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TRANSIENT HEAT TRANSFER ANALYSIS OF A WINE STORAGE CABINET

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Abstract. *Wine coolers need to guarantee a highly-controlled environment in terms of temperature and humidity for wine storage. According to the International Standard, wine coolers must keep the temperature and the relative humidity of the refrigerated compartments, respectively, in the ranges of 5 to 20 °C and 50% to 80%. Since the temperature difference between the ambient and the cooled cabinet is not so high, the cabinet can operate with a magnetic cooling system. In this work, a transient heat transfer analysis of the cabinet components has been developed. The time required to cool down the wine bottles to a prescribed temperature for different operation conditions of the magnetic cooling system and for varied sizes of the cold heat exchanger have been calculated. The results indicate that the operating conditions of the magnetic refrigeration system and the overall thermal conductance of the cold heat exchanger have a significant impact on the time required to cool down the wine bottles. This should be considered in the design of a wine cooler operating with a magnetic refrigeration system.*

Keywords: *Cabinet, wine storage, heat exchangers, thermodynamic performance, magnetic refrigeration.*

1. INTRODUCTION

Magnetic refrigeration is an emerging cooling technology based on the magnetocaloric effect (MCE) of magnetic materials and is an alternative to mechanical vapor compressor. The MCE is the thermal response of the material when subjected to a variation in the applied magnetic field. This effect can be perceived as a temperature variation, which is maximum at the material's Curie Temperature. For some materials the MCE is reversible, such as the benchmark material Gadolinium, whose Curie Temperature varies from 290 to 297 K with a temperature variation on order of 5 K when the magnetic field changes from 0 T to 2 T (Bahl & Nielsen, 2009). However, this temperature variation is not enough to cool down a conventional refrigerator. The effect is amplified with the use of an active magnetic regenerator (AMR) which is in a porous matrix made of magnetic materials.

In this work, the thermal analysis of a wine storage cabinet operating with a magnetic refrigeration system and using real heat exchangers has been carried out. A mathematical model was developed to evaluate the heat transfer between the internal components of the cabinet. The evaluation of the cooling capacity provided by the magnetic cooling system was made by a previous mathematical model (Calomeno *et al.*, 2016) based on the ε-NTU method, which was implemented in the AMR solver developed by Trevizoli (2015). The simulations consider an AMR with a gadolinium packed-sphere regenerator. A transient heat transfer analysis was proposed to estimate the pull-down time for a cabinet for different operation conditions of the AMR and varied sizes of the cold heat exchanger by means of the variation of its overall thermal conductance (UA).

2. METHODOLOGY

The wine storage cabinet is assumed to be cooled down by an AMR system. A schematic diagram of a magnetic refrigerator operating with heat exchangers is presented in Figure 1. The AMR cycle is based on a regenerative Brayton cycle in which, firstly, the AMR is magnetized adiabatically, then its temperature rises due to the MCE. Simultaneously, the cold blow crosses the AMR with a heat transfer fluid that comes from the cold heat exchanger at temperature T_{CHEx} and cools the AMR matrix and leaves the regenerators with temperature of T_{HE}. The fluid goes to the hot heat exchanger, where it rejects heat ($\dot{Q}_H$) to the external environment at temperature T_H and leaves the heat exchanger with a temperature of T_{HHEx}. The AMR is demagnetized adiabatically and the temperature of the regenerator matrix decreases, simultaneously, the fluid returns from the hot heat exchanger and performs a hot blow when crosses the regenerator while heating it. The fluid leaves the regenerator at temperature T_{CE} and goes to the cold heat exchanger, which absorbs heat ($\dot{Q}_C$) from the cooled environment at a temperature T_C. The fluid returns from the cold heat exchanger at a temperature T_{CHEx} to perform another cold blow.

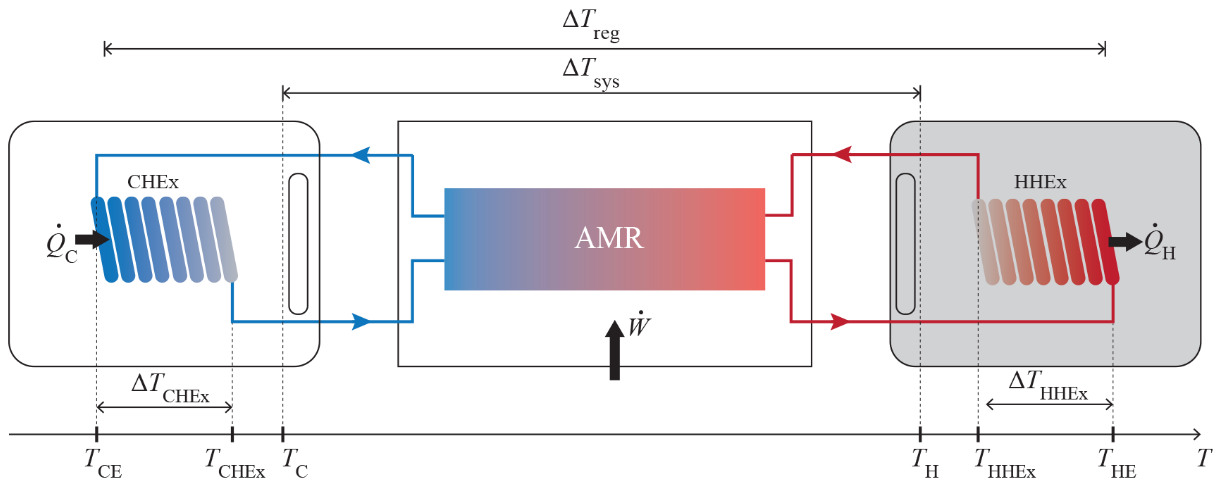


Figure 1. Schematic diagram of a refrigeration system operated by an AMR and its related variables (Calomeno *et al.*, 2016).

A mathematical model has been developed to evaluate the transient heat transfer inside the wine cooler components. The cabinet was basically subdivided in three domains: air, wine bottles and walls. In this model, it was considered that the domain of the wine bottles is divided in two materials: the glass and the water. Water has been chosen for the simulation because the International Standard IEC 62552-1 (International Electronic Commission, 2015) recommends that on experimental tests the wine bottles shall be filled up with water instead of wine. The walls consist on two different layers: liner and thermal insulation. A schematic illustration of the domains and all the heat transfer processes can be observed in Figure 2.

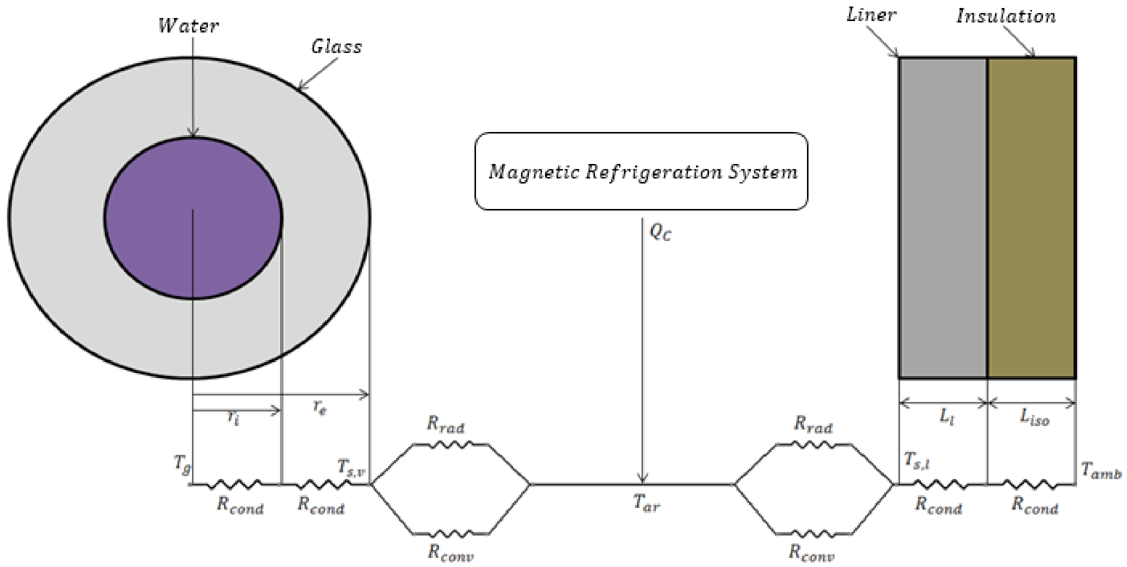


Figure 2. Schematic representation of the heat transfer domains inside the wine storage cabinet.

During the development of the mathematical model, the following hypothesis were considered:

- Thermophysical properties of glass, liner, thermal insulation and water were constant;
- One-dimensional heat transfer in wine bottles and cabinet walls;
- Natural convection inside the wine bottles was disregarded;
- There were no sources of internal heat generation.

The energy equation applied to the cabinet walls considers only pure heat diffusion and takes this form:

$$\rho . c_p \frac{dT_x}{dt} = \frac{\partial}{\partial x}\left(k . \frac{\partial T_x}{\partial x}\right) \tag{1}$$

where ρ, c_p e k are, respectively, density, specific heat and thermal conductivity of the material and T_x is the wall temperature.

The finite volume method is applied to obtain the wall temperature field as function of space and time. The wall is discretized in finite volumes on the horizontal direction, as can be observed on Figure 3.

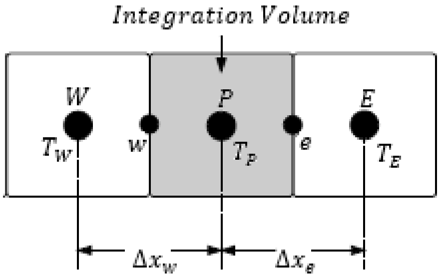


Figure 3. 1-D schematic grid of the wine cooler wall (Adapted from Trevizoli, 2015).

The control volumes on the inner and outer surfaces of the wall have half of the size of the respective material control volumes. Integrating equation 1 in time and space:

$$\int_{t_o}^{t}\int_{x_o}^{x}\int_{y_o}^{y}\int_{z_0}^{z} \rho . c_p . \frac{dT_x}{dt} dz\, dy\, dx\, dt = \int_{t_o}^{t}\int_{x_o}^{x}\int_{y_o}^{y}\int_{z_0}^{z} \frac{\partial}{\partial x}\left(k . \frac{\partial T_x}{\partial x}\right) dz\, dy\, dx\, dt \tag{2}$$

Using the full implicit formulation, equation 2 turns into:

$$\frac{\rho . c_p . \Delta x . A_W}{\Delta t}\left(T_P - T_P^o\right) = k . A_w . \frac{\partial T_x}{\partial \mathrm{x}}\bigg|_e - k . A_w . \frac{\partial T_x}{\partial \mathrm{x}}\bigg|_w \tag{3}$$

where A_W is the area of the inner walls of the cabinet and T_P^o is the temperature in the previous time step. Equation 3 can be rewritten in this form:

$$a_p . T_P = a_e . T_E + a_w . T_W + a_p^o . T_P \tag{4}$$

Equation 4 is used to obtain the temperature profile along the time inside the wall. For this, two boundary conditions are needed. The first boundary condition is on the wall surface inside the cabinet $(x = 0)$, where the liner transfer heat towards the air by radiation and natural convection. Thus:

$$k . \frac{\partial T_x}{\partial x}\bigg|_{x=0} = \left(h_{rad,w} + h_{conv,w}\right) . \left(T_a - T_x|_{x=0}\right) \tag{5}$$

The radiation heat transfer coefficient is calculated by the following equation:

$$h_{rad,w} = \varepsilon_l . \sigma . \left(T_a^2 + T_x^2|_{x=0}\right) . \left(T_a + T_x|_{x=o}\right) \tag{6}$$

where ε_l is the liner emissivity, σ is the Stefan-Boltzmann constant and T_a is the air temperature.

The convection heat transfer coefficient is calculated by the correlation of Churchill & Chu (1975) for natural convection over a vertical wall presented by Incropera and DeWitt (2006):

$$h_{conv,w} = \frac{k_a}{H}\left\{0.825 + \frac{0.387 . Ra_H^{1/6}}{\left[1 + (0.492/Pr)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \tag{7}$$

where k_a is the thermal conductivity of the air, Pr is the Prandtl number of the air and the Rayleigh number is evaluated by the following equation:

$$Ra_H = \frac{g.\beta.(T_x|_{x=0} - T_a).H^3}{\alpha.\nu} \tag{8}$$

where g is the gravitational acceleration, $\beta, \alpha\ and\ \nu$ are, respectively, the thermal expansion coefficient, thermal diffusivity and the kinematic viscosity of the air and H is the wall height.

The second boundary condition is a prescribed temperature condition, where the outer walls of the cabinet have the same temperature of the outer environment (T_∞), thus $T_x|_{x=L} = T_\infty$.

The wine bottles were simulated as horizontal cylinders where the external radius and the thickness were kept constant and the height of the bottles was adjusted to the liquid inside the cylinder has the same volume of the liquid of a normal wine bottle. In these simulations, it was considered that the number of bottles (N_B) inside the cabinet is 31, the wine bottles have external radius of $r_o = 3.85$ cm, thickness of $t_g = 4$ mm and equivalent height of $H_B = 25$ cm.

The temperature along the wine bottle radius (T_r) is obtained from the energy equation presented by the following equation:

$$N_b.\rho.c_p.\frac{dT_r}{dt} = N_b.\frac{\partial}{\partial r}\left(k.\frac{\partial T_r}{\partial r}\right) \tag{9}$$

The wine bottles were divided in finite annular volumes, in an analogous way as was used for the cabinet walls. Using the same method of integration, equation 9 becomes:

$$\frac{2\pi.r_p.(r_e - r_w).H_b.N_b.\rho.c_p.}{\Delta t}(T_P - T_P^o) = 2.\pi.H_b.N_b.k.\left(r_e.\frac{\partial T_r}{\partial \mathrm{r}}\bigg|_e - r_w.\frac{\partial T_r}{\partial \mathrm{r}}\bigg|_w\right) \tag{10}$$

One of the boundary conditions needed to solve equation 10 is that there is no heat flux in the center line of the wine bottles, then:

$$\frac{\partial T_r}{\partial \mathrm{r}}\bigg|_{r=0} = 0 \tag{11}$$

On the outer surface of the wine bottles there is combined heat transfer of radiation and natural convection. Then the boundary condition is:

$$k.\frac{\partial T_r}{\partial r}\bigg|_{r=r_o} = (h_{rad,b} + h_{conv,b}).(T_a - T_r|_{r=r_o}) \tag{12}$$

The radiation heat transfer coefficient is calculated by the following equation:

$$h_{rad,b} = \varepsilon_g.\sigma.(T_a^2 + T_r^2|_{r=r_o}).(T_a + T_r|_{r=r_o}) \tag{13}$$

where ε_g is the glass emissivity. The convection heat transfer coefficient is determined by the correlation of Churchill & Chu (1975) for horizontal cylinders is presented by Incropera and DeWitt (2006):

$$h_{conv,b} = \frac{k_a}{D}\left\{0.60 + \frac{0.387.Ra_D^{1/6}}{\left[1 + (0.559/Pr)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \tag{14}$$

where D is the outer diameter of the wine bottles and the Rayleigh number R_D is defined as the following equation:

$$Ra_D = \frac{g.\beta.(T_r|_{r=r_o} - T_a).D^3}{\alpha.\nu} \tag{15}$$

The air inside the cooled environment is in thermal contact with the wine bottles, the cabinet walls and the cold heat exchanger. In this model, an integral energy balance chosen to describe the heat transfer in the air. Thus:

$$\rho . c_p . V_a . \frac{\partial T_a}{\partial t} = \dot{Q}_w + \dot{Q}_B - \dot{Q}_C \qquad (16)$$

On the right side of the equation 16 the terms are, respectively, the heat transfer rate through the walls, the heat transfer from the wine bottles, the cooling capacity provided by the magnetic system via the cold heat exchanger. V_a is the air volume inside the cabinet. Equation 16 is also integrated in time and using the full implicit formulation. Thus:

$$\frac{\rho . c_p . V_{ar}}{\Delta t}(T_a - T_a^o) = (h_{rad,w} + h_{conv,w}) . A_w . (T_x|_{x=0} - T_a) + 2\pi . r_o . H_b . N_b . (h_{rad,b} + h_{conv,b}) . (T_a - T_r|_{r=r_o}) - \dot{Q}_c \qquad (17)$$

The cooling capacity ($\dot{Q}_C$) is obtained in terms of the air temperature according to the AMR operating conditions and the UA of the heat exchanger used in the cabinet. This value is obtained by a previous mathematical model developed by Calomeno *et al.* (2016) to evaluate the thermal performance of an AMR using real heat exchangers.

To evaluate the performance of a cabinet using a magnetic cooling device, some simulations were carried out to analyze the cooling of the wine bottles and the air inside the cabinet for specific operating conditions of the AMR. The heat transfer fluid is a mixture of water and ethylene-glycol (20%vv.). The fluid flow was simulated as a square wave (on-off), considering that the cold blow and the hot blow time are identical. Different sizes of the cold heat exchanger were simulated by varying its overall thermal conductance. The operating conditions of the magnetic system are summarized in Table 1.

Table 1 – AMR dimensions and operating conditions

Regenerator length [mm]	100	Porosity of AMR matrix [-]	0.36
Regenerator diameter [mm]	22.22	Mass flow rate per regenerator [kg/h]	50 and 100
Number of regenerators	8	Operating frequency [Hz]	1
Magnetic profile	Cosine wave form	External environment temperature [K]	300
Magnetic field (Min. / Max.) [T]	0 / 1.5	Sphere diameter [μm]	550
Overall thermal conductance (hot heat exchanger) [W/K]	15	Overall thermal conductance (cold heat exchanger) [W/K]	5 – 15

The cabinet's liner is made of high impact polystyrene and the thermal insulation is composed of expanded polyurethane. The wine cooler door has been assumed to have the same structure of the other walls. The wine bottles are composed by soda lime glass and are fill with water. The thermophysical properties of those materials can be seen in Table 2 and the most important dimensions of the cabinet are summarized in Table 3.

Table 2 – Thermophysical properties of the materials

Material	High Impact Polystyrene¹	Expanded Polyurethane²	Water¹	Soda Lime Glass¹
Density (ρ) [kg/m³]	1050	29.4	999.2	2440
Specific heat (c_p) [J/kg.K]	1350	1200	4184	840
Thermal conductivity (k) [W/m.K]	0.160	0.02	0.5767	0.937
Emissivity (ε) [-]	0.90	-	-	0.92

¹ Properties obtained from software EES (Klein, 2017).
¹ Properties obtained from Hermes (2000).

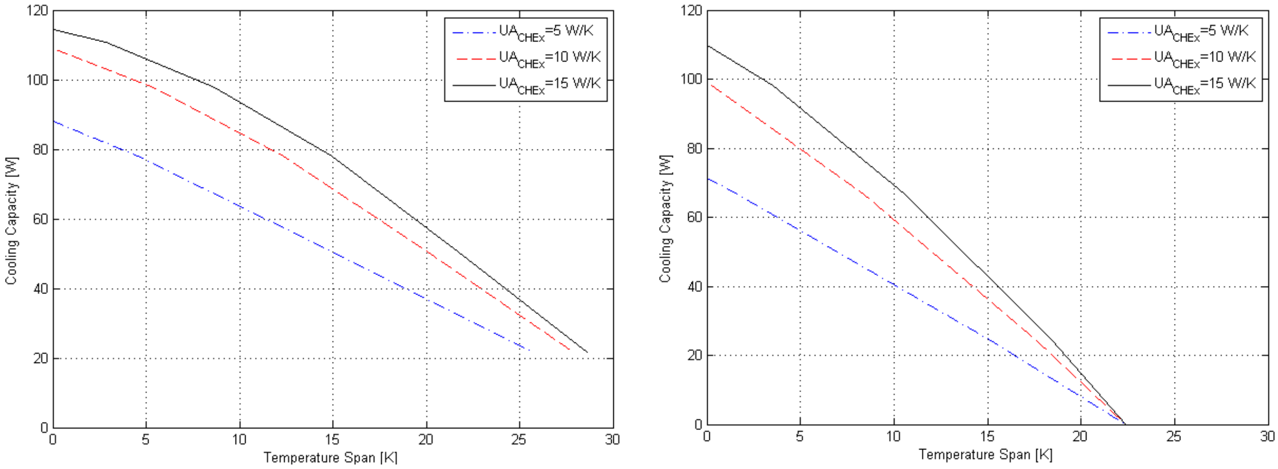
Table 3 – Cabinet dimensions

Cooled environment length [m]	0.43
Cooled environment width [m]	0.42
Cooled environment height [m]	0.83
Liner thickness [m]	0.0175
Thermal insulation thickness [m]	0.020

The energy equations for the air, walls and wine bottles are solved simultaneously. All components of the cabinet are assumed to be at the external environment temperature in the beginning and the time step is $\Delta t = 1$ s. The convergence criterion is that the difference of the wine bottle temperature and a predetermined prescribed temperature is lower then $\epsilon = 10^{-3}$ K.

3. RESULTS AND DISCUSSION

Firstly, the impact of the overall thermal conductance of the cold heat exchanger is evaluated in terms of the cooling capacity produced as a function of the temperature span. The temperature span is the temperature difference between the external environment (T_H) and the cooled environment (T_C). Simulations were carried out by varying the overall thermal conductance of the cold heat exchanger (UA_{CHEx}) in the values of 5 W/K, 10 W/K and 15 W/K while the hot heat exchanger has its overall thermal conductance (UA_{HHEx}) fixed at 15 W/K. In these simulations, the mass flow rate per regenerator was set at 50 kg/h and 100 kg/h. The cooling capacity profile produced in terms of the temperature span for these cases can be observed on Figure 4.



(a) (b)

Figure 4. Performance curves of an AMR with a hot heat exchanger with thermal conductance of 15 W/K and cold heat exchanger thermal conductance variating from 5 W/K to 15 W/K for different mass flow rate per regenerator of (a) 50 kg/h and (b) 100 kg/h.

From Figure 4, it can be observed that the cooling capacity provided by the AMR increases as the overall thermal conductance of the cold heat exchanger increases. As its overall thermal conductance varies, the same cooling capacity can be obtained, but at different temperature spans. At a given time, one half of the regenerators is being magnetized, while the other half is being demagnetized, thus the total mass flow rate at the cold and hot heat exchanger is approximately the same pump mass flow rate. Thus, for the case presented in Figure 4(a), the pump mass flow rate is 200 kg/h, while for the case in Figure 4(b) it is 400 kg/h. Although case (a) uses a lower mass flow rate, it has a better performance in terms of the cooling capacity for all values of the overall thermal conductance of the cold heat exchanger. This performance may be explained by the high utilization of the regenerators and a low effectiveness of the heat exchangers which impair the AMR performance. Also, the use of higher values of the mass flow rate clearly limits the temperature span range which cooling capacity still can be produced. Case (a) can obtain a cooling capacity of 20 W for temperature spans higher than 25 K, but higher values of the temperature span are not desirable for wine cooler appliances, because for lower cooled environment temperature the wine might freeze and loses its quality.

Also, an analysis about the pull-down time required for cooling down the wine bottles from 300 K to 281 K was made using the operating conditions shown in Figure 4(a) for the same three cases of the cold heat exchanger. This analysis uses the mathematical model described in Section 2 to estimate the cooling capacity obtained by the cold heat exchanger during the pull-down, the temperatures of the air and at the center of the wine bottle. The results can be observed, respectively, in Figures 5, 6 and 7.

The total time required to cool down the wine bottles was of 28.08 h for the case with $UA_{CHEx} = 5\ \text{W/K}$, 18.35 h for the case with $UA_{CHEx} = 10\ \text{W/K}$ and 16.00 h for the case with $UA_{CHEx} = 15\ \text{W/K}$. Thus, higher values of the overall cooling capacity of the cold heat exchanger produces higher cooling capacity and reduces the pull-down time.

The thermal size of the cold heat exchanger has a great influence on the AMR thermal performance operating in the wine cooler cabinet. The higher the overall thermal conductance, the lower the pull-down time, but the heat exchanger will occupy a larger volume.

When observing the temperature profile at the center of the wine bottle, during the first 30 minutes, the bottles are practically isothermal. After the first half hour, the temperature of the bottle starts to decrease faster for the higher overall thermal conductance of the cold heat exchanger. Comparing the cases with $UA_{CHEx} = 5\ W/K$ and $UA_{CHEx} = 10\ W/K$ there is a decrease in the pull-down time of about 10 hours and when comparing the cases with $UA_{CHEx} = 10\ W/K$ and $UA_{CHEx} = 15\ W/K$ the decrease in the pull-down time is about 2 hours. These results indicate that a heat exchanger with a reasonable overall thermal conductance which can provide a pull-down time of 16 to 19 hours and can be used for cooling wine bottles while not consuming much energy. Also, usually the wine coolers are used when the wine will not be consumed in a short period of time.

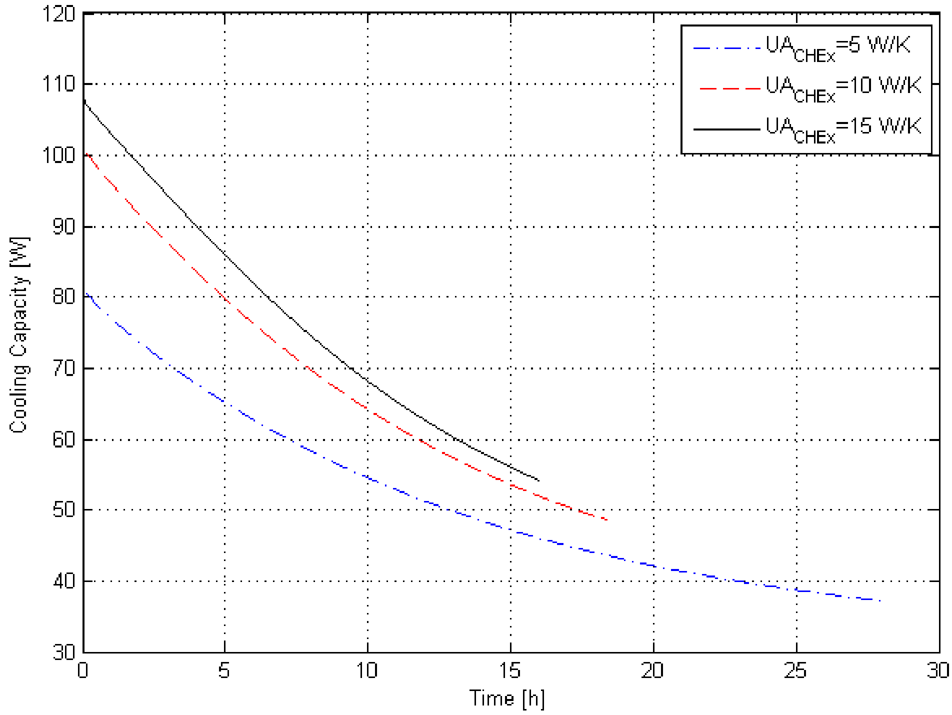


Figure 5. Cooling Capacity profile along the cooling time. AMR operating mass flow rate per regenerator of 50 kg/h with UA_{CHEx} variating from 5 to 15 W/K.

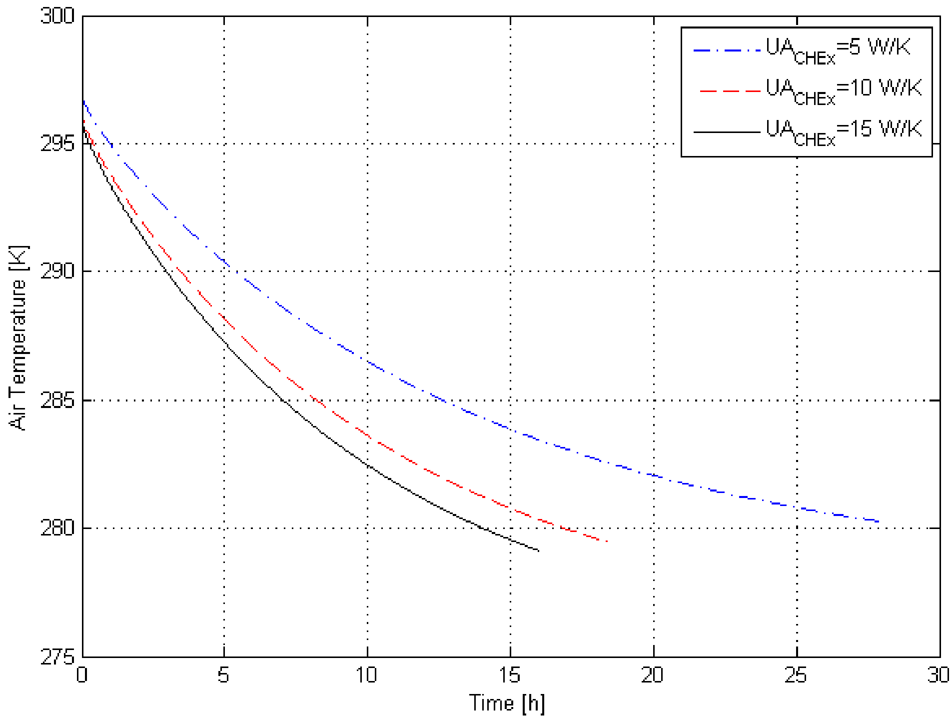


Figure 6. Temperature of the air inside the cabinet. AMR operating mass flow rate per regenerator of 50 kg/h with UA_{CHEx} variating from 5 to 15 W/K.

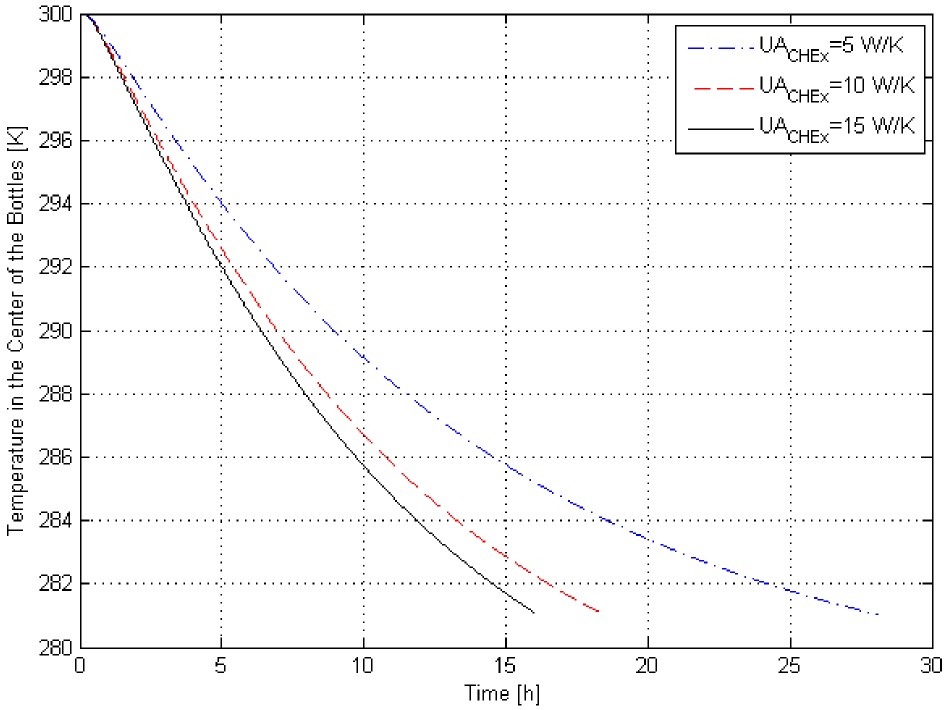


Figure 7. Temperature in the center of the wine bottles. AMR operating mass flow rate per regenerator of 50 kg/h with UA_{CHEx} variating from 5 to 15 W/K.

4. CONCLUSIONS

The thermal performance of a wine storage cabinet operating with a magnetic refrigeration system and real heat exchangers has been evaluated using a mathematical model based on the energy equation of all the analyzed domains. The pull-down time required to cool the wine bottles from 300 K to 281 K was estimated for different operating conditions of the AMR and for different sizes of the cold heat exchanger. The results indicate that for a good performance of the AMR and the cabinet, it is important that the AMR should provide higher values of the cooling capacity for a range of 10 to 20 K of the temperature span, which can be improved by using cold heat exchanger with higher values of the overall thermal conductance. An analysis of the thermal performance of the wine cooler operating with an AMR in terms of the operating conditions of the system with the heat exchangers, the coefficient of performance and the second law efficiency will be subject of further studies.

5. ACKNOWLEDGEMENTS

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