Reprinted from *J. Food Sci.*
49(2): 461-467

Sterilization of Food in Containers with an End Flat Against a Retort Bottom: Numerical Analysis and Experimental Measurements

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ABSTRACT

The heating characteristics of a metal container (filled with a conduction heating food) heated with one end flat against a retort bottom was simulated numerically using the Finite Element method. Results of heat penetration tests in a laboratory retort and in a commercial crateless retort were compared to results obtained in computer simulations. It was observed that semilogarithmic heating curves are curved under the conditions described. It was also shown that the slowest heating zone moves towards the container bottom during heating. Heat flux, which is initially directed from the retort bottom towards the can, is reversed at long heating times; i.e., heat flows from the container to the retort bottom. Suggestions are given for the handling of this type of situation in commercial practice.

INTRODUCTION

THREE PIECES OF DATA are usually required to calculate the sterilization value delivered in a thermal process: (a) the location of the lowest temperature zone (cold zone) in the retort and its relationship to the temperature measured by the retort mercury-in-glass thermometer; (b) the location of the slowest heating zone in the container, the zone which receives the smallest sterilization value; and (c) the temperature response parameter, f, and lag factor, j, of the slowest heating zone of the food product-container system.

To design the sterilization process, the heating rate parameters at the slowest heating zone of the container located at the cold zone in the retort are used with the microbial design parameters (F_0 and z-value) and process parameters (heating medium temperature, initial temperature, and cooling water temperature) to obtain the final process.

Historically, food containers have been placed in baskets in an organized or jumbled manner, with the basket, in turn, placed in a vertical or horizontal retort. The perforations in the baskets and crates were made according to industry specifications to ensure adequate contact between the heating medium and all the container walls.

To reduce the labor-intensive steps of loading and unloading batch retorts, crateless retorts were designed. In these retorting system it is possible to have a unique heating condition exist when a conduction heating food packaged in a sealed container has one of its ends flat on the retort bottom (Pflug, 1982). Containers subjected to this condition will heat more slowly than those surrounded on all sides by the heating medium. The reduction in heating rate is largely due to the buildup of a condensate layer on the retort bottom. When this condition occurs, cans are immersed in condensate rather than surrounded by condensing steam. Although several commercially available crateless retorts are equipped with false bottoms, there are still in use at the present time a large number of crateless retorts without false bottoms. In both types of retorts, inadequate condensate removal is manifested in reduced heating rates. However, since temperature mapping studies in these retorts usually show excellent heating medium uniformity, the condition described is not routinely taken into account in process design.

Conducting meaningful heat penetration studies in commercial crateless retorts during full load operation is nearly impossible. Therefore, to study this unique condition, wherein a can is heated while sitting on the retort bottom plate, a mathematical model that enables an accurate geometrical description of the nonsymmetric product-container-retort system must be employed. Neither an exact solution nor a Finite Difference solution to the governing heat transfer equations can accommodate to the geometry of the problem. Therefore, the more powerful Finite Element method must be employed.

In this study the possibility that gross underprocessing may occur in containers sitting with an end flat against a retort bottom will be evaluated. Heating rates and location of the slowest heating zone in this situation will be determined using both analytical, numerical, and experimental techniques in laboratory and commercial crateless retort systems.

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MATERIALS & METHODS

Computer

Fourier's quasiharmonic transient conduction equation for an axisymmetric body with no internal heat generation reads:

$$\frac{\partial}{\partial r}\left(rk_{rr}\frac{\partial T}{\partial r}\right)+k_{rr}\left(\frac{\partial T}{\partial r}\right)+\frac{\partial}{\partial z}\left(rk_{zz}\frac{\partial T}{\partial z}\right)=r\rho C\frac{dT}{dt} \tag{1}$$

where r and z are the axis coordinates; k_{rr} and k_{zz} are the thermal conductivities in these directions; T is temperature; ρ is density; C is specific heat; and t is time.

The initial condition in the case at hand is a uniform temperature throughout the body at Time Zero:

$$T(r, z, O) = T_o \tag{2}$$

The boundary conditions of interest in this system are either a Dirichlet type condition (a prescribed temperature, T_3, on a portion of the boundary, S_3 [Eq. (3a)] or a Cauchy condition, (convective heat transfer to a portion of the surface, S_2 [Eq. (3b)].

$$T = T_3 \text{ on } S_3 \tag{3a}$$

$$k\frac{\partial T}{\partial n}(r, z, t) = h(T - T_1); r, z \text{ on } S_2 \tag{3b}$$

where h is external heat transfer coefficient; n is direction normal to surface; and T_1 is medium temperature.

A variational Finite Element method was used to solve the governing equations. The method, and its application to thermal process calculations is described in Naveh et al. (1983). It is based

$$X=\int_V\frac{1}{2}\left[\frac{\partial}{\partial r}\left(rk_{rr}\frac{\partial T}{\partial r}\right)+\frac{\partial}{\partial z}\left(rk_{zz}\frac{\partial T}{\partial z}\right)+2r\rho C\frac{\partial T}{\partial t}\right]dv \tag{4}$$
$$+\int_{S2}X\,\frac{1}{2}[h(T-T_1)^2]\,ds$$

upon derivation of a functional (Eq. 4) that is equivalent to the governing equations which is minimized with respect to the nodal values of the body (Wilson and Nickell, 1966).

Six-noded axisymmetric quadrilateral elements were used to divide the domain modelled into a system of idealized elements. Computation was carried out on a CDC CYBER 74 Computer using software described elsewhere (Naveh, 1982).

Laboratory

Heating chamber. A cylindrical steam chamber (ϕ40 cm) with a flat bottom constructed of 3.5-mm-thick aluminum was used to simulate a flat-bottomed retort. Steam from a controlled temperature reservoir enters the upper part of the chamber through a ¾-in (1.905 cm) pipe. The unit was fitted with a stuffing box, allowing insertion of thermocouple leads. The chamber was equipped with a combined condensate-level control and pressure control. A long narrow (ϕ1.5 mm) tube, extending through the cover, was located so the inlet end could be adjusted to a few mm height above the bottom. It was fitted with an adjustable flow resistor. Before each heating run the tube was adjusted in height to maintain the desired water level in the chamber and the flow resistor was set so the pressure drop acoss it was slightly less than the difference between the desired chamber-operating pressure and atmospheric pressure. Thus, the water level was maintained by continuously removing the excess condensate. After the short come-up time period the proportional temperature controller (Honeywell, Electronic Model 15) maintained the chamber temperature to within 0.2°C.

Temperature measurement. Ecklund needle-type (CNL) thermocouples were inserted through the sides of 401 x 204 containers at 1.58 cm, 2.16 cm, and 2.54 cm (midheight) above the bottom. The thermocouple junctions were located along the container radial center line. In several of the experiments, in addition to the thermocouples inside the container and in the steam chamber above the container, a thermocouple was soldered to the container bottom, and an additional thermocouple was positioned about 1 mm below the can, immersed in the water layer trapped between the can end and chamber bottom. Temperatures were monitored using a Kaye Digistrip III Datalogger. The output was transmitted to floppy discs (WTI DMII) and analyzed using a CYBER 74 (CDC) computer with appropriate software (Naveh, 1982).

Heating studies. The 401 x 204 container, filled with thermocouples, was filled with a conduction heating starch paste and placed on the bottom of the steam chamber. Four additional cans filled with the same starch paste were placed around the thermocouple can, also with their bottom side flat against the chamber bottom. The objective was to simulate the condition occurring in a commercial situation. However, in this arrangement the exposed bottom surface area per can was larger than in the tightest container packing configuration possible in a commercial retort due to the geometrical constraints of the chamber and the Ecklund thermocouple fittings. In addition, due to the reduced heat capacitance per unit area of retort bottom, the rate of condensate formation per unit area of retort bottom was lower than in a commercial retort.

Can sitting with end on bottom. To carry out a test the cans were arranged as described above. Approximately 900 ml water were then added to the steam chamber, forming a water layer ca. 1 cm deep on the bottom. The low end of the condensate level maintaining tube was now immersed in water. After the steam chamber was closed, compressed air was introduced into the chamber, at a pressure of about 30 psig for about 20 sec. Since the pressure drop across the tube (including flow resistor) was less than 15 psia, the compressed air displaced all the water above the entrance to the condensate level tube. Steam was now introduced into the chamber through the wide ¾-in. pipe, ensuring rapid temperature come-up and pressure build-up, to maintain the condensate level at the desired depth.

Can surrounded by steam. The experiment was repeated with the can sitting on a rack well above the water level.

Mathematical modeling approaches to the heat transfer problem

The system can be modeled in two ways: (1) analytically, and (2) numerically.

Analytical model. An analytical solution can be based on either of the following treatments of the can on bottom system. (a) Assume a low surface heat transfer coefficient to the infinite slab portion of the finite cylinder to compensate for the reduced heat flow from the bottom of the retort. The infinite cylinder portion will be solved for an infinite Biot number. (b) Assume that one side of the infinite slab of the finite cylinder is insulated (semi-infinite solid). Using this assumption, some limiting values for heat transfer rates can be attained.

Numerical model. A numerical approach is necessary to describe the situation more realistically. Employing the finite element method, the system was modeled according to Fig. 1A.

The model assumes that in the worst case condition a tightly packed group of cans will be standing next to each other, all with an end flat against the retort bottom (this type of arrangement was suggested by the rust stain patterns observed on the sliding retort

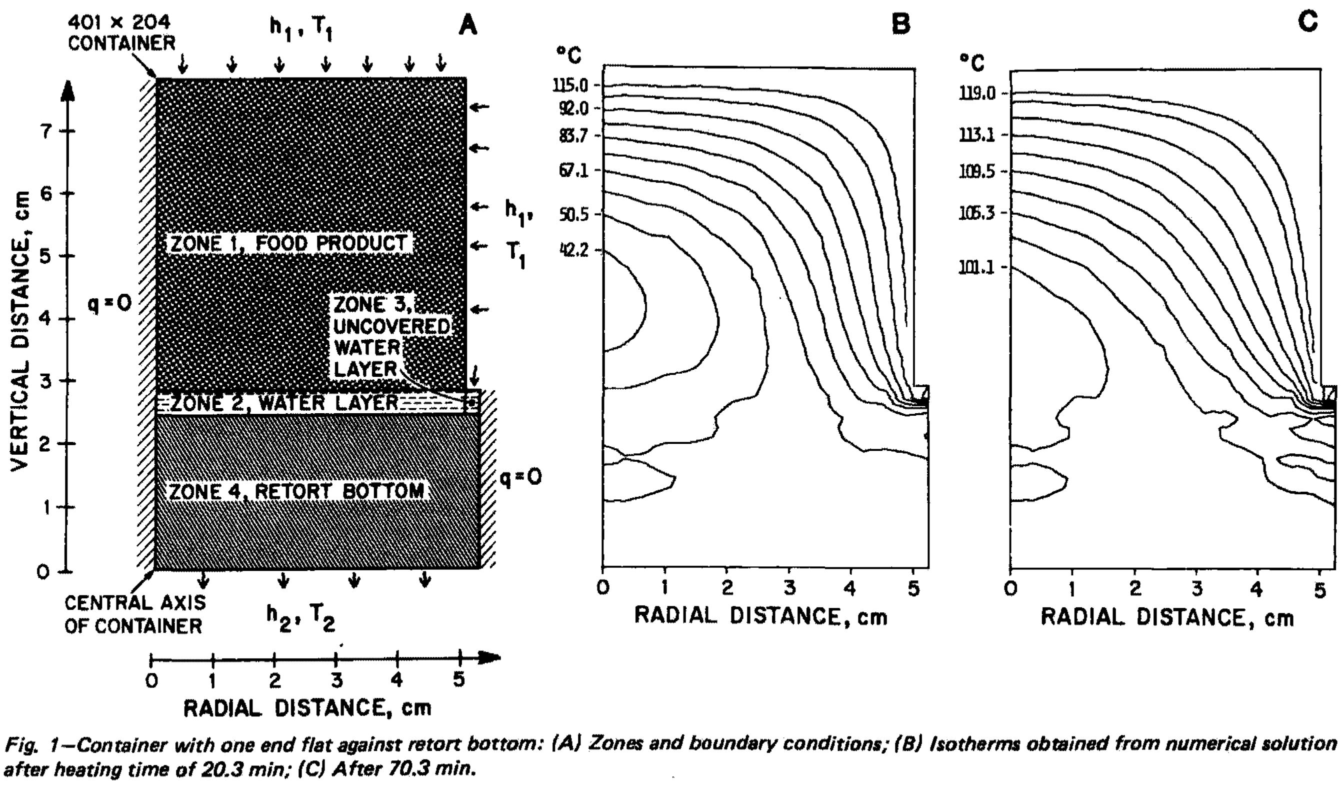

Fig. 1—Container with one end flat against retort bottom: (A) Zones and boundary conditions; (B) Isotherms obtained from numerical solution after heating time of 20.3 min; (C) After 70.3 min.

bottom in industrial systems). Using elementary plane geometry, it can be shown that the area of the retort that was not covered by cans (i.e., directly heated by steam) was 0.102 times the area covered with cans. To make the problem axisymmetric, this area was assumed to be an annulus around the container. The outside radius of the annulus was calculated so that the annular area was 0.102 times the area of the can end (in Fig. 1 the outside radius of the annulus is that of the retort bottom, Zone 4; the inside is that of the product in Zone 1).

The model also assumed that a quiescent layer of water was trapped between the flat end of the container and the retort bottom (Zone 2 in Fig. 1) with the rim acting as a dam (heat transfer along the rim was neglected). A possibility exists that small natural convection currents could develop in this water layer (Zone 2, Fig. 1), that was physically trapped in the closed cavity between the container rim, retort bottom plate and container bottom. However, since an experimental value for the heat transfer coefficient in this region was not available, introducing this possibility into the model only added an additional degree of freedom, and one more data fitting parameter to the model. As there was no benefit to further complicating the model without a physical justification, since the numerically generated data were shown to be in agreement with experimental measurements, the water layer in Zone 2 was assumed to be stagnant. Therefore, the flow of heat through this layer, which is 0.3 cm high (rim height), will occur by conduction.

Water on the noncovered surface of the retort bottom (Zone 3 in Fig. 1) was heated directly by steam in the retort. This water layer was continuously being replenished by condensate drip; heat was transferred by convection as well as by conduction. The exact flow patterns and regime in Zone 3 will depend on the retort structure, load, and drainage system, and will vary with process time (initially 15-30 gal/min condensate are typically formed in these systems and turbulence is expected; however, after long heating times much lower rates of condensate formation (< 4 gal/min) and laminar flow is expected). Overall heat transfer coefficients of 1,000 mW/cm^2°C would not be exceeded as long as the bottom area between the containers was partially wet, and would range from 100 mW/cm^2°C for slightly turbulent-to-transition regimes to 26 mW/cm^2°C for a quiescent layer (Blaisdell, 1963; McAdams, 1954). The resistance to heat transfer posed by this water layer was a major variable in the simulations.

The side and top end of the container filled with the conduction heating product (Zone 1) were exposed to condensing steam represented by a high heat transfer coefficient (h_1, T_1).

The retort bottom itself (Zone 4), a metal slab 2.74 cm thick, was cooled from beneath by ambient air, assuming natural convection (h_2, T_2).

The domain described was idealized by a system of finite elements. Two hundred eighty-five nodes were defined and connected by 124 six-noded axisymmetric quadrilateral elements with the material properties in Table 1.

RESULTS & DISCUSSION

First-term approximation analysis

The first approximation solution to the transient heat transfer equation (Carslaw and Jaeger, 1959) reads:

$$\frac{T - T_1}{T_o - T_1} = j \cdot e^{-(\beta_1^2\, \alpha t/\ell^2)} \tag{5}$$

Table 1—Thermal properties and boundary conditions used to describe the system in Fig. 1A

Zone in Fig. 1A	Heat capacitance mJ/cm^3°C	Thermal conductivity mW/cm°C
1	4,184	6.69
2	4,184	8.00
3	4,184	8.0–300.0[b]
4	3,966	200

[a] The following boundary conditions were used: h_1 = 1,000 mW/cm^2°C; h_2 = 0.58 mW/cm^2°C; T_1 = 120°C; T_2 = 30°C.
[b] This was a variable in the simulations and represents convection through the layer based on U = k/x (Blaisdell, 1963; McAdams, 1954).

where T is the unknown temperature, T_o is the initial temperature, and T_1 is the medium temperature; j is the dimensionless lag factor; α is the product thermal diffusivity; t is time; ℓ is the characteristic dimension, half the thickness of a slab or cylinder radius; and β_1 is the first root of the boundary equations for an infinite cylinder or slab.

The exponential term containing the Fourier modulus can be conveniently related to the temperature response parameter, f, and the equation roots through:

$$\frac{f\alpha}{\ell^2} = \frac{2.303}{\beta_1{}^2} \tag{6}$$

For an infinite slab and for an infinite cylinder, Eq. (6) can be written using:

$$\frac{f_c\alpha}{r^2} = k_1 \qquad \frac{f_s\alpha}{a^2} = k_2 \tag{7}$$

where f_c and f_s are temperature response parameters for an infinite cylinder and slab respectively; r is the cylinder radius, half the diameter, D; ℓ is the half thickness of a slab, half the cylinder height, L; and k_1 and k_2 are functions of the Biot number through the root equations (Carslaw and Jaeger, 1959).

From these relationships and the use of the product of solutions to describe a finite cylinder [Eq. (8)],

$$\frac{1}{f} = \frac{1}{f_c} + \frac{1}{f_s} \tag{8}$$

it can be shown that the ratio of temperature response parameter, f, of a cylinder heated from the sides, top and bottom, to the f′ of a cylinder heated only from the sides and top is:

$$\frac{f'}{f} = \left(\frac{k_1}{k_2}\left(\frac{r}{a}\right)^2 + 1\right) \Big/ \left(\frac{k_1}{4k_2}\left(\frac{r}{a}\right)^2 + 1\right) \tag{9}$$

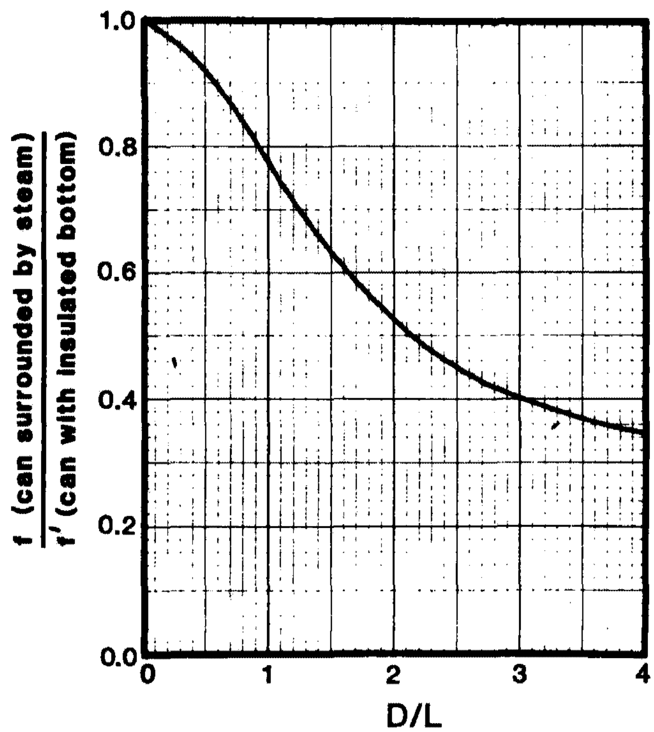

Fig. 2—Ratio of f of can surrounded by steam to f of can with insulated bottom as a function of the container length to diameter ratio. Results based on analytical solution.

For the case of a conduction heating food product heated in steam (high Biot number) $k_1 = 0.39815$ and $k_2 = 0.9332$ (Pflug et al., 1965). Fig. 2 shows that as the diameter-to-container height ratio became larger, that is, the container became more squat, the effect of the unheated bottom on the heating rate of the container became more significant.

This analysis assumes that the worst possible case occurred when the side of the container that was in contact with the flat retort bottom was insulated. The limits of the bounded temperature response parameter ratio function therefore read:

$$\lim_{r/a \to \infty} (f/f') = 1/4 \qquad \lim_{r/a \to 0} (f/f') = 1 \tag{8}$$

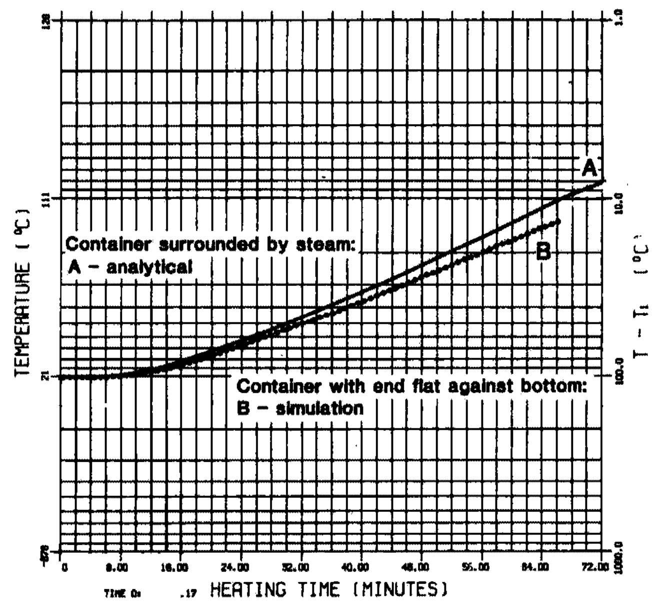

Fig. 3—Heating curves (geometric center) for 303 x 406 can filled with a water-like conduction heating product ($\alpha = 0.0016\ cm^2/sec$).

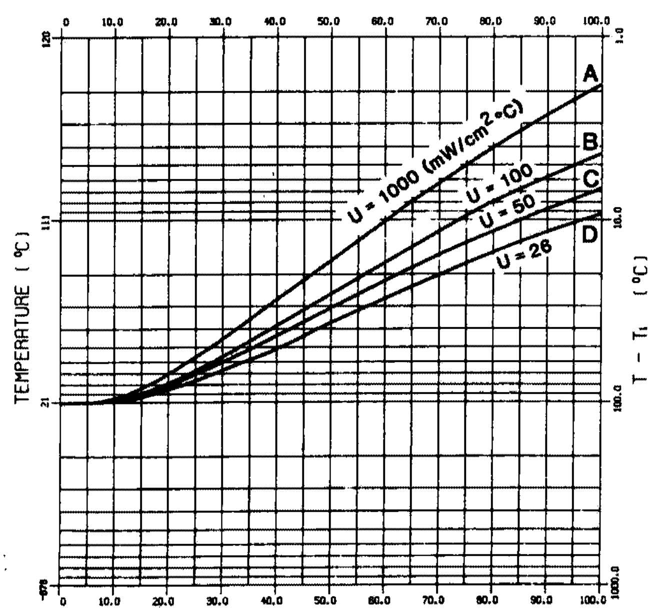

Fig. 4—Heating curves (geometric center) for 401 x 204 can with end flat against retort bottom for different values of heat transfer coefficients between steam and bottom plate.

The lower limit will be approached by containers with a very large ratio between any of the dimensions, independently of container geometry. Values of 0.251–0.27, for example, will be calculated for a conduction heat product processed in a retortable pouch flat on the retort bottom.

The value of j is a function of location and the Biot number (N_{Bi}) only. In the case of an insulated can bottom, the cold point will be located on the insulated surface which now becomes the can center from a heat flow standpoint ($dT/dr = dT/dy = 0$ @ $r = 0$, $y = -a$). Therefore, if N_{Bi} is sufficiently large (i.e., $N_{Bi} > 200$), the twofold decrease should not affect j at the new center and a value of 2.04 is expected there.

Numerical analysis

A difference of about 4 min in values of temperature response parameters (Fig. 3) was measured for a 303 x 406 container surrounded by steam and when assuming conduction through the uncovered water layer (Zone 3, Fig. 1). For the thermal diffusivity used in these simulations, a difference of over 8 min was obtained from Fig. 2 for this can size. These results indicate that in the case at hand the retort bottom was heating the can.

Comparison of the data in Fig. 3 and 4 indicates that the effect on heating rates of a can with an end flat against the retort bottom was more pronounced in a squat can, such as a 401 x 204, than in the 303 x 406 can.

A most interesting result of the numerical analysis suggests that the insulated end condition, at least theoretically, did not represent the worst case condition. Using as an example a 401 x 204 can: For an insulated can bottom the ratio f/f' is 0.535 (Fig. 2). Assuming heat transfer by conduction only through an undrained layer of water on the retort bottom (line D of Fig. 4), the ratio f/f' is 0.521. The fact that the ratio of temperature response parameters obtained using the undrained water layer model can be smaller than for an insulated bottom indicates that in this system the container was heating the retort bottom.

Using more realistic assumptions, such as moderate convection in the water layer (Zone 3), the values of the temperature response parameters (Fig. 4, Heating Curves A, B, and C) indicate that the net heat flux was from the retort bottom to the can contents for times shorter than about 80 min.

As the can contents approached heating medium temperature, a near steady state was attained in terms of heat flow to the top of the retort (bottom) plate from the can and water layer, and of heat flow from the underside of the retort plate to the surrounding air. Therefore, the temperature of the retort bottom essentially stabilized at a temperature several degrees below the heating medium temperature. By this time the direction of (net) heat flux reversed and heat began to flow from the can to the retort bottom. The descent of the cold zone towards the retort bottom (Fig. 5) at long heating times is a manifestation of this condition. Considering the dynamic boundary conditions the container experiences, the phenomenon of a moving cold zone is not surprising.

Furthermore, under these conditions there is no theoretical basis to expect straight (semilogarithmic) heating curves. Indeed, at long times, concurrent with the descent of the cold zone and the reversal of the direction of heat flow, a curvature was observed in the semilogarithmic heating curves (Fig. 4 and 6). The f-value observed at these long times was much larger than that calculated for a container with an insulated bottom, confirming the hypothesis of flow direction reversal.

Computer-generated temperature profiles in the product in a 401 x 204 can and across the retort bottom plate, assuming conduction only through the water layer in Zone

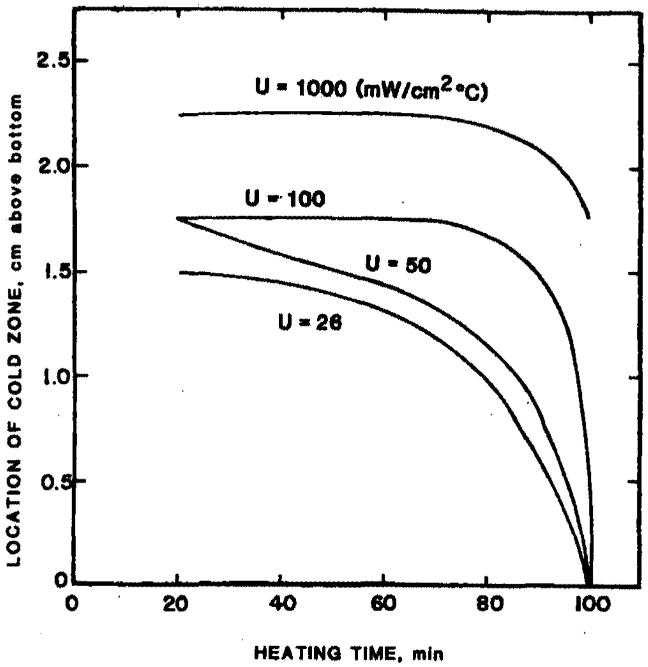

Fig. 5—Location of the moving cold zone (height above can bottom) for a conduction heating product (α = 0.0016 cm^2/sec) in a 401 x 204 can as a function of heating time and the heat transfer coefficient between steam and bottom plate. Results of computer simulations.

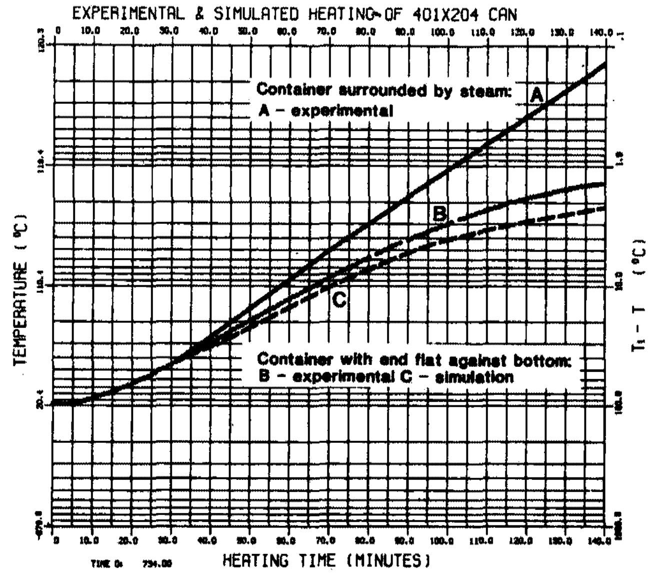

Fig. 6—Heating curves obtained in laboratory test chamber, A and B; and C, Results of computer simulations assuming a heat transfer coefficient of 50 mW/cm^2°C between retort bottom and steam.

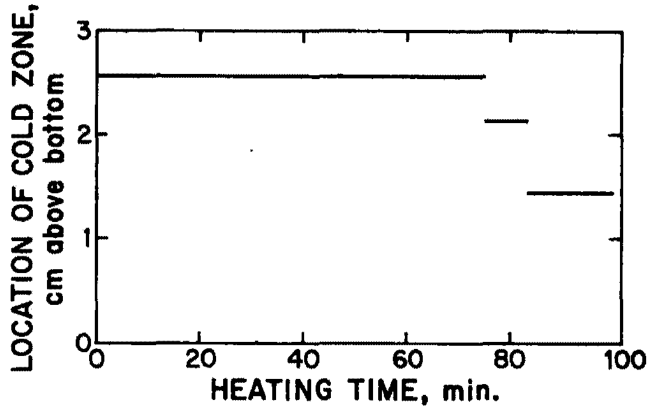

Fig. 7—Location of the lowest reading thermocouple in a 401 x 204 can filled with a starch paste heated with an end flat against the bottom of the laboratory steam chamber.

3, are shown in Fig. 1. The isotherms in Fig. 1B, after 20.3 min of heating, show that the location of the slowest heating zone (SHZ) was slightly below the midheight of the food product, along the radial line of symmetry (inside the 42.2°C isotherm). As expected, the high thermal conductivity retort bottom plate was rather uniform in temperature (no isotherms traverse it). The largest temperature gradients were observed in the uncovered water layer (Zone 3), indicating that the undrained water layer posed a major resistance to heat flow. It is interesting to note the divergence of the isotherms approximately 3 cm from the container radial centerline; forming a "valley" like shape. To the right of this opening ($3 < r < 5$ cm), the temperatures in the container were higher than in the retort bottom plate. To the left ($0 < r < 3$ cm), the temperatures in the retort bottom plate were higher than in the container. From this isotherm pattern, it can be concluded that after 20.3 min of heating, in some areas of the can-retort bottom interface ($r > 3$ cm) heat was flowing from the can to the retort bottom and, in other areas, ($r < 3$ cm) from the retort bottom to the can. The isotherm pattern in Fig. 1C, after 70.3 min of heating show the reversal of direction of heat flow in parts of the container bottom retort bottom interphase. The location of the divergence in isotherms was now closer to the container center, approximately 2-2.5 cm from the container radial centerline. Thus, the portion of container-retort interphase across which heat flow was directed from the container to the retort increased, subsequently reducing the interfacial area across which heat flow was directed upward (retort → container). The movement of the radial location of the imaginary boundary that separated these two interfacial areas towards the container center, continuously decreased the net heat flow from the retort bottom towards the container. At some point in time, the downward heat flux becomes greater than that from the retort plate towards the can. At this time the f-value will become larger than that calculated for an insulated bottom condition, and a "reversal" of the direction of net heat flow will occur. This effect can also be visualized where, after long heating times, the container becomes elongated to include the retort bottom plate.

A direct consequence of the reversal of the direction of heat flow was the change with time of the location of the SHZ. In Fig. 1B, the SHZ was roughly at the container midheight. However, in Fig. 1C, the lowest reading isotherm (101.1°C) moved to the container bottom. This phenomena is technologically significant since design of sterilization processes is based on temperatures at the SHZ, which is normally assumed to be at the geometric center in conduction heating products.

Verification of the results of the numerical model study in laboratory heating tests

The results of tests carried out to substantiate the existence of the phenomena found from the computer simulations – nonlinear heating curves, movement of cold zone, and reversal of heat flux are shown in Fig. 6 and 7.

Line B in Fig. 6 is a typical semilogarithmic heat penetration curve obtained from temperatures measured by thermocouples located at the geometric center of the 401 x 204 can of starch paste (α = 0.0015 cm^2/sec). During these experiments the condensate-level-controlling tube was adjusted to maintain a water layer of 0.3 cm. The nonlinearity of the heating curve occurred at a g (the difference between the retort temperature and the temperature at the can center) of about 3°C and was similar to the effect obtained in the computer simulations.

The movement of the cold zone location can be inferred from the results in Fig. 7. Although three thermocouples were insufficient to accurately determine the location of the cold zone, the shift after 80 min of heating in the location of the thermocouple with the lowest temperature reading from the thermocouple located at the can midheight to the thermocouple located closer to the can bottom, clearly indicated movement of the cold zone towards the container bottom.

At short heating times, i.e., 40 min, temperature readings obtained from thermocouples soldered to the bottom of the container and thermocouples located in the layer of water under the can were higher than those inside the container, indicating heat flow was directed from the retort bottom towards the container. However, at long heating times (> 70 min) these temperatures were lower than the temperatures inside the container, suggesting that heat was now flowing from the container to the retort bottom plate. In other words, a reversal of the direction of net heat flow had occurred.

The f-values of the linear portion of the heating curve obtained for the 401 x 204 can in the laboratory steam chamber ranged from 56–58 min in the several experiments, a 24% increase from the f of 44–46 min, measured in cans heated and surrounded on all sides by steam. These experimentally obtained f-values were smaller than those obtained in the computer simulations.

A test was carried out to determine whether the discrepancy in values of f obtained was due to differences between the steam chamber used in laboratory heat penetration experiments and the model assumed in the simulations, which was based on the structure of a commercially used retort. The 285-nodal point grid described earlier was modified to include a 0.35-cm-thick aluminum bottom instead of the thick steel plate. The water layer in Zone 3 now extended 2 cm beyond the can radius, with a depth of 0.3 cm, as before. An overall heat transfer coefficient of 50 mW/cm^2°C was assumed for this layer (Zone 3). Reasonable agreement was obtained between the experimental (line B, Fig. 6), thus validating the model.

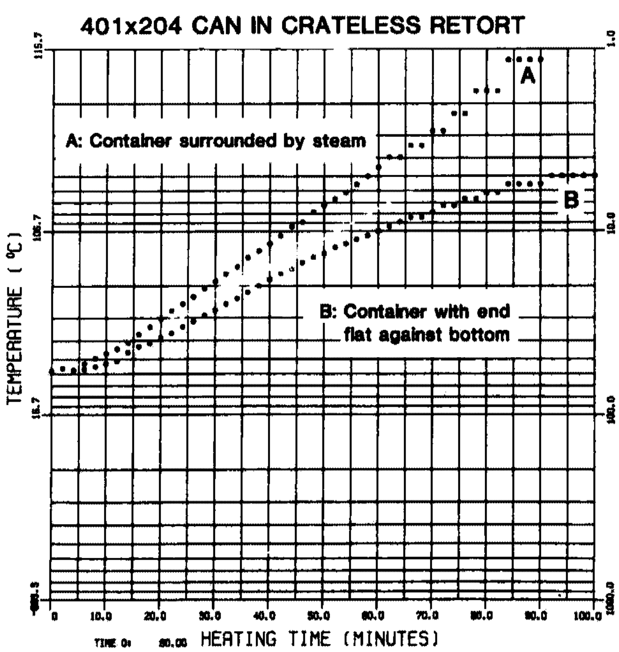

Fig. 8—Heating curves at center of 401 x 204 cans filled with salisbury steak. Data collected during full load operation of commercial crateless retort system.

Heating rates and sterilization values in commercial crateless retort systems

The numerical simulations and experiments described were necessary because it is extremely difficult to carry out controlled heat penetration studies in a commercial crateless system. Fortuitously, there was available the results of heat penetration tests of slices of meat in 401 x 204 cans held flat on the bottom of a crateless retort (Fig. 8), with the thermocouples located at the approximate geometric center of the can. These heat penetration curves had f-values and a curvature that were similar to those obtained by the finite element simulation of cans in this type of retort when an overall heat transfer coefficient through the uncovered water layer of 50 mW/cm^2°C was used. This is why the value of 50 mW/cm^2°C was selected in the computer simulation used to validate the model presented in Fig. 6, although a better fit is obtained for higher values of the heat transfer coefficient. This apparent difference between the values of heat transfer coefficient through the undrained water layer of a commercial retort during full load operation and a small laboratory heating chamber is not surprising considering the different condenstate flow pattern and exposed surface area on the bottom of these retorts. However, the fact that using a single value of h (of 50 mW/cm^2°C) for comparison of these widely differing systems gave a reasonably good fit, indicates that the assumptions made in development of the model are valid.

Table 2 shows some typical sterilization values measured during commercial operation in two types of conduction heating products. Sterilization values in the 401 x 204 containers equipped with the thermocouples that were filled with Salisbury Steak were calculated by the General Method (Patahnik, 1953), using the time-temperature data collected during the process. A threefold (2.91) reduction in sterilization value was measured for these conditions. This value (2.91) compares favorably (a 7% difference) with the threefold (3.1) reduction in sterilization value obtained in computer simulations using the physical properties shown in Table 1. To obtain an estimate of the biological F_0 distribution in crateless retorts, containers equipped with Biological Indicator Units as described in Pflug et al. (1980), were placed at the beginning, middle and end of the conveyor belt during retort loading. From Table 2, it is seen that a similar pattern in sterilization value distribution was observed: a threefold reduction in sterilization value was measured for those containers that were suspectedly sitting on the retort bottom plate during the process.

To summarize, results obtained by physical (temperature) F_0 measurements, validated by biological F_0 determination

Table 2—Sterilization values from biological indicator units (BIU) and from temperature measurements (TC) obtained in commercial crateless retort systems at different locations

Product	Can size	Can location	Measurement	F_0
Salisbury	401 x 204	Bottom	TC	6.8
		Bottom		7.3
		Center		20.4
		Center		20.7
Baked Beans	401 x 314	Bottom[a]	BIU[b]	6.0
		Bottom		7.8
		Center		15.4
		Top		18.9

[a] Position of cans on bottom is unknown.
[b] *Bacillus stearothermophilus* spores in plastic rods were used with methods as described in Pflug et al. (1980).

in a large industrial crateless retort are shown to be in agreement with data generated in a laboratory heating chamber. These three independently generated sets of data, are in turn in reasonable agreement with the results obtained from the Finite Element model developed. Thus, it is felt that the model developed adequately describes the condition of a can processed with an end flat against the retort bottom.

CONCLUSIONS & SUGGESTIONS

(1) When conduction heating foods are sterilized in retort systems where it is possible to have the end of a squat can flat against a solid retort bottom during the process, a threefold reduction in sterilization value can occur. Therefore, certain precautions must be taken in both the sterilization process design area and in retort operation to ensure that there is no public health hazard.

(2) To minimize the decrease in heating rate when the end of a container is flat against the retort bottom, the thickness of the water layer on the retort bottom plate must be small. To ensure that the water layer is small, efficient condensate drainage must be provided and its operation controlled.

(3) When this type of container-wall geometry is encountered, those carrying out the analysis of heating curves and process design should recognize: (a) the broken nature of the heating curves encountered and that, (b) the zone of least lethality will move towards the container bottom as the process proceeds.

(4) The model presented can be used to obtain heating data from numeric simulations when experimental data are unavailable. An overall heat transfer coefficient of 50 $mW/cm^2 °C$ through the uncovered water layer, in our opinion, is realistic.

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Ms received 1/23/83; revised 10/17/83; accepted 11/9/83.

This research was supported in part by a grant from the U.S.-Israel Binational Agricultural Research and Development Fund (BARD) and HHS/FDA Contract 223-79-2258. Scientific Journal Series 13,496, Minnesota Agricultural Experiment Station, St. Paul, MN 55108.