Desideri, L. Martorano, Operating maps of a rotary engine used as an expander for micro-generation with various working fluids, Applied Energy. 113 (2014) 742-750.

[38] S. Quoilin ,S. Declaye, A. Legros, L. Guillaume, V. Lemort, Working fluid selection and operating maps for Organic Rankine Cycles expansion machines. International Compressor Engineering Conference at Purdue (2012).

[39] S. Hsu, H. Chiang, C. Yen, Experimental Investigation of the Performance of a Hermetic Screw-Expander Organic Rankine Cycle, Energies. 7 (2014) 6172-6185.

[40] M. Ali Tarique, I. Dincer, C. Zamfirescu, Experimental investigation of a scroll expander for an organic Rankine cycle, International Journal Of Energy Research. 38 (2014) 1825-1834.

[41] M. Antonelli, A. Baccioli, M. Francesconi, L. Martorano, Experimental and Numerical Analysis of the Valve Timing Effects on the Performances of a Small Volumetric Rotary Expansion Device, Energy Procedia. 45 (2014) 1077-1086.

[42] M. Orosz, A. Mueller, B. Dechesne, H. Hemond, Geometric Design of Scroll Expanders Optimized for Small Organic Rankine Cycles, J. Eng. Gas Turbines Power. 135 (2013) 042303.

[43] S. Declaye, S. Quoilin, V. Lemort, Design and Experimental Investigation of a Small Scale Organic Rankine Cycle Using a Scroll Expander (2010). International Refrigeration and Air Conditioning Conference. Paper 1153.

[44] M. Yari, S. Mahmoudi, A thermodynamic study of waste heat recovery from GT-MHR using organic Rankine cycles, Heat And Mass Transfer. 47 (2010) 181-196.

[45] S. Lecompte, H. Huisseune, M. van den Broek, B. Vanslambrouck, M. De Paepe, Review of organic Rankine cycle (ORC) architectures for waste heat recovery, Renewable And Sustainable Energy Reviews. 47 (2015) 448-461.

[46] http://www.coolprop.org

[47] M. Read, N. Stosic, I.K. Smith. Optimization of Screw Expanders for Power Recovery from Low-Grade Heat Sources. Energy Technology & Policy. 1 (2014) 131-142.

[48] S. Quoilin, M. Broek, S. Declaye, P. Dewallef, V. Lemort, Techno-economic survey of Organic Rankine Cycle (ORC) systems, Renewable And Sustainable Energy Reviews. 22 (2013) 168-186.

[49] Power Plant Waste Heat Rejection Using Dry Cooling Towers, Report n° EPRI CS-1324-SY, Electric Power Research Institute, February 1980.

[50] S. Quoilin, S. Declaye, B. Tchanche, V. Lemort, Thermo-economic optimization of waste heat recovery Organic Rankine Cycles, Applied Thermal Engineering. 31 (2011) 2885-2893.

[51] www.compressorworld.com