HEAT TRANSFER IMPROVEMENT IN EVAPORATIVE CONDENSERS BY TWO SIMULATION MODELS

Alex Trigo Nakalski alexnakalski@yahoo.com.br
Paulo Smith Schneider pss@mecanica.ufrgs.br
Paulo Otto Beyer paulo.beyer@ufrgs.br
Rodrigo Danieli rdanieli@hotmail.com

Departamento de Engenharia Mecânica – Universidade Federal do Rio Grande do Sul – UFRGS – Porto Alegre, RS, Brazil

Abstract. The present paper aims to present the results obtained from two simulation models for heat transfer in evaporative condensers operating in industrial refrigeration systems, here on called Global Adjustment and Psychometric models. The condenser operates with ammonia (R-717) inside the tubes and a mixed flow of air and water on the outside, creating a strong dependency of environment conditions. Heat transfer is calculated by both approaches but in the Psychometric model it depends on the heat exchanger effectiveness and on the psychometric parameters of the air, whereas in the Global Adjustment model it is a function of heat and mass transfer coefficients. Both models depend on the determination of adjustment coefficients from experimental data. A heat transfer improvement in evaporative condensers is proposed after running the simulation models. Results from both models are displayed for different operating conditions along with their sensitivity analysis, showing similar ranges.

Keywords: Evaporative Condensers, Evaporative Coolers, Industrial Refrigeration, Ammonia

1. INTRODUCTION

Evaporative Condensers or Coolers (EC) are widely used in refrigeration systems to condensate and remove heat of refrigerant fluids aiming storage and conservation of foods in general. They are able to reach high rates of heat transfer per unity of tube surface due to the evaporation of water to the air, on an intensive outside mixed flow and they can also operate with lower discharge temperature having great effectiveness. On the other hand, incrustation of water impurities over the coil external surface may affect heat removal and consequently the equipment performance.

Mathematical models of EC have importance in the process of project and improvement, because its simulation may avoid expensive experimental procedures. Zalewski *et al* (1997) presented a model for evaporative condensers based on four differential equations and the boundary conditions, and relied on correction factors for better accuracy. Hwang *et al.* (2001) compared the performance of an EC to the one of an air condenser. They noticed that the coefficient of performance COP tends to be 11% to 21% bigger for EC. Lebrun *et al.* (2004) proposed a model based on direct and indirect contact in evaporative coolers. Heat rejection was calculated using their heat transfer effectiveness and the model presented also good accuracy. Camargo *et al.* (2004) presented a performance evaluation of an EC operating along summer periods in a Brazilian city. They noticed that the effectiveness tends to raise for higher values of dry bulb temperatures. Ertunc *et al*. and Abassi *et al*. (2005) presented models based on neural analysis, also with good accuracy. Danieli *et al*. (2006) presented a performance evaluation of four different external coils surfaces. It was noticed that the external surface has significant influences on the equipment performance.

The purpose of this paper is to analyze two different models and to propose modifications in the condenser structure to get a heat transfer improvement. One model is based on the overall heat and mass transfer coefficient and the other one is based on air enthalpy and effectiveness, as will be shown later.

Figure 1 presents an EC scheme, where a counter current flow of water and ambient air is imposed at the outside of the coils, where water evaporates to the limit of air saturation. Ammonia steam flows and condenses in the inside of the same coil. The particular EC studied in this paper is manufactured by FrostFrio (www.frostfrio.com.br) and rejects about 230 kW or 200 Mcal/h.

2. MATHEMATICAL MODELS

In the present paper, two models will be used to represent heat transfer and effectiveness in EC: the Global Adjustment, proposed initially by Parker and Treybal (1961) and a Psychrometric based model (ASHRAE, 1993). Both models relay on parameter adjustment procedures, based on experimental data, which will be described on the next section.

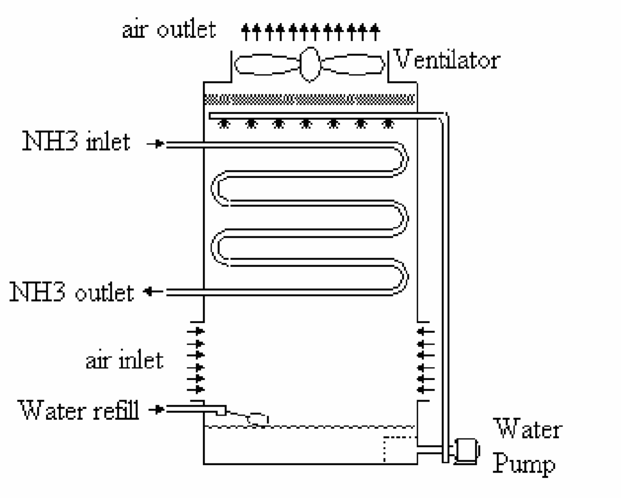

Figure 1. Evaporative Condenser Scheme.

2.1. Global Adjustment model (Parker and Treybal , 1961)

This model starts with the evaluation of heat and mass transfer coefficients U and k_m, for the interfaces of the tubes with ammonia, at the internal side, and with the air-water mixture stream at the external side. Correlations for heat and mass transfer (Bejan 1995; Welty et al. 2001) must be corrected or adjusted by experimental data (Centeno ,2005; Nakalski et al., 2006), leading to the calculation of a adjustment factor F_{AG}.

For every set of experimental data there will be a new and different value of F_{GA}, whose determination is the solution of the system of equations (1) to (3).

$$Q_{\exp} = U_C \left(T_{cond\_NH3} - T_{air-H2O}\right) = k_{m-C} \left(h_{air-H2o} - h_{air}\right) \tag{1}$$

$$U_C = \frac{1}{\frac{D_e}{D_i}\left(\frac{1}{h_i}\right) + \frac{D_e}{D_a}\left(\frac{L}{k}\right) + \frac{1}{F_{GA} h_e}} \tag{2}$$

$$k_{m-C} = F_{GA}\, k_m = F_{GA}\, 0,0924\, \dot{m}_{air}^{\,0,905} \tag{3}$$

In Equation (1), Q_{exp} is the experimental value of the rejected heat from the EC, $T_{cond-NH3}$-$T_{air-H2O}$ is the temperature difference between the refrigerant condensation state to the water film state (ºC or K), and $h_{air-H2O}$ - h_{air} is the difference of enthalpies from mixed air water to humid air (kJ/kg). Index C stands for correction. In equation (2), D is the diameter (m), h is the convective heat transfer coefficient (kW/(m^2 K)), L is the tube thickness (m), and k is the thermal conductivity of the tubes (kW/(m K)). Indexes o, i and a are respectively for the external, internal and average tube positions. An average value of F_{GA} is to be found in order to simulate the EC for a wider range of physical situations.

The Global adjustment model allows simulating the heat rejection of the EC in relation to the heat transfer surface, air and water mass flow rates, among others. Some specific cases were performed to verify the model consistency to the parameters listed above. Figure 2 shows an asymptotic behavior of the overall coefficient of heat transfer in relation to the water mass flow rate.

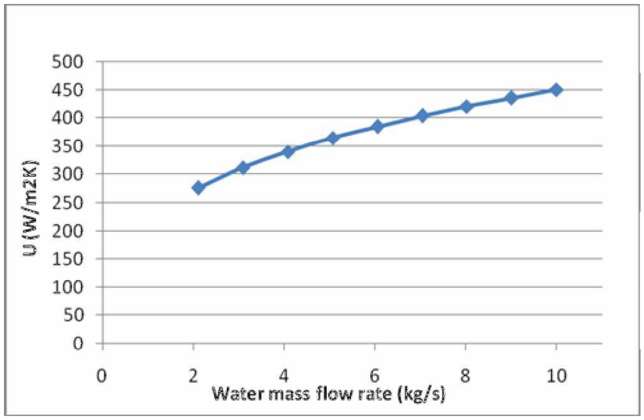

Figure 2. Overall coefficient of heat transfer *U versus* water mass flow rate.

That behavior is expected for heat exchangers, meaning that extra enhancements on heat transfer will increase the total heat transfer rate, but not its efficiency. Next figure shows the calculated heat rejection Q_{GA} as a function of the air mass flow rate for three values of heat transfer surfaces.

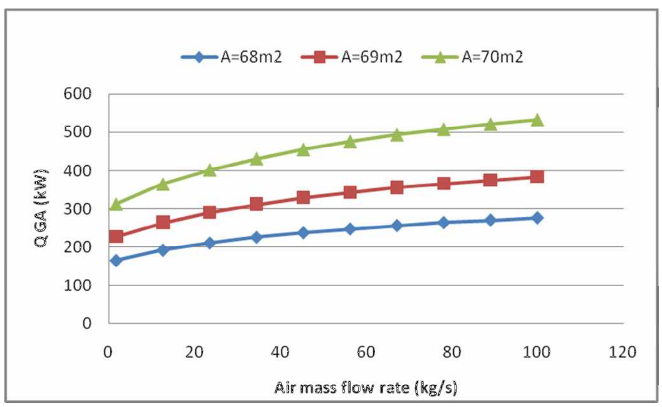

Figure 3. Simulated heat rejection rate Q_{GA} *versus* air mass flow rate for three heat transfer surfaces values.

An asymptotic behavior is again clearly identified. Heat rejection is both sensitive to the air flow rate as well as to the increase in heat transfer surface area.

Figure 4 shows the simulated behavior of the heat rejection rate Q_{GA} in relation to heat transfer surface alone. That result is similar to the one found with the Psychrometric model, to be shown latter on this text, and agrees with the expected behavior of heat exchangers.

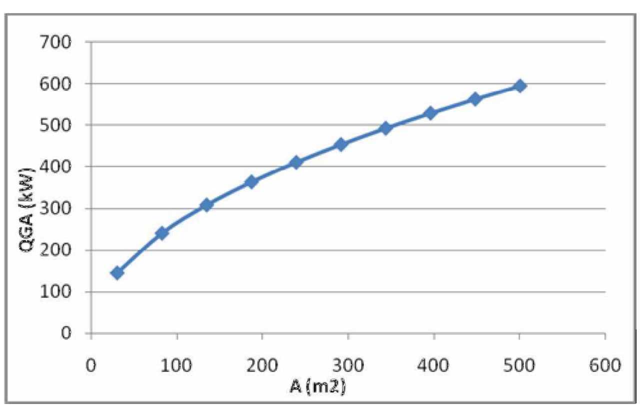

Figure 4. Simulated heat rejection rate Q_{GA} *versus* heat transfer surface.

2.2. Psychrometric model.

Described by ASHRAE (1993), it relies on the changes of state of the humid air as a predominant phenomenon, which is associated to the effectiveness of heat transfer in condensers. Effectiveness ε is a dimensionless parameter widely used to characterize heat exchanger performance, as it is the ratio between the difference on the temperature of one stream for a real heat exchanger and the same difference if that heat exchanger would have an infinite surface area.

Effectiveness ε is calculated as a function of the number of transfer units NTU by:

$$\varepsilon = 1 - e^{-NTU} \tag{4}$$

$$NTU = \frac{F_P\, A\, h_e}{\dot{m}_{air}\, c_{p-air}} \tag{5}$$

where A is the total heat transfer surface of the tubes (m^2), $\dot{m}_{air}$ the air mass flow rate (kg/s), and c_{p-air} is the specific heat of air evaluated at the inlet temperature (kJ/(kg K)). The convective heat transfer coefficient h_e is calculated using Zhukauskas' correlation to a bench of tubes where only air flows across it. Therefore, an adjustment factor F_P is introduced to better approximate the results calculated by models to the experimental data.

Heat transfer Q_P is calculated by Equation 6:

$$Q_P = \varepsilon\, \dot{m}_{air}\, \Delta h_{air} \tag{6}$$

where air enthalpy h_{air} can be evaluated using the relation:

$$h_{air} = 1,006 T_{db} + (2501 + 1.805 T_{db})\, w \tag{7}$$

with T_{db} the air dry bulb temperature (K), and w the humidity content (kg of water steam per kg of dry air). Similar to the procedure adopted in the Global Adjustment model, the determination of F_P is obtained by solving the set of equations (5) to (7).

The Psychrometric model allows the simulation of heat rejection according to a set of environmental parameters as the dry and wet bulb temperatures of air and the refrigerant condensation temperature, making it possible to evaluate heat transfer in respect to air mass flow rates and heat transfer surface. The Psychometric model was simulated to verify its consistency. Figure 5 shows the simulated heat rejection rate Q_P *versus* wet bulb temperature of air for several values of refrigerant condensation states.

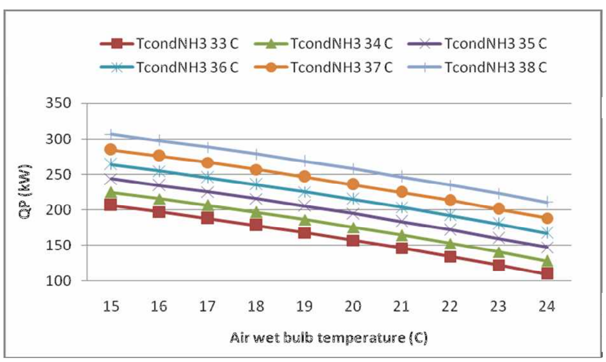

Figure 5. Simulated heat rejection rate Q_P *versus* wet bulb temperature for several levels of refrigerant condensation.

Figure 6 displays results from simulation of heat transfer rates throughout a range of air mass flow rates at several values of wet bulb temperature of the inlet air. It shows clearly that the heat rejection behavior goes in the opposite direction of the variation on the wet bulb temperature. Concerning air mass flow rates, heat rejection rate rises initially but then tends to become steady. Finally, the simulation of the heat rejection rate Q_P as a function of the heat transfer surface area in Figure 7 shows again an asymptotic behavior.

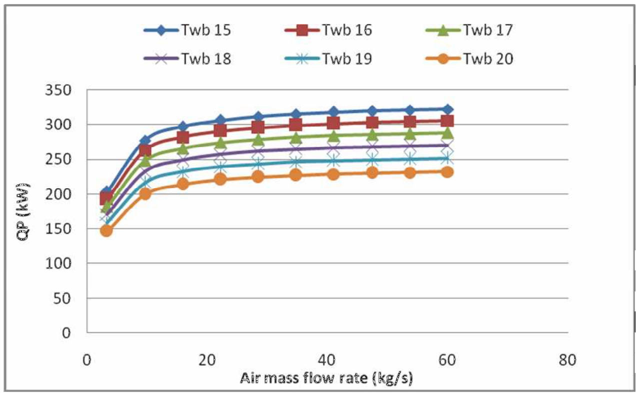

Figure 6. Simulated heat rejection rate Q_P *versus* air mass flow rates to several values of wet bulb temperature.

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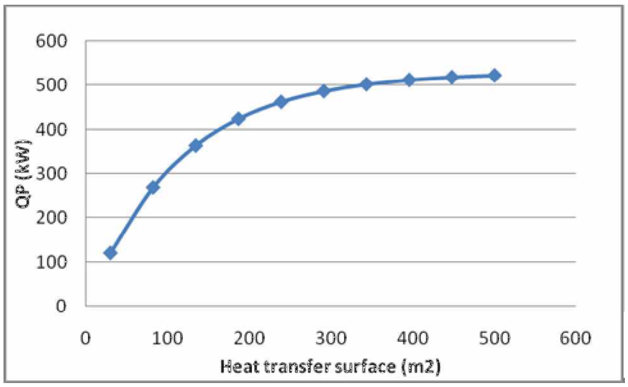

Figure 7. Simulated heat rejection rate Q_P *versus* heat transfer surface area.

3. Experimental data *versus* heat transfer models.

An experimental set of data from a real scale ammonia system, specially built for this purpose along this research project, was used to determinate the adjustment coefficients F_{GA} and F_P. These data are displayed on Table 1 and organized in two annual sequences; both for FrostFrio EC model RC200.

Table 1. Experimental set of data from a real scale ammonia system

1st sequence	1	2	3	4	5
$T_{cond\text{-}NH3}$ (°C)	32.4	33.7	34.2	30.4	30.1
$Tdb_{\text{-}i}$ (°C)	18.1	18.5	20.2	13.3	14.6
$Twb_{\text{-}i}$ (°C)	15.9	14.6	15.0	10.9	11.6
$\dot{m}_{NH3}$ (kg/s)	0.235	0.210	0.191	0.222	0.221
Q_{exp} (kW)	306.1	273.8	247.3	287.7	289.1
q_{exp} (kJ/kg)	1302.6	1303.8	1294.8	1295.9	1308.1

2nd sequence	1	2	3	4	5
$T_{cond\text{-}NH3}$ (°C)	34.0	33.8	36.2	35.1	35.1
$Tdb_{\text{-}i}$ (°C)	27.4	17.1	27.4	21.0	26.4
$Twb_{\text{-}I}$ (°C)	16.2	13.2	19.4	15.8	16.5
$\dot{m}_{NH3}$ (kg/s)	0.140	0.176	0.177	0.185	0.216
Q_{exp} (kW)	182.5	235.8	229.4	239.1	281.3
q_{exp} (kJ/kg$_{NH3}$)	1303.6	1339.8	1296.0	1292.4	1302.3

For both sequences, the temperatures of ammonia condensation $T_{cond-NH3}$, dry and wet bulb of the incoming air Tdb_{-i} and Twb_{-i} and the mass flow rate of ammonia $\dot{m}_{NH3}$ were measured to calculate the thermal rejection rate of the EC Q_{exp} (ASHRAE, 1995). These sequences were performed on two different periods of time and the condensers, although from the same model, had slight changes on their design. It is interesting to highlight that the values of specific thermal rejection rates q_{exp}, as a function of $\dot{m}_{NH3}$ (kJ/kg$_{NH3}$) are quite uniform throughout the experiments. Mean values are 1301.0 and 1306.8 kJ/kg$_{NH3}$ and standard deviations are 5.6 and 19.0 kW for sequence 1 and 2, respectively. The second set of experiments is less accurate than the first one according to their standard deviations.

By introducing experimental data on both models it was possible to evaluate the average values for F_{GA} and F_P and then calculate the heat rejection rate Q_{GA} and Q_P, shown in Table2. The relative errors or biases in respect to the experimental values of heat rejection are also displayed.

Table 2. Calculated heat rejection by Global Adjustment and Psychometric models and their errors for the two sequences of data using average values of F_{GA} and F_P.

1st sequence	**1**	**2**	**3**	**4**	**5**	**average**	σ
Q_{GA} (kW)	281.6	282.5	268.8	286.5	289.1	281.7	7.8
q_{GA} (kJ/kg$_{NH3}$)	1198.30	1345.24	1407.33	1290.54	1308.14	1309.9	76.8
Error Q_{GA} (%)	-8.00	3.18	8.69	-0.42	~ 0		
Q_P (kW)	247.8	291.6	286.8	262.2	254.6	268.6	19.6
q_P (kJ/kg$_{NH3}$)	1054.47	1388.57	1501.57	1181.08	1152.04	1255.5	183.7
Error Q_P (%)	-19.05	6.50	15.97	-8.86	-11.93		

2nd sequence	**1**	**2**	**3**	**4**	**5**	**average**	σ
Q_{GA} (kW)	218.5	228.3	228.7	231.3	242.8	229.92	8.69
q_{GA} (kJ/kg$_{NH3}$)	1303.57	1339.77	1296.05	1292.43	1302.31	1304.86	159.18
Error Q_{GA} (%)	19.73	-3.18	-0.31	-3.26	-13.69		
Q_P (kW)	212.9	236.4	239.8	238.6	232.6	232.1	11.06
q_P (kJ/kg$_{NH3}$)	1520.71	1343.18	1354.80	1289.73	1076.85	1317.06	159.71
Error Q_P (%)	16.66	0.25	4.53	-0.21	-17.31		

The deviation of the calculated values of heat rejection rates by both models from the experimental data are displayed in percentage in Table 2, and it can be noticed that they are either positive and negative as the result of mean values of the adjustment factors F. They were taken as 0.1906 and 0.01695 for F_{GA} and 1.9 and 1.4 for F_P, in respect to the first and second sequences of experimental data. The use of the specific heat rejection rate q helps to analyze the behavior of the EC in a more independent way in respect to the environmental conditions. The best result was the one from the Global Adjustment model for the first sequence of data, as the standard deviation was the smallest.

An error propagation analysis was performed with the experimental data and it was found that the uncertainties of the heat rejection rate Q_{exp} are small than the dispersions of the calculated heat rejections from the models. That was done by taking uncertainties of 0.1 ºC for all temperature measurements, 0.2 bar for pressure and 1% for the mass flow rate of ammonia. The formulation proposed by Taylor *et al*. (1994) for error propagation was applied to Equation 8

$$Q_{\exp} = \left(\dot{m}\,\Delta h\right)_{NH3} - \left(\dot{m}\,\Delta h\right)_{H2O} \tag{8}$$

proposed by ASHRAE Standard 64 (1995) to calculate the experimental heat rejection rate. From that value, the ammonia mass flow rate presented a dominant influence of 99,48% in the final result, being extremely important a good accuracy on measuring that quantity. The same procedure was performed for both simulation models and it was found, for the Psychrometric model, a influence of 96.6% of the ammonia pressure in the result of the heat rejection rate due to the fact that it is used directly to calculate the water film temperature and the air enthalpy, affecting heat rejection rates. For the Global Adjustment model, it was noticed that air and ammonia mass flow are determinant, with 51.14 % and 37.51 % of influence respectively in the heat rejection rates. Good accuracy on the measurement of those variables is required.

4. IMPROVEMENTS

One of the goals of the present project is to use the simulation routines as a tool to identify opportunities of improvement of the EC. By running the Psychrometric model, it was possible to predict that the raising on heat transfer surface would not necessarily lead to a proportional rejection of heat by the equipment. Table 3 shows both heat rejection rate Q_P and the corresponding heat flux Q''_P calculated by the Psychrometric model for a uniform air mass flow rate generated by an axial fan.

Table 3. Heat rejection rate and flux calculated by Psychrometric model equation (m_{air}=6 kg/s).

A (m2)	63	64	65	66	67	68	69	70	71	72
Q_P (kW)	233	235	238	241	243	246	248	251	254	256
Q''_P (kW/m^2)	3,70	3,68	3,66	3,65	3,63	3,61	3,60	3,59	3,57	3,56

Despite the fact that the heat rejection rates raise with the augmentation of the heat transfer surfaces area, the heat flux tends to decrease. Values of heat flux found on Stoecker and Jabardo (2002) for the heat flux in evaporative coolers are about 4 kW/m^2, meaning that a reduction on the heat transfer surface area, from the actual size of 69 m^2 to about 60 m^2, would increase its performance.

The behavior of heat transfer as a function of the air mass flow rate is displayed on Table 4, where the demanded electrical power on fans were measured for the EC.

Table 4. Demanded electrical power on fans and the corresponding heat rejection gains on EC.

Fan current (A)	2,8	3,47	4,13	4,8	5,47	6,13	6,8
Air mass flow rate (kg/s)	5,33	5,67	6,00	6,33	6,67	7,00	7,33
Electrical power (kW)	1,0	1,2	1,4	1,7	1,9	2,2	2,4
Thermal rejection rate gain on the EC (kW)	4,4	3,9	3,6	3,3	2,9	2,6	2,3

There is little gain on heat rejection rates for increasing values of air mass flow rates, but there is a undesirable increase on the pressure drop, that would practically equalize the heat rejection improvements.

An analysis of the overall heat transfer coefficient U was performed, for the three terms on the denominator of Equation 2. By evaluating separately each one of them, it was found that the convective heat transfer coefficient h_i, inside the tubes, is quite important, and the convective resistance would be of the order of 10^{-4} (m^{2}K)/W. In a similar way, the relation between tube thickness L and thermal conductivity k is of the order of 10^{-5} (m^{2}K)/W. These two resistances have little influence in the final result of U. The external heat transfer coefficient h_e, between the tubes and the water film, leads to a convective resistance of about 10^{-3} (m^{2}K)/W and must be the one to be improved. As shown by Danieli *et al.* (2006), the external tubes roughness may improve heat transfer in certain cases. Therefore, one way to raise this coefficient is making the external tubes surface somehow rougher. Certainly, the contact surface would become bigger, making water film formation easier and finally raising global coefficient U. The disadvantage side of this proposal comes from the increasing on impurity deposition in the tubes. It is necessary whatsoever to use some kind of filter to clean the feeding water.

Finally, new prototypes will be built by FrostFrio, and the experimental rig will be useful to perform tests and compare this new proposal to the current used.

5. CONCLUSION

The Global Adjustment model presented a lower deviation compared to the Psychometric model, but nothing can be stated without many more experimental essays. The uncertainty analysis showed that some parameters are determinant to calculate heat transfer, and the ammonia mass flow rate popped up as the most important one to evaluate the experimental uncertainty.

It also noticed that the convective heat transfer coefficient between the tubes and the water film must be well evaluated in order to improve the EC performance. Roughness improvement may act in a very positive direction, bur at the same time can introduce higher pressure drops.

Running both models presented in this paper showed that raising heat transfer surface area does not necessarily bring gains on the heat rejection. It was noticed that the heat flux, i.e., the heat transfer rejection by surface area, tends to become lower as the heat transfer surface area increases.

Some new ideas that come out from this study will be experimentally tested and compared to the current performance of the current model RC200.

6. REFERENCES

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