연속살균장치에서의 소고기 정육면체의 열전달특성 측정

Dynamic thermal properties of particulate foods in a continuous flow cooking system

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적 요

연속살균장치는 130℃에서 140℃의 초고온에서 연속적으로 식품을 열처리 하는 공정으로 재래 배치식 공정에 비하여 순간적인 짧은 시간이 소요되는 경제적인 공정이나, 액상과 고상으로 구성된 저산도 식품은 고상입자의 대류열전달 계수와 장치내 체류시간이 정확히 구명되지 않아서 연속살균기술이 성공적으로 적용되지 못하고 있다. 본 연구에서 연속살균장치에서의 액상식품과 고상식품사이의 대류열전달 계수를 예측하기 위하여 연속살균장치의 Hold tube에서 정육면체 모델식품내부의 온도를 측정할 수 있는 장치를 개발하였다. 연속살균장치의 홀드튜브에서 정육면체 모델 식품의 온도변화를 예측할 수 있는 유한차분법을 이용한 시뮬레이션 모델을 개발하고 소고기를 대상으로 이 시뮬레이션 모델의 입력변수인 비열, 열전도도를 실험적으로 측정하여 사용하였다. 0.0에서 15.0 centipoise의 점도를 가지는 모델 액상식품의 15.6에서 45.2 liter/min의 유속에 대하여 액상과 소고기 정육면체의 대류열전달 계수는 792에서 2107W/m^{2}K으로 예측되었다.

주요용어(Key Words): 열전도(conduction), 대류열전달(convection), 연속살균(continuous flow cooking)

INTRODUCTION

Aseptic processing is a method for thermal sterile processing of food products which move in a continuous flow through heat-hold-cool thermal processes and are then filled in sterile packages under sterile conditions. Due to ultra-high temperature used in the heating process, aseptic processing requires a much shorter time as compared to traditional preparation methods such as retorting. Therefore, it has the potential to minimize damage to food products, improve product quality, reduce energy consumption, and increase productivity. The continuous thermal sterilization process has been

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favorably used with homogeneous high acid foods as an economical and efficient means of destroying microorganisms and inactivating enzymes in foods. However, this process has not been very successful for low acid foods containing discrete particulate materials because of the problem of assuring complete sterilization.

Sterility of discrete particulate materials can be estimated by measuring temperature history of the coldest point in a particle moving through a continuous flow thermal process. To date, there is no practical method available to measure the temperature of a food particle moving through an aseptic system.

Therefore, mathematical simulation models have been popular for predicting temperature profiles of moving particles to estimate accumulated lethality during aseptic processing. Mathematical simulation models depend on input parameters such as thermal properties of both fluids and particulates, convection heat transfer coefficient between fluid and particulate, and residence time in each process. Thermal properties of both particles and fluids such as specific heat and thermal conductivity can be measured experimentally. Specific heat of sample foods can be accurately measured using Differential Scanning Calorimetry (DSC) methods (Mohsenin, 1980). Thermal conductivity of both fluids and particulates can be accurately measured using a thermal probe method as reported by Hong et al. (1998).

However, many mathematical models (de Ruyter and Brunet, 1973; Manson and Cullen, 1974; Sastry, 1986) used either infinite or assumed values for convection heat transfer coefficient because of the difficulties of determining reliable values. Since convection heat transfer at the boundary between fluids and particulates can be estimated from temperature-time profiles of fluids and particulates, several apparatus have been designed to measure the center temperature-time profile of a food particle in a fluid stream (Zuritz et al., 1987; Chandarana et al., 1988; Chang and Toledo, 1989; Alhamdan et al., 1990). Most of these studies mounted a stationary sample in a sample holder and measured particle center temperature during continuous flow heating at a constant fluid temperature. However, the effects of thermal conduction through the sample holding frame and varying characteristics of fluid flow in a real system were not considered.

The objectives of this study were:

1) to design a dynamic thermal property evaluation system to measure temperature-time profiles of a food particle and its surrounding fluid in a continuous flow cooking system,
2) to develop a finite difference model to predict the temperature distribution of a cube particle in a continuous flow cooking system using experimentally measured input parameters, and
3) to estimate the effects of viscosity and flow rate of the carrier fluid on the fluid/particulate heat transfer coefficient.

CONTINUOUS FLOW COOKING SYSTEM

An experimental apparatus was designed to simulate continuous flow cooking system. It consisted of a mixing tank, displacement pump, tube-in-shell heat exchanger, sample introducing device, sample holder, holding tubes, and a custom made flow meter. A heat exchanger for cooling processed food products and a packaging device as in a commercial aseptic system were not included. A 250-liter stainless steel mixing tank was thermally insulated with 5cm thick glass wool.

A double-diaphragm pneumatic pump with flap-valve design (Warren Rupp model SA2-A) was used because of its ability to safely handle particulate materials and high viscosity carrier fluids. A tranquilizer was placed after the pump as a surge suppression device to smooth unsteady flow. A pressure regulator and pressure tank were used to regulate supply air pressure at 552 kPa. The pump was air operated and flow rate of carrier fluid was set by adjusting the pump inlet air valve.

The Pyrex brand tube-in-shell heat exchanger consisted of a 7.62-cm I.D. by 2.44-m long outer-shell glass tube and a 5.08-cm I.D. by 3.05-m long inner glass tube. Saturated steam at 120℃ was supplied through the annulus region between the two glass tubes. The heat exchanger had an effective heating length of 2.74 m with an overall wall-to-fluid U-value of 510 W/m^2K for heating. Holding-tubes consisted of three 5.08-cm I.D. by 2.74-m long Pyrex glass tubes connected with glass tube elbows. Holding-tubes were thermally insulated with 2 cm thick glass wool. The first two holding-tube units in the fluid flow path after the heat exchanger were mounted at a slope of 2.08 cm/m to simulate commercial holding-tube installations.

A custom made flow meter was installed between the third holding-tube section and the mixing tank. It consisted of a 20-liter bucket, baffle tube, fluid level indicator, and disk valve at the bottom of the bucket which was driven by a solenoid on the top of the bucket. A control circuit board and 1.0 millisecond clock pulse generator were designed to control the flow rate measurement.

TEMPERATURE-TIME PROFILE MEASUREMENT

Figure 1 shows a sample holder which was designed to place the meat cube in the center of the flow tube while minimizing flow disturbance and heat transfer which might be caused by the sample holder frame. Brass material was formed into a cross-shaped frame of 2.0 ± 0.05 mm wide and 3.0 ± 0.05 mm thick and thirteen stainless steel needle tubes (20 gage Type 304 stainless steel #3 temper, Popper and Sons Inc.) were mounted through the brass frame. This brass frame was mounted on an aluminum structure to fit into the vertical branch of glass-T tube. Teflon insulated 40 gauge type-T thermocouples (Physitemp Instrument Inc., NJ) were installed through the stainless steel needle tubes so that the tip of each thermocouple protruded 7.5 mm from the end of each stainless steel tube. A particulate food sample was placed over the thermocouples so that only the ends of the needle tubes were in contact with the surface of the sample to minimize thermal conduction from the frame to the sample.

A sample-introducing device was designed to bypass fluid flow from a sample mounting area and was placed between the heat exchanger and holding tube section. Temperature-time profiles inside a beef cube and of fluid around the cube were measured with a CR7X measurement and control system (Campbell Scientific, Inc., UT). The data logger was interfaced to an IBM-PC compatible microcomputer through an RS-232 port for real-time display and permanent storage of temperature-time profiles and flow rates.

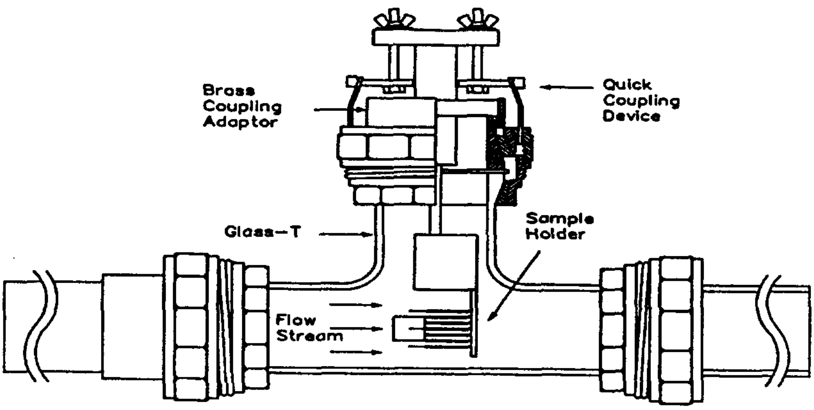

Figure 1. Schematic of the quick coupling device for the sample holder

A FINITE DIFFERENCE MODEL

A finite difference model was developed to predict the temperature distribution inside the cube particulate food in continuous flow cooking system. Since flow patterns are different at the fronted surface, parallel side surfaces, and backside of a cube with respect to flow direction, convection heat transfer rates on each side of a particulate surface will be different. However, since average value of the heat transfer rates is the value of interest for engineers in thermal process design, the heat transfer rate through each side of a cube was assumed identical. The model cube was divided into eight identical cubes as mirror images of each other to minimize computing time. Because of similarities, three-dimensional finite difference equations (FDEs) were derived using an energy balance on only four different types of control volumes (inside, side, edge, and corner). The mirror image cube was divided into nine by nine by nine nodes for a total of 729 nodes with 0.83 mm nodal distance.

Implicit solution methods have an advantage of being stable for any time step size. However, these methods involve large truncation error at every time step, although it can be minimized by choosing small time step size. Among implicit solution methods, Crank-Nicolson method gives the least accumulated truncation error (Croft and Lilley, 1977). Since in this study, parameters thermal conductivity and specific heat were influenced by temperature, Crank-Nicolson method was used to develop stable finite difference equation systems. Finite difference equations were programmed in C++ programming language (Borland International, Inc., CA).

FLUID/PARTICULATE HEAT TRANSFER COEFFICIENTS

To estimate the effects of viscosity and flow rate of the carrier fluid on fluid/particulate heat transfer coefficient h_{fp}, temperature-time profiles of a meat cube were measured at three levels of flow rate and for three levels of fluid viscosity. At each level of flow rate and viscosity, temperature-time profiles of six samples were measured. Effect of flow rate and viscosity levels on h_{fp} were tested using General Linear Models Procedures (GLM) of SAS (SAS, 1990).

Water, 1% CMC solutions, and 2% CMC solutions were used as the carrier fluid, and flow rates were around 15, 30, and 45 liter/min as shown in Table 1. CMC solutions were prepared by mixing CMC (Carboxymethyl Cellulose Sodium High viscosity, Sigma Chemical Company, MO) powder with 80°C tap water in continuous flow cooking system. Viscosity of the 1% and 2% CMC solutions were measured at 80°C using the Wells-Brookfield Cone/Plate Viscometer (Brookfield Engineering Lab., Inc., MA) with 0.8 cone spindle and were 8.4 0.66 and 15.0 0.21 centipoise, respectively. Carrier fluid temperature at the sample location was maintained around 80°C and changed less than 1°C during temperature-time profile measurement.

Table 1. Average flow rates of carrier fluids.

Carrier fluid Flow rate	Flow rate Standard deviation (liter/min)		
	Low flow rate	Medium flow rate	High flow rate
Water	15.6 0.59	29.7 0.64	45.0 0.63
1%CMC	17.3 0.34	30.3 0.19	44.9 0.12
2%CMC	18.3 0.26	29.2 0.18	45.2 0.26

The Oscar Mayer brand Beef Franks was obtained from a local grocery store. Mean moisture content and density of these beef frankfurters were 53.7 ± 0.44 % (wet basis) and 1033 ± 20 kg/m^3, respectively. Moisture content was determined using a convection oven method at 75°C for 24 hours, and density was determined using a graduated cylinder and a balance. Specific heat of the beef meat was measured continuously from 10°C to 100°C by one degree increments using a Differential Scanning Calorimetry (DSC 2920, TA instrument, Inc., NJ). Specific heat of the beef frankfurters ranged between 2729 and 3259 kJ/kgK. Thermal conductivity of the meat samples was measured using the instrument developed by Hong et al. (1998) over a temperature range of 20°C to 80°C and ranged between 0.389 to 0.350 W/mK.

To estimate h_{fp}, the temperature distribution inside a cube was computed for an initial guess value of h_{fp} and its corresponding objective function $J(h_{fp})$ was computed. This was defined as the standardized sum of squares of the difference between the calculated center temperature $Tcal(0,0,0,tn)$ and its corresponding measured center temperature $Texp(0,0,0,tn)$.

$$J(h_{fp}) = \frac{1}{N}\sum_{N=1}^{N}[T_{cal}(0,0,0,t_n) - T_{\exp}(0,0,0,t_n]^2$$

where N denotes the number of data points used in the calculations.

A minimum value of $J(h_{fp})$ was estimated using a program with the following algorithm. The value of h_{fp} was increased by step size h_{fp} while $J(h_{fp})$ decreased. When $J(h_{fp})$ increased, h_{fp} was multiplied by -1/2 and the above procedure was repeated until the absolute value of $J(h_{fp})$ was less than 1.0E-06. Then the value of h_{fp} that results in the minimum $J(h_{fp})$ value was presented as the estimated fluid/particulate heat transfer coefficient for an experiment concerned.

RESULTS AND DISCUSSION

Estimated mean h_{fp} values ranged from 792 to 2107 W/m^{2}K for beef frankfurter meat cube for three levels of flow rates and three levels of fluid viscosity as shown in Table 2. Statistical analysis showed that the fluid/particulate heat transfer coefficient is significantly different between flow rates and between viscosity levels. Estimated h_{fp} values decreased with viscosity increase, while estimated h_{fp} values increased with increase of flow rate in the 1% CMC solution and 2% CMC solution. Standard deviations of estimated h_{fp} values showed larger values in water than those in CMC solutions, which could be considered as the effects of more fluctuations in flow rates as shown in Table 1.

Table 2. Mean fluid to beef particulate heat transfer coefficient at various flow rates and in various concentrations of CMC solutio

Carrier fluid Flow rate	Heat transfer coefficient Standard deviation (W/m^{2}K)		
	Low flow rate	Medium flow rate	High flow rate
Water	1333 137	2107 691	1704 485
1%CMC	841 111	1131 273	1273 273
2%CMC	792 120	1060 392	1132 220

To check dimensional changes of a meat cube during the experimental test, its x, y, and z dimensions were measured before and after the temperature-time measurement.

During the tests, one of the dimensions increased as much as 13 percent while the other two decreased as much as five percent. As can be seen in Figure 2, measured temperature rose faster than the simulated temperatures for the first 10 seconds from the start of heating. Since heating rate of a particulate is determined by the shortest path between the surface and center point, dimensional changes of meat cubes could have caused the faster temperature rise. After 10 seconds from the start of heating, measured temperature rise showed a similar shape as the simulated center temperatures, which could be considered as no more pronounced dimensional changes occurred after the first 10 seconds. Due to the dimensional changes, the center temperature of the meat cube could rise faster than the temperature rise without the dimensional change. In this case, a larger h_{fp} value was required for the finite differential model to follow the actual center temperature-time profile. Therefore, the h_{fp} values presented in this study could have been overestimated.

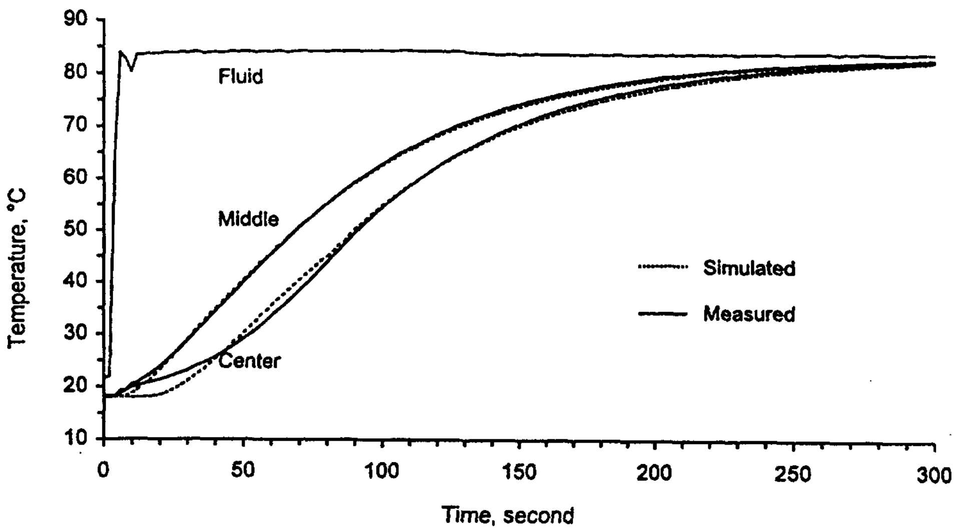

Figure 7. Simulated and measured temperature-time profiles of center adn middle points of a meat cube for h_{fp} value of 1687 W/m^2K

CONCLUSIONS

A dynamic thermal property evaluation system was designed to monitor temperature-time profiles of food particulates in a continuous flow system. The designed system was able to measure temperature-time profiles inside a meat cube and of surrounding fluid with minimum flow disturbance by the sample holder. The finite difference model was able to predict temperature distributions inside a meat cube mounted in a continuous flow cooking system. The temperature-time profiles were used effectively to estimate fluid/particulate convection heat transfer coefficient.

Estimated mean h_{fp} values ranged from 792 to 2107 W/m^{2}K for fluid flow rates 15.6 to 45.2 liter/min and viscosity from 0.0 to 15.0 centipoise. As viscosity increased, the fluid/particulate heat transfer coefficient decreased. Convection heat transfer coefficient around the meat cubes increased as flow rates of 1% CMC solution and 2% CMC solution increased. Due to dimensional change of samples, the h_{fp} values presented in this study could have been overestimated.

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