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OPTIMAL DESIGN AND PERFORMANCE ANALYSIS OF A REFRIGERATION SYSTEM FOR A FRUIT STORAGE

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ABSTRACT
Analyses performed during the design of a fruit storage refrigeration system are presented. Simulation model describing all year round behavior of the cold storage and the refrigeration system has been established and simulations have been performed in order to evaluate optimal insulation thickness of the cold room and energy costs. Costs for energy, equipment and insulation, as well as costs caused by weight loss of the product have been introduced into a cost evaluation model and the complex which consists of storage and refrigeration system has been optimized. Comparison of energy consumption and investment costs between direct expansion and brine refrigeration system has been performed. Direct refrigeration system with higher COP and thus lower energy consumption has been found as the feasible choice. Analyses have also been aimed to the air - cooler design. Oversized evaporators have been chosen in order to maintain high air humidity within the storage and thus decrease the weight loss of stored fruits.
Keywords: fruit storage, thermal insulation, direct evaporation systems, indirect brine systems, cost analysis.

1. INTRODUCTION
Water loss is one of the greatest sources of losses in fruit and vegetables distribution chain. Inside cold stores with lower air humidity and increased air movement, the product dehydration increases. Unpacked fruits and vegetables are especially vulnerable to dehydration. Dehydration reduces the salable amount and when it exceeds of 3 to 6% the marked loss of product quality appears [1].
The objective of the research has been to find optimal design of the cold storage and refrigeration system for fruit distribution storages, considering total costs for the building insulation, refrigeration equipment, consumed energy and the loss of the value of stored fruits caused by dehydration.
Water diffusion $\dot{W}$ (kg/h) from the wet surface A_{pv} (m^2) of goods to the air within the cold storage can be expressed as

$$\dot{W} = \overline{\sigma}\, A_{pv}(x_v - x_m)\,, \tag{1}$$

where $\overline{\sigma}$ (kg/m^2 h) is an average diffusion coefficient, and x_v (kg/kg) and x_m (kg/kg) average air moistures defined on Fig. 1b [2].

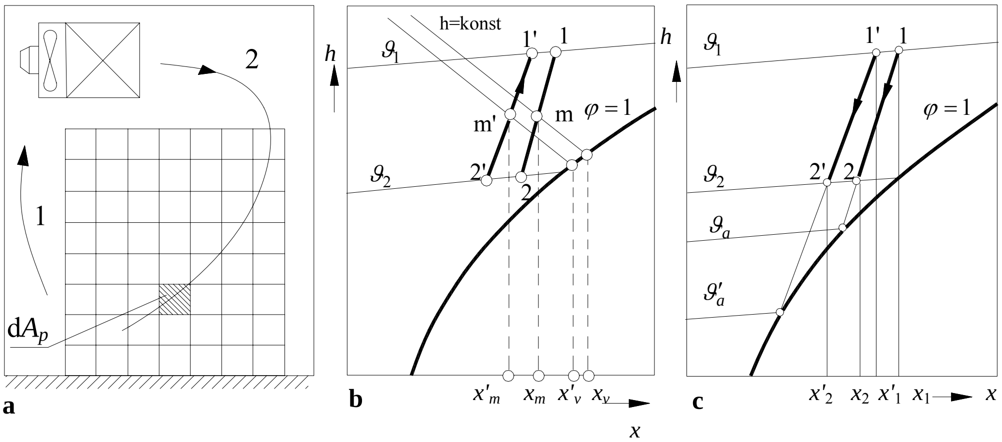

Figure 1. Influence of evaporation temperature and heat load structure on the air moisture in a cold storage a) sketch of a storage, b) air state change, c) air state change in the air cooler [2]

Total heat $\dot{Q}_t$ (W) removed in the air cooler consists of sensible $\dot{Q}_s$ (W) and latent $\dot{Q}_l$ (W) part and it can be expressed as

$$\dot{Q}_t = \dot{Q}_s + \dot{Q}_l = \dot{M}c_p(\vartheta_1 - \vartheta_2) + \dot{M}r(x_1 - x_2), \tag{2}$$

where $\dot{M}$ (kg/s) is the air mass flow rate, c_p (J/kg K) specific heat capacity of the air, r (J/kg) latent heat of evaporation for water, ϑ_1, ϑ_2 (°C) temperatures and x_1, x_2 (kg/kg) water contents of the air at evaporator inlet and outlet respectively. The change of the air enthalpy h (J/kg) within the cooler can be represented with a line (1-2) in the diagram at Fig. 1 c with inclination defined as follows:

$$dh/dx = (\dot{Q}_s + \dot{Q}_l)/\dot{W} = \dot{Q}_t/\dot{W}\,. \tag{3}$$

The air humidity in a cold store increases with lower temperature difference between the air cooler surface and the air (Fig. 1 b and c). The heat rate $\dot{Q}_t$ (W) of the air cooler can be expressed as

$$\dot{Q}_t = kA\Delta\vartheta_m \tag{4}$$

where k (W/m^2 K) is the overall heat transfer coefficient, A (m^2) heat transfer area and $\Delta\vartheta_m$ (K) mean logarithmic temperature difference. It is obvious that the higher heat transfer area of the air cooler will result in lower temperature differences for the same heat exchanged. That will result in increased surface temperature of the air cooler and thus increased air humidity within the cold store. Increased air humidity results in lower dehydration. The higher is the share of sensible heat in the total removed heat, the higher is the inclination of the line 1-2, and consequently the air humidity in the cold store is lower. For that reason the sensible heat gain must be maintained as low as possible by increased thermal insulation of the storage. Cost analysis in a particular case study has been used to determine optimal level of insulation, as well as optimal refrigeration system design including the determination of optimal air - coolers design.

2. SIMULATION OF THE SYSTEM BEHAVIOUR

Refrigeration load of a single storage includes heat transmission through the storage envelope, product load brought into a storage with new product, the heat generated by the product (fruits and vegetables), internal loads generated by refrigeration and process equipment, forklifts, people and lighting as well as infiltration air load caused by air - exchange and infiltration. All heat gains $\sum Q_i$ (J) tend to increase the temperature within the cold storage.

The case study of a refrigeration system for a cold storage with total floor area 708 m^2 (volume approximately 2700 m^3) has been analyzed. Simulation has been performed for the heat load calculation during the test reference year for location Rijeka (Croatia), using adjustable timetables for stored goods inlet and outlet, staff operation, lighting etc. Simulations resulted in known refrigeration load profile during the entire year. The heat removed from the cold room by operation of refrigeration system within a time step Δt is Q_0 (J). A simple energy balance can result in temperature change of the refrigerated room comprising walls with mass M_w (kg) and heat capacity c_w (J/kg K) and the food with mass M_f (kg) and heat capacity c_f (J/kg K):

$$\vartheta_i^{t+\Delta t} = \vartheta_i^t + \frac{\sum Q_i - Q_0}{M_w c_w + M_f c_f} . \tag{5}$$

The mass and the heat capacity of the air within the room can be neglected. Temperatures ϑ_i^t and $\vartheta_i^{t+\Delta t}$ have been assumed as temperatures of the walls, air and stored goods within the considered i-th room in times t and $t+\Delta t$. Expression (5) does not account for the time necessary for heat exchange between the air and stored goods or storage walls, but that drawback has been overcome by assuming empirical correlations describing time - dependence of temperature change of walls and food.

In order to define the cooling capacity of the refrigeration system $\dot{Q}_0$ (W), which depends on the air inlet temperatures $\vartheta_{a,e,i}$ (°C) for the air cooler and $\vartheta_{a,c,i}$ (°C) for the air cooled condenser, the design of entire refrigeration system has been done. With known table data of compressor capacities for different evaporation and condensing temperatures ϑ_e (°C) and ϑ_c (°C), correlations have been established for the refrigeration load $\dot{Q}_0$ (W), power consumption $\dot{P}_e$ (W) and heat rate rejected through the condenser to the surroundings $\dot{Q}_c$ (W). Those correlations describe considered loads by polynomials of the *j*-th order with coefficient sets A_j, B_j and C_j and exponents k and l selected for best curve fit:

$$\dot{Q}_0 = \sum_{j=0}^{n} A_j \vartheta_0^k \vartheta_c^l \; ; \qquad \dot{Q}_c = \sum_{j=0}^{n} B_j \vartheta_0^k \vartheta_c^l \; ; \qquad \dot{P}_e = \sum_{j=0}^{n} C_j \vartheta_0^k \vartheta_c^l \; . \tag{6}$$

Using correlations from Eqn. 6, and design data for condensers and evaporators, correlations have been established in an iterative process for $\dot{Q}_0$ (W), $\dot{P}_e$ (W) and $\dot{Q}_c$ (W) as functions of room temperature $\vartheta_{a,e,i}$ and surrounding temperature $\vartheta_{a,c,i}$. Such a function is suitable for analysis of the refrigeration system behavior during the operation. Those correlations have also been presented in a similar form as previous:

$$\dot{Q}_0 = \sum_{j=0}^{n} A_j \vartheta_{a,e,i}^k \vartheta_{a,c,i}^l \; ; \qquad \dot{Q}_c = \sum_{j=0}^{n} B_j \vartheta_{a,e,i}^k \vartheta_{a,c,i}^l \; ; \qquad \dot{P}_e = \sum_{j=0}^{n} C_j \vartheta_{a,e,i}^k \vartheta_{a,c,i}^l \; . \tag{7}$$

Temperature controller action was simulated by setting upper and lower temperatures of the air within the storage which influence operation of the refrigeration system. The operation time within one hour (which was the selected time step) has been determined as the quotient of energy consumption and refrigeration system load at disposal. Electric energy consumption has been achieved as the product of operation time and power for the refrigeration system operation. All year round simulation resulted in energy consumption which was one of important input parameters for the cost analysis.

3. COST ANALYSIS

Dehydration related market losses on one side, investment costs depending on insulation thickness, air cooler size and system choice on the other side as well as refrigeration system annual operation costs evaluated by simulations, have been considered in order to decide on optimal insulation thickness. The thickness of the thermal insulation has been varied between 80, 150 and 200 mm, resulting in different transmission loads and thus different refrigeration system capacities and investment prices, as well as in different annual energy costs. Specific prices included into a consideration were 267 - 293 €/m^3 for insulation (lower specific values are related to insulation with higher thickness), system specific price was 530 €/kW of refrigeration load, average price of electric energy was 0,08 €/kWh, and consideration time was 40 years for insulation and 20 years for refrigeration system. Influence of dehydration was evaluated for different insulation thicknesses and resulted in considerable differences in dehydration costs. Average price of 0,53 €/kg for stored goods has been assumed.

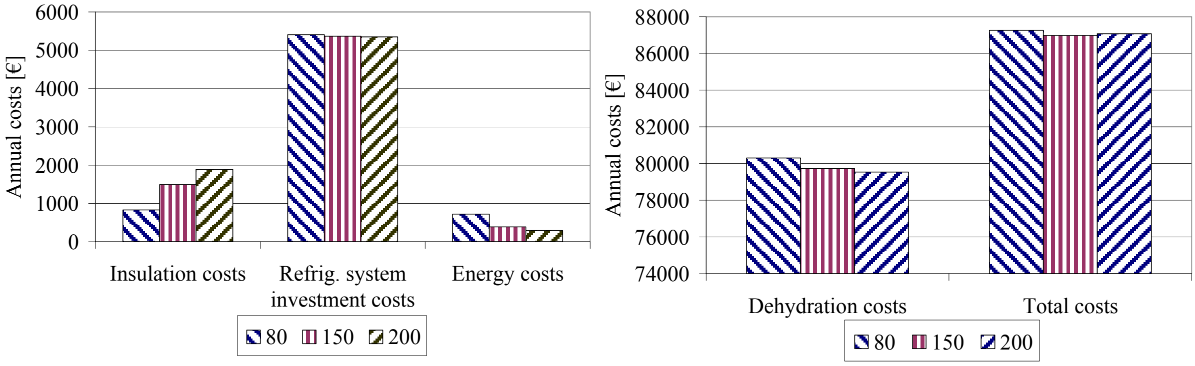

Figure 2. Structure of costs for different insulation thicknesses (80, 150 and 200 mm)

It has been found that the insulation thickness of 150 mm, which is a bit higher than usual recommendations for optimal thickness [2], represents the one which results in lowest total costs.

4. CHOICE OF THE REFRIGERATION SYSTEM

In direct expansion refrigeration system, the evaporator is placed in the refrigerated storage and the heat is transferred directly from air to the refrigerant which evaporates in the evaporator. Advantages of such system are higher COP (which is a quotient between produced energy for cooling and consumed electric energy for system operation) but there is higher probability of refrigerant leakage and the refrigerant charge is higher than for the brine system. In the brine system the heat transfer fluid circulates between the evaporator (fluid cooler) and air coolers. The most important advantages are low refrigerant charge and lower possibility of leakage. System operates with lower COP, which is caused by higher temperature differences between the refrigerant and the air. Both systems have been designed in a particular case study and simulations have been performed in order to evaluate annual costs for energy. COP has been found to be about 32% lower in the case of the brine system compared with a direct expansion system. Investment costs for the brine system were 26% higher, and annual costs comprising total energy and investment were in favor of the direct expansion system. System reliability has also found to be higher for the case of a direct expansion system.
Evaporators have been designed with mean logarithmic temperature differences between 4 and 5 K which are lower than usually and result in increased evaporation temperatures and thus increased relative humidities of the air in storages.

5. CONCLUSION

Using all year round simulations of system behavior and introducing values for energy, equipment, insulation and product costs into cost evaluation model, it has been found that optimal storage insulation thickness is higher than usually recommended, reaching 150 mm for considered storages at temperatures between 2°C and 12°C. Cost analyses resulted also in the choice of direct evaporation refrigeration system as the feasible and more reliable one. Evaporators with increased heat transfer surfaces and thus lower temperature differences between the air and the refrigerant can achieve higher evaporation temperatures and higher air moistures within cold storages which is important because the weight loss of stored goods has been found to be one of highest costs in a fruit storage refrigeration.

6. REFERENCES

[1] ASHRAE (2006): 2006 Ashrae Handbook - Refrigeration, ASHRAE, Atlanta GA
[2] Vujić, et. al. (2000): Rashladni uređaji, Mašinski fakultet Beograd