

COMPARATIVE STUDY OF KALINA CYCLES FOR WASTE HEAT RECOVERY IN THE CEMENT INDUSTRY

Elviro Pereira Barbosa Júnior, juniorelviro@gmail.com

Felipe Raul Ponce Arrieta, felipe.ponce@pucminas.br

Pontifical Catholic University of Minas Gerais, Av. Dom José Gaspar, 500 - Belo Horizonte - MG Zip Code: 30535-901, Brazil

Cláudio Homero Ferreira da Silva, chomero@cemig.com.br

Cemig GT SA, Av. Barbacena, 1200 - 20º andar - Ala B2 - B. Santo Agostinho, 131 - Belo Horizonte – MG Zip Code: 30.190-130, Brazil

Abstract. *The comparative study of Kalina cycles for waste heat recovery in the cement industry aims to optimize the power generated and the generation cost. This study compared two Kalina cycles, KCS1-1 and KCS34-SA1, for waste heat recovery from the gases of cyclonic preheater in the cement industry. The cycles were simulated in four different clinker production capacity. The basis of the study of these cycles was the thermodynamic modelling, the exergy analysis and the optimization using genetic algorithms. Models developed in the Engineering Equation Solver (EES) contain the mass, energy, entropy and exergy balances, exergetic fuels and products, exergetic efficiencies and irreversibility by components of the cycle, and the estimation of the power generation cost. Parametric studies were conducted in order to observe the influence on the performance and power generation cost of some independent variables, such as the ammonia concentration and the pressure at the turbine inlet. The performed optimization by Genetic Algorithm aims to obtain the optimal condition of the schematic cycles under a specified objective function. The comparative study indicated that in Kalina Cycle KCS1-1 the lower power generation cost presented was 126.6 R\$/MWh for a power of 4,643 kW, while for Kalina Cycle KCS34-SA1 the power generation cost was 131.8 R\$/MWh for a power of 5,091 kW. For maximum power, the cycles KCS1-1 and KCS34-SA1 yielded results of 4,927 kW and 5,746 kW, respectively. Finally, the maximum energetic and exergetic efficiencies have obtained results very close to 23.7% and 52.8%. We have concluded that, given the great similarities in the thermodynamic performance of the cycles investigated, the criteria for defining the appropriate cycle have become merely economical.*

Keywords: *Waste heat recovery, Exergy analysis, Numerical simulation, Optimization, Genetic algorithm.*

1. INTRODUCTION

The Electric System of Brazil is going through a critical moment, where generation costs have been increasing due, among other reasons, to the largest continuous use of thermoelectric plants (EPE, 2015), while the works for construction of new hydropower plants suffer constant delays. In addition to that, there is the excessive government interference in the industry (Silva et al, 2016) that discourages investments that could enlarge the supply and reduce the cost of electric power generation. Given this, it is imperative for businesses to search for alternative sources of energy in order to reduce operating costs. In this context, the use of residual heat is an option. However, according to Zhang et al (2012), the generation of electricity from low-temperature sources such as geothermal and industrial waste heat sources has generally low efficiency. For better use of these sources, it is necessary to reduce the exergetic losses. For this purpose, several power cycles were proposed. Among them, there is the Kalina cycle developed by Russian researcher Dr. Alexander Kalina in 1982 with several possible configurations. Several recent studies, some of them listed in Tab. 1, show that the Kalina cycle presents performance, in most applications, superior to other alternatives such as the conventional Rankine cycle and the organic Rankine cycle.

Table 1. Literature review of the Kalina cycle

Main conclusions	Reference
The Kalina cycle offers superior performance to the Rankine cycle	Zhang et al (2015) Guzovic et al (2010) Mergner and Weimer (2015)
The Kalina cycle presents lower cost of generation than Rankine cycle	Coskun et al (2014) Rodríguez et al (2013) Shokati et al (2015)
Maximum energy efficiency of Kalina cycle is between 15.87 and 37.1% and mass fraction of ammonia is between 80% and 95%	Modi and Haglind (2015) Zare et al (2015) Chen et al (2015) Zhao et al (2015)

The parameters that exert greater influence on the performance of the Kalina cycle are, as Wang et al (2015) state, the concentration of ammonia in the turbine inlet and the temperature difference in the evaporator. According to Thorin (2000), for decades the binary non-azeotropic mixture ammonia-water is used as working fluid in processes such as absorption refrigeration, heat pumps and power cycles. What characterizes the non-azeotropic mixtures is the change in the composition and temperature during the boiling process for any concentration. The change in temperature during boiling and condensation processes is a feature that allows the working fluid to keep its temperature profile closer to the temperature profile of the heat source, leading to a more efficient heat exchange.

When it comes to cost estimation, Ulrich and Vasudevan (2004) proposed a number of curves that relate to the cost of the equipment with a particular parameter (power, heat exchange area, operating pressure, etc.) and the material which consists of iron, stainless steel, titanium, etc. Therefore, to use this method it is required the previous calculation of several parameters, including, for instance, the areas of heat exchange.

2. METHODOLOGY

To estimate the potential for electricity generation and the generation cost of electricity for a cement plant, the following steps were performed: 1st) Definition of the residual heat source; 2nd) Choice of power cycles; 3rd) Thermodynamic modeling of cycles; 4th) Development of the economic model; 5th) Optimization.

2.1 Description of the waste heat source

The major process steps in cement industry include mining and quarrying raw materials, crushing and grinding materials in preparation for the kiln, clinker production, and cement milling. The fuel most commonly used is coal, though some kilns use natural gas, oil, and various waste fuels. Raw meal enters at the top of the kiln (Fig 1) and gradually passes through increasingly hot zones toward the flame at the bottom of the kiln. Preheating is accomplished through countercurrent flow of raw materials and exhaust gases in cyclones. The most common systems are series fourstage preheaters, which have exhaust gases leaving at approximately 613K (USA, 2008). The preheaters exhaust gases are in the medium temperature range, where there are still opportunities for waste heat recovery. However, the exhaust gases can also be used for drying and preheating kiln feed in the raw grinding stages.

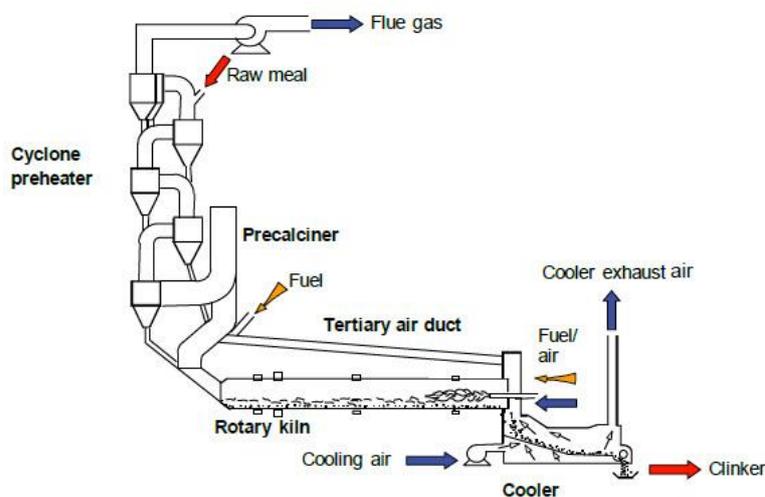


Figure 1. Diagram of standard cement kiln plant (IEAGHG, 2013)

The simulations carried out in this study were based on the clinker production daily capacity, as well as on the flow and temperature of the gas cyclone clinker preheater as Tab. 2 shows. Note that there is a correlation between a greater daily production of clinker and increased exhaust gas flow in the clinker preheater. Both grow at roughly the same proportion. The listed temperatures are within the range suggested by IFC (2014) from 573 to 663 K, and as previously stated, they can vary according to the number of stages in the clinker preheater. The chemical composition of gases by mass base was assumed to be 26.3% CO₂, 64.58% N₂, 4.94% O₂ and 4.18% H₂O.

Table 2. Features of exhaust gas from the cyclone preheater

Parameter	Mirolli (2006)	Apodi (2016)	Sui (2014)	Wang et al (2009)
Clinker production (ton/day)	3,000	3,500	4,500	5,000
Temperature [K]	658.15	583.15	588.00	613.15
Flow rate [kg/s]	76.5	88.00	113.2	126.7

2.2 Description of the heat recovery system

The Kalina cycle is a variation of the Rankine cycle, using a mixture of ammonia and water as the working fluid. The main difference between simple fluid cycles and cycles that use binary fluids is the temperature profile during the process of boiling and condensation. In simple fluid cycles, as the Organic Rankine, the temperature remains constant during boiling. Whereas in a binary mixture of water and ammonia, temperature increases during evaporation. This allows for better thermal exchange with the residual heat source and condenser cooling fluid due to smaller entropy generation. Tab. 3 presents the input parameters used for simulation and optimization of cycles. These parameters relate primarily to conditions of environment, whose assumed 295.15 K temperature is suited to the climatic conditions of Brazil. The following temperature differences are from the higher temperature to the lowest, so that the values are always positive. The quality in the separator output defines one output as saturated steam and the other output as saturated liquid. In the economizer, the fixed temperature difference ensures that the working fluid comes out as compressed liquid. Finally, the isentropic efficiencies of turbine and pump assumed typical values.

Table 3. Fixed values of parameters of the power cycles

Equipment	Variable	Value	Equipment	Variable	Value
Environment (dead state)	Pressure	101.32 [kPa]	Separator	Quality in the lower output	0 [%]
	Temperature	295.15 [K]		Quality in the upper output	100 [%]
Cooling water	Inlet and outlet pressure	250 [kPa]	LTR	ΔT	5 [K]
	Inlet temperature	295.15 [K]	Economizer	ΔT (pinch point)	5 [K]
	Outlet temperature	303.15 [K]	Turbine	Efficiency (η)	85 [%]
Condenser	Quality output (NH ₃ H ₂ O)	0 [%]	Pump	Efficiency (η)	85 [%]
	ΔT (NH ₃ H ₂ O - Water)	5 [K]			

2.2.1 Kalina Cycle KCS1-1

The following describes the Kalina cycle KCS1 typically proposed for use of residual heat in the cement industry, as shown by Mirolli (2006), Wang (2009) and Zhang at al (2012). Figure 2(a) shows the KCS1-1 Kalina cycle for the utilization of a heat source. In this cycle the working fluid is heated in the economizer (ECON) and vaporized in the evaporator (EVAP) and is then superheated in the superheater (SH). The superheated steam is expanded through the turbine (TURB) for generating work. The exhaust steam from the turbine passes through the high temperature heat recovery (HTR) and then mixes with the fluid of lower ammonia concentration. With the reduced ammonia concentration, the fluid is condensed in the condenser-1 (COND-1) and has its pressure increased by pump 2 (PUMP-2). After pumping, most of the fluid is heated by passing through the low temperature recuperator (LTR) and the HTR, then it enters the separator (SEP). In the SEP the liquid portion with a lower content of ammonia goes for LTR and is expanded in the valve (VALVE), mixing with the turbine exhaust steam. At the top of the SEP, the saturated vapor with the highest concentration of ammonia goes to the condenser-2 (COND-2) and after being condensed, its pressure increases in the pump-1 (PUMP-1), returning to the economizer restarting the cycle.

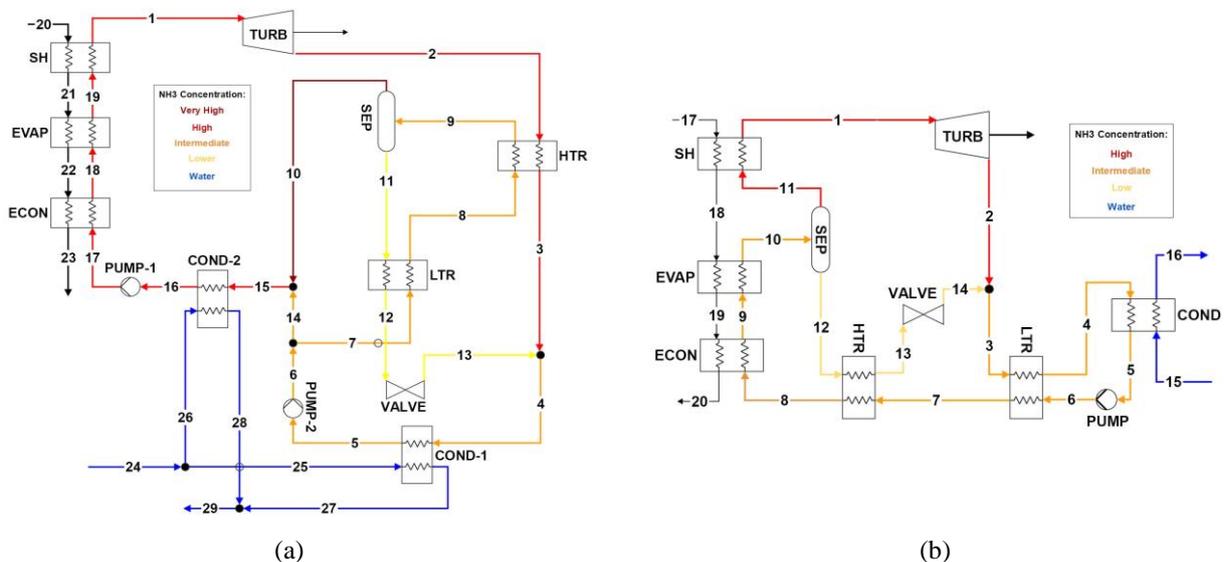


Figure 2. Diagram of Kalina Cycle KCS1-1(a) and Kalina Cycle KCS34-SA1(b)

2.2.2 Kalina Cycle KCS34-SA1

Next we describe the Kalina Cycle called KCS34-SA, also studied by Arrieta (2015), which is very similar to the KCS 34 developed for low-temperature heat sources (Zhang et al, 2012), but with the addition of a superheater. Figure 2(b) illustrates the Kalina Cycle called here KCS34-SA1. In this cycle the working fluid is heated in the economizer (ECON) and vaporized in the evaporator (EVAP). In the separator (SEP), the concentration of ammonia is increased, the fluid is then superheated in the superheater (SH). The superheated steam is expanded through the turbine (TURB) for generating work. The steam turbine exhaust is recovered and mixed with the liquid with low ammonia concentration depressurized in the VALVE. This lean mixture is condensed by rejecting heat in the cooling water through the condenser (COND). The liquid is pressurized by the pump and passes through the high and low temperature regenerators LTR and HTR respectively, before returning to ECON, and restarting the cycle.

2.3 Thermodynamic modeling

For the analysis of KCS1-1 and KCS34-SA1 Kalina cycles, calculations were carried out in accordance with the principles of first and second law of thermodynamics, through the balance of mass, energy, entropy and exergy. Through the software EES (Engineering Equation Solver), a computational tool for simulation of the Kalina cycle KCS1-1 and KCS34-SA1 was developed. The properties of ammonia-water binary mixture were obtained through an internal routine of the EES. Each equipment has been considered as a volume control, as shown in Fig. 1, determining the thermodynamic properties in each state, as well as the transfer of heat and work in each process. The main parameters of KCS1-1 Kalina cycle were calculated as follows:

$$\dot{W}_t = \dot{m}_1(h_1 - h_2) \quad (1)$$

$$\dot{W}_{p1} = \dot{m}_{16}(h_{17} - h_{16}) \quad (2)$$

$$\dot{W}_{p2} = \dot{m}_5(h_6 - h_5) \quad (3)$$

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_{p1} - \dot{W}_{p2} \quad (4)$$

$$\eta_e = \frac{\dot{W}_{CK}}{\dot{m}_{17}(h_1 - h_{17})} \quad (5)$$

$$\eta_{ex} = \frac{\dot{W}_{CK}}{\dot{m}_{20}(ex_{23} - ex_{20})} \quad (6)$$

The main parameters of KC34-SA1 Kalina cycle were calculated as follows:

$$\dot{W}_t = \dot{m}_1(h_1 - h_2) \quad (7)$$

$$\dot{W}_b = \dot{m}_5(h_6 - h_5) \quad (8)$$

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \quad (9)$$

$$\eta_e = \frac{\dot{W}_{CK}}{\dot{m}_{10}(h_{10} - h_8) + \dot{m}_{11}(h_1 - h_{11})} \quad (10)$$

$$\eta_{ex} = \frac{\dot{W}_{CK}}{\dot{m}_{17}(ex_{17} - ex_{20})} \quad (11)$$

From Eq. (1) to Eq. (11) \dot{m}_i is the mass flow rate, kg/s; h_i is the specific enthalpy in kJ/kg; ex_i is the specific exergy, in kJ/kg; \dot{W}_t , is the power generated in the turbine; in kW, \dot{W}_p is the power consumed by the pump; in kW, \dot{W}_{net} is the net power cycle Kalina, in kW; η_e is the net thermal efficiency of the cycle and η_{ex} is exergetic efficiency of the cycle. In Eq. (1), (2), (3), (7) and (8) the isentropic income was presented in Tab. 3. In Eq. (6) and (11) the specific exergy to the states '23', '20', '17' and '20' was calculated by Eq. (12) using the subroutines for calculating properties of gas mixtures made by Santos (2013).

$$ex_i = (h_i - h_0) - T_0(s_i - s_0) \quad (12)$$

In Eq. (12), T_0 is the reference environment temperature in dead state (Tab. 3); s_i is the specific entropy in kJ/kg.K; h_0 and s_0 are the specific enthalpy and specific entropy, respectively, calculated under the conditions of temperature and pressure of the reference environment in the dead state.

2.4 Cost estimate method

The method for cost estimation adopted in this work was proposed by Ulrich and Vasudevan (2004). In this method it is necessary to first define the value of the main parameters of each equipment, namely: the area of heat exchangers, the turbine power, the dimensions of the separator, and the power of the pump(s). The purchase price of the equipment is connected to these parameters through graphs provided by the authors. These graphics also allowed correcting the value of the equipment depending on the material used and working pressure. The material adopted for all the equipment was the stainless steel, as recommended by Whittaker (2009). For the calculation of the heat transfer surface area, we considered a shell-tube configuration together with the method of mean logarithmic difference temperature (MLDT), as suggested by Incropera (2007):

$$\dot{Q}_j = U_j A_j F_j \Delta T_j^{LMTD} \quad (12)$$

$$\Delta T_j^{LMTD} = \left\{ \frac{(T_{h,2} - T_{c,1}) - (T_{h,1} - T_{c,2})}{\ln \left[\frac{T_{h,2} - T_{c,1}}{T_{h,1} - T_{c,2}} \right]} \right\} \quad (13)$$

In Eq. (12) the subscript 'j' refers to the various heat exchangers that make up the cycle: SH, EVAP, ECON, HTR, LTR, COND; Q is the rate of heat transferred, in kJ/s, calculated by the energy balance; U is the overall coefficient of heat transfer, in W/m².K. The U values adopted were: for SH, 40(P₁[bar])^{1/2}, for EVAP and ECON, 105, for HTR and COND, 1,300, for LTR, 950, according to Ulrich and Vasudevan (2004), and refer to the heat exchangers of the shell-tube type. The variable F is the dimensionless correction factor depending on the configuration of the exchanger and the inlet and outlet temperatures, which is calculated using a built-in function of the EES to the heat exchangers shell-tube type. In turn, ΔT^{LMTD} is the mean logarithmic difference temperature (Eq. 13), in K; T_{h,2} is the hot side temperature at the outlet from a given heat exchanger, in K; T_{h,1} is the temperature of the hot side at the inlet of a given heat exchanger, in K; T_{c,2} is the temperature of the cold side at the outlet of a given heat exchanger, in K; T_{c,1} is the temperature of the cold side at the inlet of a given heat exchanger, in K.

The specific cost of electricity generated c_g , Eq. (14) in R\$/kWh. It was calculated considering only the portions of costs relating to the installation, operation and maintenance, since it considered that the waste gas has zero cost.

$$c_g = c_{inv} \frac{AF}{OH} + c_{O\&M} \quad (14)$$

$$AF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (15)$$

$$c_{inv} = \frac{C_{inst}}{\dot{W}_{net}} \quad (16)$$

$$C_{inst} = \sum_j C_{eq(2004)} \frac{IF_{(2004)}}{IF_{(2015)}} (C_C + C_F)(C_{SD} + C_{AB} + C_{OS}) \quad (17)$$

In Eq. (17) the subscript 'j' refers to the equipment of the cycle; IF₍₂₀₀₄₎ and IF₍₂₀₁₅₎ are dimensionless index factors for the years 2004 and 2015 amounting to 124 and 100, respectively, according to BOE (2016); $c_{eq(2004)}$ is the cost of each of the main equipment obtained by interpolation of cost curves developed by Ulrich and Vasudevan (2004) relating to the year 2004, c_{inst} is total cost of installation referring to December 2015; C_C and C_F refer to contingency expense and contractor's fee, respectively, and the sum is equal to 1.18; C_{SD} refers to site development, C_{AB} refers to auxiliary buildings, C_{OS} refers to off-site facilities and the sum is equal to 1.30. In Eq. (16) c_{inv} is the specific cost of the total investment, in R\$/kW. In Eq. (14) $c_{O\&M}$ is the specific cost of operation and maintenance, adopted as R\$/kWh 0.072, according to ICF (2015). In Eq. (15) n is the number of years of useful life, adopted as 20 years; i is the interest rate in the long term, considered at 7% p.p.y.; AF is the factor of depreciation year⁻¹; OH is the number of hours of operation per year, 8,030 h/year. All costs have been converted from USD to local currency at a rate of 3.60 [R\$/\\$].

2.5 Optimization

After the thermodynamic and cycle cost modeling, an optimization tool by Genetic Algorithm (GA) available on the EES was used. The selected variables in each model to be optimized through the GA as well as the variation range

defined for each variable are listed in Tab. 4. In this way, the variation range is defined by the maximum and minimum values among which there are valid results. The optimization had two objectives: to minimize the cost of generation (c_g) and maximize the net power cycle (\dot{W}_{net}).

Table 4. Restrictions on the variables used in the Genetic Algorithm

Power Cycle	Variable	Smallest value	Greater value	Variable	Smallest value	Greater value
KCS1-1	ΔT_{22-18} [K]	11	150	P_1 [kPa]	5,000	10,000
	ΔT_{2-9} [K]	10	35	P_2 [kPa]	102	600
	ΔT_{SA} [K]	10	60	q_{19} [%]	40	95
	x_1 [%]	40	90			
KCS34-SA1	ΔT_{12-13} [K]	10	65	q_{10} [%]	60	95
	ΔT_{19-9} [K]	11	135	P_1 [kPa]	5,000	7,750
	ΔT_{SA} [K]	11	50	x_{10} [%]	50	95

3. RESULTS AND DISCUSSIONS

After the development of a computational tool containing the thermodynamic and economic model for the simulation of the Kalina cycle KCS1-1 and KCS34-SA1, simulations were done in order to check the impact of certain variables in the cost of generation and performance of the cycle. The results of these simulations were based on the flow data of rates, composition and temperature of the exhaust from gas of clinker preheater provided by Apodi (2016) and are presented in Fig. 3 and Fig. 4. For the comparative study between cycles, optimizations were performed through GA, using the characteristics of the exhaust gas of clinker preheater listed in Tab. 2 with the purpose of minimizing the cost of generation (c_g) and maximizing the net power cycle (\dot{W}_{net}).

3.1 Influence of the main parameters

The influence of variation in temperature difference ΔT_{22-18} on the cost of generation (c_g) and liquid power cycle (\dot{W}_{net}) KCS1-1 is shown in Fig. 3 (a). Note that as ΔT_{22-18} increases, \dot{W}_{net} decreases at a constant rate, this reduction is due to the fact that a greater temperature difference between the States '22' and '18' leads to a lower heat transfer to the working fluid and, therefore, a lower power generation. The value of c_g , however, has its minimum with ΔT_{22-18} around 60 K. That occurs due to the cost of the evaporator suffers a significant increase for a temperature differences in the entry under ~ 60 K. To differences greater than ~ 60 K, the reduction in the cost of the evaporator is less significant than the reduction in power, leading to an increase in the c_g . Figure 3 (b) shows that as P_1 KCS1-1 cycle increases, \dot{W}_{net} also increases. This occurs because of the largest difference in enthalpy between the working fluid at the entrance and exit of the turbine (Eq. 1). This increase in P_1 reduces the c_g value in spite of increasing costs in most equipment especially in ECON. It is observed that with the P_1 value above 6.000 kPa, the reduction in c_g is negligible. The increase in the quality q_{19} has no influence on \dot{W}_{net} , however, it leads to a significant increase in c_g as shown in Fig. 3 (c). This is due to the increase in heat exchange area in the EVAP, needed for the biggest steam generation, be much more relevant than the reduction in the area of heat exchange of SH. Finally, Fig. 3 (d) shows that the increase in x_1 leads to an increase in \dot{W}_{net} and a reduction in c_g . This is due to the greater steam generation provided by the greatest concentration of ammonia (which evaporates easier). The reduction in c_g , however, becomes unimportant to concentrations of ammonia in x_1 above $\sim 56\%$, this is due to the very significant increase in the flow rate in the state '11' and the consequent increase in the cost of LTR.

The influence of variation in temperature difference ΔT_{19-9} on the cost of generation (c_g) and net power of the cycle (\dot{W}_{net}) KCS34-SA1 is presented in Fig. 4 (a). Note that as ΔT_{19-9} increases, \dot{W}_{net} decreases significantly. This reduction is due to the fact that a greater temperature difference between the states '19' and '9' leads to a lower heat transfer to the working fluid and, therefore, a lower power generation. The c_g value, however, reduces significantly with the decrease of ΔT_{22-18} up to ~ 100 K. For smaller differences, the c_g value remains stable. This is due to the cost of the evaporator suffers a significant increase for temperature differences in its entry under ~ 100 K, meaning that even with a significant increase in the cost of power generation keeps virtually unchanged.

Figure 4 (b) shows that as P_1 increases in the KCS34-SA1 cycle, \dot{W}_{net} also increases. This occurs due to the largest difference in enthalpy between the working fluid at the entrance and exit of the turbine (Eq. 1). It is observed that with the P_1 value above 7,000 kPa the c_g reduction is negligible. Contrary to what occurs in the KCS1-1 cycle, in KCS34-SA1 cycle an increases in q_{10} causes an increase in \dot{W}_{net} and a reduction in the amount of c_g as in Fig. 4 (c). This happens due to greater steam generation which rises the power generation. Note that for values of q_{10} above 85%, the reduction in the amount of c_g is despicable. About x_{10} , Fig. 4 (d) shows that a greater concentration of ammonia at the entrance of the KCS34-SA1 cycle leads to an increase of \dot{W}_{net} in the same way as occurs in KCS1-1. But in relation to c_g the optimal value of x_{10} is close to 90%, far above the optimal value of the previous cycle.

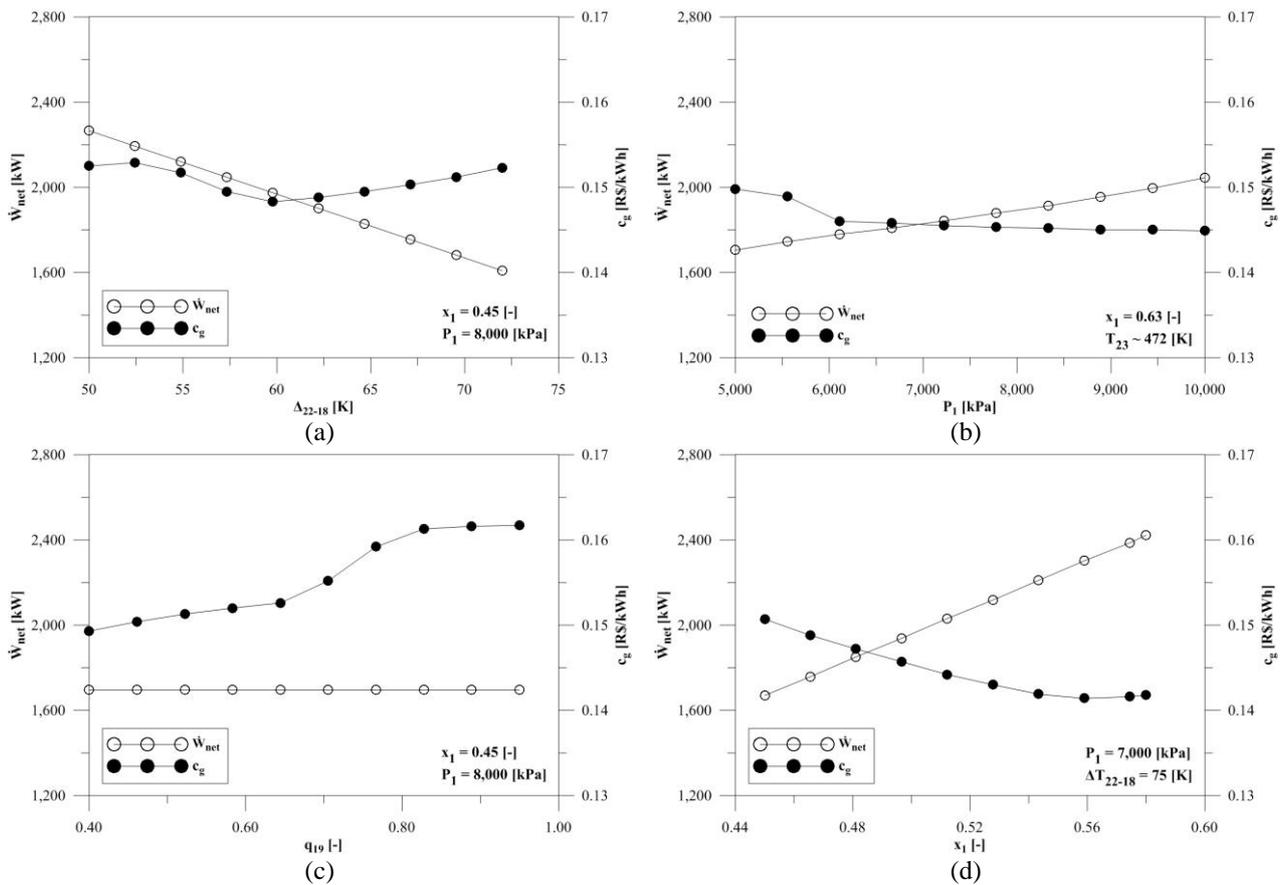


Figure 3. Impact of some variables on the cost of generation and net cycle power in cycle KCS1-1

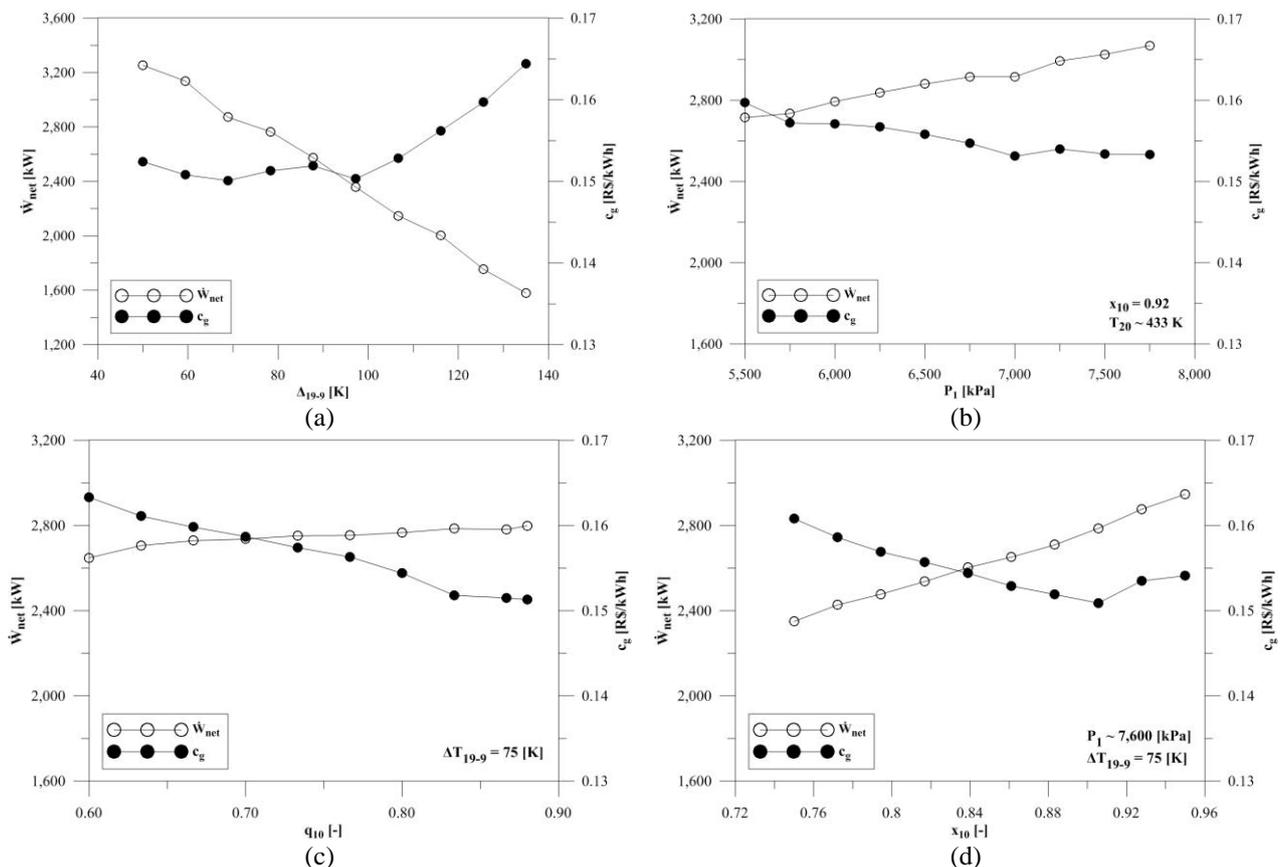


Figure 4. Impact of some variables on the cost of generation and net cycle power in cycle KCS34-SA1

3.2 Comparative study

In this section the performance and costs of KCS1-1 cycles and KCS34-SA1 simulated based on the sources in Tab. 2 are compared, both submitted to optimization of cost of generation (c_g) and net power of cycle (\dot{W}_{net}) through genetic algorithm. In Tab. 5, the results of the minimization of the c_g value are presented. The KCS1-1 cycle presents the smallest value of c_g and c_{inv} . For c_g the difference between the minimum and the maximum values are around 3%. On the other hands, for c_{inv} the difference between the minimum and the maximum values are around 6%. In terms of generated power and investment cost, the KCS34-SA1 cycle presented on average 22% more power, and also a 27% higher average c_{inst} .

Table 5. Results for c_g minimization by GA

Power Cycle	Reference	c_g [R\$/kWh]	c_{inv} [R\$/kW]	c_{inst} [R\$]	\dot{W}_{net} [kW]	η_e [%]	η_{ex} [%]
KCS1-1	Apodi (2016)	0.1498	6,617	11,218,441	1,695	22.3	49.5
KCS34-SA1	Apodi (2016)	0.1503	6,663	16,095,712	2,416	22.3	51.0
KCS1-1	Mirolli (2006)	0.1266	4,643	15,364,284	3,309	21.6	46.1
KCS34-SA1	Mirolli (2006)	0.1318	5,091	17,210,248	3,381	23.7	49.3
KCS1-1	Sui (2014)	0.1406	5,832	13,818,485	2,369	22.3	49.5
KCS34-SA1	Sui (2014)	0.1432	6,054	25,351,442	4,188	22.7	54.1
KCS1-1	Wang at al (2009)	0.1289	4,841	18,250,272	3,770	22.7	49.4
KCS34-SA1	Wang at al (2009)	0.1363	5,466	23,947,525	4,381	22.0	48.8

The costs of the equipment are shown in Fig. 5, drawn from the c_g minimization to the source Apodi (2016). It demonstrates that despite the KCS1-1 cycle has a greater number of equipment, this one has generally lower values compared to KCS34-SA1 when simulated for the same heat source. The biggest cost difference we find in the separator, which is almost eight times lower in the cycle KCS1-1. This is because the lowest working pressure of the separator in KCS1-1 cycle is around 280 kPa, while in KCS34-SA1 cycle is around 7,500 kPa.

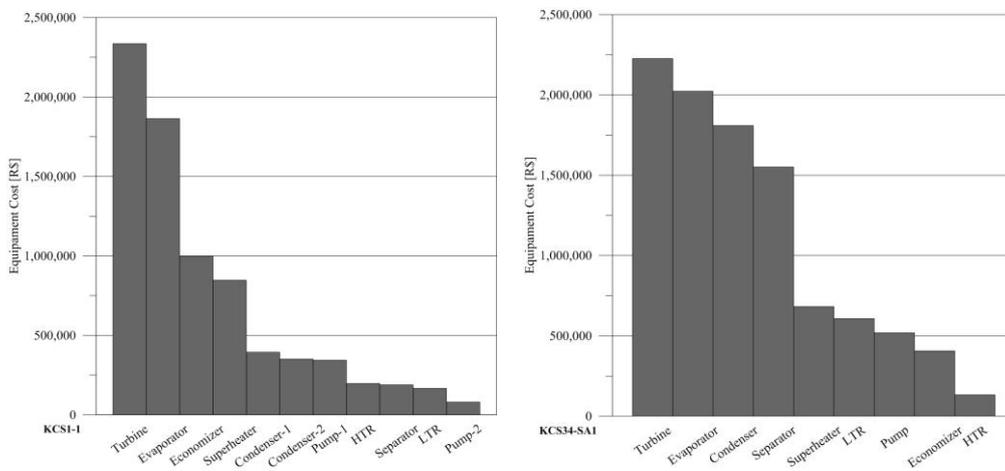


Figure 5. Cost of equipment in cycles KCS1-1 and KCS34-SA1

In Tab. 6 the results of the maximization of the value \dot{W}_{net} are presented. The cycle KCS34-SA1 has the highest value of \dot{W}_{net} in all the simulations, but only these values are, on average, only 14% higher. In addition, the KCS34-SA1 cycle presented lower c_g and c_{inv} values in most simulations, except for Sui (2014), and values of η_e and η_{ex} being 9% higher, on average.

Table 6. Results for \dot{W}_{net} maximization by GA

Power Cycle	Reference	c_g [R\$/kWh]	c_{inv} [R\$/kW]	c_{inst} [R\$]	\dot{W}_{net} [kW]	η_e [%]	η_{ex} [%]
KCS34-SA1	Apodi (2016)	0.1551	7,065	22,252,695	3,150	21.5	52.2
KCS1-1	Apodi (2016)	0.1710	8,425	23,384,891	2,776	20.2	49.1
KCS34-SA1	Mirolli (2006)	0.1369	5,521	24,753,232	4,484	23.3	52.0
KCS1-1	Mirolli (2006)	0.1375	5,572	20,759,774	3,726	21.1	46.5
KCS34-SA1	Sui (2014)	0.1494	6,586	27,877,669	4,233	21.5	52.0
KCS1-1	Sui (2014)	0.1380	5,613	22,617,631	4,029	21.0	51.3
KCS34-SA1	Wang et al (2009)	0.1395	5,741	32,984,521	5,746	22.5	52.8
KCS1-1	Wang et al (2009)	0.1433	6,067	29,892,659	4,927	19.2	45.8

Through Fig. 6 it is possible to have an overview of the performance and costs of cycles for each simulated source. The graphs show that the cycles present greater discrepancy in results for Mirolli (2016) and for Wang et al (2009) sources, whose gas temperatures of clinker preheater are higher (Tab. 2), while for Apodi (2016) and Sui (2014) sources the results were close. Note that, for higher temperatures, the KCS1-1 cycle reaches smaller c_g values, while the KCS34-SA1 cycle reaches superior values of net generated power.

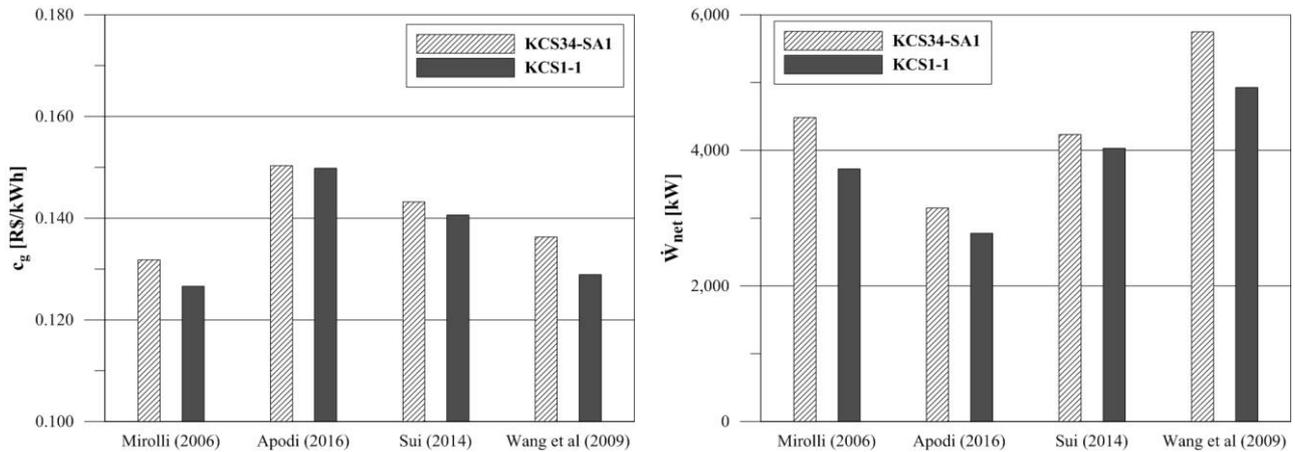


Figure 6. Results for c_g minimization and W_{net} maximization by GA

4. CONCLUSIONS

This comparative study between cycles Kalina KCS1-1 and KCS34-SA1 to the utilization of waste heat in cement industry allowed reaching the following conclusions:

- In the KCS1-1 cycle the value of ΔT_{22-18} to get the lower c_g is approximately 60 K;
- The increase of the quality of ammonia-water mixture in the output of the evaporator increases the value of the c_g in KCS1-1 cycle and reduces in the KCS34-SA1 cycle;
- The optimal value of the concentration of ammonia in the inlet of the separator that gets the smallest value of c_g to KCS1-1 cycle is around 38%, while in the KCS34-SA1 cycle it is around 90%.
- Minimizing c_g , the KCS1-1 cycle presents the smallest value of c_g and c_{inv} in every simulation;
- Minimizing c_g , the KCS34-SA1 cycle presented on average 22% higher power;
- The biggest difference between the cost of the equipment are in the separator, which is almost eight times lower in the cycle KCS1-1;
- Maximizing \dot{W}_{net} , the KCS34-SA1 cycle presents a higher \dot{W}_{net} value than in all the simulations;
- For higher temperatures the gas clinker preheater KCS1-1 cycle reaches smaller c_g values, while the KCS34-SA1 cycle reaches superior \dot{W}_{net} values

5. ACKNOWLEDGEMENTS

The authors thank FAPEMIG, FAPEMIG/CEMIG APQ-03422-12 research project, ANNEL Research and Developed program (P&D) research project GT0554 and Pontifical Catholic University of Minas Gerais – PUC Minas for the financial support to this work.

6. REFERENCES

- APODI (Ceará). WHRS - UTE CIMENTO APODI WASTE HEAT RECOVERY SYSTEM: Sistema de Recuperação De Calor. Quixeré, 2015. 15 slides: color.
- ARRIETA, Felipe Raul Ponce et al. Avaliação termoeconômica de um Ciclo Kalina para a recuperação de calor residual. In: Iberian Latin American Congress On Computational Methods In Engineering, 36, 2015, Rio de Janeiro. Proceedings... Rio de Janeiro: ABMEC, 2015.
- BOE - California State Board Of Equalization (EUA). State Of California. Equipment and Fixtures Index, Percent Good and Valuation Factors. [S. L.], 2016.
- CHEN, Yaping et al. Energy and exergy analysis of integrated system of ammonia–water Kalina–Rankine cycle. Energy, [S.I.], v. 90, p. 2028-2037, out. 2015.
- COSKUN, Ahmet; BOLATTURK, Ali; KANOGLU, Mehmet. Thermodynamic and economic analysis and optimization of power cycles for a medium temperature geothermal resource. Energy Conversion and Management, [S.I.], v. 78, p. 39-49, fev. 2014.

- EPE - EMPRESA DE PESQUISA ENERGÉTICA (BRASIL). Ministério de Minas e Energia. Anuário Estatístico de Energia Elétrica 2015: Ano base 2014. Rio de Janeiro: UP IDEIAS Comunicação e Eventos Eireli, 2015.
- GUZOVIC, Z.; LONCAR, D.; FERDELJI, N. Possibilities of electricity generation in the Republic of Croatia by means of geothermal energy. *Energy*, [S.I.], v. 35, p. 3429-3440, ago. 2010.
- IEAGHG, Deployment of CCS in the Cement Industry, 2013/19, December, 2013.
- IFC - INTERNATIONAL FINANCE CORPORATION World Bank Group. Waste Heat Recovery for the Cement Sector: Market and Supplier Analysis. s.n, 2014.
- INCROPERA, Frank P. et al. Fundamentos de transferência de calor e de massa. 6. ed. Rio de Janeiro: LTC - Livros Técnicos e Científicos, c2008. xix, 643 p. ISBN 9788521615842
- MERGNER, Hanna; WEIMER, Thomas. Performance of ammonia–water based cycles for power generation from low enthalpy heat sources. *Energy*, [S.I.], v. 88, p. 93-100, ago. 2015.
- MIROLI, M. D. Cementing Kalina cycle effectiveness. *IEEE Industry Applications Magazine*, [S.I.], v. 12, n. 4, p. 60-64, ago. 2006.
- MODI, Anish; HAGLIND, Fredrik. Thermodynamic optimization and analysis of four Kalina cycle layouts for high temperature applications. *Applied Thermal Engineering*, [S.I.], v. 76, p. 196-205, fev. 2015.
- RODRÍGUEZ, Carlos Eymel Campos et al. Exergetic and economic comparison of ORC and Kalina cycle for low temperature enhanced geothermal system in Brazil. *Applied Thermal Engineering*, [S.I.], v. 52, p. 109-119, abr. 2013.
- SANTOS, Larissa Assunção Oliveira. Modelagem de ciclos Kalina para a cogeração com calor residual no setor cimenteiro. 2014 85 f. Dissertação (Mestrado) - Pontifícia Universidade Católica de Minas Gerais, Programa de Pós-Graduação em Engenharia Mecânica
- SHOKATI, Naser; RANJBAR, Faramarz; YARI, Mortaza. Exergoeconomic analysis and optimization of basic, dual-pressure and dual-fluid ORCs and Kalina geothermal power plants: A comparative study. *Renewable Energy*, [S.I.], v. 83, p. 527-542, nov. 2015.
- SILVA, Rodrigo Corrêa da; DE MARCHI NETO, Ismael; SEIFERT, Stephan Silva. Electricity supply security and the future role of renewable energy sources in Brazil. *Renewable and Sustainable Energy Reviews*, [S.I.], v. 59, p. 328-341, jun. 2016
- SUI, Xiuwen et al. Exergetic life cycle assessment of cement production process with waste heat power generation. *Energy Conversion And Management*, [S.I.], v. 88, p.684-692, dez. 2014.
- THORIN, Eva. Power cycles with ammonia-water mixtures as working fluid: Analysis of Different Applications and the Influence of Thermophysical Properties. 2000. 54 p. Tese (Doutorado em Engenharia Química) - Royal Institute of Technology. Estocolmo, 2000
- ULRICH, Gael D; VASUDEVAN, Palligarnai T. Chemical engineering process design and economics: a practical guide. 2nd ed. Durham, N.H.: Process Publishing, c2004. xiv, 706 p. ISBN 0970876823.
- USA. U.S. Department of Energy. United States Government. Waste Heat Recovery: Technology and Opportunities in U.S. Industry. S. L: Bcs, 2008. 112 p.
- WANG, Jiangfeng; DAI, Yiping; GAO, Lin. Exergy analyses and parametric optimizations for different cogeneration power plants in cement industry. *Applied Energy*, [S.I.], v. 86, n. 6, p.941-948, jun. 2009.
- WANG, Jianyong et al. Thermodynamic analysis and optimization of a flash-binary geothermal power generation system. *Geothermics*, [S.I.], v. 55, p. 69-77, maio 2015.
- WHITTAKER, Peter. Corrosion in the Kalina cycle: An investigation into corrosion problems at the Kalina cycle geothermal power plant in Húsavík, Iceland. 2009. 40 f. Dissertação (Mestrado) - Curso de Engenharia, School For Renewable Energy Science, University Of Iceland, Akureyri, 2009.
- ZARE, V.; MAHMOUDI, S.M.S.; YARI, M.. On the exergoeconomic assessment of employing Kalina cycle for GT-MHR waste heat utilization. *Energy Conversion and Management*, [S.I.], v. 90, p. 364-374, jan. 2015.
- ZHANG, Xinxin; HE, Maogang; ZHANG, Ying. A review of research on the Kalina cycle. *Renewable and Sustainable Energy Reviews*, [S.I.], v. 16, p. 5309-5318, set. 2012.
- ZHANG, Zhi et al. Power generation and heating performances of integrated system of ammonia–water Kalina–Rankine cycle. *Energy Conversion and Management*, [S.I.], v. 92, p. 517-522, mar. 2015
- ZHAO, Pan; WANG, Jiangfeng; DAI, Yiping. Thermodynamic analysis of an integrated energy system based on compressed air energy storage (CAES) system and Kalina cycle. *Energy Conversion and Management*, [S.I.], v. 98, p. 161-172, jul. 2015.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.