ENCIT-2018-0578
OPTIMIZATION OF THE WASTE HEAT RECOVERY SUPERSTRUCTURES FOR LARGE STATIONARY DIESEL ENGINES

Leonardo Rodrigues de Araújo
Alexandre Persuhn Morawski
Marcelo Aiolfi Barone
André Chun
Renan Cristofori Lima de Oliveira
Manuel Salazar Schiaffino
Carla César Martins Cunha
João Luiz Marcon Donatelli
José Joaquim Conceição Soares Santos
Federal University of Espírito Santo, Avenida Fernando Ferrari 514, Goiabeiras, Vitória, Espírito Santo, Brazil
engmec_leoaraujo@hotmail.com, morawski.em@gmail.com, mabacz@gmail.com, andr.chun@gmail.com, renancristofori@gmail.com, manuel.s.schiaffino@gmail.com, carla@ele.ufes.br, donatelliufes@gmail.com, jjcssantos@yahoo.com.br

Abstract. *In a world with finite natural fuels resources and growing energy demand, issues related with thermal systems design, such as cost estimative, design complexity, environmental awareness and optimization are becoming increasingly important. In a typical large internal combustion engine (ICE), less than 45% of the fuel energy is converted into useful power output, while the remaining energy is mainly lost through exhaust gases and cooling water. Currently there are two important challenges regarding the use of waste heat recovery (WHR) technologies associated with ICE: (i) What is the best technology to recovery waste heat from an economic point of view? (ii) What is the optimal configuration and the best operating parameters of the most suitable technology? In this work is carried out thermoeconomic optimization of three different WHR superstructures: Organic Rankine Cycle (ORC), Kalina Cycle (KALINA) and Conventional Rankine Cycle (CRC). During the optimization process of each superstructure, both structural and parametric variables are optimized simultaneously, and after the process are defined the optimal configuration and the best operating parameters of each WHR superstructure, allowing the evaluation and selection of the WHR technology more economically feasible for an ICE. The thermoeconomic optimization problem as well as the thermodynamic and thermoeconomic modelling are formulated and solved with the EES Software, using a stochastic method. After evaluation and comparisons of the optimization results of each WHR superstructure, the ORC technology has the lowest total cost and highest net power, resulting in highest gross profit due to additional electric energy sale.*

Keywords: *Thermal Systems Design, Waste Heat Recovery, Superstructure, Thermoeconomic Optimization, Organic Rankine Cycle, Kalina Cycle, Conventional Rankine Cycle, Internal Combustion Engine.*

1. INTRODUCTION

Due to growing worldwide concern about the environmental pollution level and the rational use fossil fuels, there has been a growing number of researches aimed at developing alternative technologies for better utilization of energy resources (Yang and Yeh, 2015a). In this context; researchers develop projects that should be economically feasible to increase the energy efficiency of equipment and processes so that the consumption of fossil fuels and environmental impact are reduced.

Several studies (Kaşka, 2014; Le *et al*., 2014) have estimated that around 20-50% of energy used in industrial processes is lost as waste heat. Loss of energy due to non-utilization of waste heat contributes to increased production costs and environmental impacts. Therefore, many efforts have been made in the last decades for this heat to be recovered (Vélez *et al*., 2012). In a typical large diesel engine, less than 45% of the fuel energy is converted into shaft power output, while the remaining energy is mainly lost through the exhaust gas, the jacket cooling water and other means, such as the air intercooling system and the lubrication system (Dolz *et al*., 2012). It is apparent that the energy recovery potential from the waste heat of an engine is appreciable.

The current scenario of Brazilian electrical system has led to the dispatch of power plants, originally designed as standby plants, which have been operating continuously. These power plants were designed as simple plants, with low investment cost and efficiency, without recovering effectively the waste heat. In power plants with internal combustion engines (ICE), a large amount of heat is rejected through cooling water as well as exhaust gases.

Several technologies are available for waste heat recovery (WHR) in ICE, such as Conventional Rankine Cycle (CRC), Organic Rankine Cycle (ORC) and Kalina Cycle (KALINA).

The well-known CRC, the most developed and widespread bottoming cycle, typically combined with a gas turbine. It is most suitable for plant sizes of several hundreds of megawatts of electrical power (Paanu *et al*., 2012). The CRC can also be applied with ICE, however, with less efficiency due to the lower exhaust gas temperature and mass flow exiting the turbocharger. Even so, according to Korobitsyn (1998), for medium temperature levels, CRC is still the most indicated, due to its fluid thermodynamic properties.

The ORC applies the principle of the CRC but uses organic working fluids instead of water. These organic compounds are characterized by a lower boiling temperature than water, which allows power generation from lower temperature heat sources. In addition, ORC technology provides an interesting alternative, which is it can be used, with few modifications, in conjunction with various temperatures heat sources (Quoilin *et al*., 2013). Among low-grade heat bottoming cycles, the ORC is so far, the most commercially developed one. For low-grade heat, it is simpler and economically more feasible than the CRC (Paanu *et al*., 2012).

On the other hand, the Rankine cycles (CRC and ORC) presents a series of disadvantages in bottoming cycle applications using low and medium temperature heat in combined cycles, as well as in generation of electricity from low-temperature heat sources, due to the vaporization temperature of pure fluids being constant. Regarding to these features, in order to overcome this disadvantage of the Rankine cycles, Alexander I. Kalina designed a new power cycle in which a mixture of ammonia and water is used as working fluid (Kalina, 1984). The Kalina cycle is basically a modified Rankine cycle, which was developed as an alternative to reduce the unavoidable irreversibilities associated with heat transfer when a pure substance is used as working fluid.

According to Paanu *et al.* (2012) the Kalina cycle can produce 10 to 30% more power than a conventional Rankine cycle for the same heat source temperature. One of the reasons is the increase in the average temperature of heat addition and the reduction in the average temperature of heat rejection of this cycle, compared to Rankine cycles. However, even though the use of ammonia-water mixture may increase heat recovery, the improvement may not always be justified by the complexity of the Kalina bottoming unit (Korobitsyn, 1998).

Several researchers have been studying WHR in ICE using alternative technologies separately, for example: CRC (Gewald *et al*., 2012; Hossain and Bari, 2013; Kyriakidis *et al*., 2017; Petrov, 2006), ORC (Di Battista et al., 2015; Song et al., 2015; Song and Gu, 2015; Sprouse and Depcik, 2013; Yang and Yeh, 2015b, 2015a) and KALINA (Guo *et al*., 2015; Larsen *et al*., 2014). Some other authors compare technologies as CRC and ORC (Macián *et al*., 2013; Shu *et al*., 2016; Wang *et al*., 2017; Zhu *et al*., 2013), ORC and KALINA (Bombarda *et al*., 2010; Yue *et al*., 2015), KALINA and CRC (Jonsson and Yan, 2001). In Morawski *et al.* (2017) the authors compare all these WHR technologies using the same case study, but the structure or system configuration of WHR technologies, as well as the operating parameters were previous defined. However, it was not found studies using thermoeconomic optimization of WHR superstructures, allowing the evaluation and selection of the most economically feasible WHR technology for an ICE, defining its optimal configuration and the best operating parameters.

Evaluate and select the most feasible WHR technology for an ICE from an economic point of view is a hard task. Currently the state of the art has two important challenges regarding the use of WHR technologies associated with ICE: (i) What is the best technology to recovery waste heat from an economic point of view? (ii) What is the optimal configuration and the best operating parameters of the most suitable technology?

One study (Frangopoulos *et al.*, 2002) comments and classifies the methods obtained in the literature for the optimal synthesis of thermal systems. In relation to thermal systems optimization methods based on superstructures, the studies investigate several optimization procedures that use several different techniques or a combination of them. However, these optimizations have been carried out, in most cases, through superstructures modelled in the same or external computational environment of the optimization routine.

A superstructure intended for thermal system optimization can be defined as a large thermal system, which considers all possible (or necessary) equipment, components and interconnections capable of supplying, individually or in combination, the energy and thermal demands of processes. Therefore, the basic purpose of the superstructure is to incorporate the configuration flexibilities to be exploited in obtaining the optimum thermal system, as well as to provide the mass and energy balances for all the points (feasible or not) covered in the optimization process without incurring simulation failures.

Modern thermal systems tend to be thermodynamically and structurally complex, with a large number of components, interconnections and fluxes, and modelled by nonlinear equations systems. Therefore, the use of a professional process simulator is desirable to achieve computational efficiency, since it will take care of all thermodynamic equations, properties, component models and mass and energy balances (Pires *et al*., 2013).

For thermal systems optimization problems, computational methods using gradients may not be suitable or present good performance, instead search methods and evolutionary methods may become more attractive (Pires *et al*., 2013). In recent years, evolutionary algorithms have been employed to look for optimal solutions for thermal systems. These algorithms use a stochastic search strategy to find and compare viable solutions until the best solution is determined or a finalization criterion is met. The derivatives of the model equations are not exploited in the search process and the process modifications only depend on the value of the objective function, thus facilitating the use of process simulators.

According to Pires *et al.* (2013), the amount of published work on thermal system optimization using evolutionary algorithms ratifies the interest of the scientific community in this subject. Thermal system optimization using superstructures, process simulator and evolutionary algorithm were carried out in Koch *et al*. (2007) and Wang *et al*. (2014).

Bearing this in mind, the major innovation in in this work is the structural and parametric thermoeconomic optimization of three different WHR superstructures (ORC, KALINA and CRC), defining the optimal configuration and the best operating parameters of each WHR technology, as well as the technology more economically feasible to recover two kinds of waste heat on a diesel engine: exhaust gases and cooling water.

The optimization problem as well as the thermodynamic and thermoeconomic modelling, are formulated and solved with the Engineering Equation Solver Software (EES), using a stochastic method (genetic algorithm). During the thermoeconomic optimization process, both design configuration and parametric variables are optimized simultaneously. The objective function for the thermoeconomic optimization problem is the gross profit due to additional electric energy sale, evaluating the total cost and additional net power produced in superstructures, given sale prices energy constant.

2. CASE STUDY

UTE Viana is a power plant located in the State of Espírito Santo, Brazil, and it is equipped with 20 engine-generator sets (EG's) made up of Wärtsilä W20V32 diesel engines of 9,000 kW. Each engine-generator is set to generate 8,730 kW of electrical power. The total installed capacity is 174.6 MW. The fuel oil used in the power plant is the heavy oil OCB-1. It has a LHV (Low Heat Value) of 40,785 kJ/kg and a mass composition of 87.47 % carbon, 10.77 % hydrogen, 0.76 % oxygen and 1.00 % sulphur (Morawski, 2016).

Exhaust gases are available from 15 EG's, because the exhaust gases of the remaining five are used by recovery boilers to produce steam used in the plant itself. The exhaust gases exit the turbochargers (turbocharged engines) with a mass flow of 16.7 kg/s, temperature of 345 °C and minimum temperature of 180 ºC. The composition of exhaust gases, presented in Tab. 1, was previously calculated in Morawski (2016) using a complete combustion model with excess of humid air.

Table 1. Molar composition of exhaust gas.

CO_2 (%)	**H_2O (%)**	**N_2 (%)**	**O_2 (%)**	**Air (%)**	**SO_2 (%)**
6.36	5.58	75.53	11.60	0.90	0.03

The engine cooling water, unlike the exhaust gases, is available from all the 20 EG’s. Immediately before entering the radiators, the water has a mass flow of 27.02 kg/s and a temperature of 78.4 °C. It is important to point out that the engine cooling water must return to the engines with a temperature of 41.4 °C. In this work, the WHR superstructures were evaluated only for one engine.

3. WASTE HEAT RECOVERY SUPERSTRUCTURES

Three different (Fig. 1) WHR technologies based on bottoming cycles are considered for heat recovery of both engine cooling water and exhaust gases: ORC, KALINA, CRC.

The ORC and KALINA superstructures shown in Figs. 1a and 1b, respectively, are composed by a high temperature cycle and a low temperature cycle. The high temperature cycle is fueled mainly by exhaust gases, while the engine cooling water can be used in a preheater. On the other hand, the low temperature cycle is fueled only by the engine cooling water.

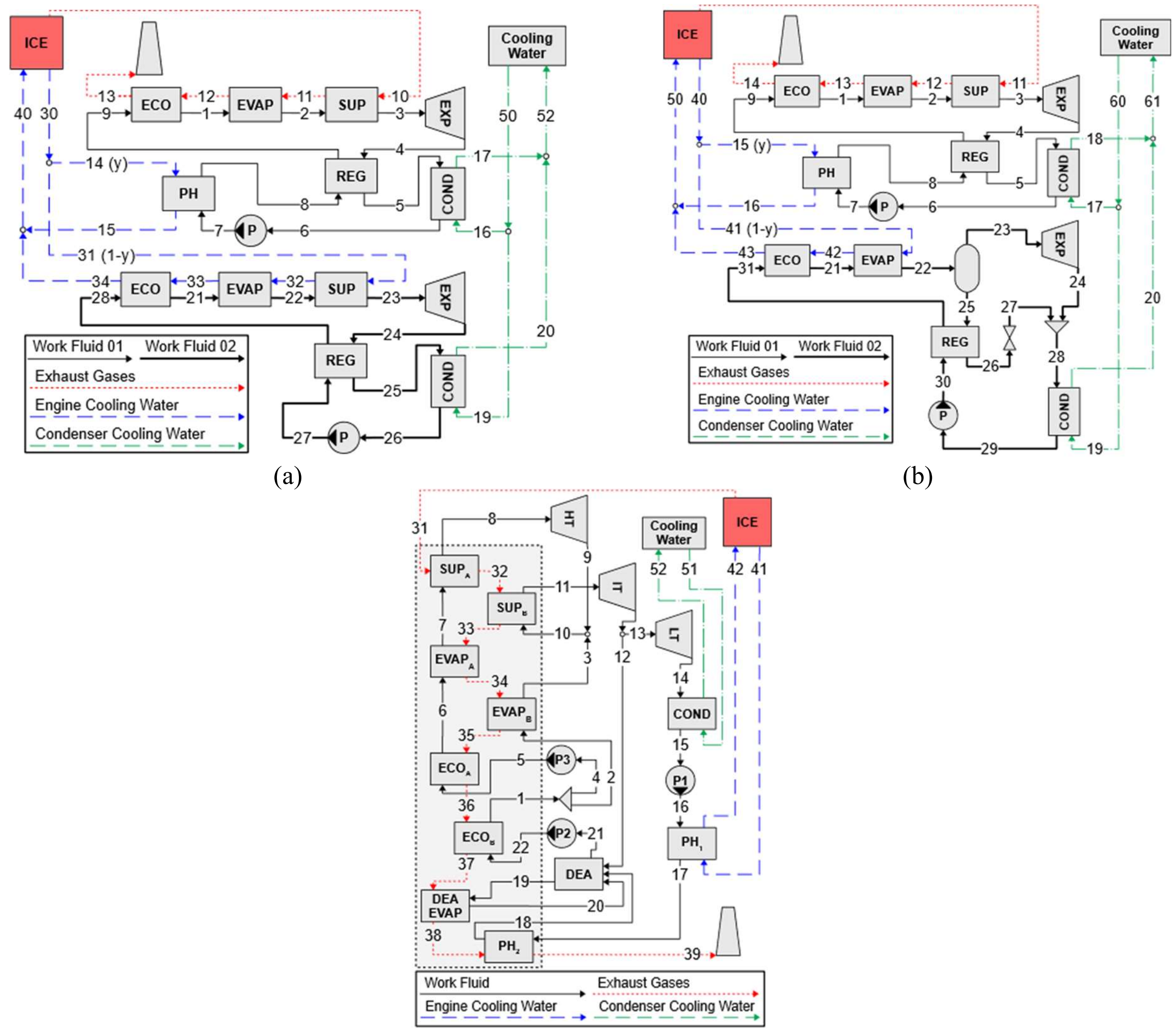
The ORC and KALINA superstructures allow flexibility of configuration by the existence or not of some equipment like economizer (ECO), superheater (SUP), regenerator (REG) on both high and low temperature cycles. In addition, the engine cooling water distribution allows the existence or not of preheater (PH) and of the low temperature cycle. In the ORC superstructure are evaluated four working fluids to each kind of heat source cycles: benzene, isobutane, isopentane and toluene in high temperature cycle, and R134a, R141b, R245fa and R236ea in low temperature cycle.

Unlike the other presented bottoming superstructures, the CRC do not allow engine cooling water as a primary heat source for a power cycle. It can be only used to preheat the working fluid (PH1) while the exhaust gases are the main heat source. The CRC superstructure can be seen in Fig. 1c.

The CRC superstructure also allows flexibility of configuration by the existence or not of some equipment like PH_1, PH_2, SUP_A, SUP_B and evaporator of the deaerator (DEA EVAP), but the main flexibility is on the recovery boiler configuration, which allows the existence or not of a second pressure level. In addition, the deaerator (DEA) can be fueled with steam from turbine extraction or from the DEA EVAP fueled by exhaust gases.

For the sake of simplicity, pressure drops and heat losses in the heat exchangers were neglected. All the condensers are design for that the working fluid to be saturated liquid. The condenser cooling water according to environment conditions of Vitória, Brazil, when cooling tower is used, are adopted as inlet and outlet of 30 °C and 35 °C, respectively.

For all turbines included at WHR superstructures, it was assumed that isentropic efficiency is equal to 90 % and, for pumps, 80 %.



(a) (b)

(c)

Figure 1. Waste Heat Recovery Superstructures: (a) ORC, (b) KALINA, (c) CRC.

4. THERMOECONOMIC OPTIMIZATION PROBLEM

In order to evaluate the WHR superstructures and point out the most feasible technology, this work uses the gross additional profit (*Pr*) resulting of the additional electric net power sale, as shown in Eq. (1).

$$\Pr = \dot{W}_{net} \cdot S_p - \dot{Z}_{Tot} \left[US\$/h \right], \tag{1}$$

where $\dot{W}_{net}$ is the electric net power generated by the superstructure, S_P is the sale price (135.00 *US\$/MWh*) and $\dot{Z}_{Tot}$ is the total cost rate associated with the total capital investment and the maintenance costs for each superstructure. The total cost rate is obtained using Eq. (2), where C_{Tot} are the total equipments costs, φ is the maintenance factor (assumed 1.06), N_{op} is the number of hours of plant operation per year (assumed 8000 hours), and *CRF* is the annual capital recovery factor, calculated by Eq. (3), where $j = 0.175$ is the interest rate and $n = 20$ is the number of years.

$$\dot{Z}_{Tot} = \frac{C_{Tot} \cdot CRF \cdot \varphi}{N \cdot 3600} \tag{2}$$

$$CRF = \frac{j \cdot (1+j)^n}{(1+j)^{n-1}} \tag{3}$$

The effective conclusion of a thermal design project requires estimation of the major costs involved in the project. Therefore, good cost estimation is a key factor in successfully completing a design project (Bejan *et al*., 1996).

There are many types of capital cost estimations and various methods often provide different results. The economic evaluation in this work is performed according to the module costing technique (MCT), extensively used for preliminary cost estimates of chemical plants (Turton *et al*., 2012). This technique relates all direct and indirect costs to the purchased equipment cost evaluated for base conditions (C_{PE}), at ambient pressure and carbon steel construction, as expressed in Tab. 2.

Table 2. Purchased equipment cost (C_{PE}) Functions (Turton *et al*., 2012; Uche, 2001).

Equipment	**Cost Equation**	
Expander	$C_{PE} = A \cdot e^{B \cdot \ln C \cdot FB1 \cdot (D \cdot F2T + E \cdot F2P)} \cdot FBN \cdot FBT$	$FB1 = \eta_T \cdot \dot{m}_k$
	$F2T = T_{in} - T_{out} - T_{out} \cdot \ln \frac{T_{in}}{T_{out}}$	$FBT = 1 + 5 \cdot e^{\left(\frac{T_{in} - 1100}{18.75}\right)}$
	$F2P = T_{out} \cdot \ln \frac{P_{in}}{P_{out}}$	$FBN = 1 + \left(\frac{1 - 0.9}{1 - \eta_T}\right)^3$
Heat Exchangers	$C_{PE} = 10^{\left\{A - B * \log_{10}(A_{HE}) + C \cdot \left[\log_{10}(A_{HE})\right]^2\right\}}$	
Pump	$C_{PE} = A \cdot e^{\left\{B \cdot \ln\left[C \cdot \dot{m} \cdot D \cdot (P_{in} - P_{out})\right]\right\}} \cdot FDN$	$FDN = 1 + \left(\frac{1 - 0.8}{1 - \eta_P}\right)^3$

The constants costs (*A, B, C, D, E*), in Tab. 2, depend on working fluid, solution concentration, equipment type and their capacity or size parameter.

The logarithmic temperature difference method is used in this present study, given by Eq. (4), where *DTML* is the logarithmic mean temperature difference, ΔT_1 and ΔT_2 are the maximum and minimum temperature differences at the heat exchangers ends, respectively.

$$DTML = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2 \right)} \tag{4}$$

The heat transfer $\left(\dot{Q}_i\right)$ in each section can be calculated by Eq. (5), where, U_i and A_i are the overall heat transfer coefficient and the heat transfer area of each section, in that order. The values of U_i are obtained by Siddiqi and Atakan (2012) and Larsen *et al*. (2014).

$$\dot{Q}_i = U_i \cdot A_i \cdot DTML \tag{5}$$

Deviations from these base conditions are handled by multiplying pressure (FP) and material (FM) factors. The pressure factor is given by Eq. (6), where P is the pressure and c_i are constants depending on equipment type.

$$\log_{10} F_P = c_1 + c_2 \log_{10}(P) + c_3 \left[\log_{10}(P)\right]^2 \tag{6}$$

The additional direct and indirect costs are considered though the bare module factor (FBM) in the module costing technique. The bare module cost is the sum of all direct and indirect cost and can be calculated by Eq. (7).

$$C_{BM} = C_{PE} \cdot F_{BM} \quad [US\$] \tag{7}$$

The values of the bare module cost factors are given for different types of equipment. For heat exchangers and pumps the expression of the bare module cost factor is given by Eq. (8), where B_i are constants depending of the heat exchanger or pump type.

$$F_{BM} = B_1 + B_2 \cdot F_P \cdot F_M \tag{8}$$

For other components the F_{BM} is directly given as a multiplier that accounts for equipment type, operating pressure and construction material. Table 3 presents the values of the coefficients adopted in this work in order to calculate the total equipments costs. Other auxiliary equipments costs, not approached by Turton *et al*. (2012), are adopted as 10% of the alternative total cost.

Table 3. Constants for cost evaluation according to the procedure suggested by Turton *et al*. (2012).

Components	B_1	B_2	F_M	c_1	c_2	c_3	F_{BM}
Heat Exchangers	1.63	1.66	1	0.03881	-0.112272	0.08183	-
Turbines	-	-	-	-	-	-	3.4
Pumps	1.89	1.35	1.6	-0.3935	0.3957	-0.00226	-

For modifications and expansions of existing thermal systems, there are also other costs that need to be accounted for, like fees and contingency costs. According Turton *et al*. (2012), when there are no other recommendations, these costs are 3% and 15% of the bare cost module, respectively. Adding these remaining costs, the total module cost is calculated by Eq. (9), where *k* represents the number of the project equipments.

$$C_{TM} = 1.18 \cdot \sum_{i=1}^{k} C_{BM,i} \quad [US\$] \tag{9}$$

All the data available in Turton *et al*. (2012) are referenced in 2001. According to Bejan *et al.* (1996), the calculated cost updated is made through an appropriated cost index. The cost index is an inflation indicator used to correct the cost of equipment items, material, labor, and supplies to the date of the estimation. For thermal design projects the Chemical Engineering Cost Index (CEPCI) is recommended for total plants, or groups of components. The newest CEPCI available at the present date is the CEPCI of March of 2017 ($CEPCI_{2017} = 562.1$). Thus, the correct total equipments cost is given by Eq. (10), where the $CEPCI_{2001}$ is 397.

$$C_{Tot} = \frac{CEPCI_{2017}}{CEPCI_{2001}} \cdot (C_{TM}) \quad [US\$] \tag{10}$$

The methodology used to determine the best technology to recovery waste heat from economic point of view and its optimal configuration and the best operating parameters is solve the thermoeconomic optimization problem of each superstructure maximizing de gross profit and after to compare the results. The Tables 4 and 5 show the decision variables, initial guesses, lower and upper bounds regarded each WHR superstructure.

Table 4. Optimization decision variables for CRC Superstructure

Decision Variables	Initial Guess	Lower Bound	Upper Bound
Evaporation temperature (1st level) (ºC)	140	133.6	158.9
Evaporation temperature (2nd level) (ºC)	160	158.9	175.4
Evaporator Pinch Point (1st level) (ºC)	120	0	250
Mass Flow Fraction (2nd level) (-)	0.40	0	1
Superheat temperature increase (1st level) (ºC)	80	0	200
Superheat temperature increase (2nd level) (ºC)	100	0	200
Deaerator pressure (bar)	1	0	10
Condensation temperature (ºC)	70	30	100
Preheater 1 effectiveness (-)	0.50	0	1
Preheater 2 temperature increase (ºC)	10	0	50
Turbine extraction (-)	0.005	0	0.1

The optimization method chosen in this work is Genetic Algorithm, with 64 individuals, 256 generations and a mutation rate of 0.058, presented in EES Optimization ToolBox. The convergence of the optimization algorithm is the difference the objective function value between the current optimization and the previous optimization. The convergence criteria value is assumed equal 0.01 US$/h.

Table 5. Optimization decision variables for ORC and KALINA Superstructures

Decision Variables	ORC Superstructure Initial Guess	ORC Superstructure Lower Bound	ORC Superstructure Upper Bound	KALINA Superstructure Initial Guess	KALINA Superstructure Lower Bound	KALINA Superstructure Upper Bound
High Temperature Cycle						
Evaporation temperature (°C)	100	45	345	150	45	345
Condensation temperature (°C)	35	30	45	40	30	45
Superheat temperature increase (°C)	40	0	200	80	0	200
Evaporator Pinch Point (°C)	150	0	200	60	0	200
Regenerator effectiveness (-)	0.50	0	1	0.50	0	1
Preheater effectiveness (-)	0.50	0	1	0.50	0	1
Engine water distribution (-)	0.40	0	1	0.05	0	1
Working fluid (-)	Isobutane	-	-	-	-	-
Ammonia concentration (-)	-	-	-	0.70	0.40	0.99
Decision Variables	**ORC Superstructure Initial Guess**	**ORC Superstructure Lower Bound**	**ORC Superstructure Upper Bound**	**KALINA Superstructure Initial Guess**	**KALINA Superstructure Lower Bound**	**KALINA Superstructure Upper Bound**
Low Temperature Cycle						
Evaporation temperature (°C)	55	45	78.4	50	45	78.4
Condensation temperature (°C)	35	30	45	33	30	45
Evaporator Pinch Point (°C)	15	0	30	20	0	40
Regenerator effectiveness (-)	0.30	0	1	0.50	0	1
Superheat temperature increase (°C)	20	0	40	-	-	-
Working fluid (-)	R245fa	-	-	-	-	-
Ammonia concentration (-)	-	-	-	0.80	0.40	0.99
Separator temperature (ºC)	-	-	-	60	10	78.4

5. RESULTS DISCUSSIONS AND COMPARISONS

The results shown in Tabs. 6 and 7 allow the evaluation and selection of the WHR technology more economically feasible for waste heat recovery of the exhaust gases and engine cooling water, defining its optimal configuration and operational parameters.

Table 6. Optimal Values CRC Superstructure

Parameters	Value
Exhaust gases mass flow (kg/s)	16.7
Engine cooling water mass flow (kg/s)	27.02
Decision Variables	
Evaporation temperature (1st level) (ºC)	158.9
Evaporation temperature (2nd level) (ºC)	158.9
Evaporator Pinch Point (1st level) (ºC)	47.51
Mass Flow Fraction (2nd level) (-)	0
Superheat temperature increase (1st level) (ºC)	170
Superheat temperature increase (2nd level) (ºC)	0.46
Deaerator pressure (bar)	0.2
Condensation temperature (ºC)	51
Preheater 1 effectiveness (-)	0
Preheater 2 temperature increase (°C)	0.34
Turbine extraction (-)	0.015
Dependent Variables	
Total cost (US$)	2,291,000
Total net power (kW)	663.9
Energy efficiency (%)	22.49
Heat source (kW)	2952
Heat sink (kW)	2288
Exhaust gases exit temperature (°C)	182
Engine cooling water exit temperature (°C)	78.4
Net power index (NPI)	3450
Objective Function	
Gross Profit (US$/h)	34.31

Table 7. Optimal Values ORC and KALINA Superstructure - First Level Optimization

	ORC Superstructure	**KALINA Superstructure**
	Parameters	
Exhaust gases mass flow (kg/s)	16.7	16.7
Engine cooling water mass flow (kg/s)	27.02	27.02
	High Temperature Cycle	
Decision Variables		
Evaporation temperature (°C)	235.9	128.4
Condensation temperature (°C)	34.6	31.7
Superheat temperature increase (°C)	0.8	117.5
Evaporator Pinch Point (°C)	40.9	80.3
Regenerator effectiveness (%)	0	73.1
Preheater effectiveness (%)	93.74	20.32
Engine water distribution (-)	1	1
Working fluid (-)	Toluene	-
Ammonia concentration (%)	-	95
Dependent Variables		
Total cost (US$)	1,533,000	2,730,000
Total net power (kW)	878	716
Energy efficiency (%)	26.22	23.32
Heat source (kW)	3349	3070
Heat sink (kW)	2471	2354
Exhaust gases output temperature (°C)	180	180
Engine cooling water output temperature (°C)	68.2	78.4
Net power index (NPI)	1746	3813
Objective function		
Gross Profit (US$/h)	81.5	30.68
	Low Temperature Cycle	
Evaporation temperature (°C)	53.5	50.0
Condensation temperature (°C)	40.8	43.9
Evaporator Pinch Point (°C)	15.4	28.3
Regenerator effectiveness (%)	0	83
Superheat temperature increase (°C)	13.7	-
Working fluid (-)	R141b	-
Ammonia concentration (%)	-	97.5
Separator temperature (ºC)	-	65.9
Dependent Variables		
Total cost (US$)	0	0
Total net power (kW)	0	0
Energy efficiency (-)	0	0
Heat source (kW)	0	0
Heat sink (kW)	0	0
Engine cooling water output temperature (°C)	-	-
Net power index (NPI)	0	0
Objective function		
Gross Profit (US$/h)	0	0

5.1 Design and Parametric Optimization

The ORC and KALINA superstructures do not contemplate the use of low temperature cycles, since all engine cooling water mass flow is directed to the preheater of the high temperature cycles. Therefore, these low temperature cycles do not produce net power and become economically unfeasible. Due to low values of overheating temperature rise and regenerator effectiveness, the optimal configuration proposed by the optimization algorithm in ORC high temperature cycle is composed by economizer, evaporator, expander, condenser, pump and preheater, besides the toluene is presented as the working fluid. The optimal design configuration proposed to KALINA high temperature cycle contemplates the existence of all equipments. The CRC superstructure is composed by recovery boiler with only one pressure level with superheating, since mass flow rate at the second level pressure is equal to zero. The deaerator is fueled by turbine extraction and low values in temperature rise (preheater 2) and effectiveness (preheater 1) show that these equipments do not exist.

5.2 Thermodynamic Analysis

All superstructures recover the exhaust gases up to practically the limit temperature of 180 °C, due to the sulfur oxide precipitation. For the engine cooling water, only the ORC superstructure achieves a significant gain of heat, with a decrease of temperature equal to 10.2 °C, reducing power demand in radiator fans, whereas for the other individual superstructures no significant variation temperature occurs. Therefore, it is necessary to use the radiator for the additional cooling of this heat source in each superstructure. The highest addition and rejection of heat, power production and energy efficiency occur in the ORC superstructure, followed by KALINA and CRC.

5.3 Economic Analysis

The use of ORC technology is the most economically feasible since it produces the highest gross profit due to its greatest net power production with lowest total cost. Furthermore, it can also be verified that the ORC technology has the lowest relation between total cost and net power produced (NPI).

6. CONCLUSIONS

This paper focuses on WHR of a diesel engine Wärtsilä 20V32. WHR technologies are used to recover waste heat from both the engine cooling water and exhaust gas. The maximum gross profit is chosen as the evaluation criterion to select the most appropriate WHR technology from the economic point of view, as well as to define its ideal configuration and optimal parameters. Technological and economic factors are considered in optimization.

The main conclusions are summarized as follows:

i. The optimization procedure is robust enough to exploit different WHR technologies to recover waste heat at ICE, in addition allow the selection of the technology more economically feasible. This procedure is also able to define the best design configuration and the optimal parameters for each WHR technology.

ii. After optimization ORC recovery system is presented as best alternative to recover waste heat for an ICE and toluene is selected as working fluid. The low-temperature engine cooling water is totally used to preheat the working fluid of the high temperature cycle, while the high-temperature exhaust gas is utilized for evaporation. The total net output power reached 878 kW, which results in a power increase of 9.75 % for the diesel engine.

iii. Because the ORC system has the lowest capital cost and the highest net power, it is selected by the optimization algorithm as the most economically feasible WHR technology to recover waste heat at ICE. Therefore, the optimized system is recommended for practical application in WHR of diesel engines. The analytical method and the optimization procedure presented in this work can be applied in similar cases.

7. ACKNOWLEDGEMENTS

The authors would like to thank the Program for Research and Technological Development of the Electric Energy Sector regulated by the National Electric Energy Agency (ANEEL), the Thermal Power Plant of Viana S.A. (TEVISA), the Support Foundation of Espírito Santo Research (FAPES) and the National Agency of Petroleum, Natural Gas and Biofuels (ANP), for the financial support.

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