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Comparison of the Thermodynamically Analysis of Vacuum Cooling Method with the Experimental Model

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Abstract: In this study, results of thermodynamically analysis were tested and compared, implementing the vacuum cooling of lettuce. According to the findings of the trial and results of the thermodynamically analysis, it is possible to determine the weight loss within an error of 2.12%, close to the other parameters to be used in the design of vacuum precooling system, such as temperature, pressure, enthalpy and entropy on specified points using the mathematical model prepared from thermodynamically equations. Moreover, the fact that the power need, the most important parameter in the design of the system, could be determined with a minimal error (0.162%) reveals that the thermodynamically analysis could be used in the design of a vacuum precooling system.

Key words: Vacuum cooling, thermodynamic analysis, lettuce, weight loss, precooling, food

INTRODUCTION

The vacuum cooling method is achieved by the rapid evaporation of water in the fruits and vegetables under vacuum (Wang and Wen, 2002a, b). For changing water from liquid to vapor, a latent heat is needed to remove (Amirante and Renzo,1989; Isik, 2002)

The removal of water from the product by evaporation in vacuum cooling starts at the point at which the pressure in the vacuum chamber falls down to the saturation, pressure value corresponding to the initial temperature of the product. This point is called as the flash point (He and Li, 2003). If the pressure in the vacuum chamber is further reduced, the evaporation of water and cooling of the product continues until the desired product temperature is reached. Product temperature is reached to 0°C pressure 0.6 kPa in the chamber. Pressure lower than 0.6 kPa cause the product to freeze (Wang and Wen, 2002a). It is disapproved to go below this pressure value, if it is aimed to evaluate the products to be cooled freshly in the market (Isenberg *et al*., 1986).

The models based on heat and mass transfer and fluid dynamics were put forward by different researchers in the mathematical definition of vacuum cooling system.

Houška *et al*. (1996) proposed a model for predicting the temperature of liquid foods during vacuum cooling. It includes a mathematical description of the ingress of ambient air into process equipment, which leads to reduced cooling rates and the potential for contamination of the foodstuff. The model considers the air flow as an adiabatic compressible flow through a nozzle. The time dependencies of total vessel pressure, liquid temperature and the mass of condensed vapor were predicted well.

The vacuum cooling of selected cooked solid food semi products (carrots, potatoes, parsley, beef and pork steaks and chicken breast) was experimentally studied with regard to an important mass transfer parameterthe product of the mass transfer coefficient and mass transfer surface, k_S. in the study of Landfeld *et al*. (2002). The data presented in this paper can serve for mathematical modeling, process kinetics prediction and engineering calculations of the vacuum cooling of semi products destined for preparing ready meals.

A model has been developed for predicting the time temperature and mass of spherical solid foods during vacuum cooling in a study by He and Li (2003). This study discusses the effects of product thermo physical properties, convection heat transfer coefficient, latent heat of evaporation as well as vacuum environmental parameters that govern the heat and mass transfer of product. The time trends of total system pressure, product temperature such as surface temperature, centre temperature, mass-average temperature, the mass of product were predicted. The model accounts for the change of temperature of solid product systematically during vacuum cooling by means of simulation.

Dostal and Petera (2003) carried out a study named Vacuum cooling of liquids: Mathematical model. This study describes a simple mathematical model of the vacuum cooling process which enables to predict a temperature evolution regarding the equipment size, vacuum pump parameters and properties of the cooled liquid. Real thermo physical properties of the cooled liquid are considered in the model along with the assumption that the main resistance against mass transport is situated on the side of liquid phase. Parameter identification of mass and heat transfer coefficients based on literature experimental data together with results of numerical simulation of real vacuum cooling equipment are described at the end.

In this study, the thermodynamically analysis of vacuum cooling was realized depending on the studies which were made before and the values of vacuum pump capacity and electrical motor power; initial, flash point and final temperatures and pressure values; weight loss resulting from vacuum cooling and energy expenditure all of which are important parameters in the design of vacuum cooling system were calculated. Results of thermodynamically analysis were tested and compared implementing the vacuum cooling of lettuce in the experimental design prepared.

MATERIALS AND METHODS

The trials of vacuum cooling were carried out in the experimental vacuum cooler at Uludag University of Agriculture, Faculty Agriculture Machinery Department of Research and Application Laboratory, Turkey (Fig. 1). The apparatus consisted of a vacuum chamber, vacuum pump(s), condenser and measuring tools.

Vacuum chamber was designed at a diameter of 0.6 m and length of 1m so as to contain 10 kg of product, in average, in one cooling period. Two rotary oil-sealed vacuum pumps are used the pumping capacity of 8.4 and 12 $m^3 h^{-1}$. Separately a compressor with power 0.368 kW is used in the cooling system. Analogue-vacuum meter is used to measure the pressure in the chamber. Weightings were done by the digital balance (Baster, Germany) that 1 g sensitivity. Digital counter, 0.01 Wh sensitivity, is used to measure energy expenditure.

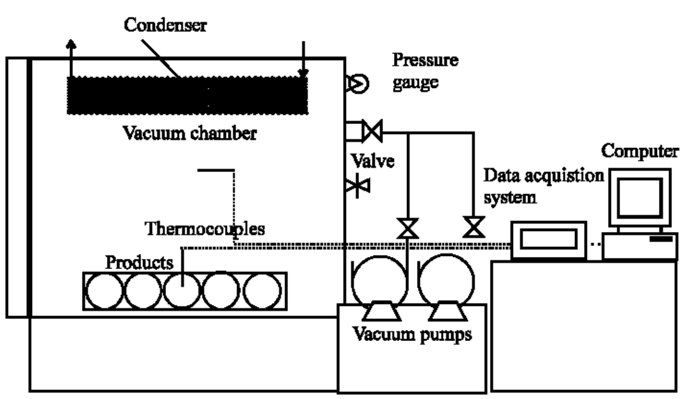

Fig. 1: Schematic diagram of the experimental vacuum cooler

Lettuce (*Lactuca sativa* L.) with an average weight of 1.54 kg and a specific volume of 1.538 $dm^3\ kg^{-1}$ was used in the trials. After the sample was weighed and placed in the vacuum chamber, the thermocouple was pressed 3 cm deep into the sample to sensitively measure temperature. The trials were carried out with 3 replicates and mean values were calculated. Weight loss, temperature, time, pressure and energy expenditure values were recorded during the trials.

Thermodynamically Analysis of Vacuum Cooling

In the relationship between pressure, volume and temperature in a vacuum cooling operation, the thermodynamic process is assumed to take place in two phases (He and Li, 2003).

The P-v (pressure-specific volume) diagram of a vacuum cooling process is given in Fig. 2. Vacuum pumps are operated after the product is placed into the vacuum chamber and after the covers are closed. The pressure falls when the vacuum pumps remove the air inside the chamber. Suction of air continues until the pressure in the chamber falls to the saturated pressure of product temperature. This is point 2. No change is observed in the product temperature during this period until the pressure inside the chamber reaches the flash point pressure (P_2) (Wang and Wen, 2001) from the atmospheric pressure (P_1). Isothermal state exchange is in effect between points 1 and 2 because the temperature is constant ($T_1 = T_2$) (Lutz and Hardenburg, 1968).

The relationship between pressure and volume, which are the measurable thermodynamic features in the isothermal state exchange, is defined by;

$$Pv = \text{Constant} \tag{1}$$

Utilizing the ideal gas equation due to the evacuation of air between points 1 and 2;

$$P_1 v_{1h} = T_1 R \tag{2}$$

where R is the gas constant. The initial specific volume of air is then calculated by;

$$v_{1h} = \frac{T_1 R}{P_1} \tag{3}$$

and the last specific volume of air is calculated by;

$$v_{2h} = \frac{T_1 R}{P_2} = \frac{P_1 v_1 h}{P_2} \tag{4}$$

The work done by the vacuum pump in relation to the evacuation of air until it reaches P_2 is given by:

$$w_{1,2} = P_1 v_{1h} \ln \frac{v_{2h}}{v_{1h}} = P_2 v_{2h} \ln \frac{P_1}{P_2} = P_2 v_{2h} \ln \frac{P_1}{P_2} \tag{5}$$

Considering that the internal energy exchange of the ideal gases during isothermal change is zero, the statement

$$q_1 = w_{1,2} \tag{6}$$

is deduced. As can be seen here, the work done during isothermal change is equal to the heat taken. Also, the latent heat between points 1 and 2 may be found from Eq. (5).

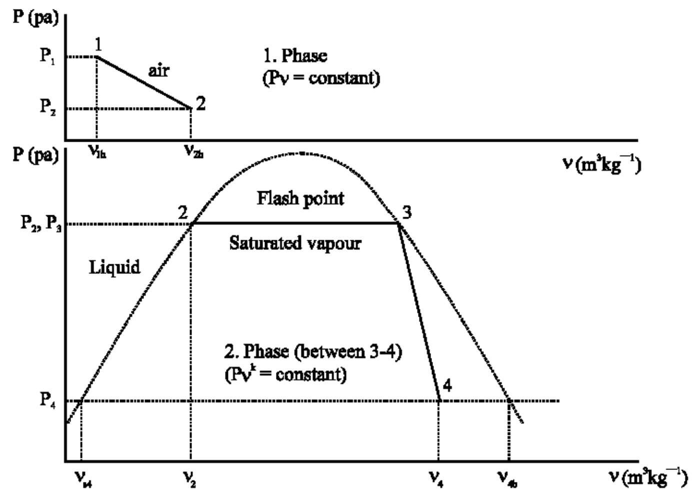

Fig. 2: P-v diagram of vacuum cooling

After the flash point is reached, water vapor occupies the space formed by the evacuation of air from the chamber, due to evaporation of water in the sample. This phase change occurs suddenly and water vapor is considered as the refrigerant in the calculations to be made after this point. Constant pressure evaporation is in effect between points 2 and 3. Therefore, when the P-v diagram (Fig. 2) is examined, the unit work done is defined by the equation

$$w_{2,3} = P_2(v_3 - v_2) \tag{7}$$

The heat taken between these points is defined as enthalpy difference by the equation

$$q_{2,3} = h_3 - h_2 = T_2(s_3 - s_2) \tag{8}$$

The temperature of the sample decreases after point 3 because the decline in pressure continues and T_4 temperature is reached at P_4 pressure.

Because the system is isolated between points 3 and 4, the phase change is considered adiabatic The relationship between pressure and volume in the adiabatic state exchange is defined by the statement,

$$PV^k = \text{Constant} \tag{9}$$

$$k = \frac{c_p}{c_v} \tag{10}$$

If the equation of the first law of thermodynamics is written as;

$$E_{ki} + h_3 + E_{pi} + q_{3,4} = E_{k_2} + h4 + E_{p_2} + W_{3,4} \tag{11}$$

then because an open system is present, kinetic and potential energies may be considered as identical at the pump inlet and outlet.

The work in the adiabatic change is given by (J kg^{-1})

$$W_{3,4} = h_3 - h_4 \quad (12)$$

Using these relations, the total work done by the vacuum pumps between points 1 and 4 is found by the equations,

$$W_b = W_{1,2}\, G_h \quad (13)$$

$$W_h = (W_{2,3} + W_{3,4})\, G_s \quad (14)$$

$$W_{vp} = W_h + W_b \quad (15)$$

If the heat quantity to be removed from the system is represented by Q, then the efficiency coefficient of vacuum cooling is found as

$$\varepsilon_v = \frac{Q}{W_{vp}} \quad (16)$$

When cooling time is taken as *t* (h), theoretical power necessary for evacuation (kW) is found as

$$N_{vp} = \frac{W_{vp}}{3600t} \quad (17)$$

while the theoretical power (W) of the necessary electric motor is found by;

$$N_{emt} = 1.5 N_{vp} \quad (18)$$

when the electric motor is considered directly connected, whereas the effective power (kW) of the electrical motor is found by the equation

$$N_{emp} = \frac{N_{emt}}{\varepsilon_{top}} \quad (19)$$

where ϵ_{top} is the total output. Moreover, the energy (Wh) spent by the system during evacuation is defined by the equation;

$$W_{et} = N_{emp}\, t \quad (20)$$

If the quantity of produce to be cooled in one cooling period via the vacuum cooling method is defined as G_u (kg), the specific heat of produce is defined as c_m (J kg^{-1} K^{-1}) and a reduction of produce temperature from T_1 to T_4 is desired, then the quantity of heat, Q (J), to be removed is found by (Amirante and Renzo, 1989):

$$Q_1 = G_u C_m (T_1 - T_4) \quad (21)$$

The respiration heat of produce (the heat emitted by the produce due to the maintenance of its vital activities along the cooling period) should also be considered when calculating the quantity of heat. If respiration heat coefficient is c_s (J kg^{-1} h^{-1}), cooling period is t (h), then the respiration heat is calculated as;

$$Q_2 = G_u C_s\, t \qquad (22)$$

Accordingly, the total heat to be removed from the produce is (Amirante and Renzo, 1989)

$$Q = Q_1 + Q_2 \qquad (23)$$

In order to remove this heat and lower the temperature of the produce from T_1 to T_4, it is necessary to reduce the system pressure from P_1 to P_4. The latent evaporation heat (J kg^{-1}) is calculated by the equation;

$$\Delta h = h_{4b} - h_{4s} \qquad (24)$$

However, point 4 is at moist vapor state. Therefore, the latent evaporation heat until point 4 will be

$$\Delta h = h_4 - h_{4s} \qquad (25)$$

At this point, the dryness degree, x_4, should be known in order to find h_4.
Since an adiabatic state change is occurring between points 3 and 4, s_3 is equal to s_4. From this point, the equation;

$$S_4 = S_{4s} + X_4\,(S_{4b}\text{-}S_{4s}) \qquad (26)$$

can be written. Then the dryness degree, x_4, is found as;

$$x_4 = \frac{s_4 - s_{4s}}{s_{4b} - s_{4s}} \qquad (27)$$

Determining the dryness degree from this equation, it is possible to find

$$h_4 = h_{4s} + x_4\,(h_{4b}\text{-}h_{4s}) \qquad (28)$$

When h_4 is replaced in Eq. 25, then (Alibas, 1982),

$$\Delta h = x_4 (h_{4b} - h_{4s}) \qquad (29)$$

The quantity of water (kg) to be evaporated from the produce for the removal of this heat is stated as;

$$G_s = \frac{Q}{\Delta h} = \frac{Q}{x_v (h_{4b} - h_{4s})} \qquad (30)$$

In this situation, the percentage weight loss in the produce is (Savas and Turk, 1992)

$$G_k = \frac{G_s}{G_u} 100 \tag{31}$$

When the air volume that must be removed by the vacuum pump to reach the flash point is shown by V_h and the amount of water vapor that should be removed after the flash point is reached is shown by V_b, the theoretical capacity of the vacuum pump (m^3) is found as;

$$V_{vp} = V_h + V_b \tag{32}$$

Here, if G_h is taken as the air mass to be removed, then;

$$V_h = G_h.v_{2h} \tag{33}$$

Because the volume of water vapor is:

$$V_b = G_s v_4 \tag{34}$$

then Eq. (33) and (34) provide

$$V_{vp} = G_h v_{2h} + G_s v_4 \tag{35}$$

The fact that the specific volume of water vapor is very high under low pressures showed that the free air capacities of the vacuum pumps to be used in the system should be very high (Dostál and Petera, 2003).

Therefore, it is necessary to condense the vapor removed by the vacuum pumps via evaporation at a certain rate and convey it again to the system in liquid state. 100% condensation is optimum with respect to avoiding the vacuum pumps consuming excess power. However, an evaporator surface area that will exceed the tank volume will be required, which is technically difficult. For this reason, cooling systems that can condense the evaporated water between 50% and 90% are used in vacuum coolers in practice (Savas and Turk). The condensation rate (Z) should be taken into consideration when determining the free air capacity of the vacuum pump.

The P-v diagram considering the condensation rate is shown in Fig. 3. When the diagram is examined, the vapor quantity at point 4' will be condensed up to the extent of Z and vapor volume ($m^3\ kg^{-1}$) will be;

$$v'_4 = v_{4s} + (1 - Z)(v_4 - v_{4s}) \tag{36}$$

Here, the volume of water vapor to be pumped is

$$V_b = G_s V_4 \tag{37}$$

The same phenomenon is also valid for the air transported between 1 and 2, so the volume of air transported will be

$$V_h = G_h O.lv_{2h} \tag{38}$$

From here, the theoretical capacity of the vacuum pump is found by Eq. (32),

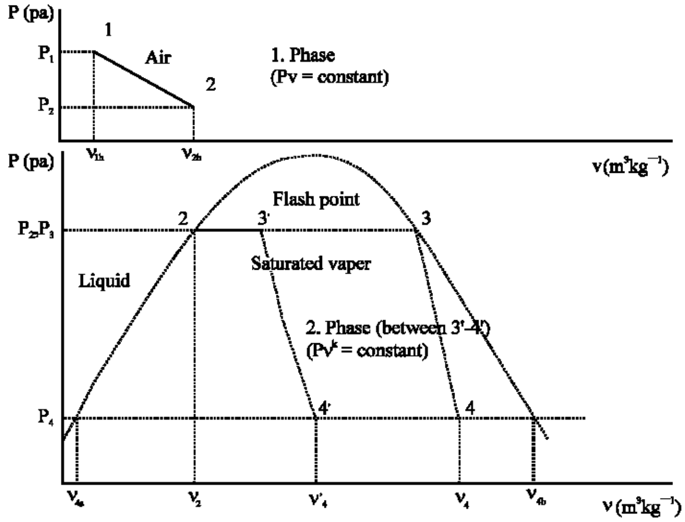

Fig. 3: P-v diagram of vacuum cooling with condenser

$$V_{vp} = V_h + V_b$$

and the pumping capacity of the vacuum pump (V_{pk}) is found by the equation;

$$V_{pk} = \frac{V_{vp}}{t} \qquad (39)$$

where t (h) is the total cooling time.

RESULTS AND DISCUSSION

Experimental Results

The results of the experiments on lettuce are given in Fig. 4. At the beginning of the trial, the temperature of lettuce was 20°C. Product temperature begins to decline parallel to the decline in pressure after the flash point. The cooling time at the point where the product temperature reached 2°C was 33 min and the pressure was 0.60 kPa. A temperature decline of 18°C occurred on the product during the 33 min cooling period and 0.30 kWh (0.22 kWh case^{-1}, Thompson *et al.* (1978)) was spent in total by the cooling compressor plus the vacuum pumps. 1.6 Wh of energy is required on average for a 1°C decline in product temperature in the cooling of 1 kg of lettuce. At the end of the trial, a weight loss of 3.13% (3-4 %, Wang and Wen (2001); 4.75, Martínez and Artes(1999) was observed in lettuce.

Approaching Limits of Thermodynamically Analysis of Vacuum Precooling to the Experimental Model

As evidenced in Table 1, the total capacity of vacuum pumps used in the trial was 20.4 $m^3 h^{-1}$. The theoretical vacuum pumping capacity for the system was found as 11.24 $m^3 h^{-1}$ in the calculation made using thermodynamically analysis. Total label power of the electrical motors used in the study was 1.458 kW. The theoretical value of necessary power was calculated as 0.617 kW.

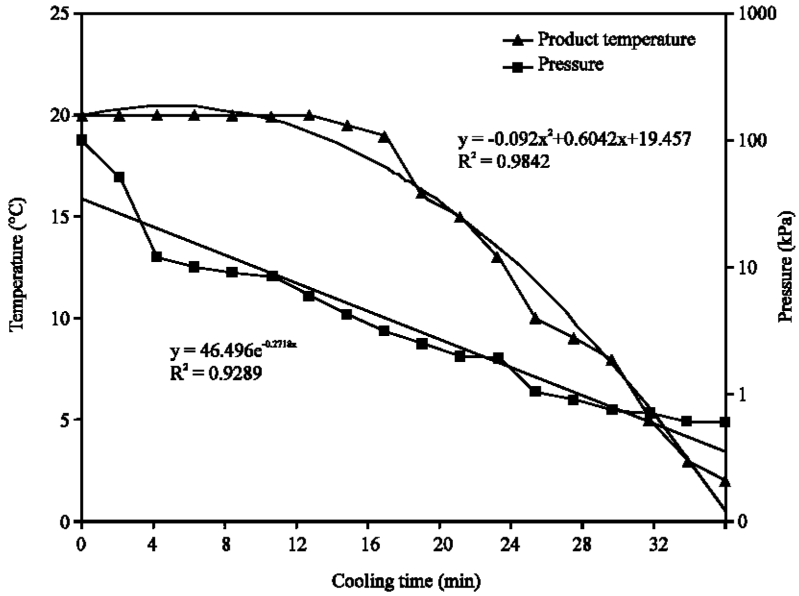

Fig. 4: Experimental results of lettuce with vaccum cooling

Table 1: Values of experimental model and thermodynamically analysis

Parameters	Unit	Results of thermodynamically analysis	Results of experiment	Difference
G_k	(%)	3.06	3.13	0.07
ϵ	-	4.196	-	-
h_2	kJ kg^{-1}	83.64	-	-
h_3	kJ kg^{-1}	2539.60	-	-
h_4	kJ kg^{-1}	2382.55	-	-
N_{emp}	kW	0.293	1.09	0.797
$^*N_{sme}$	kW	0.617	1.458	0.841
N_{vp}	kW	0.087	-	-
P_1	kPa	101.33	101.33	-
P_2	kPa	2.33	2.13	0.2
P_3	kPa	2.33	2.13	0.2
P_4	kPa	0.7	0.6	0.1
s_2	kJ kg^{-1}	0.2939	-	-
s_3	kJ kg^{-1}	8.6683	-	-
s_4	kJ kg^{-1}	8.6683	-	-
T_1	°C	20	20	-
T_2	°C	20	20	-
T_4	°C	2	2	-
Q	KJ	728.70	-	-
v_2	m^3 kg^{-1}	10017.10^{-3-}	-	-
v_3	m^3 kg^{-1}	57.78	-	-
v_4	m^3 kg^{-1}	170.32	-	-
v'_4	m^3 kg^{-1}	17.034	-	
V_{pk}	m^3 h^{-1}	11.24	20.40**	9.16
$w_{1,2}$	KJ kg^{-1}	316.90	-	-
$w_{2,3}$	KJ kg^{-1}	135.027	-	-
$w_{3,4}$	KJ kg^{-1}	157.105	-	-
W_{et}	kWh	0.159	0.161	0.002
$^*W_{set}$	kWh	0.340	0.339	0.001
W_v	kJ	173.67	-	-

*with a compressor of cooling system, **total label value of vacuum pumps

The difference between these two values originates from the much higher power of the second vacuum pump (0.89 kW) compared with the first one. However, the second vacuum pump was operated only in the 10 min part of the trial period. Therefore, the quantity of energy expended by the system should be taken into consideration so that the electrical motor power required for the system could be determined.

The total energy quantity expended by the system in the lettuce trial was measured as 0.34 kWh. From this value, the necessary power required for the system was determined as 0.618 kW. The power value calculated according to thermodynamically analysis model was found as 0.617 kW.

The weight loss rate the end of the lettuce trial was 3.13%. The average weight loss rate was calculated as 3.06%. Considering these values, the weight loss value could be determined with a deviation of only 2.12%.

CONCLUSIONS

According to the findings of the trial and results of the thermodynamically analysis, it is possible to determine the weight loss with a deviation of 2.12%, nearby the other parameters to be used in the design of vacuum precooling system, such as temperature, pressure, enthalpy and entropy on specified points using the mathematical model prepared from thermodynamically equations. Moreover, the fact that the power requirement, the most important parameter in the design of the system, could be determined within a deviation of 0.162% reveals that the mathematical model could be used in the design of a vacuum precooling system.

ACKNOWLEDGMENT

Special thanks are given to Prof. Dr. Kamil Alibas (Agriculture Faculty of uludag University) for his help in the thermodynamically analysis of vacuum cooling method.

Nomenclature

1	Initial point of vacuum cooling	T	Temperature (K, °C)
2	Initial point of flash point	t	Time of vacuum cooling (h, min)
3	Final point of flash point	v	Specific volume (m^3 kg^{-1})
4	Final point of vacuum cooling	V	Volume (m^3)
c_m	Specific heat (J $kg^{-1}K^{-1}$)	V_{pk}	Vacuum pumping capacity ($m^3 h^{-1}$)
c_s	Specific respiration heat (J $kg^{-1}h^{-1}$)		Work (J)
E_k	Kinetic energy (J kg^{-1})	w	Unit work (J kg^{-1})
E_p	Potential energy (J kg^{-1})	W_{et}	Energy expended by vacuum pump (Wh)
ϵ	Efficiency coefficient	W_{set}	Energy expended by system (Wh)
G_u	Product weight (kg)	X_v	Dryness degree
G_s	Water weight (kg)		
G_k	Weight loss (%)		Subscripts
G_h	Air weight (kg)	1	Initial point of vacuum cooling
h	Enthalpy (J kg^{-1})	2	Initial point of flash point
k	Coefficient	3	Final point of flash point
N_{vp}	Power of vaccum pump (W)	4	Final point of vacuum cooling
V_{emt}	Theoretical electricity power of vaccum pump (W)	b	Vapor
		h	Air

vcN_{emp}	Practical electricity power of vaccum pump (W)	p	Pressure
		s	Water
N_{sme}	Total power of system (W)	v	Volume
P	Pressure (Pa, mmHg)	vp	Vacum pump
Q	Total heat (J)	top	Total
Q	Specific heat (J kg^{-1})		
R	Universal gas constant (kmol)		
s	Entropy (J kg^{-1} k^{-1})		

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