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# Thermodynamic evaluation of the Kalina split-cycle concepts for waste heat recovery applications

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## Abstract

The Kalina split-cycle is a thermodynamic process for converting thermal energy into electrical power. It uses an ammonia-water mixture as working fluid (like a conventional Kalina cycle) and has a varying ammonia concentration during the preheating and evaporation steps. This second feature results in an improved match between the heat source and working fluid temperature profiles, decreasing the entropy generation in the heat recovery system. The present work compares the thermodynamic performance of this power cycle with the conventional Kalina process, and investigates the impact of varying boundary conditions by conducting an exergy analysis. The design parameters of each configuration were determined by performing a multi-variable optimisation. The results indicate that the Kalina split-cycle with reheat presents an exergetic efficiency by 2.8 % points higher than a reference Kalina cycle with reheat, and by 4.3 % points without reheat. The cycle efficiency varies by 14 % points for a variation of the exhaust gas temperature of 100 °C, and by 1 % point for a cold water temperature variation of 30 °C. This analysis also pinpoints the large irreversibilities in the low-pressure turbine and condenser, and indicates a reduction of the exergy destruction by about 23 % in the heat recovery system compared to the baseline cycle.

*Keywords:* Kalina split-cycle, Exergy analysis, Waste Heat Recovery

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## 1. Introduction

The integration of waste heat recovery (WHR) systems in various processes presents thermodynamic and environmental benefits, as it results in a greater power generation for the same fuel input and smaller specific CO<sub>2</sub> emissions. Several power cycles have been suggested in the scientific literature: they differ by the selection of the working fluid, the size of application, the temperature and pressure levels, etc. The most well-known cycles are the steam Rankine cycle, the Organic Rankine cycle (ORC) and the Kalina cycle, in which the working fluid is a mixture of ammonia and water. The two latter cycles are often suggested as alternatives to the steam Rankine cycle for waste heat recovery, as they may display a higher thermodynamic efficiency in low- and medium-temperature applications.

Both power cycles may be viable at the scale of application studied in the present work, i.e. for a net power output of 1-5 MW [1,2]. Victor et al. [3] compared the Kalina cycle and ORC in the temperature range 100-250 °C. It was suggested that, while the two cycles could produce similar power outputs, the ORC was preferable below 200 °C and the Kalina above 200 °C. Wang et al. [4] investigated WHR technologies for use in the cement industry with heat source temperatures of 340 °C. They compared the Kalina cycle and ORC with two steam cycle setups and found that the Kalina cycle had the highest efficiency, followed by the two steam cycles and the ORC. However, Bombarda et al. [5] also compared the Kalina cycle and ORC, for a heat source temperature of 346 °C, and showed that both cycles, when optimised, produced

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## Nomenclature

$T$	temperature, K	$Q$	heat
$\bar{e}$	molar exergy, J/mol	$W$	work
$\dot{E}$	exergy rate, W	ch	chemical
$\dot{Q}$	heat rate, W	ph	physical
$\dot{S}$	entropy rate, W/K	<i>Subscripts</i>	
$\dot{W}$	power, W	$d$	destruction
$\dot{m}$	mass flowrate, kg/s	$f$	fuel
$e$	specific exergy, J/kg	$j$	stream
$h$	specific total enthalpy, J/kg	$k$	component
$p$	pressure, Pa	$l$	loss
$s$	specific entropy, J/(kg·K)	$p$	product
$y$	component/sub-system exergy ratio	0	dead state
<i>Abbreviations</i>		bub	bubble point
EOS	Equation of State	cv	control volume
ORC	Organic Rankine Cycle	cw	cooling water
WHR	Waste Heat Recovery	dew	dew point
<i>Greek letters</i>		gen	generation
$\varepsilon$	exergy efficiency	in	inlet
<i>Superscripts</i>		out	outlet
*	relative	r	rich ammonia concentration

almost equal net power outputs. The present study does not directly compare the ORC with the Kalina cycle but is based on the boundary conditions used in the work of Bombarda et al. [5], to allow further evaluations of the power cycle performance.

Energy can neither be created nor destroyed, and an energy analysis illustrates the energy transformations and flows throughout the system under study. On the opposite, exergy is not conserved in any real process, illustrating therefore the locations, causes and magnitudes of the thermodynamic irreversibilities taking place. Exergy destruction also accounts for the additional exergetic fuel required because of the system imperfections. Several studies on the thermodynamic performance of the Kalina cycle exist. Marston [6] carried out a parametric study of the Kalina cycle. The turbine inlet composition and separator temperature were identified as the key parameters to optimise. These findings were supported by Nag and Gupta [7], who performed an exergetic analysis of the Kalina cycle, and identified the turbine inlet temperature and composition, as well as the separator temperature, as having the largest influence on the thermodynamic performance of this cycle. Dejfors and Svedberg [8] conducted an exergy analysis to compare the Kalina cycle with a steam Rankine cycle for a direct fired biomass-fueled cogeneration plant. They noted that the aspect of being direct fired lead to significantly higher exergy losses in the boiler for the Kalina cycle compared to the Rankine cycle. Jonsson [9] investigated the Kalina cycle as WHR system for gas engines and gas diesel engines. It was argued that the Kalina cycle presents the potential to generate more power than the steam Rankine cycle, and that the additional costs could be justified by the gains in efficiency. Singh and Kaushik [10] investigated a Kalina cycle coupled to a coal fired steam power plant. They identified the primary source of exergy destruction, and therefore the greatest potential for optimisation, as the boiler.

38 The present paper presents and evaluates a unique power generation cycle, called the Kalina split-cycle.  
39 This process is also based on the ammonia-water mixture as working fluid, like the conventional Kalina  
40 cycle, but is characterised by a varying ammonia concentration in the heat recovery system. This can result  
41 in a smaller entropy generation in the heat transfer process, and potentially in a higher exergetic efficiency  
42 of the complete power cycle. This concept was briefly mentioned in the work of Kalina [11].

43 In the system analysis presented in Larsen et al. [12], it was suggested that the components that affect  
44 the process efficiency and optimisation the most are the separator, the recuperators, the boiler and the  
45 turbine. Moreover, it was indicated that the most important variables that impact the thermal efficiency  
46 are the ammonia concentration and the cooling water temperature. A simplified cost analysis of the Kalina  
47 split-cycle was also conducted, and the payback time of this particular process layout is sensibly similar to  
48 the payback time of a conventional Kalina cycle. The major costs were related to the boiler and turbines.  
49 The boiler costs are estimated to be about 40 % higher if the Kalina split-cycle with reheat is compared to  
50 the conventional Kalina cycle, and about 45 % if compared to the Kalina cycle with reheat. The turbine costs  
51 are estimated to be about 30 % higher if the Kalina split-cycle with reheat is compared to the conventional  
52 Kalina cycle, and about 6 % if compared to the Kalina cycle with reheat.

53 The literature appears to contain little on the thermodynamic performance of such cycles, and this study  
54 aims at closing this gap, following these three objectives:

- 55 • estimation of the cycle potential, in terms of exergy efficiencies, economic costs and environmental  
56 impacts, compared to a conventional Kalina cycle, with and without reheat;
- 57 • analysis of the plant inefficiencies and of the exergy destruction trends;
- 58 • evaluation of the effect of the boundary conditions (heat source and cold reservoir temperatures) on  
59 the system performance.

60 Section 2 presents the design of the Kalina split-cycle system and the methods used in this work are  
61 reported in Section 3. The results are presented in Section 4 and concluding remarks are outlined in Section  
62 5.

## 63 2. System description

### 64 2.1. Reference Kalina cycle

65 The Kalina cycle is similar in principle to the Rankine cycle, in which heat is supplied to a closed  
66 process loop, and where thermal energy is converted into mechanical work. The main difference lies in the  
67 properties of the working fluid, which is an ammonia-water mixture in the Kalina cycle. This two-component  
68 mixture is zeotropic, which means that the vapour and liquid phases do not have the same composition  
69 when condensation and evaporation take place. At constant pressure, the evaporation temperature changes  
70 during the heat transfer process, unlike pure substances, which have a constant evaporation temperature.  
71 The temperature glide results in a better match between the temperature profiles of the heat source and  
72 receiver. The exergy destruction caused by the heat transfer process is therefore smaller, but the area  
73 requirements of the heat exchangers increase. The range of the temperature glide can be adjusted by  
74 modifying the ammonia and water fractions of the working solution, as well as the operating pressure. The  
75 Kalina cycle may therefore be suitable for both low- and medium-temperature heat recovery applications.  
76 Several configurations of this thermodynamic cycle exist. The layout considered in this work is inspired by  
77 the one presented in the study of Bombarda et al. [5] and was studied by Marston [6,13] and El-Sayed [14].  
78 The terms ‘rich’ and ‘lean’ imply ammonia-rich and ammonia-lean in the rest of this work. Four different  
79 ammonia concentrations can be found in the conventional Kalina cycle.

80 *2.2. Kalina split-cycle*

81 The Kalina split-cycle (Figure 1) is fundamentally similar to the Kalina cycle, both using the ammonia-  
 82 water mixture. The main difference lies in the change of ammonia concentration in the evaporation process,  
 83 which involves a more complex splitting and mixing arrangement to achieve the desired ammonia concentra-  
 84 tions. Five different concentrations can be found in the Kalina split-cycle and are denoted ‘very rich’  
 85 (points 18, 19 and 26), ‘rich’ (points 20 to 25, and 33), ‘basic’ (points 1 to 5), ‘lean’ (points 27 to 32), and  
 86 ‘very lean’ (points 11 to 16), in reference to their ammonia content.

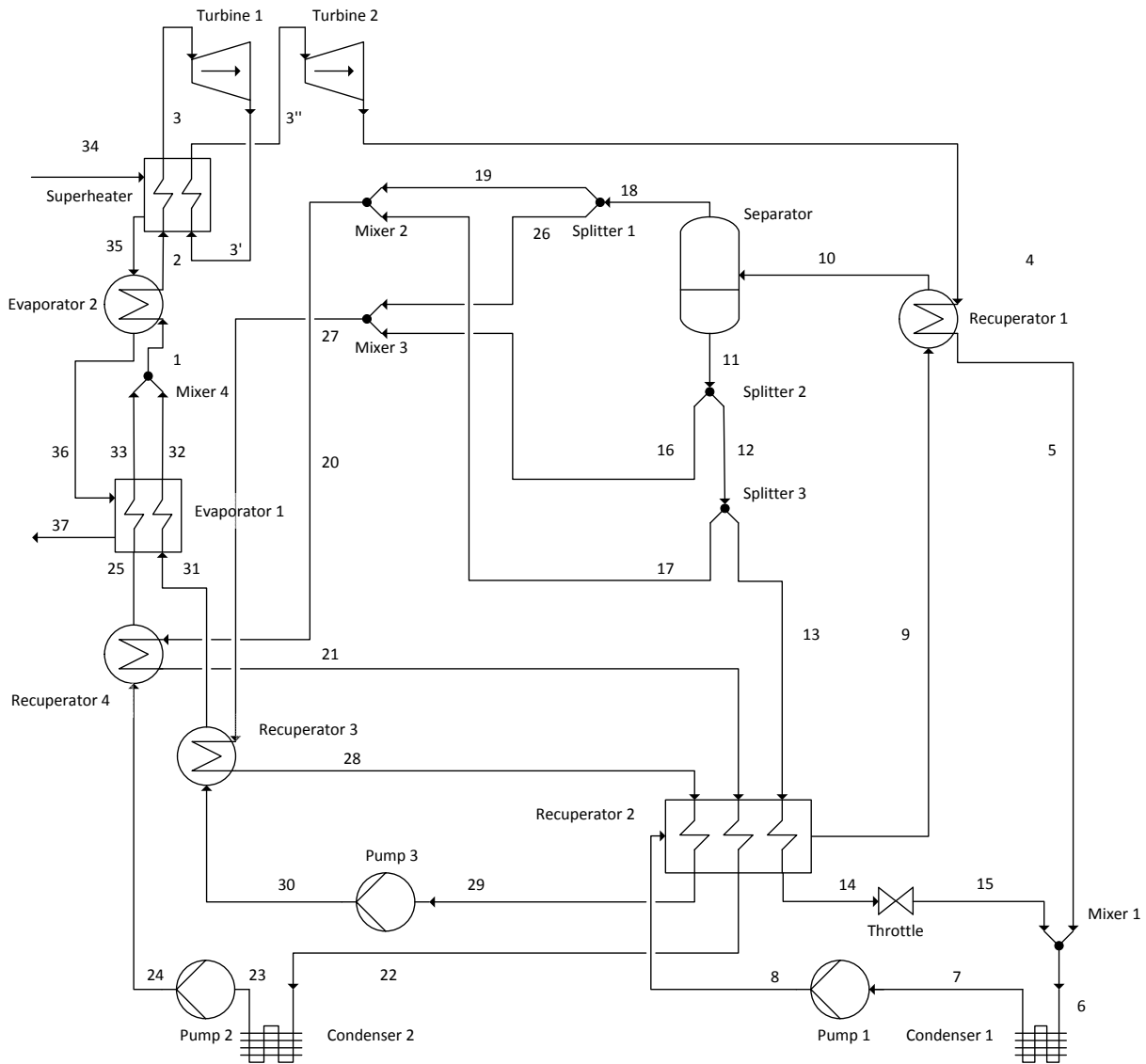


Figure 1: Flowchart of the Kalina split-cycle process

87 At the difference of the Kalina cycle where there is only one single stream, two streams with different  
 88 ammonia concentration enter the boiler, an ammonia-rich (25) and an ammonia-lean (31). In the boiler,  
 89 the former is fully evaporated (33) while the latter stays in liquid form, but is heated to its bubble point  
 90 temperature (32). They are then mixed into (1): the resulting solution is an ammonia-water solution in  
 91 vapour-liquid equilibrium, which is evaporated (2) and superheated (3). The benefits of splitting the streams  
 92 can be visualised in temperature-enthalpy diagrams shown in Larsen et al. [12].

93 In all cases, the minimum temperature difference, i.e. the pinch point location, is found in the evaporation  
 94 process, i.e after the preheating and before the mixing of the rich and lean streams. This work assumes that  
 95 the rich and lean streams are mixed at the same temperature and pressure to reduce the entropy generation  
 96 due to mixing effects. In addition, Kalina [11] suggested that the rich stream should be at its dew point,  
 97 while the lean stream should be at its bubble point. The vapour and liquid phases of each stream are  
 98 therefore at their equilibrium concentrations, which minimises the entropy generation in the mixing process.  
 99 However, such constraint fixes the mixing temperature, as the ammonia concentration of the lean and rich  
 100 streams are interdependent. The actual mixing point has little influence on the overall performance of the  
 101 system, since the pinch point is indeed located either at the bubble temperature of the ammonia-rich stream  
 102 or at the outlet of the superheater  $T_3$ .

### 103 3. Methods

#### 104 3.1. Modelling and simulation

105 This study was built on a baseline case adapted from the work of Bombarda et al. [5], where the  
 106 integration of a bottoming Kalina cycle was studied, using exhaust gases from diesel engines as heat source.  
 107 The Kalina cycle was compared to the Kalina split-cycle, considering one reheat stage as a possible option,  
 108 and the models were validated in [12]. The full power cycle was simulated using both Matlab [15] and  
 109 Aspen Plus<sup>®</sup> [16], based on the equations of state derived from Tillner-Roth and Friend (REFPROP) [17]  
 110 to estimate the physical and thermodynamic properties.

111 The modelling of the Kalina split-cycle considered the same assumptions as presented in Bombarda et  
 112 al. [5], at first for comparing the performance of this alternative configuration to the more conventional  
 113 one (Table 1). The waste heat source was exhaust gases from marine diesel engines with a total flowrate  
 114 of 35 kg/s, inlet and outlet temperatures of the waste heat recovery boiler of 346 °C and 127.7 °C, and a  
 115 composition of 74.6 % N<sub>2</sub>, 11.7 % O<sub>2</sub>, 6.7 % H<sub>2</sub>O, 5.9 % CO<sub>2</sub> and 1.1 % Ar on a molar basis.

Table 1: System parameters (baseline case)

Parameter	Description	Value
$T_{34}$	Heat source inlet temperature	346 °C
$T_{37}$	Heat source outlet temperature	127.7 °C
$T_{cw}$	Cooling water inlet temperature	25 °C
$T_{34} - T_3$	Superheater approach	16 °C
$\Delta T_{\text{evap}}$	Minimum temperature difference (boiler)	21.9 °C
$\Delta T_{\text{rec}}$	Minimum temperature difference (recuperators)	5 °C
$\Delta T_{\text{cond}}$	Minimum temperature difference (condenser)	4.5 °C
$\eta_{t,\text{pol}}$	Turbine polytropic efficiency	70.5 %
$\eta_{t,\text{mec}}$	Turbine mechanical efficiency	96 %
$\eta_{\text{pp}}$	Pump efficiency	70 %
$\eta_{\text{dr}}$	Driver efficiency	95 %

116 A multi-level approach was applied to model and optimise the two main sub-systems of the power cycle,  
 117 which were namely (i) the split-cycle boiler and turbine sub-system, where the optimal boiler *pressure* and  
 118 *inlet temperature* on the working fluid side were determined, and (ii) the mixing and splitting arrange-  
 119 ment, where the *split fractions* and corresponding *flow rates* were calculated to reach the desired ammonia  
 120 concentrations of the working mixtures.

#### 121 3.1.1. Thermodynamic level - sub-modelling

122 The prediction of the thermophysical properties of the ammonia-water mixture builds on the model  
 123 developed by Tillner-Roth and Friend. This equation of state is applicable in the liquid and vapour phase  
 124 regions and is suitable for predicting the vapour-liquid equilibrium [17]. The main difference with the  
 125 other equations of state used for ammonia-water mixtures (Stecco-Desideri and Ibrahim and Klein) lies in  
 126 the addition of a correction term to the ideal mixture behaviour in the EOS, making this EOS suitable  
 127 for pressures up to 40 MPa. This EOS was compared to the two others in a work of Thorin [18] and is  
 128 more accurate, which is of particular importance as the value of the thermal efficiency can vary up to 1.5 %

129 points when using different EOS. However, this EOS is characterised by a significantly higher computational  
 130 time than conventional cubic EOS such as Peng-Robinson or Redlich-Kwong with Soave modifications, but  
 131 presents a better accuracy.

### 132 3.1.2. Subsystem level - sub-modelling

133 The division of a system into its sub-systems eased the initialisation of the state and operating parameters  
 134 for performing the optimisation procedure, and the two sub-models were validated by using equivalent models  
 135 in Aspen PLUS [19].

136 *Heat exchanger sub-system.* A model of the boiler was developed, based on the initial waste heat source  
 137 conditions given in Bombarda et al. [5], in order to analyse the entropy generation trends. As the power  
 138 output is proportional to the flow rate of the working fluid flowing through the boiler, the aim is to determine  
 139 correlations between the maximum possible flow rate and the compositions of the lean and rich streams.  
 140 The composition of the lean stream is linked to the composition of the rich one, since (i) the temperatures  
 141 and pressures of these streams are equal at the outlet of the first evaporator, and (ii) the rich stream is in  
 142 saturated vapour state and the lean stream is in saturated liquid state. The optimisation bases therefore on  
 143 a computation of the rich stream composition against the mass flow rate of the working fluid, under these  
 144 three constraints suggested by Kalina [11]:

$$T_{\text{dew,rich}} = T(p = p_{\text{wf}}, Q = 1, x = x_{\text{rich}}) \quad (1)$$

$$T_{\text{bub,lean}} = T(p = p_{\text{wf}}, Q = 0, x = x_{\text{lean}}) \quad (2)$$

$$T_{\text{dew,rich}} - T_{\text{bub,lean}} = 0 \quad (3)$$

145 *Mixing and splitting sub-system.* The aim of this model is to calculate the necessary splitting fractions to  
 146 achieve the desired flows and concentrations of the lean and rich streams entering the boiler, based on the  
 147 temperature and pressure conditions at the flash separator inlet. This separation is assumed adiabatic.  
 148 The coupling of these two models allows for a sound and computationally more-efficient optimisation of the  
 149 Kalina Split Cycle, under fixed boundary conditions.

### 150 3.1.3. Optimisation

151 These models were further integrated into a model of the complete system to investigate its feasibility.  
 152 The optimisation was carried out using a genetic algorithm [12]: the optimisation parameters (process  
 153 variables) are emulated, as if they were genes of an individual, and the performance of each individual is  
 154 evaluated. The objective of the optimisation routine is to maximise the net power production, considering  
 155 nine process parameters as decision variables:

- 156 • temperature  $T_{10}$ , pressure  $p_{10}$  and concentration  $x_{10}$  of the working mixture at the inlet of the separa-  
 157 tion sub-system;
- 158 • working solution concentration  $x_3$ ;
- 159 • turbine inlet  $p_3$ , reheat  $p_{3'}$  and outlet  $p_4$  pressures;
- 160 • boiler temperature approach;
- 161 • rich stream concentration  $x_{20}$ .

162 The discharge temperature and mass flow rate of the heat source are kept constant, implying that  
 163 maximising the net power output is equivalent to maximising the thermal efficiency of the Kalina split-  
 164 cycle. The first generation of individuals is randomly generated, while the next ones are stochastically  
 165 selected based on the values of the optimisation function. The following generations are used in the next  
 166 iteration of the genetic algorithm, resulting in sets of best possible combinations of process parameters.  
 167 The initial population size is set to 400-800, the number of generations is 30 and the number of evaluations

168 between 12,000 and 24,000, to ensure that the global optimum is found in a search space where several local  
 169 optima are present. The number of sub- populations, the cross-over rate and the generation gap are set to  
 170 4, 1 and 0.8, while the mutation, insertion and migration rates are set to 0.5, 0.9 and 0.2. The MATLAB  
 171 GA-toolbox was used to perform the optimisations.

## 172 3.2. System analysis

### 173 3.2.1. Strategy

174 The study performed in this work can be divided into five consecutive steps:

- 175 1. the Kalina split-cycle is compared to the conventional Kalina and organic Rankine cycles, with the  
 176 working fluid and boundary conditions proposed in Bombarda et al. [5], in order to have a compre-  
 177 hensive overview of the differences between the various waste heat recovery cycles;
- 178 2. four configurations of the Kalina cycle and split-cycle were studied (denoted *Case A* for the Kalina  
 179 cycle without reheat, *Case B* for the Kalina cycle with reheat, *Case C* for the Kalina split-cycle  
 180 without reheat, and *Case D* for the Kalina split-cycle with reheat) to deduce further optimisation  
 181 possibilities;
- 182 3. a sensitivity analysis on the performance of the Kalina split-cycle for nine boundary conditions was  
 183 conducted (Table 2). The efficiency of this power cycle in other ambient conditions was investigated  
 184 by varying the cooling water temperature, and the suitability to low- and medium-temperature ap-  
 185 plications was studied by changing the hot source inlet temperature. The effect of lower exhaust  
 186 temperatures was considered, starting from a temperature of 160 °C to 100 °C, with a step of 15 °C.  
 187 Exhaust temperatures of 100 °C are currently not achievable because of practical issues with possible  
 188 sulphur condensation in the fumes, resulting in material corrosion in the chimneys. They may be  
 189 adequate in the future for fuels with a low sulphur content, natural gas, or with the development of  
 190 new technologies;
- 191 4. the relationship between the ammonia mass fraction and the exergy flows, as well as with the distri-  
 192 bution of the exergy destruction, is investigated. The ammonia mass fraction is varied between 0.7  
 193 and 0.8 with a step of 0.05. The high pressure is kept constant, meaning that the process parameters  
 194 that vary are the mass flow rate, the intermediate and low pressure levels, and the split fractions in  
 195 the mixing/separation system;
- 196 5. the entropy generation phenomenon inside the boiler is analysed by discretization of the heat exchang-  
 197 ers of the Kalina split-cycle in finite control volumes.

Table 2: Sensitivity boundary conditions

#	T <sub>34</sub> [°C]	T <sub>37</sub> [°C]	T <sub>cw</sub> [°C]
<b>1</b>	350	160	40
<b>2</b>	350	160	25
<b>3</b>	350	160	10
<b>4</b>	300	160	25
<b>5</b>	250	160	25
<b>6</b>	350	100	25
<b>7</b>	350	115	25
<b>8</b>	350	130	25
<b>9</b>	350	145	25



198 *3.2.2. Thermodynamic analysis*

199 *Exergy* may be defined as the *maximum theoretical useful work as the system is brought into com-*  
 200 *plete thermodynamic equilibrium with the thermodynamic environment while the system interacts with this*  
 201 *environment only* [20]. Exergy is not conserved in real processes as it is destroyed because of internal  
 202 irreversibilities.

203 Neglecting the potential and kinetic effects, the exergy associated with a stream of matter is a function  
 204 of its physical  $e^{\text{ph}}$  and chemical  $e^{\text{ch}}$  components [20]. It is expressed, on a specific mass basis, as follows:

$$e = e^{\text{ph}} + e^{\text{ch}} \quad (4)$$

205 Physical exergy accounts for temperature and pressure differences from the environmental conditions  
 206 and is defined as:

$$e^{\text{ph}} = (h - h_0) - T_0(s - s_0) \quad (5)$$

207 where  $s$  is the specific entropy of a stream of matter per unit-of-mass, respectively.

208 Chemical exergy accounts for deviations in chemical composition from reference substances present in  
 209 the environment. In this work, chemical exergy is calculated based on the concept of *standard chemical*  
 210 *exergy* discussed by Moran and Shapiro [21]. The specific chemical exergy of real chemical compounds is  
 211 determined using the reference environment defined by Szargut [22–24].

212 The tracing of the energy and exergy flows provides information on the system transformations and  
 213 inefficiencies. Several performance parameters were developed to illustrate the possibilities for improvements  
 214 and illustrate the components on which improvement efforts should focus [20,25–27]:

- The exergy destruction ratio  $y_d^*$  is defined as the ratio of the exergy destruction rate  $\dot{E}_{d,k}$  within a specific component  $k$  to the total exergy destruction rate in the overall system  $\dot{E}_d$ :

$$y_d^* = \frac{\dot{E}_{d,k}}{\dot{E}_d} \quad (6)$$

- The exergetic efficiency  $\varepsilon$  of a given component or sub-system  $k$ , which reflects its thermodynamic performance. It is defined as the ratio of the product exergy to the fuel exergy.

$$\varepsilon_k = \frac{\dot{E}_{p,k}}{\dot{E}_{f,k}} = 1 - \frac{\dot{E}_{d,k}}{\dot{E}_{f,k}} \quad (7)$$

217 The product exergy  $\dot{E}_{p,k}$  represents the desired effect of a given thermodynamic transformation or  
 218 process, while the fuel exergy  $\dot{E}_{f,k}$  represents the resources expended in this component/sub-system  
 219 to generate the desired result. The definitions of exergetic fuels and products for the components  
 220 existing in power cycle processes are introduced and discussed in Kotas [25–27] and in Bejan et al.  
 221 [20]. The exergy losses with cooling water and exhaust gases cannot be allocated to a single component,  
 222 but rather to the complete system, and are therefore not accounted for in the definition of the exergy  
 223 efficiency of a component.

224 The exergy accounting for the Kalina split-cycle can therefore be expressed as follows:

$$\dot{E}_{\text{heat}}^Q = \dot{E}^W + \dot{E}_{\text{cw}}^Q + \dot{E}_{\text{exh}}^Q + \dot{E}_d \quad (8)$$

225 where  $\dot{E}_{\text{heat}}^Q$  represents the exergy input to the power cycle with the heat source,  $\dot{E}^W$  denotes the exergy  
 226 associated with the net power produced,  $\dot{E}_{\text{cw}}^Q$  and  $\dot{E}_{\text{exh}}^Q$  the exergy lost to the environment with the cooling  
 227 water and exhaust gases, and  $\dot{E}_d$  the exergy destroyed in the plant. The dead state was taken to the  
 228 environmental conditions, i.e. 5 °C and 1.013 bar, and the reference environment of Szargut [28] is considered  
 229 for the chemical exergy calculations.

230 *3.2.3. Economic analysis*

231 An economic and a life cycle assessment are performed to further compare the three cycles, i.e. the  
 232 organic Rankine cycle, Kalina cycle and Kalina split-cycle. The grassroot costs are estimated from the  
 233 capacity-based correlations, based on design and operating parameters such as the heat transfer area, the  
 234 thermal and power loads, and they are characterised by an uncertainty of  $\pm 30\%$ . The estimates of the  
 235 heat transfer areas were taken from Larsen et al. [12]. For more details on the used correlations, the reader  
 236 is referred to Turton et al. [29]. The interest rate is set to 6% and the lifetime of the equipment to 15 years.

237 *3.2.4. Life cycle analysis*

238 The environmental impacts of integrating waste heat recovery cycles are estimated by taking into con-  
 239 sideration the environmental burdens during the whole life cycle, including the manufacturing, operating  
 240 and decommissioning steps, and by applying the approach of Gerber et al. [30]. The impacts are normalised  
 241 with respect to the functional unit of the power cycle, which is, in this work, taken to be 1 GJ of electricity,  
 242 and they are adjusted for an operating availability of 95%. The environmental effects that are investi-  
 243 gated are, namely, the global warming potential, over a horizon of 100 years, the ozone depletion potential,  
 244 the acidification and eutrophication effects, the human toxicity and the marine ecotoxicity. The pollutant  
 245 emissions are indexed on the CO<sub>2</sub>, CFC-11, SO<sub>2</sub>, PO<sub>4</sub><sup>-</sup> and 1,4-DCEB compounds.

246 **4. Results and discussion**

247 *4.1. Comparison of the Organic Rankine Cycle, Kalina Cycle and Kalina Split Cycle*

248 The Organic Rankine cycle, with the boundary conditions, the process parameters and the working fluid  
 249 suggested in Bombarda et al. [5], is compared to the Kalina cycle and the Kalina split-cycle. It should be  
 250 mentioned that the Organic Rankine cycle proposed in their work considers an organic fluid operated in  
 251 subcritical conditions, and that there could be more efficient and optimised cycles with fluids operating in  
 252 supercritical ones. However, such an investigation is out of scope of this study, and this comparison aims at  
 253 illustrating the most important differences between the three cycles.

Table 3: Comparison of the Organic Rankine cycle, Kalina cycle and Kalina split-cycle, based on thermodynamic, economic and environmental performance indicators.

	Organic Rankine cycle (hexamethyldisiloxane)	Kalina cycle without reheat	Kalina split-cycle with reheat
<i>Thermodynamic evaluation</i>			
Net power generation, kW	1603	1753	1910
Thermal efficiency, %	21.5	23.6	25.7
Low pressure, bar	0.12	6.5	4.9
High pressure, bar	9.74	150	125.1
<i>Economic assessment</i>			
Total grassroot costs, M\$	8.14	9.99	11.31
Recuperators, %	2.84	4.43	5.18
Condensers, %	1.99	3.32	3.01
Boiler, %	13.81	21.02	27.04
Separator, %	-	0.66	0.40
Pumps, %	1.96	4.20	6.01
Turbine, %	79.41	66.36	58.36
Production cost, \$/MWh	61.6	71.1	80.9
<i>Environmental impacts</i>			
Acidification, kg SO <sub>2</sub> -eq	-0.07	-0.069	-0.068
Eutrophication, 10 <sup>-3</sup> kg PO <sub>4</sub> -eq	-6.50	-6.40	-6.25
Global warming potential, kg CO <sub>2</sub> -eq	-58	-58	-59
Ozone depletion, 10 <sup>-6</sup> kg CFC11-eq	-2.20	-2.10	-2.15

254 The Kalina split-cycle is characterised by greater grassroot costs, as the number of components in this  
 255 configuration is much higher than for a conventional Kalina cycle, and thus even higher than for an Organic  
 256 Rankine cycle. This main increase of costs can be imputed to the greater complexity of the boiler, as streams  
 257 of different concentrations should be handled separately. Despite the higher thermal efficiency of this power  
 258 cycle, the total and production costs are 30-40% higher compared to an Organic Rankine cycle, and the

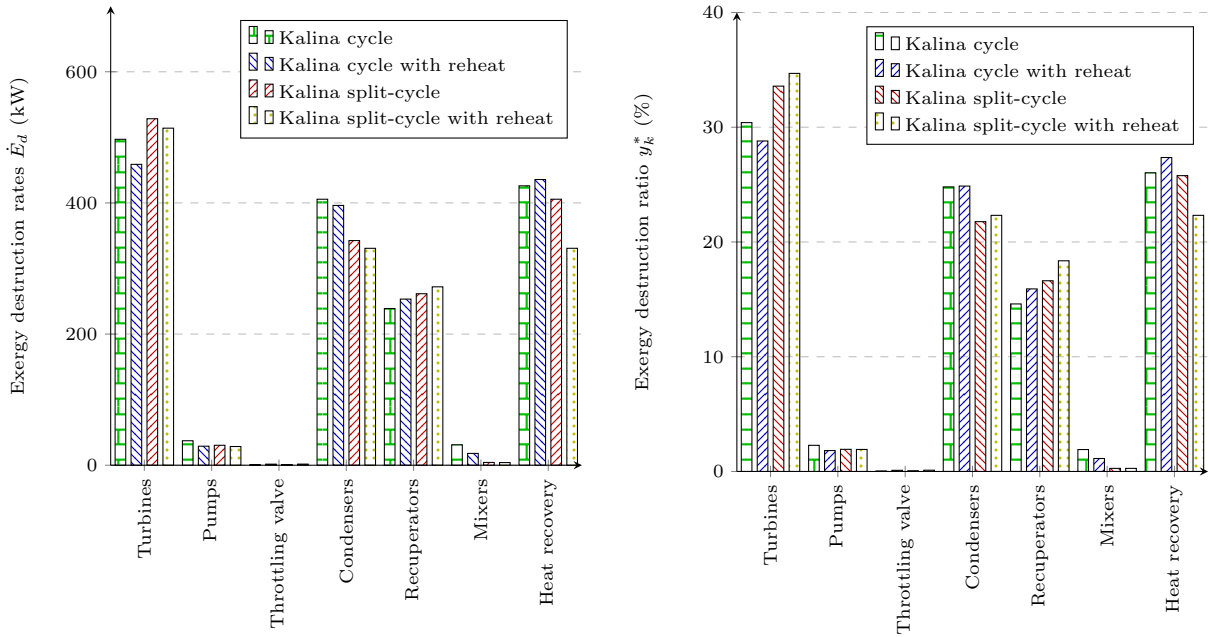


Figure 2: Comparison of the exergy destruction rates and exergy destruction ratios in the four Kalina cycle configurations (Kalina cycle (Case A), Kalina cycle with reheat (Case B), Kalina split-cycle (Case C), Kalina split-cycle with reheat (Case D)). The exergy losses associated with the rejection of the exhaust gases are not shown, since they are equal in the four cases.

259 pumps and turbines operate over larger pressure ratios (Table 3). Integrating a waste heat recovery cycle has  
 260 overall positive environmental effects, as electricity from the grid is substituted. The emissions caused by the  
 261 component construction, maintenance and operation are compensated by the reduction in most pollutants  
 262 emissions, with the exception of the ones contributing to human toxicity and marine ecotoxicity.

#### 263 4.2. Comparison of the 4 configurations

264 The exergy destruction in the Kalina cycle amount to 1726 kW (Case A), 1681 kW (Case B), 1662 kW  
 265 (Case C) and 1575 kW (Case D). The exergy efficiencies are 26.5 %, 27.5 %, 27.0 % and 28.9 %, respectively.  
 266 The exergy analysis indicates that the implementation of the reheat and split-cycle configurations result in  
 267 reduced exergy destruction and losses by 2.5–4.9 % and 2.7–5.1 % (Figure 2). In all cases, the greatest  
 268 irreversibilities take place (1) in the turbine(s), (2) in the heat recovery system, and (3) in the condensers  
 269 and recuperators.

270 The Kalina split-cycle is characterised by a significant reduction of the exergy destruction in heat transfer  
 271 processes, mainly because most desuperheating of the rich stream takes place in the recuperators, rather  
 272 than in the intermediate pressure condenser (36–38 % in relation to the baseline case) and heat recovery  
 273 system (4.8–24 %). In contrast, the exergy destruction in the recuperators is higher, because more heat  
 274 is transferred in these heat exchangers. The exergy losses associated with the rejection of the exhaust  
 275 gases to the environment are constant and equal to 5640 kW, since the comparison of the four Kalina cycle  
 276 configurations is based on fixed temperature and pressure of the exhaust gases at the outlet of the heat  
 277 recovery system. On the contrary, the exergy losses with the cooling water are slightly higher in the Kalina  
 278 split-cycle cases.

279 The Sankey diagram of the Kalina split-cycle with reheat (Figure 3) illustrates graphically the exergy  
 280 flows within the plant, as well as the main sources of inefficiencies and losses. The greatest exergy destruc-  
 281 tion take place in (i) the low-pressure turbine, (ii) the boiler, (iii) the condenser of the lean mixture, and (iv)  
 282 the recuperator placed at the outlet of the turbines. The other contributions are moderate in comparison,  
 283 accounting for less than 100 kW each. The irreversibilities corresponding to the point (i) are related to the  
 284 inefficiencies of the low-pressure turbine, which is characterised by a significant pressure ratio. The ones

285 corresponding to the points (ii), (iii) and (iv) are caused by the heat transfer from the exhaust gases to the  
 286 working mixture, and from the working mixture to the cooling water.

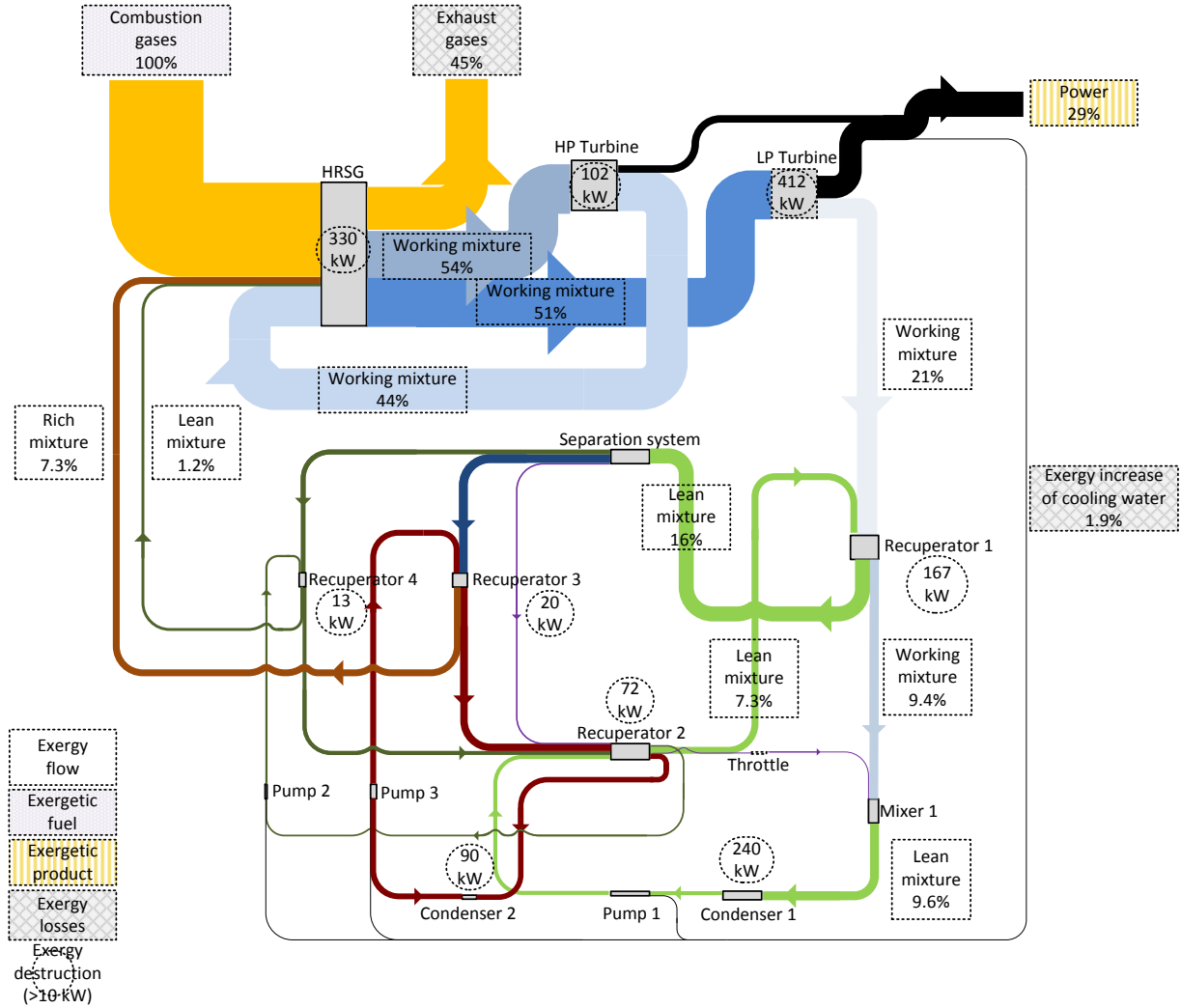


Figure 3: Sankey diagram of the Kalina split-cycle process with reheat. The arrangement of mixers and splitters to reach the desired ammonia and water concentrations in the rich and lean fluids is represented graphically as the separation system. Only the irreversibilities greater than 10 kW are shown, and the exergy accounting on the diagram considers that the exergy destruction caused by chemical mixing is negligible towards the other contributions and the physical exergy of each material stream.

#### 287 4.3. Effect of the boundary conditions

288 The optimisation results for the sensitivity analysis presented in Table 2 are shown in Table 4. It  
 289 suggests that there is no direct correlation between the optimum mass flow rate and the temperature of  
 290 the environment. Lower cooling water temperatures result in higher optimal high pressure levels, as well as  
 291 lower condensation temperatures of the working mixture. The ammonia fraction can therefore be increased,  
 292 which results in a greater power output. Higher exhaust temperatures result in smaller working mixture  
 293 flow rates, as well as higher high pressure levels, since a smaller amount of heat can be recovered in the  
 294 waste heat recovery system.

Table 4: Optimum design parameters, net power generation and exergetic efficiency of the Kalina split-cycle at different boundary conditions.

#	$\dot{m}$	$p_3$	$p_{10}$	$p_4$	$x_a$	$\dot{W}$	$\varepsilon$
1	3.31	127	13.9	4.3	0.634	1694	28.5
2	3.14	126.8	11.2	3.2	0.671	1813	28.2
3	3.45	129.5	9.9	3.5	0.754	1939	27.6
4	2.85	128.7	12.1	6.2	0.766	1211	21.9
5	1.90	65.4	11.4	4	0.657	668.5	14.5
6	3.96	62.7	10.8	2.9	0.685	2055	31.8
7	3.78	70.9	10.2	2.8	0.663	2027	31.2
8	3.78	101	10.4	3.6	0.692	2006	30.8
9	3.36	125.4	9.5	3	0.691	1974	30.4

295 The net power generation of the Kalina split-cycle is mostly sensitive (i) to the cooling water temperature,  
 296 increasing by about 7 % for a decrement of 15 °C, and (ii) to the inlet temperature of the waste heat source,  
 297 decreasing by about 40 % for a reduction of 50 °C. Similarly, the exergy efficiency of this power cycle is  
 298 very sensitive to the waste heat temperatures, while it is moderately changing with the cooling water and  
 299 exhaust gas temperatures (Table 4).

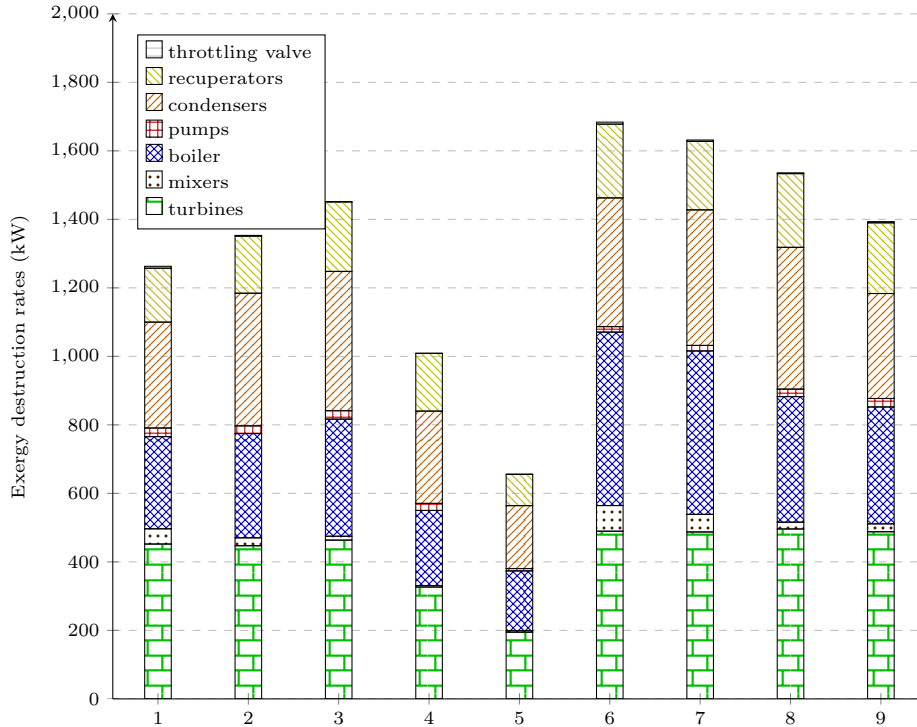


Figure 4: Exergy destruction sorted by processes, under different boundary conditions (temperatures of the heat source, heat rejection and cooling water).

300 The greatest exergy destruction can be found in the turbines, boiler and condenser, while the irreversibil-  
 301 ities taking place in the mixers, pumps and throttling valves are negligible (Figure 4). The turbines are the  
 302 most exergy-destroying system for all cases investigated in this work, with the exception of Case 6 where the  
 303 boiler ranks first. This can be explained by the larger temperature difference between the inlet and outlet  
 304 temperatures on the gas side, implying that the exergy destruction caused by heat transfer is higher.

305 When investigating each individual component, it can be seen that the boiler and the high- and low-  
 306 pressure turbines are generally the most exergy-destroying components (Figure 5), although they also display  
 307 the highest exergy efficiency, of 87–91 %, 80–84 % and 78–81 %, respectively. This high value is related

308 to the improved match between the temperature profiles of the heat source and receiver. However, the  
 309 boiler is also responsible for large exergy destruction, as the total flow rate of the working solution flows  
 310 through this component, and heat transfer takes place over a large range of temperatures. The low-pressure  
 311 condenser and high-pressure turbine rank third or fourth most exergy-destroying component in all cases, as  
 312 large amounts of exergy are transferred at low temperatures and the high-pressure turbine operates over a  
 313 large pressure ratio.

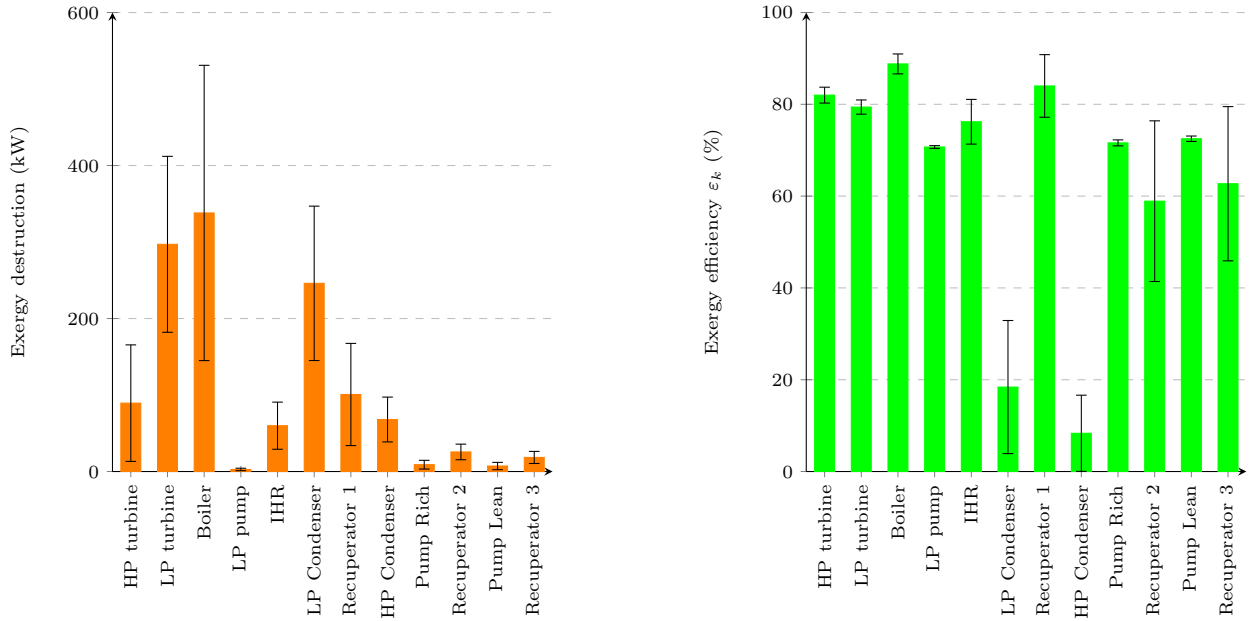


Figure 5: Exergy destruction sorted per component (left), under different boundary conditions (temperatures of the heat source, heat rejection and cooling water). Exergy efficiency per component  $\varepsilon_k$  (right), under different boundary conditions (temperatures of the heat source, heat rejection and cooling water). The range indication represents the minimum and maximum values for the specific component across the examined cases.

314 The pumps destroy little exergy and present an efficiency of about 70 %. The first and second recupera-  
 315 tors, where heat is recovered from the turbine effluent, are more efficient by 10–15 % points than the third  
 316 and fourth recuperators, where the rich and lean working fluids are preheated before entering the boiler.

317 The exergy losses with cooling water are negligible in comparison to the ones associated with exhaust  
 318 gases. The latter decrease with the limit set on the exhaust temperature, from about 3500 to 2050 kW when  
 319 it decreases from 160 °C to 100 °C.

320 The exergy destruction taking place in the turbines vary significantly with the boundary conditions, while  
 321 their exergetic efficiencies change marginally. This suggests that the variations of the exergy destruction  
 322 in these components are mostly correlated to (i) the variations of the pressure ratios of the high- and low-  
 323 pressure turbines, and to (ii) the variations of the working mixture flow rates. On the opposite, the exergy  
 324 efficiency of the condensers is directly impacted by changes in the cooling water temperature, as exergy is  
 325 dumped into the environment at low temperatures. Similarly, the exergy efficiency of the recuperators is  
 326 affected by the exhaust temperature, as this results in different operating conditions of the recuperators,  
 327 and therefore in different temperature gaps between the hot and cold streams.

#### 328 4.4. Effect of the ammonia mass fraction

329 The relation between the ammonia mass fraction and the exergy flows entering the several components,  
 330 as well as their exergy destruction and efficiency, are investigated. The ammonia mass fraction is fixed to  
 331 0.7, 0.75 and 0.8, while the high pressure is kept constant and the other process parameters (e.g. mass flow  
 332 rate) are taken as decision variables in the optimisation routine. For the same boundary conditions, the

333 total exergy destruction in the Kalina split-cycle increases (Fig 6), and this is mainly caused by the higher  
 334 exergy destruction taking place in the recuperators.

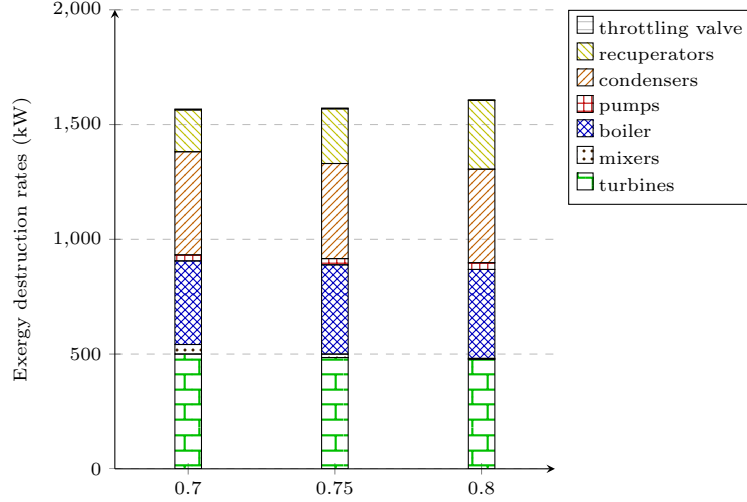


Figure 6: Exergy destruction sorted by processes, with three different ammonia concentrations (0.7, 0.75, 0.8)

335 The variations of the exergy flows and destruction for the three cases are shown in Table 5 – Table 7.  
 336 Despite the different compositions, the exergetic efficiency of the turbines is unaffected. However, as the  
 337 optimal flow rate of the working solution increases with the ammonia concentration, the amount of exergy  
 338 entering and exiting the cycle components increases in all cases.

Table 5: Exergy inflows, outflows and destruction for an ammonia fraction of 0.7. Only the physical exergy flows are presented for clarity, as they are the only ones varying in the heat exchangers and turbines. The terms HP and LP stand for high-pressure and low-pressure, T for turbine, CD for condenser, PP for pump and REC for recuperator.

	HPT	LPT	Boiler	LPP	IREC	LPCD	REC1	RCD	RPP	REC2	REC3	LPP
Inflows, kW	3228	3063	9661	136	995	541	1544	342	303	736	392	121
Outflows, kW	3121	2671	9297	132	915	168	1468	266	292	719	382	109
Destruction, kW	108	393	364	3	80	373	76	77	11	17	9	12
Efficiency, %	82.2	80.1	89.9	70.7	80.2	10.3	86.0	2.4	71.5	52.6	59.5	72.8

Table 6: Exergy inflows, outflows and destruction for an ammonia fraction of 0.75. Only the physical exergy flows are presented for clarity, as they are the only ones varying in the heat exchangers and turbines. The terms HP and LP stand for high-pressure and low-pressure, T for turbine, CD for condenser, PP for pump and REC for recuperator.

	HPT	LPT	Boiler	LPP	IREC	LPCD	REC1	RCD	RPP	REC2	REC3	LPP
Inflows, kW	3223	2950	9572	208	1082	546	1618	472	421	1002	272	90
Outflows, kW	3081	2608	9183	205	1009	234	1481	369	406	981	264	81
Destruction, kW	142	342	389	3	72	312	137	102	16	21	8	9
Efficiency, %	82.2	80.1	89.9	70.9	76.8	10.2	78.7	2.5	71.4	59.7	65.3	72.7

339 It is worth noticing that a higher ammonia fraction of the working solution results in a smaller exergetic  
 340 efficiency of all components operating at the low- and medium-pressure levels, except the ammonia-rich  
 341 condenser. This is caused by the larger temperature gaps in the recuperators and condensers, which is  
 342 evident in the case of the recuperator placed at the outlet of the turbine (REC1), with an efficiency drop of  
 343 about 15 % points. On the contrary, the components operating at the high-pressure level, i.e. the rich and  
 344 lean recuperators before the boiler, perform better, with an increase of the exergetic efficiency of about 20  
 345 % points.

Table 7: Exergy inflows, outflows and destruction for an ammonia fraction of 0.8. Only the physical exergy flows are presented for clarity, as they are the only ones varying in the heat exchangers and turbines. The terms HP and LP stand for high-pressure and low-pressure, T for turbine, CD for condenser, PP for pump and REC for recuperator.

	HPT	LPT	Boiler	LPP	IREC	LPCD	REC1	RCD	RPP	REC2	REC3	LPP
Inflows, kW	3485	3273	10151	368	1296	678	1930	610	550	1289	309	106
Outflows, kW	3368	2913	9764	366	1234	392	1722	488	531	1267	300	97
Destruction, kW	117	360	388	2	61	286	208	122	19	22	9	8
Efficiency, %	82.2	80.1	89.9	71.1	72.9	9.5	71.6	5.2	71.5	75.2	76.2	72.5

#### 4.5. Entropy generation in the heat recovery system

For the preheater, boiler and superheater, Figure 7 shows the increase in generated entropy as heat is being transferred to the Kalina cycle. For a large part of the heat transfer the slope of the four curves is almost equal, indicating similar entropy generation. The major difference stems from the initial and final parts of the heat transfer. As previously discussed, the Kalina split-cycle offers a better match with the heat source around the pinch point, which is indicated here by the smaller rate of entropy generation at the beginning of the heat exchange.

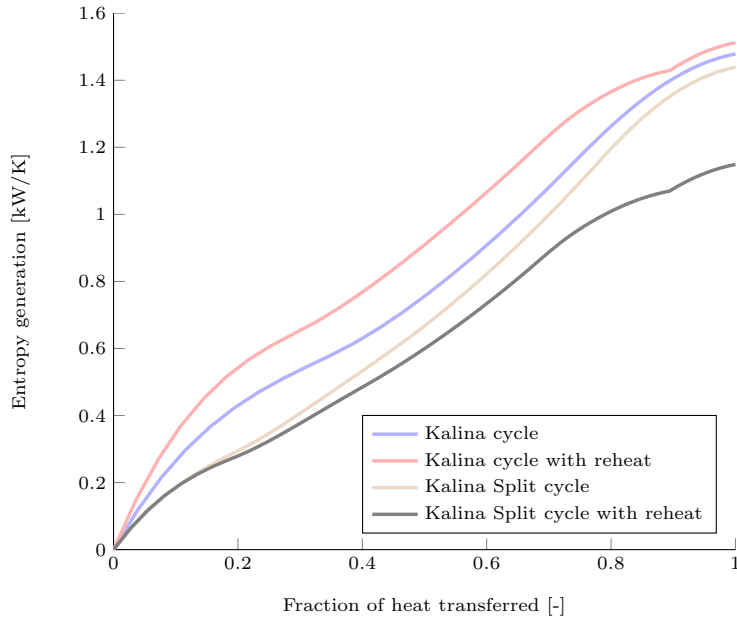


Figure 7: Entropy generation in preheater, boiler and superheater as a function of the total heat transferred. When reheating is used the heat related to the reheating is treated separately and is added at the ends of the curves

The addition of reheat increases the entropy generation in the case of the Kalina Cycle, and the main difference can again be seen at the start and end of the heat exchange. The Kalina cycle with reheat operates at a lower pressure: this results in a larger entropy generation at the start, but in a better match at the end. The overall entropy generation is therefore almost equal with and without reheat.

For the Kalina split-cycle, the opposite conclusions can be drawn, since the addition of reheat results in a reduction in entropy generation. The reheat also improves the temperature matches at the end of the heat transfer, but has little effect on the allowable inlet pressure, because of the smoothing near the pinch point.

#### 4.6. Recommendations

These analyses provide highlights on the behaviour of the Kalina split-cycle in different environmental conditions. They suggest that integration opportunities of this power cycle exist for low- and medium-temperature waste heat recovery, and for different cooling water conditions. The Kalina Split Cycle is more



364 exergy-efficient when the heat source outlet temperature is low, and when the difference between the heat  
 365 source inlet and outlet temperatures is large, as for conventional power cycles.

366 The results indicate that the process can provide near-optimum efficiencies across a large range of process  
 367 parameters, due to the large number of parameters available for adjustment. This is for example seen by the  
 368 results in Table 4, where Cases 7 and 8, yield very similar power outputs under similar boundary conditions,  
 369 while the optimum design parameters found are not entirely similar (dissimilar high pressure level, but  
 370 similar medium- and low-pressure ranges, as well as ammonia fraction).

371 In general, the different findings from this work highlight the following trends, which could be used as  
 372 guidelines for further integration, and possibly for improving the exergetic efficiency in such power cycles  
 373 (Table 8).

Table 8: Trends of the Kalina split-cycle

	$T_{cw} \searrow$	$T_{34} \searrow$	$T_{37} \searrow$	$\dot{m} \nearrow$
Ammonia mass flowrate	$\rightarrow$	$\searrow$	$\nearrow$	$\nearrow$
High-level pressure	$\rightarrow$	$\searrow$	$\nearrow$	-
Medium level-pressure	$\searrow$	$\searrow$	$\rightarrow$	$\nearrow$
Low-level pressure	$\searrow$	$\searrow$	$\rightarrow$	$\nearrow$
Ammonia mass fraction	$\nearrow$	$\searrow$	$\rightarrow$	$\nearrow$
Boiler destruction	$\nearrow$	$\searrow$	$\nearrow$	$\nearrow$
Boiler efficiency	$\nearrow$	$\searrow$	$\searrow$	$\rightarrow$
Turbine destruction	$\nearrow$	$\searrow$	$\rightarrow$	$\nearrow$
Turbine efficiency	$\nearrow$	$\searrow$	$\rightarrow$	$\rightarrow$
Exergy efficiency	$\nearrow$	$\searrow$	$\nearrow$	$\nearrow \searrow$

374 A lower cooling water temperature ( $T_{cw} \searrow$ ) allows for a lower condensation pressure of the ammonia-  
 375 water solution and therefore a higher ammonia fraction. The medium- and low-pressure levels can be  
 376 decreased: this implies that the average temperature of heat rejection is decreased, which would have  
 377 beneficial effects on the thermal efficiency (i.e. greater power generation from the turbines). Different  
 378 cooling water temperatures have a significant impact on the net power production, which increases by about  
 379 15 % when the cold reservoir is at 5 °C instead of 35 °C. Such trends are expected for power cycles.

380 A lower heat source temperature ( $T_{34} \searrow$ ), for the same exhaust temperature ( $T_{37}$ ), implies that the  
 381 amount of waste heat available for power generation decreases. The flow rate of the working solution should  
 382 therefore be reduced, and this causes a drop in the exergy destruction in the boiler and turbines, as well as  
 383 in the complete cycle. Similarly, the boiler and turbine exergetic efficiencies decrease, as a consequence of a  
 384 lower temperature at the outlet of the boiler and at the inlet of the turbines.

385 A lower exhaust temperature ( $T_{37} \searrow$ ) results in a greater uptake of waste heat, and this allows for a  
 386 higher flow rate of the working solution. The boiler pressure can as well be increased to reach the same  
 387 temperature difference at the pinch point than in the baseline case, and this leads to a greater power output  
 388 from the turbines. It is worth mentioning that: (i) the heat source temperature and superheating approach  
 389 are maintained constant, the turbine inlet temperature is therefore fixed, and this explains why the turbine  
 390 efficiency and destruction do not vary, (ii) the heat exchange in the boiler takes place over a larger range of  
 391 temperatures, resulting in greater exergy destruction in this component, and (iii) the low- and medium-level  
 392 pressure levels do not vary, as the ammonia concentration is nearly constant.

393 Regarding the different trends of the Kalina split-cycle, and the outcomes from the different steps of the  
 394 analysis, two different optimisation strategies can be followed when integrating such a power cycle based on  
 395 an ammonia-water mixture:

- 396 • enabling a high turbine inlet pressure (around 125–130 bar) and a high ammonia fraction (0.75–0.8);
- 397 or
- 398 • enabling a low turbine outlet pressure and a low ammonia fraction (0.6–0.7).

399 The presence of these two strategies makes it difficult to recommend an explicit optimal operating  
 400 range for a specific application, as there are two sets of conditions that can yield near-optimum solutions.  
 401 The presented ranges are then given as guiding suggestions. In low- and medium-temperature waste heat

402 recovery applications, and for moderate cooling water temperatures ( $T_{cw} \simeq 15\text{--}20\text{ }^\circ\text{C}$ ), the optimum low-  
403 and medium-pressure levels are in the ranges 3–4 bar and 9–11 bar, respectively. The optimal high-pressure  
404 level is generally about 120–130 bar, and may be decreased with smaller temperatures of the heat source or  
405 of the allowable rejection temperature.

406 At higher heat source temperatures than the ones investigated in this paper, the pressure may become a  
407 limiting factor. Both the Kalina cycle and Kalina split-cycle were already operating at pressures of 150 and  
408 125 bar, respectively, which is significantly higher than the operating pressure of the Organic Rankine Cycle  
409 proposed by Bombarda et al. [5]. Furthermore, the high concentration of ammonia in the rich stream of  
410 the Kalina split-cycle results in a critical pressure of only 149 bar. Since the operating pressure of the base  
411 Kalina cycle is higher, it is expected that the Kalina split-cycle could also reach a scenario where the rich  
412 stream is in a super-critical state. This does not influence the working principle of the Kalina split-cycle,  
413 but may further complicate the cycle design and operation. Such issues should be taken into account when  
414 applying the split-cycle to higher temperature heat sources.

415 Finally, practical integration may also be challenging, as using ammonia-water mixtures implies higher  
416 safety requirements. Regarding environmental aspects, ammonia-water mixtures do not have any global  
417 warming [31] or ozone depletion potentials [32] but may cause water eutrophication [33], soil acidification  
418 [34] and can affect the human health. However, the preliminary life cycle analyses on the waste heat recovery  
419 cycles suggest that the benefits of substituting electricity from the grid are more valuable than the possible  
420 harms caused by the components manufacturing and the use of an ammonia-water mixture. The exact  
421 benefits should be quantified for each specific site and location, as different facilities have different practical  
422 requirements, and different countries operate on different electricity mixes.

## 423 5. Conclusion

424 An alternative process configuration of the Kalina cycle, namely the Kalina split-cycle, was investigated  
425 in the present work. Design parameters such as the ammonia concentration of the working fluid, the turbine  
426 pressure and the splitting fractions of the mixing and separation sub-system were optimised by application  
427 of a genetic algorithm.

428 The conventional Kalina cycle and the Kalina split-cycle, with and without reheat, were compared by  
429 conducting an exergy analysis. The greatest thermodynamic performance is reached with the Kalina split-  
430 cycle with reheat, as the result of a lower turbine exhaust pressure and a higher boiler inlet temperature,  
431 compared with the conventional Kalina cycle. These benefits are achieved at the expense of a higher system  
432 complexity.

433 The irreversibilities taking place in this unique power cycle are smaller by 2.5 to 5.0 % compared to the  
434 reference Kalina cycle. The most significant reduction in exergy destruction is related to the boiler: the  
435 temperature match between the heat source and receiver is improved, and this leads to a smaller entropy  
436 generation. The largest exergy destruction takes place in the turbines and the boiler, and the least exergy-  
437 efficient components are the condensers and the recuperators.

438 A sensitivity analysis on the boundary conditions, i.e. the cooling water and exhaust gas temperatures,  
439 was performed. The results suggest that the net power output is mostly sensitive to the cooling water and  
440 inlet temperature of the waste heat source, while the exergy efficiency is mostly affected by the latter. The  
441 components that display the highest variation in exergy destruction are the boiler and the low-pressure  
442 turbine, but the ones that display the greatest sensitivity in terms of exergy efficiency are the recuperators  
443 and condensers.

## 444 Acknowledgements

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446 financial support.

447 **Appendix A. State points**

448 Table of state points for the Kalina split-cycle with reheat, reproduced from [12]:

Table A.9: Thermodynamic state points (T, P, h, x, q) for the Kalina split-cycle with reheat. The terms sup stand for superheated, sub for subcooled, x for the ammonia mass fraction and q for the vapour fraction, in the case of a two-phase behaviour.

Point	$\dot{m}$ (kg/s)	T (°C)	P (bar)	h (kJ/kg)	x	q
1	3.6	194.8	101.7	1390	0.677	0.478
2	3.6	234.9	101.7	2134	0.677	1
3	3.6	330	101.7	2500	0.677	Sup
3'	3.6	281	36.6	2331	0.677	Sup
3''	3.6	330	36.6	2602	0.677	Sup
4	3.6	133.1	2.9	2185	0.677	Sup
5	3.6	73.4	2.9	1323	0.677	0.669
6	10.6	51.3	2.9	502	0.478	0.234
7	10.6	25	2.9	24	0.478	0
8	10.6	25.1	10.1	25	0.478	Sub
9	10.6	68.4	10.1	256	0.478	0.021
10	10.6	85.8	10.1	549	0.478	0.174
11	8.8	85.8	10.1	282	0.376	0
12	7.3	85.8	10.1	282	0.376	0
13	7	85.8	10.1	282	0.376	0
14	7	41.9	10.1	79	0.376	Sub
15	7	42	2.9	79	0.376	0
16	1.5	85.8	10.1	282	0.376	0
17	0.3	85.8	10.1	282	0.376	0
18	1.8	85.8	10.1	1817	0.965	1
19	1.4	85.8	10.1	1817	0.965	1
20	1.7	85.8	10.1	1559	0.867	0.832
21	1.7	60.3	10.1	1325	0.867	0.734
22	1.7	41.9	10.1	1087	0.867	0.597
23	1.7	30	10.1	337	0.867	0
24	1.7	33.1	101.7	357	0.867	Sub
25	1.7	80.8	101.7	591	0.867	Sub
26	0.4	85.8	10.1	1817	0.965	1
27	1.9	85.8	10.1	615	0.504	0.217
28	1.9	74.5	10.1	439	0.504	0.128
29	1.9	41.9	10.1	112	0.504	Sub
30	1.9	43.8	101.7	129	0.504	Sub
31	1.9	80.8	101.7	304	0.504	Sub
32	1.9	194.8	101.7	915	0.504	0
33	1.7	194.8	101.7	1909	0.867	1
34	35	346	2	815	-	-
35	35	287.3	2	749	-	-
36	35	217.7	2	672	-	-
37	35	127.7	2	574	-	-

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