ENERGY AND EXERGETIC ANALYSIS OF AN AIR-CONDITIONING SYSTEM FOR A FRUIT WAREHOUSE
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ANALIZA ENERGETICĂ ȘI EXERGETICĂ A UNUI SISTEM DE CLIMATIZARE PENTRU UN DEPOZIT DE FRUCTE

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ABSTRACT
In the paper, a thermodynamic analysis of an air conditioning system necessary to store a quantity of 500 kg of apricots at a temperature of 10°C and a humidity of 90% in a fruit warehouse was aimed at. The storage facility is cooled by means of an air handling unit (AHU). The energetic and exergetic analysis was carried out on the air conditioning system that treats a mixture of fresh air and recirculated air for various fractions of fresh air ranging from 0 to 100%, thus being able to perform a finer analysis of the operation of the installation under different working conditions. Based on the exergy analysis, the exergy loss was estimated for each device, depending on their destination and operating mode. Numerical and graphical results are presented related to the necessary mass flow rates of the air, the refrigeration load, the necessary treatment and evolution of humid air in the system, as well as the exergy loss in the main components of the air conditioning system. This theoretical study was carried out to obtain quantitative information that will lead to a better understanding of the air conditioning irreversibility process and their distribution amongst the system component and minimizing them for optimal air conditioning cycle.

REZUMAT
Această lucrare prezintă analiza termodinamică a unui sistem de aer condiționat necesar pentru păstrarea într-un depozit de fructe a unei cantități de 500 kg de caise la o temperatură de 10°C și o umiditate de 90%. Spațiul de stocare este răcit prin intermediul unei centrale de tratare a aerului (CTA). A fost efectuată analiza energetică și exergetică a sistemului de aer condiționat care tratează un amestec de aer proaspăt și aer recirculat pentru diverse fracțiuni de aer proaspăt variind de la 0 la 100%, putându-se realiza astfel o analiză mai fină a funcționării instalației în diferite condiții de lucru. Pe baza analizei exergetice au fost estimate pierderile de exergie pentru fiecare aparat, în funcție de destinație și de regimul de funcționare. Sunt prezentate rezultate grafice și numerice legate de debitele de aer necesare, puterea frigorifică necesară, procedeele de tratare ale aerului umed în sistemul de aer condiționat precum și pierderile de exergie din procesele principale. Acest studiu teoretic s-a realizat pentru a obține informații cantitative care să conducă la o mai bună înțelegere a ireversibilitatilor aerului condiționat și a distribuției acestora între componentele sistemului și la minimizarea acestora pentru un ciclu optim de climatizare.

INTRODUCTION
Damage of the fruits and vegetables during their picking, handling, packaging, storage and sale, can be generated in part by too high temperatures that cause over-ripening, decay, as well as loss of quality and nutritional value. Other damages can be caused on the contrary, by a temperature that is too low or they can be of a mechanical nature, due to the carelessness of those who handle the products. The cold processing technologies of fruits and vegetables, the appropriate packaging, make it possible for them to arrive in a fresh state in markets that can be geographically very far from the place where they were picked. As a result, the variables to be controlled and monitored in the refrigeration of fruits and vegetables are temperature, relative humidity and ventilation.

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Fresh fruits and vegetables are alive and require fresh air to allow breathing. Without ventilation during storage, respiratory gases can accumulate and damage products. Ventilation must be done in such a way as to have control over temperature, humidity and energy consumption. All vegetables and fruits are alive and continue to breathe and metabolize. The metabolic process produces carbon dioxide and metabolic heat. There is a clear relationship between the heat of metabolism and the evolution of CO_2, and for every 2.96 watts of metabolic heat, 1 g of is generated. Respiration may also release water vapor and ethylene. The level of respiration depends on the type of product and the temperature. It also varies depending on the maturity of the product (*Heap et al., 2003*).

Cold storage is a refrigerated warehouse where fruits and vegetables can be stored for a long period of time without any deterioration of the products stored. Optimum storage temperature and relative humidity are necessary to maximize shelf life and maintain the quality of the products stored. The results showed that the shelf life of each vegetable can be increased by 50–60% when they are kept inside the cold storage prototype (*Parida et al., 2020*).

Food storage conditions, including storage time, temperature and light can have an influence on the retention of phytochemicals. In 2018, a study was conducted to investigate the effect of processing methods (freezing, preservation and drying) and storage periods on the physical-chemical and antioxidant properties of three varieties of apricots. It can be concluded from the study that canning and freezing can preserve apricot pulp for 12 months and significantly retain bioactive compounds (*Wani et. al., 2018*). A review was also made related to the thermo-physical properties of fruits which are vital for the study and optimization of post-harvest handling processes. However, the data available in the literature are not always consistent and should not be used directly. It is crucial to examine the accuracy and reliability of property data. The purpose of this review is to show gaps in fruit property data, with a focus on those properties that are important during post-harvest handling (*Mukama et al., 2020*).

Another research presented the applicability of wind and solar energies for cooling a fruit warehouse in the hot and dry Yazd region of Iran. The studied fruit warehouse had an area of 4240 m^2, resulting in a storage capacity of approximately 1000 tons. For this purpose, the heat gain of the warehouse is determined, and the obtained cooling load is then used to examine the solar and wind energy to power a conventional warehouse system (*Mostafaeipour et al., 2020*).

As part of the management of fruit refrigeration systems, it is essential to control the atmosphere with adequate ventilation. The influence of basket stacking arrangements and fan position on air flow in an industrial apple storage room was studied in this sense. Three fans situated above the vertical gaps were more effective for the ventilation of bins than four fans above the bin rows. A wide distance between stack and wall opposite to fans increased flow uniformity (*Praeger et al., 2021*). In another research, a k-ε simulation model was developed to evaluate the airflow pattern in a refrigerated container loaded with bananas, where k is the kinetic turbulent energy (k) and (ε) is the turbulent dissipation rate. The simulation results predict the location of hot spots. Moreover, it was found that the cooling distribution is improved by changing the layout of the pallets in the container, the so-called chimney layout (*Issa et al., 2014*).

The results of another study demonstrated the importance of room humidification for preserving fruit quality. Storage of fruits in a room at 95% RH (relative humidity) minimized weight loss and best preserved fruit color, firmness, size and chemical quality attributes of pomegranate. Therefore, fruit water losses can be reduced by proper packaging, control temperature and humidification of the storage room (*Mukama et al, 2019*). Another study was taken in three varieties of apricots grown in the same farm and placed in storage after sorting, the preliminary phase of storage. Storing was carried out in three different conditions: ambient temperature (+20…+22°C), refrigerated spaces (+10…+12°C) and cold conditions (+3...+5°C). The duration of preservation and level of weight (mass) and decay losses, as well as the evolution of some chemical components during storage were determined. The best results were obtained for storage in refrigerated conditions (*Vintila et al., 2015*).

Starting from the fact that storage requires a climate controlled by temperature and humidity, it is particularly important to use an efficient installation from an energy point of view. Research studies such as those presented below have been developed in this direction. So, in another study an energy and exergy analysis is made of moist air, it is used the psychometrics charts. A Visual Basic program is used to generate psychometrics charts. These charts are used to analyze the air thermodynamic behavior, considering the environmental variations, pressure, temperature and relative humidity. Also, the available energy in the cooling processes at constant enthalpy, humidification at constant temperature and heating with constant relative humidity is analyzed (*Pereyra et al., 2011*).

The following general conclusions could be drawn in another research paper: exergy efficiency is more rational than energy efficiency; exergy analysis is more helpful than energy analysis for locating and evaluating available energy saving potentials, identifying opportunities for improvements in system design and establishing cost effective system maintenance programs (*Chengqin et al., 2002*). Also, in another paper, researchers have discussed thermodynamic analysis of various psychrometric processes using the concept of exergy. They noticed that a decrease in mass flow rate of fresh air (second incoming stream) in the case of adiabatic mixing decreases the second-law efficiency of the process (*Bilal et al., 2003*).

MATERIALS AND METHODS

System description

The following analysis refers to a particular case of a fruit and vegetable cold store composed of a refrigeration chamber designed to refrigerate a quantity of products, in our case 0.5 tons of apricots. The room is provided with an access door, it is insulated with polyurethane. The external dimensions of the refrigerated warehouse are 4 m long, 3 m wide and 2.5 m high. The ambient air is at 38°C at 30% RH- relative humidity, the internal air is at 10°C at 90% RH. The walls, roof and floor are all insulated with 0.08 m polyurethane with a overall heat transfer coefficient k value of 0.308W/m^{2}K. The ground temperature is 15°C. The purpose of the air handling unit (AHU) in Fig.1 is to remove the stale air from inside the space and introduce fresh air into it. Several modules can be present inside a central unit depending on how the air is desired to be when it is introduced into the room. The modules included in the AHU are: filter module, fans, mixing module and cooling module, i.e. BRS (surface cooling battery). With the help of fans, the air is extracted from the room and the air from outside is introduced, and the amount of air introduced or extracted from the room can also be regulated, with the help of the mixing chamber, the stale air can be recirculated if it is not toxic. The cooling module helps to cool the air with water or refrigerant (depending on its supply system) before it is introduced into the room. The air coming from the mixing house will go into a cooling battery system (BRS). The air is cooled through the refrigerant medium until it reaches a lower temperature of the mixed air resulting in convection and conduction heat transfer. The entire process is shown in the Mollier (h, x) diagram (enthalpy-humidity ratio diagram) in Fig.2 and Fig.3.

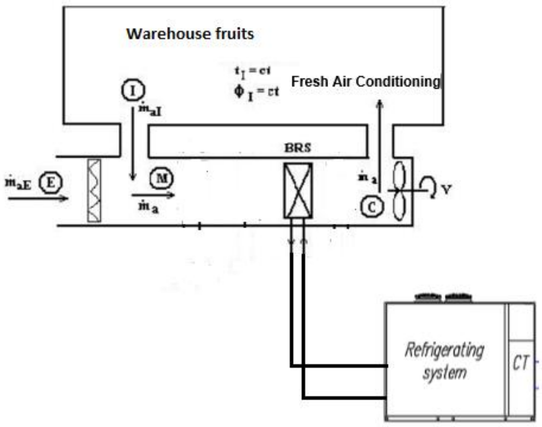

Fig. 1 –Schematic representation of the air-handling unit (AHU) serving the Warehouse

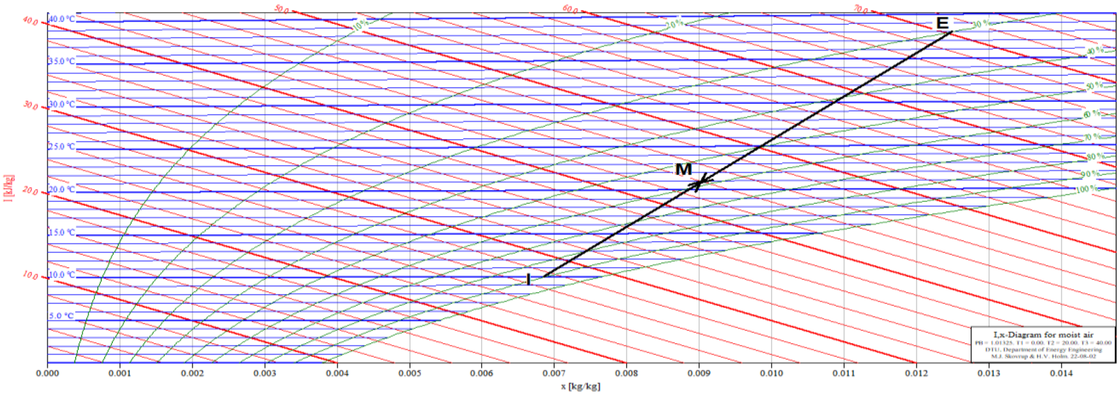

Fig. 2 – Process of mixing with 40% fresh air

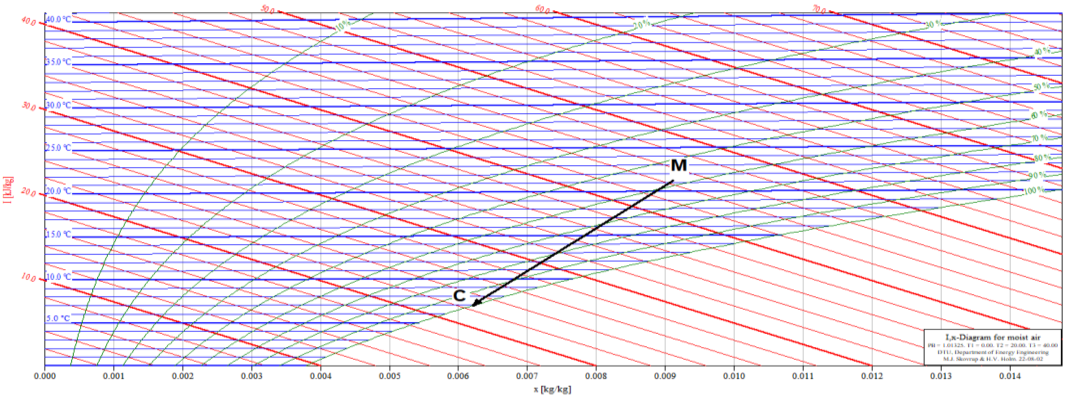

Fig. 3 – Cooling process in BRS

Mathematical model

Cooling and humidity load analysis

Calculation of thermal loads of warehouses adapted for cooling in summer is important for the accuracy of the design and the appropriate choice of equipment for the adaptation of air and air handling units in order to meet the requirements of operation.

The cold requirement for the vegetable and fruits store represents the total heat flow that must be extracted from the cold storage whose temperature must be maintained at 10°C and a relative humidity of 90% *(Pop et al., 2016; Cooling Load Calculation, 2017*):

$$\dot{Q}_{nec} = \dot{Q}_{walls} + \dot{Q}_{floor} + \dot{Q}_{lightening} + \dot{Q}_{products} [kW] \tag{1}$$

- Heat flow through walls and roof**:**

$$\dot{Q}_{walls} = \sum_{i=5} \dot{Q}_{wall,i} [kW] \tag{2}$$

where $\dot{Q}_{wall,i}$ is the heat flow coming in through wall *i*

$$\dot{Q}_{wall} = k_{wall} \cdot S_{wall} \cdot \Delta T_{wall} \tag{3}$$

where:

k_{wall} - is the overall heat exchange coefficient, $\left[\frac{W}{m^2K}\right]$;

S_{wall} - is the heat exchange area, $\left[m^2\right]$;

ΔT_{wall} is temperature difference over wall, $\left[K\right]$.

The overall heat exchange coefficient is expressed as:

$$k_{wall} = \frac{1}{\frac{1}{\alpha_{\text{int}}} + \sum_{i=1}^{n} \frac{\delta_i}{\lambda_i} + \frac{1}{\alpha_{ext}}} \tag{4}$$

where:

α_{int} is the indoor convection coefficient, $\left[\frac{W}{m^2K}\right]$;

α_{ext} is the outdoor convection coefficient, $\left[\frac{W}{m^2K}\right]$;

δ wall thickness, $\left[m\right]$;

λ thermal conductivity coefficient, $\left[\frac{W}{mK}\right]$.

- The heat load of the floor:

$$\dot{Q}_{Floor} = k_{Floor} \cdot S_{Floor} \cdot \Delta T_{Floor} \tag{5}$$

$$k_{floor} = \frac{1}{\frac{1}{\alpha_{\text{int}}} + \sum_{i=1}^{n} \frac{\delta_i}{\lambda_i}} \tag{6}$$

- The heat generated by the lighting devices is:

$$\dot{Q}_{lighting} = n \cdot \dot{Q}_{lamp} \tag{7}$$

where n is number of lamps within the cold room.

- The heat load from stored products:

$$\dot{Q}_{\mathrm{Pr}oducts} = \dot{m} \cdot c \cdot \Delta T \tag{8}$$

c is the specific heat capacity of product (kJ/kgK)

$\dot{m}$ = the mass flow of products (kg/s)

ΔT = temperature difference between the temperature of the products and the temperature within the store (K)

The heat load of the walls, roof and floor of the room dedicated to storing apricots at the temperature of 10 °C is shown in Table 1.

Table 1

Heat load of the walls, roof, floor dedicated to storing apricots

Title	A	k	ΔT	$\dot{Q}$
	[m^2]	[W/m^{2}K]	[K]	[W]
North Wall	10	0.3081	28	86.26
South Wall	10	0.3081	28	86.26
East Wall	7.5	0.3081	28	64.7
West Wall	7,5	0.3081	28	64.7
Roof	12	0.3081	28	103.5
Floor	12	0.2816	5	16.89
Total				**422.3**

To calculate the total cooling load, we will just sum all the values calculated:

$$\dot{Q}_{nec} = \dot{Q}_{walls} + \dot{Q}_{floor} + \dot{Q}_{lightening} + \dot{Q}_{products} \cong 1kW$$

Further the system is analyzed both from energy and exergy points of view. The analysis was performed with the Mathcad software program.

Energy analysis

The thermodynamic properties at each point should be specified in order to apply the energy and exergy analysis. Calculation of the physical quantities, which determine the states of the moist air, is performed using the equations given below.

Specific enthalpy, *h(t,x),* of the moist air is defined by the following equation:

$$h(t,x) = 1.004 \cdot t + x \cdot (2500 + 1.88 \cdot t) \left[{kJ}/{kgdryair} \right] \tag{9}$$

t is temperature of the moist air [°C].

The vapour pressure of water in saturated moist air (in bar) - $p_S(t)$ is expressed by the following equation (*ASHRAE Handbook, Fundamentals, 2009*):

$$A1_t(t) = 5.865604 \cdot 10^{-3} - 3.142305 \cdot 10^{3} \cdot \left(\frac{1}{t + 273.15} - \frac{1}{373.15} \right) \tag{10}$$

$$A2_t(t) = 8.2 \cdot \log\left(\frac{373.15}{t + 273.15} \right) - 2.4804 \cdot 10^{-3} - (373.15 - t - 273.15) \tag{11}$$

$$p_S(t) = 10^{A1_t(t) + A2_t(t)} \tag{12}$$

Correlation between relative humidity and humidity ratio x, can be expressed:

$$x(t,\varphi) = 0.622\ \frac{\varphi \cdot p_S(t)}{p_B - \varphi \cdot p_S(t)} \quad \text{[kg water vapor/kg dry air]} \tag{13}$$

The state parameters of the outdoor air, state E (from STAS 6648/2-82 page 3 for the month of July, Bucharest, Romania) are:

t_E=38°C x_E=12.5·10^{-3} kg vapours/kg dry air

The state parameters of the indoor air (fruit storage conditions) state I:

t_I=10°C and relative humidity φ_I=0.90

The state parameters of air conditioning with state C

An air conditioning temperature of t_C = 7°C is chosen, and then, using the angular direction of the CI process, in which the air conditioning evolves towards to state I taking over $\dot{Q}_T = \dot{Q}_{nec}$ and $\dot{m}_w$, respectively:

$$\varepsilon_{CI} = \frac{\dot{Q}_{nec}}{\dot{m}_w} \tag{14}$$

The moisture content of state C can be calculated using the formula:

$$x_C = \frac{h_I - 1.004 \cdot t_C - \varepsilon_{CI} \cdot x_I}{2500 + 1.84 \cdot t_C - \varepsilon_{CI}} \tag{15}$$

The moisture released by the products (apricots) is: $\dot{m}_w = 0.00014$ kg um /s (*Wells, 1962*).

Calculation of air flow rates

Total flow rate of dry air $\dot{m}_a$ [kg dry air/s] from the moist air is:

$$\dot{m}_a = \frac{\dot{Q}_{nec}}{h_I - h_C} \tag{16}$$

As it can be seen in Fig.1, the total flow rate is made up of recirculated air *I* and outdoor fresh air *E*, which are combined in the mixing room. The fresh air flow is the variable in relation to which the thermodynamic analysis of the system is performed.

During the analysis, the fresh air flow will be varied from 0 to 100% of the total air flow.

Recirculated air flow $\dot{m}_{aI}$ [kg dry air/s]:

$$\dot{m}_{aI} = \dot{m}_a - \dot{m}_{aE} \tag{17}$$

State of the mixed air M

Applying the mass balance as well as the energy balance, the thermodynamic properties at the outlet of the mixing chamber are obtained (parameters of mixed air under *M* state). The position and state parameters of point *M* depend on the ratio between fresh air flow and recirculated air flow.

$$h_M = \frac{\dot{m}_{aI} \cdot h_I + \dot{m}_{aE} \cdot h_E}{\dot{m}_{aE} + \dot{m}_{aI}} \tag{18}$$

$$x_M = \frac{\dot{m}_{aI} \cdot x_I + \dot{m}_{aE} \cdot x_E}{\dot{m}_{aE} + \dot{m}_{aI}} \tag{19}$$

setting the moist air treatment process from M to C.

In the particular case studied, the line MC intersects the saturation curve φ=1. Cooling in BRS (surface cooling battery) is chosen, and this battery must have a wall temperature:

$$t_p = t_C - 5grd = 2°C \tag{20}$$

The refrigerating power of the refrigerating installation that ensures the cooling of the moist air, so that the flow air $\dot{m}_a$ and the *M* state reach the *C* state, results from the heat flow that must be extracted from the air:

$$\dot{Q}_{MC} = \dot{m}_a \cdot \left(h_M - h_C\right) \qquad \dot{Q}_{BRS} = \dot{Q}_{MC} \tag{21}$$

Condensate flow rate discharged from the system is:

$$\dot{m}_{condens} = \dot{m}_a \cdot \left(x_M - x_C\right) \quad \text{[kg/s]} \tag{22}$$

The results are shown in table 2.

Table 2

The states of the moist air

Condition	**x (**humidity ratio)	**h** (specific enthalpy)	**t** (temperature)	**φ** (relative humidity)
	[kg vap/kg dry air]	[kJ/kg]	[ºC]	[-]
Outdoor Air (E)	$12.5\cdot10^{-3}$	70.295	38	0,301
Air (C)	$6.856\cdot10^{-3}$	22.622	7	0,999
Mixed Air (M)		44.50	21.27	
Room Air (I)	$6.856\cdot10^{-3}$	27.309	10	0,90

Exergy analysis

The **exergetic analysis** method allows determining the cause and magnitude of inefficiencies in an energy process or system. Exergy represents the maximum fraction of a form of unordered energy that can be transformed into mechanical work, in the ideal, reversible thermodynamic evolution of the system up to a state of reference called "dead state", chosen most of the time as the state of thermodynamic equilibrium with the ambient environment (*Dobrovicescu, 2000*). Therefore, the value assigned to the exergy of a material flow depends on the correct selection of the reference medium. The fundamental dead state is that state that would be reached if each constituent of the substance were reduced to complete equilibrium with the stable components in the environment.

The dead state temperature for the exergetic analysis is considered to be equal to that of the ambient environment:

E: $po=p_B$=1 bar; $to=t_E$=38ºC; $xo=x_E$=0.0125 kg vap/kg dry air; $\varphi_o=\varphi_E$=0.301.

According to *Bejan et al., (1988),* the exergy carried by a current of moist air contains the follwing components:

Thermal exergy of moist air, ex_{th} is:

$$ex_{th}=\left(c_{pa}+x\cdot c_{pv}\right)\left[T-T_o-T_o\cdot\ln\left(\frac{T}{T_o}\right)\right] \tag{23}$$

Mechanical exergy of moist air, ex_{mec}:

$$ex_{mec}=\left(1+1.608\cdot x\right)R_a\cdot T_o\cdot\ln\left(\frac{p}{p_o}\right) \tag{24}$$

Chemical exergy of moist air, ex_{ch}:

$$ex_{ch}=R_a\cdot T_o\left[\left(1+1.608\cdot x\right)\cdot\ln\left(\frac{1+1.608\cdot x_o}{1+1.608\cdot x}\right)+1.608x\cdot\ln\left(\frac{x}{x_o}\right)\right] \tag{25}$$

with:

c_{pa} = 1.004 kJ/kg.K mass specific heat of dry air;

c_{pv} = 1.88 kJ/kg.K mass specific heat of water vapours;

R_a =0.287 kJ/kg.K mass specific constant of dry air.

Total exergy of moist air will be:

$$ex=ex_{th}+ex_{mec}+ex_{ch} \quad \text{[kJ/kg dry air]} \tag{26}$$

Exergy flow $\dot{E}x$ transported by the moist air will be calculated with the formula:

$$\dot{E}x=\dot{m}_a\cdot ex \quad \text{[kW]} \tag{27}$$

In the exergy analysis of the air conditioning systems, the exergy flow carried by the humidity, ex_w given off by the moist air must also be taken into consideration. It can be calculated using the formula:

$$ex_w\cong-R_v\cdot T_o\cdot\ln\left(\varphi_o\right) \quad \text{[kJ/kg water]} \tag{28}$$

$$\dot{E}x_w=\dot{m}_w\cdot ex_w \quad \text{[kW]} \tag{29}$$

R_v = 0.461 kJ/kgK is the mass constant of the water vapours.

The exergetic balance equation is given by the relation:

$$\frac{d\left(E-T_o S\right)}{d\tau}=\sum Ex\left(\dot{Q}_i\right)-\dot{W}-\left(\dot{E}x_{out}-\dot{E}x_{in}\right)-\dot{E}x_D \tag{30}$$

where:

E is the total energy of the system, [J];

T_0 - reference temperature in exergetic analysis, usually ambient temperature, [K]

S – the system entropy, [J/K];

τ - the time, [s].

$\sum \dot{E}x\left(\dot{Q}_i\right)=\sum\left(1-\frac{T_0}{T_i}\right)\dot{Q}_i$ represents the exergy of the heat flows $\dot{Q}_i$ [W] exchanged by the system at the temperature level of the outer sources T_i [K];

$\dot{W}$ is the mechanical power produced or consumed [W];

$\dot{E}x_{in}$ and $\dot{E}x_{out}$ are the exergy flows carried by thermal agents (the working fluids circulating in the system), [W].

In thermodynamics, the exergy destruction represents a major inefficiency and a quantity to be minimized when the overall system efficiency should be maximized (*Tsatsaronis et al., 2011*).

According to Guy-Stodola theorem, the exergy destruction rate is computed by:

$$\dot{E}x_D=T_0\cdot\dot{S}_{gen} \tag{31}$$

where $\dot{E}x_D$ is the exergy destruction rate, [W].

Typical irreversible processes in installations (heat transfer at finite temperature difference, flow with friction and pressure loss, throttling, mixing) lead to a high rate of entropy generation, directly proportional to the destruction of exergy in each device and in the system as a whole. In conclusion, the exergy destruction rate in a device or in a plant is the best indicator of the operating efficiency.

Generally, the flow of exergy destroyed in a device can be calculated using the formula:

$$\dot{E}x_D=\sum\dot{E}x_{in}-\sum\dot{E}x_{out} \tag{32}$$

and the exergy efficiency is:

$$\eta_{ex}=\frac{\sum\dot{E}x_{out}}{\sum\dot{E}x_{in}}=1-\frac{\dot{E}x_D}{\sum\dot{E}x_{in}} \tag{33}$$

RESULTS

Next, the influence of the ratio between the fraction of fresh air and that of recirculated air is presented both on the energy consumption of the installation and on the exergy destroyed in the system.

The obtained results are graphically illustrated in Fig. 4…9. The variable in relation to which the graphic representations are made is the ratio between the flow of fresh, outside air and the total flow of air circulated in the system.

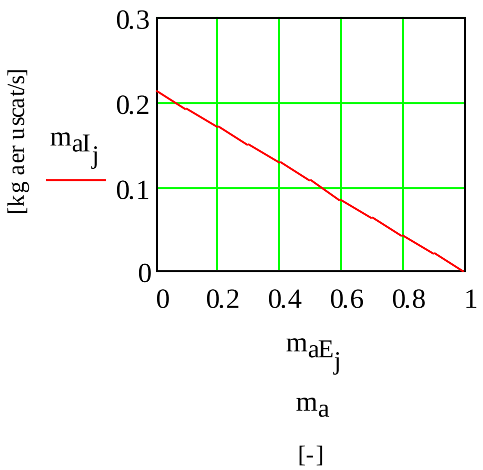

Fig. 4 – Recirculated air flow

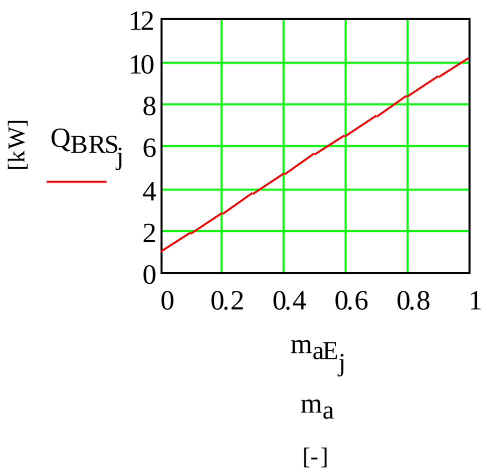

Fig. 5 – Refrigerating power necessary to cool

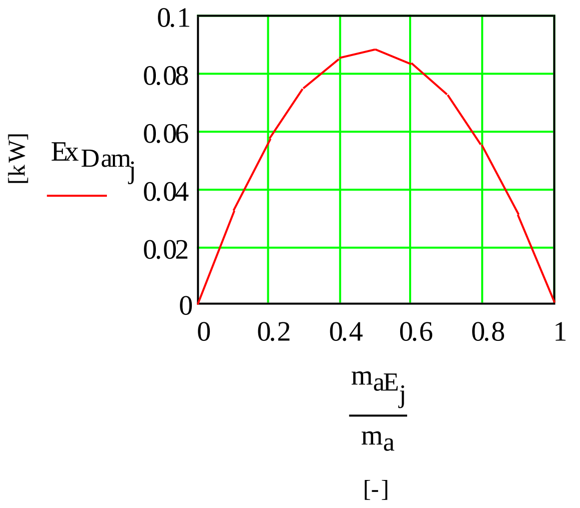

Fig. 6 – Exergy destroyed in the mixing process

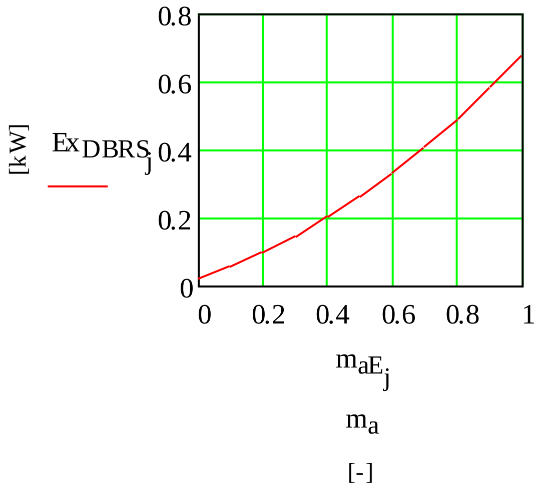

Fig. 7 – Exergy destroyed in the process of cooling the moist air

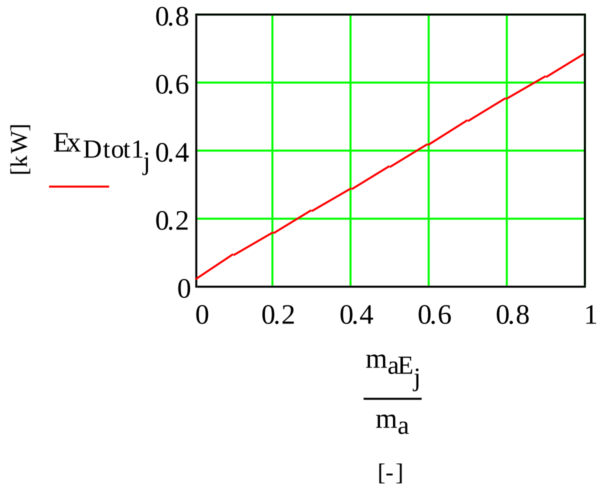

Fig. 8 – Total destroyed exergy - mixing and treating in BRS

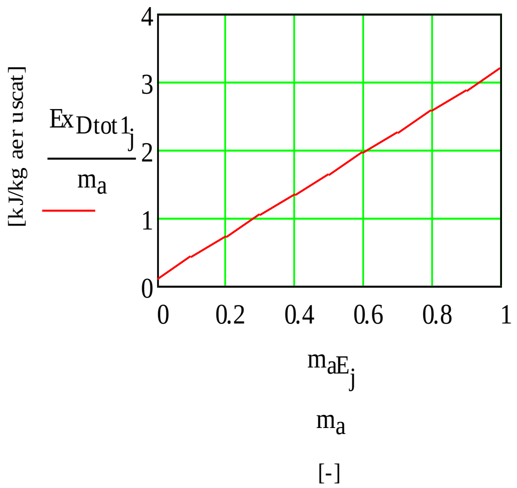

Fig. 9 - Total specific destroyed exergy - mixing and treating in BRS

According to Fig. 5, the refrigeration power required to cool the moist air increases with the increase in the ratio between the fresh air and the total air introduced into the system. The fact that the refrigerating power increases from 1 kW to 10 kW is due to the fact that it introduces fresh air, the higher the flow of fresh air, the higher the required refrigerating power and also the exergy destruction according to Fig.7.

Through mixing, it can be seen that the high exergy destruction is reached when the proportion of air is 50%, then as the fresh air flow increases, it starts to decrease according to Fig.6.

CONCLUSIONS

The energetic and exergetic analysis applied to the concrete case of a summer air conditioning system for a refrigerated storage facility for fruit-vegetables was presented. The changes introduced by the variation of the proportion of fresh air in the total flow of air circulated by the system in the humid air treatment variant with only BRS were shown.

The results indicate that the exergy destruction during the mixing process between fresh and recirculated air reaches a maximum value when the two flow rates are equal (Fig.6). The value of the exergy destroyed in mixing is, however, much lower than that in the cooling battery (Fig.7). It is also noticed that the global intensity of exergy destruction increases with the increase of the fresh recirculated air fraction, the plant also requiring a higher energy consumption to cool the moist air in BRS (Fig.8.and Fig.9).

Likewise, the refrigeration power in the BRS starts to decrease when the proportion of one of the components, either fresh air or recirculated air, decreases. The higher the proportion of fresh air, the higher the temperature of the mixture and therefore the higher the exergy destruction.

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