Research on heat transfer characteristics and cold trap capacity of a water catcher during vacuum pre-cooling

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Abstract: Effect of vacuum pre-cooling process on apples was a complex process of heat and mass transfers. The research is based on the physical properties of apples and their heat and mass transfer mechanisms during vacuum pre-cooling. As for the heat transfer characteristics of a water catcher in vacuum pre-cooling, the research studied the heat transfer mechanism and calculated the cold trap capacity by experimental means， and it cold trap capacity were evaluated to supply references for future research into the practical applications of such vacuum pre-cooling techniques. The results provide a theoretical basis for exploring better pre-cooling process conditions and the design of water catchers. The experimental results show that, when the wall temperature of the water catcher is -5°C, the optimal cold trap capacity is about 90.72g and the required cooling capacity is 210.13W in the vacuum pre-cooling of 201.9g of apples.

1. Introduction

Based on the theory that water absorbs the latent heat of vaporization under vacuum, vacuum pre-cooling is designed to improve the storage quality of fruit and vegetables by rapidly cooling them, effectively eliminating field heat and restraining respiratory heat. Simultaneously, owing to heat beings absorbed during the evaporation of water, a cooling effect is achieved without an external heat source in vacuum pre-cooling. Hence, vacuum pre-cooling is characterized by a high cooling rate, uniform temperature distribution, cleanliness, and an absence of pollution. Meanwhile, appropriate pre-cooling over time can maintain the quality of fruit and vegetables to the greatest extent and reduce decay losses [1-7]. In recent years, many scholars have focused on the physical properties of materials after pre-cooling, the physico-chemical changes in vacuum storage, the heat and mass transfer characteristics of fruit and vegetables in storage, etc [8-18]. Many research findings related to vacuum cooling have been published: Wang et al. studied a vacuum model of a meat product using the finite element method in 2002 and they also simulated the cooling process by computational fluid dynamics [19] [20]. Yan et al. (2006)studied the matching of the water catcher and the vacuum pump determined the pre-cooling quality during vacuum cooling processes [21]. Tao et al. (2006) reported the effects of vacuum cooling treatment and storage conditions on lipid oxidation, superoxide anion generation, superoxide dismutase, catalase, peroxidase and polyphenoloxidase in mushrooms [22]. Schmidt et al.

(2010) studied aimed to determine the effects of vacuum cooling on the enzymatic antioxidant system of cherry and inhibition of surface-borne bacteria during storage. By analyzing the literature, it was found that scholars mainly focus on the introduction of vacuum pre-cooling process conditions, the design of pre-cooling equipment, the control of weight-loss ratio of fruit and vegetables in pre-cooling, the physical properties of materials after pre-cooling, etc [23]. However, some scholars paid attention to the structural optimization of the water catcher used in vacuum pre-cooling, and the optimal matching of the vacuum pump, the heat transfer characteristics of water catchers and the theoretical calculation of their cold trap capacity are rarely studied.

Apples collected from Lingbao City apple plantation, Henan Province, China were taken as the study object. According to the requirements of the test, the authors divided and screened the samples. On the basis of the heat transfer characteristics of the water catcher used in vacuum pre-cooling, this research explored the heat transfer mechanism and calculated the cold trap capacity to provide a theoretical basis for the design of the water catcher.

2. The heat transfer characteristics of the water catcher

The water catcher, also called a cold trough or cold trap, is a critical component of any vacuum pre-cooling installation. The air, which contains moisture and is removed by a vacuum pump, is trapped by the water catcher. At 610 Pa and 0 °C, the air volume increases 210,000-fold. Under these conditions, the vacuum pump power consumption becomes significant, especially given its low efficiency. Therefore, the performance of the water catcher is key to the operation of the refrigerating plant.

During vacuum pre-cooling, the water vapor flowing out from the vacuum chamber condenses into liquid water when it meets the condensing surface of the water catcher, and then attaches to the condenser tube. At the same time, the non-condensable gas produced by the system will be removed by the vacuum pump. In the water catcher used for cooling air, there is single phase heat transfer in the air and also a phase change heat transfer because of the change of the moisture content therein. In the heat transfer process, the heat transfer coefficient and heat transfer quantity can be expressed as,

$$\phi=\frac{A_0\Delta t_m}{\left(\frac{1}{\alpha_i}+\gamma_i\right)\left(\frac{A_0}{A_i}\right)+\left(\frac{\delta}{\lambda}\right)\left(\frac{A_0}{\overline{A}}\right)+\left(\frac{\overline{\delta}}{\lambda_l}\right)+\frac{1}{\varepsilon\varepsilon_e\alpha_d}} \tag{1}$$

where ϕ is the amount of heat transferred, W; α_i and α_d are the heat transfer coefficients of the refrigerant in the tube and of air under dry cooling, $W/(m^2.℃)$; γ_i is the fouling coefficient for the inner side of the tube $0.0004m\cdot□$ / W; A_0, A_i, and $\overline{A}$ respectively represent the heat transfer areas of the outer and inner sides of the tube and the average area, m^2 ; δ and $\overline{\delta}$ denote the thickness of the heat exchange tube and the average water film thickness, m; λ and λ_l are the thermal conductivities of the tubes and water films, $W/(m^2.℃)$; ε_d denotes the reduced convective heat transfer coefficient (air-side)which was set to 0.8; Δt_m is the mean temperature difference, °C; and ε is the dehumidifying coefficient.

$$\varepsilon=1+\frac{\gamma_0+C_{p,v}t_m-C_w t_w}{C_{p,m}}\cdot\frac{d_m-d_w}{1000\left(t_m-t_w\right)} \tag{2}$$

where, γ_0 is the latent heat when the water vapor condenses into water at 0 °C; $C_{p,v}$ is the specific heat capacity of water vapor under constant pressure; C_w is the specific heat capacity of water; $C_{p,m}$ is the specific heat capacity of moist air under constant pressure; d_w is the moisture content of saturated wet steam when the wall temperature is , °C; d_m is the average moisture content of a single tube; t_w is the wall temperature, °C; and t_m is the average temperature of the moist air flowing through a single tube, °C.

The heat transfer coefficient in tubes may be calculated using,

$$\alpha_i = Cq^{0.67} M_l^{-0.5} P_r \left(-\lg P_r\right)^{-0.55} \tag{3}$$

Where, C is the calculated coefficient; q is the heat flux density, W/m^2; and M_l and P_r are respectively the relative molecular weight and reduced pressure of the liquid.

Condensation heat transfer plays a major role in the water catching process, while convective heat transfer contributes only slightly. Owing to the moist air flow being slow, the convective heat transfer coefficient out of the tubes was replaced with a condensation heat transfer coefficient for these theoretical calculations. The average surface heat transfer coefficient for the surface film condensation of the horizontal circular tube is,

$$h_h = 0.725 \left[\frac{\lambda_1^3 \gamma_0 \rho_1 \left(\rho_1 - \rho_v\right) g}{\mu_1 \left(T_s - T_w\right) d_0} \right]^{0.25} \tag{4}$$

Where, ρ_1 is the density of water; ρ_v is the density of water vapor; T_s is the saturated temperature corresponding to the prevailing vapor pressure. There into, the characteristic length of the vertical tube is the tube length, represented by L, and the characteristic length of the horizontal circular tube is its outside diameter d_0. Thus, the relationship between the condensation heat transfer coefficients h_v and h_h is,

$$\frac{h_h}{h_v} = 0.77 \left(L / d_0 \right)^{0.25} \tag{5}$$

The operation of the water catcher is a pressure reduction process. The convective heat transfer is less than the condensation heat transfer. Besides, the water film on the outside surface of the tube has a significant influence on the convective heat transfer of moist air. At a 25 °C initial temperature in the vacuum chamber and 60% relative humidity, the partial pressure of saturated water vapor is given by,

$$InP_{q,b} = \frac{c_1}{T_0} + c_2 + c_3 T_0 + c_4 T_0^2 + c_5 T_0^3 + c_6 InT_0 \tag{6}$$

The water vapor pressure is, $P_q = \varphi . P_{q,b}$ (7)

where, $P_{q,b}$ is the partial pressure of saturated water vapor, Pa; P_q is the water vapor pressure, Pa; c_1, c_2, c_3, c_4, c_5, and c_6 are constants and are respectively -5800.2206, 1.3914993, -0.04860239, 0.41764768 × 10^{-4}, -0.14452093 × 10^{-7}, and 6.5459673; φ is set to a value of 0.026; and T_0 is the temperature of the moist air, °C.

3. Experimental work

This research analyzed the physical properties of apples and the heat and mass transfer mechanisms during vacuum pre-cooling. Based on the heat transfer characteristics of the water catcher used in vacuum pre-cooling, the heat transfer mechanism of the water catcher and its cold trap capacity were evaluated to supply references for future research into the practical applications of such vacuum pre-cooling techniques.

3.1. The experimental device

The vacuum pre-cooling testing machine, VFD-2000 (Shanghai Vacuum Preservation Equipment Co., Ltd) was used in this experiment. It contains a refrigeration system, a vacuum system, and a data acquisition and control system (see Fig. 1a). The water catcher, as a key part of the vacuum pre-cooling device, was effectively the evaporator for the circulation of the system and reduced the load on the vacuum pump by condensing the water vapor. In addition, the vacuum system was equipped with a 2XZ-4 rotary vane vacuum pump operating at 4 L/s and at rate of rotation of1,400 rpm. Four thermocouples were used for temperature measurement and a pressure sensor was used to record the state of the applied vacuum. Meanwhile, the data acquisition and control system, comprising a programmable logic controller (PLC), host computer, and electrical system was used to collect, store, and conduct each parametric test in real-time.

a. Test device diagram for refrigeration system. b. Practicality diagram of vacuum chamber.

Figure 1. Experimental table for the vacuum pre-cooling test machine

3.2. Materials and methods

The apples used in this experiment were collected from Lingbao City Science and Technology Pilot Park, Henan Province, China. After being divided and screened, 201.9 g of samples were weighed (on an ACS-3 electronic balance whose maximum load was 3,000 g, with a sensitivity of 0.1 g, and Grade III accuracy) and placed on the shelf in the vacuum chamber. During the experiment, the pressure sensor was placed in the vacuum pump to record the change in pressure in the vacuum pump. The time-span for internal data collection was 30 s and MS-Excel® was used to post-process the raw data. The thermocouples were fixed at the geometrical centre and the margins of the apples as required (see Fig. 1b). Thereafter, the door of the vacuum chamber was locked and the apparatus was turned on and all data recorded.

The compressor and vacuum pump formed the main energy dissipating components in the vacuum pre-cooling process. During the experiment, the author adjusted the load on the compressor and switched the vacuum pump on and off in response to the observed changes in temperature and pressure. After the pre-cooling experiment, the samples were taken out and weighed. The test was then repeated with modified control parameters.

3.3. Analysis of results

By experimental observation, data recording, and analysis, the main factors affecting the cold trap capacity of a water catcher were found to have been: the initial temperature of the vacuum chamber, relative humidity, the temperature and water vapor partial pressure in the water catcher, the wall temperature of the condenser, *etc.* The atmospheric pressure, water vapor partial pressure and relative humidity in the vacuum chamber, and their changes over time, are shown in Figures 2 and 3.

It can be seen from Figure 2 that the gas pressure and dry air pressure in the vacuum chamber gradually decreased during the pre-cooling phase. Figure 3 shows changes in relative humidity in the vacuum chamber over time. Under the initial conditions, the pressure in the vacuum chamber was on the verge of atmospheric pressure. However, the pressure decreased after using the vacuum pump and its rate of change gradually decelerated in the medium-term and stabilized thereafter. At the flash vaporization point (at about 6.2 minutes and 2,338 Pa), the dry air pressure was zero and the volume of water evaporated from the apples gradually decreased. Whereas if the vacuum pump started to work before flash vaporization, almost no water was evaporated from the apples and only a little air in the vacuum chamber will be trapped by the water catcher so that the cold trap capacity becomes the water vapor remaining in the vacuum chamber. At that stage, the pressure in the vacuum chamber gradually decreased and the moisture content of the air decreased rapidly. However, when it reached the flash vaporization point, the moisture content of the apples was practically all evaporated. Figure 2 and Figure 3 show the sharp rising trend in the water vapor pressure and relative humidity in the vacuum chamber. Subsequently, the amount of evaporated water and the water vapor partial pressure were decreased. Therefore, the pre-cooling pressure should be applied on the basis of the different physical properties of different materials to obtain the best pre-cooling results in actual vacuum pre-cooling.

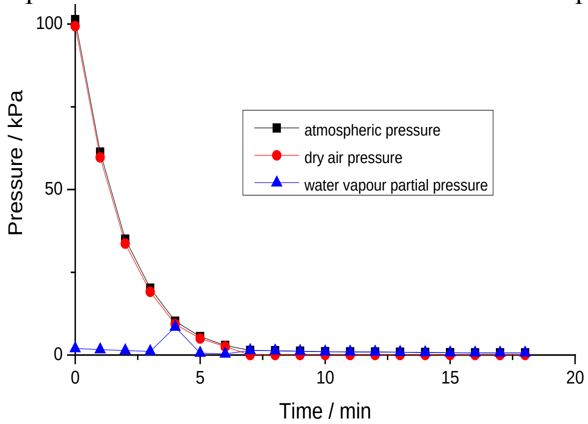


Figure 2. Changes in pressure in the vacuum chamber over time

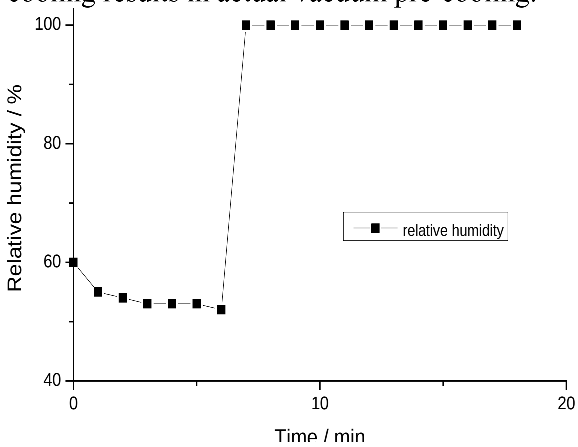


Figure 3. Changes in relative humidity in the vacuum chamber over time

3.4. The calculation of cold trap capacity

The normal operation of a vacuum pre-cooling device mainly depends on the performance of the water catcher. Based on steam condensation theory, and the aforementioned analysis, the cold trap capacity can be calculated by using the formula below. The changes of the cold trap capacity, the quantum of heat transfer from the apples, and the density of the moist air in the vacuum chamber during pre-cooling are shown in Figures 4.The quantities of capture water,

$$G = m_1 + m_2 = \phi / \gamma_0 \quad (8)$$

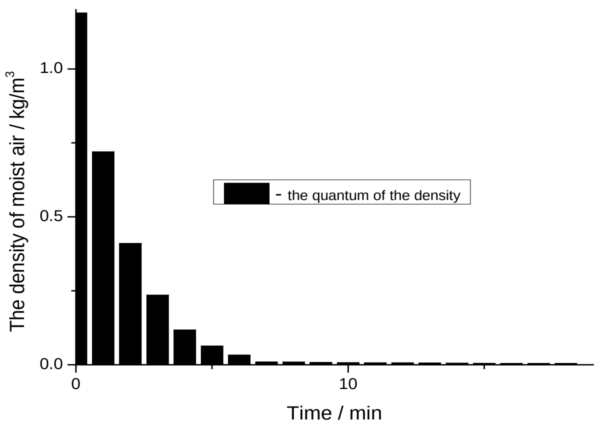
In formula, G is the quantities of capture water, γ_0 is latent heat, kJ/kg,Where $m_1 = \sum_{i=1}^{5} m_i$ is steam consumption, g, and $m_2 = \sum_{i=6}^{18} m_i$ is water consumption in evaporation, g.

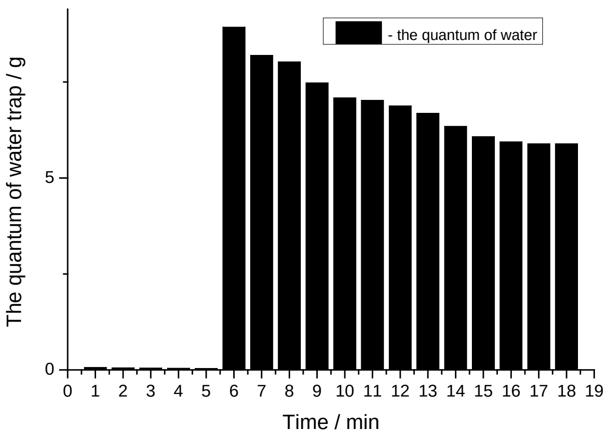
$$q = G\frac{\gamma_0}{t} \quad (9)$$

$$K_{kn} = \frac{kT}{\sqrt{2}\pi L d^2 p} = \frac{M_a}{\mathrm{Re}} \quad (10)$$

where, q is the cooling capacity, W, t is time, min, K_{kn} is Knudsen Number, L is the pipe diameter, m, M_a is Mach number, Re is Reynolds number, γ is fouling factor, $m \cdot □$ / W, k is Coefficient parameters, d is diameter, m, p is Vacuum pressure, Pa, and T is temperature,℃.

From Figures 4, it can be seen that the density of moist air decreased during pre-cooling. This situation arose because the pressure in the vacuum chamber was close to atmospheric pressure at the start of the pre-cooling phase, and under no influence from the vacuum pump, while the pressure reduced gradually during evacuation of the chamber. At the flash vaporization point, the dry air pressure was zero and both the amount of evaporated water from the apples, and the density, decreased gradually to cause the density of the moist air to continue to decrease while undergoing slight fluctuations. From the aforementioned calculation, at a wall temperature of -5 °C in the water catcher, the cold trap capacity of the vacuum pre-cooling system was 90.72g, and 210.13 W of cooling capacity was required for 201.9g of apples. The computing method and result provided a theoretical basis for the design of such water catchers.





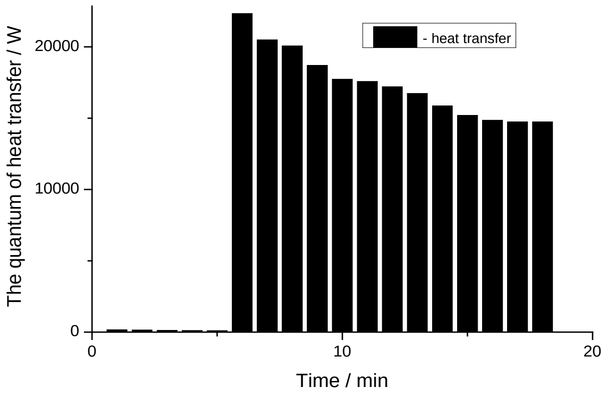


Figure. 4 Changes in the density of moist air, the cold trap capacity and the amount of heat transferred in the vacuum chamber during pre-cooling.

4. Conclusions

In this paper, the heat transfer characteristics of water-catcher used in vacuum pre-cooling system were introduced. Water-catcher was a key component, vacuum extracted with water vapor in the air and subject to capture water, trapping the water down, otherwise it would reduce the pump's performance affects the entire vacuum pre-cooling device performance. The analyses and the results could be useful for operating vacuum pre-cooling.

At first, the gas pressure, dry air pressure, and moist air density in the vacuum chamber decreased during pre-cooling. At the beginning of the pre-cooling phase, the pressure in the vacuum chamber was close to atmospheric pressure. After switching on the vacuum pump, the pressure decreased rapidly and then stabilized (albeit with minor fluctuations therein) in the medium-term, and remained stable thereafter. Besides, the cold trap capacity was related to the surface temperature, evaporating temperature, and cooling capacity of the water catcher. The results suggested that at a wall temperature of -5°C, the optimal cold trap capacity was approximately 90.72gwith 210.13W of cooling capacity required during the vacuum pre-cooling of201.9g of apples.

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