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thesis entitled

EXPERIMENTAL DETERMINATION OF THE
SURFACE HEAT TRANSFER COEFFICIENT
UNDER FOOD FREEZING CONDITIONS

presented by

Victor M. Chavarria

has been accepted towards fulfillment
of the requirements for

M. S. degree in Agricultural Engineering

A handwritten signature in cursive script that reads "D. R. Heldman".

Major professor

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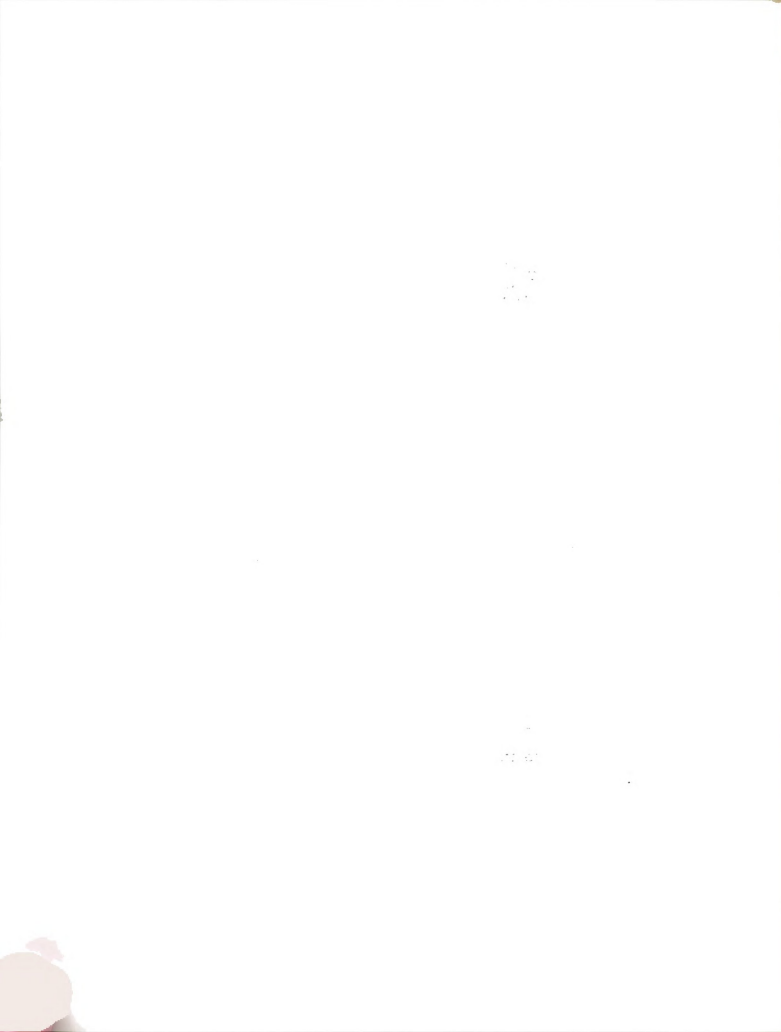
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ABSTRACT

EXPERIMENTAL DETERMINATION OF THE
SURFACE HEAT TRANSFER COEFFICIENT
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By

Victor Manuel Chavarria

The surface heat transfer coefficient is one of the limiting factors for accurate temperature fields and freezing times computation. Nevertheless, the procedures and data available for the determination of the convective transfer coefficient are scarce and unreliable in most instances. The purpose of this investigation was to empirically determine the applicability of transducer-cooling processes as an approach to measure transfer coefficients to be used as input for food freezing simulations.

Convective transfer coefficients were estimated from acrylic transducer cooling and ground beef freezing curves using nonlinear regression. The experiments were conducted in a low-speed wind tunnel using a test sample in a horizontal flat plate configuration. Heat transfer was modelled as a one-dimensional heat conduction process.

No significant differences were found at the 0.01 probability level between freezing and cooling convective transfer coefficients.

Surface Heat transfer coefficients can be successfully measured by utilizing the transducer cooling-approach given that model laws relating food freezing and transducer cooling conditions are observed.

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LIST OF SYMBOLS

Symbols

Bi	Biot Number, $h \cdot b/k_s$, (dimensionless)
b	Flat plate thickness, (cm)
C_p	Specific heat, (J/Kg °K)
e_i	Residuals, $(V_i - U_i)/\Delta T$, (dimensionless)
F	Sum-of-squares function, defined by equation (3.2.1)
h	Surface heat transfer coefficient, ($W/m^2 \text{ } ^\circ K$)
k	Thermal conductivity, ($W/m \text{ } ^\circ K$)
L	Latent heat of food product, (J/Kg)
ℓ	Characteristic length of the flat plate
m	Number of time increments for the thermal measurements
Nu	Nusselt Number, $h \cdot x/k_f$, (dimensionless)
n	Number of thermal measurement locations
P	Shape factor in equation (2.1)
Pr	Prandtl Number, ν_f/α_f , (dimensionless)
Q	Shape factor in equation (2.1)
Re	Reynolds Number, $v \cdot x/\nu$, (dimensionless)
r	Space coordinate in the y direction, y/b , (dimensionless)
s	Standard error, $(\sum_{i=1}^m e_i^2/m)^{1/2}$, (dimensionless)
T	Temperature of the solid, ($^\circ C$)
T_f	Initial freezing point, ($^\circ C$)
t	Time, (sec)

Symbols

U	Calculated sample temperature obtained from the phase-change or the cooling model, ($^{\circ}\text{C}$)
V	Experimental sample temperature, ($^{\circ}\text{C}$)
v	Air velocity, (m/sec)
x	Space coordinate in the direction of air flow
y	Space coordinate perpendicular to x
Δ	Optimum time criterion
ΔT	Maximum temperature difference, ($T_o - T_a$), ($^{\circ}\text{C}$)

Subscripts

a	Ambient, heat transfer medium
f	Fluid
i, j	
k, l	Integers
s	Solid, or frozen food product
u	Unfrozen food product
w	Fluid-solid interface
x	Denotes a variable evaluated at a given axial distance from the leading edge
o	Initial value

Greek Letters

α	Thermal diffusivity, $k/\rho C_p$, (m^2/sec)
ϵ_t	Temperature error parameter, (dimensionless)
ϵ_m	Mathematical model error parameter, (dimensionless)
δ	Operator which denotes small perturbations
λ	Eigenvalue defined by equation (3.5.2)
ρ	Density, (Kg/m^3)



ν	Kinematic viscosity, μ/ρ , (m^2/sec)
τ	Freezing time, (hr)
τ^*	Fourier Number, $\alpha t/b^2$, (dimensionless)
θ	Dimensionless temperature, $(T - T_a)/(T_o - T_a)$

1. INTRODUCTION

The freezing of food products has received considerable attention from engineers and scientists during the last decade. Its importance as a preservation method is directly related to food quality and energy consumption. The need for understanding the freezing process has fostered both experimental and mathematical analysis. In spite of the significant developments made in this food engineering area, the freezing process--phase change, effect of freezing on food quality, product structural changes, role of thermophysical properties, influence of product geometry and homogeneity--cannot be regarded as thoroughly understood subjects.

There are several factors which determine the rate at which a given food product loses heat to the freezing medium. In order of importance these factors are: (1) temperature of the freezing medium; (2) the surface heat transfer coefficient (h); (3) the size and configuration of the product; and (4) the thermophysical properties of the food product (ASHRAE, 1977a). When the transfer medium is cold air, as in air blast systems, wind tunnel freezing, and meat chilling rooms, one of the factors influencing the freezing rate is the surface heat transfer coefficient.

In the mathematical formulation of the freezing process, the heat transfer coefficient is incorporated into the convective boundary conditions coupled with the governing differential equation for the energy

balance of the food product. The influence of the surface heat transfer coefficient on time-dependent temperature fields is rather significant. Freezing time predictions can deviate considerably from true values if inaccurate transfer coefficients are chosen.

In a recent request for proposals for a survey of published surface heat transfer coefficients encountered in food refrigeration (ASHRAE, 1977a), it was indicated that inaccurate estimates of the convective coefficient could result in errors as high as 200-400% in the calculation of initial heat transfer rates. If the food freezing process is to be simulated with sufficient accuracy such that predicted values will be of practical significance, then the input data must be reliable and accurate.

The selection of surface heat transfer coefficients requires time consuming literature surveys, and numerical computation when empirical or semi-empirical correlations are used. Experiments must often be conducted for validating published or estimated values due to unreliable data matching a given set of heat transfer conditions. The procedure for determining the convective coefficient is therefore neither systematic nor practical.

The lack of a reliable and practical procedure could be attributed to such factors as:

- (1) absence of tabulated data for surface transfer coefficients reflecting commercial food freezing practices;
- (2) the number of parameters influencing the surface heat transfer coefficient (size, shape, surface characteristics, configuration and type of food product, the heat transfer medium,

product surface and transfer medium temperatures, velocity and type of flow of the medium); and

- (3) the lack of a thorough understanding of the hydrodynamic and heat transfer processes which characterize external flow problems (Lightfoot et al., 1965; Kays, 1966; ASHRAE, 1977a).

This investigation was intended to conduct an experimental determination of the surface heat transfer coefficient for food freezing conditions. Specific objectives were:

- (1) to evaluate transducer cooling experiments as an approach for estimating convective coefficients to be used for food freezing simulations;
- (2) to correlate experimental surface heat transfer coefficients to air speed and compare these correlations to published data; and
- (3) to investigate the influence of the convective coefficient on temperature fields and freezing time predictions.

This experimental approach was expected to provide insight into the limitations of published steady-state correlations for calculating transfer coefficients to be applied to convective food freezing calculations.

2. LITERATURE REVIEW

2.1 Food Freezing Simulation Models

The freezing problem has been solved analytically, for the cases of an infinite slab, infinite cylinder and a sphere with constant thermophysical properties, by a number of investigators such as Plank (1941), Hayakawa and Bakal (1974), Komori and Hirai (1974), and Golovkin et al. (1973). Numerical solutions to the problem of phase change with temperature dependent thermophysical properties have been discussed by Bonacina et al. (1973), Cleland and Earle (1977b), Comini et al. (1974), Fleming (1973), Gorby (1974), Heldman (1974), Lescano (1973) and Tarnawski (1976). Considering the lack of reliable thermophysical data of unfrozen and frozen food products, the simulation model as developed by Lescano (1973) and Gorby (1974) seems to be the most advantageous. Their finite-differences models require thermal properties of the unfrozen food product only, with thermal properties for the frozen product being determined as functions of temperature.

2.2 Influence of the Surface Heat Transfer Coefficient on Freezing Times

The influence of the surface heat transfer coefficient (h) on freezing times was analyzed by Heldman (1974 and 1975), Hsieh (1976),

Tarnawski (1976) and Lentz et al. (1977). These authors agree on the dramatic effect of the convective coefficient on freezing times, the latter being proportional to $1/h$. The freezing simulation analysis carried out by Hsieh (1976) indicated that, in the case of freezing in wind tunnels, freezing time would change negligibly for values of h greater than approximately $200 \text{ W/m}^2\text{°K}$; for h below $30 \text{ W/m}^2\text{°K}$ small reductions would dramatically increase the freezing time.

The dependence of freezing time calculations on the surface heat transfer coefficient can be analyzed using Plank's equation (Plank, 1941). The expression as modified by Mellor (1976) is:

$$\tau = \frac{\rho_s}{T_f - T_a} \left[L + C_{ps} \frac{(T_f - T_a)}{2} + C_{pu} \frac{(T_o - T_f)}{2} \frac{P \cdot b}{h} + \frac{Q \cdot b^2}{k_s} \right] \quad (2.1)$$

Plank's equation assumes constant thermal properties of the food product below the initial freezing point (T_f). Superheat removal is taken into account by the enthalpy change of the unfrozen food product.

Analyses of the influence of the surface heat transfer coefficient on freezing time calculations using numerical methods have basically yielded the same results that can be obtained from simple models such as Plank's equation (Gorby, 1974; Hsieh, 1976; Tarnawski, 1976). Although numerical models of food freezing are powerful tools for analyzing the role of critical parameters, estimation of the influence of errors in the parameters on such design variables as the freezing time is possible only through actual numerical computations using different values of the parameter of interest within a small neighborhood. The need for simple but reliable expressions or procedures to quantify the influence of the surface heat transfer coefficient on freezing time for optimum design purposes has already been emphasized by researchers in the food refrigeration field (Pflug, 1974).

2.3 Application of Transient Methods for Determining the Surface Heat Transfer Coefficient

Transient methods for estimating thermal properties and surface heat transfer coefficients have gained popularity because: (1) they require relatively simple mathematical manipulations; and (2) they can be applied to complex geometries and flow patterns for which the determination of the surface heat transfer coefficient through boundary layer theory becomes difficult if not impossible. Several investigators have resorted to experimental techniques to estimate the convective coefficient for food refrigerating processes due to lack of agreement between values calculated from published correlations (Bonacina and Comini, 1972; Cleland and Earle, 1977a), and due to unavailable data in the case of irregular geometries and complex fluid flow patterns.

The determination of the convective coefficient using nonlinear regression allows the incorporation of unsteadiness of the convective surface temperature together with temperature-dependent thermophysical properties. Therefore, insight can be provided into the influence of the nonlinear heat conduction with phase change on the surface heat transfer coefficient. Kopelman et al. (1967) and Earle (1971) employed a non-iterative transient method, but the features mentioned above were not accounted for. Both cited authors claimed that their method converged fast and that reliable mean values were obtained for their particular food freezing experimental conditions.

Bonacina and Comini (1972) used nonlinear regression to fit calculated to experimental time-temperature curves, and estimated average heat

transfer coefficients for freezing of "Tylose" (a gelatine compound used to simulate meat freezing). A two-dimensional heat transfer process in an insulated freezing chamber was simulated utilizing a finite-differences computer model. The "Tylose" sample was oriented such that the side with the largest surface area was normal to the cold air stream. The influence of inaccurate surface heat transfer coefficients on predicted temperature-history curves was illustrated. Although freezing conditions were tabulated, no reference was made to air velocity. Comini (1972) developed the nonlinear regression method further, so as to mathematically formulate the design of optimum experiments for the purpose of measuring the convective coefficient. A complete error analysis was formulated under the assumptions that thermal properties remained constant, the experimental temperature measurement error was biased rather than random, and the uncertainty in the thermophysical data was constant.

Lescano (1973) conducted experiments for determining the surface heat transfer coefficient using an aluminum transducer based on the optimum design criteria as discussed by Comini (1972). A transient lumped-parameter heat transfer process was employed to obtain heat transfer coefficients as influenced by air velocity. A mean value was obtained from the semi-logarithmic plot of dimensionless temperature of the metal transducer versus time. The experimental transducer simulation of codfish fillet freezing was based on geometric and kinematic similarity criteria. The measured convective coefficients were used as input for a finite-differences food freezing model.

A new transient method for predicting surface heat transfer coefficients was developed by Cleland and Earle (1976a). Their approach

required the measurement of surface temperatures of a transducer with a thermal conductivity closely resembling that of the food product. The test substance employed for freezing simulation was "Tylose", for which the thermal properties are well established. Both plate contact and air blast freezing transfer coefficients were determined. No reference was made to the range of air velocities associated with the air blast freezing experiments. From the application of explicit finite-differences to the convective boundary node at the surface, an expression for h was obtained as follows:

$$h = \frac{k_s}{\Delta y} \cdot \left[\frac{2T_1^m + (M - 2) \cdot T_w^m - M \cdot T_w^{m+1}}{2(T_w^m - T_a)} \right] \quad (2.2.1)$$

$$M = \Delta y^2 / (\alpha_s \cdot \Delta t) \quad (2.2.2)$$

where T_a = freezing medium temperature

T_w = measured surface temperature

T_1 = calculated temperature at node one

Equation (2.2.2) gives the estimated value of the surface heat transfer coefficient between time steps m and $m + 1$, an average value being obtainable arithmetically.

The main limitations of the method proposed by Cleland and Earle (1976a) related to:

- (1) cooling processes only;
- (2) the precooling region of the freezing curve being sufficiently long;

- (3) the isotherm center temperature remaining constant for several minutes after the onset of precooling; and
- (4) the accuracy of the method being bounded by the temperature ratio $(T_o - T_f)/(T_o - T_a) < 0.30$ and by h smaller than $70 \text{ W/m}^2\text{K}$.

The limitations of the accurate determination of the surface heat transfer coefficient utilizing equation (2.2.1) can be explained in terms of the influences of phase change and relatively high Biot Numbers (Bi) on temperature-prediction and measurement errors. The greater the surface heat transfer coefficient, the greater the Biot Number, and the faster the surface temperature responds to the freezing medium conditions. As the rate of change of the surface temperature increases, the inaccuracy of temperature measurements increases. Phase change was a second limiting factor due to the uncertainties in the calculated temperature (T_1). During the initial latent heat removal the correct prediction of thermal properties becomes a limiting factor to the accurate prediction of surface temperatures which are used for estimating the surface heat transfer coefficient.

In general, the determination of the heat transfer coefficient from time-temperature curves is an effective experimental method. The transient methods as discussed above do not require air flow assumptions of any kind. The accuracy of the estimated coefficient depends on the accuracy of both the mathematical model of heat conduction and the temperature measurements. Although the air flow need not be mathematically described, the velocity and type of flow need to be specified when reporting estimated heat transfer coefficients.



2.4 Availability and Applicability of Surface Heat Transfer Coefficients and Correlations for Calculations of Food Thermal Processes

Engineering research and design of food convective cooling or heating systems incorporate heat transfer coefficients. It has become evident to scientists and engineers in the area of food refrigeration that a high degree of accuracy in the thermophysical data does not seem necessary for thermal design. Other parameters such as contact resistances and external heat transfer coefficients are more serious sources of errors (Thermophysical Properties of Foodstuffs, 1974; Lentz et al., 1977; ASHRAE 1977a; Pflug, 1974; Bonacina et al., 1974).

It is customary for investigators to assume a value of the surface heat transfer coefficient for food freezing simulation studies, or to experimentally measure the convective coefficient if freezing time or any other design parameter is to be predicted. If the researcher resorts to published data, he will find numerous empirical or semi-empirical expressions correlating the transfer coefficient to velocity and thermal properties of the transfer medium, and the geometry and surface characteristics of the solid. The overwhelming majority of these correlations correspond to exact or approximate solutions and experimental analysis of the boundary layer flow past such simple geometries as flat plates, cylinders, and spheres (Kays, 1966; Kalinin and Dreitser, 1970; Holman, 1976; Morgan, 1975; ASHRAE, 1977c; Churchill, 1977).

Reliable published correlations are restricted to a few simple geometries and heat transfer medium flows. The availability of correlations for more complicated external flow problems, as in the case

of irregular geometries and transfer medium flows, has been hindered by the lack of a sound theory and experimental data (Churchill, 1977; Bonacina, 1972). Thus, the problem of selecting external heat transfer coefficients for food refrigeration calculations is related to the availability of such data, the applicability of published values and correlations, and ultimately the reliability and accuracy of the applicable data.

Table B-1 in Appendix B illustrates the lack of agreement, inadequate documentation and reliability of the data recommended by the cited researchers in the area of food engineering. It should become evident that there is a need for compiling, evaluating and tabulating all published values and correlations in a systematic fashion.

The inapplicability of published data to food freezing calculations might be related to assumptions made in the mathematical model. Usually the food freezing process has been modeled as a one-dimensional heat conduction problem with phase change (Hayakawa and Bakal, 1974; Iescano, 1973; Gorby, 1974; Tarnawsky, 1976; Cleland and Earle, 1977b). At the same time, it must be kept in mind that the surface heat transfer coefficient is a one-dimensional expression for describing a three-dimensional hydrodynamic-heat transfer process. The heat transfer coefficient concept arises from the need for decoupling this process into a heat conduction problem with (forced) convection at the surface of the solid, and a boundary layer flow problem with either wall temperature or heat flux obeying a given law of change with position or time (Kalinin and Dreitser, 1970). The problem of applying the heat transfer coefficient concept to a one-dimensional heat conduction model is

related to the assumption of unidirectional heat flow perpendicular to the direction of air flow along the convective surface. This assumption neglects a characteristic length of the food product geometry in the direction of air flow. It has been well established by theory and experimental evidence (Kays, 1966) that in the case of a flat plate with boundary laminar flow and a given Prandtl number, the heat transfer coefficient is velocity as well as position dependent. Air velocity and position are both measured in the direction of air flow. Therefore, with no length scale associated with the convective surface of the infinite slab, the determination of a local or a position-average heat transfer coefficient does not apply. The discrepancy between the physical problem and the mathematical modelling has given rise to disregarding published correlations, and measuring the heat transfer coefficient under local experimental conditions.

3. THEORY

3.1.1 Nonlinear Regression Formulation

In order to estimate surface heat transfer coefficients from temperature-history curves, a heat conduction model must be available for solving for the transient temperature fields in the sample being subjected to cooling or freezing. The mathematical model is represented by the predicted temperature U ,

$$U = U (y, t, h, k_s, \rho_s, c_p) \quad (3.1)$$

The nonlinear regression consists of forcing the predicted temperatures to agree with the experimental temperatures (V_i^j) by varying the surface heat transfer coefficient in a recursive manner. The measured temperatures (V_i^j) are obtained at time (j) at a given distance (i) in the direction of air flow.

The basic concept of the nonlinear regression approach consists of minimizing the sum of square errors (F) given by:

$$F(h_{\ell+1}) = \sum_{i=1}^n \sum_{j=1}^m (U_i^j(h_{\ell+1}) - V_i^j)^2 \quad (3.2.1)$$

where n is the number of positions at which the temperature is either calculated or measured, and m denotes the time at which the temperature

measurement is obtained (Bonacina and Cimini, 1972; Beck and Arnold, 1977).

In order to minimize the function $F(h_{\ell+1})$ for the value of the convective coefficient h at iteration $\ell+1$ ($h_{\ell+1}$), equation (3.2.1) is differentiated with respect to h and set equal to zero:

$$\left. \frac{dF}{dh} \right|_{h_{\ell+1}} = 2 \sum_{i=1}^n \sum_{j=1}^m \left\{ [U_i^j(h_{\ell+1}) - V_i^j] \cdot \left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell+1}} \right\} = 0 \quad (3.2.2)$$

By expanding $U(h_{\ell+1})$ in Taylor series about h_{ℓ} , and by truncating the series after the linear term:

$$U_i^j(h_{\ell+1}) \approx U_i^j(h_{\ell}) + \left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell}} \cdot \delta h_{\ell} \quad (3.2.3)$$

By estimating $\partial U/\partial h$ using a finite increment in h_{ℓ} , such that:

$$\left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell}} \approx \frac{U_i^j(h_{\ell} \cdot (1 + \gamma_{\ell})) - U_i^j(h_{\ell})}{\gamma_{\ell} \cdot h_{\ell}} \quad (3.2.4)$$

where γ_{ℓ} is a small number less than 1. Equation (3.2.4) gives an estimate of the sensitivity coefficient at time (j).

By assuming further that:

$$\left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell+1}} \approx \left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell}} \quad (3.2.5)$$

By combining equations (3.2.2) and (3.2.3):

$$\sum_{i=1}^n \sum_{j=1}^m \left\{ \left[U_i^j(h_{\ell}) + \left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell}} \cdot \delta h_{\ell} - V_i^j \right] \cdot \left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_{\ell}} \right\} = 0 \quad (3.2.6)$$



and solving for δh_ℓ :

$$\delta h_\ell = \frac{\sum_{i=1}^n \sum_{j=1}^m \left\{ \left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_\ell} \cdot [V_i^j - U_i^j(h_\ell)] \right\}}{\sum_{i=1}^n \sum_{j=1}^m \left(\left. \frac{\partial U_i^j}{\partial h} \right|_{h=h_\ell} \right)^2} \quad (3.2.7)$$

The new value of h is calculated from:

$$h_{\ell+1} = h_\ell + \delta h_\ell \quad (3.2.8)$$

The procedure is initiated by selecting an initial value of the surface heat transfer coefficient (h). By resorting to equations (3.2.4), (3.2.7 and (3.2.8) the convective coefficient is updated until $|\delta h_\ell / h_{\ell+1}| < \varepsilon$, where ε is a convergence tolerance error. If the summation is performed for all times (j), from $j = 1$ to $j = m$, then a time-average value of the heat transfer coefficient is obtained. If instead, the summation over time is carried out in a discrete fashion, from $j = \ell' \cdot \Delta t$ to $j = (\ell' + 1) \cdot \Delta t$ (where Δt is the time interval for temperature measurements and ℓ' is an integer), the coefficient obtained can be regarded as a time-dependent parameter (Bonacina and Comini, 1972).

A three-dimensional heat conduction model is required for the accurate estimation of local surface heat transfer coefficients. The difficulty associated with a multi-dimensional model is related to the calculational scheme required for specifying local values of the convective coefficient at nodes other than those at which temperature values were obtained. It then becomes necessary to resort to some means of allowing for a continuous or discrete variation of the coefficient over the convective surface. Local coefficients are more readily obtained if



a one-dimensional heat conduction process can be assumed. If so, a unique coefficient can be estimated from each local time-temperature curve for a given cooling or freezing condition.

3.1.2 Error Analysis of the Nonlinear Parameter Estimation

The nonlinear regression procedure for estimating surface heat transfer coefficients requires the minimization of the sum-of-squares function (equation (3.2.1)). The latter will usually not be minimized to zero due to experimental errors associated with the time-temperature curves, and due to the inaccuracy of the predicted temperatures. Comini (1972), based on Beck's developments in thermal properties estimation (Beck, 1966; 1967 and 1969), suggested a procedure for analyzing the influences of the uncertainties in the measured and calculated temperature-history curves, ϵ_t and ϵ_m respectively, on the determination of the surface heat transfer coefficient. The derivations of the error analysis parameters (ϵ_t , ϵ_m), and the optimum time criterion (Δ) for thermal measurements are presented in Appendix A.

3.2 Mathematical Cooling Model

The analytical solution to the cooling problem of an infinite slab with convection at one surface and adiabatic conditions at the insulated surface was obtained from Myers (1971). The formulation of the mathematical problem required the assumptions:

- (1) The thermophysical properties of the solid (transducer) are constant.

- (2) The surface heat transfer coefficient is constant.
- (3) The solid is homogeneous and isotropic.
- (4) Radiation losses from the convective surface are small compared to the convective heat flux at the surface.

The mathematical statement of the problem is:

$$\theta_{rr} = \theta_{\tau^*} \quad (3.3)$$

$$\text{with the initial condition: } \theta(r, 0) = 1 \quad (3.4.1)$$

$$\text{and the boundary conditions being: } \theta_r = 0 \text{ at } r = 0 \quad (3.4.2)$$

$$-\theta_r = H \cdot \theta(1, \tau^*) \text{ at } r = 1 \quad (3.4.3)$$

where $r = x/b$

$$\tau^* = \alpha_s t / b^2$$

$$H = h \cdot b / k_s$$

The series solution to the problem is given by (Myers, 1971):

$$\theta(r, \tau) = \sum_{n=1}^{\infty} \frac{2 \sin \lambda_n \cdot \cos \lambda_n r}{\lambda_n + \sin \lambda_n \cdot \cos \lambda_n} \cdot e^{-\lambda_n^2 \tau^*} \quad (3.5.1)$$

The eigenvalue condition is:

$$\lambda_n \tan \lambda_n = H \quad (3.5.2)$$

The exact solution together with its eigenvalue condition were implemented into a nonlinear regression FORTRAN IV program in order to estimate surface heat transfer coefficients from experimental cooling curves.



3.3 Phase-Change Finite-Differences Model

The freezing simulation model developed by Lescano (1973) was incorporated into the nonlinear regression computer program. The cited author solved the one-dimensional heat conduction problem with phase change and temperature-dependent thermophysical properties of the food product under a convective boundary condition. The mathematical formulation of the freezing problem is:

$$\rho \cdot C_p (T) \frac{\partial T}{\partial t} = \frac{\partial}{\partial y} [k (T) \frac{\partial T}{\partial y}] \quad (3.6)$$

$$\text{with the initial condition: } T (y, 0) = T_0. \quad (3.7.1)$$

and the boundary conditions being:

$$\left. \frac{\partial T}{\partial y} \right|_{y=0} = 0 \quad \text{at } y = 0 \quad (3.7.2)$$

$$-k (T) \cdot \left. \frac{\partial T}{\partial y} \right|_{y=b} = h \cdot (T (b,t) - T_a) \quad \text{at } y = b \quad (3.7.3)$$

where $C_p (T)$ and $k(T)$ are the temperature-dependent specific heat and thermal conductivity of the food sample respectively. Lescano (1973) applied the Crank-Nicolson approach to the finite-differences formulation of the heat conduction problem. The relevant assumptions considered were:

- (1) Heat conduction is a one-dimensional process.
- (2) The food product is a heterogeneous material with temperature-dependent thermal conductivity and apparent specific heat below the initial freezing point.

- (3) The food is an ideal binary solution. Below the initial freezing point, free water from the solution is solidified into ice, the concentration effects of freezing being described by the freezing point depression of the solution.
- (4) Moisture loss from the convective surface is neglected.
- (5) The initial food product temperature distribution is uniform.
- (6) The thermal properties of the food product are constant above freezing.
- (7) The surface heat transfer coefficient is constant.

The advantageous features of Lescano's freezing model are related to the automatic generation of thermal conductivity and apparent specific heat of the food as functions of temperature below the initial freezing point. Only the thermal properties of the unfrozen product are required as input. The influences of the percent of unfreezable water and density of the frozen product were also accounted for in the freezing model.

The input parameters into the finite-differences freezing model are:

- (1) computing time interval;
- (2) y-step size;
- (3) thermal properties of the unfrozen product;
- (4) surface heat transfer coefficient;
- (5) freezing air temperature;
- (6) food sample thickness;
- (7) food product initial temperature;
- (8) initial moisture content;



- (9) initial freezing point; and
 - (10) percent unfreezable water and frozen food density, if known
- (Lescano,1973).

Lescano's finite-differences computer package was incorporated into the nonlinear parameter estimation program in order to calculate surface heat transfer coefficients from ground beef freezing curves.

3.4 Assumptions Relevant to the Analysis of Surface Heat Transfer Coefficients

The analysis of the surface heat transfer coefficient obtained through nonlinear parameter estimation was carried out under the assumptions listed below:

- (1) Fluid:
 - (a) stabilized velocity and temperature fields
 - (b) constant fluid thermal properties
- (2) Solid:
 - (a) unidirectional heat conduction perpendicular to the direction of air flow
 - (b) constant thermal properties in the case of the acrylic transducer
 - (c) perfect adiabatic conditions, no lateral heat losses
- (3) Fluid-solid interface:
 - (a) There is a local effect of the position in the direction of air flow on the convective surface temperature, but heat conduction in the solid in this direction is negligible.



- (b) A local surface heat transfer coefficient can be determined from a local measured temperature-history curve.
- (c) The surface transfer coefficient is constant throughout cooling or freezing heat transfer processes.

Although no assumptions concerning the fluid are required in order to perform nonlinear regression calculations, the assumptions listed under (1) were considered relevant to the development of correlations between the Nusselt and the Reynolds Number. Those assumptions under (2) were required in order to facilitate the formulation of the mathematical linear problem of heat conduction in the case of the transducer cooling. With regards to the food freezing problem, the heat conduction equation becomes nonlinear because of the temperature-dependent thermal properties. The third type of assumptions (3) allowed for the simplification of a real three-dimensional heat transfer process to a one-dimensional formulation of the problem. Therefore, surface heat transfer coefficients estimated from transient temperatures can be considered local to the axial position at which the temperatures were obtained. The assumption of a constant heat transfer coefficient was required to obtain the exact analytical solution to the cooling problem.



4. EXPERIMENTAL

The purpose of the experimental design was the measurement of transient temperatures during cooling and freezing processes over a range of thermal and air velocity conditions. The initial and boundary conditions were chosen such that they would reflect those accounted for in the mathematical models of the one-dimensional heat transfer problems of cooling and freezing.

4.1 Apparatus

A flat plate was mounted behind a sharp leading edge in the test section of a low-speed wind tunnel in forced convection conditions as illustrated in Figure 1 . The sides of the plate were insulated with Styrofoam so as to ensure one-dimensional heat transfer normal to the air flow and prevent heat losses through the sides. The flat plate support, shown in Figure 2, was designed so that the latter could serve both as insulator of the sides and as a holder of the sample in the test section. The dimensions of the flat plate were 25 cm x 1.9 cm x 17 cm.

An acrylic plate sheet 1.9 cm thick was used to construct the flat plate transducer for the cooling experiments. The food freezing experiments were conducted with ground beef samples. The food sample holder had the same geometry and dimensions as the acrylic transducer. The

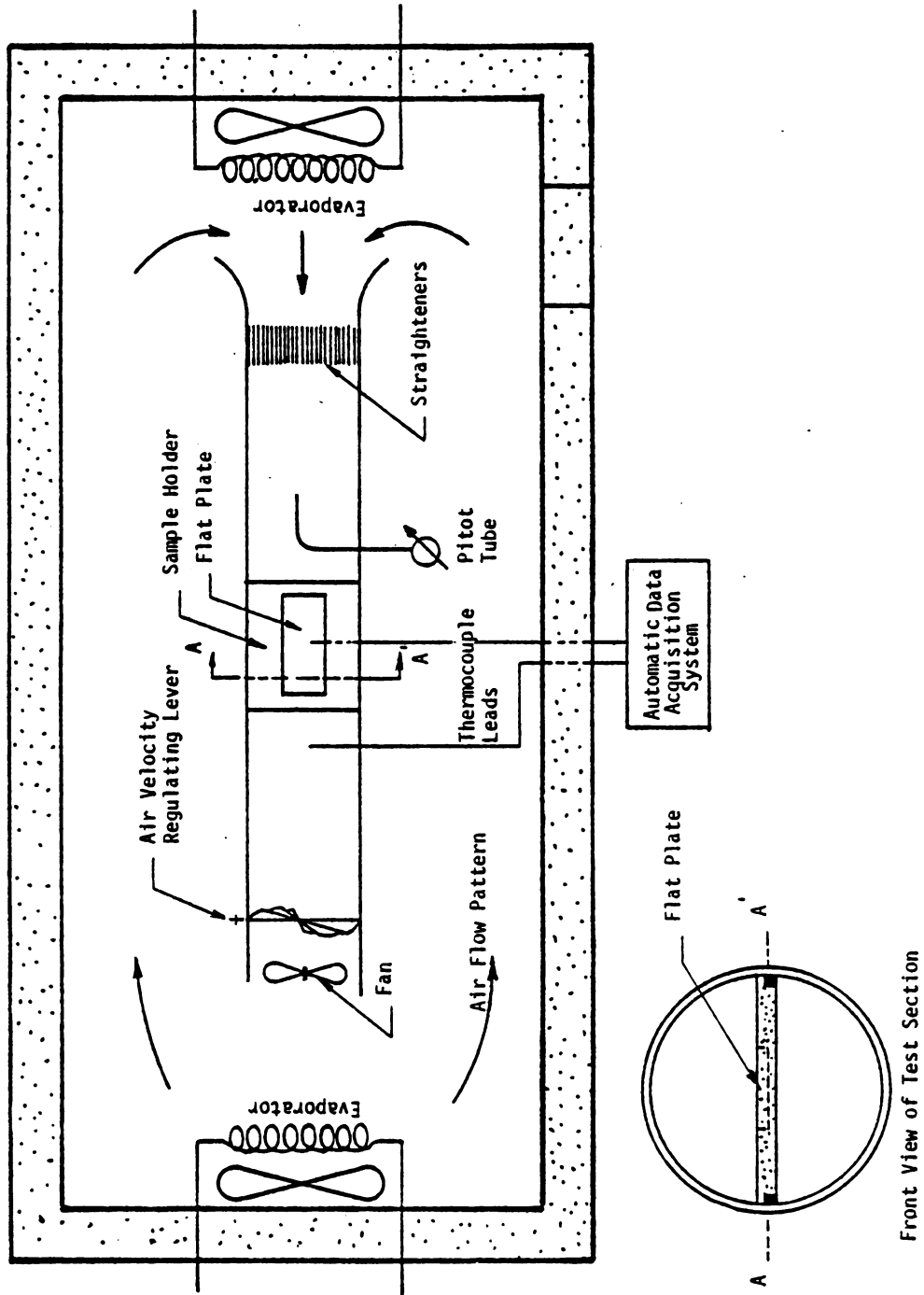


Figure 1. Schematic Diagram of the Experimental Apparatus for Cooling and Freezing Temperature Measurement.

space occupied by the acrylic plate was made available for holding the ground beef sample. The walls enclosing the food product were 0.317 cm thick bakelite sheets which were used to ease the preparation of the food sample, to imbed the bottom thermocouple junctions, and to serve as insulating walls. Temperature differences were sensed with 30-gauge copper-constantan thermocouple wires. Grooves, 1.5 ± 0.5 mm deep, running perpendicular to the direction of air flow were machined on the top and the bottom surfaces of the acrylic plate. Five thermocouples spaced every 5 cm, with the first junction positioned 0.5 cm from the plastic edge, were cemented along the surface grooves. This thermocouple installation, which is illustrated in Figure 2, allowed for the measurement of two temperatures at five positions in the direction of flow. For a given position, two thermocouple junctions were used to monitor temperatures at the surface and at the insulated bottom of the acrylic plate.

The schematic presented in Figure 2 also illustrates the thermocouple installation designed to measure temperatures of the food sample. In this case, four thermocouple pairs were spaced every 6 cm in the direction of air flow. At each location, the two measurements corresponded to the surface and the bottom temperatures of the food sample. While the bottom thermocouple wires were cemented on the bakelite sheet, which insulated the bottom of the food, the surface thermocouple wires were installed once the sample holder was filled with ground beef.

The radial thermocouple configuration was adopted in order to reduce temperature measurement errors due to the presence of the wires in the acrylic solid. In the radial configuration the wires are cemented



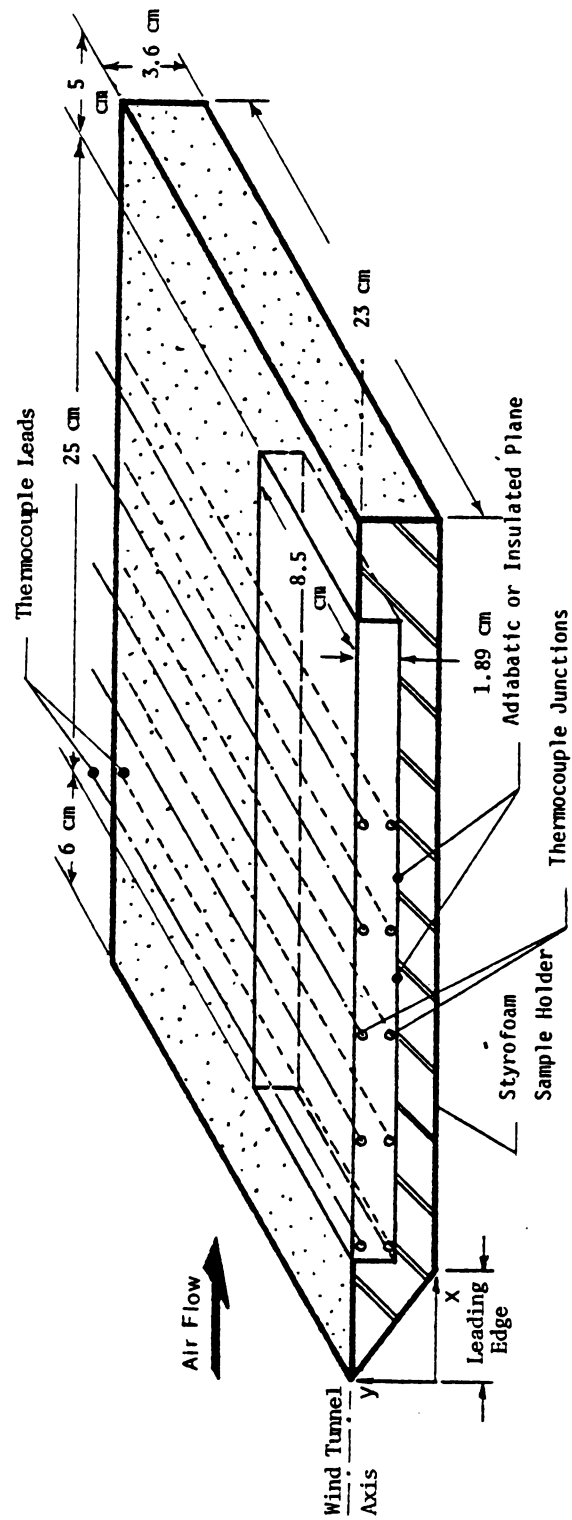


Figure 2. Schematic Diagram of Flat Plate - Sample Holder Design.

along isothermal lines which run perpendicular to the x-y plane shown in Figure 2. The heat storage capacity of the thermocouple wire rather than the thermal conductivity becomes the predominant factor influencing transient temperature measurement errors (Pfahl, 1966). Bare wires were cemented just below the acrylic surfaces, while Teflon- covered wires were installed on the food sample surface. Free stream air temperature was monitored by a set of two copper-constantan thermocouples positioned just upstream and above the flat plate.

A digital data acquisition system (Esterline Angus model PD 2064) was used to record and print the thermocouple signals, the system being sensitive to ± 0.09 °C. The air speed in the wind tunnel test section was monitored with a Pitot tube and a micromanometer accurate to ± 0.254 mm of water. The Pitot tube was positioned 40 cm upstream from the leading edge.

The wind tunnel used in the experiments was operated in a closed-circuit mode, with air being recirculated in the low-temperature room. The turbulence intensity was calculated by estimating the percentage ratio of the mean to the standard deviation of 30 randomly spaced air velocity measurements. The turbulence intensity of the air flow was about 4% at the maximum. The wind tunnel was 3.8 m long, with a circular cross-section of 46 cm in diameter. The test section was located in the center of the wind tunnel. Air was pulled into the tunnel by a blower driven by a 3.73 KW, 440/220 volt, three-phase electric motor.

The freezing room was 6.7 m x 1.8 m x 2.4 m. The refrigeration was provided by two evaporators with capacity to reduce the room temperature to -34.4 °C within a deviation of ± 1.8 °C when the wind tunnel was not

in operation. The thermal perturbation in the test section was three-fold with the tunnel operating due to the air turbulence and motor heat dissipation.

4.2 Cooling Experimental Procedure

The influence of three variables, air speed (v), flat plate thickness (b), and freezing medium temperature (T_a) on the surface heat transfer coefficient were observed during the experimental investigation. A unique combination of the three variables mentioned above defined one particular set of kinematic and thermal conditions for a given cooling run. The complete set of combinations of v , b , and T_a is shown in Table 1.

Each experimental cooling test was characterized by a set of sequential steps. These included:

- (1) setting the temperature of the refrigeration room to the desired level at least 20 hours before the experiment;
- (2) allowing for sufficient time for the acrylic transducer to reach thermal equilibrium with a room temperature of 22.0 °C;
- (3) covering the convective surface of the flat plate with a 2.5 cm thick Styrofoam sheet for insulation;
- (4) turning on the data acquisition system and keying in the channel and the scanning intervals;
- (5) opening the tunnel test section and positioning the flat plate in the test section with the sharp leading edge facing the incoming air and the convective surface parallel to the direction of air flow (as illustrated in Figure 2);

Table 1. Temperature of Cooling Medium for Various Air Speeds and Transducer Thicknesses.

Air Temperature, (°C)						
Air Speed, (m/sec)						
Natural Circulation	1.6	3.4	6.9	10.3	14.	Thickness (cm)
-17.8		-17.8		-17.8	-17.8	0.945
-28.4	-28.4	-20.	-20.	-20.	-20.	1.89
			-28.4			

Table 2. Temperature of Freezing Medium at Various Air Speeds for Food Freezing Experiments.

Air Temperature, (°C)							
Air Speed, (m/sec)							
Natural Circulation	1.7	3.4	5.0	7.4	10.0	11.3	15.2
-21.3	-28.0 ⁺	-22.4	-26.9 ⁺	-21.1	-27.5 ⁺	-17.9	-19.1
				-19.9		-19.5	

⁺The experiments run at these temperatures were performed with the food product surface directly exposed to the air.

- (6) passing the thermocouple wires through a 2 cm orifice located 30 cm behind the test section in the tunnel wall and matching the copper-constantan connectors of the transducer to those wired to the acquisition system;
- (7) removing the insulation from the flat plate;
- (8) closing the test section of the tunnel; and
- (9) switching on the electric motor to start the experiment.

The time duration of the cooling runs was selected as the time required for the dimensionless temperature to reach a value of approximately 0.1. The optimum time duration is that period of time for which the temperature data contributes to the determination of the convective coefficient with the smallest uncertainty. Comini (1972) illustrated that for this optimum time the dimensionless temperature depends on the Biot Number to a great extent. For Biot Numbers greater than 3.0, the optimum time duration becomes constant and the dimensionless temperature is about 0.37. Even though the present experimental time durations were greater than the optimum ones, the actual data used in the nonlinear procedure corresponded to the optimum time criterion. In the case of food freezing, the optimum time durations will be seen to exceed those for cooling.

4.3 Food Freezing Experimental Procedure

The procedure adopted for this part of the investigation included essentially the same sequential steps identified for the cooling experiments.

Approximately 870 grams of ground beef were used for each freezing experiment. Samples were weighted before and after each run. The food sample holder was filled with the product which was spread and leveled so as to procure a continuous uniform surface to be exposed to the cooling air. Next the food sample holder was wrapped with a Polyethylene film to prevent moisture losses from the product surface and to procure a smooth convective surface. The remaining steps are identical to those outlined for the cooling case.

Each freezing run was characterized by a given combination of air speed (v), freezing medium temperature (T_a), and moisture loss condition from the surface. The complete set of experimental conditions is summarized in Table 2.

Moisture and fat content analyses were conducted following the procedures outline by the AOAC (1965). The densities of the frozen and unfrozen ground beef samples were required as input variables to the phase-change simulation model. This property was measured by averaging 10 weight samples of food product portions of known volume. The experimental results for each ground beef sample are presented in Table C-3 in Appendix C.

The influence of moisture loss from the product surface was considered. Experiments with food samples without Polyethylene film wrapping were conducted and differences in weight loss were compared to experimental runs under the same kinematic and thermal conditions but having the food samples film wrapped. It should be emphasized that the coupled influence of both mass diffusion and surface roughness on the surface heat transfer coefficient was neglected.

The time criterion adopted to stop the freezing runs was similar to the one used for the cooling experiments. The optimum time criterion as evaluated from cooling time-temperature curves does not apply to the food freezing situation. The reasons of its inapplicability are discussed in Section 5.2.3.

4.4 Remarks on Cooling and Freezing Procedures

Transient temperatures were monitored at discrete time intervals. Typical scanning intervals used were 30, 40 and 50 seconds, depending on the air speed and the transfer medium temperature. The copper-constantan time constant, two seconds approximately, was estimated from the wire manufacturers charts (Omega Engineering, Inc., 1978).

Air velocities were monitored throughout the cooling and freezing experiments. In order to calculate mean velocities and turbulence intensities, 30 sample readings of the pressure drop across the micro-manometer at random increments of time were taken after a reasonable period of time following the onset of the heat transfer process. The stabilization period of the air velocity at temperatures below -22.0°C was significantly different than that at temperatures above -22.0°C . The wind tunnel was operated for sufficient time for the motor to warm up before installation of the sample in the test section. At the same air velocity setting the air velocities obtained at temperatures below -22.0°C were two to three orders of magnitude smaller than those velocities obtained at temperatures above -22.0°C . This effect was attributed to the increased frictional resistance of the motor shaft caused by a frozen lubrication.

The procedure as adopted for both cooling and freezing runs introduced a time delay uncertainty into the time-temperature data. The thermal and hydrodynamic processes were not started simultaneously. The heat transfer process was estimated to lead the air convection by 5 to 10 seconds. This error source was unavoidable due to the physical characteristics of the experimental setup. The uncertainty associated with the time-temperature data influenced the determination of the surface heat transfer coefficient. This influence was analyzed numerically and results are discussed in Sections 5.1.3.1 and 5.1.3.2.

5. RESULTS AND DISCUSSION

The raw data obtained from transducer cooling and ground beef freezing experiments consisted of localized time-temperature curves and air velocity measurements. A nonlinear regression computer program was developed to analyze the experimental data in order to estimate surface heat transfer coefficients from each temperature-history curve. In order to confirm the axial distance and air velocity dependence of the surface heat transfer coefficient, empirical correlations were developed. By confirming these correlations, the accuracy of the experimental transducer method was validated. The equality of the convective coefficients related to acrylic transducer cooling and ground beef freezing processes was statistically tested.

5.1.1 Experimental Cooling Curves

Five local time-temperature curves were obtained from each cooling curve from thermocouple junctions imbedded just below the convective surface of the acrylic transducer. A typical set of experimental curves was plotted in Figure 3. Inspection of the plot indicates that the rate of exponential temperature decay decreases as the distance from the leading edge increases. For the particular set presented in Figure 3, large temperature differences occur mostly for $\tau^* < 0.65$, whereas for

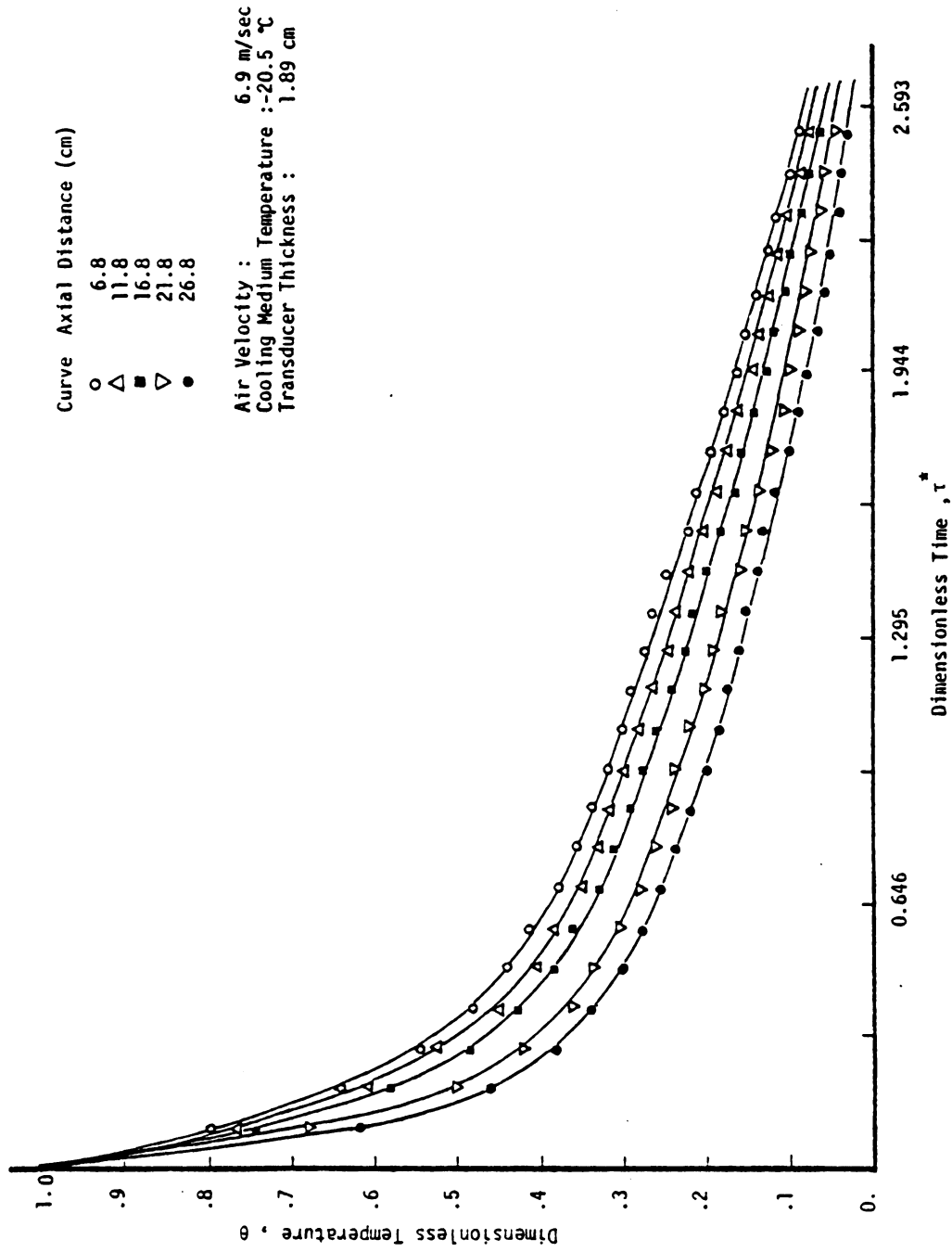


Figure 3. Experimental Time-temperature Data for Acrylic Transducer Cooling.

$\tau^* > 0.65$ the differences are small and tend to zero as the dimensionless time (τ^*) progresses.

5.1.2 Analysis of the Surface Heat Transfer Coefficient--Acrylic Transducer Cooling Results

The estimated surface heat transfer coefficients and dimensionless standard errors associated with each time-temperature curve are summarized in Table C-1 in Appendix C. Since the minimization of the sum-of-squares function (equation (3.2.1)) was performed for the entire cooling process, the transfer coefficients obtained were regarded as time-average values. The analysis of the relationships of the convective coefficient to cooling air velocity (v) and axial position (x) follows.

5.1.2.1 Influences of Air Velocity and Axial Distance

Insight into the influence of the axial distance on the convective coefficient was provided by plotting $\log(h)$ versus $\log(x)$ as illustrated in Figure 4. Inspection of the results indicates that a linear relationship exists between the two logarithmic variables if the air velocity is held constant. The influence of air velocity on the convective coefficient for a given axial distance can be established by plotting $\log(h)$ versus $\log(v)$ as shown in Figure 5. Analysis of the logarithmic plots suggests a power law relationship between the convective coefficient and air velocity or axial distance. A regression

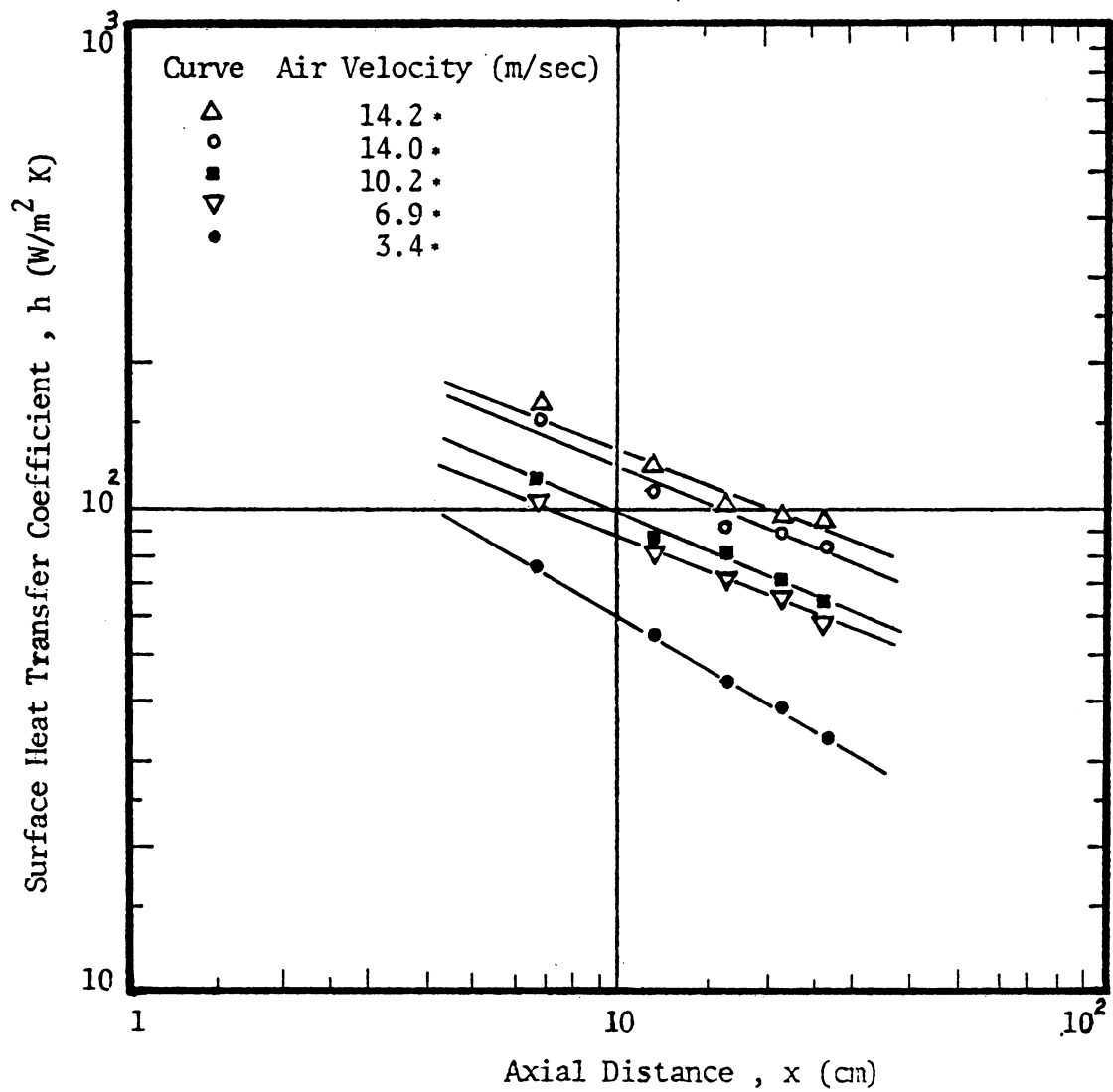


Figure 4. Surface Heat Transfer Coefficient versus Axial Distance - Acrylic Transducer Cooling Results.

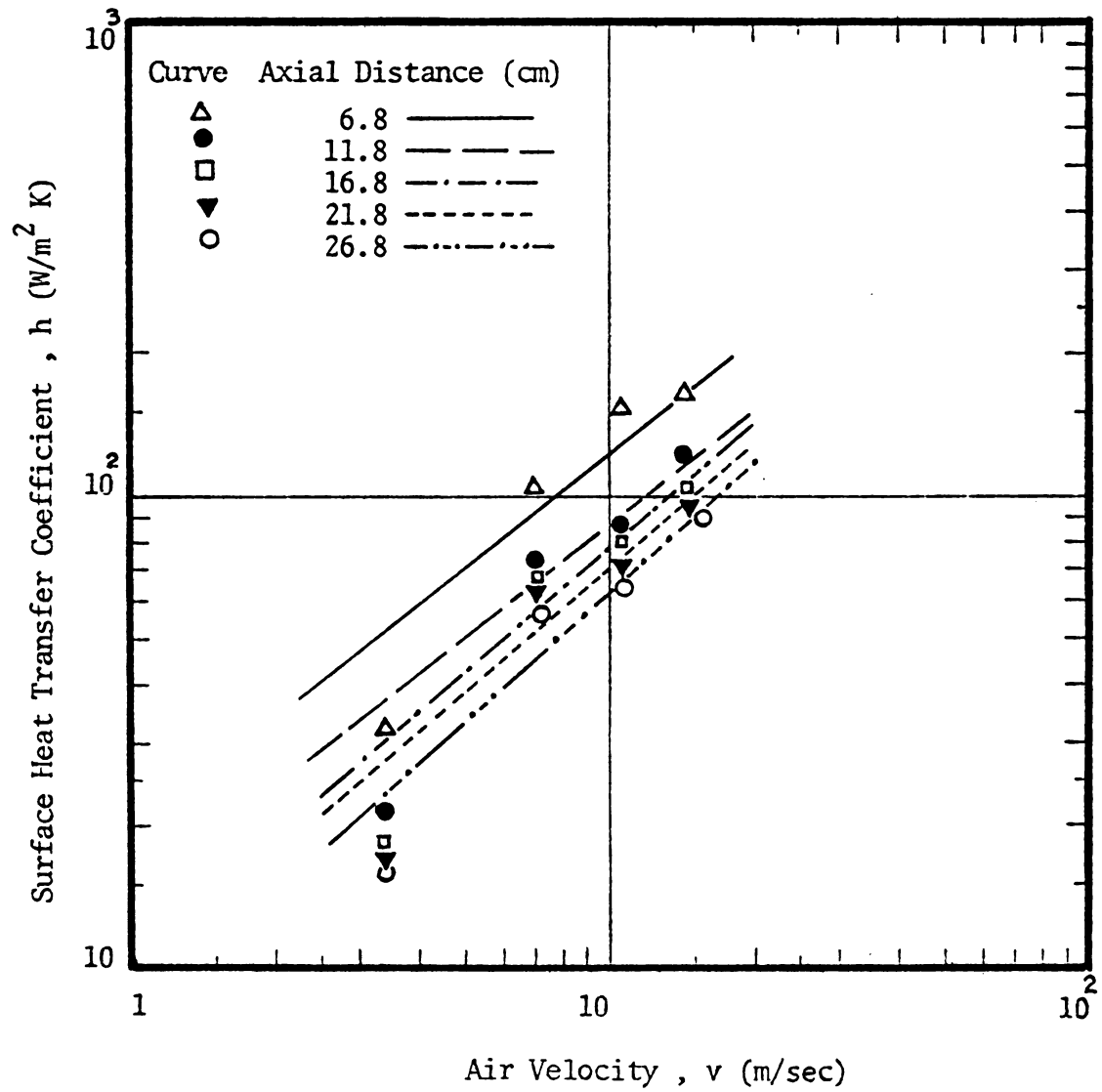


Figure 5. Surface Heat Transfer Coefficient versus Air Velocity - Acrylic Transducer Cooling Results.

analysis was performed on the experimental data in order to estimate the parameters of the regression model:

$$\eta = \beta_0 \cdot \zeta^{\beta_1}$$

$$\text{where } \eta = h_x, \bar{h}, Nu_x, \text{ or } Nu_l \quad (5.1)$$

$$\zeta = x, v, Re_x, \text{ or } Re_l$$

$$\beta_0, \beta_1 = \text{parameters}$$

The parameters β_0 and β_1 were estimated using ordinary least squares on the transformed variables: $\eta' = \log(\eta)$ and $\zeta' = \log(\zeta)$ (Beck and Arnold, 1977). The results of the power law regression analysis are summarized in Tables C-2 and C-3. By testing the regressions utilizing the t and F statistics, it was concluded that the relationships between the surface heat transfer coefficient and air velocity, and axial distance were highly significant at the 0.05 probability level.

It has become a standard procedure when correlating the surface heat transfer coefficient to combine v and x into the dimensionless Reynolds Number, which is defined as $Re = v \cdot x / (\mu / \rho) = v \cdot x / \nu$. The convective coefficient can be expressed as a part of the dimensionless Nusselt Number, which is defined as $Nu = h \cdot x / k_f$.

By utilizing experimental cooling air speed measurements and the estimated convective coefficients, dimensionless Reynolds and Nusselt Numbers were obtained. A full logarithmic plot of these two variables is presented in Figure 6. It is observed that $\log(Nu_x)$ is linearly related to $\log(Re_x)$. The transformation of the experimental data allowed for a power curve regression analysis to be performed on the

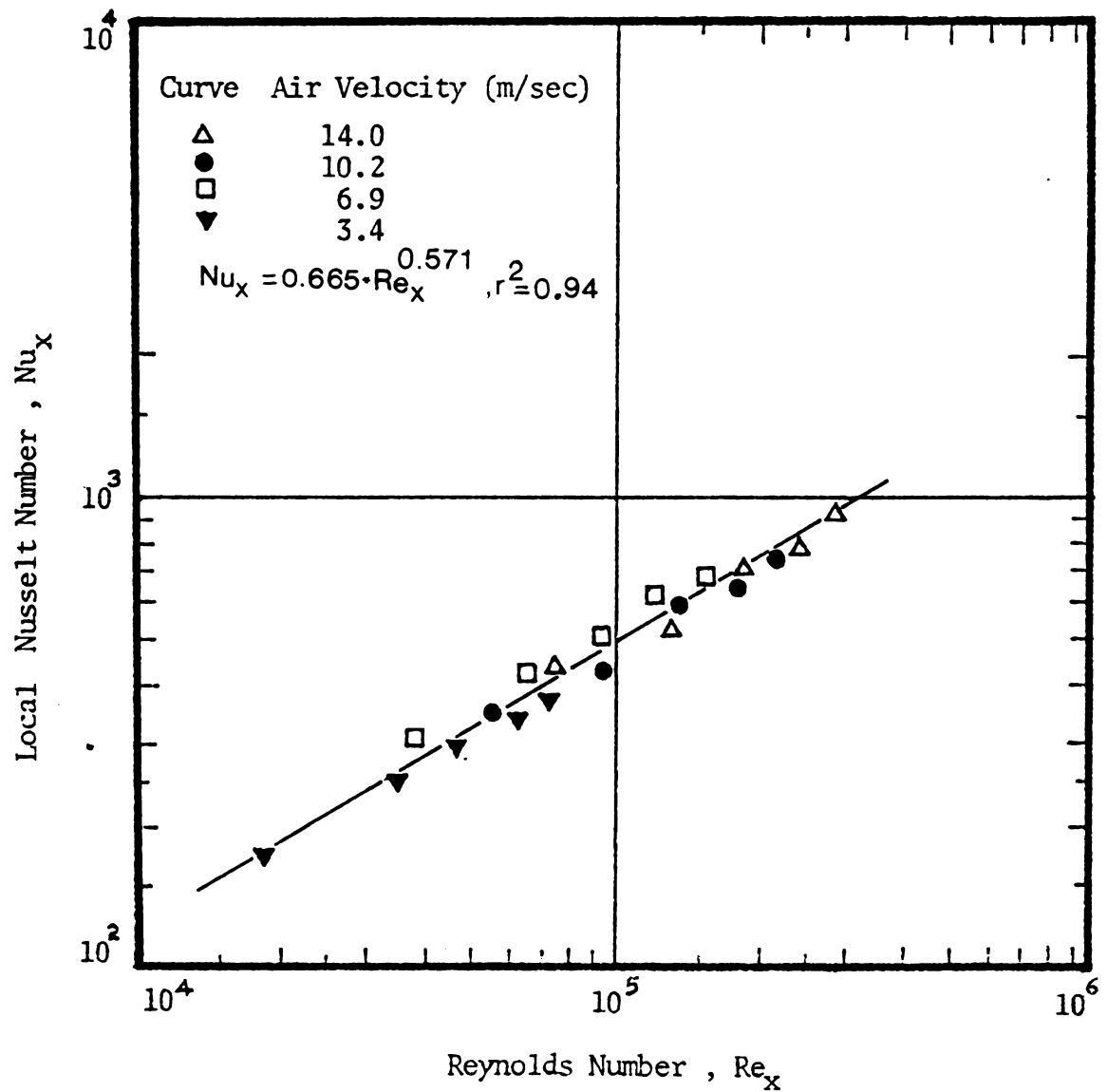


Figure 6. Local Nusselt Number versus Reynolds Number - Acrylic Transducer Cooling Results.

entire cooling data regardless of air velocity or axial distance levels. The transformed experimental results are summarized in Table 3. The correlation obtained is:

$$\text{Nu}_x = (0.172 \pm 0.437) \cdot \text{Re}_x^{(0.689 \pm 0.038)} \quad (5.2)$$

Although the correlation was tested significant at 0.01 probability level, no meaning can be attached to the parameter estimate $a = 0.172 \pm 0.437$. The variance associated with the parameter estimate $b = 0.689$ is about 5.5%. It should be emphasized that the correlations developed throughout this investigation are not intended for prediction purposes; rather, they were used to establish the equality of two sets of experimental results.

5.1.2.2 Comparison of Experimental Results with Published Steady-State Correlations

Assuming the comparison between the transient and steady-state correlations is valid, the nonlinear regression technique can be considered to yield transfer coefficients which are in good agreement with the general trend predicted by published steady-state correlations presented in Appendix D. Although the trend is correct, the experimental power curve parameters deviate considerably from the ones found in the literature

The difficulty associated with comparing the present correlation with the steady-state expressions relates to the character of the heat transfer process itself. The steady-state expressions apply either to the cases of constant interfacial temperature or uniform interfacial heat flux.

Table 3. Local Nusselt Numbers, Reynolds Numbers and Correlations - Acrylic Transducer Cooling Results.

Air Speed (m/sec)	Air			Axial Distance			Average			Power Curve				
	6.8	11.8	16.8	21.8	26.8	Nusselt Number	Parameters	†						
0.0	52.3	70.3	102.7	129.6	153.9	—	—	—	—	—	—			
3.4	128.0	18554	153.7	32196	194.2	45833	220.4	59481	258.1	73123	210.0	0.805	0.512	0.98
10.2	334.0	56356	364.2	97794	521.8	139232	593.8	180671	711.5	222109	563.0	0.592	0.572	0.92
10.2	350.0	56365	435.8	97810	600.0	139207	665.6	180720	760.9	222423	625.0	0.579	0.583	0.98
14.0	440.0	76287	517.9	131787	706.0	187734	795.8	243443	932.8	299716	754.0	0.750	0.562	0.97
14.0	461.5	76303	556.4	132185	676.1	188196	871.9	244207	965.8	300218	785.0	0.847	0.555	0.96
14.0	475.0	76958	536.5	133002	746.1	189488	838.0	245744	995.3	302108	798.0	0.855	0.556	0.95
0.0	38.2	53.3	78.4	98.9	124.0	—	—	—	—	—	—	—	—	—
3.4	172.0	18755	251.8	32511	290.1	46277	338.9	60132	377.5	73865	316.0	0.634	0.571	0.99
6.9	310.0	38190	431.8	66155	515.0	94179	625.7	122350	680.0	150342	568.0	0.685	0.579	0.99
10.5	495.0	58363	696.0	101093	886.2	143929	1066.0	186871	1085.0	229680	938.0	0.658	0.605	0.98
14.2	475.0	78238	636.6	135673	707.4	193140	950.4	250510	1116.2	308000	872.0	0.478	0.609	0.96
1.6	463.7	8987	569.0	15525	804.4	22129	1003.0	28827	1184.0	35553	916.0	0.707	0.705	0.97
6.9	425.0	40001	690.2	69284	766.0	98530	968.5	125864	1060.5	157080	874.0	0.468	0.646	0.96
10.0	840.0	56187	991.6	97502	1344.0	138784	1610.0	180298	1916.0	221476	1505.0	0.954	0.613	0.96
v	Nux	Rex	Nux	Rex	Nux	Rex	Nux	Rex	Nux	Rex	Nul	a	b	r ²

† Local Correlation: $Nu_x = a \cdot Re_x^b$

As far as the cooling case is concerned, neither the interfacial temperature nor the heat flux are time independent; both exhibit an exponential decay with time. When selecting a value for the surface heat transfer coefficient for cooling process calculations, it is found that published steady-state correlations do not apply to the heat transfer case in question (Bonacina, 1972). If no other data are available to determine the convective coefficient, the use of these correlations will result in errors due to: (1) the inapplicability, and (2) the accuracy of the published correlations (ASHRAE, 1977a). The values predicted by the constant interfacial temperature and uniform heat flux will differ by 27 to 35% depending on the value used as reference.

If the present experimental parameters are in error, then errors in the temperature measurements and temperature predictions must be analyzed. More important than the magnitude of the errors is the influence of these on the determination of the convective coefficient by means of the nonlinear regression procedure used in this investigation. A qualitative as well as quantitative treatment of errors influencing the numerical calculations is presented next.

5.1.3 Influence of Experimental Errors on the Determination of Surface Heat Transfer Coefficients from Acrylic Transducer Cooling Curves

The numerical procedure as developed by Bonacina (1972) to account for the influence of the mathematical model inaccuracy and temperature measurement error on the calculation of the surface heat transfer

coefficient was incorporated into the nonlinear regression computer program. The derivation of the model and temperature error parameters (ϵ_t and ϵ_m) is presented in Appendix A. Standard numerical techniques such as interpolatory spline function and composite Simpson's rule were used in order to evaluate the error parameters at discrete time intervals (equations (A.5), (A.6), and (A.7)) (Carnahan et al., 1969). An example of the error analysis performed for each time-temperature curve calculation will be presented.

5.1.3.1 Influence of Temperature Measurement Errors

The case of cooling the transducer under natural circulating air conditions ($v = 0$ m/sec) is discussed, and the temperature, model, and optimum time criterion parameters, together with the measured and predicted dimensionless surface temperature curves are presented in Figure 7. Curve ϵ_t represents the influence of temperature measurement errors on the estimation of the convective coefficient as the cooling time progresses. Inspection of the curve reveals that the influence of a constant measurement error is significant right after the onset of the cooling process. After a relatively short period of time, the influence decays and stabilizes to a value which is much smaller than the initial one. A constant error in transient temperature measurements was assumed because of the biased character of the residuals, and considering that a constant error would correspond to the worst uncertainty situation. In a real temperature measurement system, temperature errors are transient and stabilize as the heat transfer process stabilizes (Pfahl, 1966). For the thermocouple configuration used in this investigation,

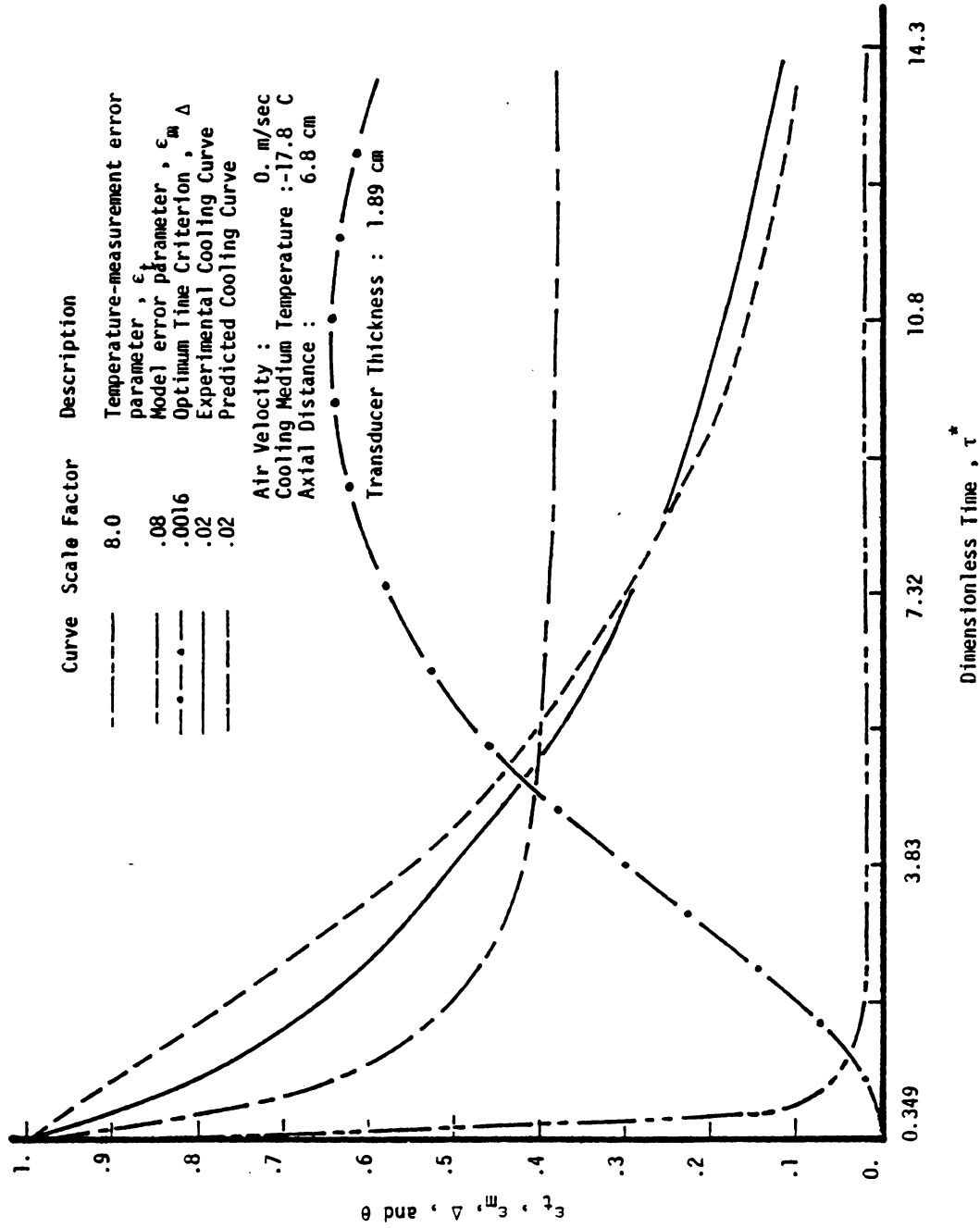


Figure 7. Error Parameters versus Time - Acrylic Transducer Cooling

measurement errors were mostly influenced by the thermal capacity difference between the bimetallic junction and the host acrylic solid, and the relative position of the junction with respect to the thickness of the solid (Pfahl, 1966; Beck, 1968; Brovkin, 1972). The magnitude of the errors have been estimated to be less than 0.5 °C; this value resulted from a finite element analysis of thermocouple temperature perturbations performed in the course of this investigation.

5.1.3.2 Influence of Mathematical Model Inaccuracy

As far as the influence of the mathematical model inaccuracy on the estimation of the convective coefficient is concerned, it can be inferred by inspection of Curve ϵ_m in Figure 7, that the model error parameter influences the regression estimation significantly for times right after the onset of cooling, and it is observed that the influence also decays to a value much smaller than the initial one. It should be noted that the temperature error parameter is about two orders of magnitude greater than the model error parameter. This difference in relative influences is attributed to the accuracy of the exact analytical solution to the acrylic transducer cooling problem.

5.1.3.3 Total Uncertainty of the Numerical Estimation

In addition to providing insight into the time-dependent character of the error parameters shown in Figure 7, Curves ϵ_t and ϵ_m can be used to estimate the magnitudes of the uncertainties on the estimated surface

heat transfer coefficient as well. For example, the values of the parameters ϵ_t and ϵ_m at the time of thermal stabilization are approximately 8.0 and 1.5. Assuming that the relative magnitudes of the measurement errors and the accuracy of the model are 1%, then the total uncertainty in the estimated coefficient is 9.5%. The total uncertainty obtained using Bonacina's graphical results is 5%. The general trend and relative magnitudes of the temperature and model parameters obtained in this investigation agree with the numerical results obtained by Bonacina (1972). The discrepancy can be attributed to the numerical methods adopted in the calculations.

5.1.3.4 Analysis of Residuals

Further insight into the influence of experimental errors on the estimation of the convective coefficient can be gained by analyzing residuals over time. Dimensionless residuals (e) were plotted versus time in Figure 8. These residuals were obtained by simply dividing the magnitudes of the residuals by the maximum temperature difference ($\Delta T = T_o - T_a$). The most important observation regarding these results concerns the biased rather than the random character of the residuals. This fact implies that temperature measurement errors can be analyzed using numerical techniques such as finite differences and finite elements to simulate the thermal perturbation of a thermocouple junction imbedded just below the convective surface of a solid (Pfahl, 1966; Beck, 1968; Brovkin, 1972). The time-dependent character of the residuals also provides a measure of the degree to which the temperatures predicted by the

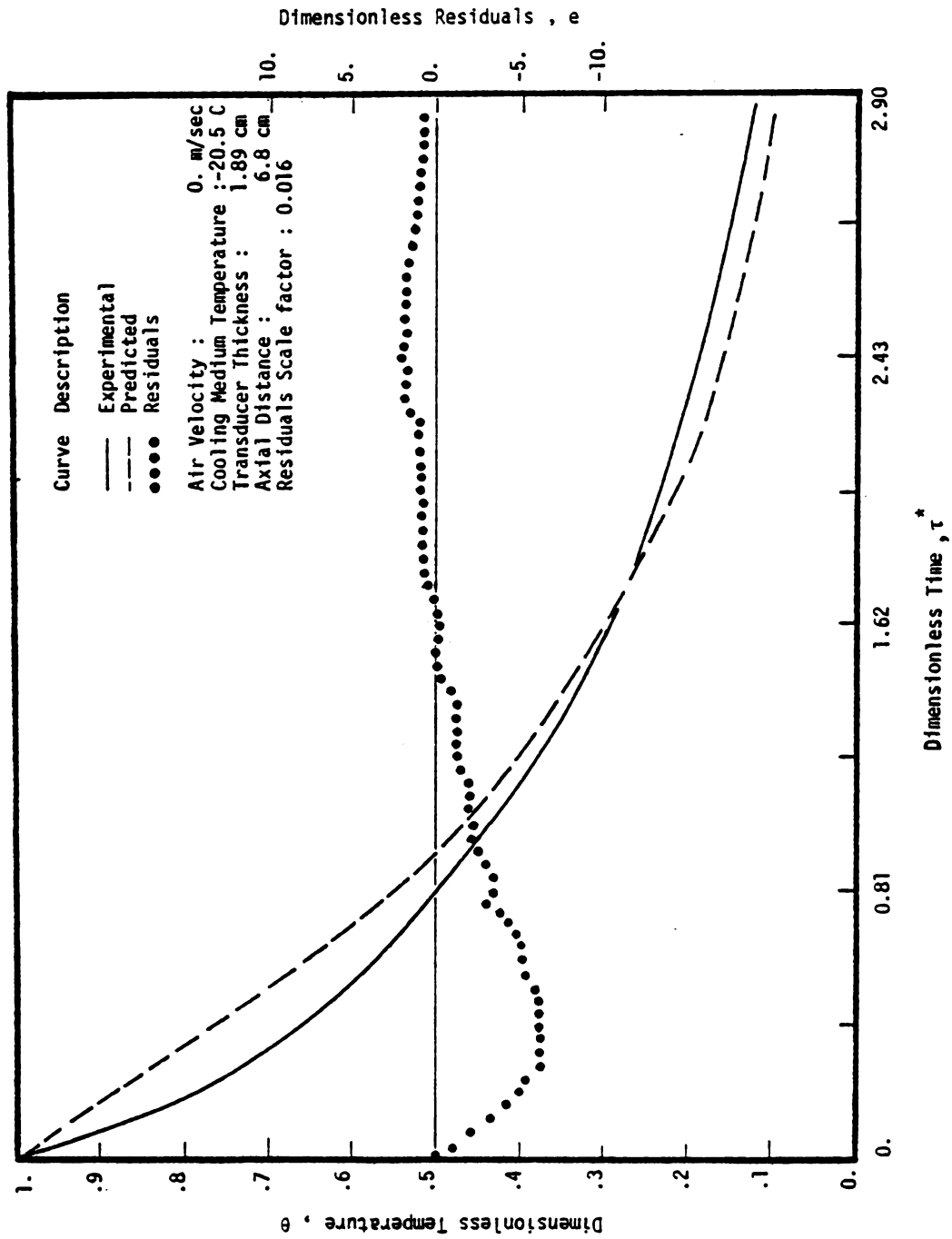


Figure 8. Agreement between Predicted and Experimental Temperatures for Acrylic Transducer Cooling.

mathematical model agree with the measured values. Inspection of Figure 8 indicates that the poorest agreement occurs after the onset of the cooling process, and that the agreement improves as the surface temperature approaches stabilization. For the particular case being discussed, the sample variance of the residuals was estimated to be 1.96%. The predicted and measured temperatures agree within a deviation of 1.4% of the total temperature difference (ΔT).

5.1.3.5 Optimization of the Numerical Calculations

The optimum time criterion (Δ) was plotted in Figure 7. The Δ criterion is determined by the sensitivity of the predicted temperatures to small perturbations in the convective coefficient. The Δ criterion reaches a maximum at some time after the onset of cooling as indicated by results in Figure 7. At this particular time ($\tau^* = 9.6$) any additional time-temperature information does not improve the accuracy of the transfer coefficient estimation any further (Beck and Arnold, 1977). Therefore, Δ is the optimum time at which the nonlinear regression procedure should be stopped. Inspection of Figure 7 indicates that the coupled contribution of the error parameters is minimized at time $\tau^* = 9.6$ when Δ reaches a maximum. At this time the dimensionless temperature is 0.22. In terms of the nonlinear regression calculations, this means that the time-temperature information, required for the successful determination of the convective coefficient, was associated with 78% of the maximum temperature difference (ΔT).

In general, the observations made above apply to all the experimental time-temperature curves analyzed. The overall agreement between the calculated and the experimental curves was 2.5% on the average. The numerical results obtained from temperature-history curves measured at $T_a = -22^\circ\text{C}$ were not as satisfactory as the rest of the computer results. The estimated surface heat transfer coefficient for these particular experiments could not be compared to those coefficients estimated from curves measured at similar air velocities. Group C in Table C-1 represents those coefficients for which the mean standard errors are greater than those associated with the rest of the estimated convective coefficients. These results are attributed to thermal instabilities of the cooling medium temperature in the wind tunnel test section. These instabilities were associated with the refrigeration temperature control of the freezing room and the heat dissipation from the electric motor driving the tunnel blowers. These two factors became most critical at temperatures close to the lower temperature limit of the refrigeration system. Variations in air temperature were not accounted for in the mathematical cooling model; in fact, one of the assumptions made in order to derive the analytical solution was to consider the air temperature constant throughout the cooling or freezing process. Therefore, cooling air temperature perturbations influenced the acrylic transducer temperature history to such an extent that the temperatures predicted by the model deviated significantly from the measured values. The lack of agreement between predicted and experimental temperatures reduced the accuracy of the estimated surface heat transfer coefficient.

5.2 Food Freezing Experimental Results

Surface heat transfer coefficients were determined from ground beef freezing curves using nonlinear regression. Four temperature-history curves were obtained from the convective as well as from the insulated surface of the food product sample at a given air velocity. The estimated parameter associated with a given local time-temperature curve was regarded as a local time-average coefficient. The influences of axial distance (x), air velocity (v), and phase change on the convective coefficient will be discussed.

5.2.1 Freezing Experimental Data

Ground beef experimental conditions are summarized in Table C-4 in Appendix C. Each freezing experiment was characterized by the freezing air temperature and air velocity, and the moisture-no-moisture loss condition at the convective surface. A typical set of experimental freezing curves for ground beef is presented in Figure 9. The shape of the curves can be recognized as that typical to food freezing processes. The temperature regions which are readily distinguished include: (1) the precooling region where the product temperature decays from its initial value to the initial freezing point; (2) the horizontal plateau region where most of the latent heat removal and ice formation occur at

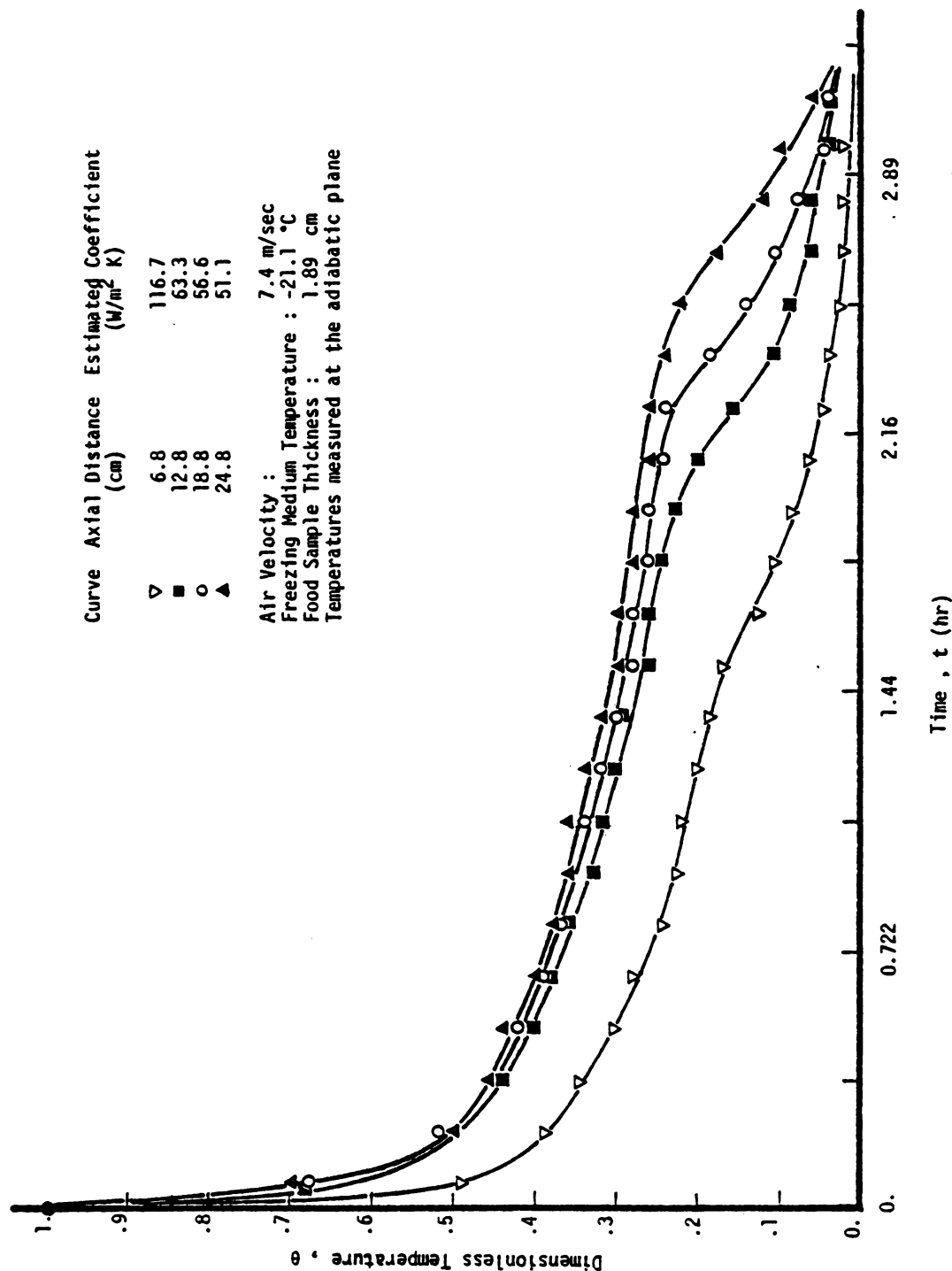


Figure 9. Experimental Time-temperature Data for Ground Beef Freezing.

a slowly depressing freezing point; and (3) the third region which is characterized by a coupled latent and sensible heat removal with the latter being predominant. Analysis of the results reveals that the distance from the leading edge influences the rate of freezing; the closer to the leading edge, the faster the freezing rate. As the axial distance increases, the freezing rate decreases and the influence of distance on the rate diminishes.

5.2.2 Analysis of the Surface Heat Transfer Coefficients--Ground Beef Freezing Results

Surface heat transfer coefficients estimated from freezing curves, standard error of the residuals (s) associated with each estimated parameter, and position-average coefficients are presented in Table C-5. For a fixed axial distance, surface heat transfer coefficients were determined from temperatures measured just below the convective surface. These values were compared to coefficients obtained from transient temperatures measured at the adiabatic or insulated plane.

5.2.2.1 Influences of Air Velocity and Axial Distance

Inspection of Table C-5 suggests that the convective coefficient is influenced by axial distance (x) and air velocity (v) as in the acrylic transducer cooling case. Better insight into the relationships between the convective coefficient and air velocity and axial

distance was provided by analyzing full logarithmic plots of h_x versus x as shown in Figure 10; h_x versus v , and average $\bar{h}$ versus v as shown in Figure 11; Nu_x versus Re_x and average Nu_ℓ versus Re_ℓ as shown in Figure 12.

By analysis of the logarithmic plots, it is observed that linear relationships exist between: (1) $\log(h)$ and $\log(x)$ at a given level of v ; (2) $\log(h)$ and $\log(v)$ for a fixed x ; (3) $\log(\bar{h})$ and $\log(v)$; (4) $\log(Nu_x)$ and $\log(Re_x)$; and (5) $\log(Nu_\ell)$ and $\log(Re_\ell)$. In addition, it is evident that a unit rise in $\log(v)$ and $\log(x)$ has opposite effects on $\log(h)$. In the former case, the air velocity is increased while in the latter it is decreased. The $\log(Nu)$ increases with a unit rise in $\log(Re)$. The linear relationship of $\log(\eta)$ to $\log(\zeta)$ was fit with the power curve model given by equation (5.1).

The parameters β_0 and β_1 were estimated following the regression procedures used to analyze the cooling experimental results. The results of the power law regression analysis are summarized in Tables C-6 and C-7 in Appendix C.

The parameter estimates of the power law relationship between h_x and x are summarized in Table C-6. The power of x (b) was on the average -0.520 and -0.529 in the case of transfer coefficients obtained from the convective and adiabatic surfaces respectively. These average parameters estimates are about 8 to 10% higher than the exact value of $b = -0.5$ obtained from boundary layer theory (Kays, 1966). The coefficient of determination (r^2) was 0.90 on the average.

As far as the influence of the axial distance (x) on the local surface heat transfer coefficient (h_x) is concerned, it can be observed from Figure 11 that the closer the location to the leading edge, the

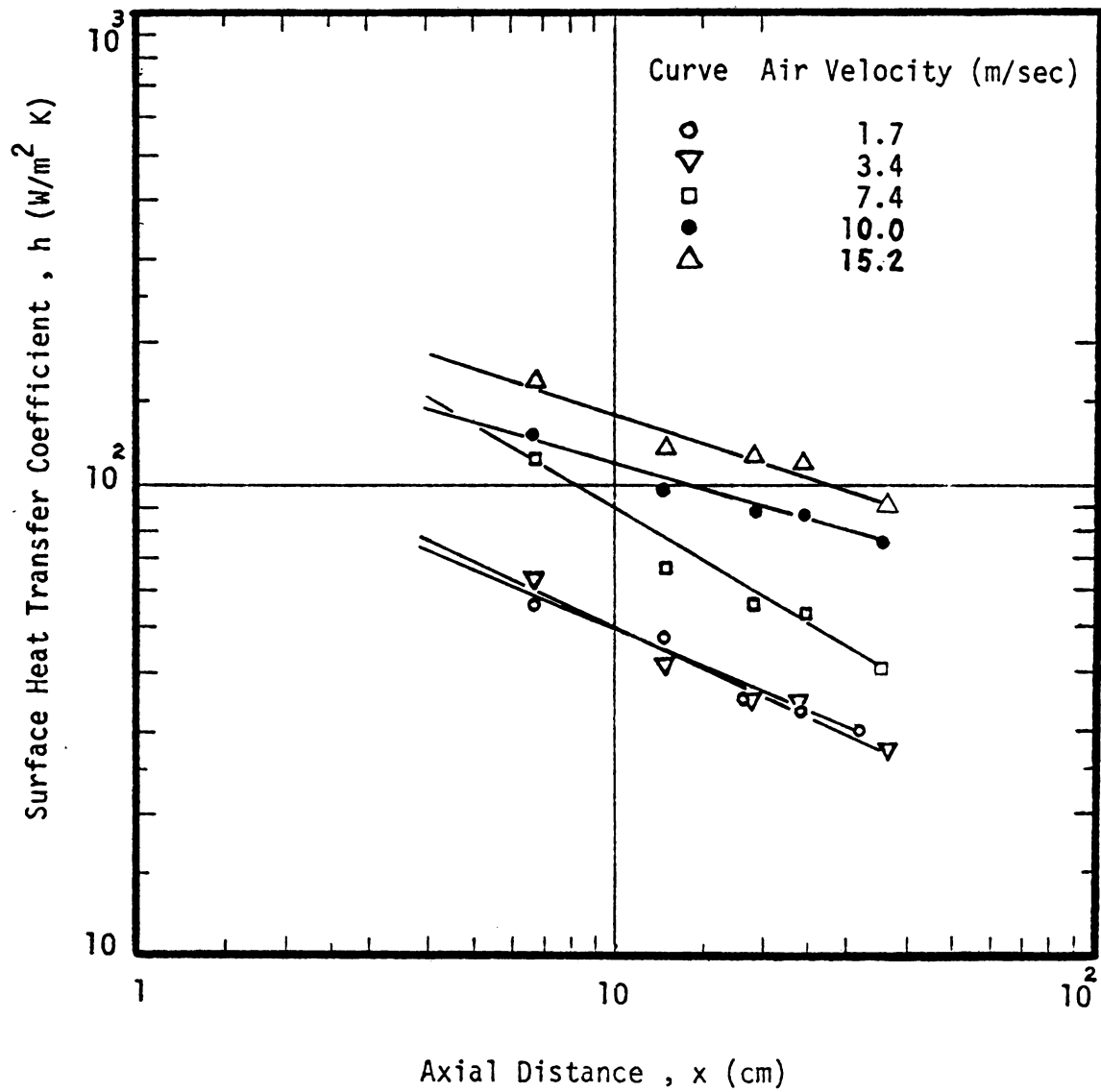


Figure 10. Surface Heat Transfer Coefficient versus Axial Distance - Food Freezing Results.

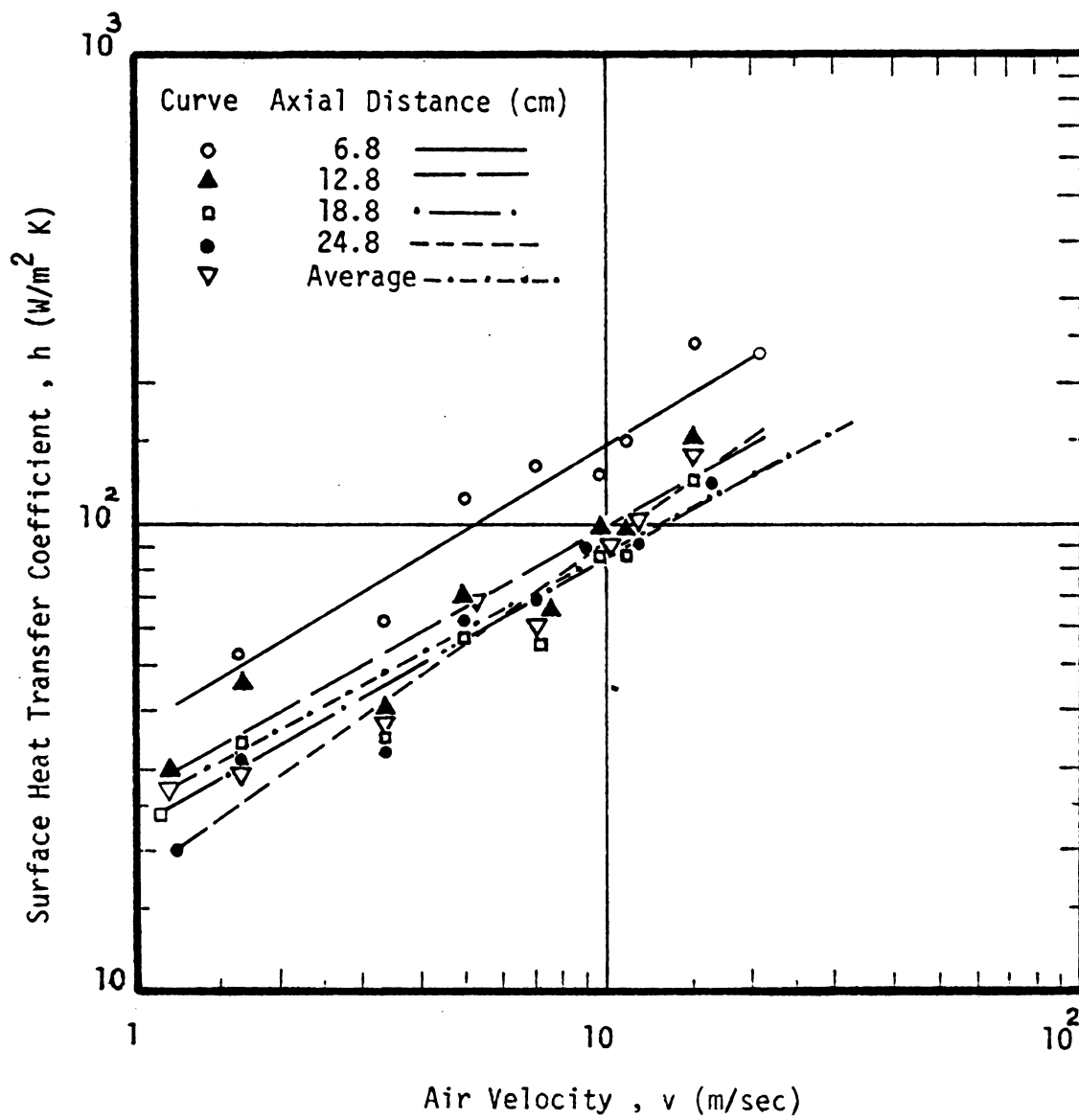


Figure 11. Surface Heat Transfer Coefficient versus Air Velocity - Food Freezing Results.

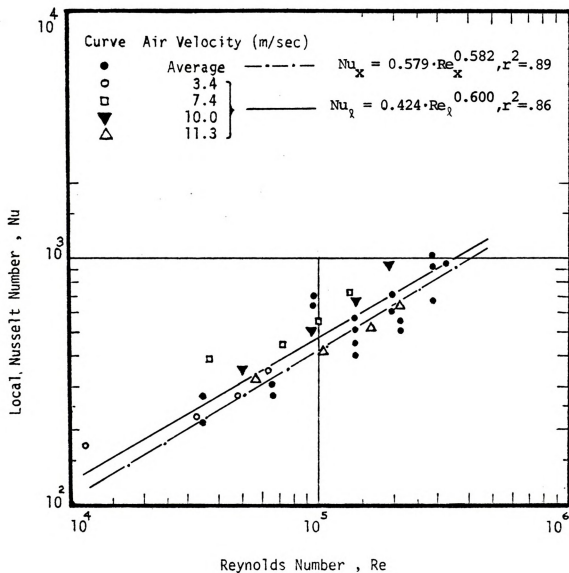


Figure 12. Local and Average Nusselt Number versus Reynolds Number - Food Freezing Results.

larger the local convective coefficient becomes. This observation is consistent with the influence of the axial distance on the measured time-temperature curves presented in Figure 9. The higher the rate of change of the temperature, the larger the surface heat transfer coefficient.

Significant statistical tests were obtained for the time-temperature curve obtained at $x = 24.8$ cm at the adiabatic food sample surface. For this particular data set, the t-test statistic indicated that the correlation between the heat transfer coefficient and the air velocity was significant at a 0.10 probability level. The overall lack of significant statistical inferences for the relationship between the local heat transfer coefficient and air velocity does not imply that the variables are uncorrelated. It was expected that the influence of experimental temperature measurement errors would influence the regressions. Analysis of the logarithmic plots indicates that as far as the influences of axial distance and air velocity are concerned, the measured coefficients follow the trend predicted by theoretical and semi-empirical boundary layer studies in a reasonable manner (Kays, 1966; Kalinin and Dreitser, 1975).

Although the regression of the local heat transfer coefficient correlation to the air velocity was not statistically significant, the correlation of the position-average coefficient ($\bar{h}$) yielded more satisfactory results. The estimated power law model obtained was:

$$\bar{h} = (24.75 \pm 0.17) \cdot v^{(0.541 \pm 0.083)} \quad (5.3)$$

By constructing the t-test statistic it was concluded that the correlation between $\bar{h}$ and v was significant at the 0.01 probability level.

The correlation presented above was compared to that obtained by curve fitting the surface heat transfer coefficients reported by Mellor (1976) and presented in Table B-1 in Appendix B. The estimated power law model becomes:

$$\bar{h} = 34.3 \cdot v^{0.49} \quad (5.4)$$

The discrepancy between the values predicted by the present correlation and Mellor's values depends on the air velocity. For the air velocity range where Mellor's data applies (refer to Table B-1), equation (5.3) predicts values about 25% lower than the published data. While Mellor's data applies to air blast systems with a velocity range between 1 and 6 m/sec the present correlation applies to low-speed wind-tunnel freezing with a velocity range twice the above. No further discussion on the discrepancy between the two correlations can proceed due to the lack of information concerning the source of Mellors' values.

The results obtained from the application of the power law regression analysis to the relationship between the local Nusselt and Reynolds Number at a fixed air velocity level are summarized in Table 4. The coefficients of determination (r^2) for the local correlations associated with each data set are greater than 0.90. These coefficients are statistically significant at a 0.05 probability level with the exception of those local correlations with r^2 less than 0.90. The parameter estimate (b) is observed not to exceed 0.70 for any local correlation. The parameter estimate (a) varies significantly as observed in Table 4. In general, (a) is much more sensitive to the scatter of the estimated transfer coefficients than is (b). This is partially attributed to the

Table 4. Local Nusselt Numbers, Reynolds Numbers and Correlations - Ground Beef Freezing Results.

Air Speed (m/sec)	Axial Distance								Average Nusselt Number	Power Curve Parameters†		
	6.8		12.8		18.8		24.8					
Convective Surface												
0.0	31.5	—	61.0	—	81.6	—	112.9	—	—	—	—	—
3.4	176.5	17713	218.4	33341	279.2	47793	354.6	63272	285.0	0.888	0.536	0.95
7.4	319.2	36809	356.2	71016	448.8	104751	546.6	138677	455.1	4.650	0.397	0.90
11.3	323.5	55762	422.1	107113	529.6	158294	634.2	209526	527.5	1.270	0.504	0.99
15.2	737.0	76100	835.0	146769	1008.0	217018	1281.0	287105	1052.5	8.200	0.396	0.89
7.3	386.3	36637	451.0	70432	552.8	99095	702.8	131219	571.8	3.220	0.450	0.91
15.2	634.3	75171	371.0	144444	—	—	786.2	282920	644.1	76.100	0.169	0.09
1.7	160.6	8872	260.3	17057	272.8	25227	347.2	33428	287.8	1.140	0.548	0.95
5.0	430.3	25253	533.2	48485	686.4	71635	926.6	94222	723.0	1.420	0.558	0.93
9.9	361.7	50242	518.0	96575	692.8	142900	921.3	189331	713.0	0.199	0.690	0.98
Adiabatic Surface												
3.4	177.1	17713	208.1	33341	279.2	47766	349.3	63286	280.8	0.939	0.529	0.93
7.4	337.7	36741	344.8	70584	452.8	104571	539.3	138486	453.2	7.730	0.352	0.81
11.3	289.4	55718	391.1	106980	501.6	158276	652.2	209562	516.5	0.538	0.573	0.99
15.2	482.0	76000	671.0	146350	895.2	216818	1172.0	286973	916.0	0.292	0.656	0.98
7.3	378.2	36235	415.0	66811	—	—	688.0	136671	565.1	2.889	0.458	0.89
15.2	743.4	75325	636.7	144343	892.8	214363	1226.0	383272	957.1	11.640	0.358	0.53
1.7	153.6	8865	213.5	17012	249.6	25215	194.2	33412	210.3	18.430	0.242	0.47
5.0	491.0	25176	466.8	48418	757.6	71585	771.3	94856	676.4	8.510	0.391	0.69
10.0	344.9	50236	494.6	96441	546.4	142898	763.0	189265	600.4	0.850	0.553	0.95
v	Nux	Rex	Nux	Rex	Nux	Rex	Nux	Rex	Nul	a	b	r ²

† Local Correlation: $Nu_x = a \cdot Re_x^b$

logarithmic transformation of the data required to obtain a linear relationship between the Nusselt and the Reynolds Numbers.

The correlation obtained by applying the power law regression analysis to all the data sets presented in Table 4 is given by:

$$Nu_x = (1.10 \pm 0.48) \cdot Re_x^{(0.536 \pm 0.043)} \quad (5.5)$$

By testing the correlation coefficient using the t-test statistic, it was concluded that the correlation between the local Nusselt and the local Reynolds Numbers is highly significant at the 0.01 probability. Even though the statistical test is significant, no meaning can be attached to the correlation presented above. By fitting a power curve to those data with a significant correlation at a 0.025 probability level, the following overall local correlation was obtained:

$$Nu_x = (0.331 \pm 0.086) Re_x^{(0.632 \pm .038)} \quad (5.6)$$

The improvement in the latter correlation suggests that errors associated with some estimated surface heat transfer coefficients influenced the regressions to such an extent that the correlations resulted meaningless. Errors in the convective coefficient are in turn due to uncertainties in temperature measurements, temperature predictions, and the validity of such assumptions as one-dimensional heat flow. The latter was not accounted for in this investigation. As far as errors in the thermal properties are concerned, these were taken into account by assuming that such errors would contribute to a time lead or lag of the predicted temperatures with respect to the measured values. The error analysis is presented in the next section.

Although it was intended to investigate the influence of moisture loss from the surface on the convective coefficient, the data collected did not allow for meaningful inferences.

5.2.2.2 Influence of the Surface Heat Transfer Coefficient on Freezing Temperature Predictions

The results presented in Figure 9 are rather significant with regards to temperature fields and freezing time predictions. The plot shows four freezing curves measured at four equally spaced axial positions for a given freezing run. A unique convective coefficient was computed from each curve. These values will be assumed to be the true coefficients associated with the heat transfer process. If a one-dimensional freezing model is utilized to predict temperature fields and freezing time, Figure 9 can be used to illustrate the extent to which inaccurate estimates of the convective coefficient influence the numerical predictions. Since the one-dimensional heat conduction model neglects the axial distance in the direction of air flow, the local influence on the transfer coefficient is ignored. The accuracy of the freezing model predictions will depend on temperature measurements. Neglecting experimental errors, the experimental freezing curve accuracy will depend on the measurement location in the air flow direction. The best agreement between predicted and experimental curves will occur when the temperature-measurement location (axial distance) corresponds to the location at which the convective coefficient applies. Any other time-temperature curve will reduce the agreement. Referring to Figure 9,

and assuming that the curve measured at 18.8 cm from the leading edge is the true unknown temperature history to be predicted, it is observed that if a value other than $56.6 \text{ W/m}^2 \text{ K}$ is selected, significantly different freezing rates will be obtained. A deviation of approximately $\pm 10\%$ of the true coefficient does not influence the precooling and initial phase-change regions to the extent that the region of sensible cooling after latent heat removal is affected. If the freezing time is defined in terms of the final isotherm-center temperature, it becomes evident from inspection of Figure 9 that gross errors in calculated temperatures will be incurred in.

In general, the observations made above suggest that when comparing curves experimentally obtained with those predicted by a one-dimensional freezing model, the temperature measurement location in the direction of air flow must be accounted for in the determination of the convective coefficient to be used as input into freezing models.

5.2.3 Influence of Experimental Errors on the Determination of Surface Heat Transfer Coefficients from Freezing Curves.

5.2.3.1 Food Freezing Model Accuracy

The determination of surface heat transfer coefficients using nonlinear regression is influenced by the degree to which the predicted freezing curve agrees with the experimental one. Disagreement can partially be attributed to the inaccuracy of the model and to the influence of uncertainties associated with the food thermal property data. If the freezing model is assumed to be accurate, it has been shown by Lescano

(1973) and Hsieh (1976) that freezing times are more sensitive to the initial freezing point and density of the food above freezing than to the thermal conductivity. The best agreement between the predicted and the measured freezing temperatures, given some uncertainty associated with the thermophysical data for ground beef, was obtained by means of the nonlinear regression approach. In Table C-5, the standard deviation of the residuals associated with each regression is presented. The average standard deviation for the total number of nonlinear regression computations was 3.7% of the maximum temperature difference (ΔT).

Predicted and experimental freezing curves, corresponding to the freezing conditions $T_a = -21.1$ °C and $v = 7.4$ m/sec, are presented in Figures 13 to 16. The measurements were obtained at the insulated plane of the food sample. Analysis of the plots reveals that: (1) higher rates are predicted by the phase-change model during the pre-cooling region; (2) the agreement is improved during the phase-change region; (3) discrepancy is increased after the initiation of the region of sensible cooling after latent heat removal; (4) the overall agreement is improved as the measurement location moves away from the leading edge; (5) the variance of the residuals does not vary much for the four cases illustrated in the plots. Inspection of the residuals reveals that: (1) they are biased rather than random; (2) tend to zero as the heat transfer tends to steady state; and (3) they seem to increase as the temperature rate of change increases.

Overall analysis of the predicted and measured freezing curves reveals that, in general, the freezing model temperature predictions follow the correct trend over the precooling, the phase-change, and the tempering regions. The greatest discrepancy between measured and predicted temperature-history curves occurs during the precooling period.

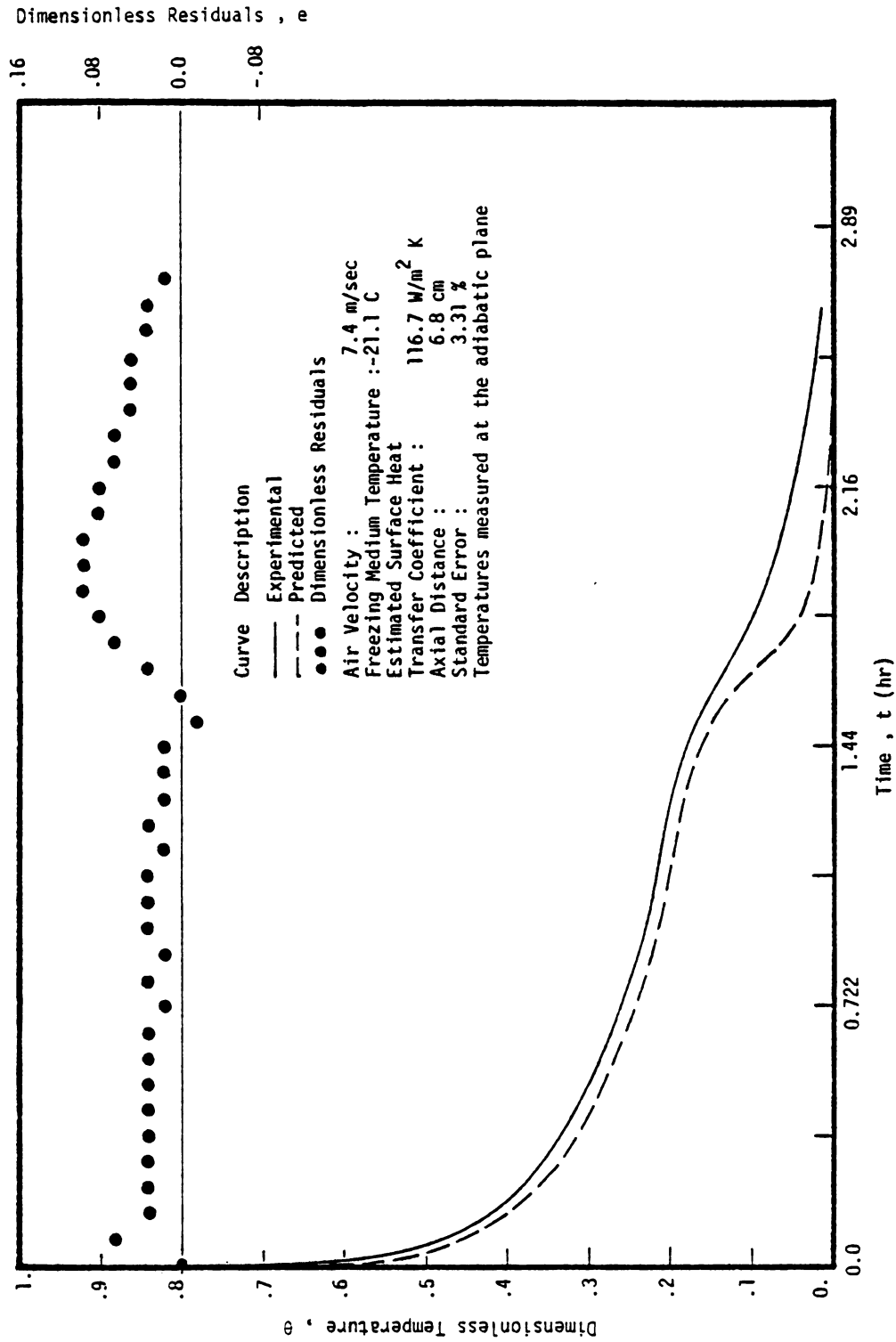


Figure 13. Agreement between Predicted and Experimental Food Freezing Curves for an Axial Distance of 6.8 cm.

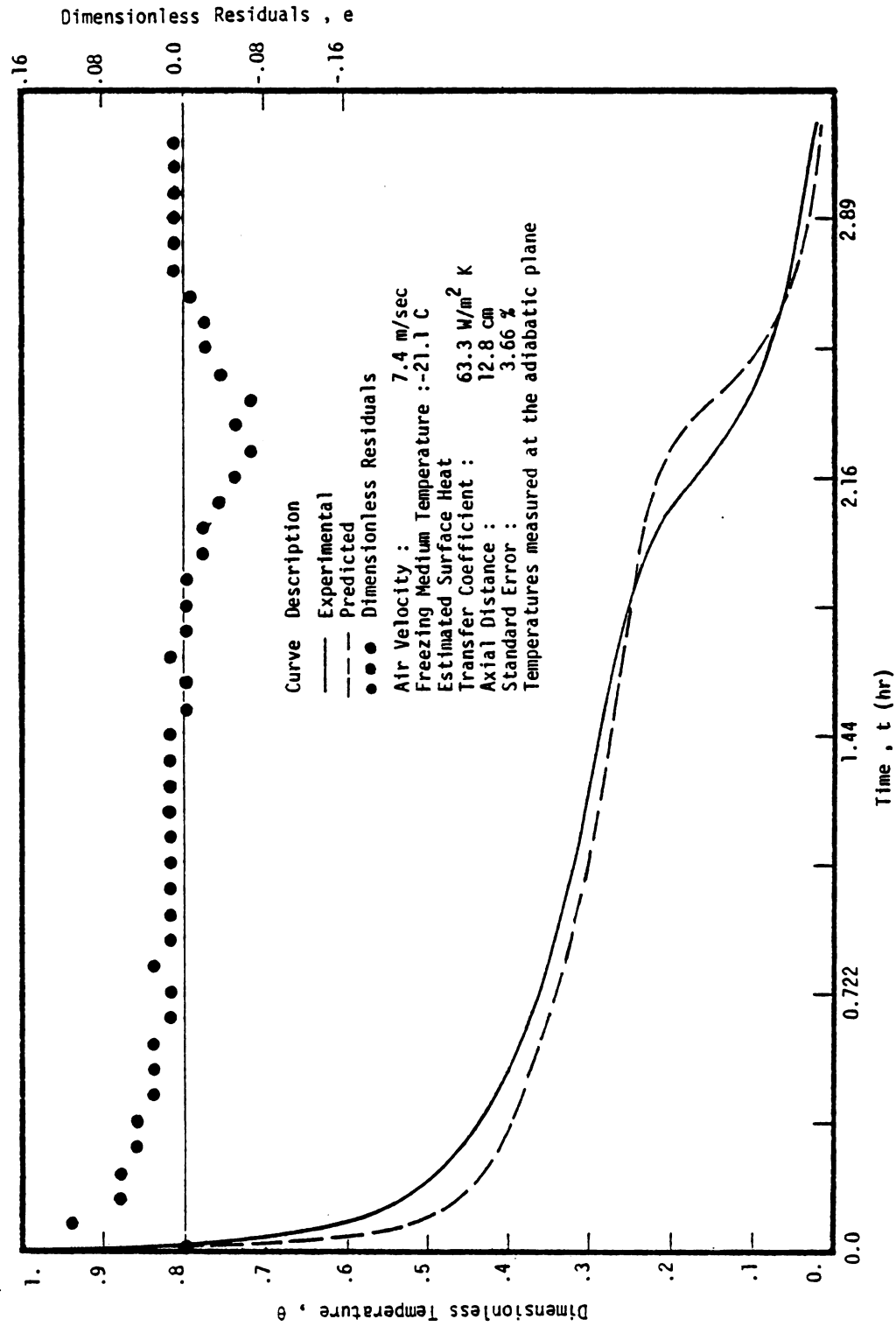


Figure 14. Agreement between Predicted and Experimental Food Freezing Curves for an Axial Distance of 12.8 cm.

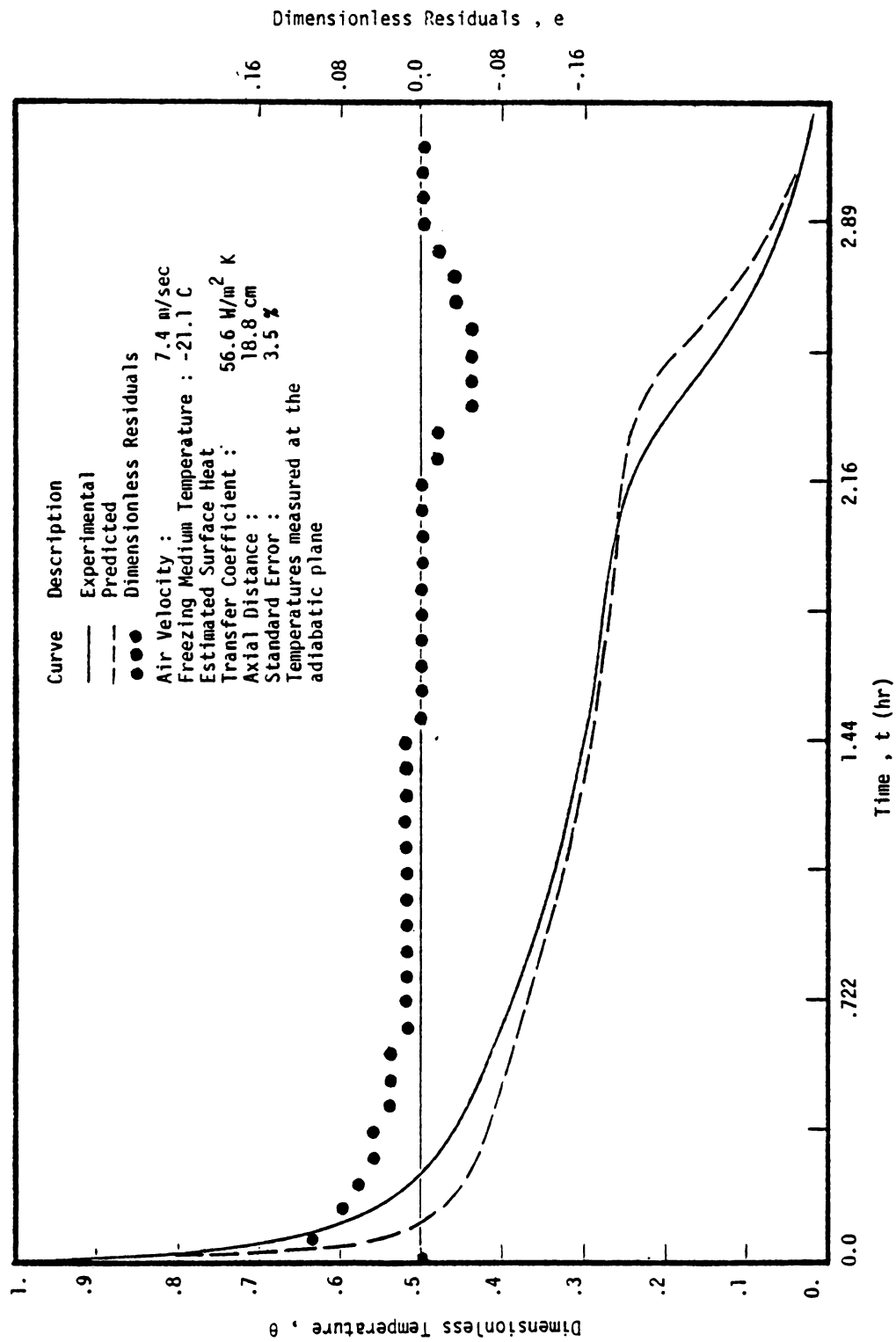


Figure 15. Agreement between Predicted and Experimental Food Freezing Curves for an Axial Distance of 18.8 cm.

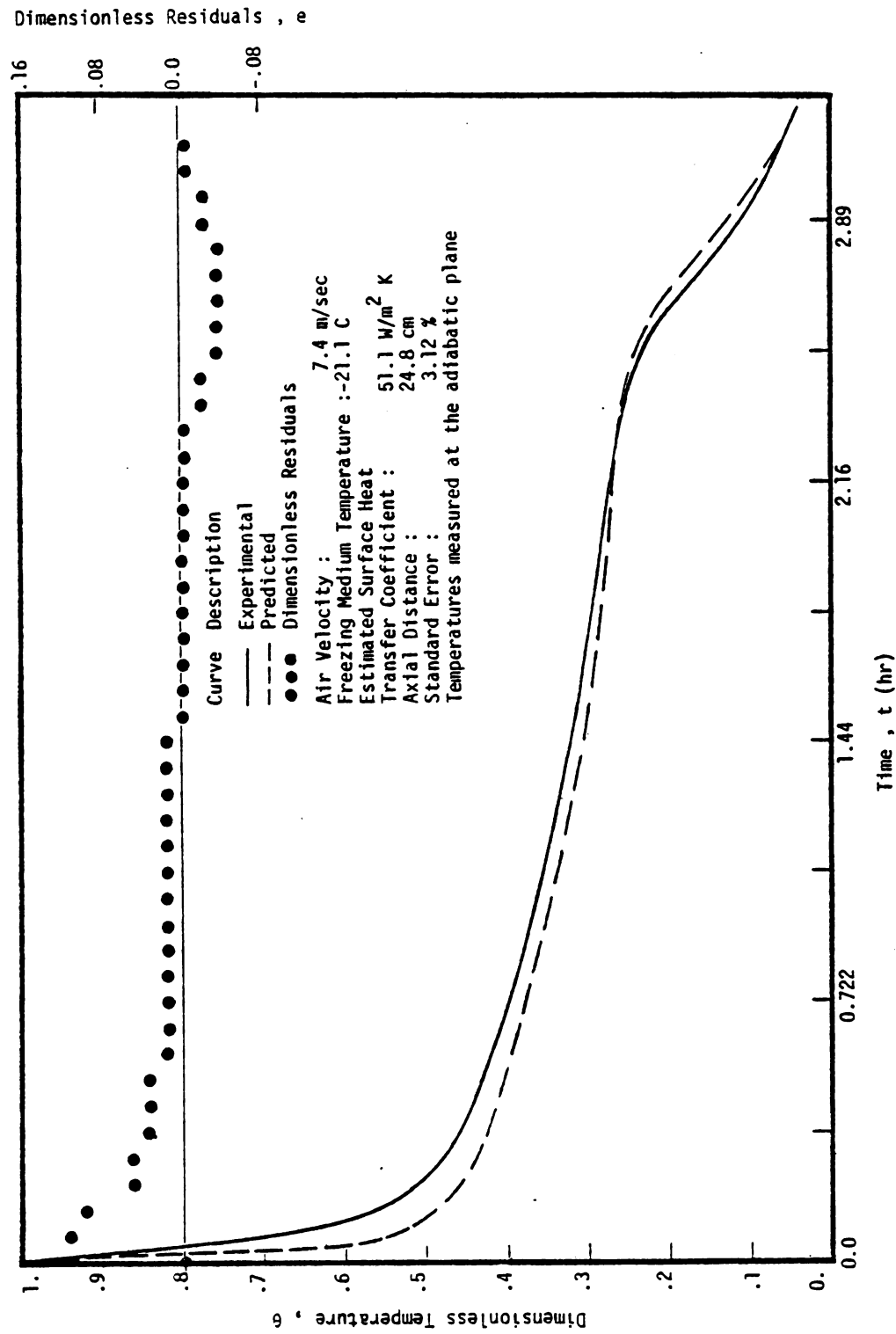


Figure 16. Agreement between Predicted and Experimental Food Freezing Curves for an Axial Distance of 24.8 cm.

The disagreement there is of the order of $\pm 3.0^{\circ}\text{C}$. During the phase-change region, the agreement improves substantially and the error is of the order of $\pm 1.0^{\circ}\text{C}$ or less. After the region of latent heat removal, the agreement is bounded by a $\pm 2.0^{\circ}\text{C}$ uncertainty.

5.2.3.2 Influence of Temperature Measurement and Model Inaccuracies

The numerical procedure discussed in Appendix A was utilized to determine the accuracy of nonlinear regression estimations of surface heat transfer coefficients. The error parameters (ϵ_t and ϵ_m), the optimum time (Δ), and the dimensionless calculated freezing curve (θ) were plotted as functions of time in Figure 17. Inspection of the plot indicates that both measurement and numerical errors tend to steady-state together with the heat transfer process. For ground beef freezing, the contributions of the error parameters (ϵ_t and ϵ_m) and the optimum time (Δ) are locally extremized at time $t = 0.07$ hr; time at which the dimensionless temperature reaches about 0.05. (Extremize means to attain a maximum or a minimum). At this particular time, the sensitivity of the freezing model temperature to the estimated parameter (h) attains a local maximum. It is interesting to note that latent heat removal is initiated at this point also. It is observed that the influence of the inaccuracy of the predicted temperatures (ϵ_m) remains reasonably steady from the onset of freezing, while the influence of temperature measurement errors (ϵ_t) on the calculation of the surface heat transfer coefficient suddenly increases by more than 50% of the value at the local maximum.

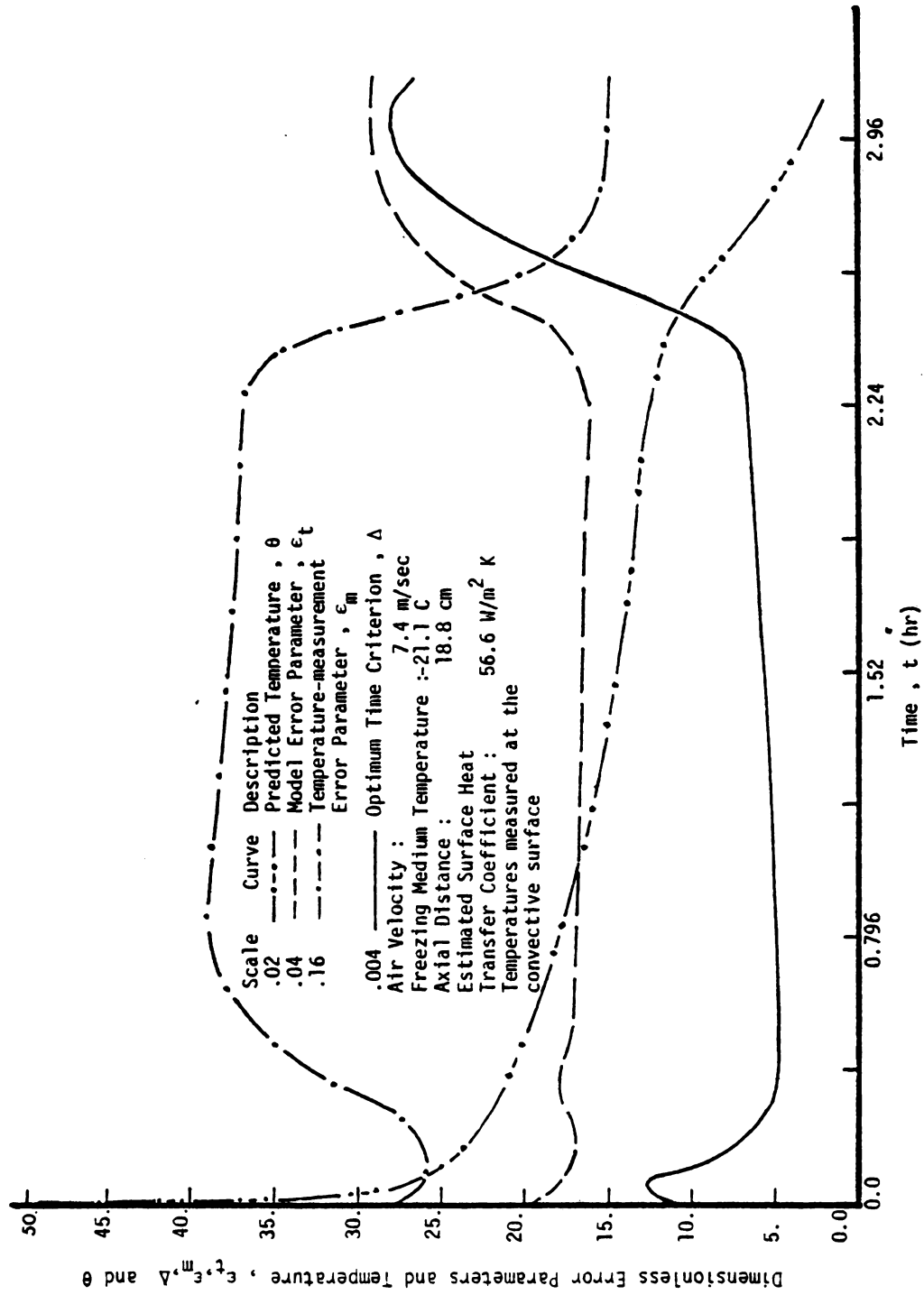


Figure 17. Error Parameters versus Time - Food Freezing Results.

5.2.3.3 Influence of Phase Change on Parameter Estimation

During the phase change region; (1) the error parameter (ϵ_t) decreases linearly and at a slow rate; (2) the error parameter (ϵ_m) remains reasonably constant; and (3) the sensitivity coefficient ($\partial\theta/\partial h^+$) increases linearly at a slow rate. These trends reveal that the temperature data associated with the phase-change region increases the uncertainty of the estimated parameter (h). According to Figure 17, 1% uncertainty in the measured temperature corresponds to approximately 6% error in the estimated surface heat transfer coefficient.

The region of sensible cooling after latent heat removal has a most significant influence on the determination of the convective coefficient. Once the food product temperature completes the plateau region, the freezing model temperature becomes most sensitive to the estimated coefficient (h) and attains a total maximum at time $t = 3.7$ hr when the dimensionless temperature is 0.95. At this time, the error parameters ϵ_t and ϵ_m are also extremized. The coupled influence of both parameters on the accuracy of the convective coefficient is minimized. During the precooling region the total uncertainty associated with the estimated parameter at time $t = 0.07$ hr is about 4.8%, while the uncertainty at time $t = 3.7$ hr is about 3.6%. The latter is the combined error due to the parameters ϵ_m (1.16%) and ϵ_t (2.4%). The relative magnitudes of these parameters is significant. It is observed that the influence of the temperature measurement error is approximately twice the model inaccuracy influence on the calculation of the convective coefficient.

This ratio suggests that the accuracy of the estimated surface heat transfer coefficient is more sensitive to temperature measurement errors than to uncertainties in the calculated temperatures because the freezing model is accurate. The degree to which the predicted temperatures agree with experimental values has been discussed in the Section 5.2.3.1 in relation to Figures 13 to 16. Additional elaborations on the freezing model accuracy are presented in the Section 5.4.

The observations considered above apply to all the nonlinear regression calculations performed on the experimental freezing curves. In general, the accuracy of the estimated surface heat transfer coefficient is improved by utilizing the temperature data associated with about 95% of the total temperature difference (ΔT). Given the accuracy of Lescano's freezing model, temperature measurement errors influenced the accuracy of the estimated parameters to a greater extent than the inaccuracy of the predicted temperatures.

5.3 Comparison of Surface Heat Transfer Coefficients Estimated from Cooling and Freezing Curves

It was confirmed in Sections 5.1.2 and 5.2.1 that the coefficients estimated from experimental cooling and freezing curves are influenced by axial distance and air velocity. The influences of both variables were combined into the dimensionless Reynolds Number, and expressions were developed in order to correlate the Nusselt to the Reynolds Number. Power law models were fit separately to the calculated coefficients associated with the transducer cooling and ground beef freezing curves respectively. The power law models allowed for the statistical testing

needed to make inferences about the equality of the surface heat transfer coefficients obtained from experimental cooling and freezing curves. In Figure 18, a block diagram illustrates the processing stages of the experimental time-temperature data. It was considered that the statistical comparison of the power law regressions, obtained from cooling and freezing estimated surface heat transfer coefficients, provided a more rigorous approach than a qualitative comparison of the estimated coefficients on a one-to-one basis.

The F-test statistic was constructed (Beck and Arnold, 1977) in order to examine the equality of the equations:

$$\text{cooling results: } Nu_x = 0.172 Re_x^{0.689},$$

$$\text{freezing results: } Nu_x = 0.331 Re_x^{0.632}.$$

The statistical test suggested that the regression functions, associated with the coefficients estimated from cooling and freezing curves, were not significantly different at a probability level of 0.01. The equality of the error variances for the two correlations was tested by the usual F-test (Beck and Arnold, 1977). It was concluded that the two regressions had equal variances at the 0.05 probability level.

Further insight into the agreement between the two power law models was obtained by evaluating both for a series of local Reynolds Numbers (applicable to the present experimental velocity range). The total and the percentage differences were calculated and are presented in Table 5. Analysis of the latter reveals that: (1) the relative differences do not exceed 15%, and (2) the closer to the upper and lower limits of the Reynolds Number range, the greater the discrepancy between the predicted Nusselt Numbers. The magnitude of the relative difference

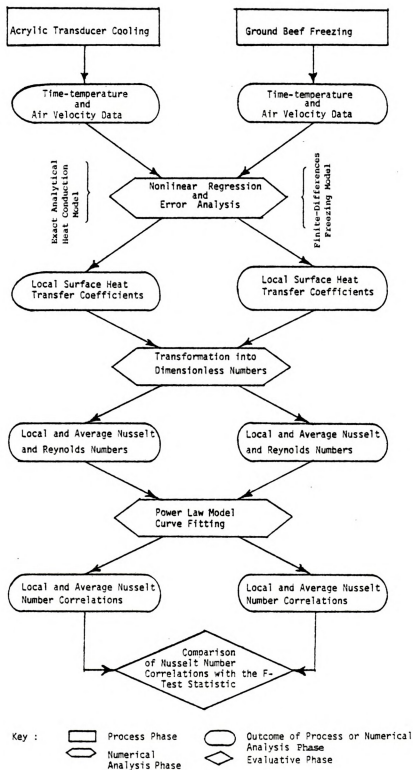


Figure 18. Block Diagram of Experimental Data Processing Stages.

Table 5. Comparison between Nusselt Number Correlations obtained from Cooling and Freezing Results.

Re_X	Acrylic Transducer Cooling Nu_X^C	Ground Beef Freezing Nu_X^F	Percent Difference	
			ΔNu_X	$ \Delta Nu_X / Nu_X^C $ %
.9 10^4	91.2	104.4	13.2	14.5
1.7 10^4	141.3	156.1	14.8	10.5
3.4 10^4	227.9	241.9	14.	6.1
4.8 10^4	289.0	300.8	11.8	4.1
5.6 10^4	321.4	331.6	10.2	3.2
6.3 10^4	348.6	357.3	8.7	2.5
7.0 10^4	374.8	381.9	7.1	1.9
.10 10^6	479.2	478.4	- .8	0.2
1.38 10^6	598.3	586.4	-11.9	2.0
1.46 10^6	621.9	607.7	-14.2	2.3
2.00 10^6	772.5	741.4	-31.1	4.1
2.83 10^6	981.3	923.3	-58.0	5.9

is considered to be within: (1) experimental errors associated with temperature measurements; (2) numerical errors associated with the inaccuracy of the heat conduction models; and (3) uncertainties associated with the thermophysical data (Hsieh, 1976; Cleland and Earle, 1977b; Lentz et al., 1977). In addition, it should be recognized that in engineering practice, knowledge or estimation of the heat transfer coefficient is limited by a 25 to 30% uncertainty.

5.4 Applications of the Transducer Experimental Approach

The applicability of the transducer experimental approach for determining surface heat transfer coefficients related to food convective freezing has been discussed. In principle, the method is not restricted to convective freezing only, but it can be utilized for other types of processes. These could be plate contact, cryogenic, and aqueous immersion freezing, retort sterilization, etc. In this section, the application of this experimental approach is discussed.

The transducer experimental method has both research and industrial applications. In the area of research, the method applies to measurement of transfer coefficients to be used as input for heat transfer simulations, while in the food processing industry the approach becomes useful for thermal-process design.

The restrictions or requirements associated with the implementation of the method are:

- (1) The mathematical model describing the heat transfer problem of the food heating and/or cooling processes must be available.

- (2) The efficient implementation of the experimental approach is accomplished by collecting time-temperature data with an automatic data acquisition system and preferably having it interface with a digital computer for direct data storage.
- (3) A digital computer must be available for performing the nonlinear regression calculations. The regression algorithm and the mathematical model must be implemented into a computer program.
- (4) A transducer must be built observing geometric and thermal similarities if the method is used for "in situ" industrial applications. For research studies, kinematic similarity has to be considered also. Although the simulation of simple geometries, such as a flat plate, facilitates the design of the transducer, modelling of more complex geometries could make the construction of the transducer and the installation of thermocouples much more difficult.

The limitations of the experimental method are directly related to the accuracy of the estimated parameters. The factors influencing the accuracy are:

- (1) The experimental temperature measurement errors.
- (2) The inaccuracy of the mathematical model of heat conduction, which in turn is related to:
 - (a) assumptions relevant to the simplification of the heat transfer problem;
 - (b) uncertainties in the thermophysical properties;
 - (c) numerical approximations associated with the mathematical solution of the heat transfer problem;

- (3) the ratio of the internal resistance of the heated or cooled body (which is related to the thermal conductivity) to the surface resistance (which is related to the contact resistance or the surface heat transfer coefficient), and the influence of the relative location of the surface thermocouple on temperature measurements.

On the other hand, the advantages of the transducer experimental method include:

- (1) The heat transfer medium flow need not be accounted for.
- (2) The experimental design can be optimized such that the time duration and the thermocouple location can be chosen in order to minimize the influence of experimental errors on the parameter estimation.
- (3) The accuracy of the method can be estimated by resorting to already developed mathematical schemes (see Appendix A).
- (4) The calculation of the coefficients can be conducted with a digital computer on-line with the temperature data acquisition system.

The steps required to implement the transducer experimental approach can be described in two parts:

- (1) Experimental Procedure:
 - (a) optimum experiment design; i.e. resort to the criteria developed by Comini (1972). Initial estimates of the transfer coefficient will be required;
 - (b) transducer design;
 - (c) experimentation and temperature data storage.

(2) Numerical Procedure:

- (a) obtain exact or numerical solution to the heat transfer problem;
- (b) implement the nonlinear regression scheme and the mathematical solution into a computer program;
- (c) perform the nonlinear regression calculations to estimate the transfer coefficients;
- (d) study the accuracy of the calculations and obtain replicates if necessary.

In order to illustrate the application of the transducer experimental approach for measuring surface heat transfer coefficients for input into freezing simulation models, the transfer coefficient obtained from cooling data (for an air velocity of $v = 11.0$ m/sec) was input into Lescano's finite-differences model to generate a freezing curve. The experimental curve for ground beef obtained at the same air velocity and similar air temperature is presented together with the predicted curve in Figure 18. The standard error of the dimensionless residuals is 3.6% of the total temperature difference (ΔT). Inspection of the plot indicates that Lescano's predicted temperatures decrease faster than the measured values during the precooling and the phase-change regions. During the region of sensible cooling after latent heat removal, the calculated values become higher than the measured ones until both intersect. The discrepancy between the two curves may be due to either or both an inaccurate surface heat transfer coefficient and/or inaccurate freezing curve prediction. Taking into consideration the accuracy of Lescano's model (Lescano, 1973) and by analysis of the residuals of all nonlinear regression calculations, it is concluded that

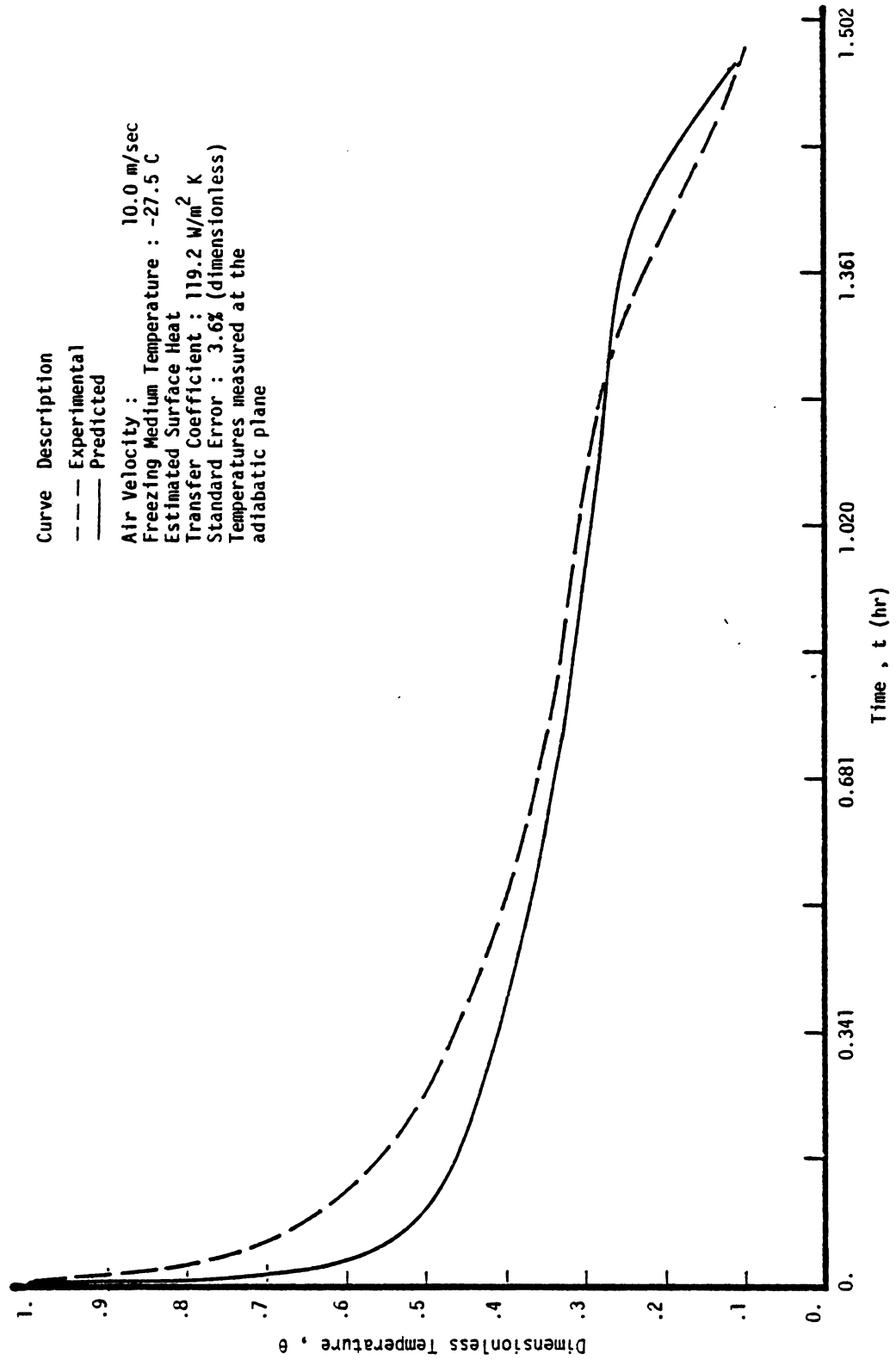


Figure 19. Influence of Surface Heat Transfer Coefficient obtained from Acrylic Transducer Cooling on Ground Beef Freezing Curve Prediction.

the major source of error accounting for disagreement between computed and experimental values is related to the uncertainties associated with such food thermal data as initial freezing point and unfrozen density. Therefore, unfortunate estimates of these properties would cause such discrepancy as the one observed in the plot.

6. CONCLUSIONS

1. The measurement of surface heat transfer coefficients using a transducer and the nonlinear regression technique can be carried out regardless of the food product configuration and airflow pattern, given that the relevant heat conduction model is available and thermal and kinematic similarities are observed.
2. The surface heat transfer coefficients determined from cooling curves are applicable to describe convective boundary conditions in one-dimensional freezing simulations.
3. Failure to account for the local influence of the surface heat transfer coefficient can partially explain the inaccurate freezing time predictions that have been reported for one-dimensional freezing simulations.
4. The influence of phase change on the determination of the surface heat transfer coefficient can be considered negligible.
5. A one-dimensional analysis of the heat transfer process associated with a flat plate configuration in a flowing medium is a valid and practical modelling approach.

7. RECOMMENDATIONS FOR FURTHER STUDY

1. To modify the cooling model to investigate the influence of packaging materials on the surface heat transfer coefficient.
2. To utilize the one-dimensional transducer-nonlinear regression approach to determine local surface heat transfer coefficients from time-temperature curves associated with cooling of infinite cylinders and/or spheres.
3. To incorporate a two-dimensional finite element heat conduction model to the transducer-nonlinear regression approach to determine surface heat transfer coefficients for irregular food product geometries.
4. To validate the assumption of a one-dimensional analysis of the heat transfer process associated with the convective heating or cooling of a flat plate configuration.

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APPENDICES

APPENDIX A

APPENDIX A

ERROR ANALYSIS OF THE NONLINEAR REGRESSION PROCEDURE

Analysis of the influence of temperature measurement errors (ϵ_t), and the influence of the mathematical model accuracy (ϵ_m) on the determination of the surface heat transfer coefficient is initiated by assuming the worst uncertainty situation: A constant temperature measurement error (Bonacina, 1972; Beck and Arnold, 1977). The uncertainty associated with the predicted temperatures can be assumed to correspond to errors in thermal properties in the case of transducer cooling because the exact analytical solution is known. In the case of food freezing the uncertainties associated with the mathematical model arise from both numerical computations and thermal property data (Bonacina, 1972; Cleland and Earle, 1977b).

The derivation of the error parameters continues by dividing Equation (3.2.7) by the transfer coefficient estimated at the ℓ -th iteration:

$$\frac{\delta h_\ell}{h_\ell} = \frac{\sum_{i=1}^n \sum_{j=1}^m \left\{ \frac{h_\ell}{\Delta T} \cdot \frac{\partial U_i^j}{\partial h} \bigg|_{h=h_\ell} \cdot \frac{V_i^j - U_i^j}{\Delta T} \right\}}{\sum_{i=1}^n \sum_{j=1}^m \left(\frac{h_\ell}{\Delta T} \cdot \frac{\partial U_i^j}{\partial h} \bigg|_{h=h_\ell} \right)^2} \quad (\text{A.1})$$

where the maximum temperature excursion (ΔT) was introduced in order to obtain dimensionless temperatures. When the iterative procedure converges $|\delta h_\ell / h_{\ell+1}| < \varepsilon$ but it will not usually be zero. Errors in the measured and in the calculated temperatures, δV_i^j and δU_i^j respectively, will influence the estimation of the surface heat transfer coefficient (h) (Bonacina, 1972).

By substituting h_ℓ by the estimated value $\hat{h}$ and separating the influences of δV_i^j and δU_i^j ;

$$\left. \frac{\delta h}{\hat{h}} \right|_t = \frac{1}{\Delta} \sum_{i=1}^n \sum_{j=1}^m \left\{ \left. \frac{\partial \theta_i^j}{\partial h^+} \right|_{h^+=1} \cdot \delta \bar{V}_i^j \right\} \quad (\text{A.2})$$

where $\theta_i^j = (U_i^j - T_a) / \Delta T$

$$h^+ = h / \hat{h}$$

and $\delta \bar{V}_i^j = \delta (V_i^j) / \Delta T$

Equation (A.2) gives an estimate of the error in the estimated parameter $(\hat{h})$ due to temperature measurement errors.

The influence of the numerical model accuracy on the estimated coefficient $(\hat{h})$ is:

$$\left. \frac{\delta h}{\hat{h}} \right|_m = \frac{1}{\Delta} \sum_{i=1}^n \sum_{j=1}^m \left\{ \left. \frac{\partial \theta_i^j}{\partial h^+} \right|_{h^+=1} \cdot \delta \bar{U}_i^j \right\} \quad (\text{A.3})$$

where $\delta \bar{U}_i^j = \delta (U_i^j) / \Delta T$

The optimum time criterion (Δ) is given by:



$$\Delta = \sum_{i=1}^n \sum_{j=1}^m \left(\frac{\partial \theta_i^j}{\partial h^+} \bigg|_{h^+ = 1} \right)^2 \quad (\text{A.4})$$

If the number of temperature observations (m) is large and only one temperature measurement location ($i = 1$) is considered, equations (A.2), (A.3), and (A.4) can be approximated in terms of integrals instead of summations.

Equation (A.2) is then expressed by:

$$\epsilon_t = \frac{\partial h}{\partial t} \bigg|_t \cdot \frac{1}{\delta \bar{V}} = \frac{1}{t \cdot \Delta} \int_0^t \frac{\partial \theta}{\partial \eta} d\eta \quad (\text{A.5})$$

where the measurement error δV_i^j was assumed constant ($\delta \bar{V}$) and it was taken out of the integral.

Equation (A.3) is given by:

$$\epsilon_m = \frac{\partial h}{\partial t} \bigg|_m \cdot \frac{\delta t}{t} = \frac{1}{t \cdot \Delta} \int_0^t \frac{\partial \theta}{\partial \eta} \cdot \eta \frac{\partial \theta}{\partial \eta} d\eta \quad (\text{A.6})$$

where errors in the mathematical model predictions were reduced to a time lag or lead with respect to measured temperatures.

The optimum time criterion is given by:

$$\Delta = \frac{1}{t} \int_0^t \left(\frac{\partial \theta}{\partial \eta} \right)^2 d\eta \quad (\text{A.7})$$

where both the dimensionless temperature (θ) and the time derivative ($\partial \theta / \partial t$) are functions of time (t) and of the dimensionless heat transfer coefficient (h^+).



If both the constant temperature measurement error ($\delta\bar{V}$) and the constant error in the predicted temperatures ($\delta\bar{U}$) are known, then the total uncertainty associated with the converged value of the heat transfer coefficient ($\hat{h}$) can be estimated using equations (A.5) and (A.6). By inspection of these expressions, it is evident that for time t^* , at which Δ reaches a maximum, the error parameters (ϵ_t and ϵ_m) are minimized. Although the measurement and numerical model errors were assumed constant, their influence on the estimation of the transfer coefficient is time dependent.



APPENDIX B



APPENDIX B

Table B-1. Literature Survey of Surface Heat Transfer Coefficients.

System	Remarks	Ambient Temp. (°C)	Air Speed (m/sec)	Surface Heat Trans. Coeff. (W/m ² K)	Overall Heat Trans. Coeff. (W/m ² K)	References
Naturally circulating air				5.8 - 17.4		
Air blast	Heating			17.4 - 58.1		
Plate contact	Cylindrical			34.9 - 116.2		ASHRAE (1977c)
Slowly agitated brine	Container			58.1 - 174.		
Rapidly agitated brine	in			174. - 581.		
Boiling liquid nitrogen	Brine			116.2 - 581.		
Boiling water				200. - 700.		
Still air		Rm Temp.	0.	7.5		
Air blast	Cardboard		1.			
	Carton		2.		6.	
		15.	1.	36.	11.	
		20.	2.	49.		
		20.	3.	60.		
Air blast		20.	4.	70.		Mallor (1976)
		20.	5.	79.		
		20.	6.	88.		
Air blast		-38.5	3.		24.	
Air blast		-20.	1.		11.	
Stockinette		to	2.		17.	
		-34.5	3.		21.	
			0.	3.4	1.7	
			2.5	14.4	5.1	
			5.1	29.	7.4	
			7.6	43.3	9.6	Mott (1964)
			10.2	57.7	11.	
			Natural	7.4	2.3	
Air blast	Carton to air				7.6	Rushbrook (1976)
	Carton slow to air				11.3	
Air blast			0.6	10.6		
			1.4	20.		Radford (1976)
			3.7	35.		
Low speed suction	Fish outlets to	-27.5	0.	26.		
Wind tunnel	air	-27.5	5.	102.		Lescaro (1973)
		-23.6	15.2	187.		



APPENDIX C



Table C-1. Surface Heat Transfer Coefficients and Standard Errors - Acrylic Transducer Cooling Results.

Air Speed (m/sec)	Axial Distance (cm)				Average Coeff
	6.8	11.8	16.8	21.8	26.8
A: Transducer Thickness: 1.89 cm, Air Temperature: -17.6 °C					
0.0	17.7	2.87	13.7	1.86	14.1
3.4	32.9	3.70	22.7	0.56	13.2
10.2	113.1	4.53	71.1	3.48	16.6
10.2	118.6	2.29	85.1	1.72	1.42
14.0	150.0	1.31	101.8	0.87	2.10
14.0	157.2	3.25	109.3	1.07	1.37
14.0	161.4	2.03	105.2	1.76	80.8
B: Transducer Thickness: 0.945 cm, Air Temperature: -20.0 °C					
0.0	12.9	1.38	10.4	1.76	10.7
3.4	77.4	1.64	54.9	1.23	10.6
6.9	104.9	3.04	84.2	3.31	1.54
10.5	167.6	2.16	135.9	2.13	34.5
14.2	162.0	3.52	124.4	3.06	58.4
C: Transducer Thickness: 0.945 cm, Air Temperature: -28.4 °C					
1.6	156.4	5.98	116.5	5.11	93.3
6.9	141.8	3.55	132.8	3.50	96.1
9.9	281.9	3.16	191.8	2.23	2.13
v	h	s	h	s	h

Surface Heat Transfer Coefficient, h ($W/m^2 K$)

Standard Error, s (dimensionless)

Table C-2. Surface Heat Transfer Coefficient -
Axial Distance Power Curve Parameters
for Acrylic Transducer Cooling.

Air Speed (m/sec)	Power Curve Parameters †		
	a	b	r ²
3.4	81.0	-0.498	0.97
10.2	236.6	-0.428	0.87
10.2 *	256.4	-0.417	0.96
14.0 *	330.8	-0.438	0.95
14.0 *	349.6	-0.445	0.94
14.0 *	355.2	-0.445	0.92
3.4 *	235.3	-0.420	0.99
6.9 *	137.5	-0.440	0.98
10.5	362.1	-0.396	0.97
14.2 *	338.5	-0.402	0.94
1.6	267.0	-0.306	0.92
6.9	285.2	-0.344	0.92
10.0	552.9	-0.386	0.90

† Local Correlation: $h_x = a \cdot x^b$

Table C-3. Surface Heat Transfer Coefficient -
Air Velocity Power Curve Parameters
for Acrylic Transducer Cooling.

Axial Position (cm)	Power Curve Parameters †		
	a	b	r ²
6.8	19.7	0.800	0.79
11.8	14.4	0.789	0.71
16.8	10.7	0.870	0.78
21.8	9.7	0.870	0.79
26.8	8.5	0.898	0.84

† Local Correlation: $h_x = a \cdot v^b$

Table C-4. Ground Beef Freezing Experimental Conditions.

Description	Air Speed (m/sec)	Freezing Air Temp (°C)	Initial Moisture (% wb)	Fat Content (% wb)	Sample Weight		Sample Density	
					Unfrozen	Frozen	Unfrozen	Frozen
					(g)		(g)	
A1	0.0	-21.3	68.47	10.76	898.1	895.0	1.063	0.977
B1	3.4	-22.4	68.58	8.88	870.0	861.0	1.050	1.040
C1	7.4	-21.1	72.24	8.33	900.0	889.4	1.072	1.006
D1	11.3	-19.1	68.50	10.06	876.8	871.0	1.057	1.001
E1	15.2	-17.9	69.10	8.62	896.9	885.6	1.050	0.984
A2	15.2	-19.5	70.90	9.30	869.0	886.5	1.056	0.985
B2	7.3	-19.9	68.73	7.21	867.0	844.7	1.061	0.997
A3	1.7	-28.0	70.90	7.10	870.0	-----	1.070	0.983
B3	5.0	-26.9	70.90	7.10	870.0	-----	1.058	1.015
C3	10.0	-27.5	70.90	7.10	870.0	-----	1.054	0.984

Table C-5. Surface Heat Transfer Coefficients and Standard Errors -
Ground Beef Freezing Results.

Air Speed (m/sec)	Axial Distance (cm)								Average Coeff
	6.8		12.8		18.8		24.8		
A: Convective Surface									
0.0	10.9	2.78	10.2	1.88	10.2	1.68	10.7	2.15	10.5
3.4	61.0	5.07	40.1	3.13	34.9	2.51	33.6	2.38	38.1
7.4	110.3	4.40	65.4	3.64	56.1	3.39	51.8	3.08	61.7
11.3	111.8	3.51	77.5	3.78	66.2	5.58	60.1	4.52	70.6
15.2	254.7	4.42	153.3	4.81	126.0	5.06	121.4	5.00	142.5
15.2	219.2	3.06	68.1	1.17	—	—	74.5	1.27	87.5
7.3	133.5	3.08	82.3	3.78	69.1	2.82	66.8	2.81	77.4
1.7	55.5	6.71	47.8	3.14	34.1	3.17	32.9	1.91	37.3
5.0	148.7	3.93	97.9	2.97	85.8	2.66	87.8	2.50	95.4
10.0	125.0	5.20	95.1	3.72	86.6	4.14	87.3	3.74	92.4
B: Adiabatic Surface									
3.4	61.2	3.19	38.2	3.12	34.9	2.26	33.1	2.37	37.5
7.4	116.7	3.31	63.3	3.66	56.6	3.50	51.1	3.12	61.7
11.3	100.0	3.32	71.8	1.72	62.7	2.57	61.8	2.05	67.9
15.2	166.6	3.85	123.2	5.12	111.9	4.22	111.0	5.48	119.3
15.2	256.9	1.74	116.9	1.01	116.6	1.33	116.2	1.17	130.5
7.3	130.7	5.55	76.2	2.54	—	—	55.2	1.97	74.8
1.7	53.1	3.14	39.2	3.94	31.2	1.87	18.4	2.65	30.0
5.0	169.7	2.00	85.7	4.13	94.7	3.71	73.1	7.04	89.2
10.0	119.2	3.60	90.8	3.90	68.3	8.01	73.2	5.48	79.5
v	h	s	h	s	h	s	h	s	$\bar{h}$

Surface Heat Transfer Coefficient, h ($W/m^2 K$)

Standard Error, s (dimensionless)

Table C-6. Surface Heat Transfer Coefficient - Axial Distance Power Curve Parameters for Ground Beef Freezing.

Freezng Expt	Air Speed (m/sec)	Convective Surface†			Adiabatic Surface†		
		a	b	r^2	a	b	r^2
B1	3.4	144.6	-0.474	0.94*	144.7	-0.479	0.92
C1	7.4	352.2	-0.592	0.95	369.0	-0.637	0.93
D1	11.3	276.4	-0.486	0.98	203.2	-0.388	0.94
E1	15.2	753.6	-0.593	0.95	300.0	-0.326	0.92*
A2	15.2	859.3	-0.827	0.68	733.6	-0.619	0.77
B2	7.3	367.4	-0.554	0.95	340.6	-0.535	0.90
A3	1.7	132.9	-0.435	0.92*	243.6	-0.753	0.89
B3	5.0	319.4	-0.429	0.86	490.0	-0.597	0.83
C3	10.0	211.3	-0.292	0.91*	269.1	-0.432	0.91

† Local Correlation: $h_x = a \cdot x^b$

Table C-7. Surface Heat Transfer Coefficient - Air Velocity Power Curve Parameters for Ground Beef Freezing.

Axial Position (cm)	Convective Surface†			Adiabatic Surface†		
	a	b	r^2	a	b	r^2
6.8	37.4	0.605	0.75	46.0	0.454	0.57
12.8	36.3	0.375	0.47	27.9	0.487	0.73
18.8	23.9	0.533	0.70	23.2	0.531	0.69
24.8	25.2	0.472	0.61	13.9	0.758	0.86

† Local Correlation: $h_x = a \cdot v^b$

APPENDIX D

Table D-1. Local Nusselt Number Steady-State Correlations

Generalized form of the local Nusselt Number $Nu_x = a Re_x^b Pr^c$

Pr	Type of Flow	Description	Coefficients			References
			a	b	c	
0.6 ÷ 15	Laminar	Exact	0.332	1/2	1/3	Kothandaraman (1975)
	Turbulent		0.0296	4/5	1/3	Kothandaraman (1975)
0.7	Laminar	Exact	0.292	1/2	1/3	Kays (1966)
	Laminar	Exact	0.332	1/2	1/3	Kays (1966)
0.70	Turbulent	Semi-empirical	0.0295	4/5	0.6	Kays (1966)
0.71	Laminar	Uniform T_w $b/\ell = 1$ $k_s/k_f > 2,000$	0.294 (0.329)	1/2 (1/2)	0 (1/3)	Sakakibara (1973)
	Laminar	Uniform q $k_s/k_f = 1$	0.404 (0.453)	1/2 (1/2)	0 (1/3)	Sakakibara (1973)
0.71	Laminar	Stagnation any value of k_s/k_f and b/ℓ	0.492	1/2	0	Sakakibara (1973)

APPENDIX E

APPENDIX E

Thermal Property Data
for
Acrylic Sheet and
Ground Beef Meat

	Thermal Conductivity (W/m·K)	Density (Kg/m ³)	Specific Heat (J/Kg·K)	Initial Freezing Point (T _f) (°C)
Ground Beef	0.4 > T _f ^a	1058 > T _f ^b	3100 ^c	-1.5 ^b
Acrylic Sheet	0.209 ^d	1185.2 ^d	1464.2 ^d	--

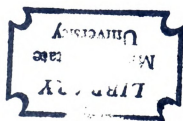
^aThermal conductivity for minced meat of approximately 66% water, 14% fat, 12% protein, 7% carbohydrates, and 1% salt. The uncertainty associated with this value is ± 0.04 W/m · K (Sörenfors, 1974).

^bExperimental values determined in this investigation.

^cData obtained from Bonacina et al., (1974).

^dData obtained from Modern Plastic Encyclopedia, (1977).





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