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KCS34 evaluation for WHR in cement industry

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***Abstract*—*The simple Kalina cycle system 34 (KCS34)-has been studied to perform energy cogeneration from the waste heat recovery (WHR) in preheater cement industries. The preheater available energy was considered from a 5000 tc/day cement production capacity. Thermodynamic and simplified exergoeconomic models were developed in the Engineering Equation Solver (EES) software. Several cycle thermodynamic parameters as ammonia-water mixture concentration and turbine operating pressure were wide-ranging in order to maximize the cycle thermal efficiency aiming to minimize the electricity generation cost. The temperature-entropy KCS34schematics were shown for different best results aiming to understand which set of parameters targets the maximum KCS34performance. The produced power, the thermal cycle efficiency, the exergetic efficiency and the exergoeconomic electricity specific cost were plotted for the different ranges of the independent parameters. The optimum results for a range specific investment price were presented. The main conclusions indicate that in the range of the studied parameters the turbine operating pressure caused a generated power variation greater than the ammonia-water mixture concentration in the KCS34performance. It was also possible to conclude that the KCS34is competitive with the existing electricity prices. In this case the KC proved to be applicable for WHR in the cement industry.*

Keywords*—*Kalina Cycle, Waste heat recovery, cement industry, CHP, cost.

I. INTRODUCTION

Currently, the industrial sector is a pioneer in energy consumption. In Brazil there is a total of 88 cement plants according to the Cement Industry Union [1], which fueled an apparent cement consumption of 353 kg/person/year in 2013, showing an increase of 1.4% over the previous year. Brazil is one of the world's largest cement producers placing fifth in the rank [1]. This way, the cement industry is in a large scale, a great investment option for energy recovery. Consequently, new technologies applied to this sector is a point to be thought out, discussed and worked mainly aim for improvements in industrial processes, the possibility of using thermodynamic cycles, especially the Kalina cycle in the utilization of waste heat during the cement production process for the cogeneration of electric energy [2]. The KCS34has been studied to perform energy cogeneration from the waste heat recovery in cyclonic preheater of the cement industries with dry production process. The preheater available energy was considered from a daily capacity of 5000 ton of clink.

II. LITERATURE REVIEW

Since the 1980s, when it was patented, the Kalina cycle has been an efficiency promising. The expected conversion efficiency is as high as 45%[3]. An optimization procedure for heat recovery boilers in combined power noticed higher exergetic efficiency with fluid ammonia-water compared with the use of pure fluids [4]. A Kalina cycle for electricity generation from the exhaust gases of a gas turbine combined cycle have attested that the Kalina cycle was found to be 10-20 % more efficient than the Rankine [5]. Other paper indicates that Kalina cycle has 3% higher performance than Rankine cycle in Husavick plant [6].In order to compute and locate the irreversibilities in a Kalina cycle, the exergetic efficiency values reached 55% for an input of the turbine steam temperature of 525°C and an ammonia fraction of 75% in the working fluid [7]. The exergy analysis of a cogeneration plant formed by a Kalina cycle and a four-stage desalination plant was performed with the purpose of the cogeneration plant was to generate electricity with the simultaneous production of fresh water from geothermal energy [8]. A detailed description of the Kalina cycle terminology for waste heat recovery in the cement industry has been presented by several authors. The advantages of using ammonia-water mixture as working fluid and points out the major design challenges for the application of waste heat recovery in the cement industry, which focus on the design of heat exchangers that recovers energy from gases and the design of the components of distillation and condensation system [9, 10]. Different Rankine cycle configurations were comparedfor waste heat recovery in the cement

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industry [11]. The authors also mention that if you want to improve or increase the generation rate per ton of clink should choice to use the Kalina cycle because with it has a higher waste heat recovery in the cement industry. A Kalina cycle was optimized for waste heat recovery application [12]. A computer code was developed in Matlab to simulate the cycle. In the ammonia concentration range, between 0.8 and 1.0, at the turbine inlet, it was observed that a higher temperature in the separator and turbine inlet, which leads to increased efficiency, and that the maximum efficiency was obtained with a concentration of 0.9. An overview about the state of the art power generation technology from thermal sources temperature range with non-aqueous fluids shows the advantage of the Kalina cycle due to the evaporation process in the boiler at a variable temperature [13].

III. METHODOLOGY

The study of waste heat recovery in the cement industry has been developed in the steps mentioned below:

- Thermal schematic definition for waste heat recovery. The focus of this work is the cement plants that have satellite clinker coolers. In these plants the only available source of heat is the exhaust gas from the cyclonic preheaters;
- Thermal cycle modeling. The modeling included the mass, energy, entropy and exergy balances. The estimated calculation of the cost of electricity generated was included into the model;
- Heat recovery optimization. Different process variables taking place in the thermal cycle were found to maximize the generation of electricity and/or minimize the cost of the generated power.

3.1 Thermal schematic definition

Considering the thermal source in focus, was selected a KCS34 for the waste heat recovery. The schematic of the cycle is shown in Fig 1. Table 1 shows the data of the preheater available energy that was considered from a daily capacity of 5000 ton of clink. These data are at state ‘17’ in Fig. 1.

The Kalina cycle can be considered simply to have a small number of components. In the state ‘1’ it is specified the steam at the turbine inlet conditions ‘1’, specifically ammonia concentration, the pressure and temperature of the working fluid. The fluid has a rich concentration of ammonia at the turbine inlet in state ‘1’. After expansion, in the state ‘2’, the fluid is mixed with a poor solution coming from the ammonia separator - EPS (8). For mixing was employed a valve (9) to equalize the pressure of the mixture at the turbine exhaust. The leaner mixture to ammonia in state ‘3’ can be condensed at a lower temperature and pressure, thus allowing greater power generation in the cycle. Before entering the condenser (11), the heat exchanges operating fluid in the low temperature regenerator - LTR (6). In the pump (7) the fluid is pressurized to the operating pressure at the turbine inlet. In LTR and high temperature regenerator - HTR (5) the working fluid is preheated before it enters the waste heat recovery boiler. The boiler is made up of the economizer - ECON (4), the evaporator - EVAP (3) and the super heater - SA (5). After passing the working fluid through the EVAP and ECON, the fluid enters SEP where it reaches the desired concentration of ammonia in the saturated vapor state ‘11’. The operating temperature at the turbine inlet is achieved by passing the fluid through SA.

Table 1: Available energy for WHR [14]

Composition	% Mol
CO_2	28.0
N_2	69.0
O_2	3.0
Parameter	Value
Temperature (K)	623.15
Pressure (kPa)	101.32
Volumetric flow (m^3/h)	269,526

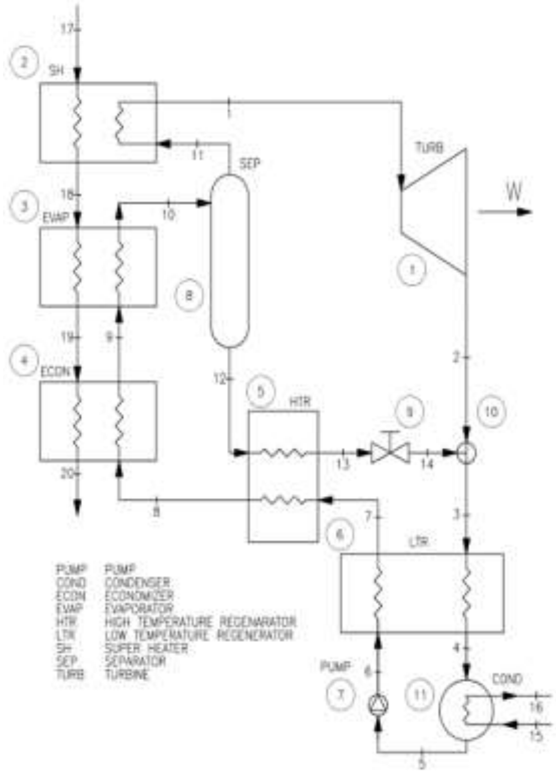

Fig. 1: The KCS34

3.2 Thermal cycle modelling

The modeling of the thermal cycle was performed in the Engineering Equation Solver (EES) software. The modeling included the mass, exergy, entropy and exergy balances as well as the calculation of exergetic Fuels and exergetic Products and exergetic efficiencies for each

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component and thermal cycle as a whole. In equations 1 to 6 we present the model for the Kalina cycle as a whole. In these equations, the different states correspond to those shown in the thermal cycle of Fig. 1. The different quantities and terms in these equations are: $\dot{Q}_{in}$, heat added to the Kalina cycle, in kW, $\dot{Q}_{out}$ heat rejected from the Kalina cycle, in kW, $\dot{W}$, Net generated power in the Kalina cycle, in kW, $\dot{m}$, mass flow, in kg/s; h, specific enthalpy, in kJ/kg; ex, specific exergy (considering physical plus chemical, as explained later), in kJ/kg $\dot{F}$, exergetic fuel, in kW; $\dot{P}$, exergetic product, in kW; η_{ex}, exergetic efficiency and η, thermal efficiency.

$$\dot{Q}_{in} = \dot{m}_8 \cdot (h_{10} - h_8) + \dot{m}_{11} \cdot (h_1 - h_{11}) \quad (1)$$

$$\dot{Q}_{out} = \dot{m}_4 \cdot (h_4 - h_3) \quad (2)$$

$$\dot{W} = \dot{m}_1 \cdot (h_1 - h_2) - \dot{m}_5 \cdot (h_6 - h_5) \quad (3)$$

$$\eta = \frac{\dot{W}}{\dot{Q}_{ent}} \quad (4)$$

$$\eta_{ex} = \frac{\dot{P}}{\dot{F}} = \frac{\dot{W}}{\dot{m}_{17} \cdot (ex_{17} - ex_{20})} \quad (5)$$

$$k^* = \frac{1}{\eta_{ex}} \quad (6)$$

For the purposes of the calculation, we used the calculation subroutines gas mixtures properties presented by [15]. With these subroutines the properties in the states '17' to '20' are calculated. Specific exergy for the working fluid (states '1' to '14') considers both the physical and the chemical portions according to equations 7 to 9.

$$\mathrm{ex} = \mathrm{ex_f} + \mathrm{ex_q} \quad (7)$$

$$\mathrm{ex_f} = \mathrm{h} - \mathrm{h_0} - \mathrm{T_0} \cdot (\mathrm{s} - \mathrm{s_0}) \quad (8)$$

$$\mathrm{ex_q} = \mathrm{x} \cdot \mathrm{ex_{q-NH_3}} + (1 - \mathrm{x}) \cdot \mathrm{ex_{q-H_2O}} \quad (9)$$

In the last equations we considered $T_0 = 295K$ e $P_0 = 101.32\ kPa$ to compute the values of h_0e s_0, x is the ammonia fraction in the working fluid. We considered 341,250 kJ/kmol, 3,120 kJ/kmol and 11,710 kJ/kmol for the specific chemical exergy of the ammonia, the liquid water and the vapor water, respectively [16]. For the states '15' to '20' only the physical exergy was considered. Additionally, for modeling the data presented in Table 2 were assumed. In this table the temperature differentials presented refer to the state of higher temperature in the thermal cycle with respect to the state of lowest temperature, so the values are positive. In the case of ΔT_{sat_9} it is referred to the subcooling degrees under the saturation temperature at the turbine operating pressure. This guarantees that the working fluid will always be subcooled at the economizer outlet. The value of ΔT_s represents the temperature difference between stages '17' and '1'. The other fixed values are the cooling water inlet pressure (P_{15}), the ammonia concentration at the turbine inlet (x_1), and the pump (η_B) and turbine (η_T) isentropic efficiencies. The values that were presented in a range were used for optimization.

Table 2. Input data for modeling

Parameter	Unit	Value
x_1	-	0.96
P_{15}	kPa	250
ΔT_{sat_9}	K	5
ΔT_{5_15}	K	5
ΔT_{3_7}	K	5
ΔT_{4_16}	K	10
ΔT_s	K	20
ΔT_{15_16}	K	8
ΔT_{12_13}	K	44.78
η_B	-	0.85
η_T	-	0.85
Parameter	Unit	Range
P_1	kPa	5,700 – 8,000
q_{10}	-	0.66 – 0.82
x_{10}	-	0.869 – 0.925
ΔT_{19_9}	K	11 - 25

The estimation of the power generated cost is done with equation (10) and data from Table 3:

$$\mathrm{C_{ger}} = \mathrm{CR} \cdot \left(\mathrm{C_{inv}} \cdot \frac{\mathrm{FA}}{\mathrm{HO}} + \mathrm{C}_{O\&M}\right)$$

where

$$\mathrm{C_{inv}} = \frac{\mathrm{IC_{1985}}}{\mathrm{IC_{2013}}} \cdot \mathrm{C_{inv_{ref}}} \cdot \left(\frac{\mathrm{C}}{\mathrm{C_{ref}}}\right)^{0,6}$$

$$\mathrm{FA} = \frac{\mathrm{i} \cdot (1+\mathrm{i})^{\mathrm{n}}}{(1+\mathrm{i})^{\mathrm{n}} - 1}$$

Table 3. Input data for cost calculation

Parameter	Unit	Value	Reference
CR	-	2.34	[17]
HO	h/year	6000	
i	% year	5	
n	year	20	
IC_{1985}	-	175	[18]
IC_{2013}	-	100	[18]
c_{ref}	kW	6000	[19]
$C_{inv_{ref}}$	US$/MW	973,000	[19]
$C_{O\&M}$	US$/MWh	0.096	[19]

The financial values were actualized and referenced for December 2013. In equations 10 to 12C_{ger} is the cost of the generated power, R$/MWh, CR is the change rate, in R$/US$, C_{inv} is the investment cost in US$/MW, FA is the amortization factor, HO is the operation time per year, $C_{O\&M}$ is the operation and maintenance cost, in US$/kWh,

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i is the interest rate, n is the life time, IC is the cost index for a given year, $C_{inv_{ref}}$ is the investment cost at the reference capacity in US$/MW, c is the calculated generation capacity of the Kalina cycle, in kW, e c_{ref} is the reference generation capacity, in kW.

3.3 Heat recovery otimization

The optimal region was located using the optimization procedure by genetic algorithms available in the EES. This optimization method was chosen because it is mathematically robust and running error-free with large number of variables and constraints. In order not to incur large calculation time was set to 3 the number of individuals, in 4 the number of generations, and in 2 the mutation rate. Mathematically the optimization problem was formulated by the Eq. (13). The values of the restrictions are higher than the values in Tab. 2 in some cases. The latter constraints are practical or conceptual, in the case of T_{20}, it is the minimum temperature of the gases in the exhaust to atmosphere and in the case of the generation entropy, $\dot{\sigma}_{ger_i}$, which requires that in each of the system components meet the second law of thermodynamics.

$$\begin{cases} Maximize\ \dot{W} \\ subject\ to: \\ 5{,}700\ kPa \leq P_1 \leq 8{,}000\ kPa \\ 0.40 \leq q_{10} \leq 0.90 \\ 0.20 \leq x_{10} \leq 0.85 \\ 0.90 \leq x_1 \leq 0.96 \\ s_{2s} \geq 0.0\ kJ/kgK \\ T_{20} \geq 340\ K \\ \dot{\sigma}_{ger_i} \geq 0.0\ \ kW/K\, i = 1, 2, \dots, 11 \end{cases} \tag{13}$$

IV. RESULTS AND DISCUSION

Note that in Tab. 3 for most parameters the values found are close to one of their extreme limits. In the cases of P_1, x_{10}, x_1, and ΔT_{12_13} the values were close to the maximum. In the case of ΔT_{19_9} and ΔT_s the values were close to the minimum. In the case of q_{10} the value was close to the middle of the range. Further simulations showed that the variation of ΔT_{12_13} and q_{10} do cause variation in the generated power and the efficiencies of the Kalina cycle. In the case of q_{10} a very little increase in the generate power was noted at 0.74. On the other hand, the value of the ΔT_s must be in the minimum to obtain the maximum power. At the optimal condition the net generated power was 2,725 kW, the thermal efficiency was 0.218, the exergetic efficiency was 0.552, a very high value which is explained by the input source of exergy to the system has a low exergy potential to be a residual gas stream of low temperature. At this condition the cost of the generated power was 278,03 R$/MWh, corroborating the breakthrough that results using KCS34 for generating electricity from waste heat in the cement industry.

Table 4. Values for the optimal condition

Parameter	Unit	Value
P_1	kPa	7,941
q_{10}	-	0.6667
x_{10}	-	0.849
x_1	-	0.960
ΔT_{12_13}	K	47.48
ΔT_{19_9}	K	11.73
ΔT_s	K	23.5

The pressure of the working fluid in the turbine inlet has a strong influence on the net generated power. As we can see in Fig. 2 (A), as increases the pressure at the inlet to the turbine rise the net generated power. The net power also increases with increasing concentration of ammonia in the working fluid in the evaporator. Note that for the maximum concentration of ammonia that was studied in this simulation below the pressure of 6211 kPa at the turbine inlet the system cannot operate because violates the thermodynamics' laws. The increase in the net generated power by the pressure is explained by the increase of the steam enthalpy at the turbine inlet. With increasing ammonia concentration at the evaporator outlet the generated steam flow increases. This happens because with more ammonia in the mixture the working fluid becomes zeotropic causing a uniform temperature profile in the evaporator, which reduces the destruction of exergy and allows you to generate more steam. Note that with the combination of these two parameters the net power varied hundreds of kW values in the ranges studied.

In Fig. 2 (B) it is noted that with increased pressure at the turbine inlet, the thermal efficiency and exergy efficiency also increase. This increase is due to the increased enthalpy of the working fluid at the turbine inlet with increasing pressure. Note further that the influence of pressure on the efficiency is enhanced causing variation in several percentage points.

Reducing the temperature difference between the evaporator outlet gas (state '19') and the working fluid at the evaporator inlet (state '9'), i.e. the ΔT_{19_9}, increases the generation of net power in simple cycle Kalina to increase the generation of steam to the turbine. Note in Fig. 3 (A) that the largest power values were achieved with the highest concentration of ammonia in the working fluid, reinforcing what was already explained before. It is also observed that although varied in the same proportion, the effect of increased ammonia concentration exceeds the reduction of the title at the evaporator outlet (which should reduce the generated steam flow). The ammonia concentration is therefore one of the most influential variables in the generation of net power in cycle. The temperature difference between the evaporator gas output

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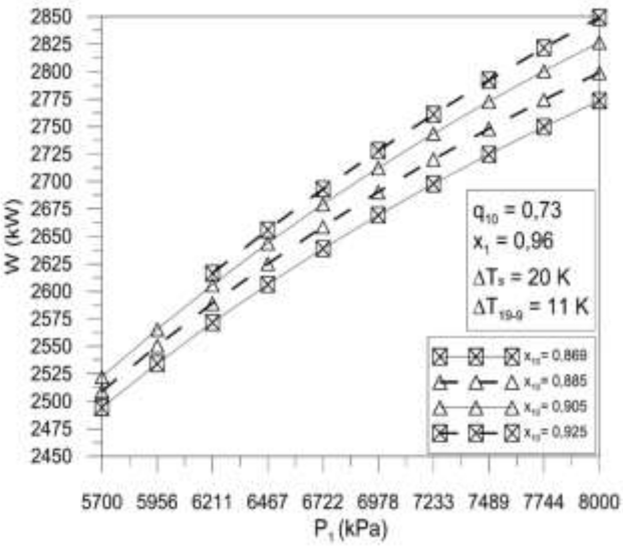
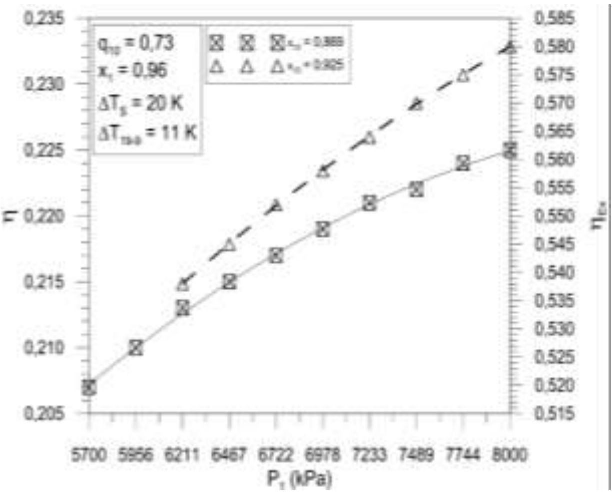
and the working fluid in the evaporator inlet ΔT_{19_9} has a strong effect in the calculation cycle of the investment cost. This variable is directly bonded to the surface of the evaporator heat exchange, and thus to the cost. Thus, operating in the minimum level of ΔT_{19_9}, will not necessarily lead to lower cost of generation.



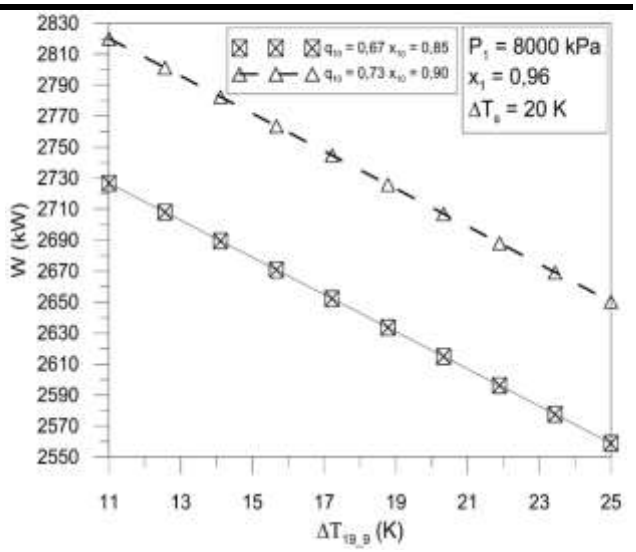
(A)



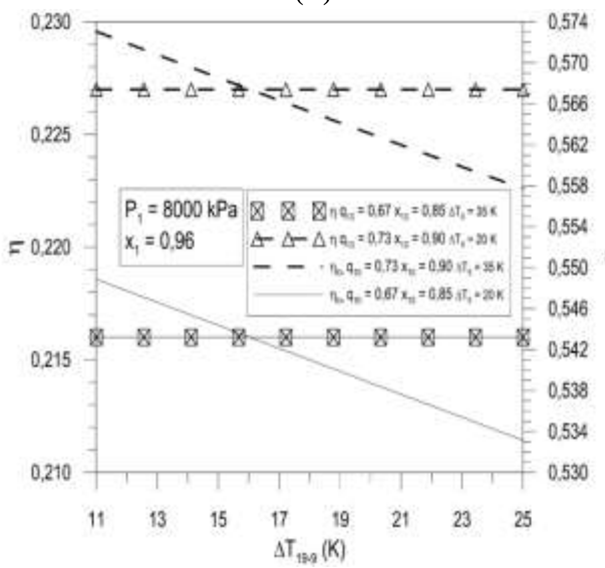
(B)

Fig.2 The influence of the turbine inlet pressure on the KCS34 performance

The influence of the temperature difference between the evaporator outlet and the gas working fluid at the evaporator inlet (ΔT_{19_9}) on the thermal efficiency and the exergetic efficiency can be observed in Fig. 3 (B). The thermal efficiency remains unchanged with the variation of this parameter because it depends mainly on the enthalpy at the inlet and outlet of the boiler and turbine, which is not altered by the ΔT_{19_9}. On the other hand, the exergetic efficiency becomes sensitive to this parameter (ΔT_{19_9}), which once changed, impacts on exergy destruction in the evaporator, which is the largest in the cycle.



(A)



(B)

Fig.3. The influence of the o ΔT_{19_9} on the KCS34 performance

The effect of varying the pressure at the inlet to the turbine on the generating cost can be seen in Fig. 4 (A). According to this figure the cost of electricity generated increases with pressure, which has a practical sense, because to adopt higher working pressures requires greater material thickness and higher cost. But the right explanation for this behavior is the cost modelthat was adopted. As it was said before, the value of the cost of the electricity generated is associated to the generation amount and the same has been observed which increases with the pressure at the turbine inlet.

Figure 4 (B) shows the effect of temperature variation in the evaporator difference (ΔT_{19_9}) on the generation cost. The cost of electricity generated decreases with increasing temperature difference in evaporator following the same trend behavior power according to the adopted cost calculation model. The trend shows that the cost model is

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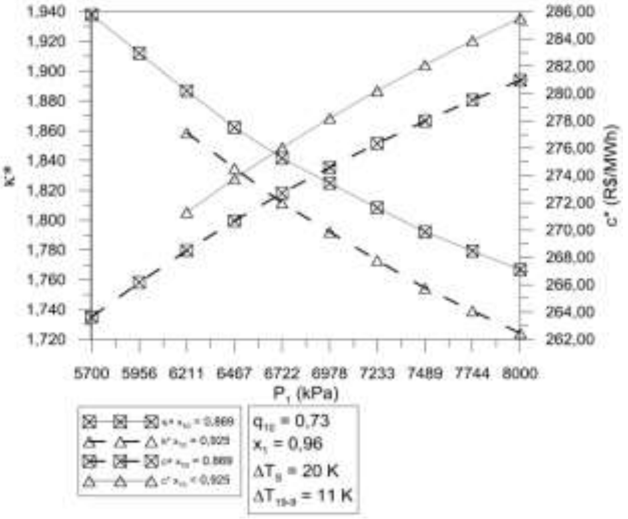
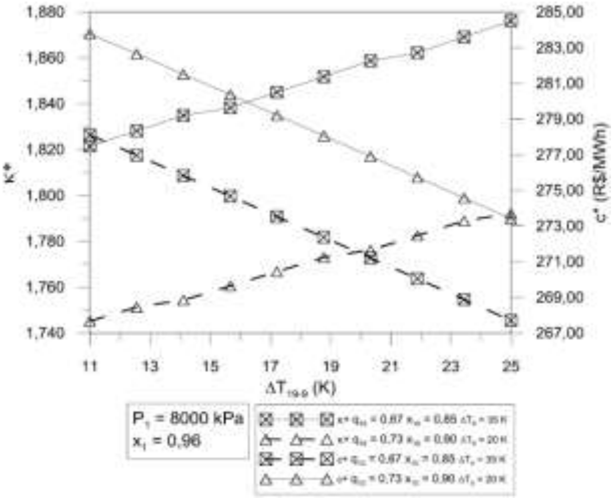
assertive in order to reflect that a greater temperature difference in the evaporator will reduce the purchased cost with thisequipment and thus causinga lower total cost of the cycle and in the generated electricity.



(A)



(B)

Fig.4. The influence of the turbine inlet pressure and o ΔT_{19_9} on the KCS34 generation cost

From the information shown in Fig. 4 the generated power range between 2,500 and 2,850 kW with generation costs values ranging from 262.00 and 286.00 R $/MWh. The optimization performed with genetic algorithms finds a middle ground in the region with power values and costs of 2,725 kW and 278.03 R$/MWh, respectively. These results allow us to state that depending on the context of this great region it is possible to opt for a solution that leads to a minimum generation cost, with less generation of power, or a solution leading to a generation of maximum electricity but with a greater generation cost. In any case, the cost of generated power is competitive in the Brazilian market. According to the Ministry of Mines and Energy [20] the average tariff value supply practiced by CompanhiaEnergética de Minas Gerais- CEMIG for the industrial sector in May 2014 was 296.38 R$/MWh.

V. CONCLUSION

The study about the waste heat recovery in the cement industry with KCS34 lets you express the following conclusions:

- The amount of electricity generated can reach 2,725 kW, with a thermal efficiency of 0.218, an exergetic efficiency of 0.552 and a generation cost of 278.03 R$/MWh;
- The pressure of the working fluid in the turbine inlet has a strong influence on the net power generated. With greater pressure at the turbine inlet, the generated power, the thermal efficiency and the exergetic efficiency increase;
- The net power generated also increases with increasing concentration of ammonia in the working fluid in the evaporator. The ammonia concentration is therefore the most influential variables in the generation of net power in the cycle;
- With the combination of the pressure values at the turbine inlet and the ammonia concentration in the evaporator that were studied, the net power ranged hundreds of kW;
- The influence of pressure on the efficiency is enhanced, causing variation in various percentage points;
- Reducing the temperature difference,ΔT_{19_9}, increases net power generation in KCS34. On the other hand, the exergetic efficiency becomes sensitive to this parameter since it impacts on exergy destruction in the evaporator;
- The cost of the generated electricity increases with pressure and decreases with increasing temperature difference in evaporator following the same trend of the generated power;
- The cost model is assertive to reflect that a greater temperature difference in the evaporator will reduce the purchased cost with this equipment and thus causing a lower total cost of the cycle and in the generated electricity;
- In the great region calculated the generation of values may vary in the range between 2,500 and 2,850 kW with generation costs ranging between 262.00 and 286.00 R$/MWh. This cost range is competitive in the Brazilian market due to the value of the average supply tariff applied by CEMIG, which for the industrial sector in May 2014 was 296.38 R$/MWh.

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