Measurement of Steam/water Ratio in the Nozzle Jet of an Oven System Using Superheated Steam and Hot Water Spray

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A heating system using superheated steam (SHS) in combination with a spray of hot water micro droplets (WMD) has been developed to prevent drying of food material during SHS heating. In the SHS+WMD system, a mixture of SHS (115°C) and WMD is generated by throttling boiling water and steam (120 to 140°C) with a nozzle. If the moisture content of food material can be controlled through regulation of the SHS/WMD ratio, the combination system is expected to provide flexible application of SHS for food processing. In this study, methods for measuring and regulating the ratio of SHS and WMD in the nozzle jet were investigated. It was found that the SHS/WMD ratio could be easily determined from the difference between the water uptake rate of the system and the theoretical steam flow rate through the nozzle (derived from the internal pressure of the nozzle). The WMD ratio increased as the nozzle internal pressure increased; by decreasing the nozzle throat diameter or by increasing the water uptake rate. SHS+WMD was generated when the nozzle internal pressure was greater than 0.188 MPa and the steam velocity at the nozzle throat was sonic, regardless of the nozzle throat diameter.

Key words: Choking flow, Convergent nozzle, Steam jet oven, Vapor–liquid two-phase flow

1. Introduction

Applications of superheated steam (SHS) for food processing are currently being studied because of its various advantages. The advantages of SHS heating include the efficient heat transfer by latent heat and gas radiation, as well as the inhibition of nutrient oxidation in food. SHS has been applied for cooking [1–4], drying [5–6], pasteurization [7] and extraction [8]. Drying capacity is one of the advantages of SHS; however, difficulties can arise when applying this technology to the cooking and blanching of fruits and vegetables. As the drying rate of the food product depends on heat flow (e.g., convection and radiation heat transfers from SHS) into the product [9], it is usually difficult to simultaneously achieve optimum product moisture content, heating time and temperature. It is expected that the development of technology that can control the moisture content of food during SHS heating will result in increased SHS application flexibility in food processing.

To prevent the drying of food during SHS heating, the authors have developed an oven system using a combination of SHS and a spray of hot water micro droplets (WMD). In the SHS+WMD system, pressurized boiling water and steam (120 to 140°C) are sprayed under atmospheric pressure through a nozzle onto the food. The saturated steam is throttled at the nozzle throat and becomes SHS at approximately 115°C. The hot water is atomized with the steam jet and is suspended in SHS. In a previous study, the SHS+WMD system was applied to the blanching of potato [10]. It was found that the modified and improved SHS system enhanced the quality of blanched potato. WMD prevented potato weight loss, as compared to a 3% loss of weight when heated with ordinary SHS at the same temperature.

It is expected that the SHS+WMD system will be applied to the cooking and blanching of other kinds of fruits and vegetables, as well as other heating processes. However, it is reasonable to suppose that optimal moisture content differs for each food material and for each heating purpose. The moisture content of food material can be regulated by changing the ratio of SHS and WMD discharging from the nozzle. Knowledge of how to
manipulate the system’s steam/water ratio, as well as the development of a ratio measurement method, are necessary for the multi-purpose application of the SHS+WMD system. However, measurement of the steam/water ratio in this system is problematic since the SHS and WMD are discharging from the same nozzle. Measuring the ratio with a throttling calorimeter and a steam table is also difficult because the mixture of steam and water is unsteady at the outlet of the nozzle. In addition, the amount of water is supposed to be in excess for throttling calorimeter under the operating condition of this system.

The main objective of this study was to propose simple and practical methods for measuring the ratio of steam and water discharging from the SHS+WMD system nozzle. In this study two methods were proposed and examined, a method using the thermodynamics of gas flow through the nozzle and a method using heat balance analysis. Results obtained using these methods were compared to verify the methods. Advantages and disadvantages of these methods were discussed. Another objective of this study was to establish a method for regulating the steam/water ratio in the nozzle jet. The steam/water ratio was measured while changing the operating conditions of the system, allowing further investigation into the relationship between operating conditions and the steam/water ratio.

2. Materials and Methods

2.1 System for superheated steam and hot water heating

2.1.1 Principle of the system

Figure 1 presents a schematic drawing of the system developed for generating SHS+WMD. In this system, a panel heater with an embedded copper pipe and heating wires is installed in a heating chamber. Water supplied by a pump boils inside the copper pipe under high pressure. The boiling water and steam are sprayed through a nozzle. The steam sprayed through the nozzle is throttled and becomes SHS. The hot water is atomized by the steam flow and is suspended as WMD in SHS. The WMD subsequently evaporates, absorbing the heat of the SHS. However, the panel heater maintains the temperature of SHS; consequently maintaining the mixture of SHS and WMD since WMD is continuously sprayed from the nozzle. In this system, pressure inside the heating chamber is almost equal to atmospheric pressure since a portion of the steam, which increases with the evaporation of the WMD and continuous supply of the SHS from the nozzle, is exhausted through an outlet. The advantages of this steam/water spray system, as compared to a system that supplies steam and hot water droplets separately, include low system manufacturing costs and the atomization of finer water droplets by the steam flow. The fine droplets are assumed to be preferable as they are uniformly distributed in SHS.

2.1.2 Description of the system

Figure 2 depicts schematic diagrams of the system. Cross sections of the heating chamber, connections of the water and steam line, and heating wires and sensors are drawn from the top and the side. Internal dimensions of the heating chamber are 1015×460×340 mm (WDH). The aluminum panel heater, with dimensions of 850×400 ×25 mm (WDH), is installed on the ceiling of the heating chamber. The copper pipe that is embedded in the panel heater is 10 m long with an inner diameter of 4 mm. The heating wires are embedded in the panel heater along the copper pipe. The heater output is 4 kW at maximum, and it is controlled to maintain the steam temperature inside the heating chamber at a set value. The panel heater is subsequently used for heating the supplied water and for maintaining the SHS temperature. The plunger pump sends reverse osmosis water from a tank to a preheater. The output of the preheater is controlled to maintain a constant water temperature at the outlet of the preheater. After preheating, the water is sent to the panel heater, where it is boiled. The boiling water is sprayed into the heating chamber through the nozzle. Thermocouples and pressure gauges are installed at the inlet of the panel heater and beside the nozzle. A 200 mm-diameter fan is installed inside the heating chamber for stirring the SHS and WMD. The heating chamber has
2.2 Methods proposal

The heat balance measurement is one of the theoretically simplest methods for determining the steam/water ratio in the nozzle jet flow in this system. In this method, the specific enthalpy of the steam and water inside the nozzle is obtained by adding the heat from the panel heater to the specific enthalpy of the water supplied to the panel heater. The heat added to the water from the panel heater is obtained from the difference between the input electricity of the panel heater and heat loss from the surface of the panel heater. The steam/water ratio can be determined from the specific enthalpy of the steam and water by the steam table or IAPWS-IF97 [11]. Although this method is theoretically certain, heat flux must be measured at several points on the heater surface to ensure accurate measurement, since heat flux distribution on the heater surface is uneven. The limitations of this method are that heat flux sensors with low thermal resistance are expensive and the number of points required for heat flux measurement increase as the system scale increases.

Gas flow measurement is an alternative method for steam/water ratio determination. In this method, the steam/water ratio is determined by the difference between the flow rate of water supplied to the system and the theoretical flow rate of steam at the nozzle throat. Parameters measured in this method are limited to the water uptake rate of the system, and the pressure and temperature of the inside and at the outlet of the nozzle, regardless of system size. However, this method involves an uncertainty whether the steam flow rate can be correctly calculated in the presence of water droplet in the nozzle throat.

If the method using gas flow measurement is valid, it is more advantageous than the heat balance measurement method, in terms of practical applicability, due to its small sensor cost. In this study, the ratio of steam and water discharging from the nozzle was determined using the two proposed methods, and the gas flow measurement method was verified by comparing the results.
obtained by these methods. Details of the methods are described in the following sections.

2.3 Nozzle flow measurement

2.3.1 Fluid measurement

Flow rates of the water supplied to the system were determined by using an electric balance to measure the mass decrease in the water tank. The mass decreases were measured for 10 minutes at 10-second intervals; flow rate \( G_m \) was subsequently calculated by regression analysis of mass changes in the water tank. The pressure \( P_i \) and temperature \( T_i \) inside the nozzle were measured for each experimental condition. The pressure and temperature were measured for 10 minutes at 100 ms intervals. The measurements were repeated five times.

2.3.2 Calculation of steam/water ratio

When only steam is blown from the convergent nozzle, the velocity and flow rate are determined by the pressure and temperature of the steam inside and outside the nozzle. In this case, velocity \( w \) and specific enthalpies \( h \) of the steam at the stagnation point and the throat of the nozzle are described by the following equation, since steam flows adiabatically through the nozzle [12].

\[
\frac{1}{2} w^2_1 + h_1 = \frac{1}{2} w^2_2 + h_2 + \Delta h
\]

Since \( w_1 \) is negligibly small, the steam flow velocity at the nozzle throat is written as

\[
w_2 = \sqrt{2(h_2 - h_1)}
\]

The heat drop \( \Delta h \) is expressed by

\[
\Delta h = h_1 - h_2 = \frac{\rho}{\gamma} w dP
\]

Since steam is adiabatically expanded by the nozzle, the relationship between pressure and specific volume \( P_i v_i = P_2 v_2 = P_v v_2 = P_v \gamma \) can be substituted for equation 3, which becomes

\[
\Delta h = \left[ \frac{\rho}{\gamma} \right] P_i dP = \frac{\rho}{\gamma} w \left[ \frac{1}{P_i} \right] \int_{P_i}^{P_2} \frac{1}{v} dP
\]

\[
= P_i \gamma \left( \frac{v_2}{v_1} \right) \frac{1}{P_2} \left[ \frac{1}{P_i} - \frac{1}{P_2} \right] \left( \frac{v_2}{v_1} \right)
\]

The steam flow velocity is consequently calculated as

\[
w_i = \sqrt{2 P_i \gamma \left( \frac{v_2}{v_1} \right) \frac{1}{P_2} \left[ \frac{1}{P_i} - \frac{1}{P_2} \right] \left( \frac{v_2}{v_1} \right)}
\]

where

\[
\phi = \sqrt{2 \gamma \left( \frac{v_2}{v_1} \right) \frac{1}{P_2} \left[ \frac{1}{P_i} - \frac{1}{P_2} \right] \left( \frac{v_2}{v_1} \right)}
\]

The flow rate \( G \) of steam is given by

\[
G = \rho \gamma A_i w_i
\]

The flow rate equation is rewritten by substituting \( \rho \gamma = 1/\gamma \) and \( P_i v_i = P_2 v_2 = \phi \) for equation 7:

\[
G = \left( \frac{P_2}{P_i} \right)^{1/\gamma} \rho \gamma A_i w_i
\]

\[
= A_2 \phi \left( \frac{P_2}{P_i} \right)^{1/\gamma}
\]

\[
= A_2 \phi \frac{P_2}{\sqrt{RT_i}}
\]

where

\[
\phi = \left( \frac{P_2}{P_i} \right)^{1/\gamma}
\]

As the internal pressure of nozzle \( P_i \) increases, \( \phi \) and \( \phi \) increase, in other words, the steam velocity \( w \) and steam flow rate \( G \) increase. However, the flow function \( \phi \) reaches maximal when \( \partial \phi / \partial (P_i/P_2) = 0 \). In this case, the relationship between \( P_1 \) and \( P_2 \) is derived from equation 9 as follows.

\[
P_2 = P_i \left( \frac{2}{1+\gamma} \right)^{\gamma/\gamma+1}
\]

where \( P_c \) is called the critical pressure against \( P_i \). When the pressure of steam outside the nozzle is atmospheric pressure (0.101 MPa), \( \phi \) reaches maximal when \( P_i \) increases to 0.188 MPa. When the steam pressure inside nozzle \( P_i \) is higher than 0.188 MPa, the steam pressure at nozzle throat \( P_2 \) increases as \( P_i \) increases due to flow choking and is given by equation 10. In this case, the steam velocity \( w_i \) is sonic and given by the following equation derived from equation 5, equation 10 and \( P_i v_i = P_2 v_2 \).

\[
w_i = \sqrt{2 \gamma \left( \frac{v_2}{v_1} \right) \frac{1}{P_2} \left[ \frac{1}{P_i} - \frac{1}{P_2} \right] \left( \frac{v_2}{v_1} \right)}
\]

In contrast, \( P_2 \) is constant and equal to atmospheric pressure when \( P_i \) is lower than 0.188 MPa and the steam velocity at the nozzle throat is subsonic. In this study, the critical pressure \( P_i \) was used as the pressure at the nozzle throat \( P_2 \) when \( P_i \) calculated from equation 10, exceeded atmospheric pressure \( P_a \).

The steam actually expanded irreversibly at the throat of the nozzle. Therefore, the actual steam flow rate \( G' \) was smaller than the value given by equation 8. It is given by

\[
G' = C_c A_2 \frac{P_2}{\sqrt{RT_i}} \phi
\]
where $C_d$ is the nozzle coefficient, or the discharge coefficient of the nozzle.

If only steam is blown from the nozzle, the flow rate is obtained by equation 12. In contrast, if the steam flow contains water, the measured flow rate $G_m$ will exceed the flow rate $G_s$, given by equation 12, because the specific gravity of water is much greater than that of steam. In this study, the difference $G_m - G_s$ determined the water flow rate through the nozzle. Thus, the steam ratio $x$ in the flow was defined as

$$x = \frac{G_s}{G_m}$$  (13)

Values of the gas constant and the specific heat ratio of steam used for calculating $x$ are presented in Table 1.

### 2.3.3 Measurement and calculation of nozzle coefficient

The nozzle coefficient $C_d$ was measured using dry air flow. First, the nozzles used in this study were connected to an air compressor. A flow meter, pressure gauge, thermocouple and a valve were connected between the nozzle and air compressor. The air flow rate $G_m$, pressure $P_1$ and temperature $T_1$ of the air at the stagnation point inside the nozzle were measured while air was blown from the nozzle by the compressor. The nozzle coefficient was calculated by

$$C_d = \frac{G_m}{G}$$  (14)

where $G$ is the ideal airflow rate, obtained by substituting $P_1$ and $T_1$ for equation 8.

The nozzle coefficient is a function of Reynolds number $Re$, and $C_d$ is expressed by the following empirical formula called nozzle function [14]:

$$C_d(Re) = \alpha - \beta \frac{1}{\sqrt{Re}}$$  (15)

In this study, the nozzle coefficient $C_d$ was measured by changing the Reynolds number of airflow regulating the nozzle internal pressure with the valve. The Reynolds number of airflow was changed from $2 \times 10^4$ to $7 \times 10^4$, which includes the Reynolds number range of steam flow examined in this study. The coefficients of nozzle function $\alpha$ and $\beta$ were determined by fitting equation 15 to $(C_d, Re)$ with the dataset obtained from measurements using the Newton method. The values of $\alpha$ and $\beta$ are function of the specific heat ratio of gas, however the differences in nozzle coefficient among the gases with the specific heat ratio of 1.3 ~ 1.4 are approximately 0.1% [15]. Hence, the error in the steam flow rate calculation is negligibly small if the nozzle coefficient obtained by the air flow is used instead of the coefficient obtained by steam flow.

The Reynolds number was given by

$$Re = \frac{D w_2}{\nu}$$  (16)

where $D$ is a characteristic length (i.e. the diameter of the nozzle throat in this study), $w_2$ is the velocity of airflow obtained from equation 5, and $\nu$ is the kinematic viscosity of the fluid. The kinematic viscosity of air was calculated as

$$\nu = \frac{\eta}{\rho} = \eta \frac{RT_2}{P_2}$$  (17)

where $R$ is the specific gas constant of air and $\eta$ is viscosity of air. The air pressure at the throat of the nozzle $P_2$ was the atmospheric pressure or the critical pressure, obtained by the same method as described in the previous section. The temperature of air $T_2$ was calculated by the following equation of adiabatic change:

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\kappa/\gamma}$$  (18)

### Table 1  Physical properties used for the calculation of gas flow.

<table>
<thead>
<tr>
<th></th>
<th>$T$ [K]</th>
<th>$P$ [MPa]</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific gas constant $R$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>steam</td>
<td>-</td>
<td>-</td>
<td>465.1 J Kg$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>air</td>
<td>-</td>
<td>-</td>
<td>287.0$^a$ J Kg$^{-1}$ K$^{-1}$</td>
</tr>
<tr>
<td>Specific heat ratio $\kappa$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>steam</td>
<td>380</td>
<td>0.1</td>
<td>1.341$^b$</td>
</tr>
<tr>
<td>air</td>
<td>300</td>
<td>0.1</td>
<td>1.402$^b$</td>
</tr>
<tr>
<td>Viscosity $\eta$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>air</td>
<td>300</td>
<td>0.1</td>
<td>18.57$^a$</td>
</tr>
</tbody>
</table>

$^a$The specific gas constant of air was calculated by an equation $R = R / M$ where $R$ is universal gas constant ($8.314$ kJ kmol$^{-1}$ K$^{-1}$) and $M$ is molecular weight of air ($28.97$ kg/kmol).

$^b$The specific heat ratio and the viscosity were assumed constant since their dependency on temperature and pressure was negligible within the range used in this study.
where \( \kappa \) is the specific heat ratio of air. The values of the gas constant, viscosity and specific heat ratio of air are given in Table 1.

In order to obtain \( C_d \) for the calculation of steam flow rate by equation 12, the Reynolds number of steam flow should be given to equation 15. The Reynolds number of steam flow was calculated by giving the nozzle diameter, steam velocity obtained from equation 5, and the kinematic viscosity of steam to equation 16. The kinematic viscosity was calculated by the Recommended Interpolating Equation of Viscosity [16] and IAPWS-IF97. When the temperature at the nozzle throat was lower than the saturated temperature, \( T_2 \) was set to the saturated temperature, and the kinematic viscosity of the saturated steam was then used for calculating the Reynolds number.

### 2.4 Heat balance measurement

#### 2.4.1 Heat flux measurement

The electricity consumption of the panel heater and heat from the panel heater were measured. The output power of the panel heater \( E \) was obtained by means of an electric power meter (2002PA, Kyoritsu Electrical Instruments Works, Ltd.). The heat loss from the panel heater \( Q \) was measured using heat flux sensors (HF-20, Captec). The heat flux sensors were attached on the panel heater by thermal paste. The thermal resistance and the accuracy of the heat flux sensor were 0.00015 m² K/W and ±5% respectively. The effect of the thermal resistance of the sensor and the thermal paste on the heat flux was neglected since the decrease in the heat flux by the thermal resistance was estimated less than 2%. Figure 3 depicts the points for the heat flux measurement on the top and bottom sides of the panel heater. The points for the heat flux measurement on the top and bottom sides were empirically decided. The heat flux measurements were repeated by changing the positions of five heat flux sensors on each side. When the number of points for the heat flux measurements exceeded 20, the heat flux distribution obtained by interpolation slightly changed by adding the measuring points. In this study, heat fluxes on 30 points were measured for the top and bottom sides.

The heat flux distributions on the top and the bottom sides were obtained with Akima interpolation by statistical computer software “R” [i] (Fig. 3). The heat losses from the top and the bottom sides were calculated with numerical integration by dividing the surface area into isosceles right-angled triangles with 5 mm long-side. Each lateral area was divided to 5 equal areas and the heat flux at the center of each area was measured. The heat losses from the lateral sides were obtained by summing the products of heat flux and rectangle area. The total heat loss \( Q \) was obtained by summing the heat losses from top, the bottom, and the lateral sides. The water supply rate \( G_m \), the pressure \( P_0 \) and the temperature \( T_0 \) at the inlet of the panel heater were also measured.

#### 2.4.2 Calculation of steam/water ratio

Because all of the heat difference of produced heat and released heat \( E-Q \) was used to increase the enthalpy of the supplied water to the panel heater, the specific enthalpy of the water and steam inside nozzle \( h_1 \) was described as

\[
h_1 = h_0 + \frac{E - Q}{G_m}
\]

where \( h_0 \) is the specific enthalpy of the supplied water at the inlet of the panel heater, calculated from the pressure \( P_0 \) and temperature \( T_0 \) by using IAPWS-IF97. The steam ratio \( x \) in the flow was calculated from

\[
h'_1 = x h_{1s} + (1-x) h_{1w}
\]

where \( h_{1s} \) and \( h_{1w} \) are the specific enthalpy of steam and saturated water, respectively, at pressure \( P_1 \). When the measured temperature \( T_1 \) was the saturated temperature at the pressure \( P_1 \), \( h_{1s} \) was the specific enthalpy of the saturated steam, calculated from the pressure \( P_1 \) by IAPWS-IF97. When the temperature \( T_1 \) was higher than
the saturated temperature, $h_{1s}$ was the specific enthalpy of the superheated steam, calculated from the pressure $P_1$ and temperature $T_1$ by IAPWS-IF97.

### 2.5 Operating conditions of the system

In this study, the temperature of the SHS phase was controlled at 115°C since, in this system, water droplets generated around this temperature were most stable. In this condition, the SHS temperature inside the heating chamber was at 113 to 116°C except in the vicinity of the panel heater and the nozzle. The temperature of steam discharging from the nozzle decreased to approximately 100°C at 50 mm from the nozzle because the steam enthalpy was converted to kinetic energy. However, the steam temperature increased to 115°C at 150 mm from the nozzle because the kinetic energy of the steam was again converted to heat by decreasing its velocity. The static pressure inside the heating chamber, at 0.100 to 0.102 MPa, was 0.1 to 0.5 kPa higher than atmosphere. Spraying nozzles with orifice diameters of 1.3, 1.5 and 1.9 mm (HB1/8U-SS0005, HB1/8U-SS00065 and HB1/8U-SS0010, Spraying System Co.) were examined to clarify the influence of nozzle orifice diameter on the steam/water ratio. The water supply rate was varied from 0.3 to 1.3 g/s by regulating the pump speed. The water temperature at the inlet of panel heater $T_0$ was maintained at 98°C by the preheater. A fan (spinning at 1,800 rpm) stirred the SHS and the droplets. The measurements were performed after the SHS+WMD system attained thermal equilibrium by operating the system for 10 to 60 minutes after changing the experiment conditions.

### 3. Results and Discussion

#### 3.1 Steam/water ratio

##### 3.1.1 Steam and water flow rate

The measured nozzle coefficients and fitted nozzle functions, described by equation 15, are depicted in Fig. 4. Although differences in the nozzle function parameters were found among nozzles with different throat diameters, nozzle coefficients increased as the Reynolds number of the air flow increased in all nozzles. The parameters of nozzle functions obtained by the curve fitting are indicated in Table 2.

Figure 5 presents the flow rates of the water and steam measured in the experiments, and the steam flow rates

![Fig. 4 Measured nozzle coefficients and fitted nozzle functions. The throat diameters of nozzles were 1.3 mm (○), 1.5 mm (△) and 1.9 mm (□).](image)

<table>
<thead>
<tr>
<th>Diameter [mm]</th>
<th>$a$</th>
<th>$\beta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.3</td>
<td>1.000</td>
<td>± 0.001</td>
</tr>
<tr>
<td>1.5</td>
<td>0.999</td>
<td>± 0.001</td>
</tr>
<tr>
<td>1.9</td>
<td>0.978</td>
<td>± 0.002</td>
</tr>
</tbody>
</table>

The values for $a$ and $\beta$ represent estimates ± standard errors of the estimates predicted by the Newton method.
calculated from the temperature and pressure inside the nozzle. When the flow rate was small and the internal pressure of the nozzle was low, the measured flow rate agreed with the calculated flow rate. This result indicated that, under these conditions, only steam flowed from the nozzle and WMD was not generated in the system. In contrast, the measured flow rate exceeded the calculated flow rate when the internal pressure was high, and the difference between the flow rates increased as the internal pressure of the nozzle increased. The difference between measured and calculated flow rates represented the amount of water that was sprayed from the nozzle, since the amount of gas phase in the fluid was given as the calculated flow rate.

3.1.2 Electricity consumption and heat release from the panel heater

Figure 6 plots the electricity consumption of the panel heater and the released heat from the panel heater. A linear relationship was found between electricity consumption and water supply rate, with nozzle throat diameter having no observed effect on electricity consumption. Heat released from the panel heater was almost constant when the flow rate was low, however when the water supply rate was higher than a certain value, the heat release increased as flow rate increased.

3.1.3 Comparison of the steam/water ratio

Figure 7 plots nozzle jet steam ratios determined by two methods. In the method by flow rate measurement, the steam ratio was obtained by giving the data presented in Fig. 5 to equation 13. In the method by heat balance measurement, the specific enthalpy of the steam and water was calculated by giving the data presented in Fig. 6 to equation 19, then after the steam/water ratio was obtained by substituting the specific enthalpy to equation 20. Although the steam ratio obtained by the heat balance measurement was, at most, 1.8% larger than the ratio obtained by the flow rate measurement, the ratios obtained by the two methods were in approximate agreement, taking into account that all the sensors used in both methods contributed a certain degree of error. The effect of water droplet presence in the nozzle throat on steam flow calculation was presumably negligible since the volume flow rate of water droplets was extremely smaller than the volume flow rate of the steam (less than 1/5000 at maximum). It was thought that the method using flow rate measurement was reliable enough, in terms of the practical application and development of the SHS+WMD system.

3.2 Characteristics of the system

3.2.1 Nozzle internal pressure and the steam/water ratio

The disagreements of the water supply rate and the calculated steam flow rate occurred when the nozzle internal pressure was higher than approximately 0.19 MPa (Fig. 5). In the case of steam, the steam flow veloc-

Fig. 6 The electricity consumption of the panel heater (open symbols) and released heat from the panel heater (solid symbols). The throat diameter of the nozzles were 1.3 mm (○, ●), 1.5 mm (△, △) and 1.9 mm ( ◆, ◆). The dashed line is a regression line of the electricity consumption and the solid line represents the energy required to heat the water (98°C) to steam (115°C) under atmospheric normal pressure.

Fig. 7 Steam ratio inside the nozzle jet estimated from the heat balance measurement and the flow rate measurement. The throat diameters of the nozzles were 1.3 mm (○), 1.5 mm (△) and 1.9 mm ( ◆). The coefficients of correlation between the steam ratios obtained by two methods were 0.997 (1.3 mm), 0.997 (1.5 mm), and 0.999 (1.9 mm).
ity at the nozzle throat attained the speed of sound when the internal pressure of the nozzle was above 0.188 MPa under atmospheric pressure. When the internal pressure of the nozzle was low and the steam velocity was subsonic, the dependence of the steam flow rate on the internal pressure was relatively large. This was presumably the reason why only steam flowed out when the internal pressure was low, since the steam flow rate could be increased by a slight increase in pressure and temperature inside the nozzle. In contrast, as the internal pressure of the nozzle increased and the steam velocity attained a sonic level, the dependence of the steam flow rate on the internal pressure decreased. In this case, the pressure and temperature inside the nozzle needs to rise considerably in order to increase the steam flow rate; however, this resulted in overheating of the heating chamber. Since the output power of the panel heater was controlled to maintain the temperature inside the heating chamber at the set value, it was thought that some supplied water was sprayed as liquid phase when the internal pressure of the nozzle was high.

### 3.2.2 Heat balance and the water flow rate

The solid line in Fig. 6 represents the energy required to heat the supplied 98°C water to steam at 115°C, 0.101 MPa. It is noteworthy that the lines denoting electricity consumption and those denoting required energy were almost parallel. In addition, the intersection obtained by extrapolating the electricity consumption plot to the heat flow axis (vertical axis) agreed with the released heat from the panel heater at a low water supply rate. This result suggests that the difference between electricity consumption and required energy represented the amount of energy that was used to maintain the temperature of the heating chamber, not to heat the supplied water.

In this system, the difference between electricity consumption and released heat indicated the energy used to heat the supplied water inside the panel heater. Hence, the supplied water had completely evaporated before it reached the nozzle, while the released heat from the panel heater was constant. In contrast, it was thought that a part of the supplied water had not evaporated inside the panel heater when the water supply rate was high, since the released heat increased as the flow rate increased. The increment of the released heat was presumably used to maintain steam temperature, which was decreased by the evaporation of water droplets sprayed from the nozzle.

### 3.2.3 Regulation method for the steam/water ratio

Figure 8 plots the changes in steam ratio in the nozzle jet by the nozzle internal pressure and the nozzle throat diameter. Fig. 8 indicates that the steam/water ratio could be regulated by controlling the nozzle internal pressure. When the nozzle internal pressure was higher than 0.188 MPa, only the pressure should be controlled for the steam/water ratio regulation since the steam and water inside the nozzle was always at saturation temperature in this condition. The nozzle internal pressure could be controlled through the water supply rate by the pump. Figure 5 and Figure 8 indicate that the steam/water ratio could be controlled also by changing nozzle throat diameter. When the nozzle internal pressure was lower than 0.188 MPa, the water ratio decreased to zero and only the steam discharged from the nozzle. Therefore, this system could be used also as an ordinary SHS heating system by setting the nozzle internal pressure lower than 0.188 MPa.

### 4. Conclusions

This study proposed and verified methods for measuring the ratio of steam and water discharging from the nozzle in a SHS+WMD system. Although the method using gas flow measurement involved an uncertainty that the effect of water droplet presence on gas flow calculation was unknown, it was concluded that this method was reliable enough for the application of SHS+WMD sys-
tem for actual food processing, as the steam/water ratio obtained by this method agreed well with the result obtained by the method using heat balance measurement. While the method using gas flow measurement requires only the water uptake rate of the system and the nozzle internal pressure and temperature for calculation of the steam/water ratio, regardless of the system size, the method using heat balance measurement requires many heat flux sensors. Therefore, the method using gas flow measurement is more advantageous when applying the SHS+WMD system to various food processing purposes.

The water ratio in the nozzle jet increased as the nozzle internal pressure increased. It was found that the water ratio could be regulated by increasing the water uptake rate of the system or by decreasing the nozzle throat diameter. The mixture of SHS and WMD was generated when the nozzle internal pressure was more than 0.188 MPa and the steam velocity from the nozzle was sonic, regardless of the nozzle throat diameter. In contrast, only SHS was generated when the nozzle internal pressure was less than 0.188 MPa and the steam velocity was subsonic.

It is expected that the results obtained in this study will enable the quantitative regulation of the steam and water flow rate of the SHS+WMD system, and that the results will accelerate the application of the SHS+WMD system to food processing.

Acknowledgments

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A)</td>
<td>sectional area</td>
<td>(m^2)</td>
</tr>
<tr>
<td>(C_d)</td>
<td>nozzle coefficient</td>
<td></td>
</tr>
<tr>
<td>(D)</td>
<td>throat diameter of nozzle</td>
<td>(m)</td>
</tr>
<tr>
<td>(E)</td>
<td>electricity consumption</td>
<td>(W)</td>
</tr>
<tr>
<td>(G)</td>
<td>flow rate</td>
<td>(kg s^{-1})</td>
</tr>
<tr>
<td>(h)</td>
<td>specific enthalpy</td>
<td>(J kg^{-1})</td>
</tr>
<tr>
<td>(M)</td>
<td>molecular weight</td>
<td>(kg kmol^{-1})</td>
</tr>
<tr>
<td>(P)</td>
<td>pressure</td>
<td>(Pa)</td>
</tr>
<tr>
<td>(Q)</td>
<td>heat release rate</td>
<td>(W)</td>
</tr>
<tr>
<td>(R)</td>
<td>specific gas constant</td>
<td>(J kg^{-1} K^{-1})</td>
</tr>
<tr>
<td>(Re)</td>
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<td></td>
</tr>
<tr>
<td>(T)</td>
<td>temperature</td>
<td>(K)</td>
</tr>
<tr>
<td>(v)</td>
<td>specific volume</td>
<td>(m^3 kg^{-1})</td>
</tr>
<tr>
<td>(w)</td>
<td>velocity</td>
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</tr>
<tr>
<td>(a)</td>
<td>parameter for nozzle function</td>
<td></td>
</tr>
<tr>
<td>(\beta)</td>
<td>parameter for nozzle function</td>
<td></td>
</tr>
<tr>
<td>(\eta)</td>
<td>viscosity</td>
<td>(Pa s)</td>
</tr>
<tr>
<td>(\nu)</td>
<td>kinematic viscosity</td>
<td>(m^2 s^{-1})</td>
</tr>
<tr>
<td>(\kappa)</td>
<td>specific heat ratio</td>
<td></td>
</tr>
<tr>
<td>(\rho)</td>
<td>specific gravity</td>
<td>(kg m^{-3})</td>
</tr>
<tr>
<td>(\phi)</td>
<td>flow function</td>
<td></td>
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<tr>
<td>(\phi)</td>
<td>velocity function</td>
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</tr>
<tr>
<td>0</td>
<td>inlet of the panel heater</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>stagnation point inside the nozzle</td>
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</tr>
<tr>
<td>2</td>
<td>throat of the nozzle</td>
<td></td>
</tr>
<tr>
<td>(a)</td>
<td>atmosphere</td>
<td></td>
</tr>
<tr>
<td>(c)</td>
<td>critical</td>
<td></td>
</tr>
<tr>
<td>(m)</td>
<td>measured</td>
<td></td>
</tr>
</tbody>
</table>

Greek symbols

\(\alpha\) parameter for nozzle function
\(\beta\) parameter for nozzle function
\(\eta\) viscosity, \(Pa s\)
\(\nu\) kinematic viscosity, \(m^2 s^{-1}\)
\(\kappa\) specific heat ratio
\(\rho\) specific gravity, \(kg m^{-3}\)
\(\phi\) flow function
\(\phi\) velocity function

Subscripts

0 inlet of the panel heater
1 stagnation point inside the nozzle
2 throat of the nozzle
\(a\) atmosphere
\(c\) critical
\(m\) measured

References


URL cited

i) http://www.r-project.org/ (Apr. 1, 2008)
微細水滴を含む過熱水蒸気に用いた食品加熱システムのノズル噴流中の水蒸気／水比の測定

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過熱水蒸気は凝縮潜熱による高い熱伝達効率や低酸素還元気による食品の酸化を防止した加熱が行えることなど様々な利点を有するから、これまでも理械加熱、殺菌、抽出および乾燥など、様々な食品加熱への応用が研究されてきた。過熱水蒸気は被加熟物を乾燥させる性質をもつが、この性質は過熱水蒸気は鮮味や養果物のプランニングに用いる際には食材の水分を最適に保持することを困難にする一因であった。著者らは過熱水蒸気による加熱中の食材の水分を制御するため、微細な水蒸気を食材に噴霧しつつ過熱水蒸気により食品加熱を行う装置を開発した。この装置をジャガイモのプランニングに用いたところ、一般的なプランニング方法である熟処理で問題になる吸水ならびに成分の流出が抑制され、かつ乾燥による歩留り低下を抑制されることが明らかになった。この装置は様々な食品の調理加熱、プランニングなどに応用可能であると期待される。もしこの装置が食品および加工目的によって異なると考えられることから、この装置のさらなる応用のためには過熱水蒸気および水滴の霧化を制御する技術、その在フローとなる過熱水蒸気および水滴の流量を測定する技術が必要であると考える。通常は水蒸気の湿り度は乾燥量計および蒸気計から求めることができが、本装置においては水滴が多いことなどの理由から乾燥量計の適用は困難と考えられた。以上の理由から本研究においては開発された装置のノズルにおける水蒸気および水滴の流量を簡便に測定する方法を提案し、さらに装置の運転条件が水蒸気および水流量に及ぼす影響について検討することを目的とした。

開発された装置において、水蒸気発生器はアルミニウム製プレート（850×400×25 mm (WDH)）に長さ10 m、内径4 mmの細管が電熱線とともに埋め込まれた構造を有しており、細管の一端から水を圧送し高圧下で沸騰させ、細管の他端に接続されたノズルから水蒸気と水を加熱室内に噴霧する。本装置のノズル内部における水蒸気／水比を知る方法として理論的に最も単純で確実なものは、水蒸気発生器の消費電力と表面から放出される熱量の差を供給された水の比エンタルビに加え、蒸気圧から比算を出す方法であるといえる。しかしながらこの方法においては水蒸気発生器の放出熱量を正確に測定するためにの高価な熱流束センサが多数必要となり、さらに装置が大型化するために必要なセンサ数が増加するという問題がある。代替の測定方法として、本研究ではノズル内圧および温度から水蒸気流量を計算し、装置に供給される水量との差を水蒸気の流量として計算する方法を考案した。この方法では装置の構成に関わらずノズル内圧および温度のみを測定すればよいため非常に簡便であるといえるが、水蒸気と水が混在してノズルから流出する場合には正確に水蒸気流量を計算することが可能であるか明確である。この方法にて水蒸気および水流量が正しく測定できることを示されれば、費用などの面において優れた測定方法といえることから、前述の熱収支を用いる方法と測定結果を比較することによりその精度について検討した。

装置にはタンクから定量ポンプを用いて水を供給し、水の流量は0.3〜1.9 g/sとした。試験用に用いた装置においては、水蒸気発生装置は加熱室内に設置されており、消費されるエネルギーよの一部が水蒸気および水蒸気の生成に使用され、残りのエネルギ（前述の熱損失と同値）が加熱室の水蒸気の加熱に使用される。水蒸気発生装置の出力は加熱室内の温度を115℃に保つように制御し、装置を運転しノズル内圧および温度、水蒸気発生装置の消費電力ならびに水蒸気発生装置の上面面積を計算した後、ノズル侧面積を計算したAkinmaの方法により2次元補間した後、面に対して数値積分することにより求め、各側面については長手方向に5等分し中央の熱流束と表面積の積から算出した。ノズル口径は1.3、1.5および1.9 mmの3種について試験を行った。

2つの方法により測定された水蒸気／水の流量比は良
く一致し（Fig. 7），ノズル内圧および温度測定による方法は開発した加熱装置を様々な食品加工に応用する上で有効な水蒸気および水流量の測定方法であると判断された。本研究で得られた結果においてはノズルにおける水の流量の質量比は最も高い場合で供給水量の約25%であったが、これは体積比にすると水蒸気流量の1/5000以下となることから、水蒸気流量計算に水滴の存在が及ぼす影響は極めて小さいと考えられる。実験に用いた装置においては、供給水量が少なくノズル内圧が0.188 MPa以下の場合においてはノズルからは水蒸気のみが亜音速で流出し、供給水量を増加させノズル内圧が0.188 MPaを超えるとノズルから流出する水蒸気の速度は音速となり水滴がともに噴霧された。実験に用いた装置においては加熱室内の水蒸気を大気圧で115℃と過熱状態に制御するため、供給された水は全て水蒸気として加熱室内に供給されるべきである。しかしながら、水蒸気流量は水蒸気流速およびノズルのど部における水蒸気密度の積で与えられるため、供給水量が増加しノズルにおける水蒸気流速が音速に達すると、ノズル内圧の増加による水蒸気流量の増加が小さくなり、供給された水の一部は水蒸気として流れ出ることができず水滴としてノズルから噴霧されたと考えられる。