SIZING AND PARAMETRIC OPTIMIZATION OF A WASTE HEAT TO POWER PLANT BASED ON TRANSCRITICAL ORGANIC RANKINE CYCLE

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ABSTRACT

In recent years, the transcritical (also called supercritical) organic Rankine cycle (Trans-ORC) has more and more aroused the attention for power generation from low-temperature heat source thanks to the better thermal matching between the heating and the cooling fluids of the high pressure heat transfer process. The absence of isothermal evaporation in transcritical ORC enables the heat source to be cooled down to a lower temperature despite an identical pinch point as in a comparable subcritical cycle. This leads to a greater utilization of the heat source. In other words, the transcritical cycle could produce more power with higher exergy efficiency compared to the subcritical one.

This paper aims at sizing a transcritical ORC to recover energy from a cooling circuit of turbine exhaust gas and optimizing the operating conditions of the cycle from a thermoeconomic point of view. Indeed, several potential organic fluids, which satisfy the screening criteria (e.g. safety, environment, thermophysical properties and availability), will be used as working fluid to evaluate the performance and the specific investment capital (SIC) of the plant. The operating conditions (i.e. turbine inlet temperature and pressure, condensing temperature and heat sink outlet temperature) of the plant will also be optimized to find out the most suitable working fluid for the studied cycle.

1. INTRODUCTION

Capturing and converting industrial waste heat into electricity is currently of a great concern. This not only improves energy efficiency of the industrial processes, but also reduces the thermal pollution caused by direct release of this heat into the environment. Over recent years, the transcritical ORC has aroused more and more attention for power generation from low-temperature heat source thanks to its advantages such as higher performance and less irreversibility. In a base Trans-ORC, the working fluid is first compressed above its critical pressure from sub-cooled liquid state (in the present work, the working fluid is assumed exiting the condenser in saturated liquid state). The working fluid is then heated, without isothermal phase change, by an external heat source (e.g. pressurized hot water) in high-pressure heat exchanger. In next step, the working medium releases its energy to drive the turbine blades during expansion process. The working fluid exits the turbine at superheated state, and is cooled down and then condensed by ambient air (dry cooling) or cooling water (wet cooling) before being pumped again to high pressure.

This work aims at sizing and then optimizing a small scale Trans-ORC to recover energy from a cooling circuit of turbine exhaust gas of a CHP (Combined Heat and Power) plant from landfill gas. According to the power range, the ORC systems could be categorized into (Vanslambrouck et al., 2011): micro-systems (0.5 – 10 kWel), small systems (10 – 100 kWel), medium systems (100 – 300 kWel) and large systems (300 – 3000 kWel or more). The process flow diagram and heat source and sink parameters for studied Trans-ORC are presented in Figure 1.
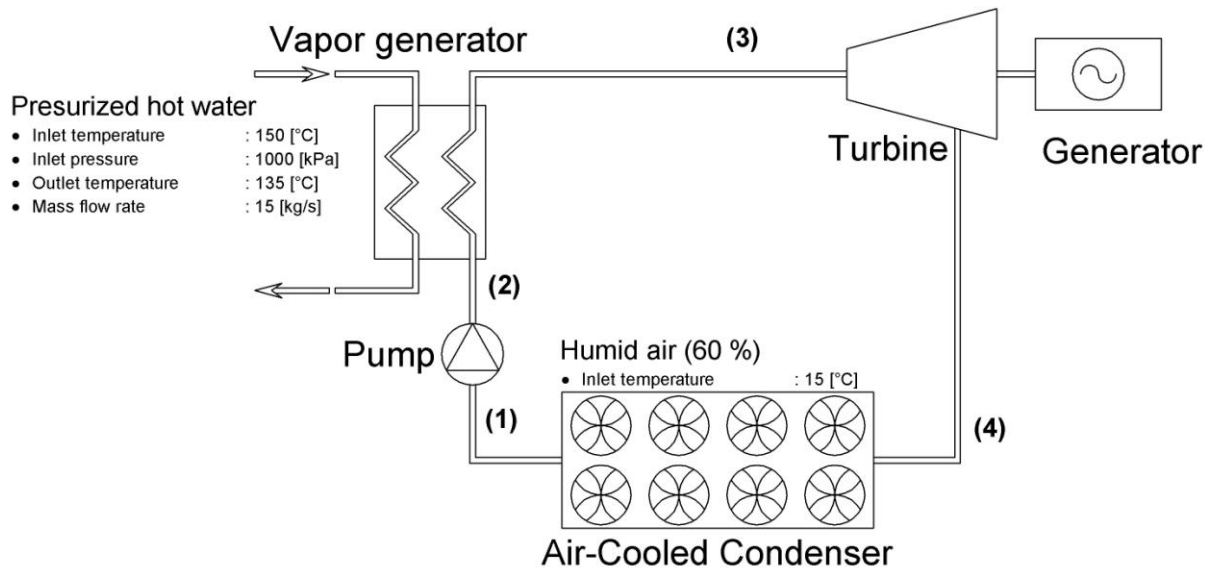
Figure 1: Process flow diagram

2. WORKING FLUID PROPERTIES

 In practice, R134a and CO_2 are often used as working fluid for Trans-ORC. Although CO_2 has many desirable characteristics such as high thermal and chemical stability, low environmental impact, low cost and abundance in nature, the 80-160 bars operating pressure of the CO_2 transcritical cycle raises safety concerns. Therefore this medium is not taken into account in this study. Indeed, many factors could influence the selection of working fluid for any particular application including the ORC one. Many criteria have been proposed to find the best balance between working fluid properties, system architecture and application environment (cf. Figure 1).
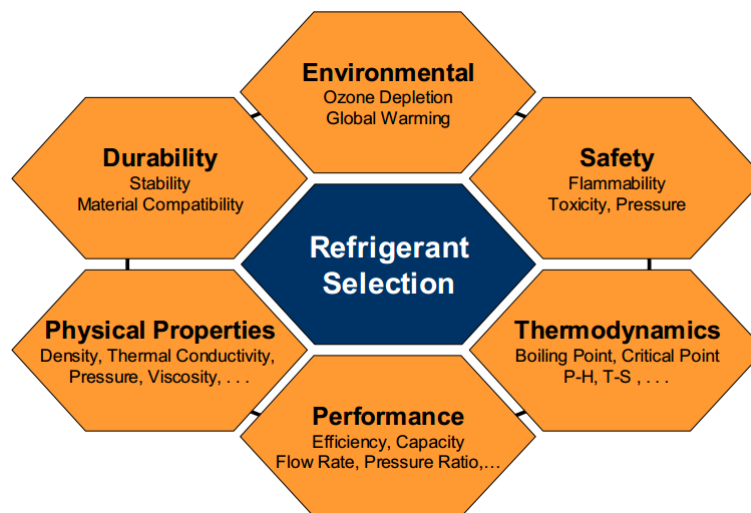

Figure 2: Factors influencing working fluid selection (Horn, 2011)

Some potential pure fluids, whose properties are available in the REFPROP database, are investigated as working medium. Those fluids satisfy selection criteria, i.e. GWP less than or equal to R134a GWP, critical temperature less than 115°C, maximum applicable temperature greater than 150°C, zero ODP, non-toxic and acceptable critical pressure. The thermophysical properties of the selected working fluids are computed by REFPROP (Lemmon et al., 2013) using Coolprop-EES interface (Bell et al., 2014). Some practical data of those working fluids are presented in Table 1.

Table 1: Working fluid data

Working fluid	R134a	R152a	R32	Propane (R290)	R1234ze(E)
Molecular weight	102.03 g/mol	66.05 g/mol	52.02 g/mol	42.08 g/mol	114.04 g/mol
T_c	101.1 $^{\circ}$C	113.3 $^{\circ}$C	78.1 $^{\circ}$C	96.7 $^{\circ}$C	109.4 $^{\circ}$C
P_c	4059 kPa	4517 kPa	5782 kPa	4251 kPa	3635 kPa
T_b	-26.1 $^{\circ}$C	-24.1 $^{\circ}$C	-51.7 $^{\circ}$C	-42.1 $^{\circ}$C	-19.3 $^{\circ}$C
ρ_c	511.9 kg/m^3	368.0 kg/m^3	424.0 kg/m^3	220.5 kg/m^3	489.2 kg/m^3
GWP	1370	133	716	< 20	6
Toxicity/Flammability**	A1	A2	A2L	A3	A2L
Materials	Good	Good	Good	Good	Good
Price (estimated)	6.85 \$/kg	3 \$/kg	3 \$/kg	1 \$/kg	37 \$/kg
Thermal conductivity*	0.01394 W.m^{-1}.K^{-1}	0.01421 W.m^{-1}.K^{-1}	0.01429 W.m^{-1}.K^{-1}	0.01832 W.m^{-1}.K^{-1}	0.008786 W.m^{-1}.K^{-1}
Density*	4.258 kg/m^3	2.759 kg/m^3	2.153 kg/m^3	1.832 kg/m^3	4.774 kg/m^3
* Computed at temperature (25°C) and pressure (1 bar)					
**ASHRAE classification : A: lower toxicity; B : higher toxicity ; 1: no flame propagation ; 2 : lower flammability ; 3 : higher flammability; 2L: lower flammability with a maximum burning velocity of < 10 cm/s					

3. EQUIPMENT SIZING AND CAPITAL COST ESTIMATION

3.1. Equipment purchase cost estimation

This work uses cost correlations as presented in (Seider et al., 2010) for estimating free-on-board purchase cost of equipment. The base cost of major equipment is estimated using the following

$$C_P^0 = e^{\{A_0 + A_1[ln(S) + A_2[ln(S)]^2 + ...\}}$$
(1)

Where

C_P^0 is base cost of equipment, US\$

A_0, A_1, A_2, are the estimation model constants
S is size factor or capacity of equipment

In addition to size, other factors, such as material of construction and operating pressure, are also taken into consideration for estimating equipment purchased cost. Therefore, the cost equations are often in the following form

$$C_P = F_M F_p C_P^0$$
(2)

Where

C_P is the purchase cost of equipment, US\$
F_M is the material factor
Fp is the pressure factor

The values of constants for base cost estimation, i.e. A_0, A_1, A_2, … in equation (1), the material and pressure factors for estimating the purchase cost of centrifugal pump, shell and tube heat exchanger and fans are found in (Seider et al., 2010). To account for the inflation, Chemical Engineering Plant Cost Index (CEPCI), published in each monthly issue of the magazine Chemical Engineering, is used. As the cost correlations in (Seider et al., 2010) were developed using a CEPCI of 500, current purchase cost of equipment should be determined by

$$C_P = C_P^{500} \frac{\text{CEPCI}}{500}$$
(3)

Where

C_P^{500} is equipment purchase cost determined with CEPCI of 500

CEPCI is the present Chemical Engineering Plant Cost Index (CEPCI = 567.3 for the year 2013)

3.2. Plant capital investment

Besides purchase costs of equipment, many other costs must be accounted for plant capital investment. In this work, the total capital investment is determined to be the sum of components costs as shown in Table 2

$$C_{TCI} = \sum C_P + C_{wf} + C_{sp} + C_{sf} + C_{CCF} \tag{4}$$

Table 2: Components of total capital investment

Component	Description
Total equipment purchase cost (excluding installation costs)	$\sum C_P$
Cost for working fluid	C_{wf}
Total cost for spares, C_{spares}	Assumed to be zero
Total cost for computer, software and associated items, C_{comp}	Neglected
Cost of site preparation, C_{sp}	5% of $\sum C_P$
Cost of service facilities, C_{sf}	5% of $\sum C_P$
Cost of contingencies and contractor's fee, C_{CCF}	18 % of ($\sum C_P + C_{sp} + C_{sf}$)
Cost of land, C_{land}	Assumed to be zero
Cost of royalties, $C_{royalties}$	Assumed to be zero
Cost of plant startup, $C_{startup}$	Assumed to be zero

3.3. Equipment sizing

For estimating the purchase cost of equipment, the size or capacity of the latter must be determined. Therefore, basic equations for equipment sizing and purchase cost estimation of some other ORC components (excluding the equipment covered in previous section, i.e. shell and tube heat exchanger, centrifugal pump and fan) are presented below.

Working fluid feed pump

A Centrifugal pump, of the radial type and with an isentropic efficiency of 70%, is used to raise the pressure of Trans-ORC working fluid. The shaft power of pump is computed as

$$\dot{W}_p = \dot{m}_{wf} \left(h_2 - h_1 \right) \tag{5}$$

The pump motor power consumption is calculated from the pump shaft power in assuming a motor efficiency (η_{motor}) to be 95%

$$\dot{W}_m = \dot{W}_p / \eta_{motor} \tag{6}$$

Vapor generator

A general shell and tube heat exchanger with one shell pass and two tube passes is used for transferring heat from pressurized hot water to ORC working fluid. The heat rate transferred in the vapor generator is calculated as

$$\dot{Q}_h = \dot{m}_{wf} \left(h_3 - h_2 \right) \tag{7}$$

The heat transfer coefficient and pressure drop of fluid flowing in heat exchanger shell side (i.e. hot water) are calculated using Kern method (Kern, 1950), while the supercritical heat transfer is computed by Jackson correlation (Jackson, 2002) with the Nusselt number determined as follows

$$\mathrm{Nu} = 0.0183 \times \mathrm{Re}^{0.82} \times \mathrm{Pr}^{0.5} \times \left(\rho_w / \rho \right)^{0.3} \times \left(\bar{C}_p / C_p \right)^n \tag{8}$$

Indeed, Jackson correlation was obtained by modifying the correlation of Krasnoshchekov et al. which is valid within the following range (Krasnoshchekov et al., 1967):

$$8 \cdot 10^4 < \mathrm{Re} < 5 \cdot 10^5, \; 0.85 < \mathrm{Pr} < 65, \; 0.90 < \rho_w / \rho < 1.0, \; 0.02 < \bar{C}_p / C_p < 4.0, \; 0.9 < T_w / T_{pc} < 2.5$$

The index n in equation (8) depends on the values of wall tube temperature, T_w, and temperature of supercritical fluid in relation to its pseudo-critical temperature, T_{pc}, as mentioned in (Jackson, 2002). Jackson (2002) and previously, Jackson and Hall (1979) assessed the accuracy of many correlations. They found the correlation of Krasnoshchekov et al. (1967) and its modified version of Jackson (2002) to be the most accurate ones. Ninety seven percent of the experimental data were correlated with the accuracy of $\pm 25\%$ (Pioro and Duffey, 2007). The pressure drop of the supercritical fluid

inside the tubes is calculated as presented in (Le, 2014). Neglecting the fouling resistance, the overall heat transfer coefficient, U, of the shell and tube heat exchanger is computed as

$$\frac{1}{U} = \frac{1}{\alpha_o} + \frac{D_o}{D_i \alpha_i} + \frac{D_o \ln(D_o / D_i)}{2k_w} \tag{9}$$

Condenser

The heat rejection step of Trans-ORC is not realized by a shell and tube heat exchanger like for the heat addition but an air-cooled condenser (ACC). In an ACC, the hot process fluid flows through a bank of tubes, and ambient air is blown across the finned tubes by fans. The heat transferred from ORC working fluid to ambient air via condenser is determined as follows

$$\dot{Q}_c = \dot{m}_{wf}(h_4 - h_1) \tag{10}$$

The air-cooled condenser is considered with horizontal finned tube bundle. The air side heat transfer and pressure drop of ACC are calculated by the correlations of Ganguli (Ganguli et al., 1985). For tube side, Gnielinski correlation (Gnielinski, 1976) is used to calculate the heat transfer of single phase fluid, while Shah correlation (Shah, 2009) is used for the condensation in plain tube. According to the empirical correlation of Shah, there are two heat transfer regimes during the condensation in plain tube. The equations of two-phase heat transfer coefficient, α_{TP}, for each regime are presented below

In regime I (turbulent regime)

$$\alpha_{TP} = \alpha_I \tag{11}$$

In regime II (mixed regime)

$$\alpha_{TP} = \alpha_I + \alpha_{Nu} \tag{12}$$

Where

$$\alpha_I = \alpha_{LS}\left(1 + 3.8 / Z^{0.95}\right)\left[\mu_l / (14\mu_v)\right]^{(0.0058 + 0.557 p_r)} \tag{13}$$

$$\alpha_{Nu} = 1.32 \, \mathrm{Re}_{LS}^{(-1/3)}\left[\frac{\rho_l(\rho_l - \rho_v) g (k_l)^3}{(\mu_l)^2}\right]^{1/3} \tag{14}$$

The boundary between the regimes I and II is given by the following relation. Regime I occurs when

$$J_v \geq 0.98(Z + 0.263)^{-0.62} \tag{15}$$

Where J_v is the dimensionless vapor velocity defined as (Shah, 2009):

$$J_v = \frac{xG}{\left[gD\rho_v(\rho_l - \rho_v)\right]^{0.5}} \tag{16}$$

The Shah's correlating parameter Z is calculated as (Shah, 2009):

$$Z = (1/x - 1)^{0.8} p_r^{0.4} \tag{17}$$

The two-phase pressure drop of the fluid flowing inside the horizontal tube is the sum of two contributions (the momentum and the frictional pressure drop) and computed as described in (Le et al., 2014). The purchase cost of air-cooled condenser is estimated from its heat transfer surface area as in the following equation:

$$C_{P,ACC} = 6466(A_{ACC})^{0.4} \, \mathrm{CEPCI} / 500 \tag{18}$$

Fan motor

The fan motor power consumption is estimated by the equation (19) from pressure drop and mass flow rate of the air through the ACC. The fan efficiency, η_{fan}, and motor efficiency, η_{motor}, in equation (19) are assumed to be equal to 70% and 95%, respectively.

$$\dot{W}_{fan} = \frac{\Delta P_{air} \dot{m}_{air}}{\rho_{air} \eta_{fan} \eta_{motor}} \tag{19}$$

Turbine

In the present work, the isentropic efficiency of turbine is assumed to be 85%. The shaft Power output of the turbine, where heat losses are neglected, is determined as

$$\dot{W}_t = \dot{m}_{wf} \left(h_3 - h_4 \right) \tag{20}$$

The purchase cost of turbine is determined as a function of its shaft power output (Seider et al., 2010):

$$C_{P,t} = 672.2 \left(\dot{W}_t \right)^{0.81} \text{CEPCI} / 500 \tag{21}$$

Electrical generator

The electrical power output of generator is determined from the turbine power output with a generator efficiency of 95%

$$\dot{W}_{gen} = \dot{W}_t / \eta_{gen} \tag{22}$$

The purchase cost of electrical generator is calculated by the equation as presented in (Pierobon et al., 2013)

$$C_{P,gen} = 60 \left(\dot{W}_{gen} \right)^{0.95}$$

Working fluid

In addition to the cost of equipment, the cost of working fluid must also be accounted for. In this work, the amount of working fluid is estimated as the liquid amount to fill the total volume of heat exchangers through which the working medium passes (Herrmann Rodrigues et al., 2010). The prices of tested working fluids were assumed from market surveys as shown in Table 1.

Global system

The net power output of Trans-ORC is considered as the difference between the power output of generator and the power consumption of the pump and fan motors

$$\dot{W}_{net} = \dot{W}_{gen} - \dot{W}_{motor} - \dot{W}_{fan} \tag{23}$$

The thermal efficiency of Trans-ORC is calculated as the ratio between system net power output to the heat transfer rate within vapor generator

$$\eta_{ORC} = \dot{W}_{net} / \dot{Q}_h \tag{24}$$

4. OPTIMIZATION

The optimization is performed by genetic method implemented in EES software (Klein, 2013). The parameters for the optimization are presented in Figure 1. The objective function is the specific investment capital (SIC) calculated hereunder:

$$\text{SIC} = C_{TCI} / \dot{W}_{net} \tag{25}$$

Many variables are taken into consideration for the optimization. In addition to the variable of operating conditions of the Trans-ORC, i.e. turbine inlet temperature (TIT); reduced pressure which is the ratio of turbine inlet pressure (TIP) to the critical pressure (P_c) of working fluid; condensing temperature and heat sink outlet temperature, the other variables concerning the shell and tube heat exchanger and the air-cooled condenser are also taken into consideration (cf. Table 3).

Table 3: Variables for SIC minimization

Variables	Lower	Upper
TIT	-	140 $^{\circ}$C
$p_r = \text{TIP}/P_c$	1.01	3
T_{cond}	25 $^{\circ}$C	50 $^{\circ}$C
T_{cso}	18 $^{\circ}$C	30 $^{\circ}$C
ACC tube pass	1	4
Fluid velocity inside condenser and vapor generator tubes	0.15 m/s	3 m/s
Baffle spacing ratio = baffle spacing/shell diameter	0.2	1.0

During the optimization, the pinch (minimum temperature difference between heating and cooling curve) of heat transfer processes must be greater than 1°C, maximum pressure drop of fluids through heat exchangers (vapor generator and condenser) must be lower than 0.5 bar and vapor quality, *x*, of fluid exiting the turbine, calculated by the equation (26), must be greater than or equal to unity.

$$x = \left(h - h_{sat,l}\right) / \left(h_{sat,v} - h_{sat,l}\right) \tag{26}$$

5. RESULTS - DISCUSSION

According to the optimization results (Table 4), the lowest SIC (2641 $/kWel) is found with the system using R152a, while the highest one (3155 $/kWel) is observed with R32. Indeed, the R152a-based Trans-ORC system presents many advantages, i.e. moderate investment capital, highest net power output and thermal efficiency, moderate operating pressure and low working fluid GWP. One thing can limit the use of R152a is the fact that this fluid is mildly flammable.

Table 4: Results of SIC optimizations

Working fluid	R134a	R152a	R32	Propane (R290)	R1234ze(E)
$\dot{W}_{net}$	88.7 kWel	102.9 kWel	93.1 kWel	89.3 kWel	95.6 kWel
η_{ORC}	9.2 %	10.7 %	9.7 %	9.3 %	9.9 %
TIT	140.0 ^{o}C	139.6 ^{o}C	140 ^{o}C	140 ^{o}C	140 ^{o}C
TIP	5420 kPa	5014 kPa	9253 kPa	6087 kPa	4206 kPa
p_r	1.335	1.11	1.6	1.432	1.157
T_{cond}	48.1 ^{o}C	44.4 ^{o}C	40.5 ^{o}C	43.8 ^{o}C	43.5 ^{o}C
T_{cso}	29.8 ^{o}C	29.9 ^{o}C	28.9 ^{o}C	29.5 ^{o}C	29.9 ^{o}C
SIC	2937 $/kWel	2641 $/kWel	3155 $/kWel	3075 $/kWel	3059 $/kWel
Avoided CO$_2$ emissions*	231.5 t/year	268.5 t/year	243.0 t/year	258.9 t/year	249.5 t/year
* Annual operating hours is assumed to be 7884 h/year at full load with specific CO$_2$ emissions for electricity generation of 331g/kWel (Clément, 2014)					

In the case of a traditional working fluid for Trans-ORC system, i.e. R134a, although the net power output and thermal efficiency are the lowest for the optimum SIC, the SIC of R134a-based system together with two other systems using propane and R1234ze are competitive. Furthermore, as shown in T-s diagrams (Figure 3) of these working fluids, they exit the turbine at highly superheated state which presents a potential to install an internal heat exchanger as recuperator for improving the cycle performance. While the higher GWP of R134a and higher flammability of propane could limit the use of these working fluids, the high cost and mildly flammability of R1234ze are the challenges for its replacement to R134a as Trans-ORC working fluid.

At the present, the commercialized ORC modules are often subcritical ones. According to the data presented in (Vanslambrouck et al., 2011), the average prices of these turbine based ORC modules, excluding the installation cost, range from 1337 $/kWel (converted from 1000 €/kWel with a 2011 yearly average exchange rate of 0.748 €/US$) for a 2 MW-unit, to 2674 $/kWel (2000 €/kWel) for a 500 kW-unit and up to 4011 $/kWel (3000 €/kWel) for a 150 kW-unit. For waste heat recovery applications, the installation costs can go from 50% to 100% of the ORC module's cost. Comparing to these data, the studied Trans-ORC systems present a real potential for waste heat recovery application at small scale. Considering the working fluid properties presented in Table 1 and the SIC values, the greater are the critical and normal boiling temperatures, the smaller is the SIC. In the case of R1234ze, the SIC is higher than that for R134a-based Trans-ORC due to the high cost of working fluid.

The amount of annual avoided CO$_2$ emission for the electricity generation by recovering waste heat using a Trans-ORC is presented in Table 4 for each working fluid. This amount is calculated in assuming 7884 operating hours per years at full load of power plant.
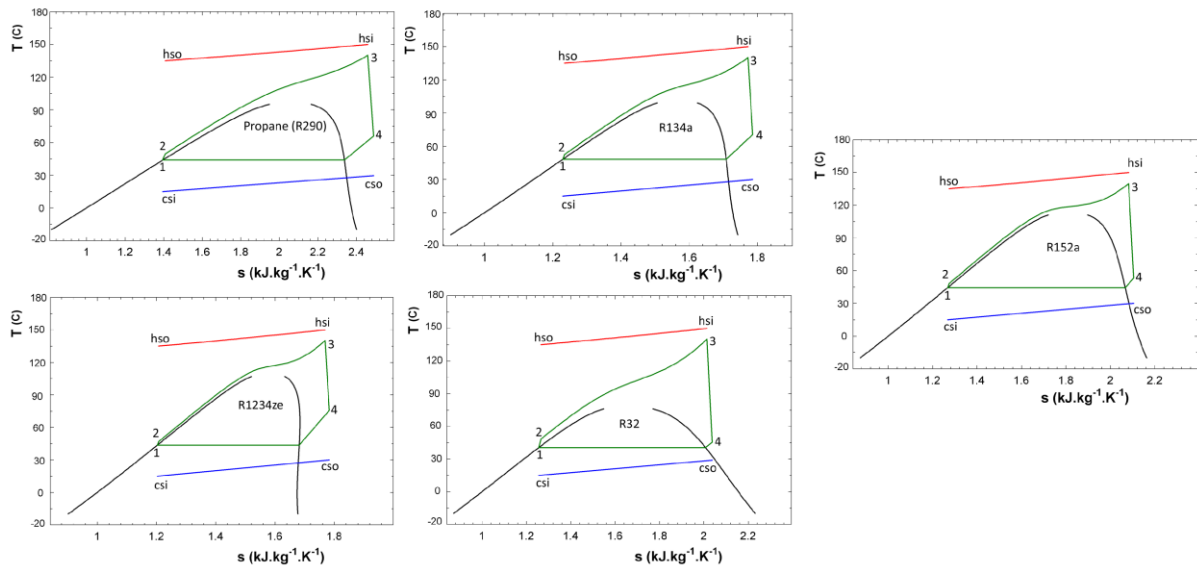
Figure 3: T-s diagram (green) of working fluids and temperature profile of heat source (red) and heat sink (blue) at minimum SIC
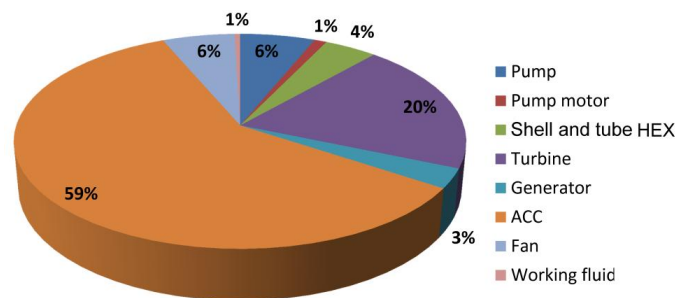

Figure 4: Repartition of purchase equipment cost of Trans-ORC (for the case of R152a)

Concerning the repartition of purchase equipment costs, the turbine and the air-cooled condenser are the most expensive components of Trans-ORC power plant (as shown in Figure 4) due to the low heat transfer coefficient and so the large required heat transfer surface area (e.g. ~11.4 m^2/kWel for the case of R152a). While the investment for the air-cooled condenser is more than half of total equipment purchase cost, this value for turbine is 20 %. In the present work, the air-cooled condenser is considered with horizontal tubes. Indeed, other technologies such as A-frame and V-frame air-cooled condenser could also be considered to increase heat transfer coefficient. The other costs included in the total capital investment are presented in Table 5.

Table 5: Capital cost components (for the case of R152a)

Component	Value
Total cost for equipments (excluding installation costs)	208361 $
Cost for working fluid, C_{wf}	948 $
Cost of site preparation, C_{sp}	10465 $
Cost of service facilities, C_{sf}	10465 $
Cost of contingencies and contractor's fee, C_{CCF}	41443 $
Total capital investment	271682 $

6. CONCLUSIONS – PERSPECTIVES

This work presents the result of Specific Investment Capital (SIC) minimization for a Trans-ORC system that converts heat from cooling circuit of a turbine exhaust gas. Some potential working fluids

have been considered for the optimization of the system. According to the optimization results, the most interesting fluid is R152a with the highest net power output (102.9 kWel) and lowest SIC (2641 $/kWel). However, the other fluids, i.e. R1234ze, R134a, or propane, present also the possibility to improve the performance and so the specific investment capital. The worst Trans-ORC in term of specific investment is found when R32 is used as working fluid. Indeed, this fluid operates at very high pressure (92.5 bars) to obtain a moderate net power output. For each tested working fluid, the optimum operating conditions are brought out by the mean of the SIC minimization. Regarding the repartition of purchase equipment costs, the most expensive component of Trans-ORC system is the air-cooled condenser followed by the turbine. Indeed, the condenser purchase cost contributes to more than half of the total equipment purchase cost, while the investment for the turbine excluding the electrical generator cost is of 20% of the total equipment cost.

NOMENCLATURE

D	Diameter	(m)
C	Cost	(US$)
$\bar{C}_p$	$\bar{C}_p = \left(h_w - h\right)/\left(T_w - T\right)$	(kJ.kg^{-1}.K^{-1})
C_p	Specific heat	(kJ.kg^{-1}.K^{-1})
g	Acceleration constant	(m/s^2)
G	Mass flux	(kg.s^{-1}.m^{-2})
h	Specific enthalpy	(kJ/kg)
$\dot{m}$	Mass flow rate	(kg/m^3)
k	Thermal conductivity	(kW.m^{-1}.K^{-1})
Nu	Nusselt number	(–)
P	Pressure	(kPa)
Pr	Prandtl number	(–)
p_r	Reduced pressure	(–)
$\dot{Q}$	Heat transfer rate	(kW)
Re	Reynolds number	(–)
SIC	Specific Investment Capital	(US$/kW)
T	Temperature	(^{o}C)
TIP	Turbine inlet pressure	(kPa)
TIT	Turbine inlet temperature	(^{o}C)
U	Overall heat transfer coefficient	(kW.m^{-2}.K^{-1})
$\dot{W}$	Power	(kW)

Greek letters

α	Heat transfer coefficient	W.m^{-2}.K^{-1}
η	Efficiency	(%)
μ	Dynamic viscosity	(kg.m^{-1}.s^{-1})
ρ	Density	kg/m^3

Subscript

1, 2, 3, 4	working fluid state points	*l/v*	liquid/vapeur
b	normal boiling point	*m*	motor
c	critical	*p*	pump
csi/cso	heat sink inlet/ outlet	*sat*	saturation
ex	external fluid	*t*	turbine
gen	generator	*TP*	two phases
h	hydraulic or hot	*wf*	ORC working fluid
hsi/hso	heat source inlet/outlet	*w*	wall

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