DESIGN PROCEDURES FOR CONVEYORIZED CHILLING AND FREEZING OF HOT BONED BEEF

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for the Degree of
DOCTOR OF PHILOSOPHY
December, 1979

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ACKNOWLEDGEMENT

I would like to express my sincere appreciation to all the individuals who in some way contributed to this work and cannot be recognized in these brief acknowledgements.

I wish to express my profound gratitude and respect to my adviser, Dr. Faye C. McQuiston, for his excellent guidance throughout this study. I would like to thank Dr. Robert L. Henrickson for his personal interest in this work. I would also like to thank Dr. Jerald D. Parker, and Dr. John A. Wiebelt for their helpful comments and discussion.

My sincere thanks to Professor Edward P. Mikol, of the University of Wisconsin, Madison, for his unselfish interest in my success.

This study was sponsored by the U. S. Department of Energy, Industrial Energy Conservation Division, DOE Constract EY-76-S-05-5097, "Energy Convservation in the Meat Processing Industry."

I thank Charlene Fries for her cooperation in typing the final manuscript.

The spiritual, material and inspirational encouragement of my parents and my grandparents is gratefully acknowledged. I am grateful to my brother, sisters, uncles, and their families for their encouragement and support. I am particularly indebted to my wife, Vijayalakshmi, for her sacrifices, help and encouragement, in making this goal possible. I am grateful to my daughter Praveena for letting me work during these past few months.

The affection and guidance provided my grandmother, V. Venkata Ramanamma, cannot be expressed in words, to whom I dedicate this thesis.

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NOMENCLATURE

Bi	-	Biot number (hL/K)
C, c	-	Specific heat (Btu/Lbm-F)
f	-	Friction factor
Fo	-	Fourier number ($\alpha\Theta/L^2$)
h	-	Heat transfer coefficient (Btu/Hr-Ft^2-F)
K	-	Thermal conductivity (Btu/Hr-Ft-F)
l	-	Characteristic length
L	-	Half thickness of solid (Ft)
$\dot{M}$	-	Mass rate of flow (Lbm/Hr)
N	-	Number of sections on the conveyor
$\dot{q}$	-	Rate of energy transfer (Btu/Hr)
Q	-	Final energy (Btu/Lbm)
Qi	-	Initial energy (Btu/Lbm)
R	-	Ratio of heat capacity flow rates of the two streams
Re	-	Reynolds number (VD/ν)
t	-	Temperature (F)
$\bar{t}$	-	Average temperature of the solid (F)
t_{bi}, t_{∞_o}	-	Constant initial temperature of the solid (F)
t_i	-	Initial temperature of the solid (F)
t_∞, t_{∞_1}, ..., t_{∞_N}	-	Fluid temperature (F)
V	-	Velocity of air (Ft/Sec)
x	-	Distance from the center of the solid (Ft)

GREEK LETTER SYMBOLS

α	-	Thermal diffusivity of the solid (Ft^2/Hr)
β	-	Roots of the transcendental equation
Θ	-	Time (Hr)
ρ	-	Density (Lbm/Ft^3)
μ	-	Absolute viscosity (Lbm/Ft-sec)
ν	-	Kinematic viscosity (μ/ρ, Ft^2/sec)
η	-	Efficiency

SUBSCRIPTS AND SUPERSCRIPTS

a	-	Fluid, air
b	-	Solid, beef
c	-	Conveyor
i	-	Initial condition
i	-	x Grid variable in finite difference equations
f	-	Final condition
∞	-	Fluid (except ∞_0)
m	-	The number of the root of the transcendental equation
n	-	The section under consideration on the conveyor, time variable ($1 < n < N$)

CHAPTER I

INTRODUCTION

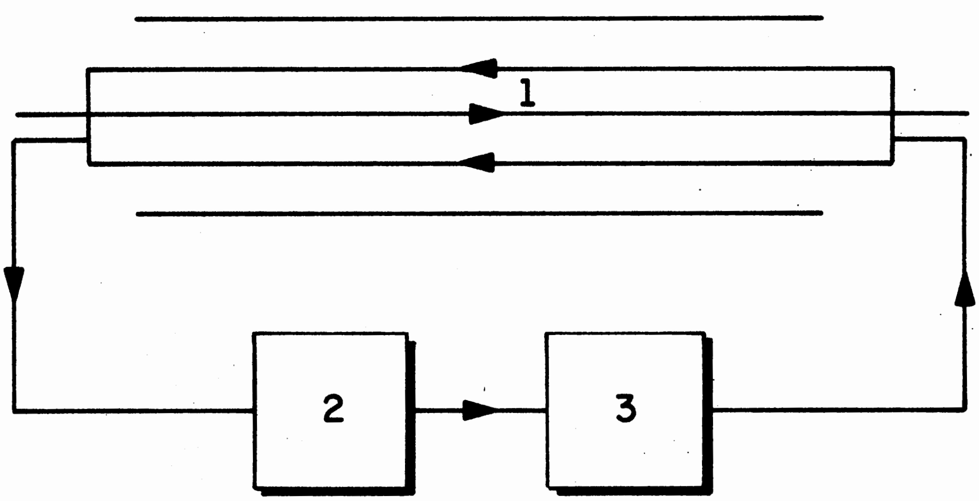
The field of heat transfer is well developed for many practical applications. There are many analytical methods and empirical correlations available in the literature; these can be directly applied or modified to suit the application under consideration. The rapidly increasing cost of energy and its limited availability demand improved system design and operating practices. An improved design will normally result in a combination of advantages such as improvement in the product quality, reduction in operating, initial equipment and building, and labor costs. A well-designed system or an improvement in the existing system can only be achieved by understanding the process and developing models to represent it closely. Thermal system modeling requires a good understanding of the heat transfer problem and exact modeling of any practical system is usually very complicated and often impossible. Hence, in practice, approximations are made which are close enough to the application to simplify the modeling procedure. With the availability of fast computers, these approximations are being relaxed more and more towards the practical system thus helping in improved understanding of the system behavior and system design.

1.1 Objectives

The primary objective of the present study is to develop a complete system modeling procedure for a one-dimensional heat transfer process of conveyorized solids. This procedure will be applied in the design of chilling and freezing processes and calculation of energy requirements for hot boned beef processing. At the present time conveyorized freezers, which are widely used in the food processing industry, are designed using highly simplified models, trial and error procedures, and experimental data. Theoretical design procedures that take into account the transient heat transfor phenomena are not available. The present study is aimed to present a rational design procedure to bridge this gap. This procedure involves finding the optimum design and operating parameters which can perform the required duty in a given amount of time with the least energy consumption. The total system, with the main subsystems, is shown in Figure 1. To predict the performance of the total system, each subsystem is modeled independently and then coupled. To achieve the above objectives, the following models are required:

1. A heat transfer model for one of the following conditions:
   a. when internal thermal resistance of the solid is negligible and the physical properties of the solid and the fluid are constant;
   b. when internal thermal resistance of the solid is important, but the cooling media can be treated as a constant temperature;
   c. internal thermal resistance of the solid is important and the temperature of the cooling medium varies along the length of the conveyor.

Practical use of the model a is very limited. Models b and c are general, have many practical applications, and will be treated for both constant and variable physical properties of the solids.



1. COUNTER FLOW CONVEYOR
2. REFRIGERATION OR HEATING SYSTEM
3. FAN OR PUMP

Figure 1. Block Diagram of the Conveyorized Heat Transfer System

2. Refrigeration system models based on actual equipment used in the respective applications.

3. A fluid moving system model.

By properly matching these subsystems in the simulation of the process, the operating points of the total system and the individual elements can be predicted. A parametric study was performed on the total system to find the sensitivity of the various parameters. These results are useful in designing and operating systems of this nature at their optimum condition.

1.2 Applications

Most of the solids which are of uniform thickness, but small compared to the dimensions of the conveyor on which they are loaded, can be represented as one-dimensional solids for heat transfer modeling. This technique can be applied to conveyorized chilling, freezing, and heating of one-dimensional solids with any fluid medium by using proper models for refrigerating or heating and circulating the fluid, as represented by the block diagram in Figure 1.

A system of this nature has many advantages. They are:

1. The counterflow design possible in conveyorized systems has the highest effectiveness of any configuration and thus is close to the ideal system.

2. The quantity and velocity of the fluid will be uniform over the solids, unlike the conditions in chill coolers and freezer rooms.

3. The velocity of the fluid over the solid surface can be maintained at the required value for optimum heat transfer and energy conditions.

4. The amount of fluid required for the process can be very much reduced due to its effective use, thus reducing the circulating energy requirement. This will also reduce the equipment load in the case of chilling and freezing of solids with air by the reduction in power for the circulating fan.

This study has direct application in the design of conveyorized chilling and freezing systems for hot boned beef. The hot boning of beef, a relatively new process in the industry, has many advantages. They are:

1. Reduced energy requirement by processing only the edible portion.
2. Reduced process time, faster throughput, and less inventory.
3. Reduced fan energy for circulating the cool air.
4. More efficient refrigeration system design.
5. Reduced cooling and freezing space and thus reduction in the building cooling load.

The present study explains the various controlling factors and the influence of various parameters in the design of such a system. It also quantifies the above advantages compared to the cold processing and sets guidelines for designing optimum systems for chilling and freezing of hot boned beef.

The present method and models can be used in other conveyorized heat transfer applications, and the models can be used with additional constraints in the design of optimum systems.

CHAPTER II

LITERATURE REVIEW

A brief description of the literature related to the heat transfer models which can be applied to the conveyorized heat transfer processes is considered here. Literature concerning cold process chilling and freezing is explored. The temperature dependence of the physical properties of the beef and the availability of the data is explained. The available literature for modeling the refrigeration system and the conveyor system is outlined.

2.1 Heat Transfer Models

An abundance of literature is available for describing the heat transfer between a solid and a fluid, where the governing equations are linear and homogeneous [1, 2, 3, 4]. Nonlinear and/or nonhomogeneous problems are addressed independently [5, 6] and there is no general method to solve them. Some of the problems associated with these models are explained in Section 3.1. There are two types of heat transfer problems which are studied here for the present application. They are for (a) constant physical properties of the solid, and (b) variable physical properties of the solid.

2.1.1 Literature Applied to Solids With Constant Physical Properties

The complexity of the model depends upon the magnitude of the physical properties of the solid, fluid, and the process which is explained in Section 3.1.1. The simplest model for $Bi < 0.1$ is given by model A1 in Section 3.1.1.1. The requirements to meet this criterion are explained in the Handbook of Heat Transfer [4]. This model uses the following simple energy balance equation with evaporation and radiation neglected.

$$\frac{dq}{d\theta} = hA(t_s - t_\infty) + \text{Evaporation} + \text{Radiation} \qquad (2.1)$$

For chilling beef, the equation above is used by Bailey [7] and Levy [8] to calculate the rate of cooling in chill rooms. In designing the equipment for chill rooms, the average product load and the peak load were taken from the ASHRAE Handbook and Product Directory, Applications Volume [9]. This procedure was adequate due to the slow and uneven rate of chilling. Using the hot processing method, the rate of chilling can be increased considerably due to the thinness of the cuts. The chilling rate can be further increased by adopting a conveyorized system which also makes the chilling rate more uniform. The fast rate of chilling necessitates the development of more complex models which can account for the internal thermal resistance of the beef.

Heldman [10], Comini and Bonacina [11], Klein et al. [12], Fikiin [13] and Bonacian and Comini [14] used the one-dimensional heat transfer equation with constant heat transfer coefficient and fluid temperature to predict the freezing time which takes into account the internal thermal resistance of the beef. This type of heat transfer model for constant physical property applications of solids is available in the literature as the Heisler charts [15], or as the Fourier solution [1, 2, 3]. This is reformulated and explained with the assumptions and applications in model A2 of Section 3.1.1.2.

For the present application, a more general model is required which can be adapted for variable fluid temperature and variable heat transfer coefficient along the length of the conveyor. This was developed and is outlined with the assumptions and applications in model A3 of Section 3.1.1.3.

The following literature was used for selecting and checking some of the parameters in the chilling process. Cutting [16] provides the various requirements of the United States Department of Agriculture (USDA) and other governmental agencies to be met during the chilling process. He also tabulated [17] the usual air temperatures, velocities used, cooling times, and evaporation loss for various methods. The ASHRAE Handbook and Product Directory [9], Sulzbacher [18], and Watt and Herring [19] give the maximum (deep round), average carcass, and the minimum (chuck surface) temperatures versus time for various chilling conditions. Bailey [20] and the ASHRAE Handbook [9] describe various parameters used in the commercial refrigeration design of chill rooms for chilling beef sides. Bailey [7] and Levy [8] give some of the factors which affect the rate of cooling and evaporation loss. Hodgson [21] provides air velocity distributions in carcass-chill rooms. Lovett et al. [22] and Radford [23] provide the data on weight loss. Radford et al. [24] provide a mathematical model for heat and mass transfer during chilling of meat for slow chilling rates.

2.1.2 Literature Applied to Solids With Variable Physical Properties

Solids whose physical properties vary with temperature are treated in this section. This type of problem can be handled best by numerical techniques. A suitable scheme can be chosen according to the type of data available and the application under consideration. Some of the frequently used procedures for heat transfer problems are explained by Ozisik [3].

During the process of freezing beef, the physical properties vary considerably and their temperature dependence is no longer negligible. In the cold processing method, the product is boxed and the boxes are loaded on pallets in the freezer rooms [9]. In designing the equipment for freezer rooms, the average product load, and the peak product load determined from the experiments are used [25, 26]. Earle [27] provides experimental data for freezing times of boneless boxed beef in air.

To predict the freezing time, Comini and Bonacina [11, 14], Fleming [28, 29], Heldman [10], Goodrich [30], and Cullwick and Earle [31] used numerical techniques and solved the one-dimensional heat transfer equation with a constant heat transfer coefficient and fluid temperature. Comini and Bonacina [11, 14] and Heldman [10] used the Crank-Nicolson analysis to solve the problem. The same approach for the freezing problem is followed in the present study because of its easiness and the availability of data for comparison. The above literature is used for the development of a method for the design of a conveyorized freezing system, which can be adopted for the variable fluid temperature and heat transfer coefficient along the length of the conveyor. This was developed and outlined with the assumptions and applications in model B2 of Section 3.1.2.

2.2 Physical Properties of Beef

The physical properties of several food materials have been determined by many investigators and are given in [32]. Mellor and Seppings [33] explained the importance of thermal physical data in the design of a refrigerated food chain. He provides an equation to predict the enthalpy as a function of temperature and the mass fraction of water frozen at different temperatures. Heldman [10] developed methods for theoretically estimating the thermal conductivity, frozen fraction of water, enthalpy and thus the specific heat for the food materials whose properties depend on temperature. Morley [34] tabulated the experimental data for many meat products and gave the references of the original work. For beef, the data developed by Riedel [35] is generally used in the related work.

For beef, the density is a very weak function of temperature over the required range of temperature [34] and is normally treated as a constant. Above the freezing temperature, thermal conductivity and specific heat are also weak functions of temperature and are usually treated as constant for chilling applications. Food materials normally contain a large quantity of water [32, 34] and the fraction of water frozen varies with the temperature. Therefore the apparent specific heat in a finite temperature interval, which is the ratio of change in enthalpy to change in temperature of that interval, is used by most of the investigators for freezing process calculations. Thermal conductivity is also a strong function of temperature over the freezing range [34]. In the present study, for the physical properties of beef, the same guidelines as explained above are used and they are given in Appendix B.

2.3 Refrigeration System and Conveyor Models

Stoecker [36] explains a procedure to simulate a refrigeration system using the characteristics of the individual components of the system. The ASHRAE Handbook of Fundamentals [32] gives the effect of evaporating and condensing temperatures on the requirement of the theoretical horse power per ton. The ASHRE Handbook, Equipment Volume [37] describes the characteristics of the individual components of different refrigeration systems and a method for balancing components of the refrigeration system. Some of the procedures and data for modeling the conveyor and the duct work are given in [38] and [4]. McQuiston and Parker [39] give the procedure for estimating the fan horsepower required to circulate air. The energy required by the fan for circulating the air is a strong function of the air velocity. The heat transfer coefficient which governs the energy transfer between the solid and the fluid is also a strong function of velocity of the fluid. Therefore the heat transfer coefficient correlation, which relates the heat transfer coefficient to velocity, should be chosen carefully. It is discussed in Appendix C with the related literature.

CHAPTER III

THEORETICAL DEVELOPMENT

The design of a conveyorized heat transfer system requires three subsystem models; they are shown by the block diagram in Figure 1, Chapter I. They are (a) the heat transfer model for the energy exchange between the solid on the conveyor and the fluid, (b) the refrigeration or heating system model which controls the temperature potential, and (c) the fan or pump model which controls the pressure potential for circulating the fluid.

Heat transfer between solids and fluids vary in many different ways in practical situations. These processes can be idealized with reasonable assumptions, so that the existing mathematical tools can be applied to model the process. Some of the modeling procedures are available in the literature and have to be reformulated for the present application. Some applications demand improved models which do not exist in the literature and they will be developed. In the present study, the heat transfer models will be developed for general application and the other subsystem models will be developed for the application of chilling and freezing of hot boned beef.

3.1 Heat Transfer Models

It is known from thermodynamics and the design procedures for heat exchangers that a counter flow configuration is the ideal design and it has the highest effectiveness. Therefore, a counterflow conveyorized system is considered. This type of system has two main energy inputs, one for creating the temperature potential, and the other for creating the pressure potential for maintaining the circulation of the given quantity of fluid at the required velocity. Both of the energy inputs are required for energy transfer between the solid and the fluid. The second energy source governs the heat transfer coefficient. The rate of energy transfer can be increased by increasing the temperature potential or by increasing the heat transfer coefficient. In designing a system of this nature, it is necessary to find the optimum combination of these two parameters--that which will require the minimum total energy input to perform the required duty. The influence of the temperature potential and the heat transfer coefficient on energy transfer between the solid and the fluid are nonlinear and strongly depend on the physical properties of the solid and the fluid. The main problem in the heat transfer model of a conveyorized system is to determine the effective temperature potential between the solid and the fluid at every section of the conveyor, which governs the energy transfer. Therefore the following models of practical importance discussed in Sections 3.1.1 and 3.1.2, will be considered. The assumptions in all the models are:

1. the solid pieces can be represented by an average thickness,
2. the heat transfer is only in one dimension, and
3. the presence of the conveyor is negligible.

Other assumptions will be stated with the particular model under consideration.

3.1.1 Solids with Constant Physical Properties

There are many applications in practice where the physical properties of the solid vary so little with the temperature that they can be treated as constant. This simplifies the modeling procedure and the amount of calculations considerably. The modeling procedure also depends upon the magnitude of the physical properties of the solid, the heat transfer coefficient, and the temperature potential. The nondimensional moduli of Biot and Fourier can be used for classifying the modeling procedures. The Biot modulus represents the ratio of internal resistance to surface resistance for energy transfer to or from the solid and the Fourier modulus represents the ratio of conduction of heat to the rate of storage of energy. The limitations and applications of different models are described below.

3.1.1.2 Model A1: For Negligible Internal Resistance Applications (or for a Thermally Thin Wall, $Bi < 0.1$). A thermally thin solid has negligible internal resistance to support temperature gradients and as such, instantaneous temperatures are uniform throughout the solid. Criteria for thermal thinness depend on material properties, thickness, boundary conditions, and exposure time. Roughly, for plane geometry and the convection boundary condition, the thin wall approximation is valid for $Bi < 0.1$. This limit does not constitute strict criteria because it ignores the aforementioned time effect. The exact requirements for the thermally thin wall criteria are explained in Reference [4].

If the thin wall approximation is valid and the heat transfer coefficient is constant, the heat exchanger equations for effectiveness of the various flow configurations can be used. This will give (a) the final outlet temperature of the fluid, (b) the time required for the process, from the known initial conditions of the fluid and the solid, and the required final average temperature of the solid. Application of this model is limited in practice. The modeling procedure is given in Appendix D.

3.1.1.2 Model A2. Constant Temperature Fluid Media Environment. At high heat transfer rates between solids of relatively low thermal conductivity (for Bi > 0.1) and a fluid, the energy transfer is not controlled by the surface temperature of the solid and the bulk average temperature of the fluid alone. The internal resistance of the solid must also be properly accounted for in calculating the energy transfer. The surface temperature of the solid is far different from the bulk average temperature of the solid and is proportional to its thermal resistance. A constant temperature fluid medium environment is valid in the following cases:

1. Applications requiring a large quantity of fluid and a fast circulation rate which results in a very small change in the fluid temperature.

2. Applications which use the latent heat of the fluid for energy transfer and result in very little or no change in the fluid temperature.

The mathematical statement of the problem is as follows: The problem is represented in Figure 2 for the case of cooling a solid with a constant temperature fluid.

The governing differential equation is:

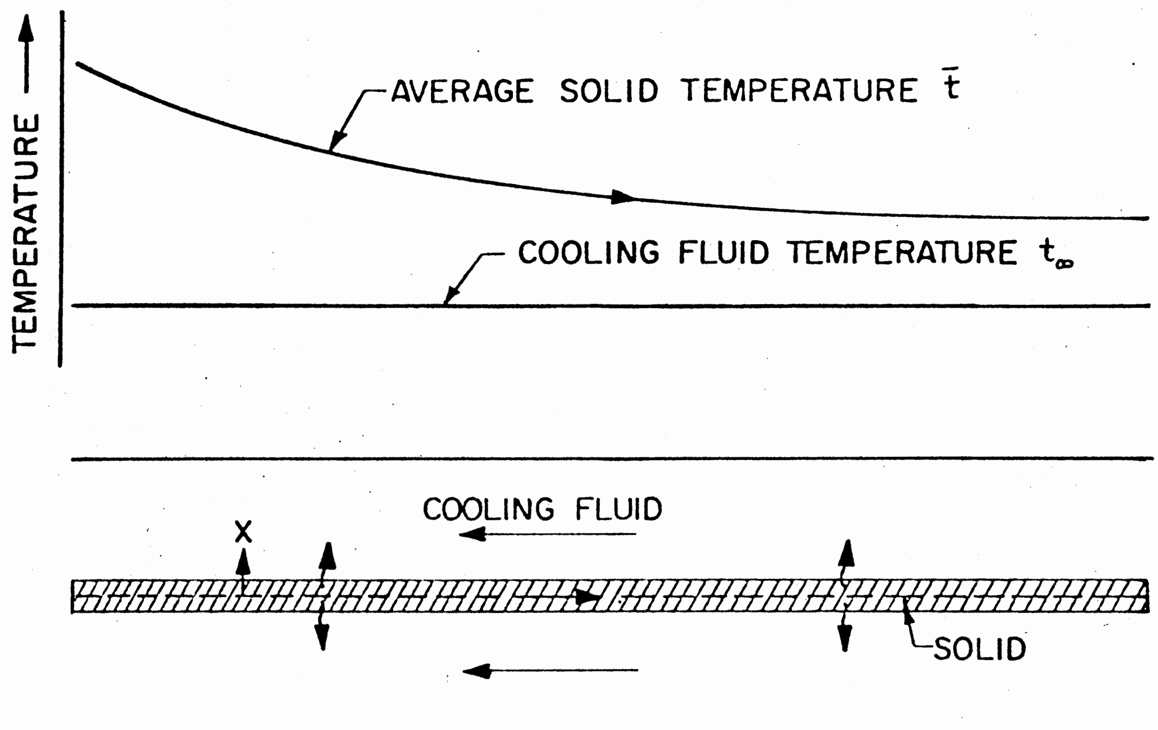


Figure 2. Schematic of the Temperature Profile and the Flow of the Two Streams for Constant Temperature Fluid Media

$$\frac{\partial^2 t(x,\theta)}{\partial x^2} = \frac{1}{\alpha} \frac{\partial t(x,\theta)}{\partial \theta} \tag{3.1}$$

The boundary conditions are:

$$\left.\frac{\partial t(x,\theta)}{\partial x}\right|_{x=0} = 0 \tag{3.2}$$

$$-K \left.\frac{\partial t(x,\theta)}{\partial x}\right|_{x=L} = h[t(L,\theta) - t_\infty] \tag{3.3}$$

The initial condition of the solid is:

$$t_i = f(x) \tag{3.4}$$

This problem is linear and homogeneous. The following two cases, which are of practical importance, are considered.

Case 1: The solution to the above problem is available in References [1, 2, 3] in different forms and these are used in deriving and checking the following form of solution, which is explained in Appendix A.

$$t(x,\theta) = t_\infty + \left\{ 2 \sum_{m=1}^{\infty} e^{-\alpha \beta_m^2 \theta} \frac{\beta_m^2 L^2}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos \beta_m x}{\cos^2 \beta_m L} \right.$$

$$\left. \frac{1}{L} \int_{x'=0}^{L} (f(x') - t_\infty) \cos \beta_m x' \, d x' \right\} \tag{3.5}$$

where the summation is taken over all the positive roots of the transcendental equation

$$(\beta_m L) \tan (\beta_m L) = Bi \tag{3.6}$$

If the function f(x) is simple, the integration can be done analytically. But in some cases it may not be possible to represent it by a simple function, then the integration has to be carried out numerically.

The average temperature of the solid can be computed using the following equation.

$$\bar{t}(\Theta) = \frac{1}{L} \int_{x=0}^{L} t(x,\Theta)\, dx \qquad (3.7)$$

Case 2: The solution to the above problem with constant t_i is explained in Appendix A and is given by:

$$t(x,\Theta) = t_\infty + 2(t_i - t_\infty) \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\Theta} \frac{Bi}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos \beta_m x}{\cos \beta_m L}$$

(3.8)

The mean temperature of the solid is given by:

$$\bar{t}(\Theta) = t_\infty + 2(t_i - t_\infty) \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\Theta} \frac{Bi^2}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{1}{\beta_m^2 L^2} \qquad (3.9)$$

The final required average temperature of the solid and thus the amount of energy transfer are known from the problem statement. The above equation can be used to calculate the time required for the process. The solution of Equation (3.9) is given in Figure 3. It can be used as follows:

Biot number is calculated from

$$Bi = hL/k$$

The energy ratio is calculated from

$$Q/Qi = (t_i - \bar{t}_f)/(t_i - t_\infty)$$

Fo is read from Figure 3.

The time required for the process is given by

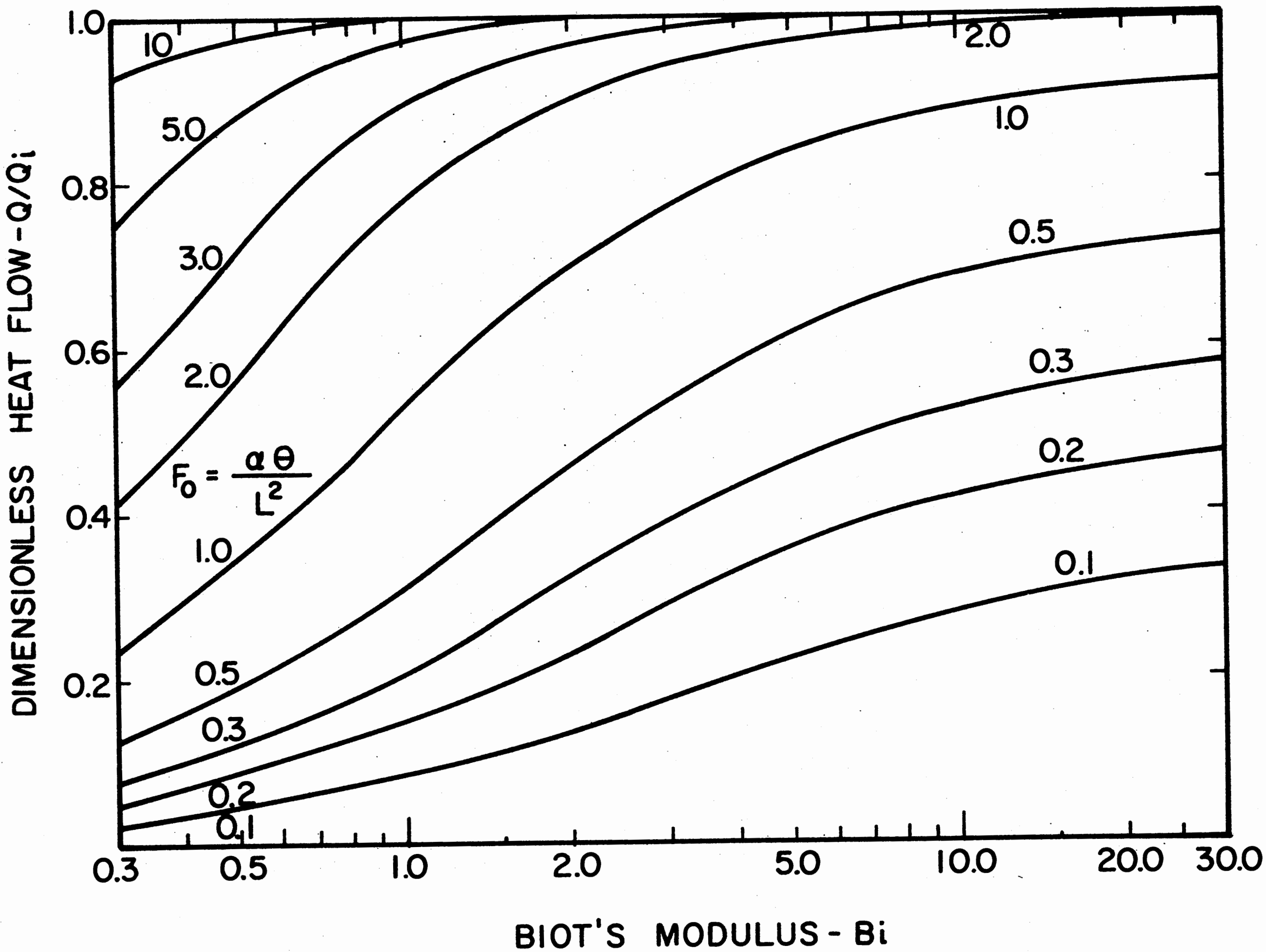


Figure 3. Dimensionless Heat Flow From a Wall Exposed to a Constant Temperature Fluid Media Environment

$$\theta = Fo * L^2/\alpha$$

The computer programs for solving Equations (3.6) and (3.8) are given in Appendix H.

3.1.1.3 Model A3: Variable Temperature Fluid Media Environment. This model is of the highest practical importance due to its flexibility for adoption to many practical cases. It is applicable to the cases where the internal thermal resistance of the solid is important, as in model A2. In many of the conveyorized heat transfer applications, the fluid temperature changes along the path of the conveyor. The fluid temperature is important in the calculation of energy transfer and the energy required by the thermal system which provides the temperature potential for energy transfer. Therefore the variation of fluid temperature along the length of the conveyor should be properly represented. This model can be used in the following applications:

1. The energy exchange between the solid and the fluid results in change of temperature of the fluid.

2. The processes in which different fluids are used at different times and which change the heat transfer coefficient.

3. When the fluid properties are functions of its temperature.

The mathematical statement of the problem is as follows: The problem is represented in Figure 4 for the case of cooling a solid with a fluid. The temperature of the fluid increases with the energy input from the solid.

The governing differential equation, the first boundary condition, and the initial condition are the same as given by Equations (3.1), (3.2) and (3.4) in model A2 of this section.

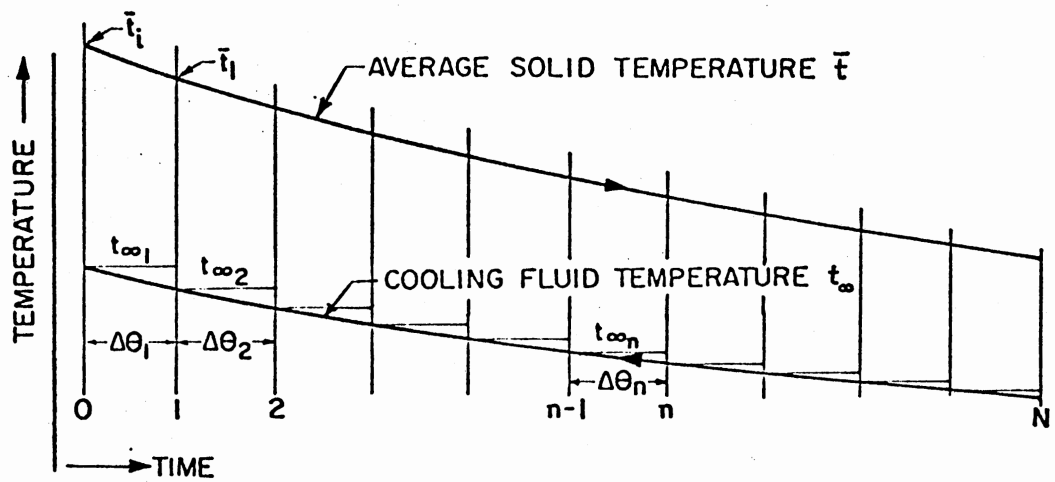


Figure 4. Schematic of the Temperature Profiles and the Step Function Approximation of the Cooling Fluid

The second boundary condition is:

$$-K \left. \frac{\partial t(x,\Theta)}{\partial x} \right|_{x=L} = h(t_\infty,\Theta)[t(L,\Theta) - t_\infty(t,\Theta)] \qquad (3.10)$$

The above equation is nonlinear and nonhomogeneous. The fluid temperature t_∞ depends upon the ratio of the flow capacity rates of the two streams and the temperature distribution in the solid. Because the fluid temperature can vary over a wide range resulting in a nonlinear temperature profile along the conveyor length, it cannot be represented by a single general function. Therefore, a complete closed form solution is not possible for this case. The next step is to divide the length of the conveyor into many sections and solve the problem stepwise with reasonable approximations at each step. The fluid temperature and heat transfer coefficient are made stepwise constant. The step function approximation of the cooling fluid temperature profile is shown in Figure 4. This reduces the problem to the linear, homogeneous case which has a solution in the form of Equation (3.5) and can be written at the nth step [bounded by sections (n-1) and n] as

$$t(x,\Theta) = t_{\infty_n} + \left\{ 2 \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\Theta} \frac{\beta_m^2 L^2}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos \beta_m x}{\cos^2 \beta_m L} \right.$$

$$\left. \frac{1}{L} \int_{x'=0}^{L} (f(x') - t_{\infty_n}) \cos (\beta_m x')\, dx' \right\} \tag{3.11}$$

where

$$0 \leq \Theta \leq \Delta\Theta_n$$

$$\text{and } f(x) = t(x, \Delta\Theta_{n-1})$$

$$\text{when } n = 1$$

$$f(x) = t(x, \Delta\Theta_0)$$

= temperature distribution in the solid before entering the conveyor.

The initial temperature distribution in the solid is known and it will be a constant. The cooling fluid temperature and the quantity of the cooling fluid used are governed mainly by the total system and the process. The final temperature distribution in the solid is fixed by the process requirements and normally the thermal average temperature of the solid is fixed. The process calculation has to be started at section 0 by assuming t_{∞_1}, as the final temperature distribution in the solid is not known at section N. This is because t_{∞_1} is normally unknown; instead t_{∞_N}, the temperature of the entering fluid, is known.

Figure 5 shows a general step bounded by the (n-1)th and the nth sections on the conveyor.

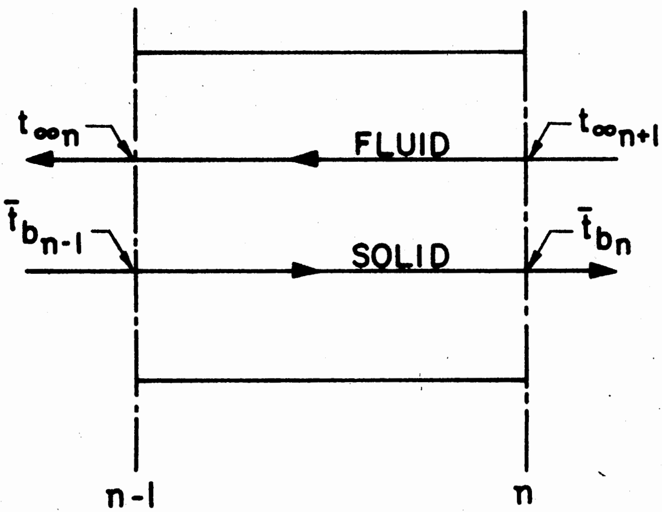


Figure 5. A General Section on the Conveyor

The following energy balance Equation (3.12) is used for calculating the fluid temperature at the next step.

$$\Delta\dot{q} = \dot{q}_{n-1} - \dot{q}_n \qquad \text{where } 1 \leq n \leq N$$

$$= \dot{M}_b\ C_b(\bar{t}_{b_{n-1}} - \bar{t}_{b_n})$$

$$= \dot{M}_a\ C_a(t_{\infty_n} - t_{\infty_{n+1}})$$

$$\text{Let } R = (\dot{M}_b\ C_b)/(\dot{M}_a\ C_a)$$

$$t_{\infty_{n+1}} = t_{\infty_n} - R(\bar{t}_{b_{n-1}} - \bar{t}_{b_n}) \qquad (3.12)$$

By successively using Equations (3.11) and (3.12) for a fixed ratio of the flow capacity rates of the two streams R, at every section on the conveyor, the final inlet temperature of the fluid can be calculated. There are actually two items to be matched. One is the cooling time required to achieve the final average temperature of the solid which fixes $\Delta\Theta$ and the other is the entering fluid temperature. The calculations are repeated by suitably adjusting t_{∞_1} until these two items are matched.

The heat transfer coefficient can also change from step to step on the conveyor either due to change in fluid properties or due to change in the cooling fluid itself. This changes the Biot number and thus the roots of the transcendental Equation (3.6). Therefore the correct Biot number and the corresponding roots should be used for the evaluation of the temperature distribution at each section.

The integration in the temperature distribution Equation (3.11) has to be performed numerically if either the temperature distribution at the earlier step is complex or h is changing from step to step. In the following case, the integration in Equation (3.11) can be done analytically. The constraints are:

1. The heat transfer coefficient is constant over the entire length of the conveyor.
2. t_∞ is stepwise constant.
3. The initial temperature of the solid is constant, i.e.,

$$t_i = t_{b_i} = f(x) = \text{constant}$$

This is a practically important case where the solid entering the conveyor is at a constant temperature and for processes when the heat transfer coefficient can be assumed constant over the entire length of the conveyor. The temperature distribution equation is the same as Equation (3.11), except f(x) is constant for the first step.

At the end of Step 1:

$$t(x,\Delta\Theta_1) = t_{\infty_1} + 2Bi(t_{b_i} - t_{\infty_1}) \sum_{m=1}^{\infty} \frac{e^{-\alpha\beta_m^2\Delta\Theta_1}}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos \beta_m x}{\cos \beta_m L} \tag{3.13}$$

$$\overline{t}(\Delta\Theta_1) = \frac{1}{L}\int_{x=0}^{L} t(x,\Delta\Theta_1)\, dx$$

$$= t_{\infty_1} + 2Bi(t_{b_i} - t_{\infty_1}) \sum_{m=1}^{\infty} \frac{e^{-\alpha\beta_m^2\Delta\Theta_1}}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{1}{\beta_m^2 L^2} \tag{3.14}$$

Now t_{∞_2} can be calculated using the Equation (3.12).

$$t_{\infty_2} = t_{\infty_1} - R[t_{b_i} - \overline{t}(\Delta\Theta_1)]$$

$$= t_{\infty_1} - 2Bi^2 R\left[(t_{b_i} - t_{\infty_1}) \sum_{m=1}^{\infty} \frac{1 - e^{-\alpha\beta_m^2\Delta\Theta_1}}{\beta_m^2 L^2 (Bi^2 + \beta_m^2 L^2 + Bi)}\right] \tag{3.15}$$

In Step 2, $f(x) = t(x,\Delta\Theta_1)$, and t_{∞_2} are used in calculating $t(x,(\Delta\Theta_1 + \Delta\Theta_2))$, $\overline{t}(\Delta\Theta_1 + \Delta\Theta_2)$ and t_{∞_3}. The procedure is repeated for all steps of the conveyor.

In general, for any section n

$$t(x, (\sum_{I=1}^{n} \Delta\Theta_I)) = t_{\infty_n} + 2Bi \left[\sum_{I=1}^{n} \left\{ t_{\infty(n-I)} - t_{\infty(n-I+1)} \right\} \right.$$

$$\left. \left\{ \sum_{m=1}^{\infty} \frac{e^{-\alpha\beta_m^2 (\sum_{k=1}^{I} \Delta\Theta_{(n-k+1)})}}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos \beta_m x}{\cos \beta_m L} \right\} \right] \quad (3.16)$$

where

$$1 \leq n \leq N$$

$t_i = t_{b_i} = t_{\infty_0}$ = constant initial temperature of the solid

$$\bar{t}(\sum_{I=1}^{n} \Delta\Theta_I) = t_{\infty_n} + 2Bi \left[\sum_{I=1}^{n} \left\{ t_{\infty(n-I)} - t_{\infty(n-I+1)} \right\} \right.$$

$$\left. \left\{ \sum_{m=1}^{\infty} \frac{e^{-\alpha\beta_m^2 (\sum_{k=1}^{I} \Delta\Theta_{n-k+1})}}{\beta_m^2 L^2 (Bi^2 + \beta_m^2 L^2 + Bi)} \right\} \right] \quad (3.17)$$

$t_{\infty_{n+1}}$ can be calculated using Equation (3.12).

Although Equations (3.16) and (3.17) do not need numerical integration, the calculations have to be done step by step for each section of the conveyor since t_∞'s at all earlier steps are required for calculating the next step. Some of the identities used in deriving Equations (3.13) to (3.17) are given in Appendix A.

3.1.2 Solids with Variable Physical Properties

There are many applications in practice where the physical properties of the solid vary considerably with temperature and cannot be treated as constant. In this section two models similar to model A2 and model A3 of the previous section will be considered due to their practical importance. A model similar to model A1 of the previous section cannot be developed and no nondimensional modulus can be used due to the temperature dependence of the physical properties of the solid.

The Crank-Nicolson method used by Comini and Bonacina [11, 14] and Heldman [10] was used in the following study. The limitations of Plank's equation and its later modification are explained in [11, 9]. The results published in [10] and the Plank equation are used to check the present model.

3.1.2.1 Model B1: Constant Temperature Fluid Media Environment. The applications of this model are similar to those of model A2 of the Section 3.1.1.2.

The mathematical statement of the problem is as follows: The problem is represented in Figure 2 for the case of freezing a solid with a constant temperature fluid.

The governing differential equation is:

$$\frac{\partial}{\partial x}\left[K(t)\ \frac{\partial t(x,\Theta)}{\partial x}\right] = \rho\ C(t)\ \frac{\partial t(x,\Theta)}{\partial \Theta} \tag{3.18}$$

The boundary conditions are:

1) $$\left.\frac{\partial t(x,\Theta)}{\partial x}\right|_{x=0} = 0 \tag{3.19}$$

2) $$\left.\frac{\partial t(x,\Theta)}{\partial x}\right|_{x=L} + \frac{h}{K(t)}\left(t(L,\Theta) - t_\infty\right) = 0 \tag{3.20}$$

The initial condition of the solid is:

$$t_i = f(x) \tag{3.21}$$

Using the Crank-Nicolson method, Equation (3.18) can be written as:

$$\rho\ C(t^n)\left\{\frac{t_i^{n+1} - t_i^n}{\Delta\Theta}\right\} = \frac{1}{2\Delta x}\left\{K^+\left[\left(\frac{t_{i+1}^{n+1} - t_i^{n+1}}{\Delta x}\right) + \left(\frac{t_{i+1}^n - t_i^n}{\Delta x}\right)\right]\right.$$
$$\left. - K^-\left[\left(\frac{t_i^{n+1} - t_{i-1}^{n+1}}{\Delta x}\right) + \left(\frac{t_i^n - t_{i-1}^n}{\Delta x}\right)\right]\right\} \tag{3.22}$$

where

n = 0 to N

N = the number of time increments

i = 1 to Lx + 1

Lx = the number of sections on the solid

$$K^+ = K\left\{t_{i+(1/2)}^n\right\} \simeq K\left\{(t_{i+1}^n + t_i^n)/2\right\}$$

$$K^- = K\left\{t_{i-(1/2)}^n\right\} \simeq K\left\{(t_i^n + t_{i-1}^n)/2\right\}$$

Rearranging Equation (3.22) gives

$$-P\ K^-\ t_{i-1}^{n+1} + [\rho\ C(t^n) + P(K^+ + K^-)]\ t_i^{n+1}$$
$$-P\ K^+\ t_{i+1}^{n+1} = P\left[K^+\ (t_{i+1}^n - t_i^n) - K^-\ (t_i^n - t_{i-1}^n)\right]$$
$$+ \rho\ C(t^n)\ t_i^n \tag{3.23}$$

where

$$P = \Delta\Theta/2\Delta x^2$$

The boundary conditions (3.19) and (3.20) reduce to

$$1)\qquad t_{i-1}^n = t_{i+1}^n \tag{3.24}$$

$$2)\qquad t_{i+1}^{n} = t_{i-1}^{n} - Q(t^{n})(t_{i}^{n} - t_{\infty}) \tag{3.25}$$

where

$$Q(t) = 2 \cdot \Delta x \cdot h/K(t_{i}^{n})$$

The change in enthalpy from time n to n+1 for the element i is

$$DH_{i}(n+1) = \bar{C}\left\{t_{i}^{n+1} - t_{i}^{n}\right\} \tag{3.26}$$

where

$$\bar{C} = \frac{1}{(t_{i}^{n+1} - t_{i}^{n})} \int_{t_{i}^{n}}^{t_{i}^{n+1}} C(t_{i})\, dt$$

The total enthalpy change from time 0 to N for the element i is

$$TDH_{i} = \sum_{n=1}^{N} DH_{i}(n) \tag{3.27}$$

The total enthalpy change from time 0 to N for the entire section is

$$TDHTA(N) = \left[\left\{(TDH_{1} + TDH_{(Lx+1)})/2\right\} + \left\{\sum_{i=2}^{Lx} TDH_{i}\right\}\right] \Big/ Lx \tag{3.28}$$

3.1.2.2 Model B2: Variable Temperature Fluid Media Environment. The applications of this model are similar to model A3 of Section 3.1.1.3.

The mathematical statement of the problem is as follows: The problem is represented in Figure 4 for the case of freezing a solid with a sensible temperature change of the fluid.

The governing differential equation, the first boundary condition and the initial condition are the same as Equations (3.18), (3.19) and (3.21) given in model B1 of Section 3.1.2.1.

The second boundary condition is:

$$\left.\frac{\partial t(x,\Theta)}{\partial x}\right|_{x=L} + \frac{h(t_\infty,\Theta)}{K(t)}\left[t(L,\Theta) - t_\infty(t,\Theta)\right] \tag{3.29}$$

$$t^n_{i+1} = t^n_{i-1} - Q(t,\Theta)(t^n_i - t^n_\infty(\Theta)) \tag{3.30}$$

where

$$Q(t, \ ) = \frac{2 \cdot \Delta x \cdot h(t^n_\infty,\Theta)}{K(t^n_i)}$$

The enthalpy change of the elements with time and the total enthalpy change of the entire section can be determined using Equations (3.26), (3.27) and (3.28). The energy balance equation similar to (3.12) is used for calculating the fluid temperature at successive steps.

$$t_{\infty_{n+1}} = t_{\infty_n} - \left(\frac{\dot{M}_b}{\dot{M}_a\ C_a}\right)(\mathrm{TDHTA}(n-1) - \mathrm{TDHTA}(n)) \tag{3.31}$$

By appropriately using Equations (3.23) through (3.31) the calculations can be repeated for the entire length of the conveyor.

3.2 Refrigeration System Model

The compression refrigeration system with ammonia as the refrigerant is widely used in the beef processing industry. Therefore, the data for the individual components of the refrigeration system are available for various capacities, for both chilling and freezing applications. The individual components are chosen according to the procedures given in catalogues [26]. The operating conditions are determined by solving simultaneous equations representing the performance of each component. The simulation procedure followed is similar to the one given by Stoecker [36] and is reformulated for use in computer simulation.

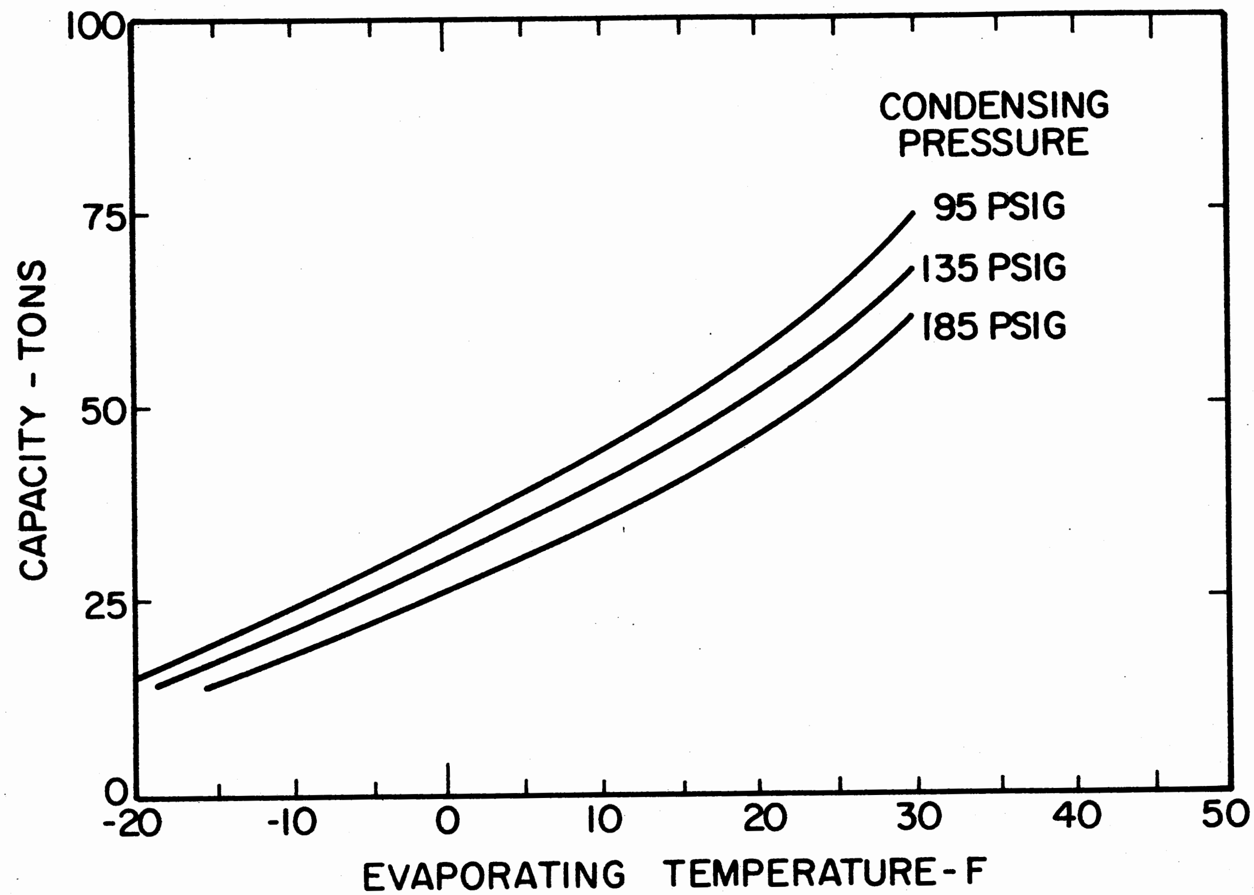
3.2.1 Compressor Data

The capacity of a typical ammonia compressor operating at a specific speed is shown graphically in Figure 6a. The capacity increases with an increase in evaporating temperature for a fixed condensing temperature and with a decrease in condensing temperature for a given evaporating temperature.

The power input to the compressor is shown graphically in Figure 6b. The power input increases with an increase in condensing pressure (temperature) for a fixed evaporating temperature. At the same time the capacity of the system decreases. This means the power input per ton increases considerably as the condensing pressure increases. The power input also increases with a rise in evaporating temperature for a fixed condensing pressure, but the capacity of the system increases at a higher rate, thus reducing the power input per ton.

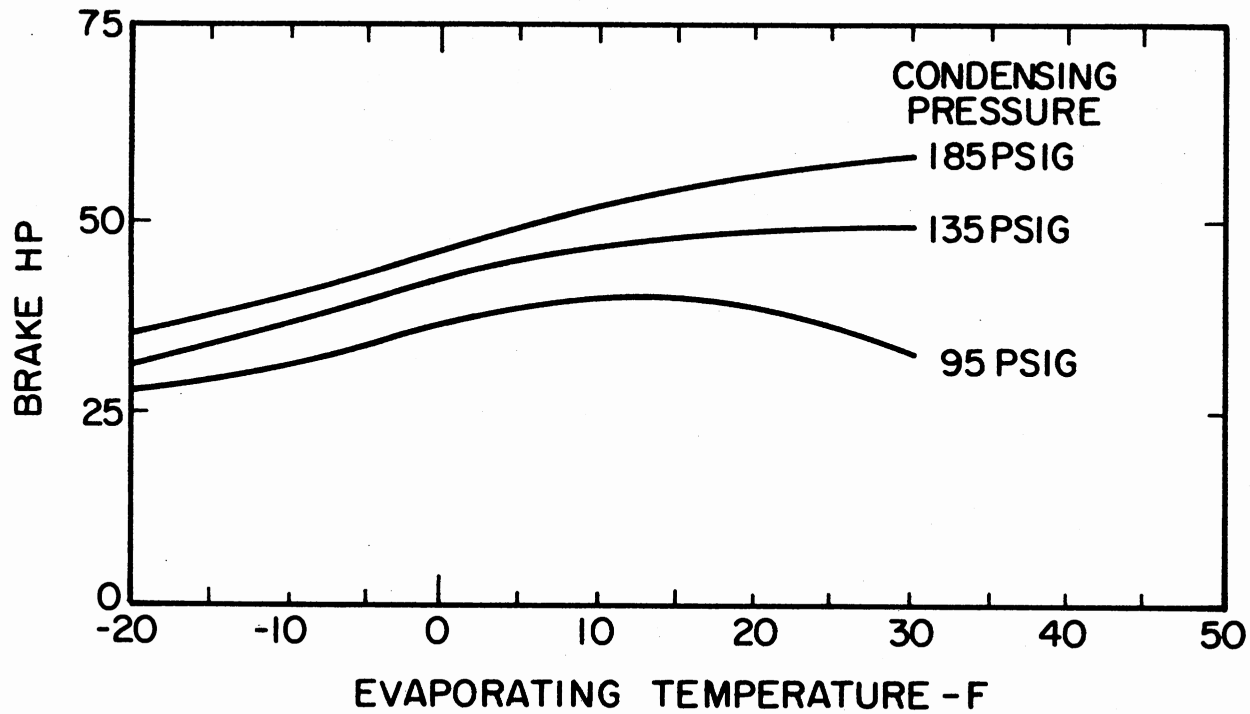
3.2.2 Condenser Data

The performance of a water cooled condenser with a constant water flow rate and a fixed entering water temperature is shown in Figure 7. The ordinate is the refrigeration capacity in tons, although the condenser itself does not develop the refrigeration capacity. The capacity



(a) Capacity

Figure 6. Performance of a Typical Ammonia Compressor



(b) Power Input

Figure 6. (Continued)

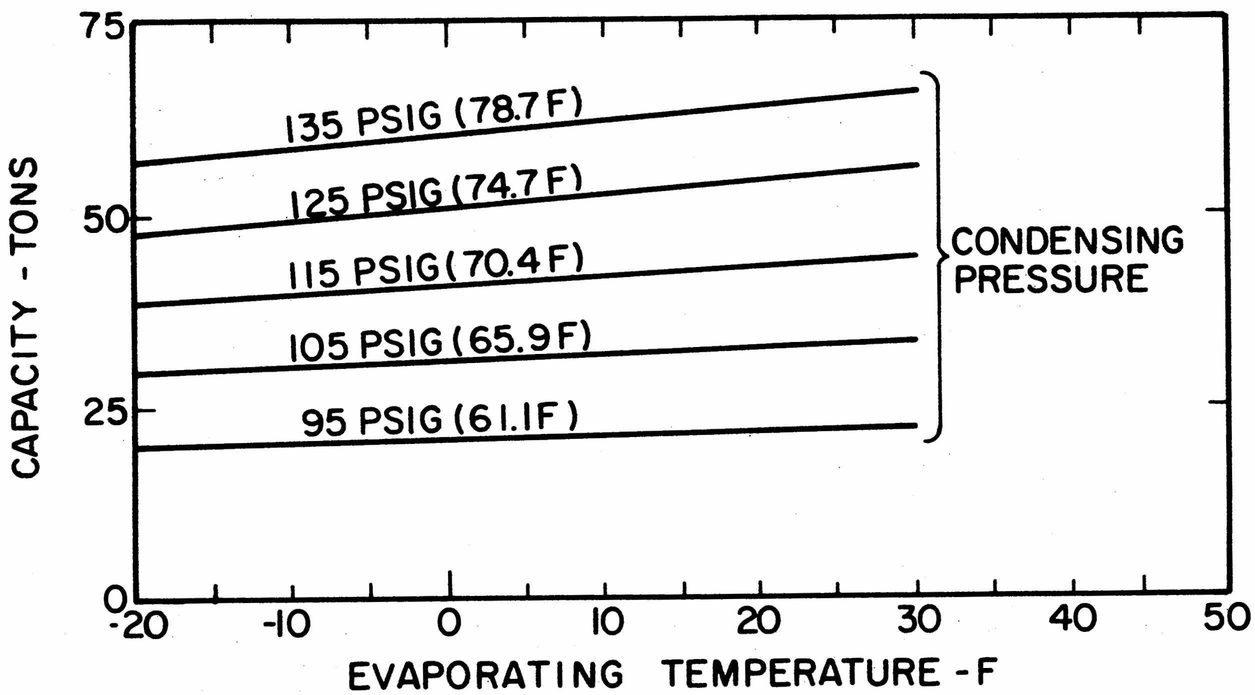


Figure 7. Performance of a Water-Cooled Ammonia Condenser

becomes greater with an increase in condensing pressure for a fixed evaporating temperature. This is due to the increase in mean temperature difference. The capacity also increases with a rise in evaporating temperature. This is due to the reduction of compressor work.

3.2.3 Throttling Device

To simplify the model, the throttling device is expected to deliver enough refrigerant to balance the compressor flow rate for all evaporating and condensing conditions. A float valve or a thermostatic expansion valve would meet this requirement.

3.2.4 Evaporator Data

The performance of a finned coil evaporator is shown in Figure 8. The capacity increases with increase in temperature difference between the entering fluid and the evaporating refrigerant temperature.

3.2.5 Condensing Unit

The combined performance of the compressor and the condensor is called the condensing unit performance. The development of this characteristic is shown in Figure 9. The intersection of the compressor and the condensor characteristics gives the capacity and the condensing temperature at the corresponding evaporating temperature. Both capacity and condensing temperature characteristics as a function of evaporating temperature are shown in Figure 10. These characteristics are expressed by equations of the form:

Capacity = CAPC(1) + (CAPC(2)*EVPT) + (CAPC(3) * (EVPT**2))

(3.32)

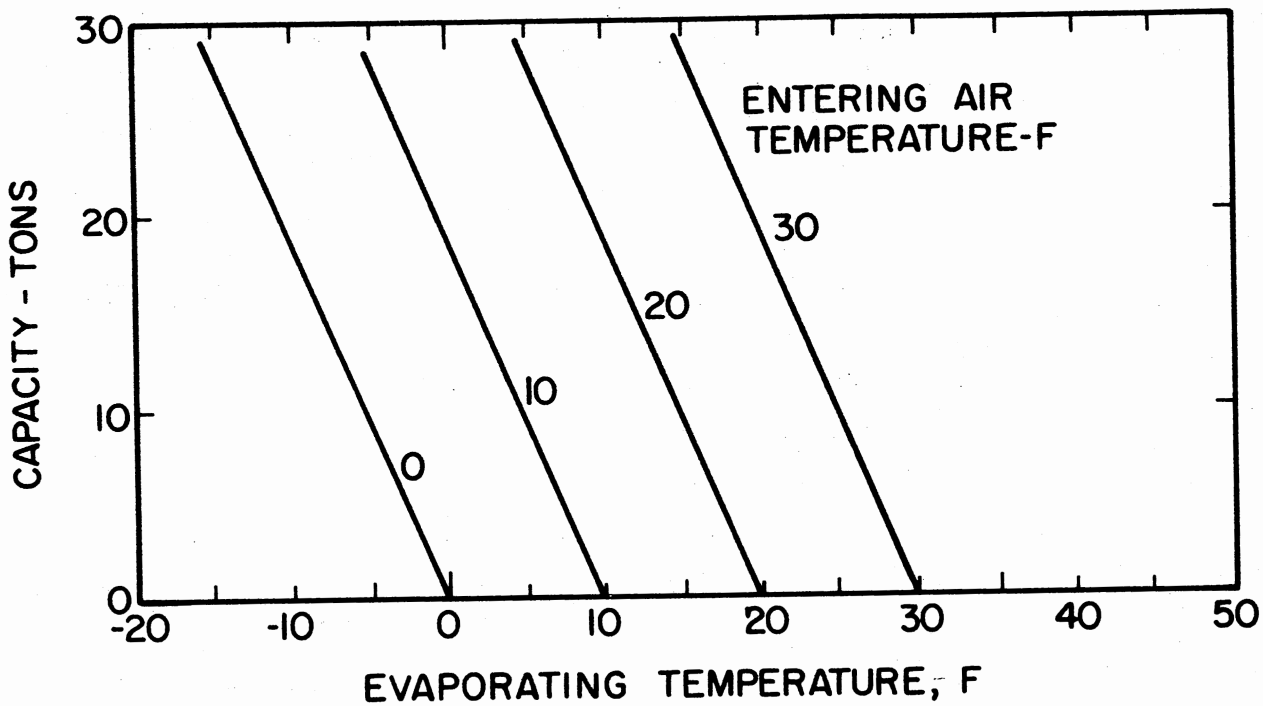


Figure 8. Performance of an Air-Cooled Evaporator

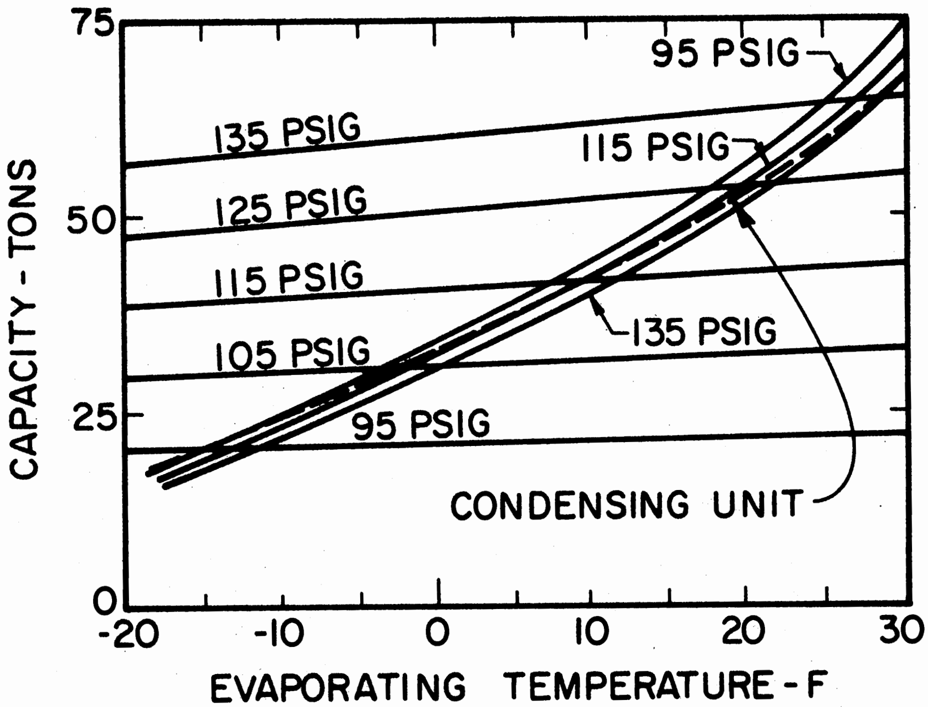


Figure 9. Performance of a Condensing Unit

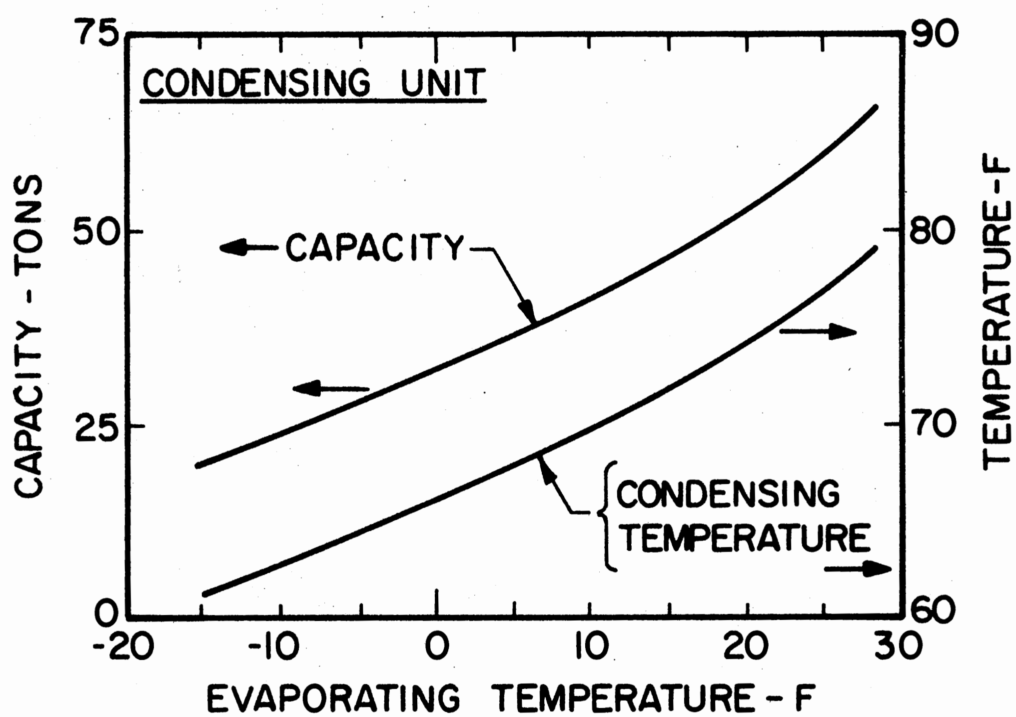


Figure 10. Condensing Unit Capacity and Condensing Temperature as a Function of Evaporating Temperature

where

capacity = refrigeration capacity in tons

CAPC(n) = constants, for n = 1, 2, 3 (Appendix E)

EVPT = evaporating temperature of the refrigerant

$$\text{CONT} = \text{CONTC}(1)+\text{CONTC}(2)*\text{EVPT}+\text{CONTC}(3)*(\text{EVPT}**2) \tag{3.33}$$

where

CONT = condensing temperature (F)

CONTC(n) = constants, for n = 1, 2, 3 (Appendix E)

3.2.6 Balance Point of the System

The evaporator characteristics can be represented by a straight line in the normal range of applications, and is of the form:

$$\text{Capacity} = -\text{Slope} * \text{EVPT} + \text{XC} \tag{3.34}$$

where

Slope = Refrigeration capacity in tons/F (Appendix E)

Xc = Constant for a given evaporator and entering air temperature (Appendix E)

Using the condensing unit characteristic and the evaporator characteristic given by Equations (3.32) and (3.34), the capacity and the evaporating temperature of the refrigerant are obtained. If the refrigeration system is operating at steady state and handling a given amount of load, the fixed temperatures of the air entering the evaporator and the evaporating temperature are determined using Equations (3.32) and (3.34). Then Equation (3.33) is used to calculate the condensing temperature of the unit. The procedure for obtaining capacity, condensing temperature, and the evaporating temperature of the system is shown graphically in Figure 11.

The power input to the compressor is represented by

$$\begin{aligned} \text{BHP compressor} = \; & C(1) + C(2) * CONT + C(3) * EVPT \\ & + C(4) * CONT * EVPT + C(5) * CONT ** 2 \\ & + C(6) * EVPT ** 2 \end{aligned} \tag{3.35}$$

where C(n) = constants, for n = 1, 2, . . ., 6 (see Appendix E).

Now Equation (3.35) is solved to get the power input to the compressor for this operating condition. This is shown in Figure 12.

The power input to the evaporator fan for a fixed (design) face velocity is represented by the equation of the form:

$$\text{Evaporator fan BHP} = CE_n * cfm \tag{3.36}$$

where CE_n = constant, for a coil n rows deep (see Appendix E).

Using the compressor motor efficiently, which is assumed constant, the input to the compressor motor is determined. From the catalogued pressure drop and water quantity to the condenser, and an assumed efficiency of the condenser water pump and its motor, the input to the circulating pump motor is determined. The fan horsepower required for circulating air through the evaporator is obtained from the evaporator catalogued data and is given by Equation (3.36). It is assumed that the evaporator fan motor is also of constant efficiency. The sum of these three inputs is the total power required by the refrigeration system. The efficiencies of the various components of the refrigeration system used in the present study are given in Appendix E.

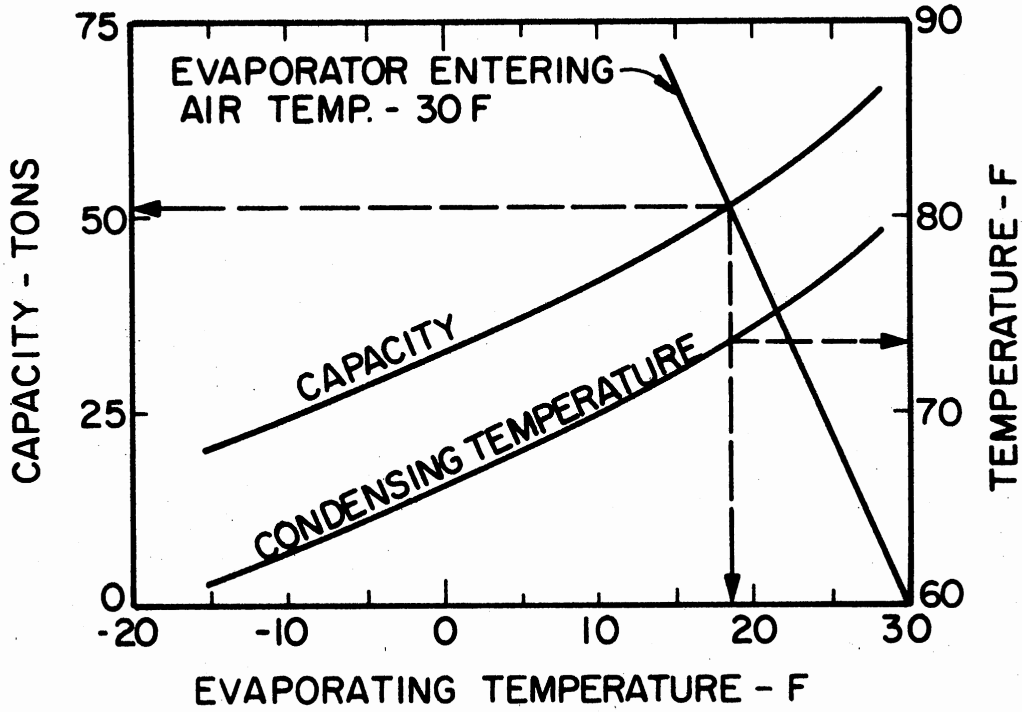


Figure 11. Performance of a Complete Refrigeration System

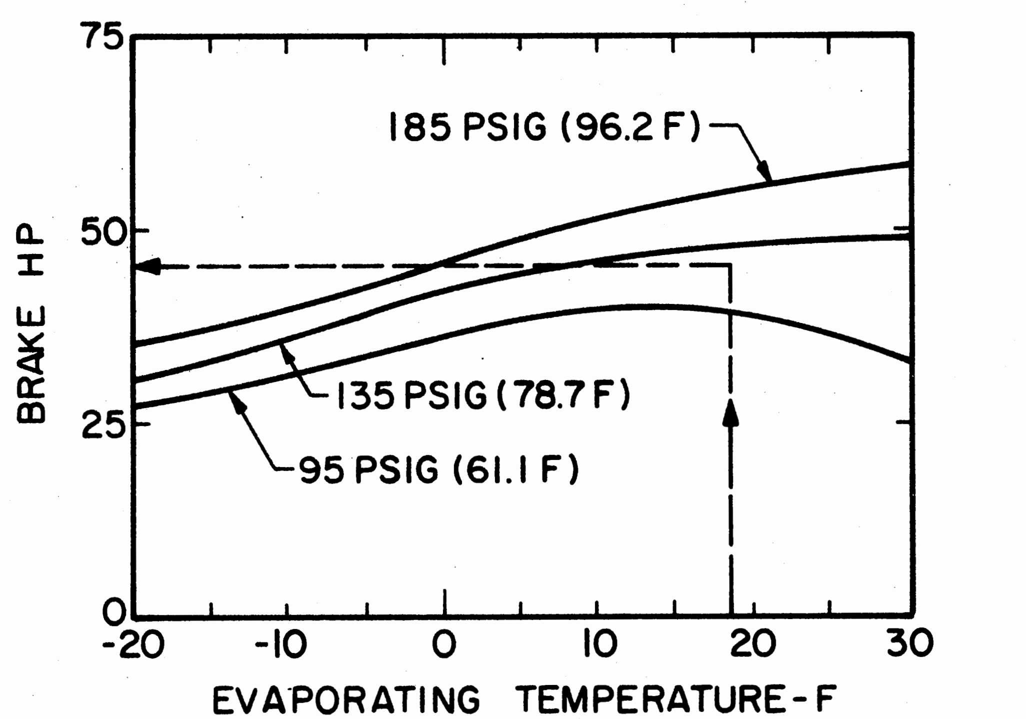


Figure 12. Power Input to the Compressor at the Operating Condition

3.3 Fluid Moving System Model

In this subsystem the main parameter to be estimated is the power requirement for circulating a given quantity of fluid at a given velocity through the conveyor system. The fluid and the solid on the conveyor will be moving in opposite directions in a duct, as shown in Figure 13. Technical information of this type of system used in practice is not available and therefore will be developed from basic fluid flow theory. The hydraulic diameter and friction factor approach, as available in References [4] and [38], will be used in estimating the power requirement.

In the following section, the procedure for estimating the fan horsepower required to circulate air through the conveyor system is developed. A similar approach can be used with other fluid media as well.

The hydraulic diameter is given by:

$$\text{Deq} = 4 * (\text{CH} - \text{BPHF}) * \text{CW}/(4 * \text{CW} + 2 * \text{CH}) \qquad (3.37)$$

The estimate of the hydraulic diameter and the average friction factor are given in Appendix F. The following equation is used to estimate the pressure drop on the conveyor.

$$f = (\Delta p)_c \, \text{Deq} \, g_c/(2 \cdot 0 * \rho * V^2 * L) \qquad (3.38)$$

$$\Delta P_c = \frac{2 \, f \, \rho \, V^2 \, L}{\text{Deq} \, g_c} \, \frac{27.7}{144} \text{ inches of water} \qquad (3.39)$$

It may be assumed that one velocity head is lost in the connecting duct work at the entrance and one velocity head at the exit.

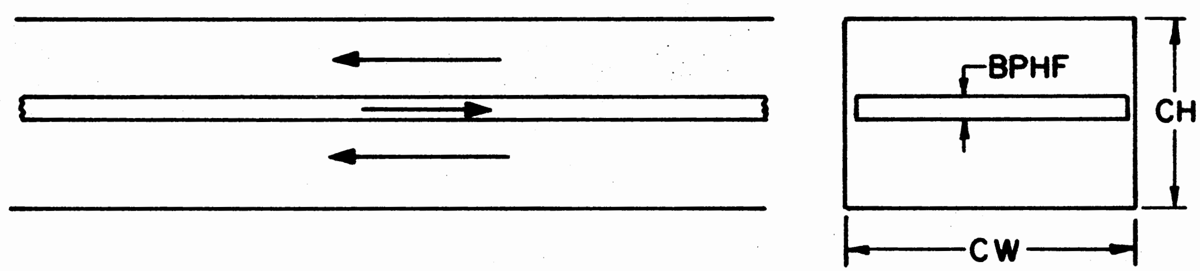


Figure 13. Conveyor System Arrangement

$$\Delta P_{\ell} = 2 * 0 \cdot 8303 * \rho * \left(\frac{V * 60}{1000}\right)^2 \tag{3.40}$$

$$\Delta P_t = \Delta P_c + \Delta P_{\ell} \tag{3.41}$$

$$\text{Conveyor fan BHP} = \frac{(\text{cfm}) * \Delta P_t}{(6350 * \eta_{fan})} \tag{3.42}$$

The Equation (3.42) is used for estimating the fan horsepower required to circulate the air on the conveyor. Using the assumed fan motor efficiency, the input to the conveyor fan motor is determined. In the present study, the efficiencies of the fan and its motor are assumed constant and are given in Appendix F.

CHAPTER IV

SELECTION OF HEAT TRANSFER MODELS FOR HOT BONED BEEF PROCESSES

In Section 3.1 various kinds of heat transfer models are treated which are applicable to conveyorized heat transfer systems. In this chapter each model will be studied to test its applicability for developing the design parameters for the chilling and freezing processes of hot boned beef. In the present study, the chilling and freezing of hot boned beef will be referred as hot boned beef processes.

4.1 Modeling of Hot Boned Beef Cuts

In the case of one-dimensional heat transfer, the thickness of beef cuts to be processed is one of the important parameters for calculating the temperature profile and the energy level. The classification and the average thickness of hot boned beef pieces were derived from a careful study at the Oklahoma State University meat laboratory and are given in Appendix G. It is assumed that different classes of cuts are processed on different conveyors.

Normally, the product is bagged at some stage before it reaches the consumer. Therefore it is assumed that the product of each group is vacuum bagged and has the average thickness as given in Appendix G. Evacuation removes the air between the bag and the product thereby allowing a high heat transfer coefficient. Bagging of the product has the following advantages:

1. It reduces the moisture loss.

2. It gives improved shelf life due to reduced contamination.

3. It reduces the contamination of the cooling fluid.

4.2 Chilling of Hot Boned Beef

The physical properties of beef are approximately constant above the freezing temperature as explained in Section 2.2, and are assumed to be constant in the chilling process. The values used in the chilling process calculations are given in Appendix B. The hot boned beef is at an average temperature of about 100 F when it enters the chilling process. The final average temperature of the product depends upon the governmental regulations [16] and/or the marketing requirements. After a careful study of these requirements and from consultations with the project advisory committee [25], it was decided to use 40 F as the average temperature to be attained at the end of the chilling process. The USDA recommends reduction of the internal temperature of meat to 40 F within 16 hours as reported by Cutting [16]. This temperature can be easily achieved in the hot processing method because of the thinness of the cuts. The problem of cold shortening due to the reduction of temperature below 50 F in less than 10 hours is eliminated by electrical stimulation of beef carcass sides [40]. The above information and the data from Appendix B are used in the following chilling models. The assumptions and limitations for each model are the same as the ones explained in Chapter III.

4.2.1 Model A1: Negligible Internal Thermal Resistance

The theoretical model A1 of Section 3.1.1.1 is studied in this section for its applicability to the hot boned beef process. The equations and other data of this model are explained in Appendix D. The counter flow heat exchanger approach was used to estimate the cooling time as a function of the heat transfer coefficient and the entering cooling air temperature; the results are shown in Figure 14.

The cooling times predicted by this method raised doubts about their practical feasibility. Therefore, the validity of this model was studied and found to be erroneous for high heat transfer rates as explained by model A1 in Section 3.1.1.1. For a four-inch thick section, the heat transfer coefficient should be less than 0.2 Btu/Hr-Ft^2-F to satisfy the thin wall criterion. Therefore the inaccuracy of the model at high heat transfer rates is obvious. This model can be used to calculate the rate of cooling in chill rooms where the heat removal is very slow [7, 8] (which is required for high quality) or to find the product load in chill rooms if the process time is experimentally established. The cool air is circulated over the carcasses in the entire room. The effective air velocity over the carcasses is one to three feet per second [21]. Thus the heat transfer coefficient is very low ($h < 1$ Btu/Hr-Ft^2-F), resulting in a slow rate of chilling and a long process time.

In the present hot processing method, the cuts are relatively thin and there is no time restriction from the quality aspect. Therefore the cooling time is completely controlled by the heat transfer process. The chilling process equipment has to be designed for a minimum energy requirement and short cooling time. For a given production rate, the

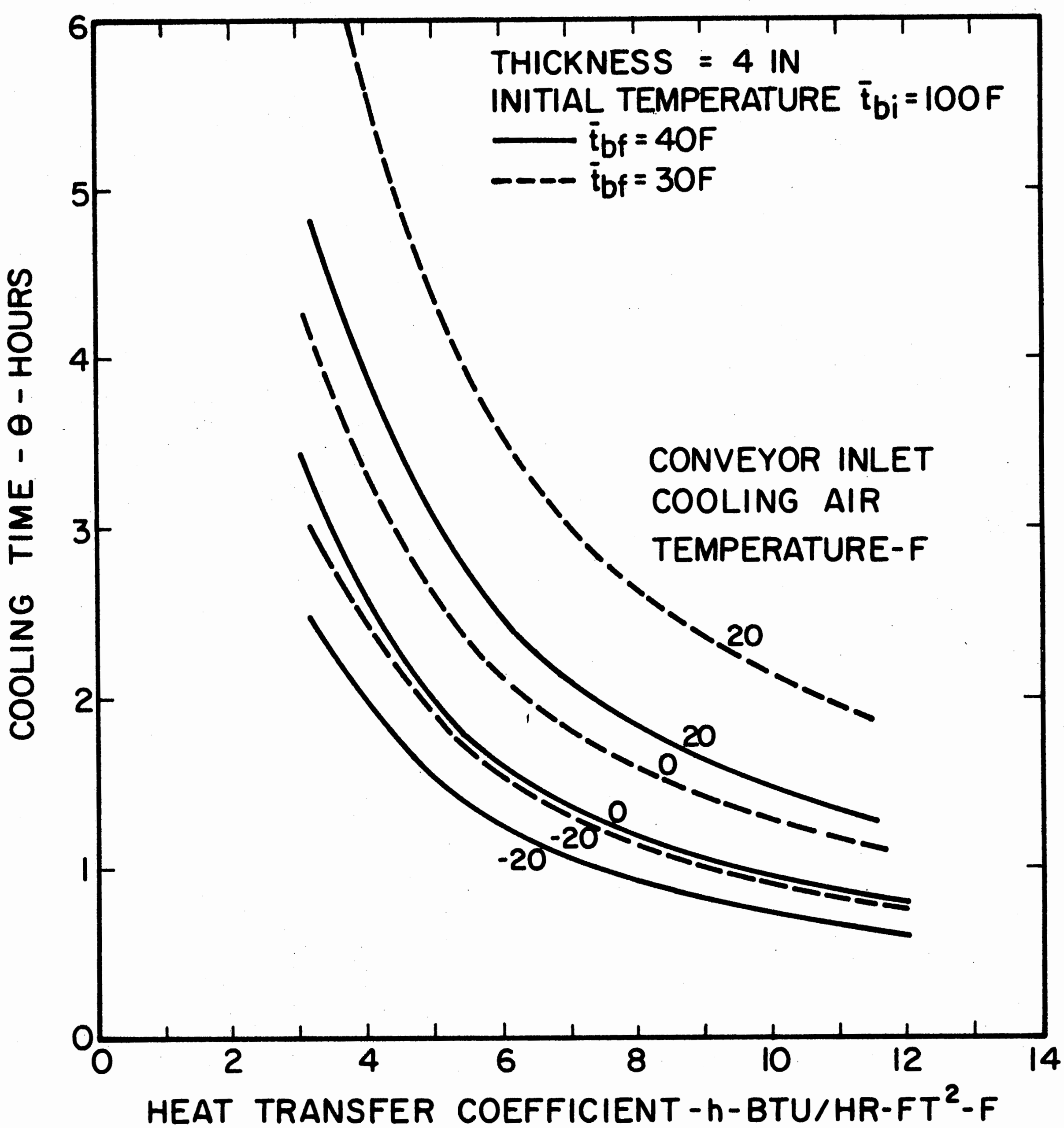


Figure 14. Cooling Time Estimate by Model A1

conveyor area and the cooling space are directly proportional to the cooling time. To meet these requirements a more realistic heat transfer model is required.

4.2.2 Model A2: Constant Temperature Fluid Media Environment

The theoretical model A2 of Section 3.1.1.2 is studied in this section for its application to the hot boned beef chilling process. Equation (3.9) is used to plot Figures 15 and 16. In these figures, the cooling time is computed for various thicknesses of the beef pieces and its average temperature as a function of the heat transfer coefficient for a fixed temperature of the cooling media. These data are summarized in Figure 17 for an average beef piece temperature of 40 F. The influence of the cooling media temperature and the product thickness in computing the cooling time is shown in Figure 18.

The influences that can be derived from these figures are:

1. The cooling time estimates by this model are reasonable compared to the known values [25].

2. Increasing the heat transfer coefficient beyond 10 Btu/Hr-Ft^2-F the reduction in cooling time is very low. This favorably supports air as the cooling media rather than liquids due to the additional problems associated with liquids as cooling media.

3. If the heat transfer coefficient h is

a. less than 1 Btu/Hr-Ft^2-F, the energy transfer is governed by the heat transfer coefficient;

b. in between 1 and 10 Btu/Hr-Ft^2-F, the energy transfer is controlled by both internal resistance and the heat transfer coefficient;

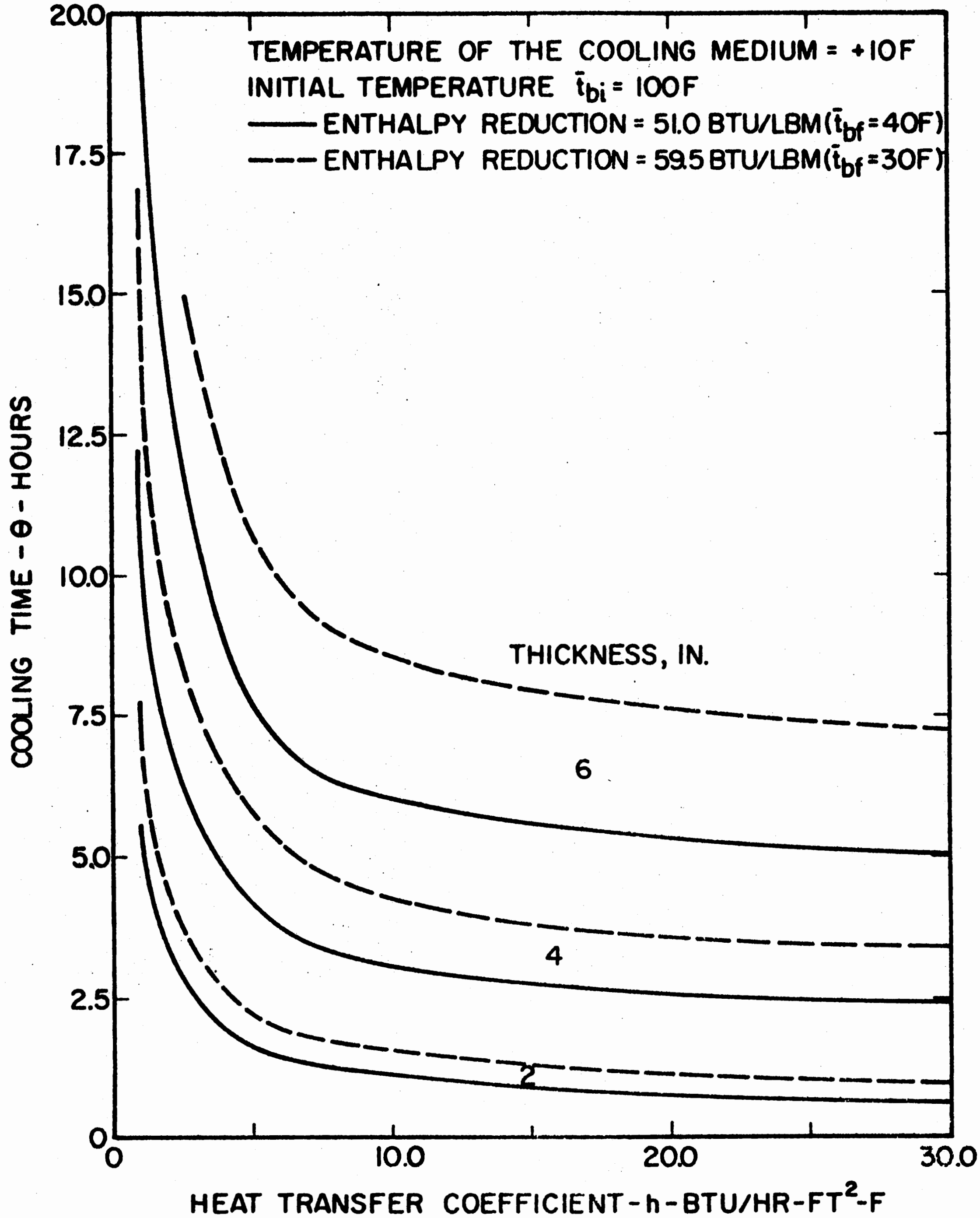


Figure 15. Cooling Time Estimate by Model A2 for Cooling Medium Temperature of 10 F

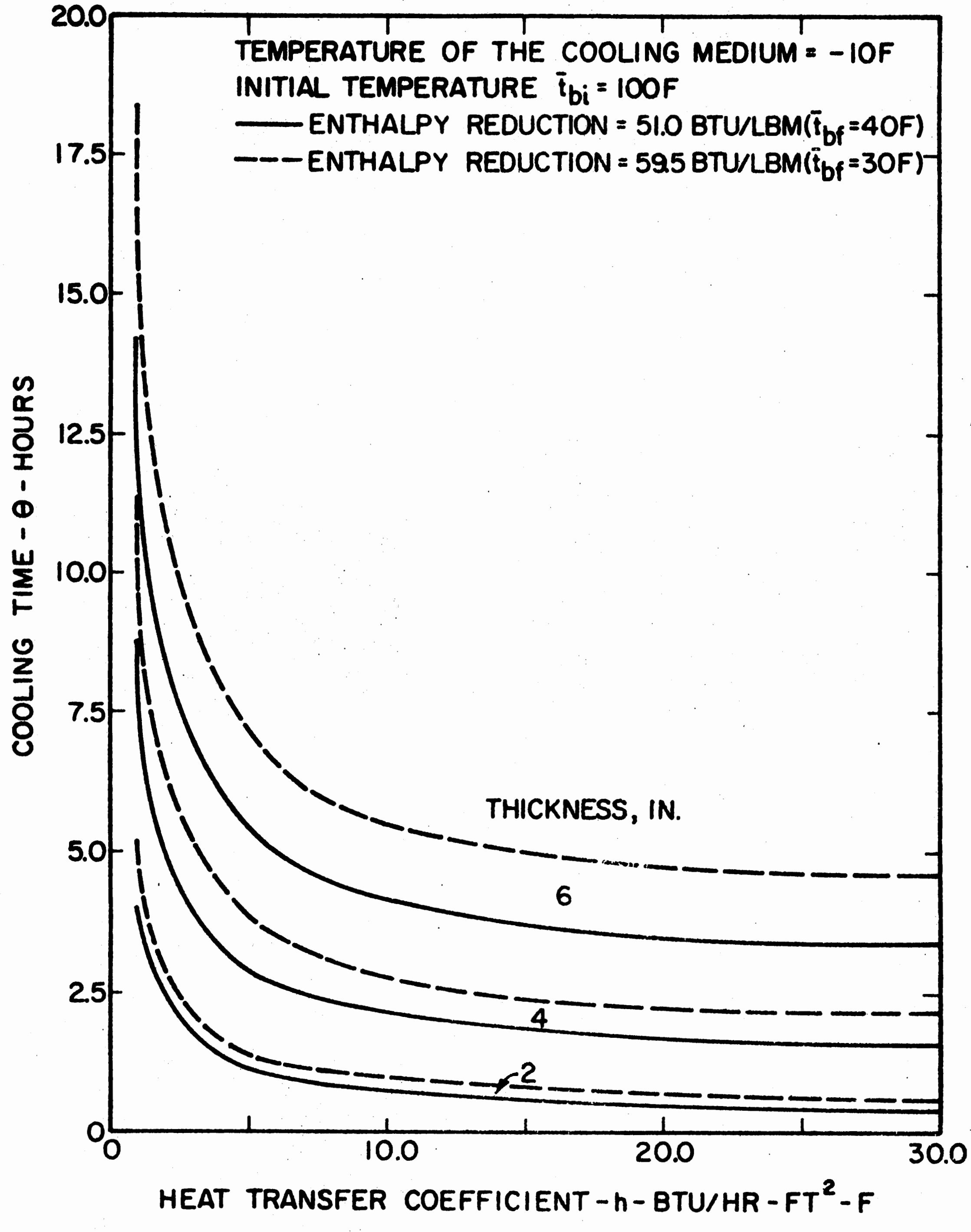


Figure 16. Cooling Time Estimate by Model A2 for Cooling Medium Temperature of -10 F

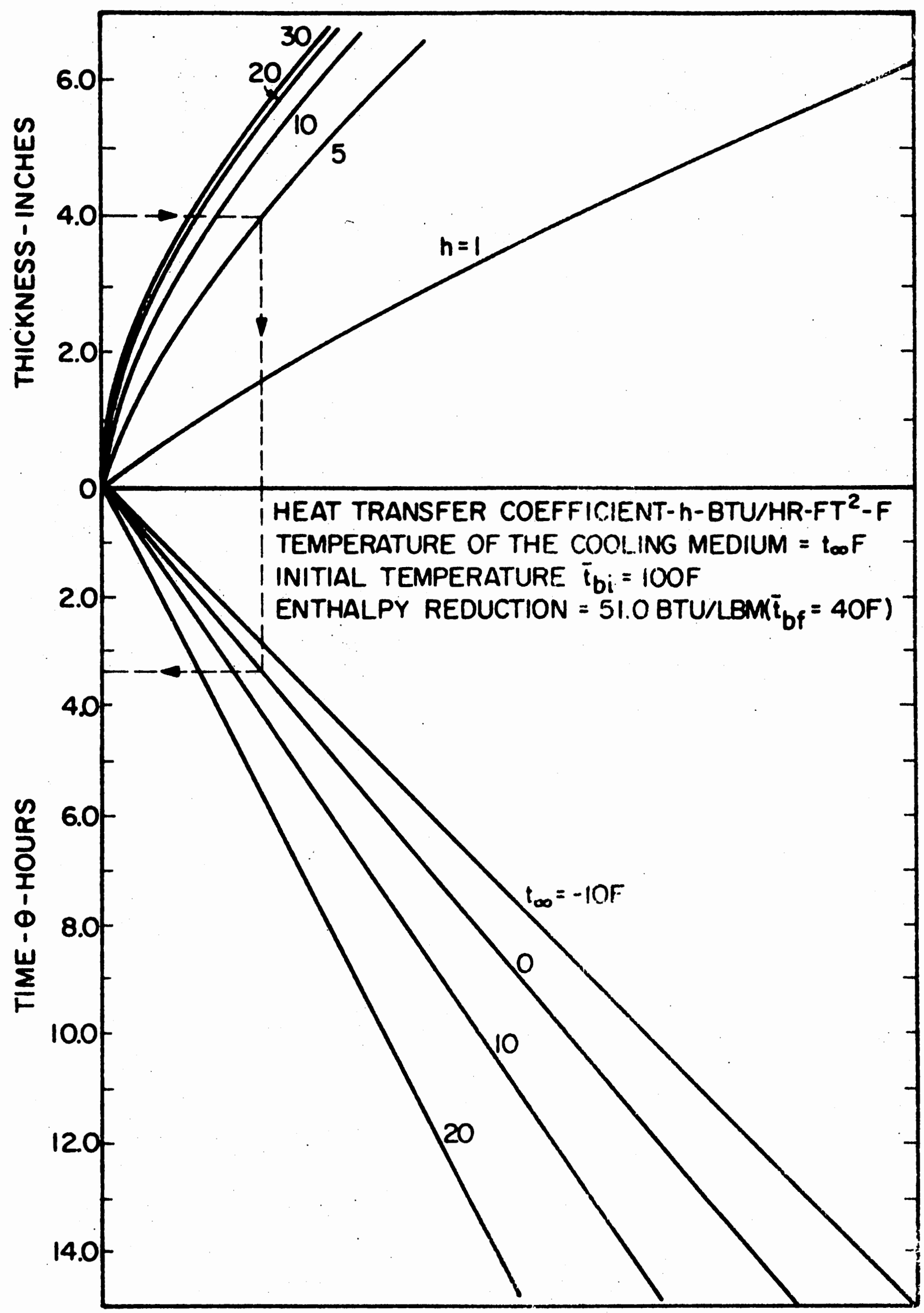


Figure 17. Influence of Various Parameters on Cooling Time by Model A2

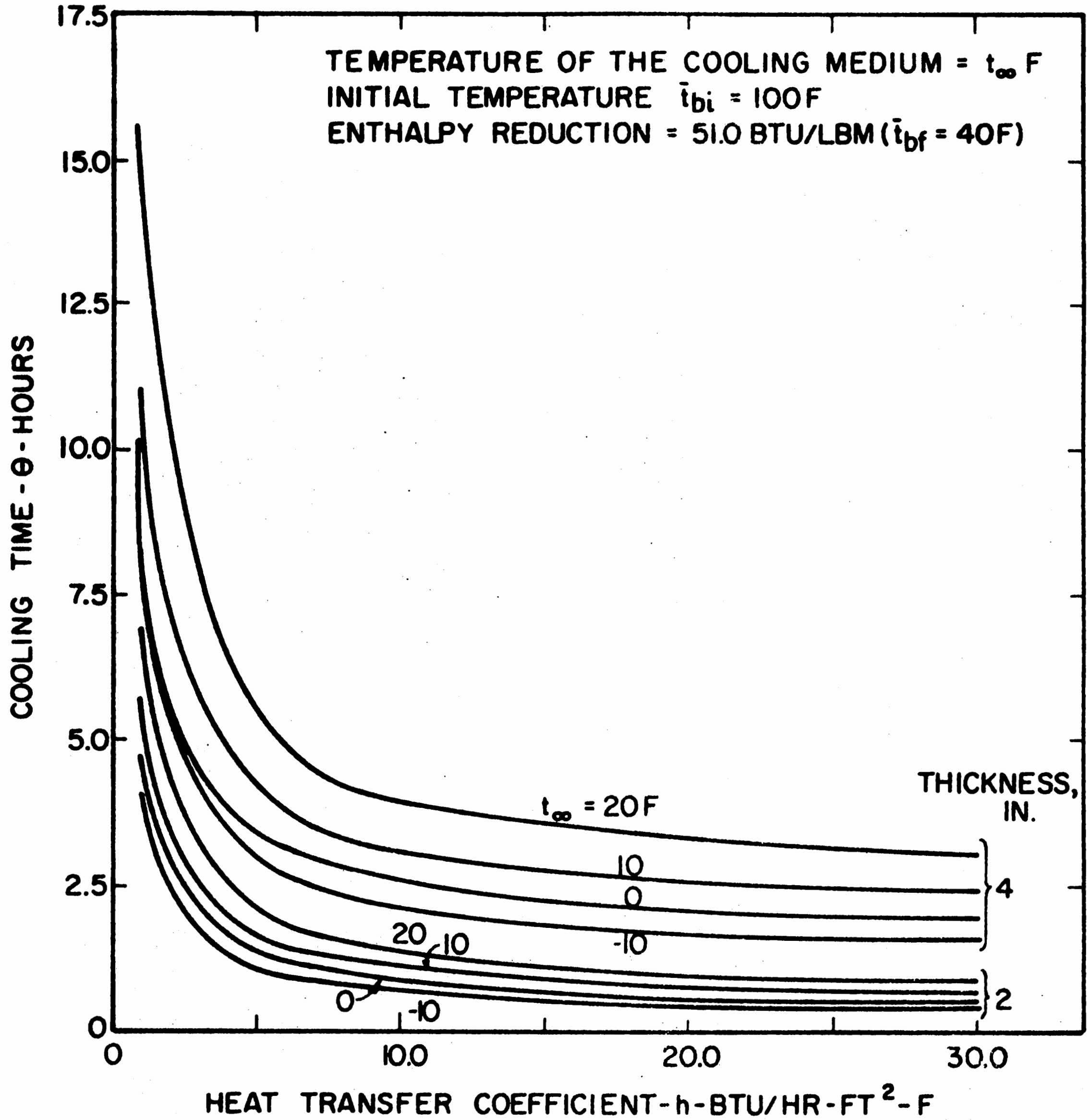


Figure 18. Influence of Cooling Medium Temperature and Product Thickness on Cooling Time

c. greater than 10 Btu/Hr-Ft^2-F, the energy transfer is controlled by the internal resistance.

4. Figure 18 shows the influence of the cooling media temperature and the product thickness in estimating the cooling time. The importance of the cooling media temperature increases

a. with the increase in thickness of the beef pieces,

b. with the increase in temperature of the cooling medium.

The cooling medium temperature is very important for estimating the cooling time at the relatively high temperature of the cooling fluid (which is the case for minimum refrigeration system energy requirements). Therefore, an improved model is required to properly represent the cooling fluid temperature in a conveyorized system. This leads to the requirement to use the third model developed in Section 3.1.1.3. In the following chapter, this model A3 is used in developing design parameters for chilling the hot boned beef.

4.3 Freezing of Hot Boned Beef

The physical properties of beef are strongly dependent on the temperature in the freezing range as explained in Section 2.2. The values used in the freezing process calculation are given in Appendix B. The hot boned beef, which is chilled to 40 to 45 F is the input material for this process. As the beef enters the freezing process, it is assumed to be at an average temperature of 45 F (a) to be conservative and (b) as used by other research workers [10, 29]. The final average temperature of the product when frozen varies from 0 to -5 F. An enthalpy reduction of 120 Btu per pound was used, which results in a final average temperature of -3 F. The freezing process normally takes 48 hours in the cold processing method. There is no time restriction in freezing the hot boned beef. The above information and the data from Appendix B are used in the following freezing models. The assumptions and the limitations for each model are the same as explained in the theoretical development chapter.

4.3.1 Model B1: Constant Temperature Fluid Media Environment

The theoretical model B1 of Section 3.1.2.1 is studied in this section for its applicability to the hot boned beef freezing process. Equations (3.23) and (3.28) were used to plot the Figures 19 and 20. The freezing time for reducing the enthalpy by 120 Btu per pound is shown for various thicknesses of the beef pieces as a function of the heat transfer coefficient and a fixed temperature of the cooling medium. These data are summarized in Figure 21. The influence of the cooling media temperature and the product thickness in computing the freezing time is shown in Figure 22.

The inferences that can be derived from these figures are similar to the ones in the chilling process, as explained in Section 4.2.2. The freezing time calculated is in between the values predicted by the modified Plank's Equation and the Cullwick and Earle approach as given in [10]. The small differences (less than 10 percent) are assumed to be due to the differences in the properties of the beef used. An improved model is required for a conveyorized freezing system design due to reasons similar to those of the conveyorized chilling process. This leads to the requirement to use model B2 which was developed in Section 3.1.2.2. Model B2 is, therefore, used in developing design parameters for freezing hot boned beef in the following chapter.

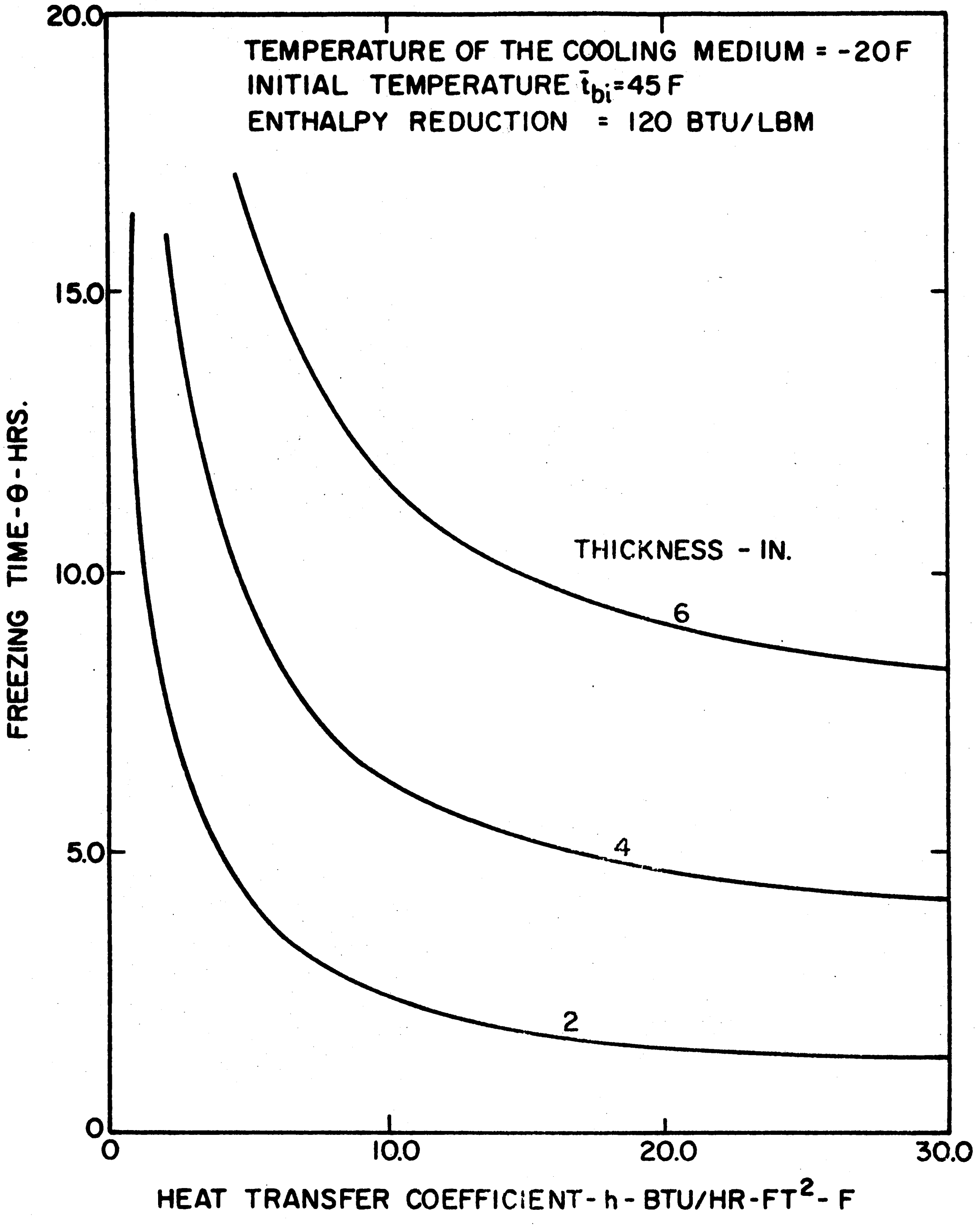


Figure 19. Freezing Time Estimate by Model B1 for Freezing Medium Temperature of -20 F

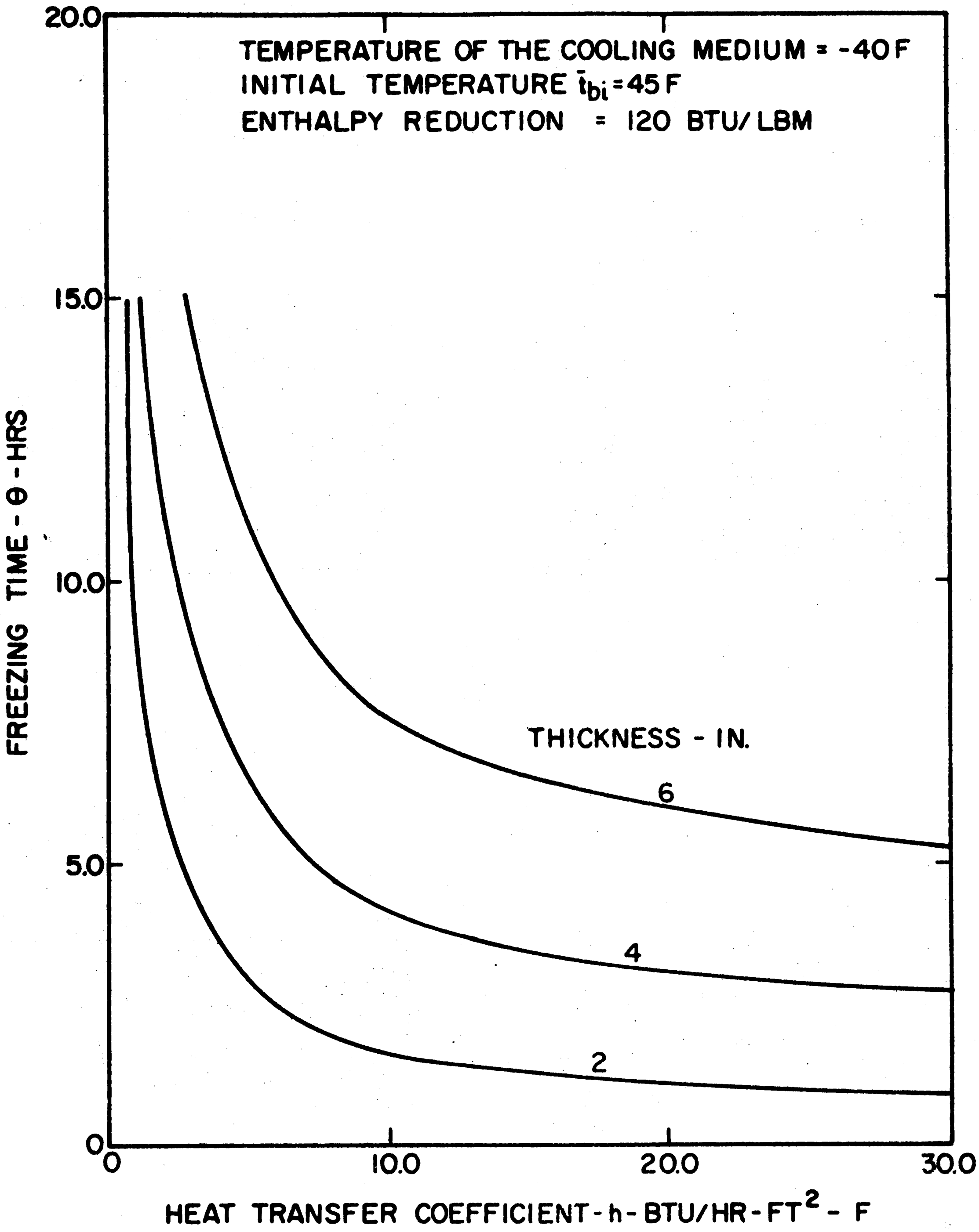


Figure 20. Freezing Time Estimate by Model B1 for Freezing Medium Temperature of -40 F

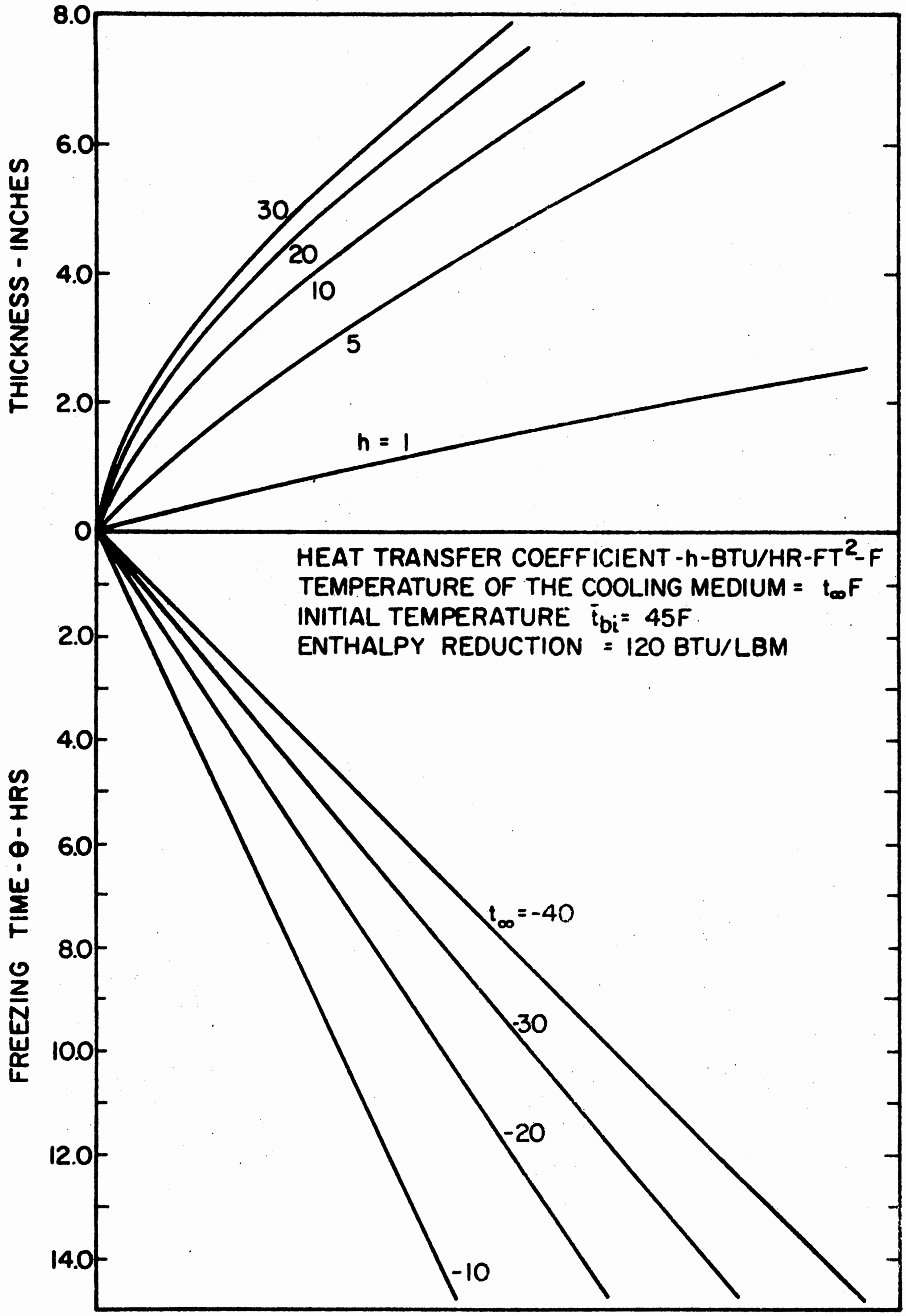


Figure 21. Influence of Various Parameters on Freezing Time by Model B1

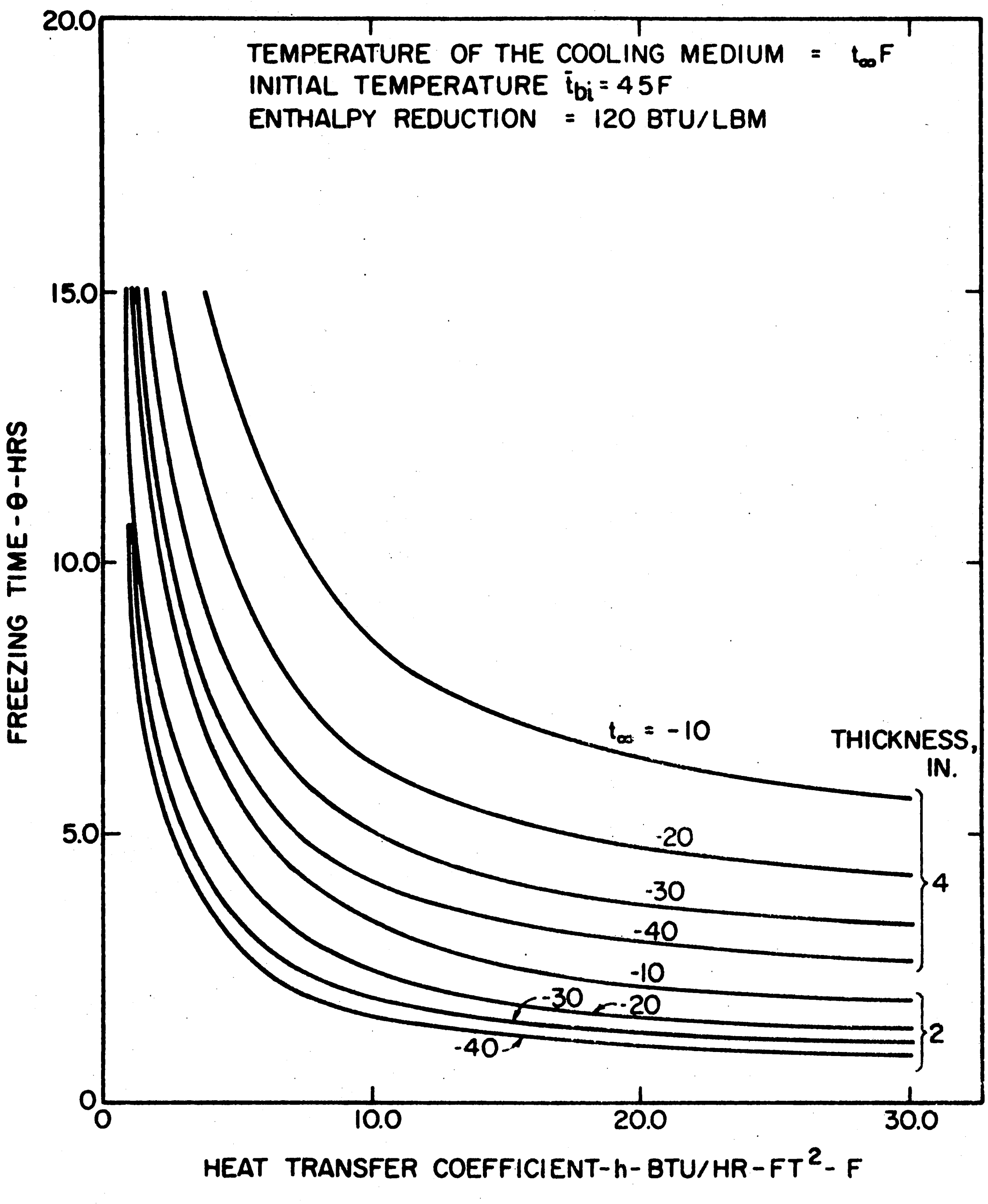


Figure 22. Influence of Cooling Medium Temperature and Product Thickness on Freezing Time

CHAPTER V

DEVELOPMENT OF DESIGN PARAMETERS FOR HOT BONED BEEF PROCESSES

The design parameters depend upon the total system, therefore it is necessary to develop a complete system model. For the hot boned beef processes the total system is made up of three major subsystems, which were explained in the theoretical development chapter. The heat transfer models required for this purpose were explained in Chapter IV. After the complete system model is developed, a parametric study was conducted to develop the design parameters which can be used in designing the conveyorized chilling and freezing systems.

5.1 Complete System Model

The total system, which is made up of three subsystems, consists of many parts. Each subsystem model was developed by modeling all the parts in it, with the provision for the input and output to the other subsystems. They are:

1. the conveyorized heat transfer system,
2. the refrigeration system,
3. the air moving system,

and are shown in Figure 23.

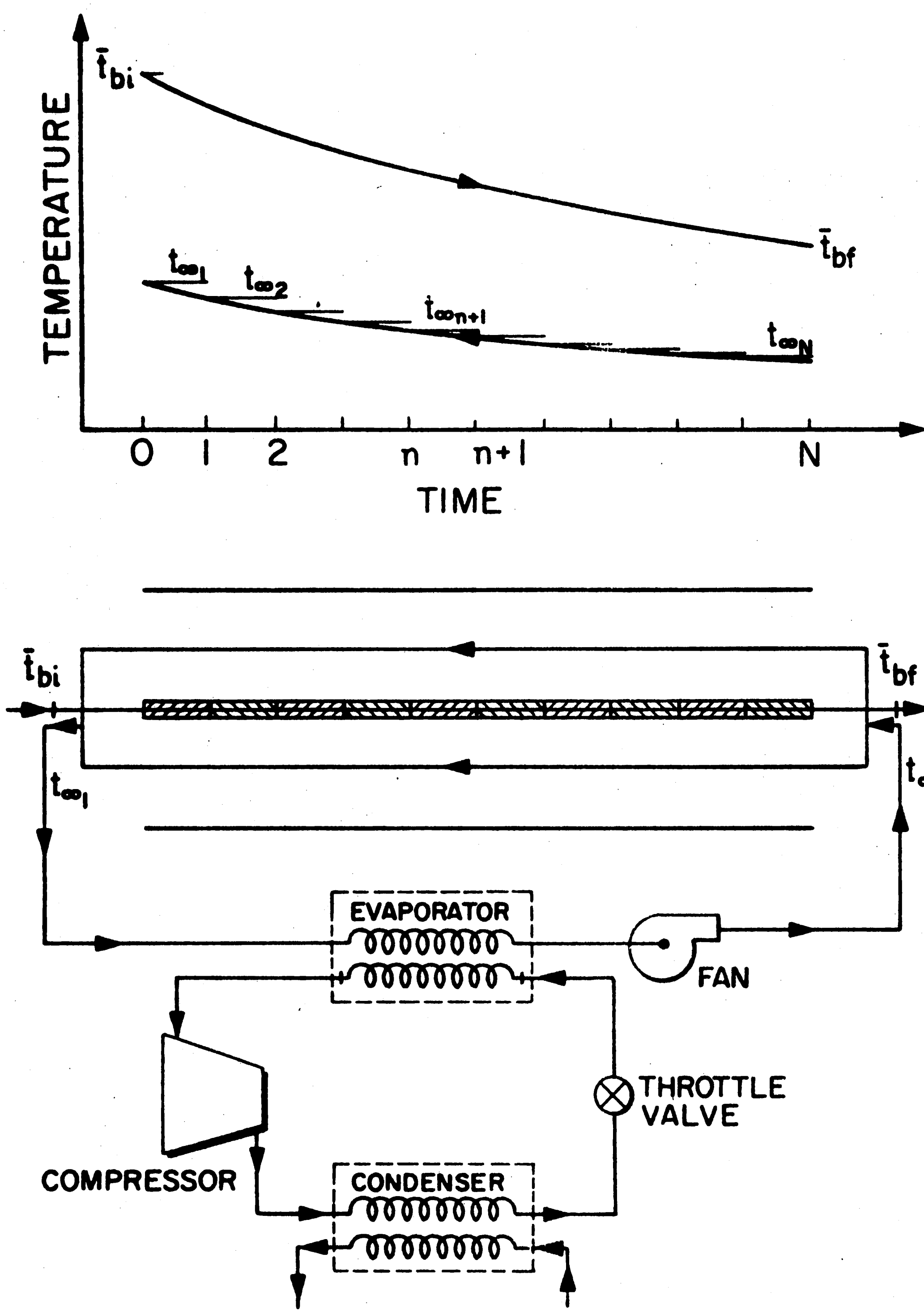


Figure 23. Schematic of the Complete System

5.1.1 Conveyorized Heat Transfer System

The model of this system consists of an appropriate heat transfer model to calculate the energy level in the solid and an energy balance equation to account for the transfer of energy between the solid and the fluid. The chilling process time is obtained using Equations (3.11) and (3.12) and is verified using Equations (3.16), (3.17) and (3.12) for the variable fluid temperature. To further check the results, two process times were calculated using Figure 3 which is for constant fluid temperature given in Chapter III. One process time was for the inlet temperature and the second for the outlet temperature of the fluid to the conveyor. The chilling process time obtained with variable fluid temperature was found to be between the two process times calculated above. The freezing process time was obtained using the Equations (3.23) through (3.31). The results for the variable fluid temperature were checked as above with the results given for the constant temperature fluid by Heldman [10] and with the results developed in model B1 in Chapter IV. A refrigeration model is used to maintain the temperature potential across the conveyor. A heat transfer correlation connects the air moving system and the heat transfer system.

5.1.1.1 Heat Transfer Coefficient. There are many correlations available in the literature for estimating the heat transfer coefficient. The following equation which applies to a flat plate is chosen because it has more flexibility in accounting for property variations.

$$st_x \, Pr^{0.4} = 0.0295 * Re_x^{-0.2} \tag{5.1}$$

For standard air with a Prandtl number of 0.72 and specific heat of 0.24 Btu/Lbm-F, the heat transfer coefficient h can be written as

$$h = 36.3 * (\rho\ V)^{0.8} * (\mu/l)^{0.2}\ \text{Btu/Hr-Ft}^2\text{-F} \qquad (5.2)$$

The range of applications of the various equations and their source are given in Appendix C.

5.1.2 Refrigeration System

Although the heat transfer from the beef is a transient, the conveyor operates at a steady state between the given inlet condition and the final required outlet condition of the beef. Several refrigeration systems were developed as explained in Section 3.2. They have various load capacities for a fixed cooling air temperature or for maintaining different cooling air temperatures for a given load. The refrigeration model computes the power requirement to produce the temperature potential for energy transfer. Equations (3.32) through (3.36) are used in this model and the constants in those equations are given in Appendix E. The constants were developed for two types of refrigeration systems. One uses reciprocating ammonia compressors for the chilling process applications and the other uses rotary compressors for the freezing process applications. In the present application, a given refrigeration system will be used for maintaining particular evaporator entering and leaving air temperatures for a fixed load. Thus the conveyor entering and leaving air temperatures are known for a fixed load and refrigeration system. The time required for the process has to be calculated iteratively until it satisfies the required final condition of the beef. The parametric study was conducted by changing the refrigeration systems, thus changing the temperature potential and the quantity of air used for the energy transfer for various product thicknesses.

5.1.3 Air Moving System

The fan model, which was developed in Section 3.3, is used for calculating the fan power required to circulate air at the required velocity and quantity fixed by the refrigeration system, using Equations (3.38) through (3.41). A friction factor value of 0.01 was used in the simulations and its influence is explained in the parametric study presented later in this chapter. The required area of the conveyor is calculated from the known product load, time required for the process, and the beef loading factor (Appendix G). The fixed quantity and velocity of air and the required length to width ratio of the conveyor, are used for calculating the dimensions of the conveyor system shown in Figure 13 in Chapter III. The length of the conveyor and the process time determine the velocity of the conveyor.

Computer programs have been developed for all of the subsystems which converge to find the operating points of the individual components. The programs estimate the power requirements of the individual subsystems and the total system. The computer programs are given in Appendix H. The programs represented by model A3 and model B2 were used in developing the final design parameters for each process.

5.2 Parametric Study

A parametric study was conducted to determine the optimum combination of velocity, temperature, and the quantity of air to be used to achieve the process requirement in a given amount of time. The optimum combination is that combination which will require the minimum energy for the total system to perform the process in a given time. This study was done for product thicknesses of one to five inches and for the condenser water inlet temperature of 75 F.

The study was conducted for each thickness of the product, by varying the evaporators and the velocity of air on the conveyor for a fixed condensing unit. The parametric influence of the heat transfer coefficient (velocity of air) and the mass ratio of the flow streams on the process is revealed by this study. The study was repeated for different condensing units to obtain different temperatures of air entering the conveyor. A tangential curve known as the optimum characteristic is drawn to all of the sets of operating curves. These curves are shown in Figure 24 for chilling beef of four inches thick.

The same procedure was repeated for product thicknesses of one to five inches; the curves for optimum power input to the total system are shown in Figure 25 for chilling and in Figure 26 for freezing. For a given process time, the power input becomes greater with an increase in thickness of the product. The influence of product thickness on power input becomes less as the process time increases. This is because a lower temperature potential and heat transfer coefficient can achieve the process requirement with a longer processing time and require less power input to the system.

The optimum characteristics were developed for cooling the beef pieces: 100 F to 40 F average temperature in the case of chilling and 45 F to -3 F (or an enthalpy reduction of 120 Btu/Lbm) in the case of freezing.

The optimum air temperature at the conveyor inlet as a function of process time for different thicknesses is given in Figure 27 for chilling and Figure 28 for freezing. For a fixed process time, the air temperature

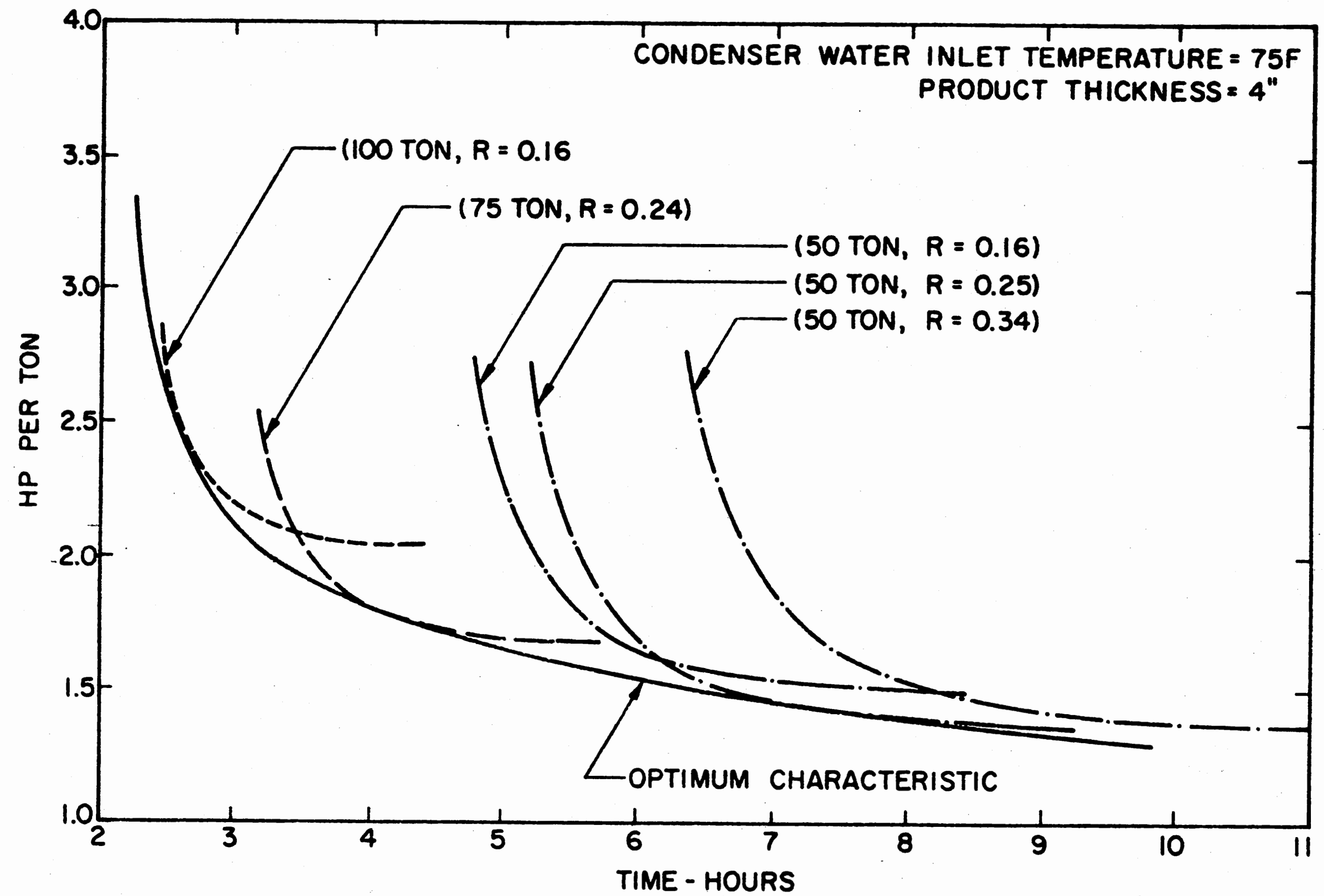


Figure 24. Development of the Optimum Characteristics of the System

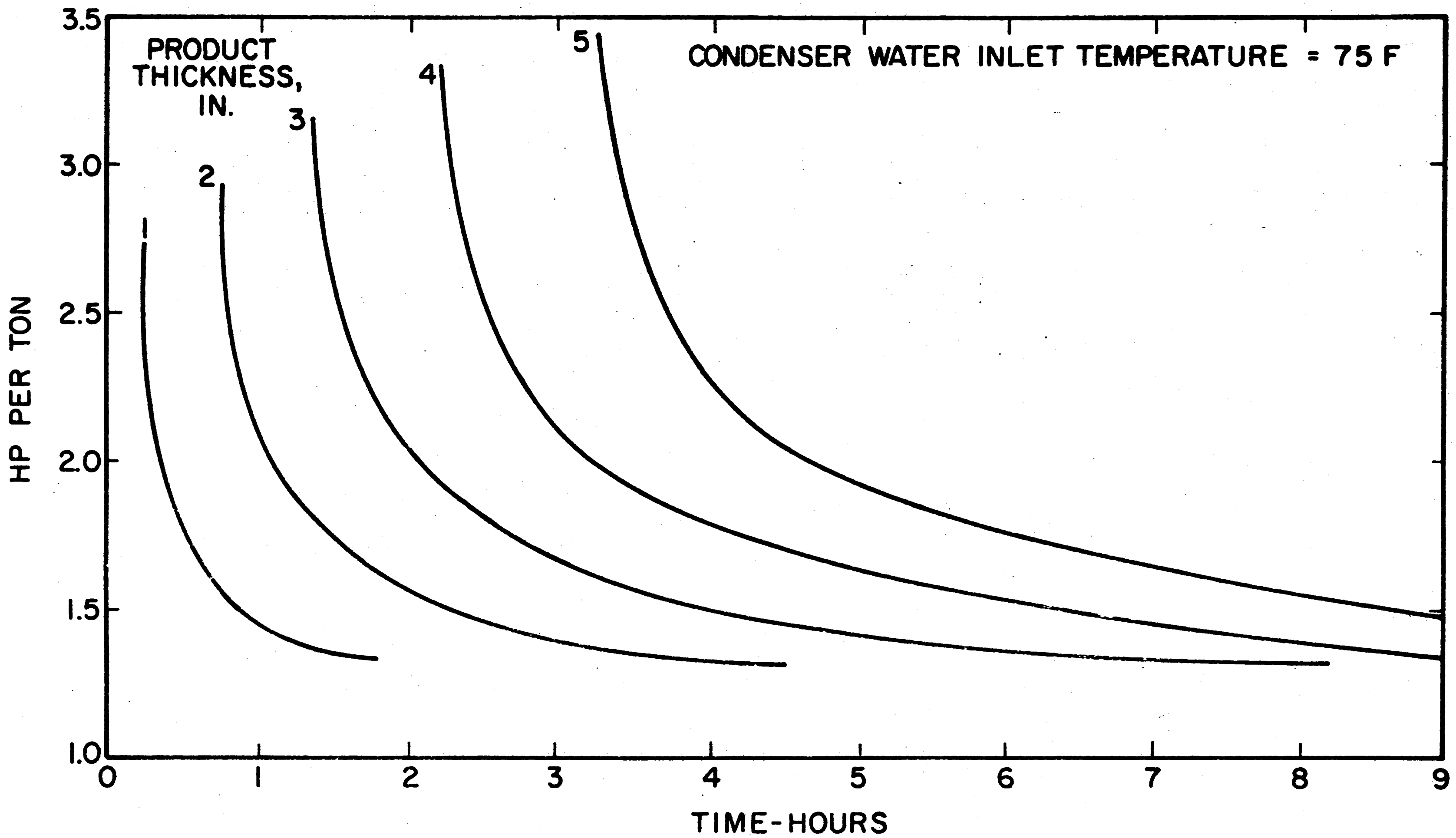


Figure 25. Optimum Power Required by the Conveyorized Chilling System

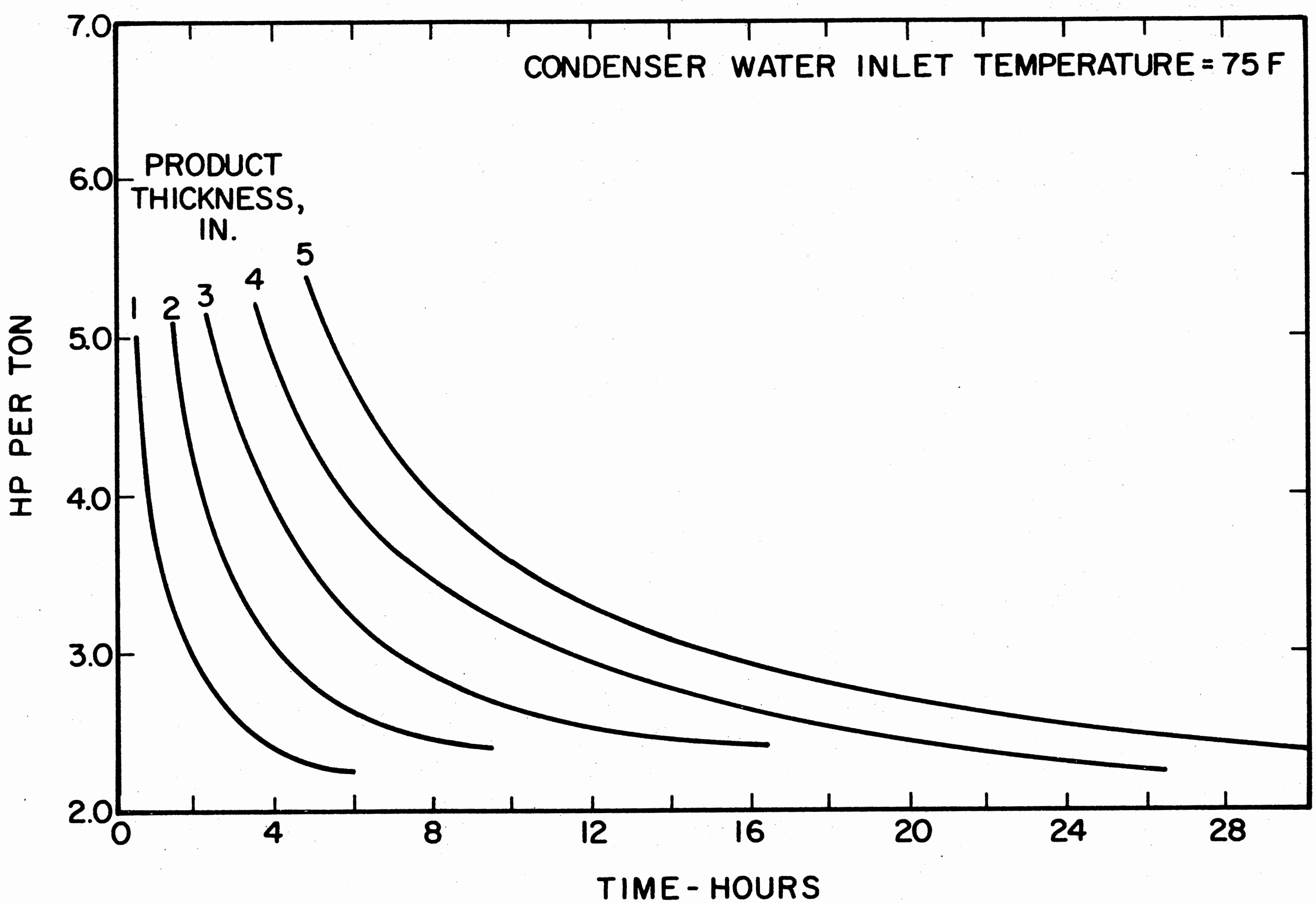


Figure 26. Optimum Power Required by the Conveyorized Freezing System

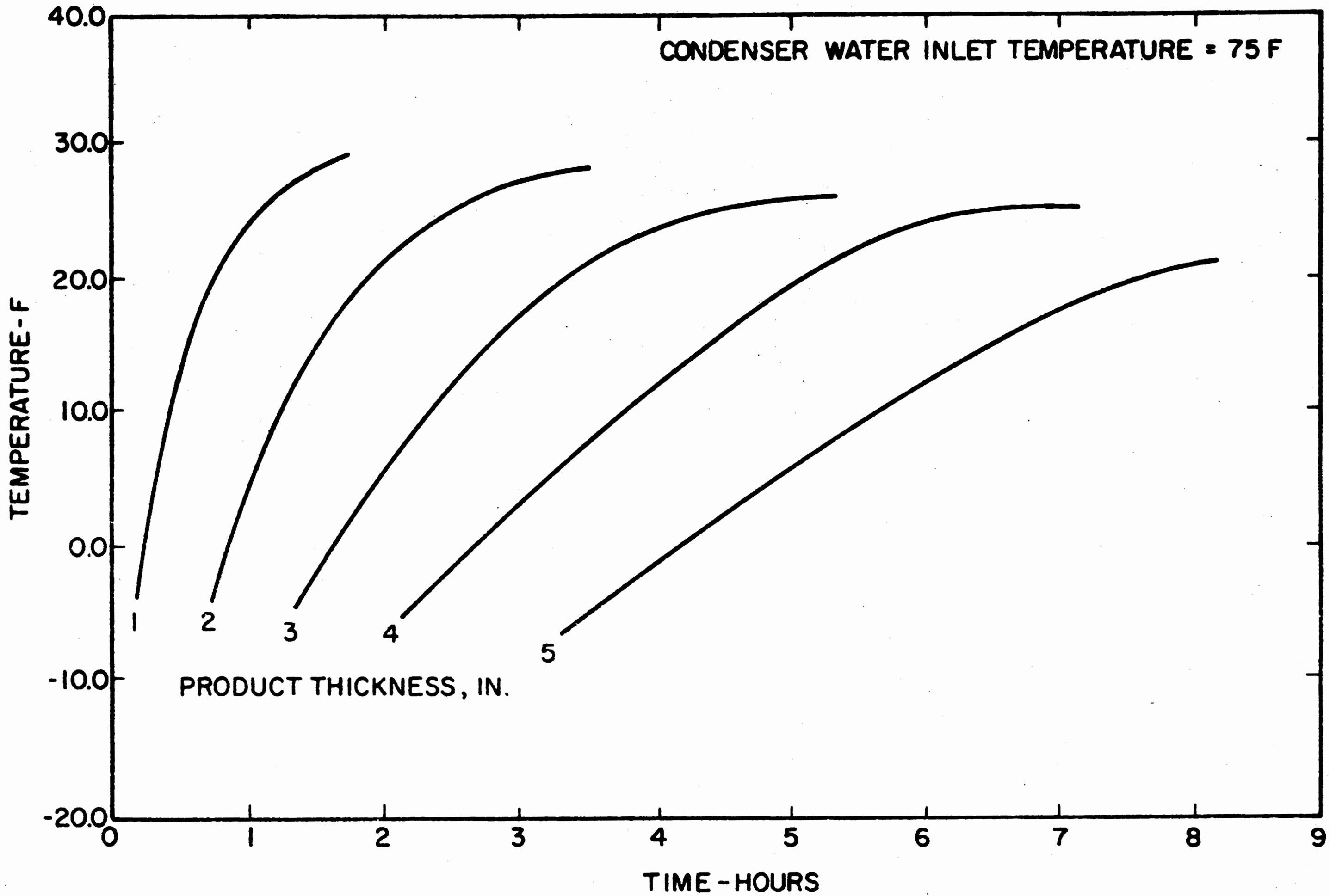


Figure 27. Optimum Temperature of Air Entering the Conveyor in the Conveyorized Chilling System

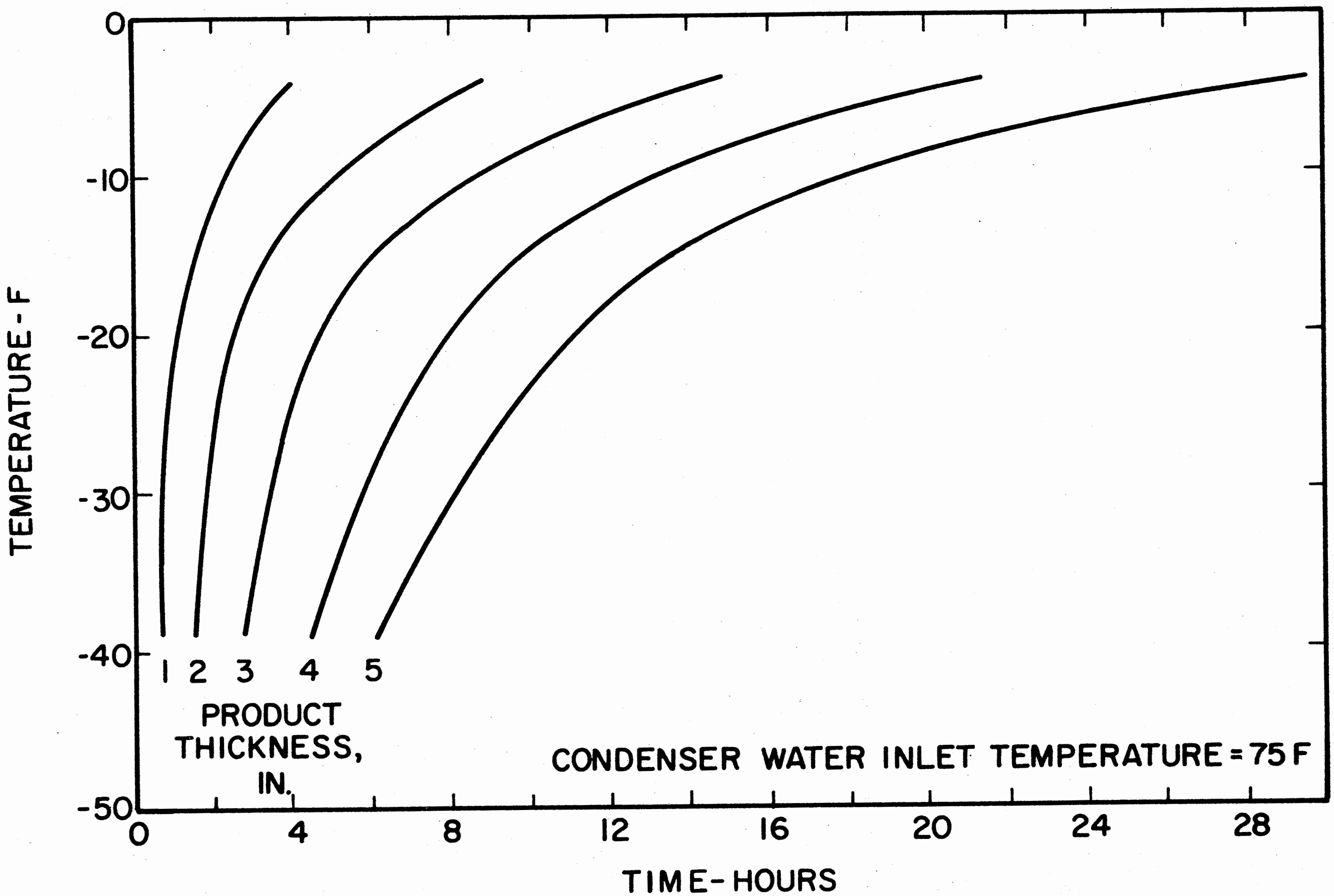


Figure 28. Optimum Temperature of Air Entering the Conveyor in the Conveyorized Freezing System

should be decreased with an increase in product thickness. The influence of product thickness on air temperature decreases as the processing time increases.

The velocity of air to be maintained on the conveyor, as a function of process time for various thicknesses, is given in Figure 29 for chilling and Figure 30 for freezing. For a fixed process time, air velocity should be increased with an increase in product thickness. The influence of product thickness on velocity also decreases as the process time increases.

In the case of chilling, the ratio of the heat capacity rates of the two streams is designated by R. In the case of freezing, no similar parameter can be defined due to the variation of the specific heat of freezing beef with temperature. Therefore the volume flow rate of air per ton of refrigeration is used. This study shows that the optimum quantity of air is nearly constant for both chilling and freezing processes and is about 700 cfm per ton ($R = 0.25$ in the case of chilling) for air velocities over the conveyor less than 35 feet per second. A lower flow rate increases the temperature drop across the coil (between the air and the refrigerant) and an increased flow rate increases the circulating power requirement, thus requiring more power for the total system. It is not advisable to use velocities higher than 30 ft per second due to the prohibitive increase in the fan power requirement and thus the total system power input. Higher velocities will not reduce the process time appreciably.

The energy input to the system is a strong function of condenser water temperature. For a given thickness of product and process time, the power input decreases with a decrease in condenser water inlet

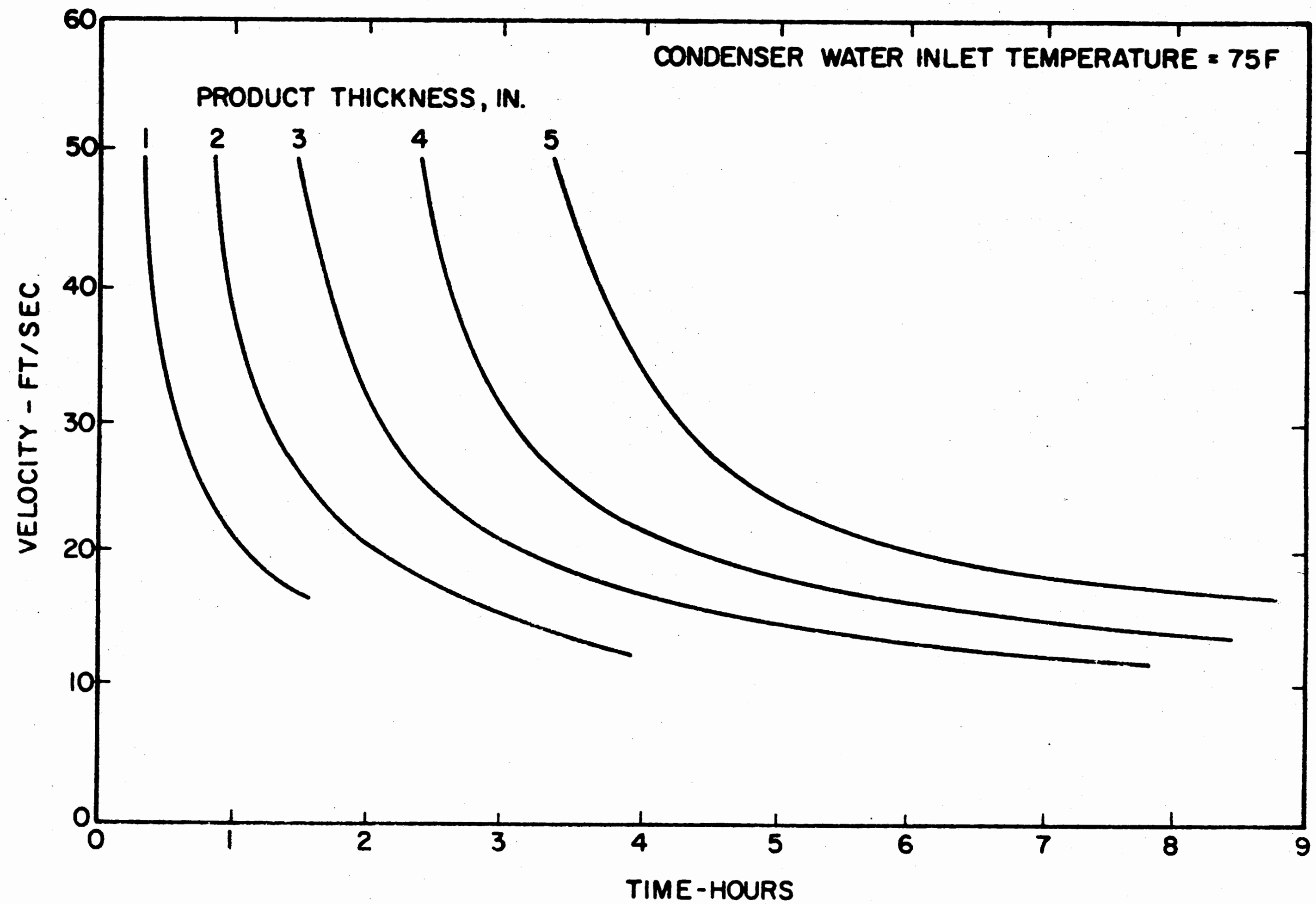


Figure 29. Optimum Velocity of Air on the Conveyor in the Conveyorized Chilling System

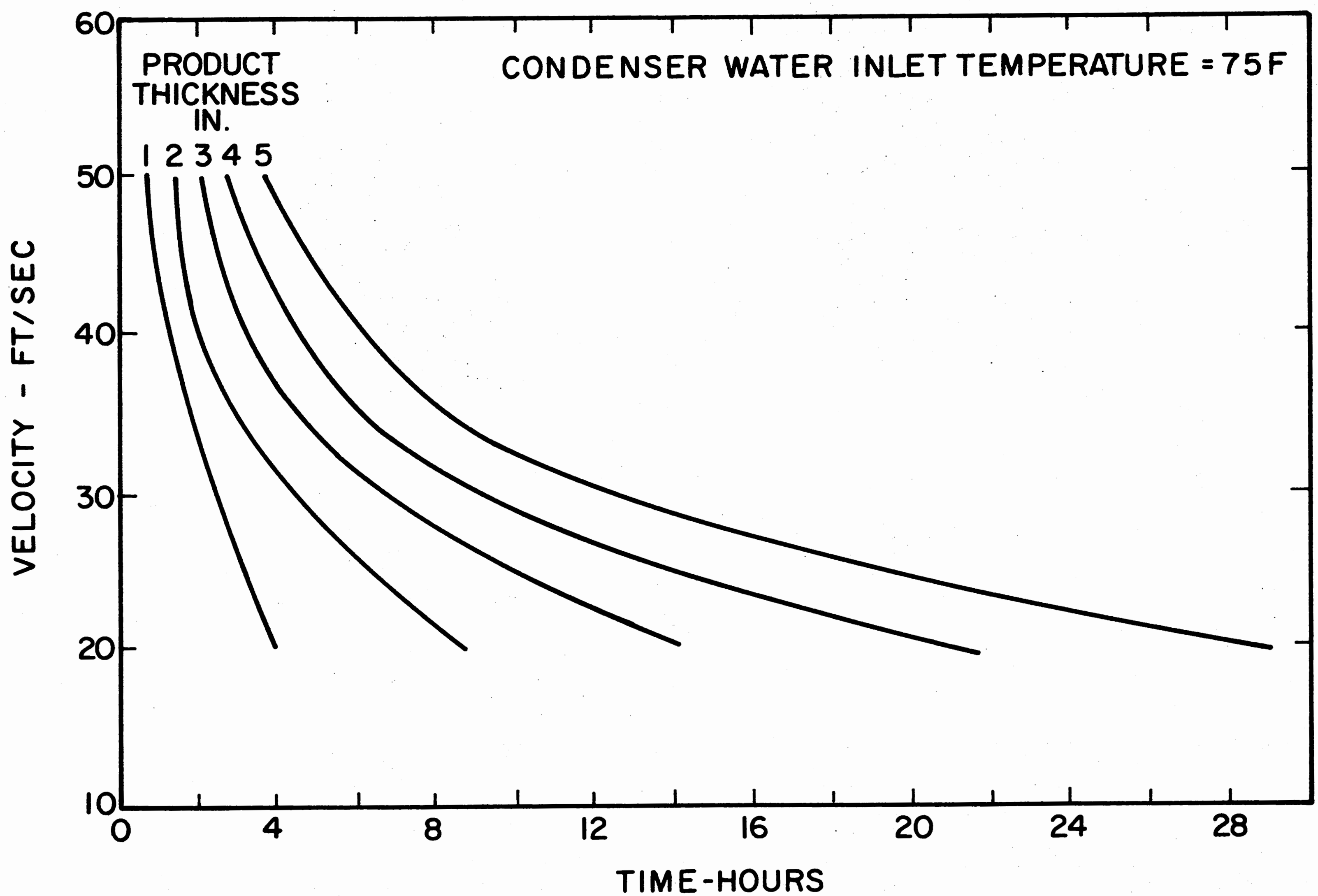


Figure 30. Optimum Velocity of Air on the Conveyor in the Conveyorized Freezing System

temperature. If ground water, which is normally at about 55 F, is available, the design characteristics similar to Figures 25, 27, and 29 are given in [40] for chilling applications. The velocity of air on the conveyor and the temperature of air entering the conveyor are weak functions of the condenser water temperature but are strong functions of the process time.

For a given process, the appropriate set of characteristics to be used is fixed. A decision has to be made on the process time for the given thickness of the beef pieces based on a study of the optimum power demand curve, initial cost of equipment and buildings, and their operating costs. The initial cost of equipment such as the conveyor and building and their operating costs increase with the increase in process time. The operating cost and the initial cost of the refrigeration system decrease with an increase in process time. The process time, therefore, has to be based on individual plant conditions. Sometimes the marketing strategy plays an important role in fixing the process time. Some economic analyses are given in [40]. Once the process time is fixed, the optimum parameters are also fixed; they are given for chilling in Figures 25, 27, and 29 and for freezing in Figures 26, 28, and 30.

The above analysis was developed on the basis of 50 tons of refrigeration load. In the case of the chilling process, it is found that the process requires up to 5 percent more power if the product load is reduced to 25 tons and 7 percent less power if the product load is increased to 200 tons. The recommended values of the parameters for the chilling process are: (a) a velocity of 15 to 20 feet per second, (b) air temperature entering the conveyor 24 to 28 F, and (c) a heat capacity flow rate ratio R of 0.24 to 0.26 (approximately 700 cfm of air per ton). In an optimally designed system, the air circulating power requirements are 10 to 15 percent of the total system power.

In the case of the freezing process, it requires up to 8 percent more power if the product load is reduced to 25 tons and 4 percent less power if the product load is increased to 100 tons. The recommended values of the parameters for the freezing process are: (a) air velocity of 20 to 25 feet per second, (b) air temperature entering the conveyor of -5 to -10 F, and (c) air quantity of approximately 700 cfm per ton of load. In an optimally designed system, the air circulating power requirements are 10 to 15 percent of the total system power.

The influence of capacity on the air velocity on the conveyor, on the temperature of the air entering the conveyor, and on the quantity of air is negligible. The length to width ratio of the conveyor is also not an important parameter and its influence is negligible. If the application requires the use of a friction factor f of 0.02 in Equation (3.38), the fan power requirement and the total power will increase about 3 percent for chilling applications and about 5 percent for freezing applications. The influence of the magnitude of friction factor on velocity, temperature and quantity of air to be circulated for optimum operation is negligible. If the product has to be processed without bagging, lower air temperature and higher air velocity should be used, which results in less moisture loss and shorter process times.

Using the data developed above, comparisons are made in the next chapter of hot and cold processing of beef for both chilling and freezing processes.

CHAPTER VI

COMPARISONS OF HOT AND COLD PROCESSES OF BEEF

In this chapter, cold and hot processing of beef is compared. The main problem in making the comparisons is to reduce the available cold process data and the developed hot process data to the same basis. Therefore, the following guidelines are set for both chilling and freezing processes:

1. The data given in the ASHRAE Handbook and Product Directory, Applications Volume [9] for cold process is used whenever possible.

2. When data is not available for the cold process, the equipment is simulated under normal operating conditions.

3. The data for the hot process is developed using actual refrigeration equipment characteristics as given in Chapter V.

4. All electric motors are assumed to have an efficiency of 90 percent.

5. The comparisons are made for condenser supply water temperature of 75 F average for the day. This temperature is normally achieved by recirculating the water using a cooling tower.

6.1 Chilling Process

The following observations are used for comparing the chilling process of both methods:

1. Twenty-four hours of chilling and 24 hours of holding time are required to bring the average temperature of the carcass to 40 F as given in [9] and as shown in Figure 31.

2. Comparisons are made for the example given in [9].

3. For a given piece, equivalent chilling is achieved in an average of 4 hours or less in the hot process.

4. Additional retention of the product in a holding cooler depends on the marketing techniques and will not be considered for either process.

5. For the hot process, a conservative estimate shows that one third of the cold process chill cooler size is sufficient and no holding cooler is required [25].

6. A conservative estimate of the building load is half the cold process chill cooler load for the hot process method; the estimate is used for comparisons.

Number of carcasses to be processed	=	520 head/day
Carcass weight [9]	=	560 Lbs
Average properties as explained in Appendix B are:		
Specific heat of carcass	=	0.78 Btu/Lbm-F
Specific heat of lean beef	=	0.85 Btu/Hr-Ft-F
Thermal conductivity of lean beef	=	0.28 Btu/Hr-Ft-F
Density of lean beef	=	65 Lbm/Ft^3

6.1.1 Load Calculations for Cold Process Method

The design capacities of the chill cooler and the holding cooler

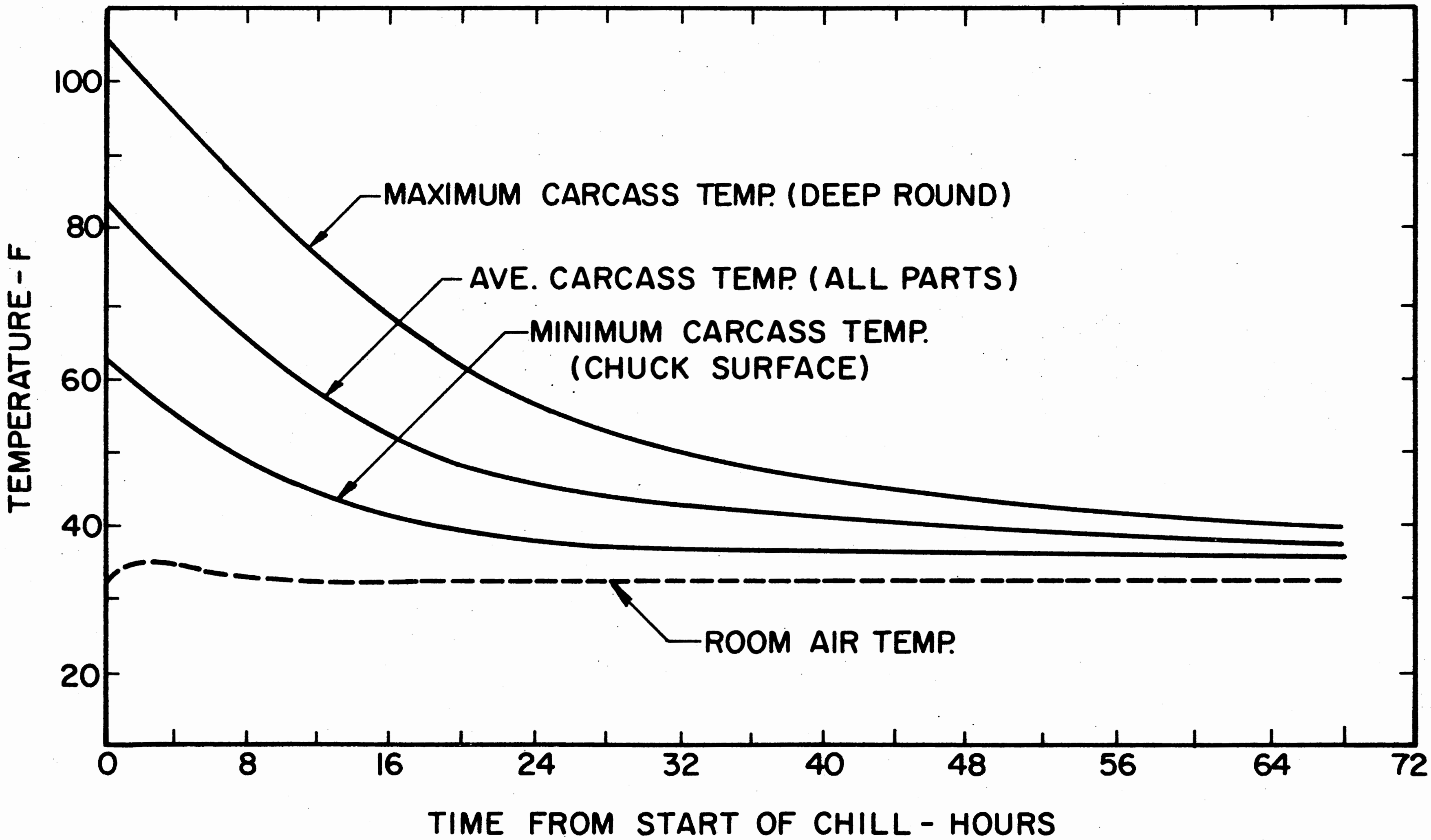


Figure 31. Typical Beef Carcass Chilling-Holding Temperature Curves

are taken from [9] and are summarized as follows:

The design cooling equipment Capacity of the chill cooler	=	1,073,000 Btu/Hr or 89.4 tons
The design cooling equipment Capacity of holding cooler	=	204,500 Btu/Hr or 17.0 tons
Total cooling equipment capacity	=	106.4 tons

The average operating conditions for the condenser water inlet temperature condition of 75 F are obtained by simulating the same equipment used in the hot process method and are as follows:

Refrigerant condensing temperature	=	93 F
Air to coil temperature [9]	=	33 F
Air off the coil temperature [9]	=	29 F
Evaporating temperature	=	21 F
Power to condenser water pump motor	=	2 HP
Input to chill cooler fan motor (45 BHP/0.9) [9]	=	50 HP
Input to holding cooler fan motor (10 BHP/0.9) [9]	=	11.1 HP
Input to compressor motor	=	1.3 HP/ton
Average total product load	=	520*560*0.78*(100-40)
	=	$1.363*10^7$ Btu/day
		or 47.3 tons
Circulating fan heat load	=	(50+11.1)*2545
	=	155,500 Btu/Hr
		or 13 tons

a. Excluding the building load:

Average total input to the system = (1.3*(47.3+13))+63.1

= 142 HP

or 105.5 KW

Energy required for processing 520 carcasses = 105.5*24

= 2530 KW-HR

b. Including the building load:

Transmission, infiltration, personnel and equipment load [9] = 422,500 Btu/Hr

or 35.2 tons

Average total input to the system = (1.3*(47.3+35.2))+63.1

= 170.4 HP

or 127 KW

Energy required for processing 520 carcasses = 127*24

= 3049 KW-HR

c. The Power Demand:

Peak power required = (106.4*1.3+63.1)

= 201.4 HP

or 150 KW

6.1.2 Load Calculations for Hot Process Method

Number of carcasses to be processed is 520 head/day; let this capacity be handled in an 8-hr shift.

Hot boning production (for 62.4% yield) (Appendix B) = 520*560*0.624

= 181,709 Lbm/shift

or 22,714 Lbm/Hr

Chilling capacity required for product to chill from 100 F to 40 F average	=	22,714*0.85*(100-40)
	=	$1.16*10^6$ Btu/Hr
		or 96.5 tons

Building load estimation:

Transmission, infiltration, personnel and equipment load in cold process chill room [9]	=	291,000 Btu/Hr
Circulating fan heat load (45 BHP/0.9) [9]	=	127,250 Btu/Hr
Difference of the above two loads	=	163,750 Btu/Hr
Taking half of the above load as the load for hot process method (due to less required space) and excluding fan load	=	81,875 Btu/Hr
		or 6.8 tons

The hot boned beef chilling process design calculations should be done for each group of beef pieces separately. The grouping is given in Appendix F. The design calculations are given in Table I. All of the design values are taken from the graphs previously explained in Chapter V. The other parameters are calculated using the design values and the data from the example.

Total conveyor area required to chill as given in Table I	=	6,706 Ft^2

TABLE I

HOT BONED BEEF CHILLING PROCESS DESIGN CALCULATIONS

Parameters	Group		
	A	B	C
Percent by weight (App. G)	29.3	30.5	40.2
Actual Weight (Lbm/Hr)	6655	6928	9131
Cooling load (tons)	28.3	29.4	38.8
Chilling time (Hrs)	4	3	2.5
Loading factor (Lbm/Ft^2)	15	10	8
Conveyor area (Ft^2)	1775	2078	2835
Process power (HP/ton)	1.5	1.4	1.3
Vel. of air on conv. (Ft/sec)	18	16	15
Temp. of air on the conv. (F)	24	27	29
Temp. of air off the conv. (F)	38.5	41.5	43.5
Quantity of air req. (cfm/ton)	700	700	700
Total cfm required	19,810	20,580	27,160
Conveyor fan motor input (HP)	2	2	2
Evp. unit fan motor input (HP)	3.5	3.5	5.0

The design cooling equipment capacity required to handle the product load	=	96.5 tons
Estimated design building load	=	6.8 tons
Fan heat load ((6+12) BHP/0.9)	=	4.2 tons
Total cooling equipment capacity	=	107.5 tons

For an 8-hr shift of the hot boning process, the chilling process takes 12 hours of operation. This is because the group A product loaded at the end of the 8th hour will require 4 more hours to complete the chilling process. Group B will take 11 hours and group C will take 10.5 hours. It is assumed that the total load is averaged over 12 hours of chilling for energy calculations.

Average total product load	=	181,709*0.85*(100-40)/12
	=	772,263 Btu/Hr
		or 64.4 tons
The average input to the refrigeration system	=	(1.5*0.293+1.4*0.305 +1.3*0.402)
	=	1.4 HP/ton

a. Excluding the building load:

Average total input to the system	=	1.4*(64.4+4.2)
	=	96 HP
		or 71.6 KW
Energy required for processing 520 carcasses	=	71.6*12
	=	860 KW-Hr
Saving in energy	=	(2530-860)/2530 = 66%

b. Including the building load:

Estimated building load	=	6.8 tons
Power required for building load	=	9.5 HP
Average input to the system when the chilling process is on	=	105.5 HP
		or 78.7 KW
Average input to the system when the chilling process is off	=	9.5 HP
		or 7.1 KW
Energy required for processing 520 carcasses	=	78.7*12+7.1*12
	=	1030 KW-Hr
Saving in energy	=	66%

c. The Power Demand:

Peak power required	=	150.5 HP
	=	112 KW
Reduction in peak power	=	(150-112)/150
	=	25%

The chilling of beef by the hot process method requires 66 percent less energy. Thirty-two percent is due to reduction in mass (37.6 percent) and the remainder is due to the improved design of the system. The capacity of the required refrigeration equipment is approximately the same for both the processes. There is a 25 percent reduction in peak power demand due to reduction in fan power requirements.

6.2 Freezing Process

For comparing the freezing process of the two methods, the following observations are used.

1. Due to the unavailability of published data for the usual cold process freezing of beef, data supplied by the Engineering Department of Wilson Foods Corporation, in Oklahoma City [26] is used for this purpose.

2. Forty-eight hours of freezing time is required in the cold process method [26, 27]; therefore, two freezer rooms are required for continuous operation.

3. A conservative estimate shows that one freezer room of the same size as above is sufficient in the hot process method for equivalent production.

Product Weight = 40 pallets*(16 boxes/pallet)*(60 lbs/box)

= 38,400 Lbm/day

Enthalpy reduction (45 F ave. to -3 F ave.):

Lean beef [34]	=	120 Btu/Lbm
Bone [34]	=	53 Btu/Lbm
Fat [34]	=	49 Btu/Lbm
Tissue (Appendix B)	=	34 Btu/Lbm

6.2.1 Load Calculations for Cold Process Method

Design of freezer room equipment as provided by [26]:

Product load for 48 hours	=	38,400*120 Btu
Average product load	=	96,000 Btu/Hr
		or 8 tons

Installation capacity required

for product load = 1.5*8

or 12 tons

Room heat gain (27'*22'*12')

at 200 Ft^2/ton = (27*22)/200

or 3 tons

The average operating conditions for a rotary screw compressor unit which is used in the hot process design characteristics development are as follows:

Condenser water inlet temperature	=	75 F
Refrigerant condensing temperature	=	90 F
Air to coil temperature	=	-25 F
Air off the coil temperature	=	-35 F
Refrigerant evaporating temperature	=	-40 F
Power to condenser water pump motor	=	0.5 HP
Input to circulating fan motor		
(6 BHP/0.9)	=	6.7 HP
Input to compressor motor		
(3.6 BHP/0.9)	=	4 HP/ton
Enthalpy reduction per unit mass		
in cold process	=	(120*0.624)+(53*0.157)
		+(49*0.199)+(34*0.02)
	=	93.6 Btu/Lbm
Average product load	=	(38,400*93.6)/48
	=	74,900 Btu/Hr
		or 6.2 tons
Circulating fan heat load	=	6.7*2545

	=	16,967 Btu/Hr
		or 1.4 tons
Total freezing equipment capacity		
for both freezer rooms	=	(12+3+1.4)*2
		or 32.8 tons

a. Excluding the building load:

Average total input to the system	=	(4.0*(6.2+1.4))+7.2
	=	37.6 HP
		or 28 KW
Energy required for processing		
38,400 Lbm of beef	=	28*48
	=	1344 Kw-Hr

b. Including the building load:

Room heat gain	=	3.0 tons
Average total input to the system	=	4.0*(6.2+1.4+3.0)+7.2
	=	49.6 HP
		or 37 KW
Energy required for processing		
38,400 Lbm of beef	=	37*48
	=	1776 Kw-Hr

c. The Power Demand:

Peak power required	=	(4.0*32.8)+7.2
	=	138.4 HP
		or 103 KW

6.2.2 Load Calculations for Hot Process

Method

This method uses the same process capacity as the cold process.

Product weight	=	38,400 Lbm/day
Hot boning production		
(for 62.4% yield)	=	38,400*0.624
	=	23, 962 Lbm/shift
		or 2995 Lbm/Hr

It is assumed to have 16 hours of freezing time, so that the last hours of production will also be processed by the beginning of the following day.

$$\text{Average freezing capacity required} = \frac{(23{,}962*120)}{(16*12{,}000)}$$

$$= 15 \text{ tons}$$

The thin cuts will cool at a much faster rate than the average load assumed above. Therefore to design the equipment capacity required, a load factor of 1.5 is recommended similar to the one used in the cold process method.

Freezing capacity required for the		
product load	=	1.5*15
		or 22.5 tons

For building load estimation, it is conservatively estimated that one freezer room is sufficient for this case.

Building load	=	3 tons

The hot boned beef freezing process design calculations should be done for each group of beef pieces separately. The grouping is the same as given in Appendix G. The design calculations are given in Table II.

All the design values are taken from the graphs previously explained in Chapter V. The other parameters are calculated using the design values and the data from the example.

Total conveyor area required for freezing as given in Table II	=	2403 Ft^2
Design freezing equipment capacity required for product load	=	22.5 tons
Estimated design building load	=	3.0 tons
Circulating fan heat load (2 BHP/0.9)	=	0.5 tons
Evaporator fan heat load (2 BHP/0.9)	=	0.5 tons
Total freezing equipment capacity	=	26.5 tons

The freezing process takes, on an average, 16 hours of operating time and 8 hours of storage.

Average product load	=	15.0 tons
Average input to refrigeration system	=	(2.7*0.293+2.6*0.305 +2.5*0.402)
	=	2.6 HP/ton

a. Excluding the building load:

Average total input to the system	=	2.6*(15.0+1.0)
	=	41.6 HP
		or 31 KW
Energy required per processing 38,400 Lbm	=	31*16
	=	496 Kw-Hr

TABLE II

HOT BONED BEEF FREEZING PROCESS DESIGN CALCULATIONS

Parameters	Group		
	A	B	C
Percent by weight (App. G)	29.3	30.5	40.2
Actual weight (Lbm/Hr)	878	913	1204
Freezing load (tons)	4.4	4.6	6.0
Freezing time (Hrs)	16	12	8
Loading factor (Lbm/Ft^2)	15	10	8
Conveyor area (Ft^2)	468	731	1204
Process power (HP/ton)	2.7	2.6	2.5
Vel. of air on conv. (Ft/Sec)	24	23	22
Temp. of air on to conv. (F)	-8	-6	-5
Quantity of air req. (cfm/ton)	700	700	700
Total cfm required	3080	3220	4200
Conveyor fan motor input (HP)	0.5	0.5	1.0
Evp. unit fan motor input (HP)	0.5	0.5	1.0

Saving in energy = (1344-496)/1344

= 63%

b. Including the building load:

Estimated building load = 3.0 tons

Average input to the system when

the freezing process is on = 2.6*(15.0+1.0+3.0)

= 49.4 HP

or 36.8 KW

Average input to the system when

the freezing process is off = 2.6*(1.0+3.0)

= 10.4 HP

or 7.8 KW

Energy required for processing

38,400 Lbm = 36.8*16+7.8*8

= 651 KW-Hr

Saving in energy = (1776-651)/1776

= 63%

c. The Power Demand:

Peak power required = 2.6*26.5

= 69 HP

or 5k KW

Reduction in peak power = (103-51)/103

= 50%

The freezing of beef by the hot process method requires 63 percent less energy. Of the 63 percent, 20 percent is due to the reduction in mass (37.6 percent) and the rest is due to the improved design possible for this method. The capacity of the refrigeration equipment required is also greatly reduced from 33 tons at -40 F evaporating temperature to 26.5 tons at -15 F evaporating temperature for a condensing temperature of 90 F in both cases. The equipment producing 33 tons at -40 F evaporating temperature can produce more than double the capacity at -15 F evaporating temperature. There is a 50 percent reduction in the peak demand due to the reduction in the equipment capacity and the fan power requirement.

CHAPTER VII

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

A rational design procedure for conveyorized heat transfer systems has been developed. These systems are widely used in the food processing industry for freezing various types of products. Herein a theory has been developed that can be applied to all conveyorized heat transfer processes. This theory is applied to develop parameters to be used in the design of conveyorized chilling and freezing systems for hot boned beef. Energy comparisons for processing beef by both cold and hot processes are made. Finally, recommendations for future work are given.

7.1 Summary and Conclusions

The outcome of this study can be summarized as follows:

1. Various heat transfer models for conveyorized heat transfer process applications are developed and given in Chapter III. The limitations and applications of the above models are also explained in that chapter. Computer programs were developed for all of the heat transfer models and are given in Appendix H.

2. Closed-form heat transfer solutions were obtained for calculating the energy exchange between the one-dimensional solid with constant properties and a variable temperature fluid by a step function approximation (Equations (3.16) and (3.17)).

3. The graphical procedure given in Reference [36] for finding the

operating points of practical refrigeration systems is reformulated to be used in computer models and is outlined in Chapter III. The characteristics of the various components of the refrigeration systems used in the chilling and freezing processes of beef were developed and are given in Appendix E. They are used to determine the operating conditions of the total refrigeration system and the individual components in the simulation of the chilling and freezing processes.

4. All the heat transfer models explained in Chapter III are tested for their applicability to conveyorized hot boned beef processes and the results are presented in Chapter IV.

5. A procedure is developed for simulating a conveyorized heat transfer system in general. It is used in the development of characteristics for the design of chilling and freezing equipment for hot boned beef. The procedure can be applied in designing conveyorized freezers with various fluids which are widely used in the food industry.

6. By hot boning the beef, the nonedible fat and bone are removed before the beef is cooled. This mass reduction of 37.6 percent reduces the chilling process energy by 32 percent and the freezing process energy by 20 percent.

7. The hot boned beef cuts are relatively thin (maximum thickness up to four inches) compared to the beef carcass sides (maximum thickness up to ten inches) and hence the process time is reduced. The thinness of the cuts and the reduced process time reduces the size of the refrigerated area, and thus the building load and the equipment load. This is shown in Chapter VI.

8. The size of the cooler and freezer rooms is reduced because of a reduced product weight and process time and because of the compactness

with which the cuts can be loaded on the conveyor and stored. The estimates are given in Reference [40] and in Chapter VI.

9. The thin hot boned cuts can be processed with higher temperature air than is possible in cold processing. This increases the coefficient of performance of the refrigeration system and thus reduces the energy required for processing. Some of the values are presented in Chapter VI.

10. The thin cuts made it possible to use a conveyor system for chilling and freezing which was not possible with carcass sides. The conveyor arrangement reduces the amount of air necessary to obtain the required velocity over the product surface and this in turn reduces the fan power requirement up to 70 percent in both the chilling and freezing processes. This also reduces the equipment load.

11. There are many advantages of the hot processing method as outlined above. The overall energy saving made by adopting the hot boning process and the conveyorized method for processing is estimated to be about 66 percent in chilling and 63 percent in freezing.

12. The hot boning of beef and conveyorized processing reduces the peak power demand considerably. It is estimated that the reduction is up to 25 percent for the chilling application and 50 percent for the freezing application. The equipment operates at a more uniform load than possible in the cold processing. This increases the system efficiency and the electrical load factor for the installation.

13. For a given production rate, there will be considerable reduction in the freezing equipment capacity needed for freezing hot boned beef on the conveyor compared to the present freezer room method.

14. The data given in graphical form in Chapter V can be used in

designing equipment for chilling or freezing beef on a conveyor. The sensitivity of the various parameters used in the design is explained in that chapter.

15. It can be qualitatively stated that processing of unbagged beef needs colder air (less than 28 F) than processing of bagged beef. This results in more economical processing due to reduction in moisture loss from the beef. The moisture loss will be minimal if the crust of the beef pieces can be frozen quickly and maintained in the frozen state during the process. Moisture loss is normally an important economic concern to the processer. The same design characteristics as those developed in Chapter V can be used to some extent for designing the equipment for the unbagged product; using very cold air results in less moisture loss, a short process time, and higher process energy.

7.2 Recommendations

Although the results given in Chapter V can be applied to some extent, further work is required in developing the design parameters for systems which use air-cooled condensers.

Some of the beef processing installations use two-stage refrigeration systems for freezing. Additional work needs to be done to develop the design parameters for these systems with the appropriate refrigeration system models.

There will be considerable moisture loss in processing the unbagged product; this is an important concern to the beef processer. Therefore, a mass transfer model is required for calculating the moisture loss. The mass transfer model requires the development of diffusion coefficient data for beef as a function of temperature. It is recommended that a heat and mass transfer model be developed to find the optimum combination of velocity and temperature of air to be used in designing a system for an economic balance between moisture loss and process energy requirement.

BIBLIOGRAPHY

[1] Carslaw, H. S., and J. C. Jaeger. Conduction of Heat in Solids. 2nd ed. Fair Lawn, N.J.: Oxford University Press, 1959, pp. 119-120.

[2] Luikov, A. V. Analytical Heat Diffusion Theory. New York: Academic Press, Inc., 1968, pp. 214-240.

[3] Ozisik, M. N. Boundary Value Problems of Heat Conduction. Scranton, Penn.: International Text Book Company, 1968, pp. 54-59, 388-405.

[4] Handbook of Heat Transfer. New York: McGraw-Hill, Inc., 1973, pp. 3.26-3.52, 7.5-7.6.

[5] Luikov, A. V. "Methods of Solving the Nonlinear Equations of Unsteady State Heat Conduction." J. of Heat Transfer--Soviet Research, Vol. 3, No. 3 (May-June, 1971), pp. 1-51.

[6] Goodman, T. R. "Application of Integral Methods to Transient Nonlinear Heat Transfer." Advances in Heat Transfer, Vol. 1 (1964), pp. 51-122.

[7] Bailey, C. "Factors Affecting Rate of Cooling and Evaporation." Meat Chilling--Why and How. Symposium at MRI, Langford, April, 1972, pp. 13.1-13.11.

[8] Levy, F. L. "Energy, Time-Temperature and Weight Loss During Meat Cooling." Meat Chilling--Why and How. Symposium at MRI, Langford, April, 1972, pp. 14.1-14.15.

[9] ASHRAE Guide and Data Book. Applications, 1971, pp. 239-336.

[10] Heldman, D. R. "Computer Simulation of Food Freezing Processes." Proc., 4th Intl. Congress of Food Sci. and Technol., Vol. 4 (1974), pp. 397-406.

[11] Comini, G., and C. Bonacina. "Application of Computer Codes to Phase Change Problems in Food Engineering." Intl. Inst. of Refrigeration, Bressanone, 1973-1974, pp. 15-27.

[12] Klein, S., E. Slevicek, and M. Kminek. "Rapid Cooling of Meat I." Vysoka Sk Chem.-Technol. Sb Potravinarska Technol., Vol. 35 (1972), pp. 103-115.

[13] Fikiin, A. "Determination De la Duree De Refrigeration Des Products Alimentaires (Determining the Cooling Times of Food Stuffs)." 12th Congress Intl. du Froid (Madrid), Paris, 1967, pp. 705-716.

[14] Bonacina, C., and G. Comini. "On a Numerical Method for the Solution of the Unsteady State Heat Conduction Equation With Temperature Dependent Parameters." Proc., 13th Intl. Congress of Refrigeration, Washington, 1971 (published 1973), pp. 329-336.

[15] Kreith, F. Principles of Heat Transfer. 3rd ed. New York: Intext Educational Publishers, 1973, pp. 165-168.

[16] Cutting, C. L. "Current Issues in Meat Chilling." Meat Chilling --Why and How. Symposium at MRI, Langford, April, 1972, pp. 0.1-0.4.

[17] Cutting, C. L. "Rapid Chilling Procedures." Meat Chilling--Why and How. Symposium at MRI, Langford, April, 1972, pp. 18.1-18.3.

[18] Sulzbacher, W. L. "Meat Chilling in the United States." Meat Chilling--Why and How. Symposium at MRI, Langford, April, 1972, pp. 31.1-31.4.

[19] Watt, D. B., and H. K. Herring. "Rapid Chilling of Beef Carcasses Utilizing Ammonia and Cryogenic Systems: Effect on Shrink and Tenderness." J. Animal Sci., Vol. 38, No. 5 (1974), pp. 928-934.

[20] Bailey, C. "An MRI View of Commercial Refrigeration Design." Meat Chilling--Why and How. Symposium at MRI, Langford, April, 1972, pp. 35.1-35.4.

[21] Hudgson, T. "Air Velocity Distribution in Carcass Chill Rooms." Proc., 13th Intl. Congress of Refrigeration, Washington, 1971 (published 1973), pp. 25-36.

[22] Lovett, D. A., L. S. Herbert, and R. D. Radford. "Chilling of Meat--Experimental Investigation of Weight Loss." Towards an Ideal Refrigerated Food Chain. International Institute of Refrigeration, Annexe 1976-1, pp. 307-314.

[23] Radford, R. D. "Water Transport in Meat." Towards an Ideal Refrigerated Food Chain. International Institute of Refrigeration, Annexe 1976-1, pp. 315-322.

[24] Radford, R. D., L. S. Herbert, and D. A. Lovett. "Chilling of Meat: A Mathematical Model for Heat and Mass Transfer." Towards an Ideal Refrigerated Food Chain. International Institute of Refrigeration, Annexe 1976-1, pp. 323-330.

[25] Project Advisory Committee. Energy Conservation in Meat Processing Industry. Final report, Department of Energy Contract EY-76-S-05-5097, 1976.

[26] McGinness, M. F. Personal communication. Wilson Food Corporation, Engineering Division, Oklahoma City, Oklahoma, 1978-1979.

[27] Earle, R. L. "The Freezing Time of Boneless Beef in Air Blast Freezers." 2nd Intl. Congress of Food Sci. and Technol., Vol. 4 (1966), pp. 79-88.

[28] Fleming, A. K. "The Numerical Calculation of the Freezing Process." Proc., 13th Intl. Congress of Refrigeration, Washington, 1971 (published 1973), pp. 303-311.

[29] Fleming, A. K. "Application of a Computer Program to the Freezing Process." Proc., 13th Intl. Congress of Refrigeration, Washington, 1971 (published 1973), pp. 403-410.

[30] Goodrich, L. E. "Efficient Numerical Technique for One-Dimensional Thermal Problems With Phase Change." Intl. J. Heat and Mass Transfer, Vol. 21 (1978), pp. 615-621.

[31] Cullwick, T. D. C., and R. L. Earle. "Prediction of Freezing Time of Meat in Plate Freezers." Proc., 13th Intl. Congress of Refrigeration, Washington, 1971 (published 1973), pp. 397-401.

[32] ASHRAE Handbook. 1977 Fundamentals, pp. 28.1-29.20.

[33] Mellor, J. D., and A. H. Seppings. "Thermophysical Data for Designing a Refrigerated Food Chain." Towards an Ideal Refrigerated Food Chain. International Institute of Refrigeration, Annexe 1976-1, pp. 349-359.

[34] Morley, M. J. "Thermal Properties of Meat: Tabulated Data." MRI Special Report No. 1. MRI, Langford, 1972, pp. 26-47.

[35] Riedel, L. "Calorimetric Investigations of the Meat Freezing Process." Kaltetechnik, Vol. 9, No. 2 (1957), pp. 38-40.

[36] Stoecker, W. F. Refrigeration and Air Conditioning. New York: McGraw-Hill Book Company, Inc., 1958, pp. 158-170.

[37] ASHRAE Guide and Data Book. 1972 equipment volume, pp. 199-202.

[38] White, F. M. Viscous Fluid Flow. New York: McGraw-Hill Book Company, 1974, pp. 123-129.

[39] McQuiston, F. C., and J. D. Parker. Heating, Ventilating, and Air Conditioning Analysis and Design. New York: John Wiley and Sons, 1974, pp. 383-410.

[40] Energy Conservation in Meat Processing. Final report, DOE Contract EY-76-S-05-5097. U.S. Department of Energy, June, 1979, pp. 196-251, 354-365.

[41] McAdams, W. H. Heat Transmission. 3rd ed. New York: McGraw-Hill Book Company, Inc., 1954, p. 249.

[42] Loginov, L. I. "Application of Numerical Methods for Cooling Process Calculation." 12th Congress International du Froid (Madrid), Paris, 1967, pp. 717-729.

[43] Kays, W. M. Convective Heat and Mass Transfer. New York: McGraw-Hill Book Company, Inc., 1966, p. 239.

APPENDIX A

DERIVATION OF THE SOLUTIONS TO THE HEAT TRANSFER EQUATIONS

A plane wall, whose thickness is small compared to the other two dimensions, is known as the one-dimensional solid in the heat transfer problems. A solid of the above nature exposed to convective heat transfer on both surfaces, can be treated as a solid of half thickness L with an insulated surface on one side and convective heat transfer from the other surface as shown in Figure 32.

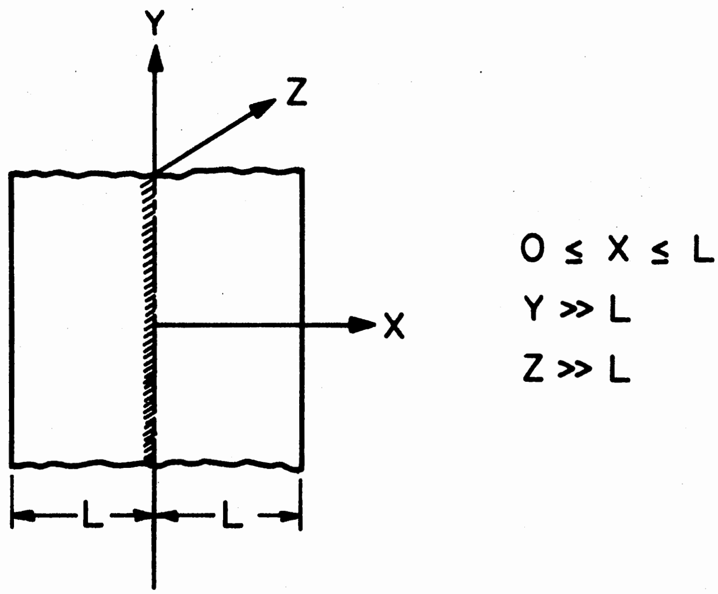


Figure 32. Nomenclature for Transient Heat Flow in a One-Dimensional Solid

The problem can be expressed mathematically as follows:

The governing differential equation is:

$$\frac{\partial^2 t(x,\theta)}{\partial x^2} = \frac{1}{\alpha}\frac{\partial t(x,\theta)}{\partial \theta} \tag{A.1}$$

The boundary conditions are:

$$1) \qquad \left.\frac{\partial t(x,\theta)}{\partial x}\right|_{x=0} = 0 \tag{A.2}$$

$$2) \quad -k \left.\frac{\partial t(x,\theta)}{\partial x}\right|_{x=L} = h[t(L,\theta) - t_\infty] \tag{A.3}$$

The initial condition is:

$$t_i = f(x) \tag{A.4}$$

Let

$$T = t - t_\infty$$

The above equations reduce to:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial T}{\partial \theta} \tag{A.1a}$$

$$1) \qquad \left.\frac{\partial T}{\partial x}\right|_{x=0} = 0 \tag{A.2a}$$

$$2) \quad \left.\frac{\partial T}{\partial x}\right|_{x=L} + \frac{h}{K} T = 0 \tag{A.3a}$$

$$t_i = T_i + t_\infty = f(x)$$

or

$$T_i = t_i - t_\infty = f(x) - t_\infty = F(x) \tag{A.4a}$$

Separating the variables:

$$T(x,\theta) = X(x)\,\Theta(\theta)$$

$$\frac{1}{X}\frac{\partial^2 X}{\partial x^2} = \frac{1}{\alpha}\frac{1}{\Theta}\frac{\partial \Theta}{\partial \theta} = \begin{cases} +\beta^2 \\ 0 \\ -\beta^2 \end{cases} \qquad \beta > 0$$

For $+\beta^2$:

$$\dot{\Theta} - \alpha\beta^2\Theta = 0$$

$$\Theta = e^{\alpha\beta^2\theta}$$

For this solution, the temperature of the slab grows exponentially with time, which is physically impossible. For 0:

$$\frac{X''}{X} = 0$$

$$X = c_1 x + c_2$$

From the boundary conditions, both c_1 and c_2 are zero. For $-\beta^2$:

$$X'' + X\beta^2 = 0$$

$$X(x) = c_1 \cos \beta x + c_2 \sin \beta x \qquad \text{(A.6)}$$

Using the boundary conditions:

$$c_2 = 0$$

$$\beta L \tan \beta L = \frac{hL}{K} = Bi \qquad \text{(A.7)}$$

$$X(x) = c_1 \cos \beta x \qquad \text{(A.6a)}$$

$$\Theta' + \alpha\beta^2\Theta = 0$$

$$\Theta(\theta) = e^{-\alpha\beta^2\theta} \qquad \text{(A.8)}$$

The solution is:

$$T(x,\theta) = X(x)\,\Theta(\theta)$$

$$= \sum_{n=1}^{\infty} c_n \cos \beta_n x \; e^{-\alpha\beta_n^2\theta} \tag{A.9}$$

$$T(x,0) = T_i = t_i - t_\infty = f(x) - t_\infty = F(x) \tag{A.10}$$

$$= \sum_{n=1}^{\infty} c_n \cos \beta_n x \tag{A.10a}$$

$$\int_{x'=0}^{L} F(x') \cos (\beta_m x') \, dx' = \sum_{n=1}^{\infty} c_n \int_{x'=0}^{L} \cos (\beta_n x') \cos (\beta_m x') \, dx' \tag{A.10b}$$

$$= 0 \qquad \text{for } n \neq m$$

For n = m:

$$\int_{x'=0}^{L} F(x') \cos \beta_m x' \, dx' = c_m \int_{x'=0}^{L} \cos^2 (\beta_m x') \, dx' \tag{A.10c}$$

$$\therefore \quad c_m = \frac{2 \int_{x'=0}^{L} F(x') \cos (\beta_m x') \, dx'}{\left[L + \dfrac{\sin 2\beta_m L}{2\beta_m} \right]} \tag{A.11}$$

$$T(x,\theta) = \sum_{m=1}^{\infty} \frac{e^{-\alpha\beta_m^2\theta} \cos \beta_m x \;\; 2 \int_{x'=0}^{L} F(x') \cos (\beta_m x') \, dx'}{\left[L + \dfrac{\sin \beta_m L \cos \beta_m L}{\beta_m} \right]} \tag{A.12}$$

Using

$$\frac{\beta_m^2 L^2}{\cos^2 \beta_m L} = Bi^2 + \beta_m^2 L^2$$

$$T(x,\theta) = 2 \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\theta} \frac{\beta_m^2 L^2}{(Bi^2 + \beta_m^2 L + Bi)} \frac{\cos \beta_m x}{\cos^2 \beta_m L} \frac{1}{L} \int_{x'=0}^{L} F(x') \cos (\beta_m x') \, dx'$$

(A.13)

$$t(x,\Theta) = t_\infty + 2 \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\Theta} \frac{\beta_m^2 L^2}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos\beta_m x}{\cos^2\beta_m L} \frac{1}{L} \int_{x'=0}^{L} (f(x') - t_\infty)$$

$$\cdot \cos (\beta_m x') \, dx' \qquad \text{(A.14)}$$

Case 1: $t_i = f(x) = \text{constant}$

$$t(x,\theta) = t_\infty + 2(t_i - t_\infty) \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\theta} \frac{Bi}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos \beta_m x}{\cos \beta_m L}$$

(A.15)

Case 2: The average temperature of the solid is

$$\bar{t}(\theta) = t_\infty + 2 \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\theta} \frac{Bi}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{1}{\cos(\beta_m L)} \frac{1}{L} \int_{x'=0}^{L} (f(x') - t_\infty) \cos(\beta_m x') \, dx'$$

(A.16)

Case 3: The average temperature of the solid when $t_i = f(x) = \text{constant}$

$$\overline{t}(\theta) = t_\infty + 2(t_i - t_\infty) \sum_{m=1}^{\infty} e^{-\alpha\beta_m^2\theta} \frac{Bi^2}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{1}{\beta_m^2 L^2} \tag{A.17}$$

Some of the relations used in the derivation of step function approximation method for variable t_∞ are:

1) $$\frac{1}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{Bi^2}{\sin^2 \beta_m L} \left[1 + \frac{\sin 2\beta_m L}{2\beta_m L}\right] = 1$$

2) $$2\, Bi \sum_{m=1}^{\infty} \frac{1}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{\cos\beta_m x}{\cos \beta_m L} = 1$$

3) $$2\, Bi^2 \sum_{m=1}^{\infty} \frac{1}{(Bi^2 + \beta_m^2 L^2 + Bi)} \frac{1}{\beta_m^2 L^2} = 1$$

APPENDIX B

PHYSICAL PROPERTIES OF AIR AND BEEF

B.1 Air Properties

The physical properties of air as a function of temperature are computed for the temperature range -40 F < t < 100 F by the following equations:

c	=	0.24	Btu/Lbm-F
Pr	=	0.72	
$\rho(t)$	=	(519.0 * 0.0765)/(460.0 + t)	Lbm/Ft^3
$\mu(t)$	=	1.71875E-8 * t + 1.11E-5	Lbm/Ft-Sec
$\nu(t)$	=	4.6875E-7 * t + 0.13E-3	Ft^2/Sec
K(t)	=	2.1875E-5 * t + 0.0133	Btu/Hr-Ft-F

B.2 Beef Properties

1. Temperature between 28 F and 102 F: The physical properties of beef are nearly constant above freezing and the following properties are used for the temperature range 28 F < t < 102 F. The specific heat data were obtained from [34] and the density data were developed in the Oklahoma State University Meat Lab. The weight fraction of the various components of the carcass were established by studying 25 carcasses in the OSU Meat Lab. The data are given in Table III.

Average specific heat of carcass

= (0.85*0.624)+(0.157)+(0.69*0.199)+(0.7*0.02)

= 0.78 Btu/Lbm-F

Average thermal conductivity of lean beef

= 0.28 Btu/Hr-Ft-F

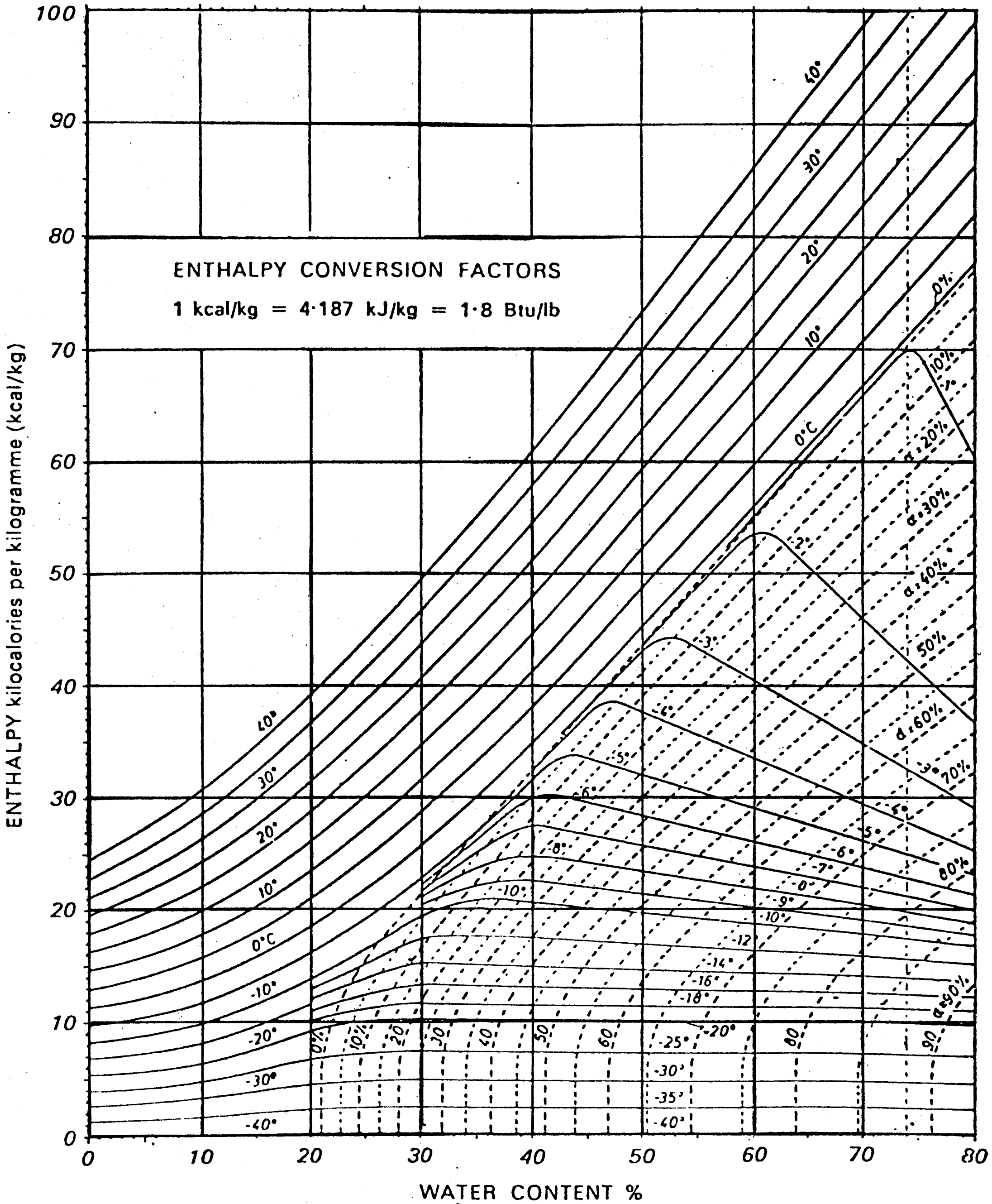
2. Temperature between -40 F and 28 F: The physical properties of beef in the freezing temperature range are a strong function of tempera-

ture. They have been studied by many investigators as given in Section 2.2. Riedel [35] studied the enthalpy as a function of percent water content for various temperature levels of beef and the data are given in Figure 33 (which is taken from Reference [34]). An average value for the water content of 74.5 percent is used by most of the investigators, and is used in this study also. The apparent specific heat is calculated using the enthalpies and the corresponding temperatures from Figure 33. The thermal conductivity as a function of temperature is available in [34]. The thermal conductivity, enthalpy, and apparent specific heat as a function of temperature, which are used in the freezing process calculations, are given in Figure 34.

TABLE III

PHYSICAL PROPERTIES OF BEEF ABOVE FREEZING TEMPERATURE

Component	Specific Heat (Btu/Lbm-F)	Density (Lbm/Ft^3)	% Weight of Carcass
Lean	0.85	65.0	62.4
Bone	0.60	82.6	15.7
Fat	0.69	56.1	19.9
Tissue	0.70	----	2.0



α is the percentage of ice, related to the total water content

Figure 33. Enthalpy-Water Content Diagram for Lean Beef Between -40 and +40 C

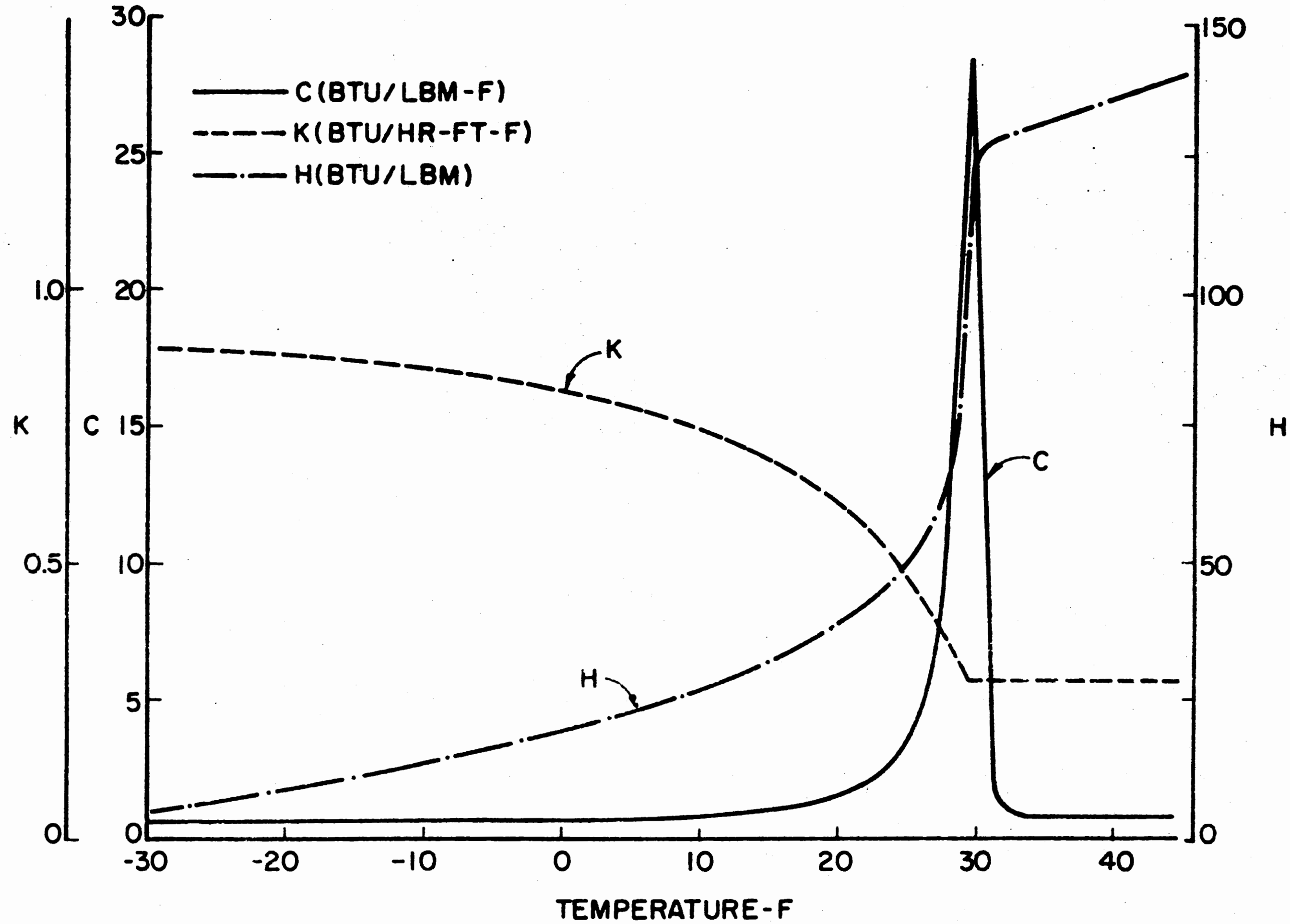


Figure 34. Thermal Physical Properties of Beef in the Freezing Temperature for Water Content of 74.5 Percent

The following equations were developed to use in the computer programs:

Apparent specific heat (Btu/Lbm-F)

$$
\begin{aligned}
Cp(t) &= 0.85 && (31.1 < t \leq 102F) \\
&= -14.722*t + 459.76 && (29.3 < t \leq 31.1\ F) \\
&= 10.889*t - 290.64 && (27.5 < t \leq 29.3\ F) \\
&= 63.874 - 15.15*t + 1.3405*t^2 && \\
&\quad -(5.1796E-2*t^3)+(7.4306E-4*t^4) && (12.2 < t \leq 27.5\ F) \\
&= 2.7778E-2*t + 0.6111 && (1.4 < t \leq 12.2\ F) \\
&= 5.42E-3*t + 0.6424 && (-40 \leq t \leq 1.4\ F)
\end{aligned}
$$

Thermal conductivity (Btu/Hr-Ft-F)

$$
\begin{aligned}
BK(t) &= 0.28 && (29 < t \leq 102\ F) \\
&= 0.79805 - (5.43226E-3*t) && \\
&\quad +(3.71516E-5*t^2)-(4.19922E-6*t^3) && \\
&\quad -(4.04242E-7*t^4) && (-20 < t \leq 29\ F) \\
&= 0.89 && (-40 \leq t \leq -20\ F)
\end{aligned}
$$

The density is a weak function of temperature and is assumed as given above. The same weight fractions of carcass as given above are used in the freezing process calculations also.

APPENDIX C

HEAT TRANSFER COEFFICIENT

Some of the heat transfer coefficient correlations available in the literature for flat plates which are used for chilling and freezing calculations of beef are as follows. They all predict the heat transfer coefficient of about the same magnitude over the ranges specified for those equations.

$$h = 1.2+0.24V \qquad \text{for } V < 16 \text{ Ft/Sec [27]} \tag{C.1}$$

$$h = 1.09+0.23V \qquad \text{for } V < 16 \text{ Ft/Sec [41]} \tag{C.2}$$

$$Nu = 0.37(Re)^{0.6} \qquad \text{for turb. flow [42]} \tag{C.3}$$

$$St_x\, P_r^{0.4} = 0.0295(Re_x)^{-0.2} \qquad \text{for turb. flow [43]} \tag{C.4}$$

$$h = 0.53(V)^{0.7} \qquad 16 < V < 100 \text{ Ft/Sec [41]} \tag{C.5}$$

Equation (C.4) is chosen because of its flexibility in accounting for property variations. The characteristic length is estimated by averaging the length and width of all of the pieces in each group. The average length was found to be one foot. Therefore one foot was used in computing the average heat transfer coefficient.

The average heat transfer coefficient:

$$St_x\, Pr^{0.4} = 0.0295\, Re_x^{-0.2}$$

For standard air with Prandtl number of 0.72 and specific heat of 0.24 Btu/Lbm-F, the heat transfer coefficient h can be written as:

$$h_x = 8.0742E\text{-}3 * (\rho V)^{0.8} * (\mu/x)^{0.2}$$

$$h = (1/1)\int_{x=0}^{L} h\, dx$$

$$= 36.3 * (\rho V)^{0.8} * (\mu/\ell)^{0.2}$$

For example, at $t = 0$ F, $l = 1$ Ft.

$$h = 0.52(V)^{0.8} \text{ Btu/Hr-Ft}^2\text{-F}$$

which is approximately the same as the McAdams equation.

APPENDIX D

HEAT TRANSFER MODEL FOR NEGLIGIBLE INTERNAL RESISTANCE

The heat exchange between a solid which has negligible internal resistance and a fluid can be treated similar to the heat exchange between two fluids in a heat exchanger. The limitations of this model are explained in model A1 of Section 3.1.1. This procedure can be used in calculating the process time and the final outlet temperature of the fluid for the known inlet temperature of the fluid and the solid and the required final average temperature of the solid. In this model the conveyor dimensions should be fixed first. The equations for this model are explained below and are used for the application of chilling hot boned beef as given by the computer program model A1, listed in Appendix H.

Calculate:

AMFR = Fluid mass flow rate (Lbm/Hr)

ACP = Fluid specific heat (Btu/Lbm-F)

AMC = AMFR * ACP

= Fluid capacity rate (Btu/Hr-F)

BMFR = Solid mass flow rate (Lbm/Hr)

BCP = Solid specific heat (Btu/Lbm-F)

BMC = BMFR * BCP

= Solid capacity rate (Btu/Hr-F)

CL = Conveyor length (Ft)

CW = Conveyor width (Ft)

Hc = Heat transfer coefficient (Btu/Hr-Ft^2-F)

If AMC > BMC

CMAX = AMC

CMIN = BMC

If BMC > AMC

CMAX = BMC

CMIN = AMC

C = CMIN/CMAX

NTU = Hc*2*CL*CW/CMIN

Using the flow capacity ratio C and the NTU, the effectiveness for any flow configuration can be determined. This effectiveness can be used to calculate the outlet temperatures of the fluid and the solid. The calculations have to be repeated by varying the time and thus the solid mass flow rate for a given conveyor arrangement, until the required final average temperature of the solid is obtained. A parametric study has to be conducted to find the design parameters. This model was not developed any further due to its limited practical applicability.

APPENDIX E

REFRIGERATION SYSTEM CHARACTERISTICS

The performance of the refrigeration equipment used in the simulation program is represented by the algebraic equations of the individual components. The characteristics of each component are developed from the data given by the manufacturers catalogues and to suit the simulation procedure given in Section 3.2.

Four sizes of condensing units ranging from 100 to 25 tons capacity at approximately 20 F evaporating and 95 F condensing temperatures are considered for the chilling process. These units provide the temperature of -5 F through 30 F air entering the conveyor for a 50-ton load. For the freezing application, five sizes of the condensing units ranging from 130 to 50 tons of capacity at approximately -10 F evaporating and 95 F condensing temperatures are considered. These units provide the temperature of -45 F through 5 F air entering the conveyor for a 50-ton load.

Five evaporator units with a capacity ranging from 35 tons through 10 tons per unit, and a 16 F temperature difference between the entering air and evaporating temperature are considered. By varying the number of units and the capacity of the units, the heat capacity ratio R or cfm per ton of refrigeration can be varied.

The equations describing the characteristics of the equipment are:

1. The condensing unity capacity (tons):

$$= \mathrm{CAPC(1)+(CAPC(2)*EVPT)+(CAPC(3)*EVPT**2)} \qquad \text{(D.1)}$$

2. The condensing temperature (F):

$$= \mathrm{CONTC(1)+(CONTC(2)*EVPT)+(CONTC(3)*EVPT**2)} \qquad \text{(D.2)}$$

3. The power intput to the compressor (BHP):

= C(1)+(C(2)*CONT)+(C(3)*EVPT)+(C(4)*CONT*EVPT)
+(C(5)*CONT**2)+(C(6)*EVPT**2) (D.3)

4. Evaporator capacity (tons):

= -Slope * EVPT + XC (D.4)

5. The power input to the evaporator fan (BHP):

$$= CE_n * cfm \qquad (D.5)$$

where

CAPC(n)	=	Condensing capacity constants for n = 1,2,3
CONTC(n)	=	Condensing temperature constants for n = 1,2,3
C(n)	=	Power input to compressor constants for n = 1, 2, ..., 6
CONT	=	Condensing temperature (F)
EVPT	=	Evaporating temperature (F)
	=	EVPEAT - TD
EVPEAT	=	Evaporator entering air temperature (F)
TD	=	Temperature drop (F)
Slope	=	Evap. Cap./TD
XC	=	Constant for a given evaporator and EVPEAT
	=	Slope * EVPEAT
CE_n	=	Constants for evaporator fan motor for n rows deep coil (CE_{10} = 1.724E-4, CE_8 = 1.63E-4, (CE_6 = 1.538E-4)

The constants for the condensing units are given in Table V for chilling and Table VI for freezing process applications. The constants for various evaporator units are given in Table IV. The evaporator capacity (CAPEVP) is given for a 16 F temperature difference between the entering air and the evaporating temperature. The other variables used are as follows:

CAPEVP	=	Evaporator capacity (tons)
BHPPE	=	Power input to the evap. fan per unit (HP)
CFMPE	=	Quantity of air circulating per evaporator unit (cfm)
BHPCPM	=	Power input to cond. water pump motor (HP)

TABLE IV

CONSTANTS FOR THE EVAPORATOR UNITS

Evaporator	1	2	3	4	5
CAPEVP (tons)	10.6	14.53	20.27	28.93	36.8
BHPPE (BHP)	1.5	2.0	3.0	6.0	6.0
CFMPE (cfm)	8900	12600	17500	26300	33000
Slope (tons/F)	0.6625	0.9081	1.2669	1.8081	2.3

The efficiencies of the various components of the refrigeration system used in this study are:

a. Compressor motor efficiency = 0.9

b. Condenser water pump efficiency = 0.7

c. Cond. water pump motor efficiency = 0.8

d. Evaporator fan motor efficiency = 0.9

TABLE V

CONDENSING UNIT CONSTANTS FOR CHILLING APPLICATION

Reference Capacity (tons)	25	50	75	100
CAPC(1)	14.33374	28.9371	42.80355	56.60441
CAPC(2)	0.432741	0.871198	1.306554	1.71661
CAPC(3)*E3	3.5330	6.41834	10.9346	15.1441
CONTC(1)	84.19632	85.32847	84.84091	84.22805
CONTC(2)	0.323237	0.366707	0.328399	0.362137
CONTC(3)*E3	0.94053	0.48017	1.83557	1.1004
C(1)	-3.245533	-6.3802264	-11.211423	-12.98955
C(2)	0.481926	0.922569	1.396513	1.77375
C(3)	-0.179985	-0.352299	-0.501875	-0.64199
C(4)*E3	4.96941	9.61513	13.89467	17.9039
C(5)*E3	-2.03147	-3.90014	-5.98402	-7.5214
C(6)*E3	-3.5929	-7.02916	-10.04125	-12.9809
BHPCPM	0.5	0.75	1.0	1.5

TABLE VI

CONDENSING UNIT CONSTANTS FOR FREEZING APPLICATION

Reference Capacity (tons)	50	65	75	95	100
CAPC(1) *E-1	6.32577	8.28671	9.17829	10.44573	16.39696
CAPC(2) *E 0	1.47990	2.06869	2.15896	1.82465	4.13863
CAPC(3) *E 2	1.27931	1.98666	1.73319	0.38395	3.60439
CONTC(1)*E-1	9.19377	9.76819	9.98519	10.54701	11.35493
CONTC(2)*E 1	4.10236	4.67681	4.61339	5.90278	5.59889
CONTC(3)*E 3	0.19506	1.70538	1.09931	3.33867	0.00000
C(1) *E-2	-0.10614	-0.11814	-0.17008	0.11981	0.68633
C(2) *E 0	1.30386	1.62822	1.97953	1.74556	1.27871
C(3) *E 0	-0.88022	-1.27162	-1.31710	-1.23687	-1.74763
C(4) *E 2	1.56218	2.15232	2.33967	2.47203	3.53088
C(5) *E 4	-5.15677	2.34308	-9.13913	22.92582	103.70860
C(6) *E 3	-3.53070	-5.66550	-5.28520	-6.09906	-8.58650
BHPCPM	1.0	1.0	1.0	1.0	1.0

APPENDIX F

FRICTION FACTOR FOR THE CONVEYOR

AND THE DUCT

The friction factor for the conveyor and the duct is to be estimated from the basic fluid flow theory as the technical information of this type of system used in practice is not available. To estimate the friction factor, some of the parameters which are not available are reasonably assumed. The influence of these parameters, their assumed average values, and affect of deviation from the assumed average values are studied. The hydraulic diameter for the conveyor system arrangement shown in Figure 13 in Chapter III is given by the following equation:

$$Deq = 4 * (CH - BPHF) * CW/(4 * CW + 2 * CH)$$

An average estimate of Deq is made as follows:

$$CW >> CH$$

$$Deq = CH - BPHF \simeq 2 \text{ Ft}$$

The average relative roughness is estimated as follows:

$$\text{The relative roughness for duct work from [31]} = k1/Deq = 0.00005/2$$

The relative roughness for conveyor and meat as estimated from samples:

$$k2/Deq = 0.03/2$$

$$\text{The average relative roughness} = (k1 + k2)/(2 * Deq) = 0.008$$

$$\text{The Reynolds number } Re = V * Deq/\nu$$

For air, and for V = 10 Ft/Sec,

$$Re = 1.5 * 10^5$$

At this high Reynolds number, the friction factor remains essentially constant and is equal to 0.0093. Therefore, f = 0.01 is proposed as the average value. For the friction factor f to become double (f =

0.02), the relative roughness has to increase approximately six times. The error, therefore, involved in the estimation of the average friction factor is negligible. In the case that a better estimate on friction factor is available, the fan horsepower and the total power requirement can be corrected. The influence of friction factor on the system power demand is explained in the parametric study in Section 5.2. The fan-power required is small compared to the total system power (less than 15 percent). This further reduces the influence of any error in the friction factor estimation on the results.

In the present study, the efficiency of the fan and its motor are assumed constant and are as follows:

a. Efficiency of the fan = 0.7.

b. Efficiency of the fan motor = 0.9.

APPENDIX G

MODELING OF BEEF CUTS

There are two main advantages for adopting the hot processing of beef rather than the cold processing. One is the reduction in mass and the other is the reduction in the size of the cuts. Relatively thin cuts to be chilled in hot processing compared to the thick carcass in the case of cold processing. The average thickness of the carcass is approximately three to four inches and the thickest section is up to ten inches thick. In the case of hot processing the grouping of the cuts and their average thickness are determined after careful study of the hot boned beef from 25 carcasses in the OSU meat lab. There are 11 different specific cuts which represent 59.8 percent of the hot processing yield. The remaining 40.2 percent is of relatively small pieces known as the lean. Out of the 11 cuts, 4 have an average thickness ranging from 2.5 to 3.1 inches. These represent 29.3 percent by weight of the total yield and this class has an average thickness of 2.8 inches. The other 7 cuts have an average thickness ranging from 1.2 to 1.9 inches. These represent 30.5 percent by weight of the total yield and this class has an average thickness of 1.8 inches. The third group is the lean, 40.2 percent by weight. The lean was bagged so that the maximum thickness will not exceed two inches and this gave an average thickness of 1.5 inches. The groups can be summarized as shown in Table VII.

The weight of the product of a given group that can be loaded on a unit conveyor area is the loading factor. It is given by:

Loading factor = Average thickness * Density of the product.

TABLE VII

CLASSIFICATION OF HOT BONED BEEF CUTS

Group	Percent By Weight	Average Thickness (in.)
A	29.3	2.8
B	30.5	1.8
C	40.2	1.5

APPENDIX H

COMPUTER PROGRAMS

```
C
C COMPUTER PROGRAM FOR MODEL - A3
C ********************************
C
C   THIS PROGRAM COMPUTES ALL THE NECESSARY VALUES FOR DEVELOPING THE
C   OPTIMUM CHARACTERISTICS FOR DESIGN OF THE CONVEYARISED CHILLING
C   SYSTEM OF ANY 1-D SOLIDS.
C   THE PRESENT EXAMPLE IS SET FOR HOT BONED BEEF
C
C  INPUTS :
C  * * * * * *
C
C   INPUTS ARE IN NAMELIST FORM, THE VALUES IN THE PARENTHESIS ARE
C   THE DEFAULT VALUES
C   CARD(S)-1
C   /NAME1/
C   BTI    =BEEF INITIAL TEMPERATURE (100.0 F)
C   BTF    =BEEF FINAL AVE. TEMP. REQUIRED (40.0F)
C   BPLF   =BEEF PIECE AVERAGE LENGTH (1.0 FT)
C   BMFR   =BEEF MASS FLOW RATE (11765.0 LBM/HR)
C   BPHI   =BEEF PIECE THICKNESS (4.0 IN)
C   NV     =NUMBER OF REFRIGERATION SYSTEMS TO BE STUDIED (1)
C   CARD(S)-2
C   /NAME2/
C   TIMET  =ESTIMATED TOTAL CHILLING TIME (1.5*BPHI HRS)
C   EVPEAT =ESTIMATED EVAPORATOR ENTERING AIR TEMPERATURE (30.0 F)
C   CNCWT  =CONDENSER COOLING WATER TEMPERATURE FOR PRINTING (75.0 F)
C   ICU    =CONDENSING UNIT SELECTED
C   IEVP   =EVEPORATOR UNIT SELECTED
C   NUEVP  =NUMBER OF EVAPORATORS SELECTED
C
C  OUTPUT :
C  * * * * * *
C
C   TEMP   =TEMPERATURE DISTRIBUTON IN THE SOLID FROM CENTER TO BOUNDARY
C   THE REST IS SELF EXPLANITARY
C
C  SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
C   * * * * * * * * * * * * * * * * * * * * * * *
C
C    SUBROUTINE TEMPR(TLXTH,TINF,B,BL,TLPXTH)
C    SUBROUTINE BN(C,B,NR)
C    SUBROUTINE REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,IRSYS,ICE)
C    SUBROUTINE CONDU(C,CAPC,CONTC,BHPCPM,IRSYSC,ICU)
C    SUBROUTINE EVPR(CAPEVP,BHPPE,CFMPE,IEVP)
C
C  KEY TO SYMBOLS
C  * * * * * * * *
C
C   ACP    =AIR SPECIFIC HEAT (BTU/LBM-F)
C   AK     =AIR THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C   ALPHA  =BEEF THERMAL DIFFUSIVITY (FT**2/HR)
C   AMC    =AIR CAPACITY RATE (BTU/HR-F)
C   AMU    =AIR ABSOLUTE VISCOSITY (LBM/FT-SEC)
C   ANU    =AIR KINEMATIC VISCOSITY (FT**2/SEC)
C   ARO    =AIR DENSITY (LBM/FT**3)
C   B(N)   =B   IN (BL)*(TAN(BL))=C EQUATION   (N=1,..NR)
C   BCP    =BEEF SPECIFIC HEAT (BTU/LBM-F)
C   BHPCMM =BHP INPUT TO COMPRESSOR MOTOR
C   BHPCOM =BHP INPUT TO COMPRESSOR
C   BHPCPM =BHP INPUT TO CONDENSER PUMP MOTOR
C   BHPEFM =BHP INPUT TO EVAPORATOR FAN MOTOR
C   BHPPE  =BHP INPUT TO EVAPORATO FAN
C   BHPT   =TOTAL BHP INPUT TO REFRIGERATION SYSTEM
C   BHPT1  =TOTAL BHP INPUT TO THE SYSTEM INCLUDING CONVEYOR FAN
C   BHPTPT =TOTAL BHP PER TON TO THE SYSTEM
C   BIOT   =BIOT NUMBER
C   BK     =BEEF THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C   BL(N)  =B*L IN (BL)*(TAN(BL))=C EQUATION  (N=1,..NR)
C   BLF    =BEEF LOADING FACTOR (LBM/FT**2)
C   BMC    =BEEF CAPACITY RATE (BTU/HR-F)
C   BMFR   =BEEF MASS FLOW RATE (LBM/HR)
C   BPHI   =BEEF PIECE THICKNESS (IN)
C   BPHF   =BEEF PIECE THICKNESS (FT)
C   BPLF   =BEEF PIECE AVERAGE LENGTH (FT)
C   BRO    =BEEF DENSITY (LBM/FT**3)
C   BTI    =BEEF INITIAL TEMPERATURE (F)
C   BTF    =BEEF FINAL AVE. TEMPR. REQUIRED (F)
C   C(N)   =CONSTANTS FOR COMPRESSOR INPUT CALCULATION (N=1,..6)
C   CAPC(N)=CONSTANTS FOR THE CAPACITY OF THE SYSTEM (N=1,2,3)
C   CAPEVP =CAPACITY OF EVEPARATOR IN TONS FOR 1L F DROP IN TEMPERATURE
C   CAR    =CONVEYOR AREA (FT**2)
C   CART   =CONVEYOR TOTAL AREA (FT**2)
C   CFMPE  =CFM PER EVAPORATOR
C   CFMPT  =CFM OF AIR PER TON OF COOLING LOAD
C   CH     =CONVEYOR HEIGHT (FT)
C   CL     =CONVEYOR LENGTH (FT)
C   CLTWR  =CONVEYOR LENGTH TO WIDTH RATIO (CL/CW)
C   CNCWT  =CONDENSER COOLING WATER INLET TEMPERATURE (F)
C   CONT   =CONDENSING TEMPERATURE OF THE REFRIGERANT (F)
C   CONTC(N)=CONSTANTS FOR CONDENSING TEMPERATURE OF THE SYSTEM (N=1,2,3
C   CVEL   =CONVEYOR VELOSITY (FT/HR)
C   CW     =CONVEYOR WIDTH (FT)
C   DEQ    =HYDRAULIC DIAMETER OF THE DUCTWORK AND THE CONVEYOR (FT)
C   DPC    =PRESSURE DROP OF AIR ON CONVEYOR IN IN. OF WATER
C   DPCON  =PRESSURE DROP IN THE CONNECTING DUCT WOTK IN IN. OF WATER
C   DPCPSI =PRESSURE DROP OF AIR ON CONVEYOR IN PSI
C   DPTOT  =TOTAL PRESSURE DROP OF AIR ON COVEYORIN IN. OF WATER
C   DTIME  =TIME AT EACH SECTION ON CONVEYOR (HR)
C   DXB    =INCREMENTAL LENGTH IN BEEF PIECE (FT)
C   EFCM   =EFFICIENCY OF COMPRESSOR MOTOR
C   EFEFM  =EFFICIENCY OF CIRCULATIG FAN AND ITS MOTOR
C   EVPEAT =EVEPERATOR ENTERING AIR TEMPERATURE (F)
C   EVPLAT =EVEPERATOR LEAVING  AIR TEMPERATURE (F)
C   EVPT   =EVEPORATING TEMPERATURE OF THE REFRIGERANT (F)
C   FR     =FRICTION FACTOR
C   HC     =HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-F)
C   HCP    =(K/THICKNESS) FOR PLASTIC BAG (BTU/HR-FT**2-F)
C   ICU    =CONDENSING UNIT SELECTED (1-25TON,2-50TON,3-75TON,4-100TON
C             AT APPROX. 20F EVPT. AND 95F CONT. TEMPERATURES)
C   IEVP   =EVAPORATOR UNIT SELECTED  (1-10.67TON,2-14.53TON,3-20.27TON,
C             4-28.93TON,5-36.8TON, FOR A 16F TEMPR. DROP)
C   NBS    =NUMBER OF SECTIONS IN THE BEEF PIECE (10)
C   NCS    =NUMBER OF SECTIONS ON THE CONVEYOR
C   NR     = NUMBER OF ROOTS OF TRANSCENDENTAL EQN. REQUIRED
C   NUEVP  =NUMBER OF EVAPORATORS SELECTED
C   RCAP   =REQUIRED COOLING CAPACITY (TONS)
C   RMC    =RATIO OF HEAT CAPACITY FLOW RATES OF THE TWO STREAMS
C   TAV    =AVERAGE TEMP. OF BEEF PIECE AT SECTION (N+1) ON THE CONVEYOR
```

```
C     TAVI   =AVERAGE TEMP. OF BEEF PIECE AT SECTION N ON THE CONVEYOR
C     TIMET  =TOTAL TIME
C     TINF   =AIR TEMPERATURE (F)
C     TLPXTH =TEMP. AT SEC. X IN BEEF PIECE AND AT SEC. (N+1) ON CONV.
C     TLXTH  =TEMP. AT SEC. X IN BEEF PIECE AND AT SEC. N ON CONV.
C     U      =OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-F)
C     VEL    =VELOCITY OF AIR ON CONVEYOR (FT/SEC.)
C     XBI    =HALF THICKNESS OF BEEF PIECE (IN)
C     XBF    =HALF THICKNESS OF BEEF PIECE (FT)
C
      DIMENSION  B(10),BL(10),TLXTH(21),TLPXTH(21)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,DTIME
      COMMON/BR/EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
      NAMELIST /NAME1/BTI,BTF,BPLF,BMFR,BPHI,NV
     $ /NAME2/ TIMET,EVPEAT,CNCWT,ICU,IEVP,NUEVP
C
C     AIR PROPERTIES FOR -40.0 . LE. T . LE . 100.0 F
C
      ARO(T)=(519.0*0.0765)/(460.0+T)
      AMU(T)=1.11E-05+(1.71875E-08*T)
      ANU(T)=0.13E-03+(4.6875E-07*T)
      AK(T)=0.0133+(2.1875E-05*T)
      ACP=0.24
C
C     BEEF PROPERTIES CONSTANT ABOVE FREEZING TEMPERATURE OF 28.5 F
C
      BCP=0.85
      BRO=65.0
      BK=0.28
      BTI=100.0
      BTF=40.0
      BPLF=1.0
      BMFR = 11765.0
      BPHI =4.0
      NV=1
      CNCWT=75.0
      READ(5,NAME1)
      BLF=BPHI*BRO/12.0
      BPHF=BPHI/12.0
      XBI=BPHI/2.0
      XBF=XBI/12.0
      ERROR=(BTI-BTF)/(BTI*2.0)
      NBS=10
      NBSP1=NBS+1
      DXB=XBF/NBS
      ALPHA=BK/(BRO*BCP)
      WRITE(6,210)
      WRITE(6,220) BCP,BRO,BK,ALPHA,BTI,BTF,BPHI,BMFR,BLF
      WRITE(6,330)
      WRITE(6,NAME1)
C
C     CONVEYOR
C
      NCS=10
      NCSP1=NCS+1
      CLTWR=4.0
      CAR=BMFR/BLF
C
C     SYSTEM
C
      HCP=550.0
      BMC=BMFR*BCP
      RCAP=BMC*(BTI-BTF)/12000.0
      NR=5
      FANEFF=0.7
      VAC=10.0
C
C     INITIAL ESTIMATE OF EVPEAT AND TIMET
C
      EVPEAT=30.0
      TIMET=BPHI*1.5
C     LOOP 10 IS FOR NUMBER OF REFRIGERATION SYSTEMS TO BE STUDIED
      DO 10 IV=1,NV
      READ(5,NAME2)
      DO 15 IVEL=1,5
      IPAGE=IVEL/2
      IPAGE2=IPAGE*2
      IF(IPAGE2.NE.IVEL)  WRITE(6,210)
      IF(IVEL.EQ.1)  WRITE(6,NAME2)
      WRITE(6,320)
      VAC=IVEL*10
      IVT=0
C     LOOP 20 IS FOR THE CONVERGENCE ON THE TOTAL CHILL TIME REQUIRED
      DO 20 IT=1,100
C
C     CONVEYOR SIZE CALCULATIONS
C
      CART=CAR*TIMET
      CW=SQRT(CART/CLTWR)
      CL=CART/CW
      CVEL=CL/TIMET
      DTIME=TIMET/NCS
      TIMET1=TIMET
      IEC=0
      IF(IVT.EQ.1)  GO TO 45
   26 CONTINUE
      IEC=IEC+1
C
      CALL REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,IRSYSC,
C
     1  IEC)
      T=EVPEAT
      AMC=CFM*ARO(T)*60.0*ACP
      EVPEAT=EVPLAT+(RCAP*12000.0/AMC)
      IECF=1
      IF(IEC.GT.50)  GO TO 25
      IECF=0
      IF(ABS((RCAP-SYSCAP)/RCAP).GT.0.005) GO TO 26
      IVT=1
      CFMPT=CFM/RCAP
   25 CONTINUE
      IF(IECF.EQ.1)  WRITE(6,250)
      IF(IECF.EQ.1)  GO TO 10
      T=EVPEAT
      AMC=CFM*ARO(T)*60.0*ACP
      RMC=BMC/AMC
      T=(EVPEAT+EVPLAT)/2.0
      HC=36.334*((ARO(T)*VAC)**0.8)*((AMU(T)/BPLF)**0.2)
      UI=(1.0/HC)+(1.0/HCP)
```

```
      U=1.0/UI
      INIT=0
      BIOT= U*XBF/BK
C
      CALL BN(BIOT,BL,NR)
C
      DO 40 J=1,NR
      B(J)=BL(J)/XBF
40    CONTINUE
45    CONTINUE
      TIME=0.0
      TINF=EVPEAT
      TAVI=BTI
      TAV=BTI
      DO 50 I=1,NBSP1
      TLXTH(I)=BTI
50    CONTINUE
      CH=BPHF+(CFM/(CW*VAC*60.0))
      DEQ=(4.0*(CH-BPHF)*CW)/((4.0*CW)+(2.0*CH))
C     LOOP 60 IS FOR VARIOUS SECTIONS ON THE CONVEYOR
      DO 60 N=1,NCS
      TIME=TIME+DTIME
      TINF=TINF-(RMC*(TAVI-TAV))
      TAVI=TAV
C
      CALL TEMPR(TLXTH,TINF,B,BL,TLPXTH)
C
      INIT=1
C     LOOP 70 IS FOR TRANSFERING THE TEMPERATURE ARRAY
      DO 70 J=1,NBSP1
      TLXTH(J)=TLPXTH(J)
70    CONTINUE
      TAV=(TLXTH(1)+TLXTH(NBSP1))/2.0
C     LOOP 80 IS FOR CALCULATING THE AVERAGE TEMPERATURE
      DO 80 J=2,NBS
      TAV=TAV+TLXTH(J)
80    CONTINUE
      TAV=TAV/NBS
60    CONTINUE
      TDIF=TAV-BTF
C     CORRECTING THE TOTAL CHILL TIME
      DCOR=0.01
      IF(TDIF.LT.0.5)  DCOR=0.005
      TIMET=TIMET+(TDIF*DCOR*BPHI)
      ITF=1
      IF(ABS(TDIF).GT.ERROR)  GO TO 20
      WRITE(6,260) VAC, U,CH,DEQ,RMC,BIOT
      WRITE(6,230) TIMET1,RCAP,CART,CW,CL,CVEL
      WRITE(6,310) EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
      WRITE(6,240) IRSYSC,ICU,IEVP,EVPEAT,EVPLAT,SYSCAP,CFM,BHPT
     $  ,CFMPT
      WRITE(6,270) TIME,TINF,TAV
      WRITE(6,280)  (TLXTH(J),J=1,NBSP1)
C     LOOP 90 IS FOR CALCULATING THE CIRCULATING FAN POWER REQUIRED
      ITF=0
      FR=0.005
      DO 90 IFR=1,3
      DPCPSI=(FR*2.0*ARO(T)*VAC*VAC*CL)/ (DEQ*32.2*144.0)
      DPC=27.7*DPCPSI
      DPCON=2.0*0.8303*ARO(T)*((VAC*60.0/1000.0)**2)
      DPTOT=DPC+DPCON
      FANHP=(CFM*DPTOT)/(6350.0*FANEFF)
      BHPT1=BHPT+FANHP
      BHPTPT=BHPT1/SYSCAP
      WRITE(6,290)  FR, DPC,DPCON,DPTOT,FANHP,BHPT1,BHPTPT
      FR=FR*2.0
90    CONTINUE
      GO TO 15
20    CONTINUE
      IF(ITF.EQ.0)  GO TO 15
      WRITE(6,300)
      WRITE(6,260) VAC, U,CH,DEQ,RMC,BIOT
      WRITE(6,230) TIMET1,RCAP,CART,CW,CL,CVEL
      WRITE(6,310) EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
      WRITE(6,240) IRSYSC,ICU,IEVP,EVPEAT,EVPLAT,SYSCAP,CFM,BHPT
     $  ,CFMPT
      WRITE(6,270) TIME,TINF,TAV
      WRITE(6,280)  (TLXTH(J),J=1,NBSP1)
15    CONTINUE
10    CONTINUE
210   FORMAT(1H1)
220   FORMAT(//,5X,'AVERAGE BEEF PIECE SPECIFICATIONS',//,
     1 10X,'SP. HEAT        =',F5.2,2X,'BTU/LB-F',/,
     2 10X,'DENSITYY        =',F5.1,2X,'LB/FT**3',/,
     3 10X,'TH. COND.       =',F5.2,2X,'BTU/HR-FT-F',/,
     4 10X,'TH. DIFUSIV.    =',E11.4,2X,'FT**2/HR',/,
     5 10X,'AV. TEMP. IN    =',F6.1,2X,'F',/,
     6 10X,'AV. TEMP. OUT   =',F6.1,2X,'F',/,
     7 10X,'AV. THICKNESS   =',F5.2,2X,'IN',/,
     8 10X,'MASS FLOW RT.   =',F8.1,2X, 'LB/HR',/,
     9 10X,'LOADING FACT.   =',F5.1,2X,'LB/FT**2',/)
230   FORMAT(10X,'COOLING TIME =',F5.2,2X,'HRS',
     1 14X,'REQUIRED CAPACITY   =',F5.1,2X,'TONS',/,
     3 10X,'TOTAL CONV. AREA    =',F5.0,2X,'FT**2',
     4  6X,'CONV. WEDITH        =',F5.1,2X,'FT',/,
     5 10X,'CONV. LENGTH        =',F6.1,2X,'FT',
     6  8X,'CONV. VELOCITY      =',F5.1,2X,'FT/HR',/)
240   FORMAT(10X,'RATED CAP. OF COND. UNIT =',I4,2X,'TONS',
     1 5X,'ICU=',I2,5X,'IEVP=',I2,/,
     2 10X,'EVP. ENT. AIR TEMP. =',F5.1,2X,'F',
     3 10X,'EVP. LEV. AIR TEMP. =',F5.1,2X,'F',/,
     4 10X,'SYSTEM CAPACITY      =',F6.1,2X,'TONS',
     5  6X, 'CFM    =',E12.4,2X,/,
     6 10X,'TOT. BHP OF REFRIG. SYS. =',F6.1,
     7  7X,'CFM/TON =',F6.0,/)
250   FORMAT(10X,'NOT CONVERGED IN SYSCAP   * * * * * '  )
260   FORMAT(5X,'VEL. OF AIR ON CONV. =',F5.1,'FT/SEC',
     1 10X,'HEAT TRANS. COEF.  =',F6.2,'BTU/HR-FT**2-F',/,
     2 10X,'CONV. HEIGHT        =',F5.1,'FT',
     3 11X,'EQ. DIAMETER        =',F5.1,'FT',/,
     4 10X,'RATIO OF MCP        =',F5.3,
     5 13X,'BIOT NUMBER         =',F6.2,/)
270   FORMAT(6X,'TIME=',F5.2,2X,'HR',10X,'TINF=',F6.1,2X,'F',
     1 10X,'AV. BEEF PIECE TEMP.  =',F6.1,2X,'F',/)
280   FORMAT(8X,'TEMP=',11(F6.2,4X),/)
290   FORMAT(8X,'FR=',F5.3,5X,'DP. CONV.  =',F4.1,2X,'IN H20',
     1 10X,'DP. CONCT.  =',F4.1,2X,'IN H20',
     2 10X,'DP. TOTAL   =',F4.1,2X,'IN H20',/,
     3 10X,'FAN HP      =',F5.1,2X,
     4 10X,'TOTAL HP    =',F6.1,2X,'HP',
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     5 10X,'HP/TON       =',F6.4,/)
300   FORMAT(10X,'NOT CONVERGED IN TIME  * * * * *',
     1 10X,'THE LAST ITERATION VALUES ARE',/)
310   FORMAT(8X,'EVPT=',F6.2,4X,'CONT=',F6.2,4X,'BHPCMM=',F6.1,4X,
     $ 'BHPEFM=',F6.1,4X,'BHPCPM=',F6.2,4X,'NUEVP=',I2)
320   FORMAT(/)
330   FORMAT(//)
      STOP
      END
      SUBROUTINE TEMPR(TLXTH,TINF,B,BL,TLPXTH)
C     *******************************************
C
C     THIS ROUTINE COMPUTES THE TEMPERATURE AT EVERY SECTION IXB WHICH
C     VARIES FROM 1 AT THE CENTER TO NBSP1 AT THE OUTER BOUNDARY OF
C     THE 1 - D SOLID
C
      DIMENSION B(1),TLXTH(1),TLPXTH(1),BL(1)
      DIMENSION EXPT(10),CSBNX(21,10),FX(10),FXT(21),CONSTR(10)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,DTIME
      IF(INIT.GT.0)  GO TO 10
      DO 20 NBL=1,NR
      XB=0.0
      DO 30 IXB=1,NBSP1
      CSBNX(IXB,NBL)=COS(B(NBL)*XB)
      XB=XB+DXB
30    CONTINUE
      CONST1=BL(NBL)*BL(NBL)
      CONST2=(BIOT*BIOT)+CONST1+BIOT
      CONSTR(NBL)=CONST1/CONST2
20    CONTINUE
10    CONTINUE
      DO 40 NBL=1,NR
      EXPP=-ALPHA*B(NBL)*B(NBL)*DTIME
      EXPT(NBL)=0.0
      IF(EXPP.LT.-10.0)  GO TO 40
      EXPT(NBL)=EXP(EXPP)
40    CONTINUE
      DO 50 NBL=1,NR
      DO 60 IXB=1,NBSP1
      FXT(IXB)=(TLXTH(IXB)-TINF)*CSBNX(IXB,NBL)
60    CONTINUE
      FX(NBL)=(FXT(1)+FXT(NBSP1))/2.0
      DO 70 IXB=2,NBS
      FX(NBL)=FX(NBL)+FXT(IXB)
70    CONTINUE
      FX(NBL)=FX(NBL)*DXB/XBF
50    CONTINUE
      DO 100 IXB=1,NBSP1
      DUM1=0.0
      DO 110 NBL=1,NR
      DUM1=DUM1+(EXPT(NBL)*CONSTR(NBL)*CSBNX(IXB,NBL)*FX(NBL)/
     $ (CSBNX(NBSP1,NBL)*CSBNX(NBSP1,NBL)))
110   CONTINUE
      TLPXTH(IXB)=TINF+(2.0*DUM1)
100   CONTINUE
      RETURN
      END
      SUBROUTINE BN(C,B,NR)
C     *******************************************
C
C     COMPUTATIONS OF THE ROOTS OF TRANSCENDENTAL EIGEN CONDITIONS FOR T
C     EQUATION   ***  B(N)*TAN(B(N))=C  ***
C     WHERE  N = 1,NR      NR - THE NUMBER OF ROOTS REQUIRED
C     B(N) IN THIS ROUTINE IS SAME AS THE BL(N) INTHE MAIN PROGRAM
C
      DIMENSION B(1)
      DATA  PI/3.141593/,ERROR/1.0E-05/
      N=1
      B(1)=PI/SQRT((8.0/C)+4.0)
      NIL=1
      IF(C.GE.2.0)  GO TO 30
      NIL=2
      DO 10 IN=1,100
      TB1=TAN(B(1))
      B1=C/TB1
      IF(ABS(B(1)-B1).LE.ERROR)  GO TO 30
      B(1)=(B(1)+B1)/2.0
10    CONTINUE
20    WRITE(6,110)  N
110   FORMAT(10X,'DID NOT CONVERGE IN THE 100  ITERATIONS OF THE ROOT',
     $I5)
      RETURN
30    CONTINUE
      DO 40 N=NIL,NR
      BNP1=B(1)
      DO 50 IN=1,100
      BNP2=ATAN(C/(((N-1)*PI)+BNP1))
      IF(ABS(BNP2-BNP1).LE.ERROR)  GO TO 60
      BNP1=BNP2
50    CONTINUE
      GO TO 20
60    B(N)=((N-1)*PI)+BNP1
40    CONTINUE
      RETURN
      END
      SUBROUTINE REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,
     1IRSYSC,IEC)
C     *******************************************
C
C     THIS ROUTINE COMPUTES THE OPERATING POINTS OF ALL THE COMPONENTS
C     OF THE REFRIGERATION SYSTEM CHOOSEN BY ICU,IEVP,NUEVP  - AT THE
C     SYSTEM BALANCE POINT
C
      DIMENSION  C(6),CAPC(3),CONTC(3)
      COMMON/BR/EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
C
C     GENERAL CONSTANTS
C
      TD=16.0
      EFCM=0.9
      EFEFM=0.9
      ERROR=0.001
      IF(IEC.GT.1)  GO TO 5
C
C     CONDENSING UNIT SELECTION
      CALL CONDU(C,CAPC,CONTC,BHPCPM,IRSYSC,ICU)
      XIRSYS=IRSYSC
C
C
```

```
      5 10X,'HP/TON        =',F6.4,/)
300   FORMAT(10X,'NOT CONVERGED IN TIME  * * * * *',
     1 10X,'THE LAST ITERATION VALUES ARE',/)
310   FORMAT(9X,'EVPT=',F6.2,4X,'CONT=',F6.2,4X,'BHPCMM=',F6.1,4X,
     $ 'BHPEFM=',F6.1,4X,'BHPCPM=',F6.2,4X,'NUEVP=',I2)
320   FORMAT(/)
330   FORMAT(//)
      STOP
      END
      SUBROUTINE TEMPR(TLXTH,TINF,B,BL,TLPXTH)
C     ******************************************
C
C     THIS ROUTINE COMPUTES THE TEMPERATURE AT EVERY SECTION IXB WHICH
C     VARIES FROM 1 AT THE CENTER TO NBSP1 AT THE OUTER BOUNDARY OF
C     THE 1 - D SOLID
C
      DIMENSION B(1),TLXTH(1),TLPXTH(1),BL(1)
      DIMENSION EXPT(10),CSBNX(21,10),FX(10),FXT(21),CONSTR(10)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,DTIME
      IF(INIT.GT.0)  GO TO 10
      DO 20 NBL=1,NR
      XB=0.0
      DO 30 IXB=1,NBSP1
      CSBNX(IXB,NBL)=COS(B(NBL)*XB)
      XB=XB+DXB
30    CONTINUE
      CONST1=BL(NBL)*BL(NBL)
      CONST2=(BIOT*BIOT)+CONST1+BIOT
      CONSTR(NBL)=CONST1/CONST2
20    CONTINUE
10    CONTINUE
      DO 40 NBL=1,NR
      EXPP=-ALPHA*B(NBL)*B(NBL)*DTIME
      EXPT(NBL)=0.0
      IF(EXPP.LT.-10.0)  GO TO 40
      EXPT(NBL)=EXP(EXPP)
40    CONTINUE
      DO 50 NBL=1,NR
      DO 60 IXB=1,NBSP1
      FXT(IXB)=(TLXTH(IXB)-TINF)*CSBNX(IXB,NBL)
60    CONTINUE
      FX(NBL)=(FXT(1)+FXT(NBSP1))/2.0
      DO 70 IXB=2,NBS
      FX(NBL)=FX(NBL)+FXT(IXB)
70    CONTINUE
      FX(NBL)=FX(NBL)*DXB/XBF
50    CONTINUE
      DO 100 IXB=1,NBSP1
      DUM1=0.0
      DO 110 NBL=1,NR
      DUM1=DUM1+(EXPT(NBL)*CONSTR(NBL)*CSBNX(IXB,NBL)*FX(NBL)/
     $ (CSBNX(NBSP1,NBL)*CSBNX(NBSP1,NBL)))
110   CONTINUE
      TLPXTH(IXB)=TINF+(2.0*DUM1)
100   CONTINUE
      RETURN
      END
      SUBROUTINE BN(C,B,NR)
C     ******************************************
C
C     COMPUTATIONS OF THE ROOTS OF TRANSCENDENTAL EIGEN CONDITIONS FOR T
C     EQUATION   ***  B(N)*TAN(B(N))=C  ***
C     WHERE  N = 1,NR      NR - THE NUMBER OF ROOTS REQUIRED
C     B(N) IN THIS ROUTINE IS SAME AS THE BL(N) INTHE MAIN PROGRAM
C
      DIMENSION B(1)
      DATA  PI/3.141593/,ERROR/1.0E-05/
      N=1
      B(1)=PI/SQRT((3.0/C)+9.0)
      NIL=1
      IF(C.GE.2.0)  GO TO 30
      NIL=2
      DO 10 IN=1,100
      TB1=TAN(B(1))
      B1=C/TB1
      IF(ABS(B(1)-B1).LE.ERROR)  GO TO 30
      B(1)=(B(1)+B1)/2.0
10    CONTINUE
20    WRITE(6,110)  N
110   FORMAT(10X,'DID NOT CONVERGE IN THE 100  ITERATIONS OF THE ROOT',
     $I5)
      RETURN
30    CONTINUE
      DO 40 N=NIL,NR
      BNP1=B(1)
      DO 50 IN=1,100
      BNP2=ATAN(C/(((N-1)*PI)+BNP1))
      IF(ABS(BNP2-BNP1).LE.ERROR)  GO TO 60
      BNP1=BNP2
50    CONTINUE
      GO TO 20
60    B(N)=((N-1)*PI)+BNP1
40    CONTINUE
      RETURN
      END
      SUBROUTINE REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,
C     ******************************************
C
C     THIS ROUTINE COMPUTES THE OPERATING POINTS OF ALL THE COMPONENTS
C     OF THE REFRIGERATION SYSTEM CHOOSEN BY ICU,IEVP,NUEVP  - AT THE
C     SYSTEM BALANCE POINT
C
     1IRSYSC,IEC)
      DIMENSION  C(6),CAPC(3),CONTC(3)
      COMMON/BR/EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
C
C     GENERAL CONSTANTS
C
      TD=16.0
      EFCM=0.9
      EFEFM=0.9
      ERROR=0.001
      IF(IEC.GT.1)  GO TO 5
C
C     CONDENSING UNIT SELECTION
      CALL CONDU(C,CAPC,CONTC,BHPCPM,IRSYSC,ICU)
      XIRSYS=IRSYSC
C
C
```

```
C     EVAPORATOR SELECTION
      CALL EVPR(CAPEVP,BHPPE,CFMPE,IEVP)
C
C
C     TOTAL SYSTEM
C
5     CONTINUE
      BHPEFM=(BHPPE*NUEVP)/EFEFM
      CAPEVT=CAPEVP*NUEVP
      CFM=CFMPE*NUEVP
      SLOPE=-CAPEVT/TD
      Y1=CAPEVP*(NUEVP+1)
      X=EVPEAT-TD
      XC=CAPEVT-(SLOPE*X)
      DO 10 I=1,100
      Y2=CAPC(1)+(CAPC(2)*X)+(CAPC(3)*X*X)
      X=(Y1-XC)/SLOPE
      IF(ABS((Y1-Y2)/Y2).LE.ERROR)  GO TO 20
      Y1T=Y1
      Y1=Y2
10    CONTINUE
      WRITE(6,30)  Y1T,Y2
30    FORMAT(5X,' * * * * * DIDNOT CONVERGE IN REFSYS',
     1 5X,'Y1 =',F6.1,5X,'Y2 =',F6.1)
      DEVP=2.0
      IF(CAPEVT.LT.XIRSYS)  DAVP=-DAVP
      EVPEAT=EVPEAT+DEVP
      IEC=IEC+1
      IF(IEC.LE.50) GO TO 5
      SYSCAP=0.0
      RETURN
20    CONTINUE
      EVPT=X
      SYSCAP=Y2
      CONT=CONTC(1)+(CONTC(2)*X)+(CONTC(3)*X*X)
      A=CONT
      B=EVPT
      BHPCOM=C(1)+(C(2)*A)+(C(3)*B)+(C(4)*A*B)+(C(5)*A*A)+(C(6)*B*B)
      BHPCMM=BHPCOM/EFCM
      BHPT=BHPCMM+BHPEFM+BHPCPM
      RO=519.0*0.0765/(460.0+EVPEAT)
      AMC=CFM*RO*60.0*0.24
      EVPLAT=EVPEAT-(SYSCAP*12000.0/AMC)
      RETURN
      END
      SUBROUTINE CONDU(C,CAPC,CONTC,BHPCPM,IRSYSC,ICU)
C     *******************************************
C
C     THIS ROUTINE ASSIGNS THE CONSTANTS FOR COMPUTING REFR. CAP. ,
C     COND. TEMPR., AND BHP INPUT TO COMPRESSOR
C     THE COMPRESSOR CAPACITY MENSIONED IN COMENT CARD IS AT APPROX.
C     20 F EVPT AND 95 F CONT , IT IS ONLY FOR IDENTIFICATION
C
      DIMENSION C(1),CAPC(1),CONTC(1)
      IF(ICU.GT.5)  ICU=5
      GO TO (25,50,75,100,150)  ,ICU
150   CONTINUE
      WRITE(6,110)
110   FORMAT(10X,'NO CONDENSING UNIT IS DEFINED')
      STOP
```

```
C
C
100   CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 100 TON
C     CONDENSER - 20 IN * 14 FT, 4 - PASS, 394 GPM
C
      C(1)=-0.12939546E02
      C(2)=0.17737462E01
      C(3)=-0.64199337
      C(4)=0.17903361E-01
      C(5)=-0.75213602E-02
      C(6)=-0.12939867E-01
      CAPC(1)=56.69441355
      CAPC(2)=1.716698392
      CAPC(3)=1.514407627E-02
      CONTC(1)=84.22804927
      CONTC(2)=0.3621374973
      CONTC(3)=1.100450227E-03
      BHPCPM=1.3
      IRSYSC=100
      RETURN
75    CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 75  TON
C     CONDENSER - 16 IN * 18 FT, 4 - PASS, 247 GPM
C
      C(1)=-0.11211423E02
      C(2)=0.13965135E01
      C(3)=-0.5018749
      C(4)=0.13894672E-01
      C(5)=-0.59840245E-02
      C(6)=-0.10041252E-01
      CAPC(1)=42.80354994
      CAPC(2)=1.306553888
      CAPC(3)=1.09345556E-02
      CONTC(1)=84.84090735
      CONTC(2)=0.3283994243
      CONTC(3)=1.835567383E-03
      BHPCPM=1.0
      IRSYSC=75
      RETURN
50    CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 50  TON
C     CONDENSER - 16 IN * 10 FT, 4 - PASS, 247 GPM
C
      C(1)=-0.63802264E01
      C(2)=0.92256919
      C(3)=-0.35229891
      C(4)=0.96151268E-02
      C(5)=-0.39001451E-02
      C(6)=-0.70291588E-02
      CAPC(1)=28.93710291
      CAPC(2)=0.87119772
      CAPC(3)=6.418343422E-03
      CONTC(1)=85.32847492
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      CONTC(2)=0.3667074
      CONTC(3)=4.80712905E-04
      BHPCPM=0.75
      IRSYSC=50
      RETURN
25    CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 25  TON
C     CONDENSER - 12 IN * 10 FT, 4 - PASS, 247 GPM
C
      C(1)=-0.3245533E01
      C(2)=0.4319265
      C(3)=-0.1799345
      C(4)=0.4969407E-02
      C(5)=-0.2031468E-02
      C(6)=-0.3592901E-02
      CAPC(1)=14.33374
      CAPC(2)=0.432741
      CAPC(3)=3.533006E-03
      CONTC(1)=84.196325
      CONTC(2)=0.3232368
      CONTC(3)=9.405337E-04
      BHPCPM=0.5
      IRSYSC=25
      RETURN
      END
      SUBROUTINE EVPR(CAPEVP,BHPPE,CFMPE,IEVP)
C     ********************************************
C
C     THIS ROUTINE ASSIGNS THE CHARACTERISTIC VALUES OF THE EVAPORATOR
C     CHOOSEN. ALL EVP. ARE 10 ROWS DEEP
      IF(IEVP.GT.6)  IEVP=6
C
      GO TO (1,2,3,4,5,6)  , IEVP
5     CONTINUE
C
C     EVAPORATOR - 2.3 TON/TD, 2 HP/FAN, 1140 RPM
C
      CAPEVP=36.8
      BHPPE=6.0
      CFMPE=33000.0
      RETURN
4     CONTINUE
C
C     EVAPORATOR - 1.81 TON/TD, 3 HP/FAN, 1140 RPM
C
      CAPEVP=28.93
      BHPPE=6.0
      CFMPE=26300.0
      RETURN
3     CONTINUE
C
C     EVAPORATOR - 1.27TON/TD, 1.5 HP/FAN, 1140 RPM
C
      CAPEVP=20.27
      BHPPE=3.0
      CFMPE=17500.0
      RETURN
2     CONTINUE
C
C     EVAPORATOR - 0.91 TON/TD, 1.0 HP/FAN, 1140 RPM
C
      CAPEVP=14.53
      BHPPE=2.0
      CFMPE=12600.0
      RETURN
1     CONTINUE
C
C     EVAPORATOR - 0.67 TON/TD, 0.75 HP/FAN, 1140 RPM
C
      CAPEVP=10.67
      BHPPE=1.5
      CFMPE=3900.0
      RETURN
6     CONTINUE
      WRITE(6,110)
110   FORMAT(10X,'* * * NO EVPERATOR IS DEFINED * * * ')
      STOP
      END
```

```
C
C COMPUTER PROGRAM FOR MODEL - A2
C ********************************
C
C   FOR MODEL - A2 TINF HAS TO BE KEPT CONSTANT
C   FOR VARIABLE NAMES REFER TO COMPUTER PROGRAM FOR MODEL - A3
C   THIS ROUTINE IS USED FOR EVALUATING THE EQUATION 3.11
C   WITH AN ASSUMED VARIATION OF TINF ALONG THE CONVEYOR LENGTH
C   THIS ROUTINE IS CHECKED USING THE ROUTINE DEVELOPED FOR EQN. 3.16
C   AND HEISLER CHARTS
C
      DIMENSION  B(10),BL(10),TLXTH(21),TLPXTH(21)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,TIME
110   FORMAT(5X,'BIOT=',F8.5,5X,'B=',10F10.3)
120   FORMAT(5X, 8F15.5)
130    FORMAT(1H1)
      NR=6
      XBI=3.0
      TIMET=2.0
      HC=30.0
      DTINF=10.0
      TINF=-20.0
      XBF=XBI/12.0
      NBS=10
      NBSP1=NBS+1
      DXB=XBF/NBS
      BK=0.28
      ALPHA=0.005
      BTI=100.0
      HCT=0.0
      DO 35 INF=1,4
      TIME=TIMET
      ITFL=0
      WRITE(6,130)
      DTIME=0.2
      TINF=TINF+DTINF
      DO 10 I=1,NBSP1
      TLXTH(I)=BTI
10    CONTINUE
      DO 20 NT=1,200
      IF(HC.EQ.HCT)  GO TO 40
      INIT=0
      BIOT=(HC*XBF)/BK
      CALL BN(BIOT,BL,NR)
      DO 5 J=1,NR
      B(J)=BL(J)/XBF
5     CONTINUE
      WRITE(6,110)  BIOT,B
40    CONTINUE
      TIME=TIME+DTIME
      CALL TEMPR(TLXTH,TINF,B,BL,TLPXTH)
      HCT=HC
      INIT=1
      TAV=0.0
      DO 30 J=1,NBSP1
      TAV=TAV+TLPXTH(J)
30    CONTINUE
      TAV=(TAV-(.5*(TLPXTH(1)+TLPXTH(NBSP1))))/NBS
      IF(TAV.GT.43.0)  GO TO 20
      ITFL=ITFL+1
      IF(ITFL.EQ.1)  TIMET=TIME
      IF(TAV.LT.28.0)  GO TO 35
      FO=ALPHA*TIME/(XBF*XBF)
      QR=(BTI-TAV)/(BTI-TINF)
      DTIME=0.02
      IF(TAV.LT.38.0.AND.TAV.GT.32.0)  DTIME=0.1
      WRITE(6,120)  TIME,TINF,TAV,HC,XBI,QR,FO,BIOT
20    CONTINUE
35    CONTINUE
      STOP
      END
      SUBROUTINE TEMPR(TLXTH,TINF,B,BL,TLPXTH)
C     ******************************************
C
C     THIS ROUTINE COMPUTES THE TEMPERATURE AT EVERY SECTION IXB WHICH
C     VARIES FROM 1 AT THE CENTER TO NBSP1 AT THE OUTER BOUNDARY OF
C     THE 1 - D SOLID
C   THIS ROUTINE IS BASED ON EQN. 3.11
C
C  INPUTS :
C  * * * * * *
C   ALPHA  =SOLID THERMAL DIFFUSIVITY (FT**2/HR)
C   B(N)   =B   IN (BL)*(TAN(BL))=C EQUATION   (N=1,..NR)
C   BIOT   =BIOT NUMBER
C   BK     =SOLID THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C   BL(N)  =B*L IN (BL)*(TAN(BL))=C EQUATION  (N=1,..NR)
C   BTI    =SOLID INITIAL TEMPERATURE (F)
C   DTIME  =TIME AT EACH SECTION ON CONVEYOR (HR)
C   DXB    =INCRIMENTAL LENGTH IN THE SOLID
C   INIT   = 0 FOR FOR NEW BIOT NUMBER CASE, =1 FOR SAME BIOT NUMBER
C   NBS    =NUMBER OF SECTIONS IN THE SOLID
C   NCS    =NUMBER OF SECTIONS ON THE CONVEYOR
C   NR     = NUMBER OF ROOTS OF TRANSCENDENTAL EQN. REQUIRED
C   TINF   =COOLING FLUID TEMPERATURE AT THE PRESENT SECTION ON CONVEYOR
C
C  OUTPUT :
C  * * * * * *
C   TLPXTH(N) =AT THE END OF THE PRESENT SECTION ON THE CONVEYOR (F)
C   WHERE N=1 TO NBSP1
C
      DIMENSION B(1),TLXTH(1),TLPXTH(1),BL(1)
      DIMENSION EXPT(10),CSBNX(21,10),FX(10),FXT(21),CONSTR(10)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,DTIME
      IF(INIT.GT.0)  GO TO 10
      DO 20 NBL=1,NR
      XB=0.0
      DO 30 IXB=1,NBSP1
      CSBNX(IXB,NBL)=COS(B(NBL)*XB)
      XB=XB+DXB
30    CONTINUE
      CONST1=BL(NBL)*BL(NBL)
      CONST2=(BIOT*BIOT)+CONST1+BIOT
      CONSTR(NBL)=CONST1/CONST2
20    CONTINUE
10    CONTINUE
      DO 40 NBL=1,NR
      EXPP=-ALPHA*B(NBL)*B(NBL)*DTIME
```

```
      EXPT(NBL)=0.0
      IF(EXPP.LT.-10.0)  GO TO 40
      EXPT(NBL)=EXP(EXPP)
40    CONTINUE
      DO 50 NBL=1,NR
      DO 60 IXB=1,NBSP1
      FXT(IXB)=(TLXTH(IXB)-TINF)*CSBNX(IXB,NBL)
60    CONTINUE
      FX(NBL)=(FXT(1)+FXT(NBSP1))/2.0
      DO 70 IXB=2,NBS
      FX(NBL)=FX(NBL)+FXT(IXB)
70    CONTINUE
      FX(NBL)=FX(NBL)*DXB/XBF
50    CONTINUE
      DO 100 IXB=1,NBSP1
      DUM1=0.0
      DO 110 NBL=1,NR
      DUM1=DUM1+(EXPT(NBL)*CONSTR(NBL)*CSBNX(IXB,NBL)*FX(NBL)/
     $ (CSBNX(NBSP1,NBL)*CSBNX(NBSP1,NBL)))
110   CONTINUE
      TLPXTH(IXB)=TINF+(2.0*DUM1)
100   CONTINUE
      RETURN
      END
      SUBROUTINE BN(C,B,NR)
C     ********************************************
C
C     COMPUTATIONS OF THE ROOTS OF TRANSCENDENTAL EIGEN CONDITIONS FOR T
C     EQUATION  ***  B(N)*TAN(B(N))=C  ***
C     WHERE  N = 1,NR      NR - THE NUMBER OF ROOTS REQUIRED
C     B(N) IN THIS ROUTINE IS SAME AS THE BL(N) INTHE MAIN PROGRAM
C
      DIMENSION B(1)
      DATA  PI/3.141593/,ERROR/1.0E-05/
      N=1
      B(1)=PI/SQRT((8.0/C)+4.0)
      NIL=1
      IF(C.GE.2.0)  GO TO 30
      NIL=2
      DO 10 IN=1,100
      TB1=TAN(B(1))
      B1=C/TB1
      IF(ABS(B(1)-B1).LE.ERROR)  GO TO 30
      B(1)=(B(1)+B1)/2.0
10    CONTINUE
20    WRITE(6,110)  N
110   FORMAT(10X,'DID NOT CONVERGE IN THE 100  ITERATIONS OF THE ROOT',
     $I5)
      RETURN
30    CONTINUE
      DO 40 N=NIL,NR
      BNP1=B(1)
      DO 50 IN=1,100
      BNP2=ATAN(C/(((N-1)*PI)+BNP1))
      IF(ABS(BNP2-BNP1).LE.ERROR)  GO TO 60
      BNP1=BNP2
50    CONTINUE
      GO TO 20
60    B(N)=((N-1)*PI)+BNP1
40    CONTINUE
      RETURN
      END
```

```
C
C COMPUTER PROGRAM FOR MODEL - A2
C ********************************
C
C   FOR MODEL - A2 TINF HAS TO BE KEPT CONSTANT
C   FOR VARIABLE NAMES REFER TO COMPUTER PROGRAM FOR MODEL - A3
C   THIS ROUTINE IS USED FOR EVALUATING THE EQUATIONS 3.16 AND 3.17
C   WITH AN ASSUMED VARIATION OF TINF ALONG THE CONVEYOR LENGTH.
C   IT CAN ALSO BE USED FOR CHECKING EQUATION 3.8
C   THIS ROUTINE IS CHECKED USING THE ROUTINE DEVELOPED FOR EQN. 3.11
C   AND HEISLER CHARTS
C
      DIMENSION  B(10),BL(10),TINF(10), TLPXTH(21)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,BTI,DTIME(20)
110   FORMAT(5X,'BIOT=',F8.5,5X,'B=',10F10.3)
120   FORMAT(5X,14F8.2)
130   FORMAT(10X,'TIME=',F5.1,10X,'TINF=',F6.2)
      NR=6
      XBI=2.0
      XBF=XBI/12.0
      NBS=10
      NBSP1=NBS+1
      NCS=10
      NCSP1=NCS+1
      DXB=XBF/NBS
      BK=0.28
      HCT=0.0
      TIME=0.0
      ALPHA=0.005
      BTI=100.0
      DO 20 NB=1,NCS
      HC=5.0
      IF(HC.EQ.HCT)  GO TO 40
      INIT=0
      BIOT=(HC*XBF)/BK
      CALL BN(BIOT,BL,NR)
      DO 5 J=1,NR
      B(J)=BL(J)/XBF
5     CONTINUE
      WRITE(6,110)  BIOT,B
40    CONTINUE
      XNB=(NB-1)
      DTIME(NB)=0.1*NB
      TINF(NB)=25.0-XNB
      TIME=TIME+DTIME(NB)
      WRITE(6,130)   TIME,TINF(NB)
      CALL TEMPR(NB,TINF,B,BL,TLPXTH,TAVE)
      INIT=1
      TAV=0.0
      DO 30 J=1,NBSP1
      TAV=TAV+TLPXTH(J)
30    CONTINUE
      TAV=(TAV-(.5*(TLPXTH(1)+TLPXTH(NBSP1))))/NBS
      WRITE(6,120)  (TLPXTH(J),J=1,NBSP1),TAV,TAVE
      HCT=HC
20    CONTINUE
      STOP
      END
      SUBROUTINE TEMPR(NB,TINF,B,BL,TLPXTH,TAVE)
C     ******************************************
C
C     THIS ROUTINE COMPUTES THE TEMPERATURE AT EVERY SECTION IXB WHICH
C     VARIES FROM 1 AT THE CENTER TO NBSP1 AT THE OUTER BOUNDARY OF
C     THE 1 - D SOLID
C   THIS ROUTINE IS BASED ON EQN. 3.16
C   THIS ROUTINE ALSO COMPUTES THE AVERAGE TEMPERATURE OF THE SOLID
C   USING EQN. 3.17
C
C INPUTS :
C * * * * * *
C   ALPHA  =SOLID THERMAL DIFFUSIVITY (FT**2/HR)
C   B(N)   =B   IN (BL)*(TAN(BL))=C EQUATION    (N=1,..NR)
C   BIOT   =BIOT NUMBER
C   BK     =SOLID THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C   BL(N)  =B*L IN (BL)*(TAN(BL))=C EQUATION   (N=1,..NR)
C   BTI    =SOLID INITIAL TEMPERATURE (F)
C   DTIME  =TIME AT EACH SECTION ON CONVEYOR (HR)
C   DXB    =INCRIMENTAL LENGTH IN THE SOLID
C   INIT   = 0 FOR FOR NEW BIOT NUMBER CASE, =1 FOR SAME BIOT NUMBER
C   NB     =SECTION NUMBER ON THE CONVEYOR
C   NBS    =NUMBER OF SECTIONS IN THE SOLID
C   NCS    =NUMBER OF SECTIONS ON THE CONVEYOR
C   NR     = NUMBER OF ROOTS OF TRANSCENDENTAL EQN. REQUIRED
C   TINF(N)=COOLING FLUID TEMPERATURE   (N=1,..NCS)
C
C OUTPUT :
C * * * * * *
C   TLPXTH(N) =AT THE END OF THE PRESENT SECTION ON THE CONVEYOR (F)
C   WHERE N=1 TO NBSP1
C   TAVE   =AVERAGE TEMPERATURE OF THE SOLID
C
      DIMENSION B(1),BL(1),TINF(1),TLPXTH(1)
      DIMENSION EXPT(10,10),CSBNX(21,10),CONSTR(10),DTINF(10),AB2(10)
      DIMENSION  BL2(10)
      COMMON/B1/NBS,NBSP1,INIT,NR
      COMMON/B2/XBF,DXB,BIOT,ALPHA,BTI,DTIME(20)
      IF(INIT.GT.0)  GO TO 10
      DO 20 NBL=1,NR
      XB=0.0
      DO 30 IXB=1,NBSP1
      CSBNX(IXB,NBL)=COS(B(NBL)*XB)
      XB=XB+DXB
30    CONTINUE
      CONST1=BL(NBL)*BL(NBL)
      BL2(NBL)=CONST1
      CONST2=(BIOT*BIOT)+CONST1+BIOT
      CONSTR(NBL)=1.0/CONST2
      AB2(NBL)=ALPHA*B(NBL)*B(NBL)
20    CONTINUE
10    CONTINUE
      DTINF(NB)=BTI-TINF(1)
      DO 50 NS=1,NB
      DT=0.0
      DO 60 K=1,NS
      DT=DT+DTIME(NB-K+1)
60    CONTINUE
      IF(NS.EQ.NB)  GO TO 70
      DTINF(NS)=TINF(NB-NS)-TINF(NB-NS+1)
```

```
70    CONTINUE
      DO 40 NBL=1,NR
      EXPP=-AB2(NBL)*DT
      EXPT(NS,NBL)=0.0
      IF(EXPP.LT.-10.0)  GO TO 40
      EXPT(NS,NBL)=EXP(EXPP)
40    CONTINUE
50    CONTINUE
      DO 100 IXB=1,NBSP1
      DUM2=0.0
      DO 110 NS=1,NB
      DUM1=0.0
      DO 120 NBL=1,NR
      DUM1=DUM1+(EXPT(NS,NBL)*CONSTR(NBL)*CSBNX(IXB,NBL)/CSBNX(NBSP1,NBL
     $))
120   CONTINUE
      DUM2=DUM2+(2.0*BIOT*DTINF(NS)*DUM1)
110   CONTINUE
      TLPXTH(IXB)=TINF(NB)+DUM2
100   CONTINUE
      DUMA2=0.0
      DO 130 NS=1,NB
      DUMA1=0.0
      DO 140 NBL=1,NR
      DUMA1=DUMA1+(EXPT(NS,NBL)*CONSTR(NBL)/BL2(NBL))
140   CONTINUE
      DUMA2=DUMA2+(2.0*BIOT*BIOT*DTINF(NS)*DUMA1)
130   CONTINUE
      TAVE=TINF(NB)+DUMA2
      RETURN
      END
      SUBROUTINE BN(C,B,NR)
C     ******************************************
C
C     COMPUTATIONS OF THE ROOTS OF TRANSCENDENTAL EIGEN CONDITIONS FOR T
C     EQUATION  ***  B(N)*TAN(B(N))=C  ***
C     WHERE  N = 1,NR     NR - THE NUMBER OF ROOTS REQUIRED
C     B(N) IN THIS ROUTINE IS SAME AS THE BL(N) INTHE MAIN PROGRAM
C
      DIMENSION B(1)
      DATA  PI/3.141593/,ERROR/1.0E-05/
      N=1
      B(1)=PI/SQRT((8.0/C)+4.0)
      NIL=1
      IF(C.GE.2.0)  GO TO 30
      NIL=2
      DO 10 IN=1,100
      TB1=TAN(B(1))
      B1=C/TB1
      IF(ABS(B(1)-B1).LE.ERROR)  GO TO 30
      B(1)=(B(1)+B1)/2.0
10    CONTINUE
20    WRITE(6,110)  N
110   FORMAT(10X,'DID NOT CONVERGE IN THE 100  ITERATIONS OF THE ROOT',
     $I5)
      RETURN
30    CONTINUE
      DO 40 N=NIL,NR
      BNP1=B(1)
      DO 50 IN=1,100
      BNP2=ATAN(C/(((N-1)*PI)+BNP1))
      IF(ABS(BNP2-BNP1).LE.ERROR)  GO TO 60
      BNP1=BNP2
50    CONTINUE
      GO TO 20
60    B(N)=((N-1)*PI)+BNP1
40    CONTINUE
      RETURN
      END
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C
C COMPUTER PROGRAM FOR MODEL - A1
C ********************************
C
C   FOR VARIABLE NAMES REFER TO COMPUTER PROGRAM FOR MODEL - A3
C   THIS ROUTINE USES THE HEAT EXCHANGER THEORY FOR COMPUTING THE MASS
C   AVERAGE TEMPERATURES AND ASSUMES NO INTERNAL RESISTANCE IN THE SOLID
C
C
C     AIR PROPERTIES FOR -40.0 . LE. T . LE . 100.0 F
C
      ARO(T)=(519.0*0.0765)/(460.0+T)
      AMU(T)=1.11E-05+(1.71875E-08*T)
      ANU(T)=0.13E-03+(4.6875E-07*T)
      AK(T)=0.0133+(2.1875E-05*T)
      ACP=0.24
C   ATI     =AIR TEMPERATURE ON TO THE CONVEYOR
      ATI=0.0
      CH=2.0
      BTI=100.0
      BPHI=4.0
      BTF=30.0
      BPLF=1.0
C
C     BEEF PROPERTIES
C
      BRO=65.0
      BCP=0.85
      BK=0.28
      CW=1.0
      CL=100.0
      WRITE(6,100)  BPHI,ATI,CH,CL
      WRITE(6,110)  BTI,BTF,BPLF
      BPHF=BPHI/12.0
      AFH=CH-BPHF
      BPHF=BPHI/12.0
      DO 10 IV=1,5
      VEL=IV*10.0
      AV=VEL
      ATO=ATI
      DO 20 IT=1,20
      DO 15 ITT=1,2
      T=(ATI+ATO)/2.0
      HC=36.334*((ARO(T)*VEL)**0.8)*((AMU(T)/BPLF)**0.2)
      R=(1.0/HC)+(1.0/550)
      U=1.0/R
      XI=IT*0.5
      TIME=XI
      CV=CL/(3600.0*XI)
      BV=CV
      BMFR=CW*BPHF*BV*BRO*3600.0
      BMC=BMFR*BCP
      AMFR=CW*AFH*AV*ARO(T)*3600.0
      AMC=AMFR*ACP
      CMIN=AMC
      CMAX=BMC
      IFLAG=0
      IF(AMC.LT.BMC)  GO TO 30
      IFLAG=1
      CMIN=BMC
      CMAX=AMC
   30 CONTINUE
      ARAB=2.0*CL*CW
      C=CMIN/CMAX
      XNTU=ARAB*U/CMIN
      TEMP=-XNTU*(1.0-C)
      EPSILN=(1.0-EXP(TEMP))/(1.0-(C*EXP(TEMP)))
      IF(IFLAG.EQ.1)  GO TO 40
      ADT=EPSILN*(BTI-ATI)
      ATO=ADT+ATI
      DQ=AMC*ADT
      BTO=BTI-(DQ/BMC)
      GO TO 50
   40 BDT=EPSILN*(BTI-ATI)
      BTO=BTI-BDT
      DQ=BMC*BDT
      ATO=ATI+(DQ/AMC)
   50 CONTINUE
   15 CONTINUE
      WRITE(6,115)  TIME,IFLAG
      WRITE(6,120) AV,HC,DQ,AMFR,BMFR,ATO,BTO
   20 CONTINUE
   10 CONTINUE
  100 FORMAT(1H1,/,10X,'BEEF THICKNESS = ',F6.2,2X,'IN'
     *,/,10X,'AIR INLET TEMPERATURE ON TO CONV. =',F6.1,2X,'F'
     *,/,10X,'CONV. HEIGHT =',F5.1,2X,'FT'
     *,/,10X,'CONV. LENGTH =',F6.1,2X,'FT'
     *,/)
  110 FORMAT(10X,'BEEF INITIAL TEMPERATURE =',F6.1,2X,'F',/
     *,10X,'BEEF FINAL TEMPERATURE REQUIRED =',F6.1,2X,'F',/
     *,10X,'BEEF PIECE CHARACTERISTIC LENGTH =',F6.1,2X,'FT',/)
  115 FORMAT(/,10X,'TIME =',F6.2,2X,'HR',10X,'IFLAG =',I2)
  120 FORMAT(/,10X,'AIR VEL. =',F5.1,2X,'FT/SEC',
     *14X,'HEAT TRANS. COEF. =',F6.2,2X,'BTU/HR-FT**2-F',
     *10X,'DQ =',F8.0,2X,'BTU/HR',
     */,10X,'AIR MASS FLOW RT. =',F8.0,'LBM/HR',
     * 7X, 'BEEF MASS FLOW RT. =',F8.0,'LBM/HR',
     */,10X,'AVE. AIR TEMP. OUT = ',F6.1,2X,'F',
     * 10X,'AVE. BEEF TEMP. OUT = ',F6.1,2X,'F',
     */)
      STOP
      END
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C
C COMPUTER PROGRAM FOR MODEL - B2
C ********************************
C
C   THIS PROGRAM COMPUTES ALL THE NECESSARY VALUES FOR DEVELOPING THE
C   OPTIMUM CHARACTERISTICS FOR DESIGN OF THE CONVEYARISED FREEZING
C   SYSTEM OF ANY 1-D SOLIDS.
C   THE PRESENT EXAMPLE IS SET FOR HOT BONED BEEF
C
C  INPUTS :
C  * * * * * *
C
C   INPUTS ARE IN NAMELIST FORM, THE VALUES IN THE PARENTHESIS ARE
C   THE DEFAULT VALUES
C   CARD(S)-1
C   /NAME1/
C   BTI    =BEEF INITIAL TEMPERATURE (45.0 F)
C    DHF=ENTHALPY REDUCTION REQUIRED (120 BTU/LBM)
C   BPLF   =BEEF PIECE AVERAGE LENGTH (1.0 FT)
C   BMFR   =BEEF MASS FLOW RATE (5000.0 LBM/HR)
C   BPHI   =BEEF PIECE THICKNESS (4.0 IN)
C   NV     =NUMBER OF REFRIGERATION SYSTEMS TO BE STUDIED (1)
C   NBS    =NUMBER OF SECTIONS IN THE SOLID (10)
C   IXPRT  =X-DIRECTION PRINT REQUIRED AT INTERVALS OF IXPRT
C   DTM    =TIME STEP IN MINUTES (1 MIN)
C   CARD(S)-2
C   /NAME2/
C   TIMET  =ESTIMATED TOTAL FREEZING TIME (6.0*BPHI HRS)
C   EVPEAT =ESTIMATED EVAPORATOR ENTERING AIR TEMPERATURE (30.0 F)
C   CNCWT  =CONDENSER COOLING WATER TEMPERATURE FOR PRINTING (75.0 F)
C   ICU    =CONDENSING UNIT SELECTED
C   IEVP   =EVEPORATOR UNIT SELECTED
C   NUEVP  =NUMBER OF EVAPORATORS SELECTED
C
C  OUTPUT :
C  * * * * * *
C
C   DHF    =ENTHALPY DROP OF EACH NODE IN SOLID FROM CENTER TO BOUNDARY
C   TEMP   =TEMPERATURE DISTRIBUTON IN THE SOLID FROM CENTER TO BOUNDARY
C   THE REST IS SELF EXPLANITARY
C
C  SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
C  * * * * * * * * * * * * * * * * * * * * * * * *
C
C    SUBROUTINE FREEZ(TDHTA,DHDT,TINF,DT)
C    SUBROUTINE TRIDAG
C    SUBROUTINE REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,IRSYS,ICE)
C    SUBROUTINE CONDU(C,CAPC,CONTC,BHPCPM,IRSYSC,ICU)
C    SUBROUTINE EVPR(CAPEVP,BHPPE,CFMPE,IEVP)
C    FUNCTION CP(TF)
C    FUNCTION BK(TF)
C
C  KEY TO SYMBOLS
C  * * * * * * * *
C
C   ACP    =AIR SPECIFIC HEAT (BTU/LBM-F)
C   AK     =AIR THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C   AMC    =AIR CAPACITY RATE (BTU/HR-F)
C   AMU    =AIR ABSOLUTE VISCOSITY (LBM/FT-SEC)
C   ANU    =AIR KINEMATIC VISCOSITY (FT**2/SEC)
C   ARO    =AIR DENSITY (LBM/FT**3)
C   BCPA   =BEEF AVERAGE SPECIFIC HEAT (BTU/LBM-F)
C   BHPCMM =BHP INPUT TO COMPRESSOR MOTOR
C   BHPCOM =BHP INPUT TO COMPRESSOR
C   BHPCPM =BHP INPUT TO CONDENSER PUMP MOTOR
C   BHPEFM =BHP INPUT TO EVAPORATOR FAN MOTOR
C   BHPPE  =BHP INPUT TO EVAPORATO FAN
C   BHPT   =TOTAL BHP INPUT TO REFRIGERATION SYSTEM
C   BHPT1  =TOTAL BHP INPUT TO THE SYSTEM INCLUDING CONVEYOR FAN
C   BHPTPT =TOTAL BHP PER TON TO THE SYSTEM
C   BIOT   =BIOT NUMBER
C   BK     =BEEF THERMAL CONDUCTIVITY (BTU/HR-FT-F)
C   BLF    =BEEF LOADING FACTOR (LBM/FT**2)
C   BMCA   =BEEF AVERAGE CAPACITY RATE (BTU/HR-F)
C   BMFR   =BEEF MASS FLOW RATE (LBM/HR)
C   BPHI   =BEEF PIECE THICKNESS (IN)
C   BPHF   =BEEF PIECE THICKNESS (FT)
C   BPLF   =BEEF PIECE AVERAGE LENGTH (FT)
C   BRO    =BEEF DENSITY (LBM/FT**3)
C   BTI    =BEEF INITIAL TEMPERATURE (F)
C   BTF    =BEEF FINAL AVE. TEMPR. REQUIRED (F)
C   C(N)   =CONSTANTS FOR COMPRESSOR INPUT CALCULATION (N=1,..6)
C   CAPC(N)=CONSTANTS FOR THE CAPACITY OF THE SYSTEM (N=1,2,3)
C   CAPEVP =CAPACITY OF EVEPARATOR IN TONS FOR 1L F DROP IN TEMPERATURE
C   CAR    =CONVEYOR AREA (FT**2)
C   CART   =CONVEYOR TOTAL AREA (FT**2)
C   CFMPE  =CFM PER EVAPORATOR
C   CFMPT  =CFM OF AIR PER TON OF COOLING LOAD
C   CH     =CONVEYOR HEIGHT (FT)
C   CL     =CONVEYOR LENGTH (FT)
C   CLTWR  =CONVEYOR LENGTH TO WIDTH RATIO (CL/CW)
C   CNCWT  =CONDENSER COOLING WATER INLET TEMPERATURE (F)
C   CONT   =CONDENSING TEMPERATURE OF THE REFRIGERANT (F)
C   CONTC(N)=CONSTANTS FOR CONDENSING TEMPERATURE OF THE SYSTEM (N=1,2,3
C   CVEL   =CONVEYOR VELOSITY (FT/HR)
C   CW     =CONVEYOR WIDTH (FT)
C   DEQ    =HYDRAULIC DIAMETER OF THE DUCTWORK AND THE CONVEYOR (FT)
C   DHDT   =ENTHALPY DROP AT A GIVEN TIME INTERVAL (BTU/LBM)
C   DHF    =TOTAL ENTHALPY REDUCTION REQUIRED (BTU/LBM)
C   DPC    =PRESSURE DROP OF AIR ON CONVEYOR IN IN. OF WATER
C   DPCON  =PRESSURE DROP IN THE CONNECTING DUCT WOTK IN IN. OF WATER
C   DPCPSI =PRESSURE DROP OF AIR ON CONVEYOR IN PSI
C   DPTOT  =TOTAL PRESSURE DROP OF AIR ON COVEYORIN IN. OF WATER
C   DT     =TIME AT EACH SECTION ON CONVEYOR (TIME STEP) (HR)
C   DXB    =INCREMENTAL LENGTH IN BEEF PIECE (FT)
C   EFCM   =EFFICIENCY OF COMPRESSOR MOTOR
C   EFEFM  =EFFICIENCY OF CIRCULATIG FAN AND ITS MOTOR
C   EVPEAT =EVEPERATOR ENTERING AIR TEMPERATURE (F)
C   EVPLAT =EVEPERATOR LEAVING  AIR TEMPERATURE (F)
C   EVPT   =EVEPORATING TEMPERATURE OF THE REFRIGERANT (F)
C   FR     =FRICTION FACTOR
C   HC     =HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-F)
C   HCP    =(K/THICKNESS) FOR PLASTIC BAG (BTU/HR-FT**2-F)
C   HFMP   =ENTHALPY OF BEEF AT THE GIVEN TIME INTERVAL
C              (H=0.0 AT T=-40 F)  (BTU/LBM)
C   ICU    =CONDENSING UNIT SELECTED (1-25TON,2-50TON,3-75TON,4-100TON
C              AT APPROX. 20F EVPT. AND 95F CONT. TEMPERATURES)
C   IEVP   =EVAPORATOR UNIT SELECTED  (1-10.67TON,2-14.53TON,3-20.27TON,
C              4-28.93TON,5-36.8TON, FOR A 16F TEMPR. DROP)
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```
C     NBS    =NUMBER OF SECTIONS IN THE BEEF PIECE (10)
C     NCS    =NUMBER OF SECTIONS ON THE CONVEYOR
C     NR     = NUMBER OF ROOTS OF TRANSCENDENTAL EQN. REQUIRED
C     NUEVP  =NUMBER OF EVAPORATORS SELECTED
C     RCAP   =REQUIRED COOLING CAPACITY (TONS)
C     RMC    =RATIO OF HEAT CAPACITY FLOW RATES OF THE TWO STREAMS
C     TAV    =AVERAGE TEMP. OF BEEF PIECE AT SECTION (N+1) ON THE CONVEYOR
C     TAVI   =AVERAGE TEMP. OF BEEF PIECE AT SECTION N ON THE CONVEYOR
C     TDHTA  =TOTAL ENTHALPY DROP FROM BEGINING TO PRESENT TIME(BTU/LBM)
C     TIME   =TOTAL TIME(HR)
C     TF     =TEMPERATURE (F)
C     TINF   =AIR TEMPERATURE (F)
C     TLPXTH =TEMP. AT SEC. X IN BEEF PIECE AND AT SEC. (N+1) ON CONV.
C     TLXTH  =TEMP. AT SEC. X IN BEEF PIECE AND AT SEC. N ON CONV.
C     U      =OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-F)
C     VEL    =VELOCITY OF AIR ON CONVEYOR (FT/SEC.)
C     XBI    =HALF THICKNESS OF BEEF PIECE (IN)
C     XBF    =HALF THICKNESS OF BEEF PIECE (FT)
C
      COMMON/F1/NBS,NBSP1,INIT
      COMMON/F2/XBF,DXB,BTI,BRO,HTCOF
      COMMON/F3/HFM(33),TDH(33),TLXTH(33)
      COMMON/BR/EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
      NAMELIST /NAME1/BTI,DHF,BPLF,BMFR,BPHI,NV,NBS,IXPRT,DTM
     $ /NAME2/ TIMET,EVPEAT,CNCWT,ICU,IEVP,NUEVP
C
C     AIR PROPERTIES FOR -40.0 . LE. T . LE . 100.0 F
C
      ARO(T)=(519.0*0.0765)/(460.0+T)
      AMU(T)=1.11E-05+(1.71875E-03*T)
      ANU(T)=0.13E-03+(4.6875E-07*T)
      AK(T)=0.0133+(2.1875E-05*T)
      ACP=0.24
C
C
      BRO=65.0
      BTI=45.0
      DHF=120.0
      BPLF=1.0
      BMFR=5000.0
      BPHI =4.0
      DTM=1.0
      IXPRT=1
      NBS=10
      NV=1
      CNCWT=75.0
      READ(5,NAME1)
      LXP1=NBS+1
      BLF=BPHI*BRO/12.0
      BPHF=BPHI/12.0
      XBI=BPHI/2.0
      XBF=XBI/12.0
C     FINDING THE AVE. FINAL TEMPR. FOR THE GIVEN INITIAL TEMPR. AND
C       ENTHALPY DROP
      HFMP=(71.6*1.8)+(0.85*(BTI-32.0))
      HFMM=HFMP-DHF
      TF=-40.0
      DHMT=-CP(TF)*0.1
      DO 5 I=1,72
      DO 5 J=1,10
      TFP=(J-1)*0.1+(I-1)
      TFM=-40.0+TFP
      CPM=CP(TFM)
      DHM=CPM*0.1
      DHMT=DHMT+DHM
      IF(DHMT.GE.HFMM)  GO TO 6
    5 CONTINUE
      WRITE(6,215)
      STOP
    6 CONTINUE
      BCPA=DHF/(BTI-TFM)
      BMCA=BMFR*BCPA
      DO 8  IX=1,LXP1
      TLXTH(IX)=BTI
      HFM(IX)=HFMP
    8 CONTINUE
      NBSP1=NBS+1
      DXB=XBF/NBS
      WRITE(6,210)
      WRITE(6,220)  BPHI,BTI,BRO,BLF,BMFR,DHF,TFM,BCPA
      WRITE(6,225)  (HFM(J),J=1,LXP1,IXPRT)
      WRITE(6,330)
      WRITE(6,NAME1)
C
C     CONVEYOR
C
      CLTWR=4.0
      CAR=BMFR/BLF
C
C     SYSTEM
C
      HCP=550.0
      DTIME=DTM/60.0
      RCAP=BMFR*DHF/12000.0
      FANEFF=0.7
C
C     INITIAL ESTIMATE OF EVPEAT AND TIMET
C
      EVPEAT=0.0
      TIMET=BPHI*6.0
C     LOOP 10 IS FOR NUMBER OF REFRIGERATION SYSTEMS TO BE STUDIED
      DO 10 IV=1,NV
      READ(5,NAME2)
      DO 15 IVEL=1,5
      IPAGE=IVEL/2
      IPAGE2=IPAGE*2
      IF(IPAGE2.NE.IVEL)  WRITE(6,210)
      IF(IVEL.EQ.1)  WRITE(6,NAME2)
      WRITE(6,320)
      VEL=IVEL*10
      IVT=0
      NCS=(TIMET/DTIME)+1
      IEC=0
      INIT=0
   26 CONTINUE
      IEC=IEC+1
C
      CALL REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,IEC)
C
      T=EVPEAT
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      AMC=CFM*ARO(T)*60.0*ACP
      EVPEAT=EVPLAT+(RCAP*12000.0/AMC)
      IECF=1
      IF(IEC.GT.100)  GO TO 25
      IECF=0
      IF(ABS((RCAP-SYSCAP)/RCAP).GT.0.005) GO TO 26
      IVT=1
      CFMPT=CFM/RCAP
25    CONTINUE
      IF(IECF.EQ.1)  WRITE(6,250)
      IF(IECF.EQ.1)  GO TO 10
      T=EVPEAT
      AMC=CFM*ARO(T)*60.0*ACP
      RMCA=BMCA/AMC
      T=(EVPEAT+EVPLAT)/2.0
      HC=36.334*((ARO(T)*VEL)**0.8)*((AMU(T)/BPLF)**0.2)
      UI=(1.0/HC)+(1.0/HCP)
      U=1.0/UI
      HTCOF=U
      TIME=0.0
      TINF=EVPEAT
C     LOOP 40 IS FOR VARIOUS SECTIONS ON THE CONVEYOR WITH DT=DTIME/10
      DHDT=0.0
      DT=DTIME/10.0
      DO 40 N=1,20
      TIME=TIME+DT
      DHDTT=DHDT*BMFR
      TINF=TINF-(DHDTT/AMC)
      CALL  FREEZ(TDHTA,DHDT,TINF,DT)
      INIT=1
40    CONTINUE
C     LOOP 50 IS FOR VARIOUS SECTIONS ON THE CONVEYOR WITH DT=DTIME/2
      DT=DTIME/2.0
      DO 50 N=1,16
      TIME=TIME+DT
      DHDTT=DHDT*BMFR
      TINF=TINF-(DHDTT/AMC)
      CALL  FREEZ(TDHTA,DHDT,TINF,DT)
50    CONTINUE
C     LOOP 60 IS FOR VARIOUS SECTIONS ON THE CONVEYOR WITH DT=DTIME
      DT=DTIME
      DO 60 N=11,NCS
      TIME=TIME+DT
      DHDTT=DHDT*BMFR
      TINF=TINF-(DHDTT/AMC)
      CALL  FREEZ(TDHTA,DHDT,TINF,DT)
      IF(TDHTA.GE.DHF)   GO TO 19
60    CONTINUE
      WRITE(6,300)
19    CONTINUE
C
C     CONVEYOR SIZE CALCULATIONS
C
      CART=CAR*TIME
      CW=SQRT(CART/CLTWR)
      CL=CART/CW
      CVEL=CL/TIME
      CH=BPHF+(CFM/(CW*VEL*60.0))
      DEQ=(4.0*(CH-BPHF)*CW)/((4.0*CW)+(2.0*CH))
      WRITE(6,260) VEL, U,CH,DEQ
      WRITE(6,230)   RMCA,RCAP,CART,CW,CL,CVEL
      WRITE(6,310) EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
      WRITE(6,240)  ICU,IEVP,EVPEAT,EVPLAT,SYSCAP,CFM,BHPT,CFMPT
      WRITE(6,270) TIME,TINF,TDHTA
      WRITE(6,280)  (TLXTH(J),J=1,NBSP1,IXPRT)
      WRITE(6,285)   (TDH(J),J=1,NBSP1,IXPRT)
C     LOOP 90 IS FOR CALCULATING THE CIRCULATING FAN POWER REQUIRED
      FR=0.005
      DO 90 IFR=1,3
      DPCPSI=(FR*2.0*ARO(T)*VEL*VEL*CL)/ (DEQ*32.2*144.0)
      DPC=27.7*DPCPSI
      DPCON=2.0*0.3303*ARO(T)*((VEL*60.0/1000.0)**2)
      DPTOT=DPC+DPCON
      FANHP=(CFM*DPTOT)/(6350.0*FANEFF)
      BHPT1=BHPT+FANHP
      BHPTPT=BHPT1/SYSCAP
      RFTTP=(FANHP+BHPEFM)/BHPT1
      WRITE(6,290)  FR, DPC,DPCON,DPTOT,FANHP,BHPT1,BHPTPT,RFTTP
      FR=FR*2.0
90    CONTINUE
15    CONTINUE
10    CONTINUE
210   FORMAT(1H1)
215   FORMAT(5X,'*** CHECK INITIAL TEMP. AND ENTHALPY REDUCTION ***')
220   FORMAT(//,5X,'AVERAGE BEEF PIECE SPECIFICATIONS',//,
     1 10X,'AV. THICKNESS  =',F5.2,2X,'IN',/,
     2 10X,'AV. TEMP. IN   =',F6.1,2X,'F',/,
     3 10X,'DENSITYY       =',F5.1,2X,'LB/FT**3',/,
     4 10X,'LOADING FACT.  =',F5.1,2X,'LB/FT**2',/
     5 10X,'MASS FLOW RT.  =',F5.1,2X, 'LB/HR',/,
     6 10X,'ENTHALPY REDUC.=',F6.1,2X,'BTU/LBM',/,
     7 10X,'FINAL AVE. TEMP=',F6.1,2X,'F',/,
     8 10X,'APPRT. SP. HEAT=',F6.2,2X,'BTU/LBM-F',/)
225   FORMAT(//,10X,'ENTHALPY LEVEL OF EACH NODE WITH -40.0 F AS 0.0
     1BTU/LBM IS',//,10X,11(F6.2,4X),/)
230   FORMAT(10X,'RATIO OF MCPA      =',F5.3,
     1 14X,'REQUIRED CAPACITY  =',F5.1,2X,'TONS',/,
     3 10X,'TOTAL CONV. AREA   =',F5.0,2X,'FT**2',
     4  6X,'CONV. WEDITH       =',F5.1,2X,'FT',/,
     5 10X,'CONV. LENGTH       =',F6.1,2X,'FT',
     6  8X,'CONV. VELOCITY     =',F5.1,2X,'FT/HR')
240   FORMAT(10X,'ICU = ',I2,10X,'IEVP =',I2,/,
     2 10X,'EVP. ENT. AIR TEMP. =',F5.1,2X,'F',
     3 10X,'EVP. LEV. AIR TEMP. =',F5.1,2X,'F',/,
     4 10X,'SYSTEM CAPACITY       =',F6.1,2X,'TONS',
     5  6X, 'CFM     =',E12.4,2X,/,
     6 10X,'TOT. BHP OF REFRIG. SYS. =',F6.1,
     7  7X,'CFM/TON =',F6.0,/)
250   FORMAT(10X,'NOT CONVERGED IN SYSCAP   * * * * * '  )
260   FORMAT(5X,'VEL. OF AIR ON CONV. =',F5.1,'FT/SEC',
     1 10X,'HEAT TRANS. COEF.  =',F6.2,'BTU/HR-FT**2-F',/,
     2 10X,'CONV. HEIGHT          =',F5.1,'FT',
     3 11X,'EQ. DIAMETER          =',F5.1,'FT')
270   FORMAT(6X,'TIME=',F5.2,2X,'HR',10X,'TINF=',F6.1,2X,'F',
     1 10X,'AV. BEEF PIECE H RED. =',F6.1,2X,'BTU/LBM',/)
280   FORMAT(3X,'TEMP=',11(F6.2,4X),/)
285   FORMAT(3X,' DHF=',11(F6.2,4X),/)
290   FORMAT(8X,'FR=',F5.3,5X,'DP. CONV.  =',F4.1,2X,'IN H2O',
     1 10X,'DP. CONCT.  =',F4.1,2X,'IN H2O',
     2 10X,'DP. TOTAL   =',F4.1,2X,'IN H2O',/,
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      3 10X,'FAN HP      =',F5.1,2X,
      4 10X,'TOTAL HP    =',F6.1,2X,'HP',
      5 10X,'HP/TON      =',F6.4,
      6 5X,' FAN/TOTAL  =',F6.4,/)
 300  FORMAT(10X,'NOT CONVERGED IN TIME  * * * * *',
     1/,' * * * INCREASE TIMET IN INPUT  * * * * ')
 310  FORMAT(3X,'EVPT=',F6.2,4X,'CONT=',F6.2,4X,'BHPCMM=',F6.1,4X,
     $ 'BHPEFM=',F6.1,4X,'BHPCPM=',F6.2,4X,'NUEVP=',I2)
 320  FORMAT(/)
 330  FORMAT(//)
      STOP
      END
C
      SUBROUTINE FREEZ(TDHTA,DHDT,TINF,DT)
C     ****************************
C
      COMMON/F1/NBS,NBSP1,INIT
      COMMON/F2/XBF,DXB,BTI,BRO,HTCOF
      COMMON/F3/HFM(33),TDH(33),U(33)
      COMMON/TRIDA/LI,LXP1,A(33),B(33),C(33),D(33),V(33)
      DIMENSION DH(33),X(33),CPU(33),CPV(33)
      IF(INIT.GT.0)  GO TO 20
C     U=TEMPERATURE ARRAY AT (N-1) TH STEP
C     V=TEMPERATURE ARRAY AT N TH STEP
      LI=1
      X0=0.0
      TDHTAI=0.0
      LX=NBS
      DX=DXB
      LXP1=LX+1
      LXP2=LX+2
C    *** FINDING X-SECTIONS,U(X,0.0),ENTHALPY    ..... LOOP 10  .....
      DO 10 IX=1,LXP1
      X(IX)=X0
      U(IX)=BTI
      CPU(IX)=CP(U(IX))
      TDH(IX)=0.0
      X0=X0+DX
  10  CONTINUE
      CBL=HTCOF*2.0*DX/BK(U(LXP1))
      U(LXP2)=U(LX)-(CBL*(U(LXP1)-TINF))
  20  CONTINUE
      P=DT/(2.0*DX*DX)
      TDHT=0.0
C    *** LOOP 40 IS FOR CALCULATION OF COEFF     @@@@@  40  @@@@@  40
       DO 40 IX=2,LX
       IXP1=IX+1
       IXM1=IX-1
       UAP=(U(IXP1)+U(IX))/2.0
       UAM=(U(IX)+U(IXM1))/2.0
       CPP=CPU(IX)*BRO
      BKP=BK(UAP)
      BKM=BK(UAM)
      PBKP=P*BKP
      PBKP=P*BKP
      PBKM=P*BKM
      A(IX)=-PBKM
      B(IX)=(CPP+(PBKM+PBKP))
      C(IX)=-PBKP
      D(IX)=(PBKP*(U(IXP1)-U(IX)))-(PBKM*(U(IX)-U(IXM1))) + (CPP*U(IX))
  40   CONTINUE
       UAMP=(U(1)+U(2))/2.0
       CPP = CPU(1)*BRO
      BKMP=BK(UAMP)
      PBKMP=P*BKMP
      A(1)=0.0
      B(1)=(CPP+(2.0*PBKMP))
      C(1)=-2.0*PBKMP
      D(1)=(2.0*PBKMP*(U(2)-U(1))) + (CPP*U(1))
      CBL=HTCOF*2.0*DX/BK(U(LXP1))
       UDIF=U(LXP1)-TINF
       U(LXP2) = U(LX)-(CBL*UDIF)
      UAP=U(LXP1)
      UAM=(U(LXP1)+U(LX))/2.0
       CPP=CPU(LXP1)*BRO
      BKP=BK(UAP)
      BKM=BK(UAM)
      PBKP=P*BKP
      PBKM=P*BKM
      A(LXP1)=-(PBKM+PBKP)
      B(LXP1)=CPP+(PBKM+PBKP)+(PBKP*CBL)
      C(LXP1)=0.0
      D(LXP1)=(PBKP*(U(LXP2)-U(LXP1))) -(PBKM*(U(LXP1)-U(LX)))
     $ +(CPP*U(LXP1)) + (PBKP*CBL*TINF)
      CALL TRIDAG
C    *** LOOP 50 IS FOR CALCULATING ENTHALPY     ++++  50  ++++  50
      DO 50 IX=1,LXP1
      CPV(IX)=CP(V(IX))
      DUV=(U(IX)-V(IX))/3.0
      UT1=U(IX)-DUV
      UT2=UT1-DUV
      CPM1=CP(UT1)
      CPM2=CP(UT2)
      CPA=(CPU(IX)+CPV(IX)+CPM1+CPM2)/4.0
      IF(IX.LT.LX) GO TO 55
      DUV2=DUV/2.0
      UT3=U(IX)-DUV2
      UT4=UT3-DUV
      UT5=V(IX)-DUV2
      CPM3=CP(UT3)
      CPM4=CP(UT4)
      CPM5=CP(UT5)
      CPA=((CPA*4.0)+CPM3+CPM4+CPM5)/7.0
  55  CONTINUE
      DH(IX)=CPA*(U(IX)-V(IX))
      CPU(IX)=CPV(IX)
      TDH(IX)=TDH(IX)+DH(IX)
      TDHT=TDHT+TDH(IX)
  50  CONTINUE
      TDHTA=(TDHT-(0.5*(TDH(1)+TDH(LXP1))))/LX
      TDH(LXP1)=TDH(LX)
      DHDT=TDHTA-TDHTAI
      TDHTAI=TDHTA
C    *** LOOP 60 IS FOR TRANSFERING THE TEMPERATURE ARRAY   --- 60 ---60
      DO 60 IX=1,LXP1
      U(IX) = V(IX)
  60  CONTINUE
      RETURN
      END
C
```

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      SUBROUTINE TRIDAG
C     *********************************
C
C     SUB. TRIDAG SOLVES A TRIDIAGONAL SYSTEM OF EQUATIONS
C    A-SUB. DIAG., B-DIAG., C-SUP. DIAG., D-CONST. MATRIX
C    LI-SUBSCRIP. OF FIRST AND LX- OF LAST EQATION
      COMMON/TRIDA/LI,LX   ,A(33),B(33),C(33),D(33),V(33)
      DIMENSION BETA(34),GAMMA(34)
      DATA IOT/6/
      IF(LX.LT.LI) GO TO 3
      BETA(LI)=B(LI)
      GAMMA(LI)=D(LI)/BETA(LI)
      LIP1=LI+1
      DO 1 I=LIP1,LX
      BETA(I)=B(I)-A(I)*C(I-1)/BETA(I-1)
1     GAMMA(I)=(D(I)-A(I)*GAMMA(I-1))/BETA(I)
      V(LX)=GAMMA(LX)
      LAST=LX-LI
      DO 2 K=1,LAST
      I=LX-K
2     V(I)=GAMMA(I)-C(I)*V(I+1)/BETA(I)
      RETURN
3     WRITE(IOT,110)
110   FORMAT(5X,'FIRST EQUATION NUMBER IS BIGGER THAN LAST EQ. NUMBER')
      RETURN
      END
C
      FUNCTION BK(TF)
C     ****************
C
C   THIS FUNCTION COMPUTES THE BEEF THERMAL CONDUCTIVITY AS A
C       FUNCTIN OF TEMPERATURE
C
      IF(TF.GE.29.0)  GO TO 10
      IF(TF.LE.-20.0)  GO TO 20
      TF2=TF*TF
      BK=0.79805-(5.43226E-03*TF)+(3.71516E-05*TF2)-(4.19922E-06*TF*TF2)
     $ -(4.04242E-07*TF2*TF2)
      RETURN
10    BK=0.28
      RETURN
20    BK=0.89
      RETURN
      END
C
      FUNCTION CP(TF)
C     ******************
C
C   THIS FUNCTION COMPUTES THE BEEF SPECIFIC HEAT AS A FUNCTION OF TEMP.
      IF(TF.GE.31.1)  GO TO 10
      IF(TF.LE.1.4)  GO TO 20
      IF(TF.LT.31.1.AND.TF.GT.29.3)  CP=-14.722*TF+459.76
      IF(TF.LE.29.3.AND.TF.GT.27.5)  CP=10.889*TF-290.64
      IF(TF.LE.27.5.AND.TF.GT.12.2)  CP=63.874-(15.15*TF)+(1.3405*TF*TF)
     $ -(5.17964E-02*TF**3)+(7.4306E-04*TF**4)
      IF(TF.LE.12.2.AND.TF.GT.1.4)  CP=(2.7778E-02*TF)+0.61111
      RETURN
20    CP=(5.42E-03*TF)+0.6424
      RETURN
10    CP=0.85
      RETURN
      END
C
      SUBROUTINE REFSYS(EVPEAT,CFM,SYSCAP,BHPT,EVPLAT,ICU,IEVP,IEC)
C     ******************************************
C
C     THIS ROUTINE COMPUTES THE OPERATING POINTS OF ALL THE COMPONENTS
C     OF THE REFRIGERATION SYSTEM CHOOSEN BY ICU,IEVP,NUEVP  - AT THE
C     SYSTEM BALANCE POINT
C
      DIMENSION  C(6),CAPC(3),CONTC(3)
      COMMON/BR/EVPT,CONT,BHPCMM,BHPEFM,BHPCPM,NUEVP
C
C     GENERAL CONSTANTS
C
      TD=16.0
      EFCM=0.9
      EFEFM=0.9
      ERROR=0.0001
      IF(IEC.GT.1)  GO TO 5
C
C     CONDENSING UNIT SELECTION
      CALL CONDU(C,CAPC,CONTC,BHPCPM,ICU)
C
C
C     EVAPORATOR SELECTION
      CALL EVPR(CAPEVP,BHPPE,CFMPE,IEVP)
C
C
C
C     TOTAL SYSTEM
C
5     CONTINUE
      BHPEFM=(BHPPE*NUEVP)/EFEFM
      CAPEVT=CAPEVP*NUEVP
      CFM=CFMPE*NUEVP
      SLOPE=-CAPEVT/TD
      Y1=CAPEVP*(NUEVP+1)
      X=EVPEAT-TD
      XC=CAPEVT-(SLOPE*X)
      DO 10 I=1,100
      Y2=CAPC(1)+(CAPC(2)*X)+(CAPC(3)*X*X)
      X=(Y1-XC)/SLOPE
      IF(ABS((Y1-Y2)/Y2).LE.ERROR)  GO TO 20
      Y1T=Y1
      Y1=Y2
10    CONTINUE
      WRITE(6,30)  Y1T,Y2
30    FORMAT(5X,' * * * * * DIDNOT CONVERGE IN REFSYS',
     1 5X,'Y1 =',F6.1,5X,'Y2 =',F6.1)
      DEVP=2.0
      IF(CAPEVT.LT.XIRSYS)  DAVP=-DAVP
      EVPEAT=EVPEAT+DEVP
      IEC=IEC+1
      IF(IEC.LE.50) GO TO 5
      SYSCAP=0.0
      RETURN
20    CONTINUE
      EVPT=X
      SYSCAP=Y2
```

```
      CONT=CONTC(1)+(CONTC(2)*X)+(CONTC(3)*X*X)
      A=CONT
      B=EVPT
      BHPCOM=C(1)+(C(2)*A)+(C(3)*B)+(C(4)*A*B)+(C(5)*A*A)+(C(6)*B*B)
      BHPCMM=BHPCOM/EFCM
      BHPT=BHPCMM+BHPEFM+BHPCPM
      RO=519.0*0.0765/(460.0+EVPEAT)
      AMC=CFM*RO*60.0*0.24
      EVPLAT=EVPEAT-(SYSCAP*12000.0/AMC)
      RETURN
      END
C
      SUBROUTINE CONDU(C,CAPC,CONTC,BHPCPM,ICU)
C     *******************************************
C
C     THIS ROUTINE ASSIGNS THE CONSTANTS FOR COMPUTING REFR. CAP. ,
C     COND. TEMPR., AND BHP INPUT TO COMPRESSOR
C     THE COMPRESSOR CAPACITY MENSIONED IN COMENT CARD IS AT APPROX.
C     AT (EAVP.,COND.) TEMPERATURES
C     IT IS ONLY FOR IDENTIFICATION
C
      DIMENSION C(1),CAPC(1),CONTC(1)
      IF(ICU.GT.6)  ICU=6
      GO TO (1,2,3,4,5,6)  ,ICU
C
6     CONTINUE
      WRITE(6,110)
110   FORMAT(10X,'NO CONDENSING UNIT IS DEFINED')
      STOP
C
1     CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 50 TON AT APPR. (-10,90)
C     CONDENSER - 16 IN * 16 FT, 4 - PASS, 247 GPM
C
      C(1)=-0.1061434E02
      C(2)=0.1303359E01
      C(3)=-0.8302237E00
      C(4)=0.156218E-01
      C(5)=-0.5156777E-03
      C(6)=-0.3530699E-02
      CAPC(1)=63.25769
      CAPC(2)=1.4799
      CAPC(3)=1.279305E-02
      CONTC(1)=91.93774
      CONTC(2)=0.4102358
      CONTC(3)=1.95059E-04
      BHPCPM=1.0
      RETURN
2     CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 50 TON AT APPR. (-20,90)
C     CONDENSER - 16 IN * 16 FT, 4 - PASS, 247 GPM
C
      C(1)=-0.1181357E02
      C(2)=0.162822E01
      C(3)=-0.1271624E01
      C(4)=0.2152322E-01
      C(5)=0.234308E-03
      C(6)=-0.5366555E-02
      CAPC(1)=82.86708
      CAPC(2)=2.0686336
      CAPC(3)=1.9366557E-02
      CONTC(1)=97.6819346
      CONTC(2)=4.67631E-01
      CONTC(3)=1.705332E-03
      BHPCPM=1.0
      RETURN
3     CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 50 TON AT APPR. (-25,90)
C     CONDENSER - 16 IN * 16 FT, 4 - PASS, 247 GPM
C
      C(1)=-0.1700303E02
      C(2)=0.197953E01
      C(3)=-0.1317936E01
      C(4)=0.2339667E-01
      C(5)=-0.9139132E-03
      C(6)=-0.528525E-02
      CAPC(1)=91.73291
      CAPC(2)=2.158959
      CAPC(3)=1.7331922E-02
      CONTC(1)=99.851913
      CONTC(2)=4.6133839E-01
      CONTC(3)=1.0993118E-03
      BHPCPM=1.0
      RETURN
4     CONTINUE
C
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 50 TON AT APPR. (-30,90)
C     CONDENSER - 16 IN * 16 FT, 4 - PASS, 247 GPM
C
      C(1)=0.1198058E02
      C(2)=0.174556E01
      C(3)=-0.1236872E01
      C(4)=0.247203E-01
      C(5)=0.2292582E-02
      C(6)=-0.6099057E-02
      CAPC(1)=104.4573
      CAPC(2)=1.8246528
      CAPC(3)=3.8394765E-03
      CONTC(1)=105.4701
      CONTC(2)=5.9027778E-01
      CONTC(3)=3.3386752E-03
      BHPCPM=1.0
      RETURN
5     CONTINUE
C     CONDENSING UNIT IS MADE UP OF
C     COMPRESSOR - 50 TON AT APPR. (-40,90)
C     CONDENSER - 16 IN * 16 FT, 4 - PASS, 247 GPM
      C(1)=0.6863323E02
      C(2)=0.127371E01
      C(3)=-0.1747627E01
      C(4)=0.3530834E-01
      C(5)=0.1037036E-01
      C(6)=-0.85865E-02
```

```
      CAPC(1)=163.96956
      CAPC(2)=4.138629
      CAPC(3)=3.6043912E-02
      CONTC(1)=113.5493
      CONTC(2)=5.593893E-01
      CONTC(3)=0.0
      BHPCPM=1.0
      RETURN
      END
      SUBROUTINE EVPR(CAPEVP,BHPPE,CFMPE,IEVP)
C     ******************************************
C
C     THIS ROUTINE ASSIGNS THE CHARACTERISTIC VALUES OF THE EVAPORATOR
C     CHOOSEN. ALL EVP. ARE 10 ROWS DEEP
C
      IF(IEVP.GT.6)  IEVP=6
      GO TO (1,2,3,4,5,6)  , IEVP
5     CONTINUE
C
C     EVAPORATOR - 2.3 TON/TD, 2 HP/FAN, 1140 RPM
C
      CAPEVP=35.8
      BHPPE=6.0
      CFMPE=33000.0
      RETURN
4     CONTINUE
C
C     EVAPORATOR - 1.81 TON/TD, 3 HP/FAN, 1140 RPM
C
      CAPEVP=23.93
      BHPPE=6.0
      CFMPE=26300.0
      RETURN
3     CONTINUE
C
C     EVAPORATOR - 1.27TON/TD, 1.5 HP/FAN, 1140 RPM
C
      CAPEVP=20.27
      BHPPE=3.0
      CFMPE=17500.0
      RETURN
2     CONTINUE
C
C     EVAPORATOR - 0.91 TON/TD, 1.0 HP/FAN, 1140 RPM
C
      CAPEVP=14.53
      BHPPE=2.0
      CFMPE=12600.0
      RETURN
1     CONTINUE
C
C     EVAPORATOR - 0.67 TON/TD, 0.75 HP/FAN, 1140 RPM
C
      CAPEVP=10.67
      BHPPE=1.5
      CFMPE=8900.0
      RETURN
6     CONTINUE
      WRITE(6,110)
110   FORMAT(10X,'* * * NO EVPERATOR IS DEFINED * * * ')
      STOP
      END
```

```
C
C     COMPUTER PROGRAM FOR MODEL - B1
C     ********************************
C
C     FOR VARIABLE NAMES REFER TO COMPUTER PROGRAM FOR MODEL - B2
C     THIS MODEL IS CHECKED WITH THE RESULTS PUBLISHED IN REFERENCE 10
C
       IMPLICIT REAL*8 (A-H,O-Z)
C     X-PRECIDING AVARIABLE CHANGES THE VARIABLE TO REAL
C     I-PRECIDING AVARIABLE CHANGES THE VARIABLE TO INTEGER VARIABLE
       COMMON/TRIDA/LI,LXP1,A(33),B(33),C(33),D(33),V(33)
       DIMENSION U(33),X(33)
       DIMENSION HFM(33),DH(33),TDH(33),CPU(33),CPV(33)
       NAMELIST /NAM1/LX,IXPRT,ITPRT,XLI,TINF,TMAX,DTM,DHF,HTCOF
     *          ,HPRT,ITPRT1
       DATA IIN,IOT/5,6/
       DATA X0,T,IXPRT,ITPRT/2*0.0D00,1,10/
       DATA XL,LX/1.0D00,10/
200    FORMAT(///,4X,'THE EQUATION TO BE SOLVED IS',//
     * ,10X,'D/DX(K(U)*DU/DX)=RHO*C(U)*DU/DT',//
     *,8X,'THE BOUNDARY CONDITIONS ARE',//
     *,10X,'  AT X=0.0  DU/DX = 0.0',//
     * ,10X,'  AT X=L  DU/DX + ((HTCOF/K(U))*(U-UINF)) = 0.0',//
     *)
205    FORMAT(5X,'INPUT DATA',//
     *,10X,'LENGTH VARIES FROM ',F5.3,' TO ',F5.3,2X,'IN',//
     *,10X,'NO. OF SECTIONS ALONG LENGTH =',I3,//
     * ,10X,'TINF = ',F7.1,2X,'F',//
     * ,10X,'HTCOF= ',F7.1,2X,'BTU/HR-FT**2-F',//
     * ,10X,      'ENTHALPY REDUCTION =',F7.1,2X,'BTU/LB',//
     *)
210    FORMAT(5X,'THE VALUES OF U(X,T) AT TIME=0.0 IN F ARE',/)
215    FORMAT(//,5X,'THE VALUES OF H(X,T) AT TIME=0.0 IN BTU/LB ARE',/)
220    FORMAT(3(5X,3('U(',F5.3,',0.0) =', G12.4,2X),//))
225    FORMAT(8(5X,3('H(',F5.3,',0.0) =', G12.4,2X),//))
230    FORMAT(//,5X,'TIME = ',F8.4, 5X,'AVE. ENTHALPY REDUCTION = ',F6.1
     * ,2X,'BTU/LB',/)
240    FORMAT(/, 10X,'X = ',11F10.3)
250    FORMAT(/, 10X,'U = ',11F10.1)
255    FORMAT(/,10X,'H = ',11F10.1)
260    FORMAT(1H1)
       DO 5 J=1,4
       RHO=65.0
       X0=0.0
       T=0.0
       LX=10
       ITPRT=10
       TINIT=45.0
       ITPRT1=2
       HPRT=115.0
       LI=1
       HTCOF=5.0
       XLI=1.0
       TINF=-20.0
       DTM=1.0
       DHF=120.0
       TMAX=10.0
       READ(IIN,NAM1)
       NTMAX=((TMAX*60.0)/DTM)+1
       XL=XLI/12.0
       XLX=LX
       DX=XL/XLX
       LXP1=LX+1
       LXP2=LX+2
       WRITE(IOT,260)
       WRITE(IOT,NAM1)
       WRITE(IOT,200)
       WRITE(IOT,205)  X0,XLI,LX,TINF,HTCOF,DHF
C     *** FINDING X-SECTIONS,U(X,0.0),ENTHALPY      ..... LOOP 10  .....
       DO 10 IX=1,LXP1
       X(IX)=X0
       U(IX)=TINIT
       CPU(IX)=CP(U(IX))
       TDH(IX)=0.0
       HFM(IX)=(71.6*1.8)+(0.35*(U(IX)-32.0))
       X0=X0+DX
10     CONTINUE
       CBL=HTCOF*2.0*DX/BK(U(LXP1))
       U(LXP2)=U(LX)-(CBL*(U(LXP1)-TINF))
       WRITE(IOT,210)
       WRITE(IOT,220)  (X(IX),U(IX)  ,IX=1,LXP1,IXPRT)
       WRITE(IOT,215)
       WRITE(IOT,225)  (X(IX),HFM(IX),IX=1,LXP1,IXPRT)
       WRITE(IOT,260)
C     *** LOOP 20 IS FOR EVERY TIME INCRIMENT DT       ******20 ******* 20
       DO 15 ITT =1,3
       DT=DTM/600.0
       NTMA=20
       IF(ITT.EQ.1)  GO TO 16
       IF(ITT.EQ.2)  GO TO 17
       NTMA=NTMAX
       DT=DTM/60.0
       GO TO 16
17     CONTINUE
       DT=DTM/120.0
       NTMA=(((10.0/60.0)-T)/DT+0.5)
16     CONTINUE
       P=DT/(2.0*DX*DX)
        DO 20 IT=1,NTMA
       TDHT=0.0
       T=T+DT
C     *** LOOP 40 IS FOR CALCULATION OF COEFF      @@@@@  40   @@@@@  40
        DO 40 IX=2,LX
        IXP1=IX+1
        IXM1=IX-1
        UAP=(U(IXP1)+U(IX))/2.0
        UAM=(U(IX)+U(IXM1))/2.0
        CPP=CPU(IX)*RHO
       BKP=BK(UAP)
       BKM=BK(UAM)
       PBKP=P*BKP
       PBKP=P*BKP
       PBKM=P*BKM
       A(IX)=-PBKM
       B(IX)=(CPP+(PBKM+PBKP))
       C(IX)=-PBKP
        D(IX)=(PBKP*(U(IXP1)-U(IX)))-(PBKM*(U(IX)-U(IXM1))) + (CPP*U(IX))
40      CONTINUE
        UAMP=(U(1)+U(2))/2.0
```

```
      CPP = CPU(1)*RHO
      BKMP=BK(UAMP)
      PBKMP=P*BKMP
      A(1)=0.0
      B(1)=(CPP+(2.0*PBKMP))
      C(1)=-2.0*PBKMP
      D(1)=(2.0*PBKMP*(U(2)-U(1))) + (CPP*U(1))
      CBL=HTCOF*2.0*DX/BK(U(LXP1))
      UDIF=U(LXP1)-TINF
      U(LXP2) = U(LX)-(CBL*UDIF)
      UAP=U(LXP1)
      UAM=(U(LXP1)+U(LX))/2.0
      CPP=CPU(LXP1)*RHO
      BKP=BK(UAP)
      BKM=BK(UAM)
      PBKP=P*BKP
      PBKM=P*BKM
      A(LXP1)=-(PBKM+PBKP)
      B(LXP1)=CPP+(PBKM+PBKP)+(PBKP*CBL)
      C(LXP1)=0.0
      D(LXP1)=(PBKP*(U(LXP2)-U(LXP1))) -(PBKM*(U(LXP1)-U(LX)))
     $ +(CPP*U(LXP1)) + (PBKP*CBL*TINF)
      CALL TRIDAG
C     *** LOOP 50 IS FOR CALCULATING ENTHALPY      ++++  50  ++++  50
      DO 50 IX=1,LXP1
      CPV(IX)=CP(V(IX))
      DUV=(U(IX)-V(IX))/3.0
      UT1=U(IX)-DUV
      UT2=UT1-DUV
      CPM1=CP(UT1)
      CPM2=CP(UT2)
      CPA=(CPU(IX)+CPV(IX)+CPM1+CPM2)/4.0
      IF(IX.LT.LX) GO TO 55
      DUV2=DUV/2.0
      UT3=U(IX)-DUV2
      UT4=UT3-DUV
      UT5=V(IX)-DUV2
      CPM3=CP(UT3)
      CPM4=CP(UT4)
      CPM5=CP(UT5)
      CPA=((CPA*4.0)+CPM3+CPM4+CPM5)/7.0
55    CONTINUE
      DH(IX)=CPA*(U(IX)-V(IX))
      CPU(IX)=CPV(IX)
      TDH(IX)=TDH(IX)+DH(IX)
      TDHT=TDHT+TDH(IX)
50    CONTINUE
      TDHTA=(TDHT-(0.5*(TDH(1)+TDH(LXP1))))/LX
      TDH(LXP1)=TDH(LX)
      IF(TDHTA.GE.DHF)  GO TO 70
      IF(TDHTA.GE.HPRT)  ITPRT=ITPRT1
            IT1=IT/ITPRT
            IT2=IT1*ITPRT
            IF(IT2.NE.IT)  GO TO 25
             WRITE(IOT,230)  T,TDHTA
             WRITE(IOT,240)  (X(IX),IX=1,LXP1,IXPRT)
             WRITE(IOT,250)  (V(IX),IX=1,LXP1,IXPRT)
             WRITE(IOT,255)  (TDH(IX),IX=1,LXP1,IXPRT)
25    CONTINUE
C     *** LOOP 60 IS FOR TRANSFERING THE TEMPERATURE ARRAY   --- 60 ---60
```

```
      DO 60 IX=1,LXP1
      U(IX) = V(IX)
60    CONTINUE
20    CONTINUE
15    CONTINUE
70     CONTINUE
            WRITE(IOT,230)  T,TDHTA
            WRITE(IOT,240)  (X(IX),IX=1,LXP1,IXPRT)
            WRITE(IOT,250)  (V(IX),IX=1,LXP1,IXPRT)
            WRITE(IOT,255)  (TDH(IX),IX=1,LXP1,IXPRT)
5     CONTINUE
      STOP
      END
      SUBROUTINE TRIDAG
      IMPLICIT REAL*8 (A-H,O-Z)
C     SUB. TRIDAG SOLVES A TRIDIAGONAL SYSTEM OF EQUATIONS
C    A-SUB. DIAG., B-DIAG., C-SUP. DIAG., D-CONST. MATRIX
C    LI-SUBSCRIP. OF FIRST AND LX- OF LAST EQATION
      COMMON/TRIDA/LI,LX  ,A(33),B(33),C(33),D(33),V(33)
      DIMENSION BETA(34),GAMMA(34)
      DATA IOT/6/
      IF(LX.LT.LI) GO TO 3
      BETA(LI)=B(LI)
      GAMMA(LI)=D(LI)/BETA(LI)
      LIP1=LI+1
      DO 1 I=LIP1,LX
      BETA(I)=B(I)-A(I)*C(I-1)/BETA(I-1)
1     GAMMA(I)=(D(I)-A(I)*GAMMA(I-1))/BETA(I)
      V(LX)=GAMMA(LX)
      LAST=LX-LI
      DO 2 K=1,LAST
      I=LX-K
2     V(I)=GAMMA(I)-C(I)*V(I+1)/BETA(I)
      RETURN
3     WRITE(IOT,110)
110   FORMAT(5X,'FIRST EQUATION NUMBER IS BIGGER THAN LAST EQ. NUMBER')
      RETURN
      END
      FUNCTION BK(TF)
      IMPLICIT REAL*8 (A-H,O-Z)
      IF(TF.GE.29.0)  GO TO 10
      IF(TF.LE.-20.0)  GO TO 20
      TF2=TF*TF
      BK=0.79805-(5.43226D-03*TF)+(3.71516D-05*TF2)-(4.19922D-06*TF*TF2)
     $ -(4.04242D-07*TF2*TF2)
      RETURN
10    BK=0.28
      RETURN
20    BK=0.89
      RETURN
      END
      FUNCTION CP(TF)
      IMPLICIT REAL*8 (A-H,O-Z)
      IF(TF.GE.31.1)  GO TO 10
      IF(TF.LE.1.4)  GO TO 20
      IF(TF.LT.31.1.AND.TF.GT.29.3)  CP=-14.722*TF+459.76
      IF(TF.LE.29.3.AND.TF.GT.27.5)  CP=10.889*TF-290.64
      IF(TF.LE.27.5.AND.TF.GT.12.2)  CP=63.874-(15.15*TF)+(1.3405*TF*TF)
     $ -(5.17964D-02*TF**3)+(7.4305D-04*TF**4)
      IF(TF.LE.12.2.AND.TF.GT.1.4)  CP=(2.7778E-02*TF)+0.61111
```

```
      RETURN
20    CP=(5.42D-03*TF)+0.6424
      RETURN
10    CP=0.85
      RETURN
      END
```

VITA

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