

FOOD PROCESS DESIGN

Zacharias B. Maroulis

*National Technical University of Athens
Athens, Greece*

George D. Saravacos

*Rutgers, the State University of New Jersey
New Brunswick, New Jersey, U.S.A.
and National Technical University of Athens
Athens, Greece*

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Preface

The design of processes, processing equipment, and processing plants in the food industry is still based on practical experience and empirical knowledge, although significant progress has been made on the underlying transport phenomena and unit operations. The main difficulty lies in the complexity of food composition and structure and the limited data on reliable physical and transport properties of food materials. In addition to engineering and economic considerations, food processes must produce food products that are safe, nutritious and acceptable to the consumers.

Recent advances in Food Process Engineering, especially in unit operations and physical properties of foods, can be utilized in Food Process Design, which should be developed as a new area, utilizing the established field of Chemical Process Design.

Advances in modeling and computer simulation can be applied to the design of more efficient food processes, which can be controlled and operated more effectively. The development of new PC software, such as the Excel spreadsheets, has simplified the computer implementation in Process Design, eliminating the need for detailed computer programming and coding. Food Process Engineering, and especially Food Process Design, can benefit greatly from the application of this computer technology.

The purpose of this book is to introduce the application of computer spreadsheets to the design of industrial food processes. The most important food processes, which can be modeled and simulated (mainly heat and mass transfer processes), are designed, using fundamental engineering and economic relationships, and literature data of physical and transport properties of foods. The design of food mechanical processes, such as mechanical processing/separations and packaging, is still based on empirical knowledge of equipment suppliers and food plant operators.

Chapter 1 introduces the basic concepts of Food Science, which are related to process design, i.e., chemical kinetics, food microbiology, food safety, and food quality. Chapter 2 reviews the principles of Process Design, with emphasis on food processes. The concepts of process cost and profitability are introduced with an example of design of an entire tomato-paste-processing plant. Chapter 3 deals with the principles of computer-aided process design, using computer spread-

sheets. The spreadsheet implementation is demonstrated, using as an example the design of a liquid/liquid solvent extraction.

Chapters 4 to 10 cover the design of the most important heat and mass transfer processes of the food processing industry, using the computer spreadsheet technique. Each process is designed and optimized with respect to the total annualized cost, which consists of the equipment and utilities operation costs of the specific process. The design results of the individual processes can be utilized in the economic evaluation (profitability) of the entire food processing plant, as shown in the example of Chapter 2.

Chapters 4 and 5 cover the design of heating, cooling and freezing of food products; the spreadsheet technique used in the design of heat exchangers, coolers, and freezers of typical food products; and the analysis of process cost and the optimum operating conditions, using literature and estimated technical and economic data.

Chapters 6 and 7 deal with the design of food evaporation and food dehydration, using literature data; the spreadsheets are used to estimate the annualized cost of the main process, and optimize the evaporation and dehydration operations.

Chapter 8 deals in detail with the design of thermal processing of foods (pasteurization and sterilization), covering both continuous flow and in-container processing; Process optimization is concerned with both microbial inactivation and heat damage to important food components. The spreadsheet technique is compared to the traditional Ball formula method for in-container sterilization.

Chapter 9 covers the design of two important mass transfer processes of interest to foods, i.e., distillation and extraction. Chapter 10 is concerned with the design of membrane separations of importance to food processing, i.e., ultrafiltration and reverse osmosis.

Finally, the Appendix contains engineering data and physical and transport properties related to Process Design, i.e., conversions to SI units, and properties of foods, air, water, and steam.

We wish to acknowledge the contributions and help of our colleagues, associates and graduate students at National Technical University of Athens, especially D. Marinos-Kouris and M. Krokida.

Finally, we must thank the staff of the publisher Marcel Dekker, Inc., especially Maria Allegra and Theresa Stockton for encouraging our efforts in this project, and helping in editing the manuscript.

We hope that this book will contribute to the recognition of Food Process Design as an important part of Food Engineering in both academia and industry. We welcome any suggestions and criticism on the contents of the book. We regret any errors that may have escaped our attention.

Zacharias B. Maroulis
George D. Saravacos

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1

Food Science in Process Design

I. INTRODUCTION

Food Process Design, in addition to heat/mass transfer and reaction engineering, should consider important aspects of Food Science, related to food safety, nutrition, and food quality. The objective of food process or food plant design is to economically produce food products, which are safe, nutritious, and organoleptically acceptable to the consumers.

Food Science and Technology, consisting basically of Food Chemistry, Food Microbiology, and Food Engineering, is concerned with the processing, storage, and use of food products in human nutrition.

Food Chemistry is concerned with the chemical composition and chemical changes during processing, storage, and use. Chemical reactions in foods are mostly undesirable, since they may result in degradation of food quality, e.g., oxidation, hydrolysis, and polymerization. However, some food processes are based on chemical reactions, e.g., cooking and roasting. Chemical kinetics of food reactions are very important in quantifying the various changes in food quality.

Food Microbiology is concerned with the growth and inhibition or destruction of microorganisms, which may cause food spoilage or illness to the consumers. Food safety is based mainly on the control of food-borne bacteria. Microbial destruction kinetics is similar to chemical kinetics, and quantitative data are essential on designing various food processes.

Nutritional considerations of food processes are related mostly to the degradation of important food nutrients, such as vitamins and proteins.

Food Engineering is important in the design of food processes, processing equipment, and processing plants. Engineering principles, practical experience, and economics should be applied, while taking into consideration the principles and experience of Food Science and Technology.

II. FOOD PRESERVATION PROCESSES

1. Conventional Processes

The design of conventional physical preservation processes for foods, i.e., thermal processing, refrigeration and freezing, evaporation, and dehydration is discussed in individual chapters of this book. The objective of these processes is to inactivate or reduce substantially the action of spoilage microorganisms by heat, cold, or low water activity (Heldman and Hartel, 1997; Fellows, 1990). Conventionally processed foods are shelf-stable products, i.e., they can be stored at ambient temperatures for several months. Canning technology is described by Downing (1996).

Modeling of the heating, cooling, freezing, or concentrating processes can be simplified by realistic assumptions, and the solution of the basic equations can be facilitated by computers. Modeling of inactivation of the microorganisms is simplified by assuming, in most cases, first-order reactions and Arrhenius dependence on temperature. Quality deterioration is assumed to follow first-order kinetics, and optimization can yield a process with safe microbial reduction and minimum quality damage. Detailed analysis of the inactivation kinetics of microorganisms and food components is presented in Chapter 8. An introduction to computer applications in Food Technology was presented by Singh (1996).

2. Minimal Processing

Minimal processing is used for the preservation of short shelf-life fruits and vegetables, and meat products, with minimum damage to the freshness of the product. Minimally processed foods retain their freshness and they are more acceptable to the consumers than conventionally preserved products.

Minimal processing can be achieved using a combination of mild processing methods, which control the growth of spoilage and pathogenic microorganisms (hurdle technology), without serious damage to the product quality (Singh and Oliveira, 1994; Alzamora et al., 2000).

Hurdles (obstacles) used in food preservation include temperature, water activity, pH, preservatives, and competing microorganisms (e.g., lactic acid bacteria). Recent nonthermal preservation methods, such as irradiation, high pressure, and pulsed electric fields, may also be used.

Minimally processed fruits can be produced using the following hurdles: Steam blanching for 1-3 min, reduction of water activity (a_w) to 0.98-0.93 by sugar addition, lowering the pH to 4.1-3.0 by addition of citric or phosphoric acid, and addition of antimicrobials, e.g., 1,000 ppm of sorbate and sulfite. The treated fruits can be packaged and stored at ambient (room) temperature (up to 35 °C) for about 4 months (Leistner, 2000).

Minimally processed raw vegetables can be produced by modified atmosphere packaging (MAP) and storage at 1-8 °C for 5-7 days. Vegetable and potato dishes (salads), vacuum packaged (sous vide) and heated mildly at 65-95 °C, can be stored under refrigeration (1-8 °C) for about 42 days. The growth of anaerobic microorganisms, such as toxin-producing *Cl. botulinum*, should be prevented by some additional hurdle.

Minimally processed meat products (e.g., sausages), which are shelf-stable (storage at ambient temperature) can be produced by mild heating (70-110 °C) of the plastic-packaged product, lowering the water activity or *pH*, and adding an antimicrobial, e.g., nitrite.

Development of minimally processed foods by hurdle technology should be combined with *GMPs* (good manufacturing practices) and *HACCP* (hazard analysis) for better process control.

3. Nonthermal Processing

Nonthermal food preservation is still in the development stage, and only limited technical information and engineering data are available for reliable process design. These processes are suitable for partial inactivation of pathogenic and spoilage microorganisms, e.g., by a factor of 10^{-4} . The economics of these novel processes is not yet established, but there is a commercial potential, because of the important advantages (better food quality, no heat damage to sensitive foods, and less energy requirement). The following nonthermal preservation methods are listed in order of diminishing importance:

i. Irradiation

Irradiation preservation of foods is based on the inactivation of spoilage and pathogenic microorganisms by ionizing radiations, i.e., high-energy electrons, X-rays, or gamma rays (Saravacos and Kostaropoulos, 2002). Low irradiation doses (about 0.1 kGy) can prevent the sprouting of potatoes, and 0.15-0.75 kGy will kill the storage insects. Pasteurization (inactivation of pathogenic and most spoilage microorganisms) requires 1-10 kGy, while sterilization requires doses of 10-30 kGy. The dose of 1 Gy (Gray) corresponds to energy absorption of 1 J/kg or 100 rad (1 rad = 100 erg/g).

Low doses of irradiation have been approved by Health Authorities of various countries for specific food products, while other products and processes are waiting approval before commercialization.

ii. High Pressure Processing

High (hydrostatic) pressure in the range of 1-8 kbar (100-800 MPa) can inactivate vegetative cells of pathogenic and spoilage microorganisms, without heat damage to the quality of sensitive foods. High pressure processing (HPP) can be applied to pasteurization and minimal processing. Destruction of microbial spores may require combined heat and HPP treatment (Barbosa-Canovas et al., 2000).

iii. Pulsed Electric Fields

Pulsed electric fields can inactivate vegetative cells by dielectric breakdown of bacterial membranes, e.g., electric fields of 30-60 kV/cm and pulse duration of about 1 μ s (Barbosa-Canovas and Zhang, 2000). Limited technical data are available for the engineering design of the process.

III. CHEMICAL KINETICS

Chemical aspects of Food Science of importance to food process design are discussed by Walstra and Jemnes (1984), Valentas et al. (1991), Fennema (1996), and Warthesen and Muehlenkamp (1997). The role of chemical kinetics in food systems is discussed by Villota and Hawkes (1992), and Taoukis et al. (1997).

Most of the chemical changes in food systems are considered first-order reactions. Only a few food reactions can be represented by zero-order reactions. The first-order kinetics is described by the equation:

$$\frac{dC}{dt} = -kC \quad (1-1)$$

where C (kg/m^3) is the concentration and k ($1/\text{s}$) is the reaction constant.

The half-time or half-life $t_{1/2}$ of a chemical change, i.e., the time t (s) at which the initial concentration of a component is reduced by 50% ($C/C_o = 0.5$) is given by the equation:

$$t_{1/2} = \ln(2)/k = 0.693/k \quad (1-2)$$

The effect of temperature on the reaction rate (constant) is given by the Arrhenius equation:

$$\ln\left(\frac{k}{k_o}\right) = \frac{E}{R}\left(\frac{1}{T_o} - \frac{1}{T}\right) \quad (1-3)$$

where k , k_o are the reaction constants at temperatures T , T_o , respectively (K), E is the energy of activation (J/mol), and R is the gas constant (8.31 J/mol K).

The activation energy E is a strong indication of the underlying mechanism of the food reaction. Low E values are characteristic of enzymatic reactions, while very high values are found in protein denaturation. Table 1.1 and Figure 1.1 show some typical energies of activation of chemical food reactions (Villota and Hawkes, 1992).

It is shown that the activation energy of microbial destruction (bacterial spores and vegetative cells) is much higher than the E value of most food chemical reactions. The similarity of the activation energies of microbial destruction and protein denaturation suggests that microbial death may be caused by some irreversible denaturation of a key microbial protein.

Food proteins are more heat labile than other basic food components, due to physico-chemical denaturation (conformation change) of the large and complex molecules (Stanley and Yada, 1992).

Phase and state transitions of food polymers and carbohydrates are defined by the glass transition temperature T_g , above which the food material is rubbery, and it changes to a glass material at lower temperature. The effect of temperature on the reaction constant near the glass transition temperature is better described by the Williams, Landel and Ferry (WLF) equation (Roos, 1992):

$$\log\left(\frac{k_g}{k}\right) = -\frac{C_1(T - T_g)}{C_2 + (T - T_g)} \quad (1-4)$$

where k_g , k are the reaction constants at temperatures T_g , T , respectively, and empirical constants $C_1 = 17.4$ and $C_2 = 51.6$.

Table 1.1 Activation Energies of Food Reactions

| Reaction | Energy of Activation, kJ/mol |
|-----------------------------|------------------------------|
| Enzymatic reactions | 4 - 60 |
| Chlorophyll degradation | 20 - 100 |
| Ascorbic acid degradation | 20 - 150 |
| Non enzymatic browning | 50 - 150 |
| Lipid oxidation | 40 - 100 |
| Bacterial spore destruction | 200 - 350 |
| Vegetative cell destruction | 200 - 600 |
| Protein denaturation | 300 - 500 |

Data from Villota and Hawkes, 1992

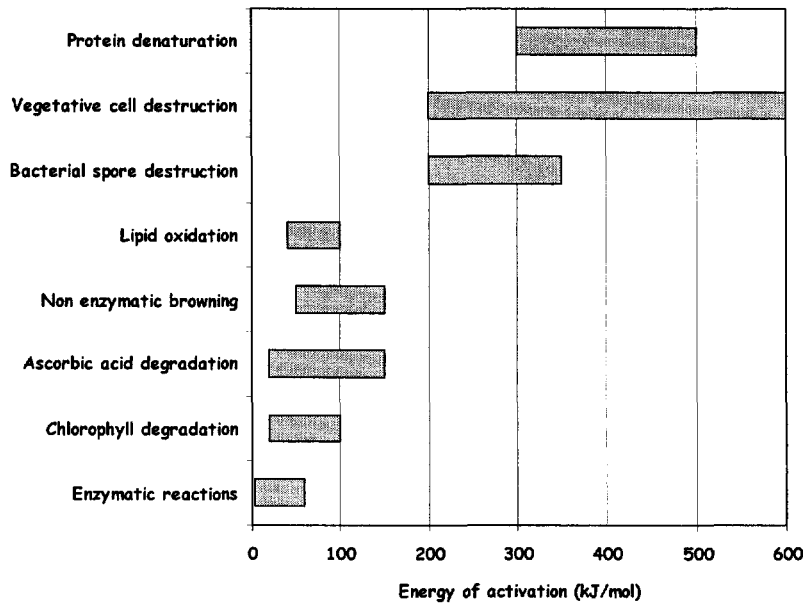


Figure 1.1 Activation energies of food reactions.

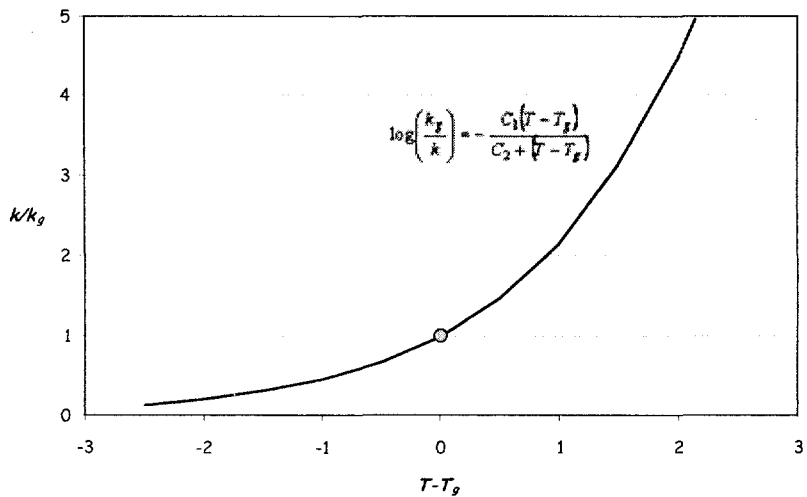


Figure 1.2 Effect of temperature on reaction constant near the glass transition temperature using the WLF equation.

The WLF equation predicts a sharp increase of the reaction constant above the T_g (see Figure 1.2). The same equation can be applied to predict the sharp decrease of viscosity or increase of mass diffusivity just above the T_g .

IV. FOOD MICROBIOLOGY AND FOOD SAFETY

1. Food Microbiology

The inhibition and control of microbial action is of particular importance in the design of food preservation and manufacturing processes. Microbial control is essential to food safety for the consumers (Lund et al., 1999).

Food preservation processes, such as thermal processing, refrigeration/chilling, freezing and drying, are based on the control or inactivation of spoilage and pathogenic microorganisms. Bacteria and bacterial spores are of primary importance, because of their high resistance to heat treatment and their potential as public health hazards.

Microbial growth depends on the food composition and the environmental factors, such as temperature, moisture (water activity), pH, chemical composition, and gas composition (oxygen, carbon dioxide). The life cycle of a microbial population consists of four phases, i.e., lag, growth, stationary, and death. The growth and death phases are usually exponential, while the lag and stationary phases are asymptotic.

The microbial growth is represented by a sigmoid curve, which can be expressed by the following two simplified empirical models (Wilcox et al., 1993):

Logistic model

$$\log\left(\frac{N}{N_o}\right) = \frac{C}{1 + \exp(-B(t - M))} \quad (1-5)$$

Gompertz model

$$\log\left(\frac{N}{N_o}\right) = C \exp(-\exp(\exp(-B(t - M)))) \quad (1-6)$$

where N_o is the initial (asymptotic) microbial density (colonies/mL), N is the microbial density after time t , C is the number of log cycles of the growth, M is the time at which the growth rate is maximal, and B is the relative growth rate at time M (see Figure 1.3).

The effect of temperature T on the growth rate of microorganisms is usually modeled by the Arrhenius Equation (1-3). In some systems, the square root model for the maximal growth rate constant B is used:

$$\sqrt{B} = b(T - T_{min}) \quad (1-7)$$

where T_{min} is the theoretical minimum temperature for growth, and b is a constant (Ratkowsky regression parameter).

The microbial death is treated in detail in Chapter 8 (Thermal Processing). The traditional design of thermal processing of foods is based on first-order kinetics of inactivation (death) of microorganisms. In thermal processing of foods, the reaction constant of microbial inactivation k at a given temperature (Equation 1-1) is usually replaced by the decimal reduction time (D , min):

$$D = 2.3/k \quad (1-8)$$

The effect of temperature on k (or D), expressed by the activation energy E of the Arrhenius Equation (1-3), is usually replaced by the z value, i.e., the increase of temperature, which results in a 90% reduction of D . The z -value is related to E by the approximate equation:

$$z = 2.3 \frac{RT_o^2}{E} \quad (1-9)$$

where T_o is the reference temperature and $R = 8.31$ J/mol K the gas constant.

The first-order kinetic model for microbial inactivation is not applicable to some food systems. Instead, non-log-linear curves are obtained, which can be expressed by alternate microbial survivor curves, based on statistical interpretations (IFT, 2003).

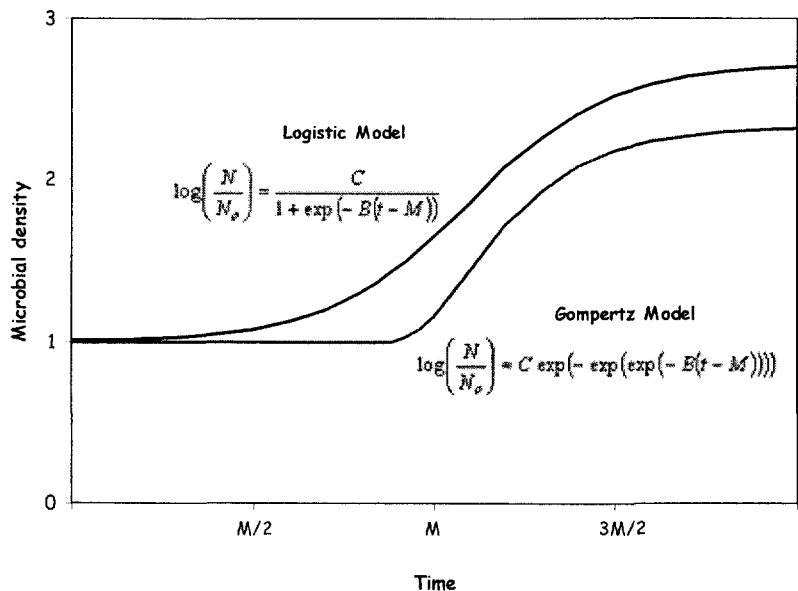


Figure 1.3 Microbial growth curves.

2. Food Safety

Food safety is concerned mainly with the growth of pathogenic microorganisms, which may cause disease to the consumers. Food safety is taken into consideration by the programs GMP (Good Manufacturing Practices) and HACCP (Hazard Analysis for Critical Control Points), and Risk Analysis.

In addition to the GMP and HACCP, the Food Safety Objectives (FSO) should be considered for the safe processing and distribution of food products. The FSOs were proposed recently by the International Commission on Microbiological Specifications of Foods (ICMSF), and they examine the maximum frequency or concentration of a microbial standard in a food at the time of consumption that provides the appropriate level of protection (Arthur, 2002).

Food pathogens include bacteria, such as *Salmonellae*, *Listeria monocytogenes*, *Escherichia coli* 0157:H7, and *Campylobacter*. Viruses, biotoxins, and parasites may be also involved in food-borne diseases.

Food safety can be assessed and controlled by Predictive Microbiology, which is concerned with microbial responses (microbial growth) of compositional and environmental factors in food systems (McMeekin et al., 1993, Banks, 1992). The growth of pathogenic bacteria (e.g., *Salmonellae*, *Listeria*) in foods stored at room or refrigerated temperatures is of particular importance in food preservation.

Empirical models are used to predict the growth of spoilage or pathogenic bacteria in a given food as a function of temperature, water activity, pH, salt content, organic acids, antimicrobial agents, and gas composition. The predictive models allow the development of software, which can estimate quantitatively the

hazard from pathogenic bacteria under definite conditions of the food process. The models can estimate the growth, survival, and death of indicator microorganisms, as affected by intrinsic parameters of the food or extrinsic factors applied to the food. Both microbial safety and shelf life of the food product can be predicted.

a. Good Manufacturing Practices (GMP)

Hygiene (Sanitation) is a fundamental requirement of all food plant operations, i.e., processing, packaging, storage, buildings, and personnel. Microbial and non-microbial contamination should be prevented by proper design and operation of all processing equipment and the entire plant.

The principles and practices of Good Manufacturing Practices (*GMP*) should be taken into serious consideration in food plant and equipment design (Popham, 1996). *GMPs* are a combination of manufacturing and management practices aimed at ensuring that food products are consistently produced to meet specifications and customer expectations. They are practical rules and recommendations, based on experience, which, when followed in the various food processing operations, will result in safe and high-quality food products (Gould, 1994). In the *USA*, the following Agencies have responsibilities on food processing plants and processing equipment: *FDA* (Food and Drug Administration), *USDA* (US Department of Agriculture), *EPA* (Environmental Protection Agency), *FTC* (Federal Trade Commission), and *CS* (Customs Service).

The Code of Federal Regulations in the *USA* (part 110, title 21) contains the practices, which must be followed in food plants, processing foods for human consumption (Gould, 1994). These regulations are enforced by the Food and Drug Administration (*FDA*). They are updated regularly and published in the Federal Register (Washington, DC). The rules cover the buildings, processing equipment, and personnel of the processing plant. They also cover processing, hygienic, and control operations, receiving, warehousing, shipping, and distribution of the food products.

Each country has rules and regulations, related to foods, which should be considered carefully, when building or operating a food plant. The European Union (*EU*) is developing uniform food legislature for its 15 member-countries (as of 2002).

Current Good Manufacturing Practices (*cGMPs*) cover a wide spectrum of manufacturing practices, but the main emphasis is on food plant hygiene (sanitation), while food quality receives the proper consideration (Troller, 1993).

In the design and layout of food plants, the following aspects related to *GMPs* should be taken into consideration: 1. Single floor versus multi-story buildings; 2. land space for future expansion; 3. waste disposal; and 4. building details (drainage, doors, lighting, ventilation, plumbing). Regulations similar to the *GMPs*, related to the design of food plants processing meat and poultry, are administered by the *USDA*.

Both *GMP* and *USDA* require adequate space for equipment installation and storage of materials, separation of operations that might contaminate food (cross-contamination), adequate lighting and ventilation (Saravacos and Kostaropoulos, 2002).

Process utilities (steam, water, and refrigeration) must be placed in separate rooms, and the process fluids transported to the processing equipment through overhead piping. Special treatments are needed for plant floors (tiles, polymer resins, and sealed concrete). Although the major hygienic hazard in food processing plants is microbial contamination, plant design should also provide for elimination of various pests from food areas, like insects, rodents, and birds.

Plant design should consider cleaning of food processing equipment and buildings, with appropriate preparation room for cleaning solutions and *CIP* piping (Chapter 2).

b. Food Safety Programs and HACCP

The need for uniform standards in world trade has led to the adoption of international standards, such as the series of *ISO 9000*, which detail the quality assessment procedures for industrial products in general. Food quality usually refers to the nutritional, sensory, compositional, and convenience attributes of food products. Sometimes, food quality includes food safety, which refers to the absence of microbial, chemical, biological or physical hazards.

Food safety programs are required for securing food safety and for complying with the regulations of government and international organizations. A food safety program consists of documents, records, systems and practices, including *HACCP*. Most modern food safety programs are implemented by the Hazard Analysis Critical Control Point (*HACCP*) system. The *HACCP* system was introduced first in 1989 by the U.S. National Advisory Committee on Microbiological Criteria for Foods (Gould, 1994). It is a system that identifies, evaluates and controls hazards, which are significant to the production of safe food. *HACCP* was first applied to meat, poultry, and dairy products, which are sensitive to microbial spoilage and hazards. It has been extended to most other food products.

The *HACCP* system is based on 7 principles, which detail the inspection and control procedure for a food processing plant (Codex Alimentarius, 1997; *NZIFST*, 1999). They include determination of the critical control points (*CCPs*), establishment of critical limits for each *CCP*, establishing a system to monitor each *CCP*, establishment of corrective action and procedures for verification of the system, and documenting all procedures for the given application.

Prerequisite tasks, needed for successful application of the *HACCP* programs are: Assembly of the *HACCP* team, description of the food and its distribution, intended use and consumers of the food, development of the flow diagram which describes the process, and verification of the flow diagram (*NACMCF*, 1997). Computer software is available for performing effective *HACCP* and food safety surveys (Mermelstein, 2000; Mortimore and Wallace, 2001).

In addition to GMPs and *HACCP*, the Food Safety Objectives (*FSO*) were proposed recently by the International Commission on Microbiological Specifications and Codex Alimentarius for risk management (Busta, 2002).

Proper plant design is a prerequisite for an effective *HACCP* program (Kvenberg, 1996). Consideration should be given to eliminating or substantially reducing the potential hazards. The following factors are important for effective design: 1. Product flow through the processing system without cross-contamination, 2. Prevention of contamination of foreign bodies, 3. Restriction of employee traffic, and 4. Positive air pressure in the processing areas.

V. QUALITY CHANGES IN FOOD PROCESSING

Food Processing involves several mechanical, thermal, and mass transfer operations, which are used to transform, preserve, and manufacture various food products (Potter and Hotchkiss, 1995; Fellows, 1990; Brennan et al., 1990).

Food processing and storage may cause significant changes in food quality, such as losses of color, texture, flavor, and nutritive value. Quantitative quality changers can be modeled by simple kinetic equations, which have been applied for the important food processes of thermal processing (sterilization, pasteurization) and dehydration. Quality changes are usually expressed by first-order kinetics as function of temperature, moisture content, water activity, and gas composition (Karel, 1983; Taoukis et al., 1997).

New food preservation processes, such as aseptic processing and high-pressure preservation, should be evaluated with respect to both food safety and food quality.

The design of thermal food processes is based on maximizing nutrient retention, while assuring an acceptable reduction of the spoilage and pathogenic microorganisms (Chapter 8). Time-temperature combinations of the first-order kinetic model are expressed by the two parameters k and E of Equations (1-1) and (1-3). Optimization of the sterilization processes is possible because microbial inactivation is much faster than nutrient retention (basis of HTST and UHT processes). In thermal processes, the parameters k and E are replaced by D and z , respectively, as defined in Equations (1-8) and (1-9). Table 1.2 shows typical D , E , and z values for the thermal degradation of food components (Lund, 1977; Karel, 1983).

Using the z -value instead of E should be confined to small temperature ranges, since significant errors may occur in large differences (Datta, 1993).

Table 1.2 Typical Degradation Rate Parameters in Thermal Processing

| Component | Temperature, °C | D_{121} , min | E , kJ/mol | z , °C |
|------------------|-----------------|-----------------|--------------|----------|
| Thiamin | 110 - 150 | 150 | 110 | 45 |
| Ascorbic acid | 50 - 100 | - | 25 | 70 |
| Chlorophyll | 80 - 140 | 150 | 80 | 50 |
| Browning | 40 - 130 | 300 | 100 | 50 |
| Enzymes | - | 10 | 200 | 30 |
| Vegetative cells | - | 0.003 | 400 | 5 |
| Bacterial spores | - | 1 | 250 | 10 |

Data from Lund (1977) and Karel (1983).

The traditional calculations of thermal process time are based on the lethality of the target microorganism at the coldest spot (lowest heating point) in the packaged food. The lethality F_c at the coldest spot is calculated from the equation (Stumbo, 1973):

$$F_c = \int_0^t \frac{T - T_o}{z} dt \quad (1-10)$$

where T_o is the reference temperature (121°C for sterilization) and z is the temperature factor for sterilization (usually $z = 10^\circ\text{C}$).

The integrated sterility value F_s , representing the volume average survival of the target microorganism is given by the equation (Silva et al., 1992):

$$F_s = -D_o \log \left(\frac{N}{N_o} \right)_{av} = -D_o \log \left(\frac{1}{V} \int_0^V 10^{-\frac{F_c}{D_o}} dV \right) \quad (1-11)$$

where N_o is the initial microbial population, N is the population after time t , and V is the volume of the food material.

In thermal processing the “cook value” C is an overall index of quality degradation, and it can be estimated from the models used for microbial inactivation.

The cook value at the coldest spot is defined by an equation similar to (1-10):

$$C_c = \int_0^t \frac{T - T_o}{z} dt \quad (1-12)$$

In quality degradation the reference temperature may be taken as $T_o = 100^\circ\text{C}$, and the temperature factor is about $z = 50^\circ\text{C}$.

The integrated cook value C_s , representing the volume average quality degradation is given by the equation:

$$C_s = -D_o \log \left(\frac{C}{C_o} \right)_{av} = -D_o \log \left(\frac{1}{V} \int_0^V 10^{-\frac{C_c}{D_o}} dV \right) \quad (1-13)$$

where C_o is the initial cook value, and C is the cook value after time t .

Low cook values are normally obtained in HTST and UHT thermal processing of foods, since the cook D_o value is relatively small, and the temperature factor is large.

NOMENCLATURE

| | |
|-------------|--|
| b | Constant |
| B | Relative growth rate |
| C | Concentration |
| C | Number of log cycles of the microbial growth |
| C_1 | Empirical constant |
| C_2 | Empirical constant |
| C_c | Cook value |
| C_o | Initial cook value |
| C_s | Integrated cook value |
| D | Decimal reduction time |
| E | Activation energy |
| F_c | Lethality |
| F_s | Integrated sterility |
| k | Reaction constant |
| k_g | Reaction constant at temperature T_g |
| k_o | Reaction constant at temperature T_o |
| M | Time at which the microbial growth rate is maximal |
| N | Microbial density |
| N_o | Initial microbial density |
| R | Gas constant |
| T | Temperature |
| t | Time |
| $t_{1/2}$ | Half-time or half-life |
| T_g | Glass transition temperature |
| T_{min}^* | Theoretical minimum temperature for growth |
| T_o | Reference temperature |
| V | Volume |
| z | Thermal resistance factor |

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2

Principles of Food Process Design

I. INTRODUCTION

Process Design was developed as a component of Chemical Engineering and it is applied mainly in the chemical process industries. It is based on unit operations, transport phenomena, reaction engineering, process control, and process economics. Process Design uses empirical techniques, based on long experience of operating plants. The recent trend is for application of fundamental physical, chemical, and engineering principles, use of computer modeling, and molecular and process simulations (Edgar, 2000).

Process Design has been applied successfully to the design, construction, and operation of continuous large processing plants, handling mostly homogeneous gas and liquid materials, where the required physical property data are available or can be predicted using reliable techniques. Application of process design to solids processing and to heterogeneous systems is more difficult, because of the limited theoretical knowledge of solids and heterogeneous processes, the inadequate data on physical properties, and the empirical nature of most solids processing equipment (Peters and Timmerhaus, 1991).

Modern Process Design is part of Process Systems Engineering, and it is extended to heterogeneous, structured, and formulated materials in continuous, batch, and flexible processes. The characteristics of both process and product are considered at the molecular, nano-, micro-, and macroscopic scale. Product quality and safety are emphasized, in addition to the conventional engineering considerations of energy, process cost, and environmental impact. Detailed process modeling and model validation are essential before engineering design and construction of the processing plant.

The recent trends in Process Design are directly applicable to the developing field of Food Process Design, which deals mostly with solids, semi-solids, and heterogeneous materials, emphasizing the quality and safety of the processed food products.

Process Design is used in the design of entirely new (grass-roots) plants, but mostly in the expansion of existing production facilities, and in plant upgrading and modernization. New and improved processing facilities are necessary to meet the demands for new products, improved quality, process control, energy saving,

and strict safety and environmental requirements. Economics and profitability must always be taken into account when designing new or upgraded processing plants.

Process and Plant Design are the basic parts of a Feasibility Study of an industrial project. The design and construction of an industrial processing plant involve the stages presented in Table 2.1.

Process and plant design are based on the following traditional engineering disciplines:

Chemical Engineering, which is concerned with the physical and engineering properties of materials, construction of process flowsheets, material and energy balances, equipment sizing, and plant utilities.

Mechanical Engineering, which deals with detailed engineering design of process and utility equipment, piping and material transport equipment, installation and maintenance of industrial equipment, and heating / air conditioning of industrial buildings.

Electrical Engineering is concerned with electrical power (e.g., motors), industrial lighting, process control, and automation of the processing plant.

Industrial Engineering is concerned with efficient plant operation, better utilization of material and labor resources, time-motion studies, and application of the various occupational and public health regulations at the local, state, and federal levels.

The design of food processes and food processing plants is based on the same principles of process (chemical) design, with the additional requirements for food safety and food quality. Food processes, processing equipment, and processing plants must comply with strict hygienic (sanitary) regulations.

Optimization of food processes is based on maximum preservation effect with minimum quality damage to the food product, and minimum health hazard to the consumers.

Table 2.1 Design and Construction of Industrial Plant

1. Feasibility Study
 - a. Process Design
 - b. Plant Design
 2. Engineering Design
 - a. Process Equipment
 - b. Plant Utilities
 3. Plant Construction
 - a. Civil Engineering Works
 - b. Installation of Equipment
 - c. Plant Start-up
-

The purpose of Process Design is to meet the needs of an industrial plant for economic production of one or more products. Process Design determines the required type and size of equipment, and operating conditions, which will realize the objectives of a particular process. The results of process design are used in the optimization of the particular process and in the detailed engineering design, construction, and operation of processing plants.

Process Design involves the following stages (Figure 2.1): (1) Selection of the proper flowsheet to realize the required production; (2) material and energy balances, which are specifying the process requirements of the plant; (3) sizing and rating of the required industrial process equipment; (4) cost estimation; (5) financial and profitability analysis; (6) parametric optimization; and (7) structural optimization of the process.

Flowsheets and material balances are discussed with respect to the design of food processes (Section II). Sizing of process equipment is discussed in the various chapters of this book, and in the specialized literature (Perry and Green, 1997; Walas, 1988; Saravacos and Kostaropoulos, 2002). Practical aspects of plant and equipment design are discussed by Bhartia (1979-1983), and Sandler and Luckiewicz (1987). Mathematical modeling and process optimization is discussed in Chapter 3 of this book.

The most commonly used food processes are classified and discussed in Section II, while the special requirements of food plants, related to product quality and safety are discussed in the Section III of this chapter.

Plant Design involves detailed engineering and construction of process equipment, utilities, buildings, storage facilities, and waste treatment. Some important aspects of Plant Design are discussed briefly in Section IV, while the process economics are presented in Section V of this chapter.

Most of the process and plant design procedures were developed in the chemical process industries, where large quantities of gases and/or liquids are processed continuously into a small number of major products. The same principles can be adapted to batch processes and to processing of solids, which characterize most food processing industries (Douglas, 1988).

Design projects that are related to laws and regulations, product safety, employee welfare and accidents, natural disasters, and so forth, are not, in general, evaluated only on the basis of financial profits. This is particularly important in food processing operations, where product safety and quality to the consumers are of primary importance.

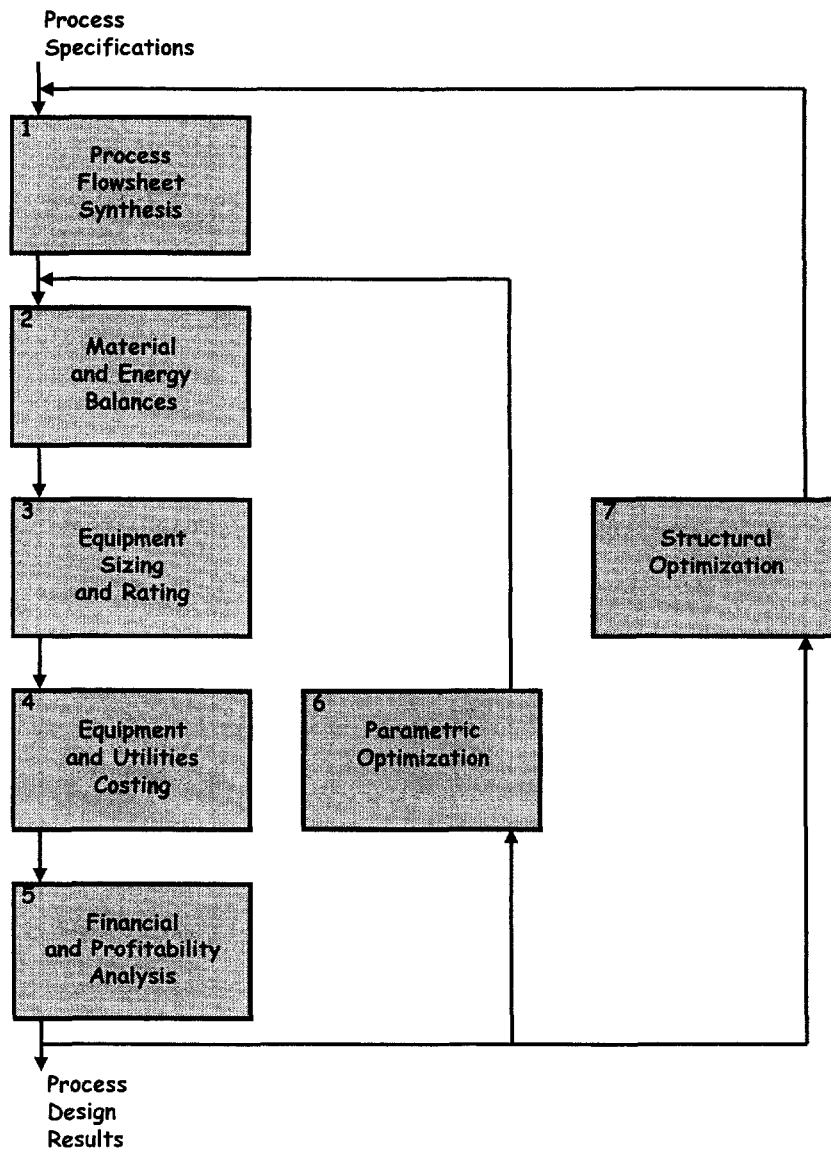


Figure 2.1 Information flow diagram of process design.

II. DESIGN OF FOOD PROCESSES

1. Introduction

The food processing industry has developed through the years from small, specialized food factories to large processing plants, based mostly on empirical experience, and supported, whenever necessary, by the principles of the underlying sciences of Chemistry and Microbiology. The engineering disciplines, mainly Mechanical and Civil Engineering, were utilized in the construction of food processing equipment and plant facilities. During the recent years, Chemical Engineering has entered into the design, operation and control of food processes, through the application of the successful industrial concepts of Unit Operations, Transport Phenomena, Process Design and Process Control.

Systematic Process Design is being adopted in the design of food processes, replacing the empirical approaches of the past. In addition to the principles and techniques of Chemical Process Design, the design of food processes must be based on the principles and technology of Food Science and Engineering.

Successful and efficient manufacturing technologies, developed in other industries, can be adapted, modified, and implemented in the food industry. Application of the engineering principles and techniques, used extensively in the process industries, must be accompanied by food safety and quality considerations, which have an overriding role in the overall process design and plant operation. While the terms “processes” and “operations” are used interchangeably in the literature, a process may involve some chemical, biochemical or biological reactions in addition to the strictly physical (or mechanical) operation.

Food processing involves several physical unit operations and microbiological, biochemical and chemical processes, which aim at preservation and improvement of food quality, or conversion to safe and nutritional food products in large, economic scale. Food preservation and conversion technology has advanced considerably during the recent years (Fellows, 1990; Heldman and Hartel, 1997).

Food Engineering has evolved into an interdisciplinary area of Applied Science and Engineering, based primarily on Chemical Engineering and Food Science. The traditional unit operations of Chemical Engineering have been adapted to Food Processing, taking into consideration the complexity of food materials and their sensitivity to processing conditions (Leniger and Beverloo, 1975; Loncin and Merson, 1979; Heldman and Lund, 1992; Valentas et al., 1997; Ibarz and Barbosa-Canovas, 2002).

The physical operations of food processing can be analyzed by applying the established concepts of unit operations and transport phenomena of Chemical Engineering (Gekas, 1992; Fryer et al., 1997; Welti-Chanes et al., 2002). In addition to the traditional engineering considerations of process cost, energy optimization and process control, demands on food quality and safety should be satisfied.

The need for improved product quality in all industries (Product Engineering) should be taken into consideration in all stages of Process Design. In the food industry, advances in the new field of Food Materials Science should be considered, in respect to the effect of food handling, processing and storage on the structure, physical properties, and quality of food products (Aguilera, 2000).

Process Design involves basically material, energy, and economic balances. Most food engineering data, such as physical and transport properties, biochemical and microbiological kinetics, are not very accurate, and rough models are used. Approximate estimates may be sufficient for a feasibility study of a particular process or for comparing one process over another (Zaror and Pyle, 1997).

Food process design must integrate processing, packaging, and materials handling for conventional and new technologies and products. Food processes and plants must meet the specific requirements of the food industry for hygienic (sanitary) standards, food quality, process control, and maintenance.

Food safety is a strict requirement in all aspects of food process design. Potential health hazards should be considered at the design and specifications of processing equipment and food plant. Good manufacturing practices (*GMP*) and *HACCP* standards should be considered at the design stage (Chapter 1).

Process simulations and computer techniques, discussed in Chapter 3, can be used to explore the effect of engineering and economic parameters, such as temperature, flow rate, and heat transfer coefficients, on the process performance (Fryer et al., 1997; Georgiadis et al., 1997).

Process control and automation, adapted from other industries, must take into consideration the requirements of accurate control of safe thermal processing, time-temperature effects on product quality, and desired micro and macro-structural properties of food products.

Plant layout of the processing equipment should be based on the requirements of materials flow, hygienic operation, access to equipment, process control, and maintenance. An optimized design layout should combine the equipment needs with the architectural, civil, structural, and environmental requirements.

Process and plant design procedures should comply with the relevant national and international standards, e.g., *BS 5750*, *ISO 9000*.

2. Unit Operations in Food Processing

The basic Unit Operations of Chemical Engineering, i.e., Fluid Flow, Heat Transfer, and Mass Transfer, have been applied to the food processing industry for many years. The theory on these operations was developed originally for gases and liquids (Newtonian fluids), but in food processing (or food manufacturing) non-Newtonian fluids, semi-solid and solid food materials are handled, and adaptation or extension of the theory is necessary. Some food processing operations, dealing with such complex materials are still treated empirically, using rules, practices and equipment developed through experience (Brennan et al., 1991). Generalized models of unit operations were analyzed by Diefes and Okos (1997).

Several specialized unit operations were developed in the food processing industry, and more than 150 such food processing operations were listed by Farkas (1977, 1980).

For the purposes of this book, the unit operations of food processing are classified on the basis of the processing equipment, with typical examples shown in Table 2.2 (Saravacos and Kostaropoulos, 2002).

The mechanical transport operations include pumping of liquids, pneumatic and hydraulic transport, and mechanical conveying. Pumping can be modeled and simulated, based on the principles of Fluid Flow and the Rheology of fluid food

materials. However, the other mechanical transport operations are based on empirical rules and specialized equipment, developed through experience of manufacturers of equipment and industrial food processors.

Mechanical processing operations constitute a very important part of food processing, dealing mostly with solid and semi-solid materials. Modeling and simulation of these operations is difficult, and empirical rules and equations are sometimes used. Size reduction, agglomeration, mixing, and extrusion, developed in the chemical process industries, are adapted and applied to various food processes. Sorting, grading, peeling, slicing, expression, and forming require specialized equipment, which has been developed for the various food products and processes (Saravacos and Kostaropoulos, 2002).

General mechanical separations, such as screening, filtration, and centrifugation, have been adapted from the chemical process industries to various food processes. Cleaning and washing are empirical operations, using specialized equipment, developed for specific food materials.

Most of the heat and mass transfer processes used in food processing can be modeled and simulated, using established techniques of Chemical Engineering. Novel non-thermal preservation methods, e.g., irradiation and high pressure processing, are still in the development stage, and specialized process design is required. Food packaging operations are highly specialized, they are difficult to model and simulate, and they are based on the selection of appropriate equipment.

Food quality considerations are very important in the selection and operation of processes. Food materials can be considered as either living or non-living plant (or animal) tissues. Fresh fruits and vegetables are considered as living tissues. In most food processing operations, the food materials consist mostly of non-living tissues (Farkas, 1980).

A practical description of the unit operations, used in the processing of fruits and vegetables, was presented by Gould (1996). Fruit and vegetable processing is a large industry, consisting of a large number of small to medium-sized processing plants, and producing several diverse food products. These plants utilize several unit operations, since the materials being processed are solid or semi-solids, sensitive to mechanical and thermal processing. On the other hand, the dairy, edible oil, milling, and beer industries deal with large amounts of fewer products, utilizing a smaller number of conventional unit operations.

The scale-up methods, used extensively in Chemical Engineering, are difficult to apply, even to continuous food processing operations, due to insufficient physical property data, and the complex physical, chemical and biological changes in the food systems. Pilot plant data, under similar processing conditions, are necessary for scale-up of industrial operations of complex food processes, like extrusion cooking of starch-based foods (Valentas et al., 1991), or processing of new foods.

The pilot plant is useful in determining new food processes, and in testing new processing equipment under industrial operating conditions. It is often used for the production of large samples of new food products, used for storage and marketing tests.

Table 2.2 Classification of Unit Operations of Food Processing

| Group of Operations | Typical Food Processing Operations |
|--------------------------|--|
| Mechanical Transport | Pumping of Fluids Pneumatic Conveying Hydraulic Conveying Mechanical Conveying |
| Mechanical Processing | Peeling, Cutting, Slicing Size Reduction Sorting, Grading Mixing, Emulsification Agglomeration Extrusion, Forming |
| Mechanical Separations | Screening Cleaning, Washing Filtration Mechanical Expression Centrifugation |
| Heat Transfer Operations | Heating, Blanching Cooking, Frying Pasteurization Sterilization Evaporation Cooling, Freezing, Thawing |
| Mass Transfer Operations | Drying Extraction, Distillation Absorption, Adsorption Crystallization from Solution Ion Exchange |
| Membrane Separations | Ultrafiltration Reverse Osmosis |
| Non-Thermal Preservation | Irradiation High Pressure Pulsed Electric Fields |
| Packaging | Filling, Closing Metallic, Plastic Packages Aseptic Packaging |

3. Food Process Flowsheets

In Food Process Design, flowsheets similar to those of Chemical Process Design are used, i.e., process block diagrams (*PBD*), process flow diagrams (*PFD*), process control diagrams (*PCD*), and process instrumentation and piping diagrams (*PID*).

The selection of a process flowsheet in the chemical and petrochemical industries requires extensive computer calculations and simulations, due to the large number of possible process configurations.

The selection of most food process flowsheets is confined to a limited number of alternatives, due to the well-defined basic process, e.g., sterilization, evaporation, drying or packaging (Saravacos and Kostaropoulos, 2002). Auxiliary equipment, such as heat exchangers, can increase the number of alternatives, e.g., for energy optimization, in parts of the basic flowsheet. Food processes handling liquid foods, such as milk and vegetable oils, require extensive piping and more complex flowsheets.

Two-dimensional (*2D*) flowsheets are mainly used for various process equipment, and plant representations. In special cases, three-dimensional (*3D*) diagrams offer a better visualization of complex processing equipment or processing plants (CE, 1999).

Process block diagrams (*PBD*) are widely used for simple representation of the process, and for preliminary calculations of material and energy balances. Each rectangular block represents individual unit operations or group of operations. The process flow diagrams or process flowsheets (*PFD*) show more details of the process or plant, using specific symbols for equipment, piping, and utilities.

Both *PBD* and *PFD* flowsheets can show process details, like material flow rates (kg/h), energy flows (kW), temperatures ($^{\circ}\text{C}$), and pressures (bar), and they can be combined with tables of data.

The process control diagrams (*PCD*) show the position of the control units in the processing lines, and their connection to the sensors. The process instrumentation and piping diagrams (*PID*) indicate the type and location of instrumentation, and the type and connections of pipes. In addition to the *PBD* and *PFD*, layout diagrams, showing the position of the processing equipment in the food plant are used.

For illustrative purposes, a block diagram, a process flowsheet, and a layout diagram for the same food processing plant are shown in Figures 2.2, 2.3, and 2.4. The plant chosen is a tomato-processing facility, producing tomato paste, involving several unit operations and processes and a variety of processing equipment, which will be analyzed in detail in the examples of subsequent chapters of this book. Figure 2.5 shows a *3D* flowsheet for the same tomato-paste-processing plant for better visualization of the plant and equipment (Saravacos and Kostaropoulos, 2002).

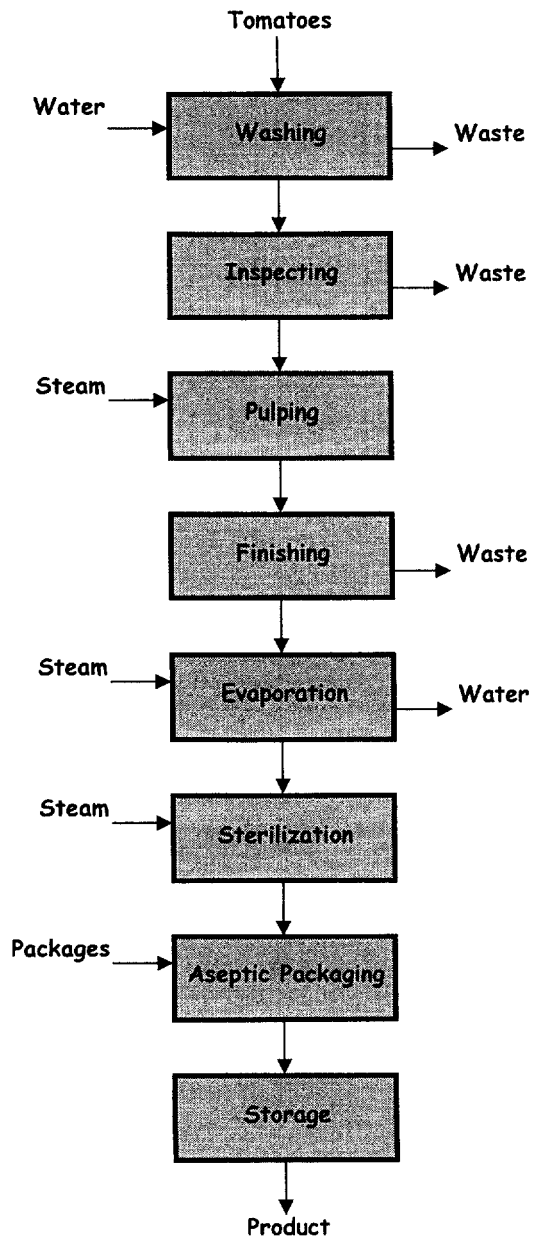


Figure 2.2 Simplified process block diagram for tomato-paste-processing plant.

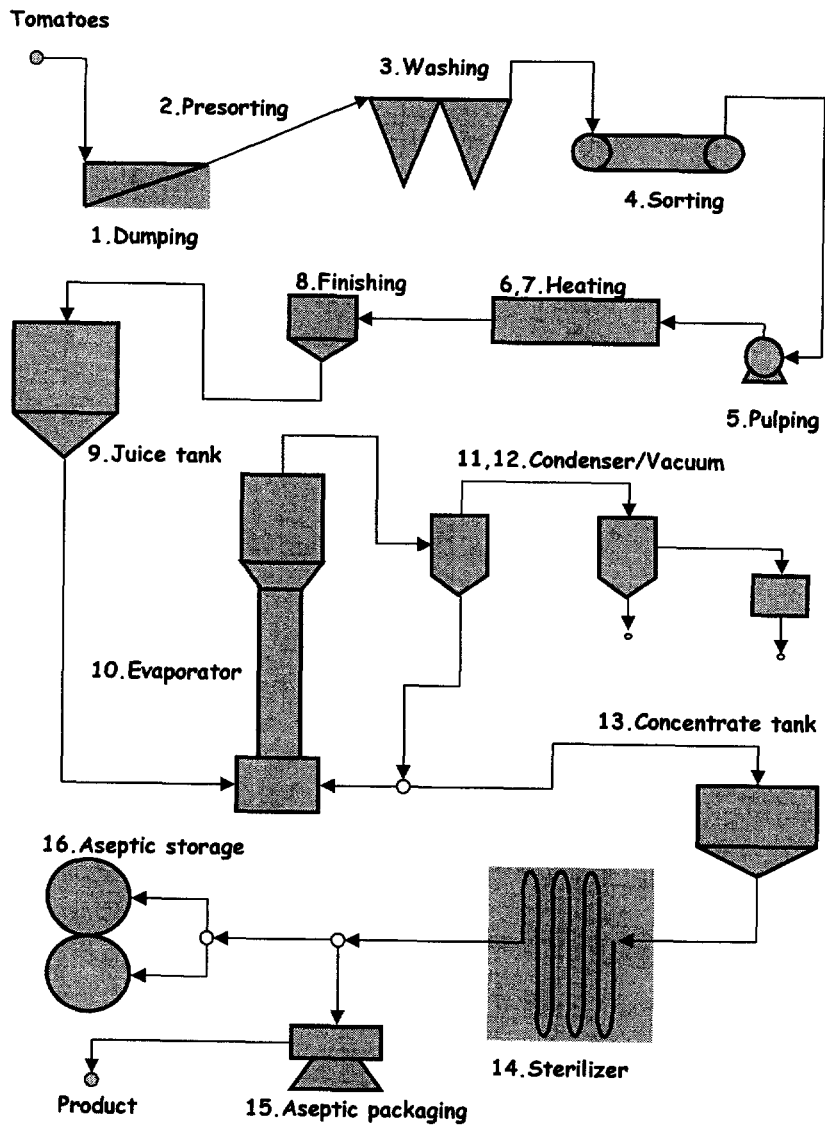


Figure 2.3 Simplified process flow diagram for tomato-paste-processing plant (see Figure 2.2).

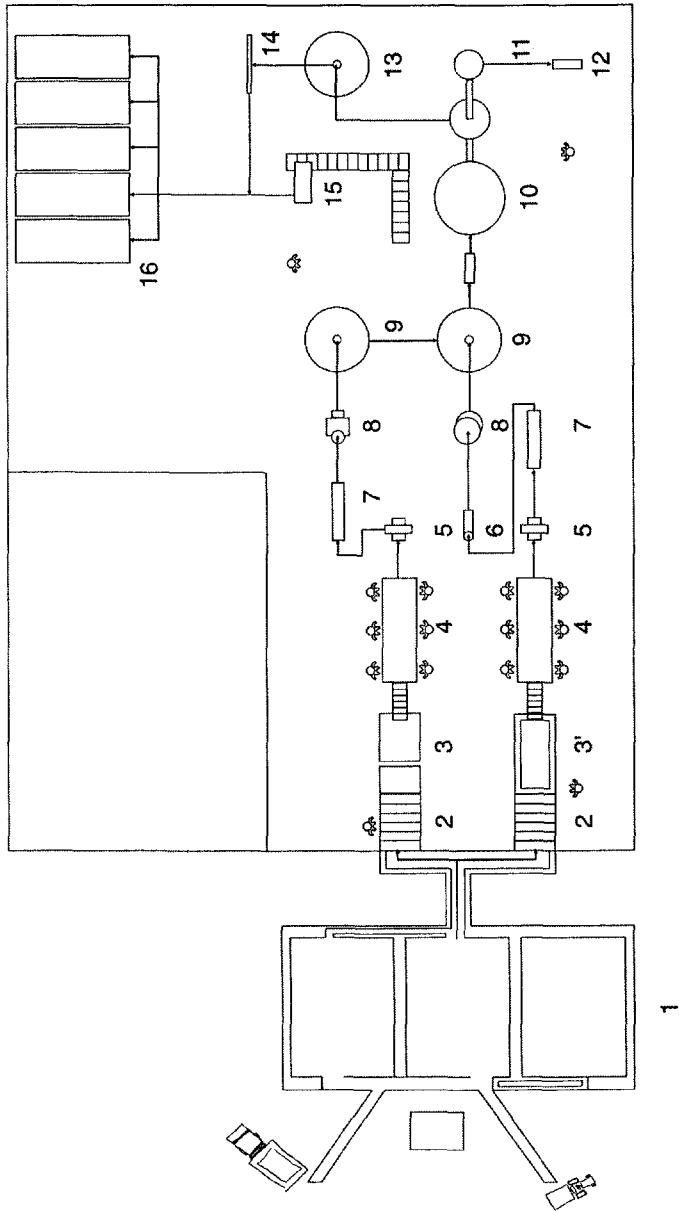


Figure 2.4 Floor plan (layout of equipment) of a tomato-paste plant (see Figure 2.3), Saravacos and Kostaropoulos (2002).

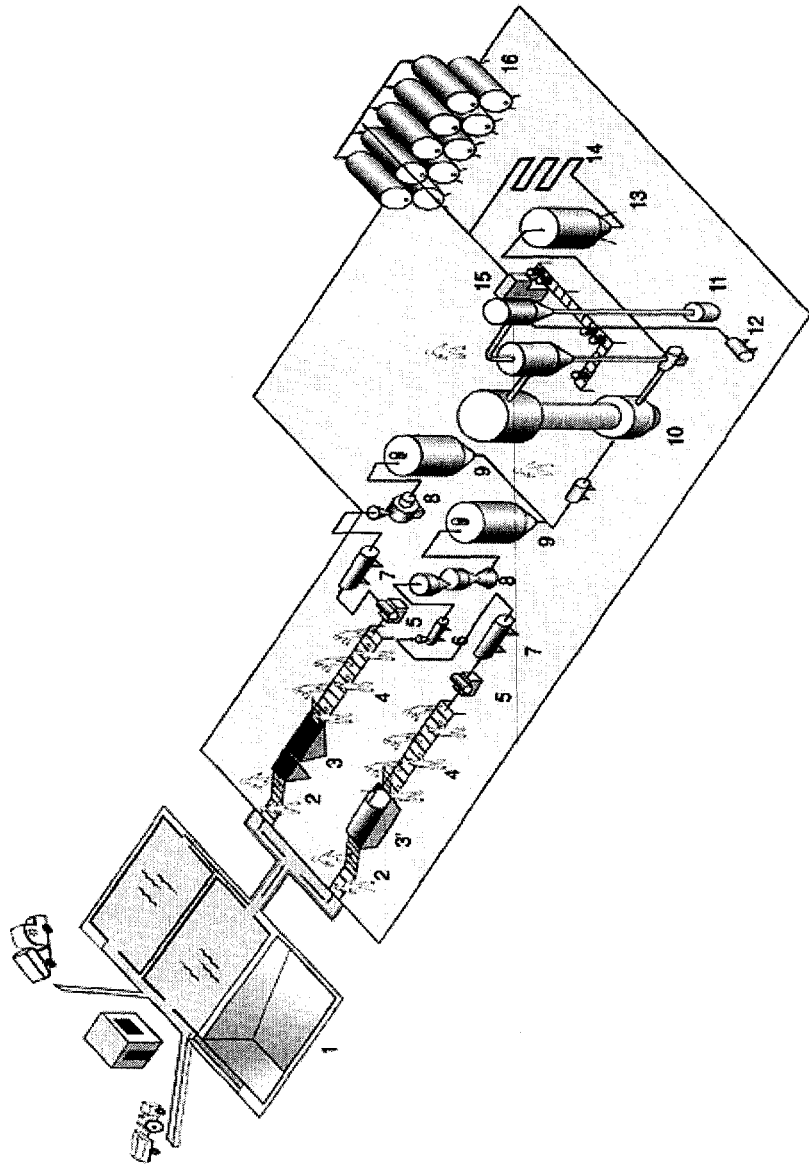


Figure 2.5 Simplified 3D process flow diagram for a tomato-paste-processing plant (see Figure 2.3), Saravacos and Kostaropoulos (2002).

4. Material and Energy Balances

The material and energy balances used in Chemical Engineering are applicable to most food process calculations. However, food processes require special attention, due to the complexity of food materials, and the importance of minor food components to food quality. Variations in food composition due to variety, growing conditions, and age of the raw materials, may be significant. Required food composition data can be obtained directly by chemical analysis, or taken from the literature, e.g., the *USDA* food composition data (Watt and Merrill, 1963). Due to the significant variability of some raw food materials, material and energy balances may be required periodically, during the operation of the food processing plants, e.g., in the seasonal processing of fruits and vegetables (Farkas and Farkas, 1997).

Overall and component material balances are calculated at the boundaries of a food process, from the mass conservation equations in the system (Saravacos and Kostaropoulos, 2002):

$$(total\ mass\ in) - (total\ mass\ out) = (total\ mass\ accumulated)$$

$$(component\ mass\ in) - (component\ mass\ out) = (component\ mass\ accumulated)$$

Material balances are simplified in continuous operations, since the accumulated materials (total and component) are equal to zero.

The component material balance can be written for one or more food components, which are important in a given processing operation. Typical components, involved in food processing, are water (moisture), total solids (*TS*), soluble solids (*SS*), fat, oil, salt, and protein. The soluble solids are usually expressed as °Brix (% sucrose by weight), measured with refractometers, which are used widely in the laboratory and the processing plant. The concentration of components is normally expressed as mass or weight fractions. Volumetric flows should be converted to mass flows using the density of the material.

Energy balances are calculated at the boundaries of a food process, from the energy conservation equation (first law of Thermodynamics) in the system:

$$(total\ energy\ in) - (total\ energy\ out) = (total\ energy\ accumulated)$$

For most process design calculations and equipment sizing, the main energy form considered is heat, and only heat balances are calculated. The mechanical and electrical requirements for pumping, transportation, refrigeration, and operation of the various pieces of process and utility equipment are considered in the detailed process, equipment and plant design. Heat balances involve the enthalpy and specific heats of the various process and utility streams.

Thermophysical and thermodynamic data for foods are obtained from food engineering and food properties handbooks and databases (Rahman, 1995; Rao and Rizvi, 1995; Saravacos and Maroulis, 2001). Transport properties of foods are of particular importance in food process and equipment design. Typical physical properties of foods are given in the Appendix.

The material and energy balances are essential in the design of food processes, processing equipment, process utilities, and waste treatment facilities, in process optimization and control, and in cost analysis of the process and the processing plant.

Detailed material and heat balance calculations are given in Example 2.1 at the end of this chapter.

5. Mechanical Processes

Mechanical processing operations are very important in food processing. They include mechanical transport of food materials, mechanical processing, and mechanical separations. Mechanical transport includes pumping and mechanical conveyors; mechanical processing is concerned with size reduction, agglomeration size reduction, mixing, and forming; and mechanical separations involve filtration, centrifugation, and expression.

Mechanical operations are based on equipment, specific for various processes, developed mostly from experience of equipment manufacturers and users in the chemical and food processing industries. Some empirical equations are applied to specific processes and equipment, but exact mathematical modeling and simulation of the mechanical processes is not feasible, except for pumping, which is based on Fluid Flow and Rheology. Description of mechanical processing equipment, in general, is found in Perry and Green (1997), Walas (1988), and Bhatia (1979-1983), while equipment used in food processing is described by Saravacos and Kostaropoulos (2002). Details of mechanical equipment can be found in bulletins and catalogs of manufacturers and suppliers of processing equipment. Lists of directories of equipment suppliers are by Saravacos and Kostaropoulos (2002).

The equipment used in mechanical processing of foods must comply with the principles of hygienic design and operation. The equipment surfaces coming into contact with the food should be made of corrosion-resistant stainless steel, and should be cleanable with the cleaning in place (CIP) system (Jowitt, 1980; Troller, 1993).

a. Mechanical Transport Operations

Liquid and semi-fluid foods are transported in food processing plants using various types of pumps, while food particles (powders) and grains are transported by pneumatic conveyors. Hydraulic conveying is used for large food pieces, while various types of mechanical conveyors are used for solid foods, food packages and containers.

i. Transport of Fluid Foods

Rheological data are required for the design of pumping of fluid foods in piping systems. Most fluid foods are non-Newtonian fluids (Saravacos and Maroulis, 2001), and the power-law model yields useful parameters for the estimation of the Reynolds number, the pressure drop, and the energy requirements (Bernoulli equation). Friction losses in pipes and fittings are estimated using empirical equations (Holland and Bragg, 1995; Saravacos and Kostaropoulos, 2002).

The design of most food processes is concerned mainly with heat and mass transfer operations, and the equipment required to carry out these basic operations, e.g., sterilizers, evaporators, and dryers. Transport of food materials is considered

an auxiliary operation, which is carried out with empirically-designed specialized equipment.

Table 2.3 shows some important fluid transport equipment used in food processing.

ii. Mechanical Conveyors

Several types of mechanical conveyors are used in transporting solid foods, food packages, and containers. Typical units are listed in Table 2.4:

b. Mechanical Processing Operations

Mechanical processing equipment, used in chemical and mineral processing, such as grinders, mills, agglomerators, and mixers, has been adapted to food processing. Specialized equipment, such as homogenizers, paste and dough mixers, and forming/extrusion, has been developed by equipment manufacturers for specific food processes and products.

Some typical mechanical processes, used in food processing, are shown in Table 2.5.

c. Mechanical Separation Operations

A wide range of mechanical separation equipment is used in food processing. Some basic equipment, such as screens, filters, centrifuges, and cyclones, are adapted to the food industries. Specialized equipment, such as sorters, peelers, and juice extractors, has been developed for specific food processes and food products.

A list of mechanical separation equipment is shown in Table 2.6.

Table 2.3 Fluid Transport Equipment

| Equipment | Applications |
|--|--|
| Pumps | |
| Centrifugal | Low viscosity fluids |
| Radial, axial flow | Dilute suspensions |
| Positive Displacement | |
| Lobe, gear, piston, progressive cavity | Viscous, sensitive fluids and pastes |
| Pneumatic Conveyors | Particles and grains suspended in air |
| Hydraulic Conveyors | Fruits and vegetables suspended in water |

Table 2.4 Mechanical Conveyors

| Equipment | Applications |
|------------------|-----------------------------|
| Belt Conveyors | Particles, pieces, packages |
| Roll Conveyors | Packages, heavy products |
| Screw Conveyors | Pastes, grains |
| Chain Conveyors | Containers |
| Bucket Elevators | Particles, grains |

Table 2.5 Mechanical Processing Operations

| Equipment | Applications |
|--|--|
| Cutting Equipment Slicers, dicers | Fruits / vegetables meat |
| Grinders Roll mills, hammer mills disc grinders, pulpers | Cereal grains fruits / vegetable, meat |
| Agglomerators Rotary pan, drum Fluidized-bed, drying Compression, pelletizing | Food granules from food powders |
| Homogenizers Pressure, colloid mills | Milk products food emulsions |
| Mixers Agitated tanks mixers of solids | Liquid/liquid liquid/solid solid/solid |
| Forming/Extrusion Equipment Forming extruders Twin extrusion cookers | Forming of cereal foods extrusion cooked products |

Table 2.6 Mechanical Separations

| Equipment | Applications |
|---|-------------------------------------|
| Screens Sieving equipment | Food particles, flour |
| Sorters | Sizing of fruits |
| Filters Cake filters, frame, vacuum depth filters | Fruit juices, wine water, air |
| Centrifuges Centrifugal separators filtering centrifuges, decanters | Milk, vegetable oil fruit juices |
| Cyclone Separators | Particles / air |
| Mechanical Expression Expression equipment, Screw presses, juice extractors | Vegetable oil fruit juices |
| Removal of Food Parts Peeling, pitting Skinning equipment | Fruit products animal products |
| Removal of External Parts Wet cleaners, air cleaners | Fruit/vegetables grain cleaning |

Table 2.7 Food Packaging Equipment

| |
|---|
| Container Preparation |
| Metal, glass, plastic, paper |
| Filling Equipment |
| Dosing, weighing, valves |
| Closing Equipment |
| Metallic containers, glass closures |
| Plastic containers, cartons and cardboard |
| Aseptic Packaging |
| Form-fill-seal equipment |
| Monoblock, combiblock systems |
| Group Packaging |
| Wrapping |
| Palletizing |

6. Food Packaging Processes

Food Packaging is highly dependent on packaging equipment, and there is little room for mathematical process modeling and simulation. Specialized equipment is used for container preparation, product filling and closing, and aseptic packaging of foods (Kostaropoulos and Saravacos, 2002).

Table 2.7 summarizes the main equipment used in Food Packaging.

III. FOOD SAFETY AND QUALITY

Food safety and quality are very important requirements of food process design and food processing operations. Food safety refers mainly to microbiological hazards from consuming foods, and secondarily to chemical and physical hazards. Food quality refers to the sensory and nutritive value of foods (Chapter 1).

Processing and packaging technologies and equipment must be designed to protect and ensure both food safety and quality. Developments in these areas are discussed in national and international conferences and symposia of scientific organizations. The Institute of Food Technology (*IFT*) organizes special conferences on food safety and quality (*IFT*, 2002), covering subjects such as predictive microbiology and rapid microbiological testing methods.

Food quality is of fundamental importance to any food process design. It distinguishes food process design from the design of other (chemical) processes. All other considerations of process design (principally economics, but also environmental and energy considerations) must be taken into account. In the final analysis, the food product must be wholesome, safe, and organoleptically acceptable to the consumer.

Food quality is an indispensable part of any food process design. The kinetics of quality deterioration is combined with the basic equations of heat and mass transfer in all food processes (see Chapter 1).

1. Plant Safety

Safety of personnel, products, and plant facilities should be considered in any process and plant design.

Plant safety is particularly important in the chemical and petrochemical industry, where major accidents and emissions of hazardous chemicals can cause serious injuries, loss of lives, and destruction of facilities (Frank and Arendt, 2002). Process safety regulations should be followed in the design and operation of any processing plant.

The following federal agencies regulate the safety of personnel and plant facilities in the USA: Occupational Safety and Health Administration, *OSHA* (Process Safety Management Standards, Hazardous Operations, *HAZOP*); Environmental Protection Agency (*EPA*); and National Fire Protection Agency (*NFPA*). In addition, the Center of Chemical Process Safety (*CCPS*) of the American Institute of Chemical Engineers (*AIChE*) is concerned with process safety.

There are specific requirements, recommended by various organizations, for fire hazards, electric motors, dust explosions, etc. Dust explosions are particularly important in processing and storage of food powders. Moisture-proof electric motors must be used in damp environments, such as canning. Explosion-proof electric motors should be used for dusty environments, such as grain mills and powder conveyors.

Steam boilers should be located in a separate boiler house to confine any explosion hazard. Plant layout and construction should prevent accidents caused on the personnel, e.g., special floor coatings, and protective rails in silos.

The noise level in the processing areas should be not too high to avoid health problems to the operators. Maximum noise levels, according to Directive 86 / 188 of the European Union, should not exceed 90 dB in 8-hour work, and 93 dB for 4-hour work near the noisy equipment. In some processing areas, the noise level may be excessive, for example 90-110 dB in a bottling plant. In such cases, the operators must take protective measures, e.g., using earmuffs.

Noise can be reduced by proper selection of equipment, better foundations and seating of equipment with moving parts, gentle conveying, and isolation of noisy equipment in special rooms.

In Food Processing, food safety (protection of the consumers' health) is of paramount importance, and it deserves special consideration over the general plant safety, outlined previously (see following sections).

2. Hygienic Food Process Design

Hygienic or sanitary design of food processes and food processing equipment is based on proper selection of construction materials and fabrication techniques, which will facilitate food processing and thorough cleaning of the equipment (Saravacos and Kostaropoulos, 2002).

Hygienic design of process equipment must be accompanied by a thorough hygienic design of the whole food process and processing plant (Jowitt, 1980; Brennan et al., 1990; Gould, 1994). Engineering implications of hygienic process design should be considered from the outset of the design process, especially for new food processing systems. The principles of food processing sanitation are described by Kutsuyama (1993). A guide to the sanitary design of food plants and

food processing equipment was published by Imholte (1984). Hygienic aspects of food processing equipment are discussed by Kessler (1981), and Kessler and Lund (1989).

In the European Union (*EU*), hygienic design of food equipment is developed by the European Hygienic Equipment Design Group (*EHEDG*, 1997). The *EU* research and development program LINK includes a project on advanced and hygienic food manufacturing, consisting of hygienic processing, and food process simulation and modeling.

3. Hygienic Standards and Regulations

The design and operation of food processes and processing equipment should ensure the microbiological safety of the final food products. Design engineers, equipment manufacturers and food processors should follow strict hygienic standards and government regulations.

Government regulations of food processing equipment are essential for the manufacture of safe and wholesome foods and the protection of Public Health. In the USA, the following Government Agencies and Private Organizations have published sanitary standards for food processing equipment (Troller, 1993):

- 1) *USDA* (US Dept. of Agriculture), Washington, DC: 1. Publ. MPI-2 "Accepted Meat and Poultry Equipment", 2. Agric. Handbook No. 191 "US Inspected Meat Packing Plants. A Guide to Construction, Equipment, Layout", 3. "Poultry Processing Equipment".
- 2) US Dept. of Interior, Washington, DC: Bureau of Commercial Fisheries "Fishery Products and Processing Equipment".
- 3) US Public Health Service, Washington, DC: FDA (Food and Drug Administration). *GMPs*.
- 4) *IAMFES* (International Association of Milk, Food, and Environmental Sanitarians, Inc.), Ames, Iowa: Committee on Sanitary Procedures "3-A Sanitary Standards".
- 5) *ASME* (American Society of Mechanical Engineers), New York: *ANSI-ASME* F2-1: "Food, Drug and Beverage Equipment".
- 6) *BISSC* (Baking Industry Sanitation Standards Committee), New York: "BISSC Sanitation Standards".
- 7) *AFDOUS* (Association of Food and Drug Officials of the United States), Littleton, CO: "AFDOUS Frozen Food Code".
- 8) National Sanitation Foundation, Ann Arbor, MI: 1. "Food Service Equipment Standards" 2. "Food Preparation and Service Equipment".

The 3-A sanitary standards were developed originally for the milk industry, but they have been extended to other food products in the *USA* and other countries. They resulted from the collaboration of equipment manufacturers, the users of food equipment, the International Association of Milk, Food, and Environmental Sanitarians, the Dairy and Food Industries Suppliers Association (*DFISA*), the Poultry and Egg Institute of America, and representatives of the *USPH/FDA* and *USDA*. A list of the 3-A standards and 3-A accepted practices is presented by Rao (1992). The 3-A standards refer mainly to milk processing equipment, including storage tanks, heat exchangers, pasteurizers, freezers, evaporators, drying equipment, and various fittings. The 3-A accepted practices include *HTST* pasteur-

izers, and culinary (potable) steam production. Special E-3-A standards apply to processing equipment for egg products (*IAMFES*, 1988).

In addition to the 3-A standards, the following two rules/regulations should be considered in the USA: The Pasteurized Milk Ordinance, and the Good Manufacturing Practices (*GMPs*) of the *FDA*.

Equipment used in USDA-inspected food plants must previously be approved and listed in the “Compendium of *USDA* Approved Equipment”. In addition to the US Federal Regulations, the Departments of Health of some states have specific requirements for dairy processing equipment (Clark, 1997a). Special caution is needed, when equipment used in a less regulated industry is applied to strictly regulated food industries, such as meat and poultry.

In the European Union (*EU*), certain general rules apply to the hygienic design of food processing equipment, in addition to the regulations of the individual member-countries. A review of the European regulations for hygienic design in food processing, with emphasis on milk, was presented by Grasshoff (1992). The European Hygienic Equipment Design Group (*EHEDG*) is concerned with the sanitation of food processing equipment (*EHEDG*, 1997).

The design of modern food processing equipment should be based on databases of hygienic requirements and regulatory standards. Special attention should be given to the sealing spots of moving parts, e.g., rotating and reciprocating shafts, where microbial contamination is possible (Hauser, 1992). Developments in hygienic design of food processing equipment and food plants in Europe are presented in the annual meetings of *EHEDG*.

The *EU* “Machinery Directives” (e.g., 89 / 392, and 91 / 368) specify that food processing machinery must be designed and constructed as to avoid any risk of infection and sickness. *EU* documents, such as CEN / TC 153 (CEN = European Standardization Committee, *TC* = technical committee) specify machinery, safety, and hygienic requirements for various food industries. The standard *CEN* 1672-2 is concerned about food machinery, safety, and hygienic requirements. The European food industry has, in general, adopted the US 3-A standards within the framework of the standards being developed by the *EHEDG*.

New regulations and standards for various food processing equipment are being published with the notation prEN (provisional European standards). Future requirements in the hygienic design of food factories were discussed by Curiel (2001).

Other specifications used in the food industry are the International Standardization Organization (*ISO*), the German Standardization Authority (*DIN*) requirements for fittings, the bulletins of the International Dairy Foundation (*IDF*), and the British standards BS 5750. The symbols “*CE*” (Conformite Europeenne) on equipment is used as evidence of compliance.

A number of guidelines have been published by *EHEDG*, which are voluntary and complementary to the corresponding national and international hygienic standards. The *EHEDG* guidelines include the following:

- 1) Microbiologically safe continuous pasteurization of liquid foods.
- 2) A method for assessing the in-place cleanability of food processing equipment.
- 3) Microbiologically safe aseptic packing of food products.
- 4) A method for the assessment of in-line pasteurization of food processing equipment.

- 5) A method for the assessment of in-line steam sterilizability of food processing equipment.
- 6) The microbiologically safe continuous flow thermal sterilization of liquid foods.
- 7) The EC Machinery Directive and food processing equipment.
- 8) A method for the assessment of bacteria tightness of food processing equipment.
- 9) Hygienic equipment design.
- 10) Welding stainless steel to meet hygienic requirements.
- 11) Hygienic design of closed equipment for the processing of liquid food.
- 12) The continuous and semi-continuous flow thermal treatment of particulate foods.
- 13) Hygienic design of valves for food processing.

The need for thorough hygienic design and operation of the entire food processing line is very important in food processing: A weak link in the processing line is sufficient to nullify the whole hygienic operation.

4. Cleaning of Process Equipment

The principles of cleaning, rinsing, and sanitizing of food processing equipment are discussed by Loncin and Merson (1979). Cleaning and sanitation should be considered an integral part of food process design and food processing operations (Plett, 1992). The food processing equipment should be designed to facilitate the removal and draining of all the process effluents (steam condensate, waste solids, e.g., peels). All dead ends in tanks, containers, and piping should be eliminated.

Fouling is particularly important in heat exchangers and other installations involving fluid flow (evaporators, filters, cyclones, etc.). Empirical models have been suggested to describe heat-induced fouling and its relationship to the overall heat transfer coefficient (U) and the pressure drop (ΔP), Fryer (1997).

The food processing equipment must be cleaned easily either by quick dismantling and cleaning the parts, or by Cleaning In Place (*CIP*) techniques. The equipment of small food processing plants is usually cleaned by periodic dismantling of the principal units, such as pumps, plate heat exchangers, filters etc. Quick dismantling and re-assembling of process piping is facilitated by various hand-opening clumps.

The design and installation of *CIP* systems in large food processing plants requires specialized experience in pipe flow, sanitation, processing operations, and process control (Jowitt, 1980; Seiberling, 1997; Saravacos and Kostaropoulos, 2002).

The *CIP* system involves the following sequential operations: 1) Pre-rinsing with cold (soft) water; 2) alkali wash (supplemented with sodium hypochlorite); 3) intermediate water rinse; 4) acid rinse; 5) final water rinse; and 6) rinse with sanitizing solution (sodium hypochlorite) or flushing with hot (90 °C) water.

The *CIP* system is actually a chemical cleaning operation, in which the chemical solution is brought into contact with all soiled surfaces. Addition of surface active substances, reducing substantially the surface tension of water, facilitates the penetration of water and aqueous cleaning solutions into crevices of the equipment. The required tanks, pumps, pipes, valves and heaters (heat exchangers

or steam injection devices) are used either as single-use or re-use (re-circulation) systems. Air-operated piston or diaphragm-type pumps are used to feed the chemical solutions. For safety reasons, the pumps and the chemical supply containers are enclosed in a separate compartment of the processing plant.

Ball spray devices are used to clean process and storage tanks. Cylindrical and rectangular tanks are cleaned using liquid feed rates of 8-12 L/min m² internal surface, while vertical silos require liquid rates of 25-35 L/min m tank circumference. Adequate inclination (slope) of piping and process vessels is essential for self-draining of process and cleaning liquids.

Special *CIP* systems are applied to dry-food processing equipment, such as conveyors (belt, screw, pneumatic), dryers (spray, rotary etc), and dry food processing lines, e.g., cereals (Seiberling, 1997).

Usually, food equipment must be cleaned daily, after a processing period. However, when different products are processed in the same equipment, cleaning depends also on the frequency of product changes.

Effective *CIP* requires automation of the whole system. Microprocessor controllers (*PLC*) are used in connection with on-line sensors for temperature, level, flow rate, pressure, and valve position (Grasshoff, 1992). The concentration of cleaning agents and organic effluents can be measured with pH meters, redox potential meters, and optical density meters. The degree of surface contamination can be determined by pressure drop measurements in the pipeline.

IV. FOOD PLANT DESIGN

1. General Aspects

Some aspects of food plant design and food process economics must be considered in order to have a better picture of the whole food processing plant. The food plant consists of the processing equipment, the process utilities, the plant buildings, and the associated storage and office facilities.

There are some important differences in food-related plant design and other industrial applications:

- 1) The raw materials and final products of the food industry are sensitive biological materials, limiting the processing operations. Thus, raw materials in many food industries (e.g., fruits and vegetables), can be stored only for a limited time.
- 2) Large quantities of sensitive raw materials must be processed in the shortest possible period, and the equipment must be able to withstand sudden extreme processing conditions.
- 3) Hygienic factors are important not only in the plant, but also in the process-food product-operating personnel interactions.
- 4) In many cases, such as fruits and vegetables, food processing is seasonal, and a significant part of the employees are unskilled labor.
- 5) Since most seasonal raw materials are perishable, relatively short-term cash flow must be provided for their purchase.

Plant design refers either to the construction of new processing/manufacturing plants or to the improvement or expansion of existing/operating plants. In the latter cases, a detailed evaluation of the existing operation is neces-

sary, before any commitments are made. The adopted solutions should be introduced into the plant as smooth as possible, without disturbing the existing operation.

Most investments (70-75%) for food processing plants in the USA (Young, 2000) and in Germany (BEV, 2000) are related to plant renovation and expansion, and only 25-30% involve construction of new plants.

In all types of food plant design, the main goal is the achievement of the best possible results with respect to quality requirements, high productivity, and low cost. Some typical requirements of food processing operations and equipment are the following:

1) Production Rate. Processing should be as fast as possible, in order to reduce the danger of microbial spoilage and infection, and prevent quality degradation, e.g., vitamin loss due to prolonged heat treatment.

2) Heat Application. In most cases heat must be applied at the lowest possible level to prevent quality losses. However, in cold chains (refrigeration storage and transport systems), temperature should be kept at the highest possible level for saving energy.

3) Sanitation. Hygienic (sanitary) conditions must cover the entire manufacturing spectrum, from raw materials lines to final products, including processes, equipment, buildings, and personnel.

2. New Food Plants

Continuous operation of food processing plants is desirable, since it is more cost-effective, particularly in large plants, and the processes can be controlled more effectively. However, batch processing is still practiced in several food plants, due to the complexity of some processes, and the diversity and low volume of the processed products. The output of a batch food processing plant can be maximized by judicious plant design, which optimizes the use of the available equipment (Cadbury, 1992). Batch processing requires intermediate storage tanks for further processing of the materials (Sinnott, 1996). Optimization of the process cycles should consider the entire plant operation. Time-utilization (Gantt) charts should include both processing and cleaning of the process equipment.

Plant layout is particularly important in food processing because of the uniqueness of processes, and the strict requirements for food hygiene and product quality. One-storied (ground level) buildings are generally preferred, while multi-level installations are advantageous, when gravity flow of large-volume materials is important, as in grain milling plants.

In planning a new food plant, several factors should be considered, including the following basic requirements: location, product/process, food hygiene (sanitation), plant safety, and flexibility.

3. Plant Improvement

Improvement or renovation of existing plants is a continuous task in food manufacturing plants. Even newly designed food plants may need improvement. The difficulties in plant design are manifested especially in seasonal processing, e.g., of fruits and vegetables. In some cases, the same processing facility may be used

for processing different food products, e.g., thermal processing of fruit juices and milk. Plant improvements may include the following:

- 1) Production. Food quality and plant productivity may be improved through new and better processing methods and equipment.
- 2) Conveyance. Increasing the speed of conveyance / transportation of raw material and products will reduce losses through spoilage and mechanical injury.
- 3) Storage. Better storage facilities and conditions will reduce quality losses and minimize logistic costs.
- 4) Energy. Reduction e.g., of heat losses by insulation of equipment.
- 5) Buildings. Better use of building space; insulation of buildings.
- 6) Environment. Utilization of specific environmental conditions, e.g., low air temperature in potato storage. Elimination or reduction of environmental burden of water effluents, air pollutants, solid waste, thermal and noise pollution.
- 7) Management. Greater effectiveness in all fields of the enterprise. Improvement of information flow through the various departments, and feed back adjustments, using computer technology. Labor conditions can be improved, and total number of personnel reduced by judicious automation.

Plant improvement is based on a thorough analysis of the existing plant and in finding alternative solutions of the identified problems. The consequences of proposed changes must be considered carefully, before any plant alterations are made. Most improvements require “individual” or “custom-made” solutions, in which the expertise and experience of the plant designer is very important.

4. Plant Expansion

Expansion of existing plants is necessary to meet increased demand of the company's products or to expand the activities to related new fields. The difference between new plant construction and plant expansion lies in the fact that, in the latter case, the existing substructure of the enterprise is taken into consideration. In planning plant expansion, the following points should be considered:

- 1) Avoid causing problems to existing installations (e.g., energy and effluent networks), transport systems, or creating production “bottlenecks”. Typical examples of the last problem are packaging machines and storage facilities, which are designed to be used also in the plant expansion.
- 2) Use existing installations and equipment for the purpose of reducing investment and operating cost of the expanded facility.
- 3) Purchase, if possible, the same type of equipment, if similar products are to be produced, thus reducing maintenance cost.
- 4) Purchase new equipment, whose combination with existing similar machines will increase plant flexibility. For example, new small compressors, combined with existing larger units, will meet the additional refrigeration load of expanded cold storage facilities.

5. Mobile Food Plants

Mobile food plants may be used in special food processing operations, as in the introduction of a new process (pilot plant), or in seasonal food processing of small food operations (Kostaropoulos, 2001). A mobile plant consists of a number of

small processing units, usually mounted on a trailer-truck, which can be transported to the site of food processing, and connected to process utilities (steam, water, refrigeration).

Mobile factories are particularly attractive for processing relatively small quantities of similar raw materials, which are harvested at different seasons and locations, e.g., apricots (summer) and peaches (fall). A special type of mobile food plants are the fish-processing ship-factories, which move to various fishing areas (Kostaropoulos, 1977).

6. Advanced Food Plants

New food processing and manufacturing concepts, food structure engineering, process design and control, and hygienic aspects are the elements of advanced modern food plants (SIK, 2001). Progress in food processing can be accelerated by adopting new technologies from other manufacturing industries.

Automatic control of food processes can be achieved by interfacing (combining) modern analytical techniques (optical/color, MW, X-rays) with computers. Future plants should be able to adjust their production by taking into consideration the consumers' sensory and texture evaluation of the food products.

Total automation of food processing plants, using robotics and computer technology, can prevent microbial contamination, and improve hygienic operation and product safety.

7. Economic Aspects

The principles of Process Economics, outlined in section V of this chapter, are applicable, in general, to the design of food processes and food processing plants. In addition, some unique economic aspects should be considered, when analyzing food processes.

Capital Cost

Cost estimates in food processing plants is empirical and less accurate than in the chemical process industries. There is a large variety of food plants producing several food products in large or small quantities. General estimation techniques, outlined in section V of this chapter can be used for preliminary calculations. Some useful empirical data from the food industry were presented by Clark (1997b). Basic data, required in a food plant project, include materials, manufacturing operations, packaging operations, and warehousing.

The cost distribution depends on the size and capacity of the food processing plant, e.g., plant site and buildings 25%, process equipment 40%, equipment installation 15%, engineering 10%, and other 10%.

The following procedures can be used in food plant estimations:

- 1) Firm quotes from equipment suppliers and contractors, which can be accurate within 5%.
- 2) Factored estimates, used on processing equipment and correlations. Processing equipment represents about 40-60% of the total food plant cost, which is a very rough approximation.

3) Square meter costs of plant area, based on experience. For example, for large food plants (over 10000m²) the cost may be about 4 k\$/m², out of which 2.5 k\$/m² is for installed equipment and 1.5 k\$/m² for plant buildings (year 2000 prices).

4) Ratio estimates, based on sales or production volumes. The sales to assets (fixed capital) ratio in the food industry may vary from 2-5, depending on the type of industry and the age of the plant. The plant cost per ton capacity may vary from \$ 500 (wet products) to \$ 2500 (dry products).

The plant production capacity can be expressed in kg/h, ton/day, ton/year, cases/day, bottles/min, etc., both as average and range of values. Operating time is specified in hours/day, hours/year, shifts (8h)/day, or days/week. Material and energy balances are in kg/h and kW respectively.

Operating Cost

The major operating cost in food processing is the cost of raw and packaging materials, which can amount to 60-70% of the total cost. Other significant costs are labor, energy, and depreciation of equipment. Other components of the cash flow are sales and distribution, marketing, administration, taxes, and insurance.

Economic analysis (profitability) can be estimated by the return on investment (*ROI*), the net present value (*NPV*) or the Cumulated Cash Flow (*CCF*). The payback period in the food industry is taken usually as 5 years, which is significantly shorter than in the chemical process industries (6-10 years)

Preliminary plant cost estimation is based on the cost of the major processing equipment. In the chemical industry the cost of a plant is estimated as 3.5 to 5 times the cost of the major equipment. However, the cost of the major food processing equipment is relatively higher, because smaller units are used, the materials of construction are expensive (stainless steel, hygienic design), and there is less piping and instrumentation. As a result, the estimated plant cost in food processing is about 1.5 to 2.5 times the cost of the processing equipment (Bartholomai, 1987; Pyle, 1997).

Mathematical modeling and simulation are used to evaluate and optimize the operation of food processing plants. Thus, modeling and linear programming were applied to an apple juice concentrate plant in Argentina (Bandoni et al., 1988). The plant processed 164 ton/day of apples, producing 20.6 tons/day apple juice concentrate 72 °Brix, 1.2 ton/day of aroma essence, and 54.5 ton/day of pomace. Raw material represented 60-65% of the total cost of apple juice concentrate. Availability of raw material and apple variety has an important effect on process economics.

An economic analysis of a citrus processing plant in Italy was presented by Moresi (1984). The plant processed 20 ton/h of oranges and alternatively 10 tons/h lemons, producing frozen citrus juice concentrates, peel oil, and dried pomace. The profitability of the citrus plant depends strongly on the cost of raw material (oranges and lemons), accounting to about 70% of the total product cost.

The high cost of some food raw materials, e.g., seasonal fruits and vegetables, is sometimes subsidized by local governments, in order to provide a reasonable income to the growers, keeping the prices of the processed products at competitive levels in international trade.

V. PROJECT EVALUATION AND COST ESTIMATION

1. Financial Analysis and Process Profitability

The economic analysis and profitability of general processing plants (with emphasis to chemical processing) is discussed by Perry and Green (1984, 1997), Douglas (1988), Peters and Timmerhaus (1991), Pyle (1997), and specialized economics books.

A simplified money flow for project evaluation is presented in Figure 2.6. This diagram is used to describe the economic model for the financial analysis in this book.

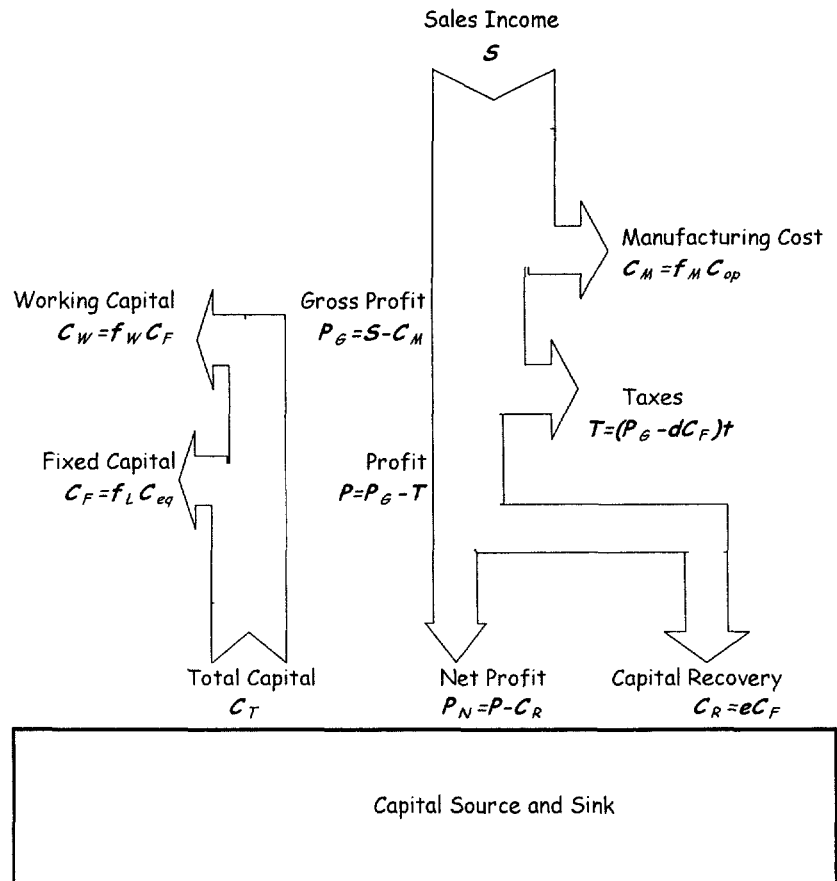


Figure 2.6 A simplified money flow diagram for financial analysis model.

a. Investment Cost

The total capital investment C_T in a processing plant consists of the fixed capital C_F or fixed investment, needed to supply the necessary manufacturing and plant facilities, and the working capital C_W , necessary for the operation of the plant:

$$C_T = C_F + C_W \quad (2-1)$$

The fixed capital C_F consists of the fixed manufacturing investment (equipment, piping, foundations, site preparation) and the fixed non-manufacturing investment which includes the land, processing buildings, utility and waste disposal / treatment, warehouse, shipping and receiving facilities, administrative and other offices, shops, laboratories and other permanent parts of the plant. Construction overhead costs (engineering and supervision expenses, contractor's fees, and contingencies) are also considered non-manufacturing costs.

The working capital C_W in an industrial plant consists of the total amount of money invested in raw materials and supplies carried in stock, finished and semi-finished products, accounts receivable and payable, and cash kept on hand. It is usually estimated as a fraction of the fixed capital:

$$C_W = f_W C_F \quad (2-2)$$

The ratio of working to fixed capital f_W varies from 10-20% in most process industries, but it may be as high as 50% in plants processing products of seasonal demand, or seasonal production of raw materials, e.g., fruits and vegetables. The following value is suggested for food industries:

$$f_W = 0.25 \quad (2-3)$$

The capital investment in an industrial processing plant can be estimated by a number of methods of varying degree of accuracy, depending on the objective of the process design (Peters and Timmerhaus, 1991). The following five estimates are listed in order of increasing accuracy:

- 1) Order-of-magnitude or ratio estimate, based on similar previous data;
- 2) study or fractional estimate, based on knowledge of major items of equipment;
- 3) preliminary estimate, based on sufficient data to authorize the budgeting of the project;
- 4) definitive (project control) estimate, based on nearly complete data before completion of drawings and specifications; and
- 5) detailed (contractor's) estimate, based on complete engineering drawings, specifications, and site surveys.

Estimates (1), (2), and (3) are pre-design cost estimates, which are very important for determining if a proposed project should be considered in more detail. Estimates (4) and (5) are considered firm estimates.

The cost of making capital investment estimates depends on the type of estimate and the size of the processing plant (project), varying from about \$ 3000 (ratio estimate) to nearly \$ 1 000 000 (detailed estimate). The approximate accuracy of the various estimates is given in Table 2.8.

Table 2.8 Accuracy of Capital Cost Estimates

| Estimate of Capital Cost | Approximate Accuracy, % |
|--------------------------|-------------------------|
| Ratio estimate | over 30 |
| Study estimate | less than 30 |
| Preliminary estimate | 20 |
| Definitive estimate | 15 |
| Detailed estimate | 5 |

The cost of the various components in a processing plant varies widely, because of the important differences in the processing of gases, liquids, and solids. Thus, the cost of piping is much higher in gases or liquids processing than in solids processing, where size reduction and mechanical separation equipment is more important.

Table 2.9 presents an approximate cost breakdown for the important components of a typical processing plant (Peters and Timmerhaus, 1991). The installed utilities or auxiliary services include steam, industrial water, compressed air, and waste treatment. Refrigeration is usually considered within the analysis of the specific process, e.g., freezing of foods. The contingency factor refers to unexpected costs of the installation/construction project.

The fixed capital investment in a processing plant can be estimated from empirical rules or approximations, which yield results similar to those of Table 2.9. For example, the Lang factor f_L can be used to estimate the fixed capital C_F from the process equipment cost C_{eq} , using the empirical equation:

$$C_F = f_L C_{eq} \quad (2-4)$$

where

$$f_L = \begin{cases} 4.7 & \text{for fluids processing} \\ 3.6 & \text{for mixed fluids/solids processing} \end{cases} \quad (2-5)$$

The higher f_L factor for gases and liquids processing is due to the higher requirements for piping and valves.

Table 2.9 Approximate Percentage Cost of Process Plant Components

| Component | % of Fixed Capital Cost |
|-----------------------------|-------------------------|
| Purchased equipment | 25 |
| Installation of equipment | 10 |
| Installed piping | 8 |
| Instruments / Control | 8 |
| Electrical installed | 5 |
| Utilities installed | 15 |
| Buildings and Construction | 10 |
| Engineering and supervision | 10 |
| Contractor's fee | 3 |
| Contingency | 6 |
| Total | 100 |

In food processing plants, the f_L factor is much smaller because of the higher cost of equipment (stainless steel) and less piping. Cost data of typical food processing plants are shown in Table 2.10 (Bartholomai, 1987; Clark, 1997b). These data are presented in Figure 2.7, from which the Lang factor f_L is estimated:

$$f_L = 1.6 \text{ for food processing plants} \quad (2-6)$$

In modern food processing plants, using more instrumentation and process control, the Lang factor varies in the range 1.5 to 2.5.

Table 2.10 Cost Data of Typical Food Plants (Year 2000 Prices)

| Food plant | Capacity ton/h | | Equipment Cost M\$ | Total cost M\$ |
|--------------------------|----------------|---------|-----------------------|-------------------|
| | Raw | Product | | |
| Orange juice concentrate | 20 | 1.5 | 1.35 | 3.80 |
| Tomato paste | 15 | 2.5 | 1.30 | 2.20 |
| Apple processing | 5 | 4.0 | 2.40 | 3.50 |
| Milk powder | 18 | 1.7 | 3.40 | 4.80 |
| Ice cream | 2 | 2.0 | 1.80 | 3.00 |
| Milk processing | 2 | 2.0 | 1.00 | 1.40 |
| Vegetable oil | 2 | 1.8 | 1.70 | 2.80 |

1M\$ = 1 000 000 USD

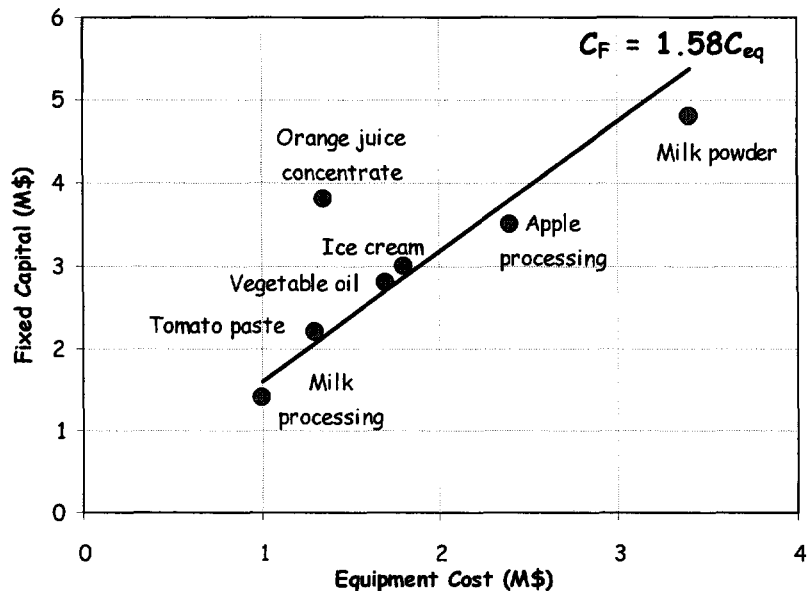


Figure 2.7 Fixed capital versus equipment cost for the typical food industries in Table 2.10.

b. Process Profitability

Process profitability can be estimated by the following simple economic calculations based on the money flow diagram in Figure 2.6:

The gross profit P_G (annual, before taxes) is the sales income S minus the manufacturing cost C_M :

$$P_G = S - C_M \quad (2-7)$$

The manufacturing cost C_M consists of the direct production cost C_{op} and other expenses, which are analyzed in detail in the literature, e.g., Peters and Timmerhaus (1991). For quick estimation the direct production cost C_{op} is multiplied by a factor f_M to obtain the manufacturing cost C_M .

$$C_M = f_M C_{op} \quad (2-8)$$

For food industries the following value is suggested:

$$f_M = 1.6 \quad (2-9)$$

The taxable income R is the annual gross profit P_G minus the depreciation allowed by the tax authorities:

$$R = P_G - d C_F \quad (2-10)$$

where d is annual depreciation allowed by the tax authorities for computing taxable income. Thus the taxes T are calculated by the following equation, considering an income tax rate t :

$$T = R t \quad (2-11)$$

Consequently the annual profit after taxes P , alternatively called Annual Cash Flow ACF , is calculated by the equation:

$$P = ACF = P_G - T \quad (2-12)$$

The net profit is obtained after subtraction of the capital recovery payment:

$$P_N = P - e C_F \quad (2-13)$$

where e is the capital recovery factor.

Generally, the two depreciation rates d and e are different. d is decided by the tax authorities while e is negotiated by the loaner bank. Usually the following equations are used:

$$d = 1 / N_T \quad (2-14)$$

That is, a straight line depreciation is accepted by the tax authorities for the expected project life N_T .

$$e = \frac{i_L(1+i_L)^{N_L}}{(1+i_L)^{N_L} - 1} \quad (2-15)$$

Equation (2-15) calculates the annual payment for a loan with i_L interest rate and N_L loaning period.

It must be noted that both N_T and N_L are generally different from the real life of the project N . However, the following approximation is usually made:

$$N = N_T = N_L = 10 \text{ years} \quad (2-16)$$

The most important magnitudes in financial analysis are:

- the Cumulated Cash Flow CCF , and
- the Net Present Value NPV

which are defined by the following equations:

$$CCF = -C_T + \sum_{y=1}^N P \quad (2-17)$$

$$NPV = -C_T + \sum_{y=1}^N \frac{P}{(1+i)^y} \quad (2-18)$$

where i is the interest rate, expressing the time value of money. It must be noted that NPV (instead of CCF) takes into account the time value of money:

$$NPV = CCF, \text{ when } i=0 \quad (2-19)$$

If P is constant over the years then it can be proved that:

$$CCF = -C_T + N P \quad (2-20)$$

$$NPV = -C_T + \frac{P}{e}, \text{ where } e = \frac{i(1+i)^N}{(1+i)^N - 1} \quad (2-21)$$

Figure 2.8 is a graphical representation of Equation (2-21). It represents net present value NPV versus time for the following values of interest rate i :

- $i = 0: NPV = CCF$
- $i = 0.5$
- $i = IRR$

where the internal rate of return IRR is defined by the following relation:

$$i = IRR \Rightarrow NPV = 0 \quad (2-22)$$

CCF and NPV curves intersect the x-axis at the points SPB and DPB , respectively, which are defined by the following equations:

Simple Payback Period SPB :

$$N = SPB \Rightarrow CCF = 0 \quad (2-23)$$

Discounted Payback Period DPB :

$$N = DPB \Rightarrow NPV = 0 \quad (2-24)$$

It can be proved that:

$$SPB = \frac{C_T}{P} \quad (2-25)$$

$$DPB = \frac{\ln\left(\frac{1}{(1-i)^{SPB}}\right)}{\ln(1+i)} \quad (2-26)$$

The inverse of simple payback period SPB is often called the Return on Investment ROI :

$$ROI = \frac{P}{C_T} \quad (2-27)$$

Figure 2.9 is a modification of Figure 2.8 taking into account the construction period and the salvage value of the equipment at the end of the project life.

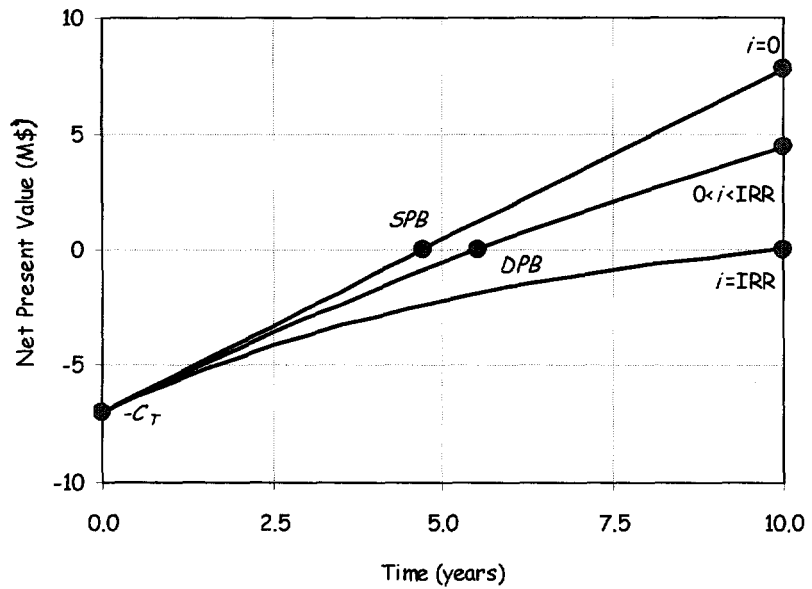


Figure 2.8 Net present value NPV and cumulated cash flow CCF versus time.

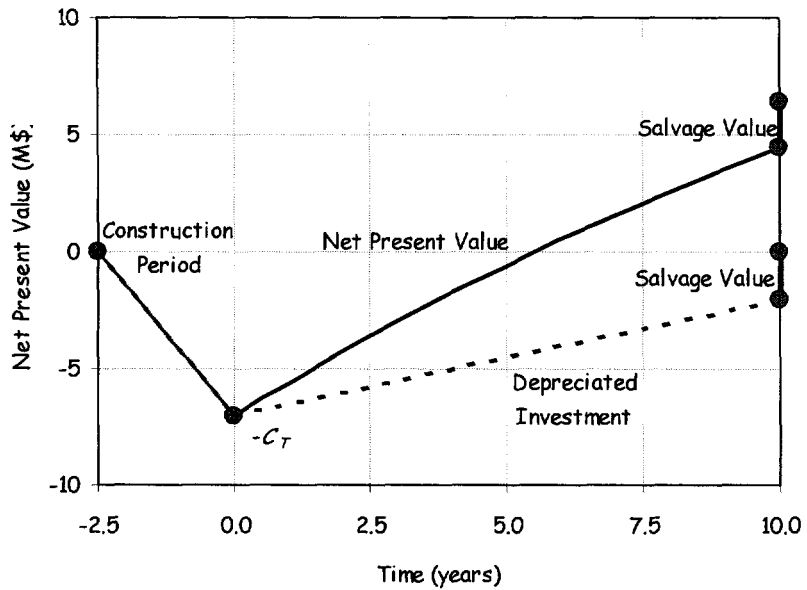


Figure 2.9 Net present value NPV versus time. Construction period and equipment salvage value are taken into account.

c. Individual Processes

The previous financial analysis refers to complete industries or to processes with income. But, usually, in process design the processes analyzed have no income, and they just undertake a subtask of a major system. In these cases, process design is based on the total annualized cost TAC , defined by the Equation:

$$TAC = eC_T + C_M \quad (2-28)$$

where C_T is the total investment cost and C_M the annual manufacturing cost. Thus, the total annualized cost TAC is a weighted sum between the investment cost C_T and the manufacturing cost C_M . The weighting factor is the capital recovery factor e , which is calculated from Equation (2-29).

$$e = CRF(i, N) = \frac{i(1+i)^N}{(1+i)^N - 1} \quad (2-29)$$

Taking into account Equations (2-4) and (2-9), Equation (2-28) is written:

$$TAC = ef_L C_{eq} + f_M C_{op} \quad (2-30)$$

The reader is reminded that for food industries (Equations 2-6 and 2-9):

$$f_L \approx f_M \approx 1.6 \quad (2-31)$$

In individual processes, incorporating one or two operations in an existing or designed processing plant, only the cost of the major process equipment and its operating cost (utilities) are considered. Thus, the following assumptions can be made:

$$f_L \approx f_M \approx 1 \quad (2-32)$$

Thus, Equation (2-30) is written:

$$TAC = eC_{eq} + C_{op} \quad (2-33)$$

The total annualized cost of the process TAC in the form of Equation (2-33) is usually selected as the objective function to be optimized in this book.

The following equations calculate the equipment cost C_{eq} , and the annual operating cost C_{op} , respectively.

$$C_{eq} = \sum_i C_{oi} \left(\frac{M_i}{M_{oi}} \right)^{n_i} \quad (2-34)$$

$$C_{op} = \sum_j c_j F_j t \quad (2-35)$$

which are analyzed in the following sections.

The process cost equation for TAC (2-33) is used throughout this book for the economic evaluation and optimization of the various food processes (Chapters 3-10).

The complete economic analysis and profitability of food processing plants requires additional cost data on raw materials (foods and packages), labor, and overhead, as shown in Example VI of this chapter. It should be noted that the cost of raw materials in food processing is about 60-70% of the manufacturing cost, which is much higher than in usual chemical processing.

2. Cost of Equipment and Utilities

The most accurate cost estimation for process equipment is to obtain a price quotation from a reliable manufacturer or supplier (vendor) of equipment. Specification sheets for each process unit should be prepared for the equipment, which should contain basic design data, materials of construction, and special information that will help the supplier to provide the appropriate equipment. Standardized equipment is preferred because of lower cost and faster delivery.

It should be noted that very strict and detailed specifications could increase substantially the price of the equipment, while an available “off-the-shelf” unit at a lower cost might be satisfactory. Second-hand equipment at a lower cost may, in some cases, be satisfactory for the intended application.

Suppliers of processing equipment are found in various directories and in international equipment fairs.

When approximate cost data are required for preliminary design, empirical methods and rules are used, which will yield fast results within the accepted accuracy (Chilton, 1970). A popular method is to use the Guthrie charts of equipment cost versus capacity (Guthrie, 1969; Peters and Timmerhaus, 1991; Perry and Green, 1984; Douglas, 1988). Plotted on log-log scales, the Guthrie charts show straight lines. These charts are represented by the generalized cost-capacity Guthrie equation:

$$C = C_0 \left(\frac{M}{M_0} \right)^n \quad (2-36)$$

where, C and C_0 are the equipment costs (e.g., USD=\$) at plant capacities M and M_0 (e.g., kg/h), respectively.

The scale index (exponent, n) varies with the type of equipment over the range 0.5 to 1.0, and it is often taken approximately as $n=2/3$. The “2/3” factor is related to the cost of spherical vessels: $C=kV^{2/3}$, where V is the vessel volume and k is a constant (Biegler et al., 1997).

Exponent values near $n=1$ are characteristic of complex mechanical or electrical units, such as motors, compressors, homogenizers, and distillation columns. Low values near $n=0.5$ characterize large processing units, such as evaporators, heat exchangers, and tanks.

Table 2.11 and Figure 2.10 show typical equipment costs for general process equipment, which is also used in food processing.

The plant capacity/cost relationship (Equation 2-36) is normally applied to equipment and utilities of the main industrial processes. Better cost estimates can be obtained by modifying Equation (2-36), considering the cost of all auxiliaries outside the main process, such as environmental installations, and materials handling and storage (Haseltine, 1986).

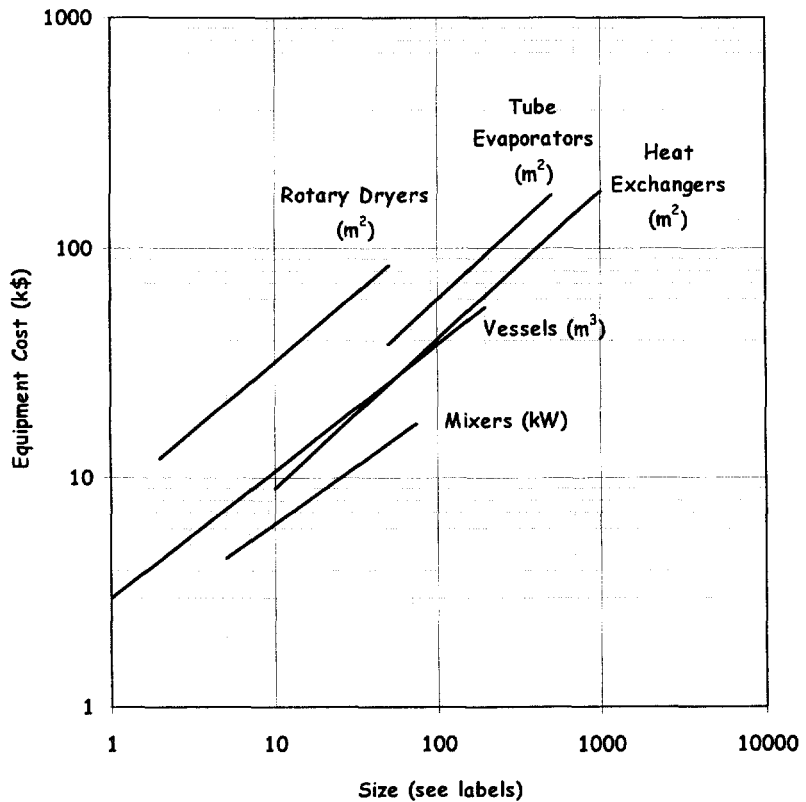
To take into account the effect of inflation, operating pressure and temperature, Equation (2-36) is modified to Equation (2-37)

$$C = f_I f_T f_p C_0 \left(\frac{M}{M_0} \right)^n \quad (2-37)$$

Table 2. 11 Estimated Costs of Equipment and Scale-up Factors (year 2000)

| Equipment | Size (Unit) | Size Range | | Unit Cost k\$ | Scale Index |
|------------------|----------------------------|------------|------|------------------|-------------|
| | | min | max | | |
| Heat Exchangers | Area (m ²) | 10 | 1000 | 2.00 | 0.65 |
| Mixers | Power (kW) | 5 | 75 | 3.00 | 0.50 |
| Rotary Dryers | Area (m ²) | 2 | 50 | 8.00 | 0.60 |
| Tube Evaporators | Area (m ²) | 50 | 500 | 3.00 | 0.65 |
| Vessels | Capacity (m ³) | 1 | 200 | 3.00 | 0.55 |

Note: 1 k \$ = 1000 USD

**Figure 2.10** Equipment cost for various unit operations (year 2000).

where the correction factors for inflation f_I , temperature f_T , and pressure f_P are estimated from the following equations:

$$f_I = \left(\frac{M \& S}{1100} \right) \quad \text{or} \quad \left(\frac{CEP}{400} \right) \quad (2-38)$$

and the cost indices $M\&S$ and CEP are discussed in the next paragraph.

$$f_T = \exp\left(\frac{T - 150}{750}\right) \quad (2-39)$$

$$f_P = \exp\left(\frac{P - 1}{75}\right) \quad (2-40)$$

The correction factors of empirical equation (2-39) for temperatures 100-1800 °C and (2-40) for pressures 1-70 bar come from fitting exponential equations to data presented by Biegler et al. (1997). Figure 2.11 shows the effect of pressure on the cost factor f_P .

Most food processes operate at temperatures lower than 150°C and temperature correction of the equipment cost is not necessary. The pressure correction factor may be used at pressures up to 70 bar, but not at the very high pressures of supercritical fluid extraction (about 300 bar) or the ultrahigh pressures of high-pressure food processing (1-8 kbar).

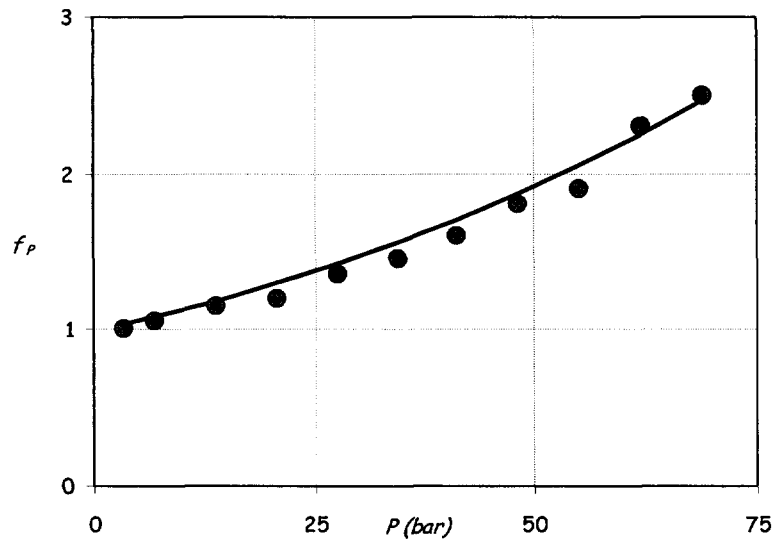


Figure 2.11 Equipment cost correction factors for pressure.

The cost of processing equipment or processing plants at a given time can be estimated by multiplying the known cost of the equipment/plant at a past time by the ratio of an appropriate index, corresponding to the same time points. Two common engineering indices are used in the process industries:

a) The Marshall and Swift Equipment Index (*M&S*) (formerly Marshall and Stevens Index), published periodically in the journal "Chemical Engineering", is the weighted average of the cost of equipment of 8 chemical process industries (chemicals, petroleum, paper, rubber, paint, glass, cement, and clay products). The *M&S* all-industry index is the arithmetic average of indices for 47 different industrial, commercial and housing equipment. The *M&S* all-industry index is about 2% lower than the *M&S* equipment, and it is used in this book. The base year of both *M&S* indices is 1926 (= 100).

b) The Chemical Engineering Plant Cost Index (*CEP*) shows construction costs for processing plants. It consists of the weighted average of four major components: equipment, machinery and supports; erection and installation labor; buildings, materials and labor, and engineering and supervision. The major component, equipment, consists of fabricated equipment; process machinery; pipe, valves, and fittings; process instruments and controls; pumps and compressors; electrical equipment and materials, and structural supports, insulation, and paint. The base year of the *CEP* cost index is 1957-1959 (=100).

Both *M&S* and *CEP* cost indices are used in process design, giving similar results (CE, 1997). Table 2.12 and Figure 2.12 show the cost indices *M&S* (all-industry) and *CEP* during the period 1960-2000. These data can be extrapolated to the near future using linear regression equations fitted to recent years. The following simplified equations are also presented in Figure 2.12.

$$CEP = 400 + 5 (Year-2000) \quad (2-41)$$

$$M\&S = 1100 + 20 (Year-2000) \quad (2-42)$$

The increases of the cost indices are caused by inflation and rises in the prices of certain expensive materials, e.g., stainless steel. The sharp increase of the cost indices during the decade 1970-1980 is mainly due to the rising costs of energy (world-wide petroleum crisis).

Table 2.12 *M&S* (all-industry) and *CEP* Cost Indices

| Year | M&S | CEP |
|------|------|-----|
| 1960 | 230 | 100 |
| 1965 | 240 | 105 |
| 1970 | 300 | 120 |
| 1975 | 444 | 182 |
| 1980 | 560 | 261 |
| 1985 | 790 | 325 |
| 1990 | 904 | 356 |
| 1995 | 1030 | 380 |
| 2000 | 1100 | 385 |

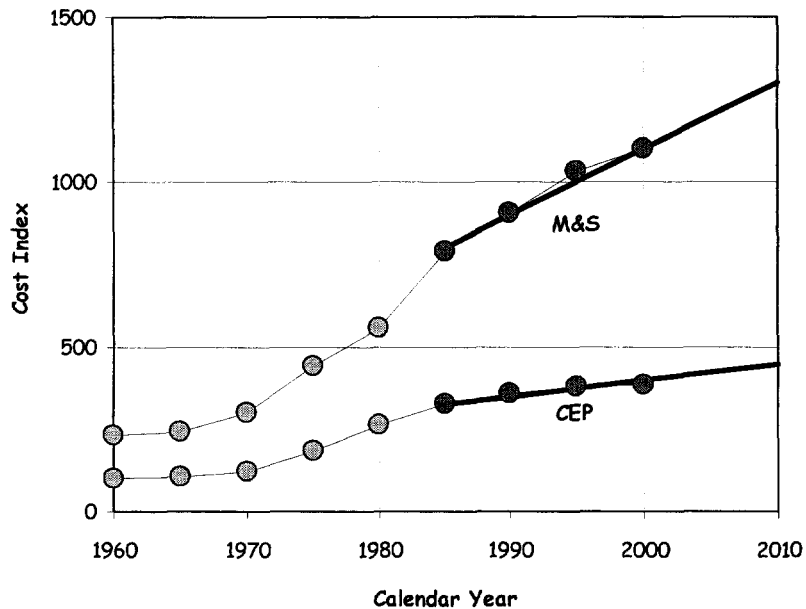


Figure 2.12 Marshal and Swift all-industry equipment (*M&S*) and chemical engineering plant (*CEP*) cost indices in the period 1960-2000.

Concerning the cost of the utilities, Table 2.13 can be used to make preliminary estimates. Generally, the term utilities includes power, steam, compressed air, cooling and process water. The prices differ from company to company. They will depend on the primary energy sources, the technologies applied, and the plant location. The current cost of utilities supplied by the utility companies electricity, gas and water can be obtained from their local area offices.

Table 2.13 Typical Costs of Utilities for Process Plants (year 2000)

| Utility | \$/MWh | Cost \$/ton |
|---------------|--------|-------------|
| Fuel | 40 | 450 |
| Steam | 50 | 30 |
| Power | 100 | |
| Cooling Water | 5 | 0.15 |
| Refrigeration | 20 | |

VI. APPLICATION TO TOMATO-PASTE-PROCESSING PLANT

Prepare a preliminary design for a tomato processing plant, processing 20 ton/day of tomatoes of 7 % TS, and producing tomato paste of 32 % TS. The block diagram of the plant will be based on the block diagram of Figure 2.2 and the diagrams of Figures 2.3 to 2.5. Tomato processing is discussed by Luh and Woodroof (1988), Gould (1992), and *NFPA* (1992).

The plant will use the “hot break” method of producing tomato juice, which is concentrated in a triple- effect forced-circulation evaporator. The product will be heat-sterilized and packed aseptically in 150 L drums. The solid tomato wastes will be de-watered mechanically in a screw press and dehydrated in an air dryer, to a dried pomace of 8 % moisture content.

Estimate the approximate cost of the processing equipment of the food plant, using cost data from the literature, and the suppliers of food processing equipment.

1. Material Balances

The material balances, based on a feed of 100 kg/h of tomatoes are shown on the block diagram of Figure 2.13. The raw tomatoes are assumed to contain 7 % TS (total solids) and 93 % water.

The tomato solids consist mainly of soluble carbohydrates, which are usually measured with a refractometer as °Brix (% sucrose), and converted to % TS with special tables (*NFPA*, 1992). The TS include small amounts of skins and seeds, which are separated mechanically from the tomato juice.

Consecutive material balances (overall and % TS) around each processing unit show that out of a feed of 100 kg tomatoes (7 % TS), the following products are obtained: Tomato juice (6 % TS) 95 kg, which is concentrated in a 3-effect evaporator to 17.8 kg of tomato paste (32 % TS). Culls (tomatoes unfit for processing) 2 kg, mixed with tomato waste from juice finishing 2 kg to tomato waste (23.5 % TS) 4 kg. The mixed waste is dewatered partially in a screw press to 2 kg of concentrated waste (40 % TS), which is dehydrated in a rotary air dryer to 0.87 kg of dried pomace (92 % TS). The evaporated water in the juice concentration unit is 77.2 kg, and the remaining small amount of tomato solids is lost in the liquid wastes of washing and dewatering steps.

The amount of chlorinated water required for tomato receiving and washing depends on the design of the system, and for 20 ton/h of tomatoes, approximately 2 m³/h of water may be needed.

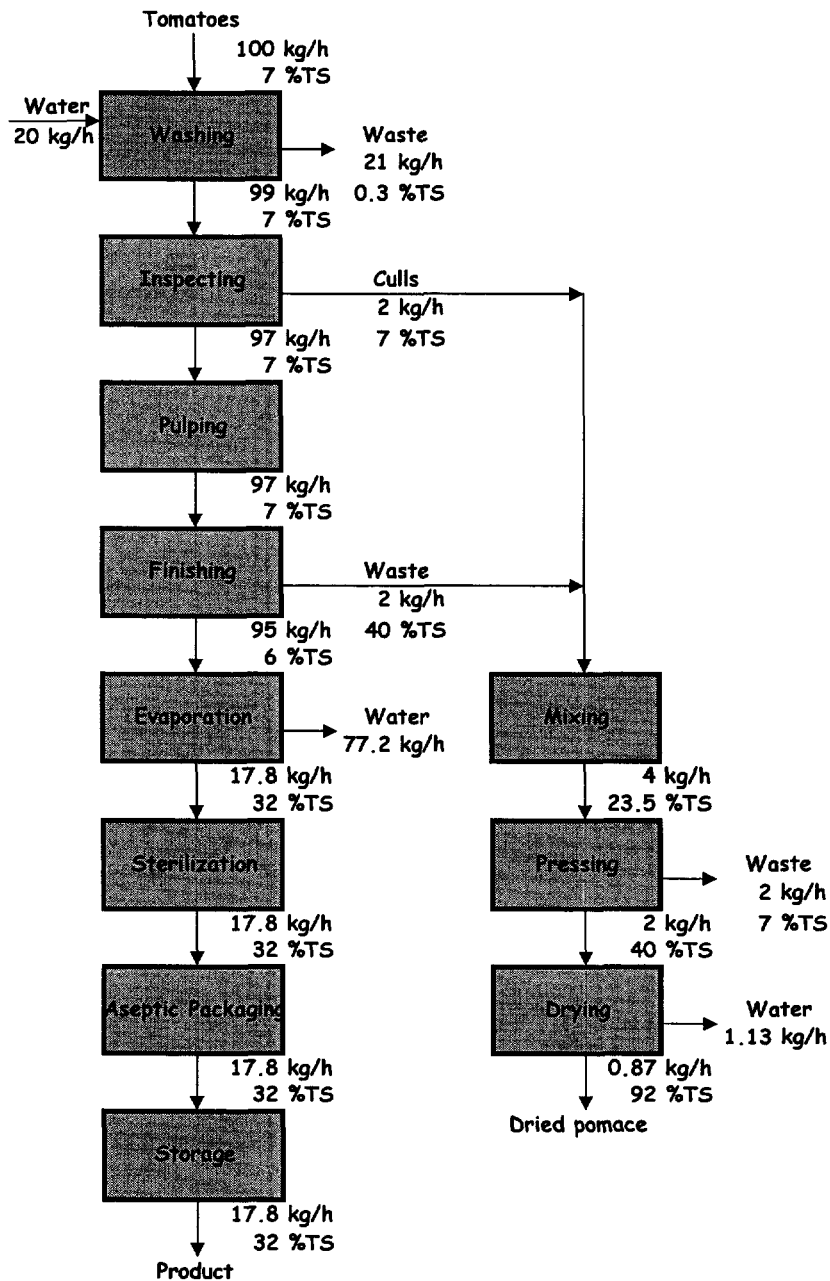


Figure 2.13 Material balances in tomato-paste-processing plant (see Figure 2.2). Basis 100 parts (weight) of raw tomatoes 7%TS (Total solid content).

2. Energy Balances

The energy requirements in the tomato-paste plant are mainly for heating the evaporator, heating tomatoes during pulping, and heat-sterilization of tomato paste. Cooling water is needed mainly for condensing the water vapors from the evaporation system, and secondarily for cooling the sterilized tomato paste. A relatively small amount of electrical power will be needed for the operation of the electric motors, moving the pumps, conveyor belts, tomato pulpers/finishers, and the dryer.

For preliminary calculations the specific heat (C_p) of tomatoes (7 % TS) is taken as 4.0 kJ/kg K and that of tomato paste (32 % TS) 3.40 kJ/kg K. The (C_p) of water is 4.18 kJ/kg K, and the thermodynamic properties (enthalpies) of steam are taken from steam tables (Haar et al., 1984).

A triple-effect evaporator will be used to concentrate 19 ton/h of tomato juice from 7 % TS to 3.56 ton/h of tomato paste of 32 % TS , removing 15.44 ton/h of water. Assuming a heat economy of 2.5 (kg of water evaporated per kg of heating steam) the steam requirement will be $15.44/2.5 = 6.2$ ton/h.

Heating of washed and sorted tomatoes from 25–85 °C for pulping (“hot break”) will require a heating energy rate (power) of $20000 \times 0.97 \times 4.0 \times (85-25) = 4656000$ kJ/h = $4656/3600 = 1297$ kW. Using direct saturated steam of 120 °C (2 bar) of enthalpy (120 °C - 85 °C) 2350 kJ/kg, the amount of required steam will be $4656000/2350 \times 1000 = 1.98$ ton/h. The steam required for sterilization of 3.56 ton/h of tomato at 95 °C from an initial temperature of 45 °C will be $3560 \times 3.40 \times (95-45) = 605200$ kJ/h = $605200 / 2202 = 275$ kg/h = 0.275 ton/h. Total steam needed, $6.2 + 1.98 + 0.275 = 8.45$ ton/h.

The rotary dryer will dehydrate 400 kg/h of pressed tomato waste (40 % TS) to 174 kg/h of dried pomace (92 % TS), removing $400-174 = 226$ kg/h water. Assuming that the heat capacity of the rotary dryer is 4 MJ/kg evaporated water, the required energy for drying will be $226 \times 4 = 904$ MJ/h. If natural gas of 40 MJ/kg is used as a fuel, $904/40 = 22.6$ kg/h will be needed.

Cooling water (CW) will be needed mainly for the condensation of the water vapors from the 3rd effect of the evaporator. Total water evaporated $20000 \times 77.2/100 = 15440$ kg/h, and water vapors to be condensed in the 3rd effect $15440/3 = 5146$ kg/h. A mixing condenser can be used, in which the vapors are condensed by direct contact with the cooling water. Assuming a CW temperature of 20 °C and condensation temperature of 60 °C, the CW required will be $5146 \times 2357/40 \times 4.18 = 72542$ kg/h, or 72.5 m³/h. The CW required for cooling $20000 \times 0.178 = 3560$ kg/h of sterilized tomato paste from 95 to 45 °C, assuming final water temperature of 35 °C, will be $3560 \times 3.40 \times (95-45)/4.18 \times (35-20) = 9652$ kg/h or 9.6 m³/h. Total cooling water required $72.5 + 9.6 = 82.1$ m³/h.

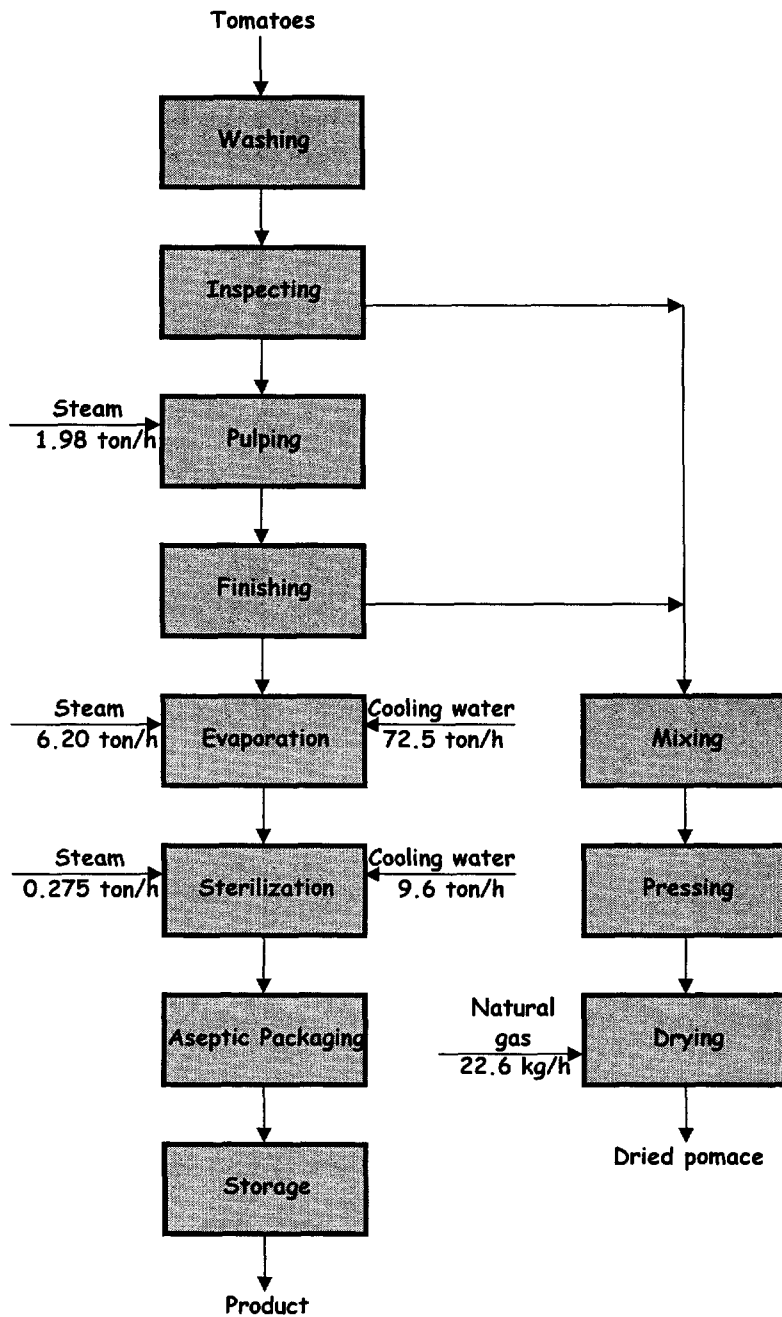


Figure 2.14 Energy balances in tomato-paste-processing plant (see Figure 2.2).

3. Sizing and Cost of Process Equipment

The sizing and cost of the major processing equipment, i.e., evaporators, sterilizers, and dryers, is discussed in the corresponding chapters of this book, and by Saravacos and Kostaropoulos (2002). Empirical data on the mechanical processing equipment are provided by manufacturers and suppliers. Cost data are also provided by Bartholomai (1987), and Clark (1997b).

Two parallel mechanical preparation lines for tomatoes are used with identical equipment, since the feed rate is relatively large. A large evaporation system is used, since it is more economical, while the size of the aseptic packing system is normal. A rather small dryer will be used, because the amount of waste dehydrated and the water evaporated are relatively small. A triple-effect forced-circulation evaporator of 15440 kg/h evaporative capacity will be used. The first effect of the evaporator will be heated with saturated steam at 120°C (2 bar), and the last (3rd) effect will operate at 60°C. The overall heat transfer coefficients are assumed to be $U_1=2.0$, $U_2=1.5$ and $U_3=1.0$ kW/m²K. The estimated total heat transfer area of the evaporator will be 330 m². The detailed design of evaporators is discussed in Chapter 6.

The aseptic processing system will sterilize and pack 3560 kg/h tomato paste. Assuming an evaporative capacity of the dryer 30 kg/m²h, a drying surface of $226/30 = 7.5$ m² will be needed. A cylindrical rotary dryer of 1.5 m diameter and 2.0 m length with a drying surface of $3.14 \times 1.5 \times 2.0 = 9.7$ m² can be used.

Table 2.14 shows preliminary capacity and cost data of processing equipment for the tomato-paste plant. Assuming a Lang factor $f_L = 2$ (Equation 2-4), the cost of the fixed capital investment for the tomato-paste plant will be about:

$$C_F = 2 \times 1.5 = 3 \text{ M\$}$$

Table 2.14 Preliminary Data of Processing Equipment for a Tomato-Paste Plant

| Equipment | Capacity | Approximate Cost k\$ |
|---|--------------------------------|-------------------------|
| Washing | 2x(10 ton/h) | 150 |
| Sorting | 2x(10 ton/h) | 100 |
| Heater | 2x(10 ton/h) | 100 |
| Pulping | 2x(10 ton.h) | 100 |
| Finishing | 2x(10 ton/h) | 120 |
| Juice tanks | 2x(25 m ³) | 20 |
| Evaporator | 1x(330 m ² surface) | 500 |
| Aseptic processing | 1x(4 ton/h) | 250 |
| Screw press | 1x(1 ton/h) | 50 |
| Dryer | 1x(250 kg/h evaporation) | 80 |
| Pumps | 10 (various capacities) | 60 |
| Total cost of main processing equipment | | 1.5 M\$ |
| Fixed capital investment | | 3 M\$ |

4. Plant Profitability

The plant under consideration is assumed to process 28 000 ton/yr in order to produce $(20\,000 \times 0.178 =)$ 5000 ton/yr of tomato paste. This corresponds to an annual operating time of about 1400 h (about 3 months for 16 h daily operation).

The cost of raw materials (tomatoes) is about 53 \$/ton and that of tomato paste 800 \$/ton (year 2000 prices). Thus, the sales income will be 5000 ton/yr \times 800 \$/ton = 4 M\$/yr and the annual cost of the raw materials $28\,000 \times 53 = 1.50$ M\$/yr. The annual cost of utilities is: $(82 \text{ ton/h C/W} \times 0.15 \text{ $/ton} + 8.5 \text{ ton/h steam} \times 30 \text{ $/ton} + 23 \text{ m}^3/\text{h natural gas} \times 0.40 \text{ $/m}^3) \times 1400 \text{ h/yr} = 0.40$ M\$/yr. Thus, the operating cost is estimated as $1.50 + 0.40 = 1.90$ M\$/yr and the total (manufacturing) cost $1.9 \times 1.20 = 2.3$ M\$/yr. The raw materials account for $1.5 / 2.3 = 65\%$ of the total product cost.

The financial model presented in Section V is applied to the present tomato processing plant and the results are presented in Table 2.15 and in Figure 2.15.

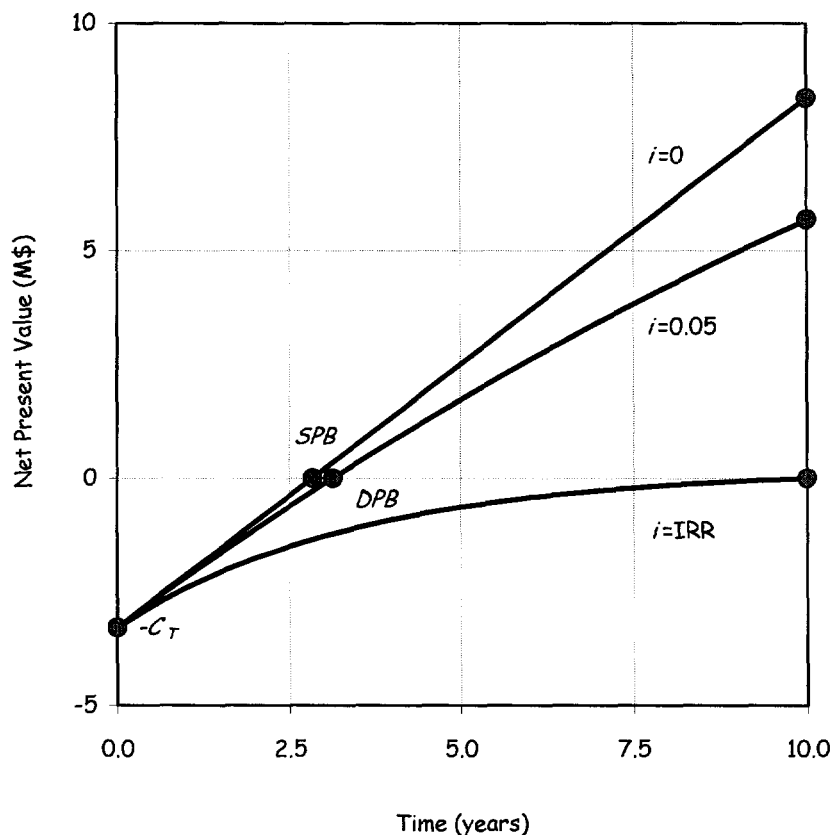


Figure 2.15 Net present value of a tomato-paste-processing plant.

Table 2.15 Tomato Plant Profitability

Basic data

| | | |
|----------------|------------|-------------|
| Equipment cost | $C_{eq} =$ | 1.50 M\$ |
| Operating cost | $C_{op} =$ | 1.90 M\$/yr |
| Sales income | $S =$ | 4.00 M\$/yr |

Factors estimates

| | | |
|---------------------------|---------|--------|
| Manufacturing cost factor | $f_M =$ | 1.20 - |
| Working capital factor | $f_W =$ | 0.10 - |
| Lang factor | $f_L =$ | 2.00 - |

Capital and manufacturing costs

| | | |
|--------------------|---------|-------------|
| Total capital | $C_T =$ | 3.30 M\$ |
| Fixed capital | $C_F =$ | 3.00 M\$ |
| Working capital | $C_W =$ | 0.30 M\$ |
| Manufacturing cost | $C_M =$ | 2.28 M\$/yr |

Economic environment

| | | |
|--------------------------|-------|--------|
| Depreciation coefficient | $d =$ | 0.10 - |
| Tax rate | $t =$ | 0.40 - |
| Interest rate | $i =$ | 0.05 - |
| Capital recovery factor | $e =$ | 0.13 - |
| Project life | $N =$ | 10 yr |

Profit analysis

| | | |
|--------------|---------|-------------|
| Gross Profit | $P_G =$ | 1.72 M\$/yr |
| Taxes | $T =$ | 0.56 M\$/yr |
| Profit | $P =$ | 1.16 M\$/yr |
| Net profit | $P_N =$ | 0.74 M\$/yr |

Financial appraisal

| | | |
|---------------------------|---------|----------|
| Net present value | $NPV =$ | 5.69 M\$ |
| Simple payback period | $SPB =$ | 2.84 yr |
| Discounted payback period | $DPB =$ | 3.13 yr |
| Return on investment | $ROI =$ | 0.35 - |
| Internal rate of return | $IRR =$ | 0.33 - |

NOMENCLATURE

| | |
|----------|-------------------------------------|
| C_{eq} | Equipment cost |
| C_F | Fixed capital |
| C_i | Cost of the i equipment |
| c_j | Unit cost of the j utility |
| C_M | Manufacturing cost |
| C_{oi} | Unit cost of the i equipment |
| C_{op} | Operating cost |
| C_T | Total capital |
| C_W | Working capital |
| d | Depreciation for tax reduction |
| e | Capital recovery factor |
| F_j | Flow rate of the j utility |
| f_L | Lang factor |
| f_M | Manufacturing cost factor |
| f_W | Working capital factor |
| i | Interest rate |
| M_i | Size of the i equipment |
| M_o | Reference size of the i equipment |
| N | Real life of the project |
| n_i | Scaling index of the i equipment |
| N_L | Loan period |
| N_T | Project life for tax depreciation |
| P | Pressure |
| P | Profit |
| P_G | Gross profit |
| P_N | Net profit |
| R | Taxable income |
| S | Sales income |
| t | Tax rate |
| T | Taxes |
| T | Temperature |
| TAC | Total annualized cost |
| t | Time |

Abbreviations

| | |
|----------|---|
| $2D$ | Two-dimensional |
| $3D$ | Three-dimensional |
| ACF | Annual Cash Flow |
| $AFDOUS$ | Association of Food and Drug Officials of the United States |
| $AIChE$ | American Institute of Chemical Engineers |
| $ASME$ | American Society of Mechanical Engineers |
| BE | Break Even Point |
| $BISSC$ | Baking Industry Sanitation Standards Committee |
| BS | British Standards |
| CCF | Cumulated Cash Flow |

| | |
|----------------|--|
| <i>CCP</i> | Critical Control Point |
| <i>CCPS</i> | Center of Chemical Process Safety |
| <i>CE</i> | Cofornite Europeenne |
| <i>CEP</i> | Chemical Engineering Plant Cost Index |
| <i>CIP</i> | Cleaning In Place |
| <i>CRF</i> | Capital Recovery Factor |
| <i>CS</i> | Customs Service |
| <i>DFISA</i> | Dairy and Food Industries Suppliers Association |
| <i>DIN</i> | German Standardization Authority |
| <i>DPB</i> | Discounted Payback Period |
| <i>EHEDG</i> | European Hygienic Equipment Design Group |
| <i>EPA</i> | Environmental Protection Agency |
| <i>EU</i> | European Union |
| <i>FDA</i> | Food and Drug Administration |
| <i>FSO</i> | Food Safety Objectives |
| <i>FTC</i> | Federal Trade Commission |
| <i>GMP</i> | Good Manufacturing Practice |
| <i>HACCP</i> | Hazard Analysis Critical Control Point |
| <i>HAZOP</i> | Hazardous Operations |
| <i>HPP</i> | High Pressure Processing |
| <i>HTST</i> | High Temperature Short Time |
| <i>IAMFES</i> | International Association of Milk, Food, and Environmental Sanitarians |
| <i>IDF</i> | International Dairy Foundation |
| <i>IFT</i> | Institute of Food Technologists |
| <i>IRR</i> | Internal Rate of Return |
| <i>ISO</i> | International Standardization Organization |
| <i>M&S</i> | Marshal and Swift Equipment Cost Index |
| <i>MAP</i> | Modified Atmospheric Packaging |
| <i>MP</i> | Maximum Profit |
| <i>NFPA</i> | National Fire Protection Agency |
| <i>NPV</i> | Net Present Value |
| <i>NTP</i> | Non-Thermal Processes |
| <i>OSHA</i> | Occupational Safety and Health Administration |
| <i>P&I</i> | Pipes and Instruments |
| <i>PBD</i> | Process Block Diagram |
| <i>PCD</i> | Process Control Diagram |
| <i>PFD</i> | Process Flow Diagram |
| <i>R&D</i> | Research and Development |
| <i>ROI</i> | Return on Investment |
| <i>SPB</i> | Simple Payback Period |
| <i>SS</i> | Soluble Solids |
| <i>TAC</i> | Total Annualized Cost |
| <i>TI</i> | Total Income |
| <i>TS</i> | Total Solids |
| <i>USA</i> | United States of America |
| <i>USD</i> | United States Dollars |
| <i>USDA</i> | United States Department of Agriculture |

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3

Computer-Aided Process Design

I. INTRODUCTION

Computer software is a tool-of-the-trade for engineers. The capabilities provided by computers for fast calculation, large storage, and logical decisions, plus the available technical and mathematical software, permit engineers to solve large problems and to do it much more rapidly than ever before. The engineers' emphasis can therefore shift from problem solving to planning, conceiving, and interpreting with the information made available. Process design is one of the engineering functions that has been impacted by computers (Peters and Timmerhaus, 1991).

Complex programs that simulate the operation and a complete process or individual units, have been developed by central design departments of the major manufacturing companies, by university departments, and by commercial software organizations. Hundreds, even thousands, of programs have been written to solve problems commonly encountered by chemical or food engineers. Many of these have become available for purchase.

An engineer may write a programming language code to solve a particular design problem. Before the explosive growth of software, this was done commonly. There is still a place for it, when appropriate software is not available or as a learning technique.

Because program writing and debugging is usually quite time-consuming, it is recommended that available software be used and that a specific design program be written only after determining that appropriate software is not available.

Alternatively, spreadsheet software has become an indispensable tool for engineers, because of the availability of personal computers, ease of use, and adaptability to many types of problems. Spreadsheet software has achieved great popularity because of its availability for microcomputers at reasonable cost, the ease of learning and using the software, and its flexible applications to many problems. Solutions for many engineering problems can be obtained more rapidly by spreadsheet than by writing a language code program, such as FORTRAN.

Furthermore, general purpose spreadsheet software can be used effectively in process design. For example, Microsoft Excel with Visual Basic for Applications is an effective tool for process design. Spreadsheets offer sufficient process model “hospitality”. They are connected easily and on-line with charts and graphic objects, resulting in powerful and easy-to-use graphical interfaces. Excel also supports mathematical and statistical tools. For instance, Solver is an excellent tool for solving sets of equations and performing optimization. Databases are effectively and easily accessed. In addition, Visual Basic for Applications offers a powerful object-oriented programming language, capable for constructing commercial graphics interfaces.

It is the main scope of this book to develop complete mathematical models and integrated design procedures for most common food processes. All models and methods presented in this book are appropriate for direct computer implementation. Furthermore, all the application examples are based on programming techniques on Excel. Thus, it is the aim of this chapter to present step-by-step procedures in order to apply the proposed models into the Excel environment. The chapter refers to two main topics. Firstly, the principles for solution of a process design problem are presented and then the principles for Excel implementation are described.

The reader is encouraged to become familiar with the following topics regarding Excel software, using the relative literature:

- Modeling and Spreadsheets
- Analyzing the Solver
- Sensitivity Analysis Using Excel Tables
- Controls and Dialog Boxes to Input Data
- Graphics to Get the Results
- Databases
- Visual Basic as a Programming Language

II. PRINCIPLES OF SPREADSHEET-AIDED PROCESS DESIGN

Computer-aided design is based on computer simulators and computer simulators are based on process modeling. The basic terms modeling, simulation and design are defined in Table 3.1. Modeling is the procedure of translating the physical laws of a process to mathematical equations in order to analyze or design the process. Simulation is the appropriate software, which predicts the real performance of a process. It is based on mathematical modeling plus the appropriate graphics interface in a computer environment. Design is a procedure of sizing and rating a process in order to obtain a specific goal, such as economic production, product quality, and protection of the environment.

Modeling and simulation are useful tools in process design. Table 3.2 summarizes a step-by-step procedure for process modeling, while Table 3.3 summarizes a step-by-step procedure for process simulation. These steps are further analyzed as follows:

Table 3.1 Basic Definitions

Modeling

is the procedure to translate the physical laws of a process to mathematical equations

Simulation

is the appropriate software which guesses the real performance of a process

Design

is a procedure to **size** and **rate** a process in order to obtain a specific goal

Size

Given the process specifications
calculate the equipment size and characteristics

Rate

Given the process specifications and the equipment size and characteristics
calculate the operating conditions

Table 3.2 Process Modeling

1. Process model formulation
 2. Degrees-of-freedom analysis
 3. Alternative problems formulations
 4. Problem-solution algorithm
 5. Cost estimation and project evaluation analysis
 6. Process optimization
-

Table 3.3 Process Simulation Procedure in a Spreadsheet Environment

1. Model development in a spreadsheet
 2. Implementation of alternative problems solution and/or optimization procedures
 3. Development of graphics interface
-

The equations of a model describe the physical laws which are valid for the process. They are derived from material and energy balances, thermodynamic equilibria, transport phenomena, geometry, equipment characteristics, and so forth.

A degrees-of-freedom analysis is shown in Table 3.4 and in Figure 3.1. Suppose that M variables are incorporated into the mathematical model of N equations. Generally, M is greater than or equal to N , and the difference $M-N$ corresponds to the degrees of freedom of the process. The degrees of freedom is characteristic of the process. In process design, some variables have given values, due the design specifications, and the remainder correspond to design variables.

The number of design variables is characteristic of the problem. Several different problems could be defined for every process (see for example Table 3.5). The values for the design variables are decided by the design engineer. The remainder $N \times N$ set of equations is solved using mathematical techniques. In chemical and food engineering the resulting system is rare, that is every variable appears in a few equations. In that case the system can be solved sequentially (down triangle matrix) or by using a few trial variables.

Table 3.4 Degrees-of-Freedom Analysis

| | | |
|---------------------------|---------|------------------------|
| Total number of variables | M | |
| Total number of equations | N | |
| Degrees of freedom | $F=M-N$ | Process characteristic |
| | | |
| Degrees of freedom | F | |
| Problem specifications | K | |
| Design variables | $D=F-K$ | Problem characteristic |

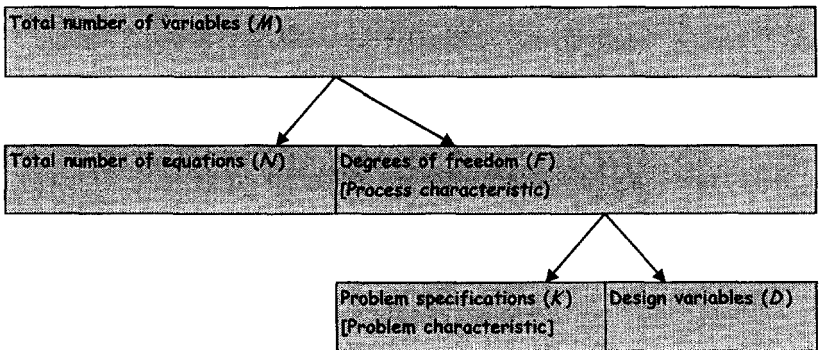


Figure 3.1 Degrees-of-freedom analysis.

Table 3.5 Some Typical Problems

Direct

| | |
|-----------|---|
| Given | the characteristics of input streams the equipment characteristics the operating conditions |
| Calculate | the characteristics of the output streams |

Design

| | |
|-----------|---|
| Given | the characteristics of input streams the characteristics of output streams |
| Calculate | the equipment characteristics the operating conditions |

Rating

| | |
|-----------|--|
| Given | the characteristics of input streams the characteristics of output streams the equipment characteristics |
| Calculate | the operating conditions |

Identification

| | |
|-----------|---|
| Given | the characteristics of input streams the characteristics of output streams the operating conditions |
| Calculate | the equipment characteristics |

The problem of selection of the design variables and the precedence order in which the equations are to be solved is usually based on the structural array. The structural array describes the structure of the process model. The columns of the structural array correspond to all the variables of the process and the rows to all the process equations. An "1" is placed wherever a variable appears in an equation and a "0" otherwise. See for example Table 3.6.

Several algorithms have been proposed in the literature for design variables selection. See for example the Lee-Christensen-Rudd algorithm presented in Table 3.7 (Rudd and Watson, 1968).

Figure 3.2 represents two solution algorithms for the system in Table 3.6. The second is based on the Lee-Christensen-Rudd algorithm.

The values of the design variables are selected so as to optimize an objective function, usually of economic origin as analyzed in Chapter 2. An information flow diagram for process optimization is shown in Figure 3.3.

Table 3.6 Example of a Simple Mathematical Model Along with the Corresponding Structural Array

| Model equations | |
|-----------------|-----------------------|
| E1 | $X_1 / X_2 = a$ |
| E2 | $X_1 + X_2 + X_4 = 0$ |
| E3 | $X_3 = \ln(X_2/X_1)$ |

| Structural array | | | | |
|------------------|-------|-------|-------|-------|
| | X_1 | X_2 | X_3 | X_4 |
| E1 | 1 | 1 | | |
| E2 | 1 | 1 | | 1 |
| E3 | | 1 | 1 | 1 |
| Sum | 2 | 3 | 1 | 2 |

Table 3.7 The Lee-Christensen-Rudd Algorithm for Design Variables Selection

1. Locate a column which contains only one "1" and delete the column and the corresponding row (equation).
2. Repeat step 1 until all equations have been eliminated
3. The remaining variables are the design variables
4. The precedence order of solving the remaining equations is the reverse of the order in which the equations were deleted
5. Should this algorithm not eliminate all equations, a recycle loop has been detected, which cannot be eliminated by the proper selection of the design variables, and certain of the equations must necessarily be solved simultaneously

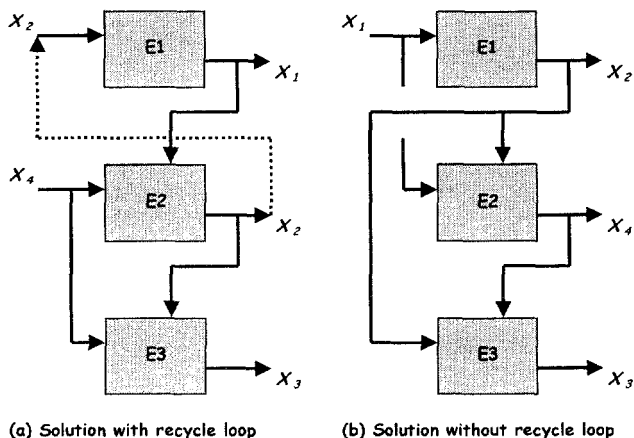


Figure 3.2 Solution algorithms for problem in Table 3.6.

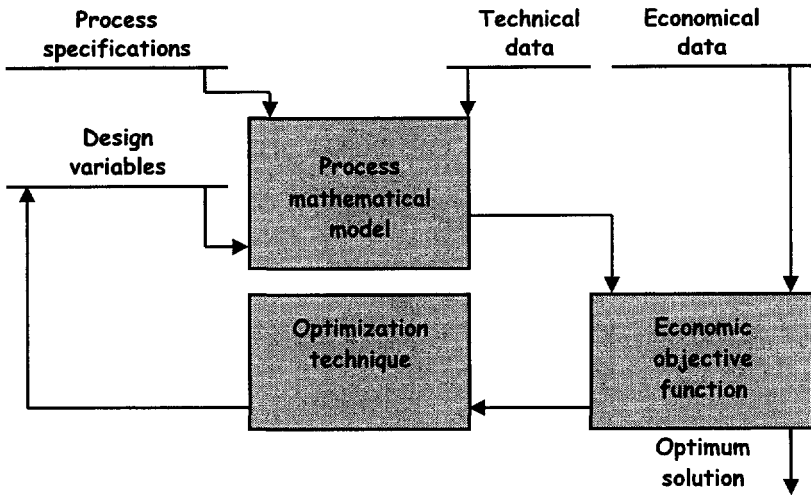


Figure 3.3 Information flow diagram for process optimization.

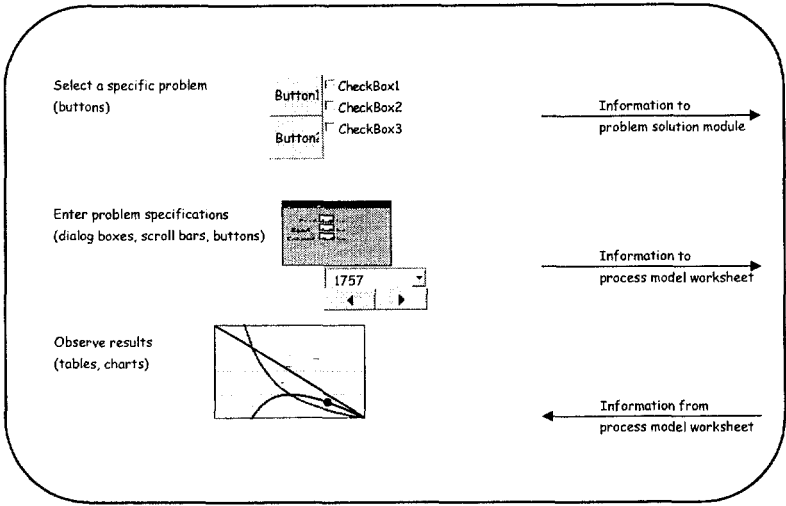
The above procedure is suitable for implementation in a spreadsheet environment. The resulting simulator has generally the outline presented in Figure 3.4. Four different units are distinguished, with each one developed in a different sheet (Maroulis and Kremalis, 1995).

“Process Model Worksheet” is the heart of the system calculations. It contains the process model. When no interactions are needed, the model solution uses only worksheet functions. In that case, when any change in input variables (free variables) occurs, the solution is obtained automatically on this worksheet.

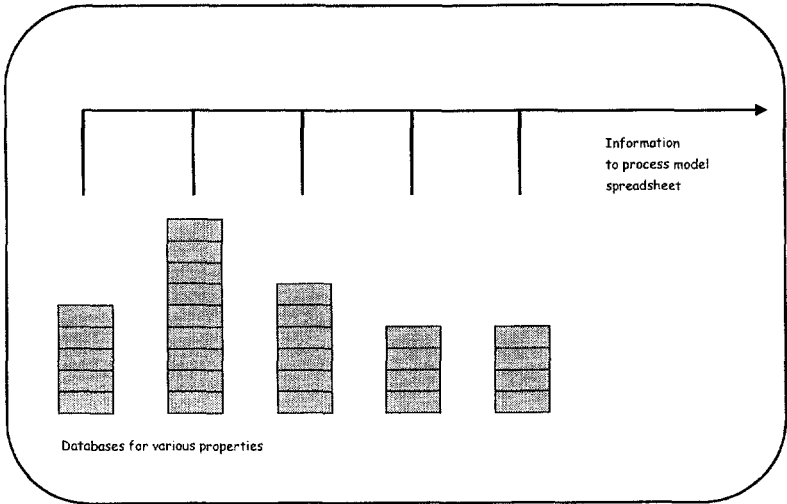
Since the use of the simulator requires the solution of different problems, some different problems are formulated on the “Problem Solution Visual Basic Module”. Their solution is based on the simplest problem of the process model worksheet above, and uses the Solver or the Goal Seek utilities of Excel via a Visual Basic program, to obtain a solution for the alternative problems.

All technical and required data are retrieved from the “Database worksheet”, which contains all the required information in the form of “data lists”. These data are extended and modified via appropriate dialog boxes.

“Graphics interface worksheet” is a user-friendly way for man-machine communication. It usually consists of three parts: (a) Problem specifications: The specifications and the required data for the problem to be solved are entered by the user or estimated from the databases. Data are inserted via dialog boxes or buttons for changing some important magnitudes. (b) Problem type selection: The type of problem to be solved is selected via buttons. (c) Results presentation: The results are obtained automatically, and are presented in the form of tables or charts. Since these charts are updated automatically, the user has at his disposal all the information needed for sizing, rating, sensitivity analysis, or comparison of alternative solutions.

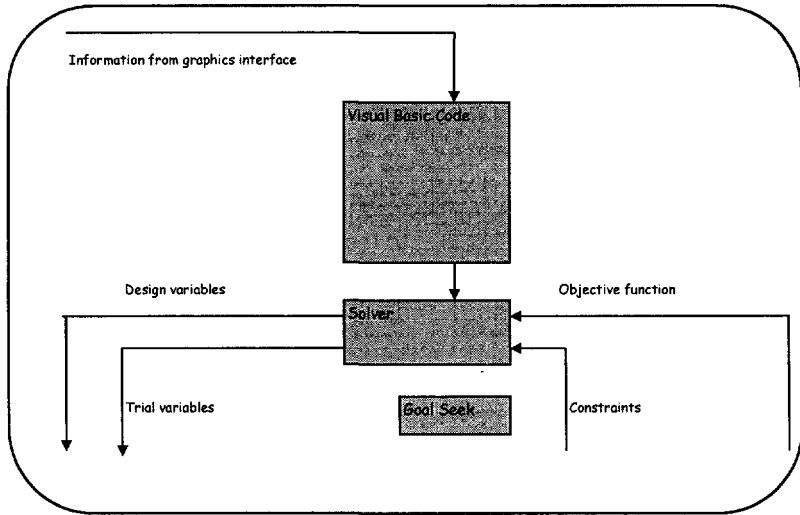


Graphics interface worksheet

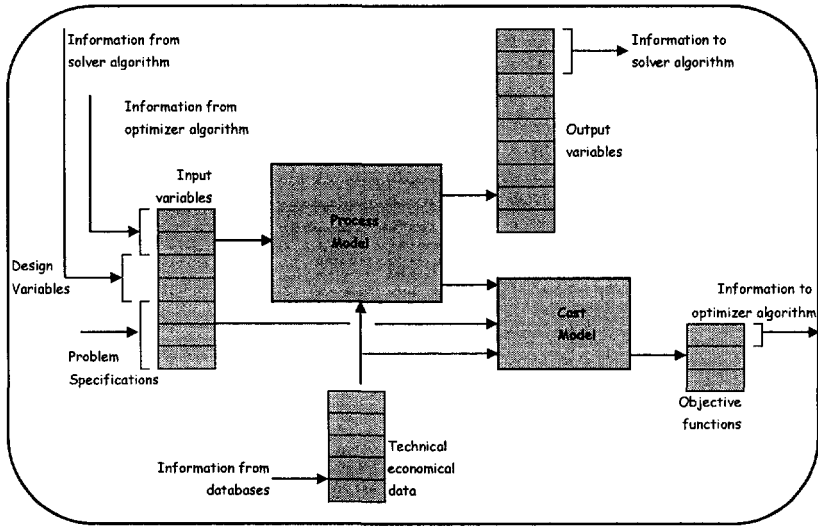


Database worksheet

Figure 3.4 Simulator architecture on a spreadsheet environment.



Problem solution visual basic model



Process model worksheet

Figure 3.4 Continued.

III. APPLICATION EXAMPLE

A step-by-step Excel implementation of a simple design problem on a simplified liquid-liquid extraction system is presented in this section.

1. Process Description

It is proposed to recover a useful compound C from a water solution W by extraction with an organic solvent S (Figure 3.5). The extractor is a mixer-settler system, which uses a pump for mixing and a vessel for settling. The two phases (aqueous and organic) are considered immiscible and the compound C distributes itself between the aqueous and organic phases. The aqueous phase with a flow rate W (kgW/s) and concentration X_o (kgC/kgW) enters the system and exits at the same flow rate and a new composition X (kgC/kgW). The organic phase with a flow rate S (kgS/s) and concentration Y_o (kgC/kgS) enters the system and exits at the same flow rate and a new composition Y (kgC/kgS).

2. Process Model

A simplified mathematical model of the process presented in Figure 3.5 is summarized in Table 3.8. It includes 2 equations: The material balance for the compound C (Equation E1) and the equilibrium of the compound C between the aqueous and the organic phases (Equation E2). The profit P of the process is considered as the objective function for process optimization (Equation O1). The variables involved in the process model as well as the required technical and economical data are also presented in Table 3.8. Considering a typical design problem with three process specifications a degrees-of-freedom analysis suggests one design variable as shown in Table 3.8, and if we select the waste concentration X as design variable, a simple sequential solution algorithm is obtained as shown in Table 3.8. The optimum value for waste concentration X can be obtained by maximizing the profit P . Waste concentration must be varied in the range between zero and feed concentration X_o (process constraints in Table 3.8).

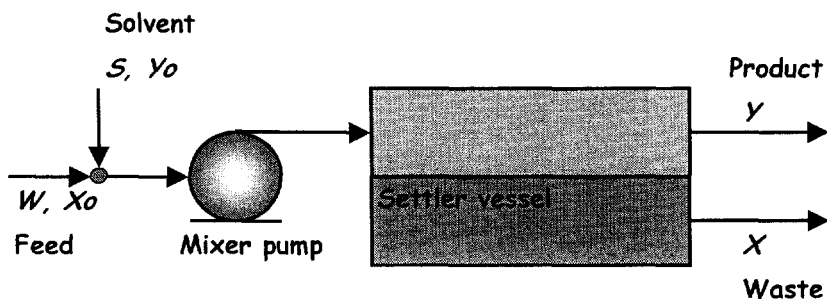


Figure 3.5 Diagram of a simplified liquid/liquid extractor.

Table 3.8 Process Model**Process Equations**

$$W(X_o - X) = S(Y - Y_o) \quad (E1)$$

$$Y = mX \quad (E2)$$

Process Constraints

$$X > 0 \quad (C1)$$

$$X_o - X > 0 \quad (C2)$$

Objective Function

$$P = YS C_p - S C_s \quad (O1)$$

Process Variables

$$W \quad \text{kg/s} \quad \text{Feed flow rate} \quad (V1)$$

$$S \quad \text{kg/s} \quad \text{Solvent flow rate} \quad (V2)$$

$$X_o \quad \text{kgC/kg}W \quad \text{Feed concentration} \quad (V3)$$

$$Y_o \quad \text{kgC/kg}S \quad \text{Solvent concentration} \quad (V4)$$

$$X \quad \text{kgC/kg}W \quad \text{Waste concentration} \quad (V5)$$

$$Y \quad \text{kgC/kg}S \quad \text{Product concentration} \quad (V6)$$

Technical Data

$$m \quad - \quad \text{Equilibrium constant} \quad (T1)$$

Cost Data

$$C_s \quad \$/\text{kg}S \quad \text{Solvent} \quad (R1)$$

$$C_p \quad \$/\text{kg}C \quad \text{Compound} \quad (R2)$$

Process Specifications

$$W \quad \text{kg/s} \quad \text{Feed flow rate} \quad (S1)$$

$$X_o \quad \text{kgC/kg}W \quad \text{Feed concentration} \quad (S2)$$

$$Y_o \quad \text{kgC/kg}S \quad \text{Solvent concentration} \quad (S3)$$

Degrees-of-Freedom Analysis

$$\text{Process variables} \quad 6$$

$$\text{Process equations} \quad \underline{2}$$

$$\text{Degrees of freedom} \quad 4$$

$$\text{Specifications} \quad \underline{3}$$

$$\text{Design variables} \quad 1$$

Design Variables

$$X \quad \text{kgC/kg}W \quad \text{Waste concentration} \quad (D1)$$

Solution Algorithm

$$(E2) \rightarrow Y$$

$$(E1) \rightarrow S$$

$$(O1) \rightarrow P$$

3. Excel Implementation

The following steps comprise an Excel implementation procedure:

1. Workbook preparation
2. Process modeling in a spreadsheet
3. Using “Solver” for process optimization
4. Using graphs and tables for presentation of the results
5. Introducing dialog boxes and controls to modify data
6. Towards an integrated graphics interface

Step 1: Workbook Preparation

Create a new workbook and name it “Extraction.xls”. Insert and name blank sheets as it is presented in Table 3.9.

Step 2: Process Modeling in a Spreadsheet

Into the spreadsheet “Process” consider seven separate ranges, as follows:

- Technical Data
- Design Variables
- Process Specifications
- Economic Data
- Process Model
- Process Constraints
- Economic Model

Each range consists of three columns and some rows, one row for every variable in the range. In each range the first column contains the variable names, the second the variable values or variable formulas, and the third the units used. Name all cells in second columns according the names in the first column. (You can use the “Ctrl Shift F3” option).

The ranges “Technical Data”, “Design Variables”, “Process Specifications” and “Economic Data” contain only data. Insert these data accordingly to Table 3.10. The Ranges “Process Model”, “Process Constraints” and “Economic Model” contain formulas. Insert these formulas accordingly to Table 3.10.

Having inserted data and formulas, the process model implementation has been completed. The resulting spreadsheet “Process” looks like that presented in Figure 3.6. The cell ranges can be colored with different colors. The drawn arrows show the information flow in the spreadsheet.

The spreadsheet process model is ready for use. Any changes in process data, economic data, process specifications, design variables are taken into account and the results are updated immediately.

Any optimization technique, graphical or tabulated reports, any scenario analysis, or sensitivity analysis, any sophisticated graphics interface can be based on the “Process” spreadsheet. Some examples follow.

Step 3: Using “Solver” for Process Optimization

Create a Visual Basic subroutine with the name “optimum” in the “Optimize” module. The appropriate code is shown in Table 3.11.

Table 3.9 Sheets in “Extraction.xls” Workbook

| Sheet Name | Purpose |
|-----------------------------|---|
| Spreadsheets | |
| • Process | Process model |
| • Flowsheet | Process flowsheet |
| • Report | Summary report of results |
| • Control | Graphics interface |
| Visual Basic Modules | |
| • Optimize | Process optimization subroutines |
| • Controls | Subroutines for dialog boxes and controls |
| Dialog Box Sheets | |
| • Spec | Process specifications |
| • Tech | Technical data |
| • Cost | Economical data |

Table 3.10 Cell Content in “Process” Spreadsheet

| Cell name | Cell content | Cell name | Cell content |
|-----------|--------------|-----------|------------------|
| F | 1.00 | S | =F*(Xo-X)/(Y-Yo) |
| Xo | 1.00 | Y | =m*X |
| Yo | 0.00 | P | =Y*S*Cp-S*Cs |
| X | 0.75 | X | =X |
| m | 4.00 | Xo-X | =Xo-X |
| Cs | 1.00 | | |
| Cp | 1.00 | | |

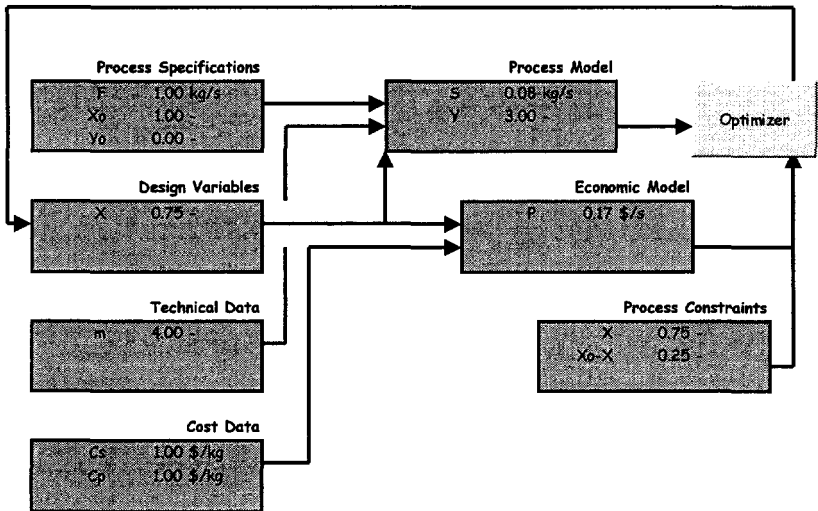


Figure 3.6 Model implementation in the “Process” spreadsheet.

Table 3.11 Visual Basic Subroutine for Process Optimization

```
Sub optimum()  
1 Sheets("Process").Activate  
2 SolverReset  
3 SolverOk SetCell:=Range("objective"), MaxMinVal:=1, ByChange:=Range("variables")  
4 SolverAdd CellRef:=Range("constraints"), Relation:=3, FormulaText:=0#  
5 SolverSolve UserFinish:=True  
6 Beep  
End Sub
```

Statement 1 activates the “Process” spreadsheet. Statement 2 resets the Solver. Statement 3 selects the cell with the name “objective” to be the objective function [SetCell:=Range(“objective”)], requires the maximization of the objective function [MaxMinVal:=1], and selects the range “variables” to be the decision variables [ByChange:=Range(“variables”)]. Statement 4 suggests that all cells in the range “constraints” [CellRef:=Range(“constraints”) must be greater than [Relation:=3] zero [FormulaText:=0#]. Statement 5 activates the solver to find the optimum.

The above mentioned cell names must be defined. Thus, in the sheet “Process” name:

- the cells which contain the values of the design variables as “variables”
- the cells which contain the process constraints as “constraints”
- the cell which contains the profit as “objective”

In the sheet “Process” insert a new button, name it “optimizer” and assign it to the subroutine “optimum”.

Press the button “optimizer” and the optimum is reached in a few seconds as shown in Figure 3.7. A comparison between initial and optimum design can be made using Figures 3.6 and 3.7.

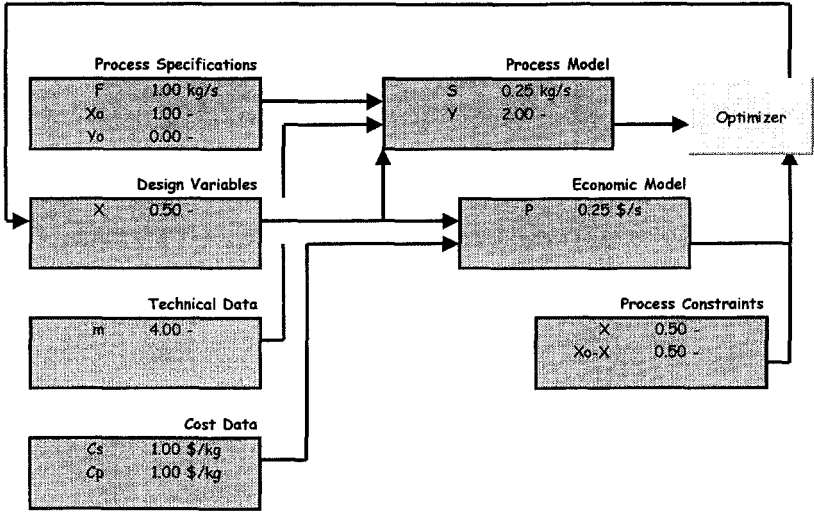


Figure 3.7 Comparison between initial and optimum design.

Step 4: Using Excel Tables and Charts for Presentation of the Results

The process design results can be further analyzed using the tools “Tables” and “Charts” supported by Excel.

For example, a process flowsheet can easily be constructed in Excel as follows: In the sheet “Flowsheet” draw a flowsheet like that presented in Figure 3.8 by using the drawing toolbar. Any information concerning process conditions can be inserted in cells near the desired point of the flowsheet. For each piece of information there need three cells, one for the variable name, one for the variable value, and one for the variable units. That is, in order to insert the feed characteristics select a cell near the icon of the feed arrow and insert the text “W=”, in a neighboring cell insert the formula “=W” to get the value from the “Process” sheet, and in another cell, nearby, insert the text “kgW/s”. You can add any information you like. See for example Figure 3.8. Any changes in data are updated immediately.

In order to plot the effect of the design variable on a technical and an economical variable the following steps can be used: Construct a one-dimensional Excel Table in which the “Column Input Cell” is the cell with the name “X”. The second and third output columns refer to the cells “S” and “P”, respectively. Next construct a “XY(Scatter)” Chart in which the first column of the Table corresponds to x-values and the second to y-values. Similarly, construct a second “XY(Scatter)” Chart in which the first column of the Table corresponds to x-values and the third to y-values. The results are presented in Figure 3.9, in which the optimum design is also shown.

Any other tabulated results or desired reports can be easily obtained as follows: Select a spreadsheet to incorporate the required information. Insert text or graphics as you like. Get the information from the “Process” sheet as described previously in the flowsheet construction procedure. See for example Figure 3.10, which is a summary report of the results obtained.

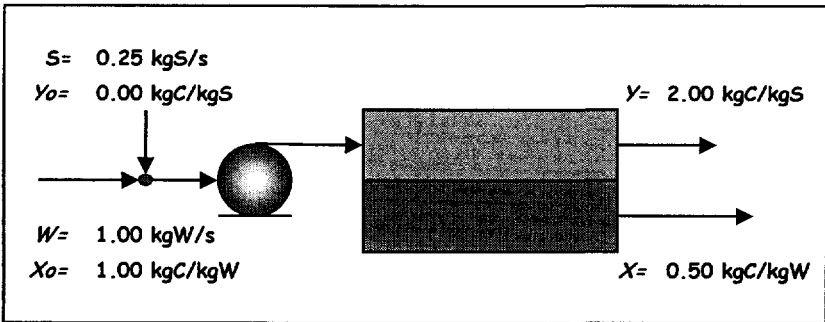


Figure 3.8 Process flowsheet implemented in the spreadsheet “Flowsheet”.

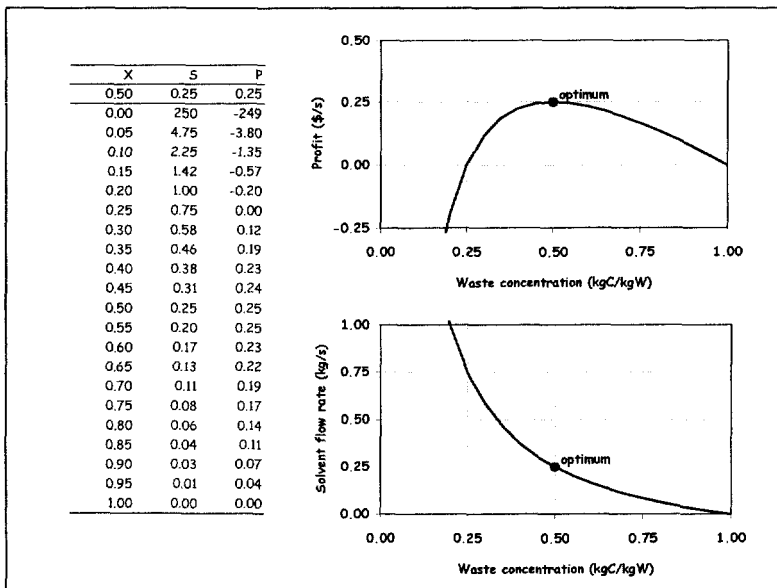


Figure 3.9 Analyze the effect of the design variable on a technical and an economic variable, using the “Tables” and “Charts” tools, supported by Excel.

| Solute Recovery from a Rejected Aqueous Stream Using an Organic Solvent | |
|--|--------------|
| Process | |
| Liquid / liquid extraction | |
| Technical data | |
| Equilibrium constant | 4.00 - |
| Cost data | |
| Solvent cost | 1.00 \$/kg |
| Solute cost | 1.00 \$/kg |
| Specifications | |
| Feed flow rate | 1.00 kgW/s |
| Feed concentration | 1.00 kgC/kgW |
| Solvent concentration | 0.00 kgC/kgS |
| Design variable | |
| Waste concentration | 0.25 kgC/kgW |
| Process design results | |
| Product concentration | 1.00 kgC/kgS |
| Required solvent flow rate | 0.75 kgS/s |
| Economics | |
| Profit | 0.00 \$/s |

Figure 3.10 A summary report of the results obtained.

Step 5: Introducing Dialog Boxes and Controls to Modify Data

A dialog box is used in order to modify the values of process specifications, which are included in the range "Process Specifications" in the spreadsheet "Process".

In the Dialog Module "db_spec" insert for every variable one "Label" (from the toolbar "forms") for its description, one "Edit Box" (from the toolbar "forms") for its value, and one "Label" for its units. Name all the Edit Boxes with the name of the corresponding variable.

In the Visual Basic Module "vb_controls" type a subroutine to use the dialog box in the sheet "d_spec", as described in Table 3.12.

In the spreadsheet "Process" insert a button, name it "specifications", and assign it to the subroutine "DialogSpecifications".

Press the button "specifications" and a dialog box appears in order to modify data for process specifications (see Figure 3.11).

A scrollbar is used for each design variable in order to modify the values of the design variables, which are included in the range "Design Variables" in the spreadsheet "Process".

A scrollbar, in order to handle the waste concentration X , can be inserted as follows:

- Insert the scroll bar icon from the toolbar "forms" as shown in Figure 3.11.
- Insert the minimum allowable value in a cell named "X.min".
- Insert the maximum allowable value in a cell named "X.max".
- Insert the coded value in a cell named "X.CV".

The coded value ranges between 0 and 100 and is defined as follows:

$$X.CV = (X - X.min) / (X.max - X.min) * 100$$

- Insert a scrollbar from the toolbar "forms" and assign the "Cell Link" (in the "Format Object" menu) to the coded value "X.CV".
- Replace the content of the cell named "X" with the following formula:
$$= X.min + X.CV * (X.max - X.min) / 100$$

It must be noted that the range "variables" which is handled by the solver during optimization must be redefined in order to refer to coded values, instead of the actual values. This modification guarantees the proper performance of the optimization and of scrollbars.

Table 3.12 A Subroutine to Activate the Dialog Box

```
Sub DialogSpecifications()  
dbName = "d_spec"  
DialogSheets(dbName).EditBoxes("W").Text = Range("W").Value  
DialogSheets(dbName).EditBoxes("Xo").Text = Range("Xo").Value  
DialogSheets(dbName).EditBoxes("Yo").Text = Range("Yo").Value  
If DialogSheets(dbName).Show Then  
Range("W").Value = DialogSheets(dbName).EditBoxes("W").Text  
Range("Xo").Value = DialogSheets(dbName).EditBoxes("Xo").Text  
Range("Yo").Value = DialogSheets(dbName).EditBoxes("Yo").Text  
End If  
Beep  
End Sub
```

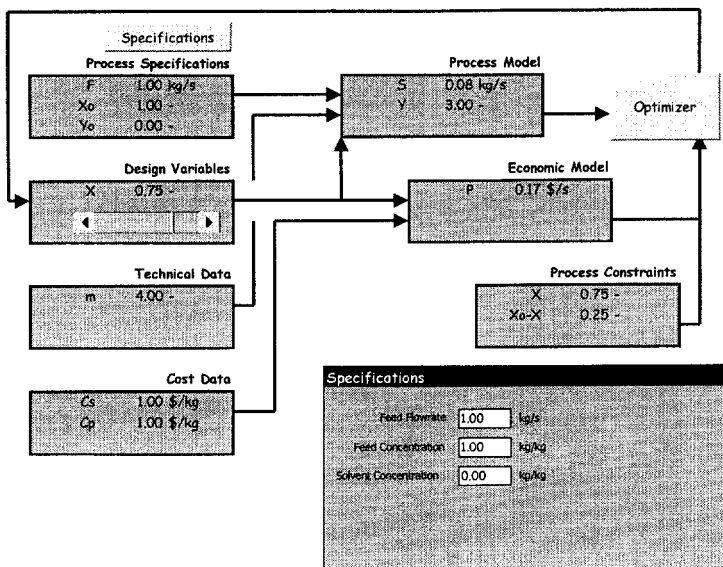


Figure 3.11 Process specifications dialog box.

Step 6: Towards an Integrated Graphics Interface

Any desired graphics interface can be developed in the spreadsheet “Control”. A simple example, presented in Figure 3.12, can be constructed as follows:

- Draw a process flowsheet in sheet “Controls”, as described in Step 5.
- Insert buttons to appear and disappear the crucial graphs
- Insert buttons to activate the desired dialog boxes
- Insert scroll bars to modify the desired process variables
- Insert buttons to solve different problems, e.g., process optimization.

A view of the resulting graphics interface with only one graph activated is shown in Figure 3.13. The user has now at his disposal a process simulator. He can enter data via scroll bars or dialog boxes and observe the results via buttons, which activate the desired graphs or reports.

The graphics interface could be further improved to look professional using appropriate programming code in Visual Basic.

Some integrated graphics interfaces are presented for various processes in Figures 3.14 through 3.17.

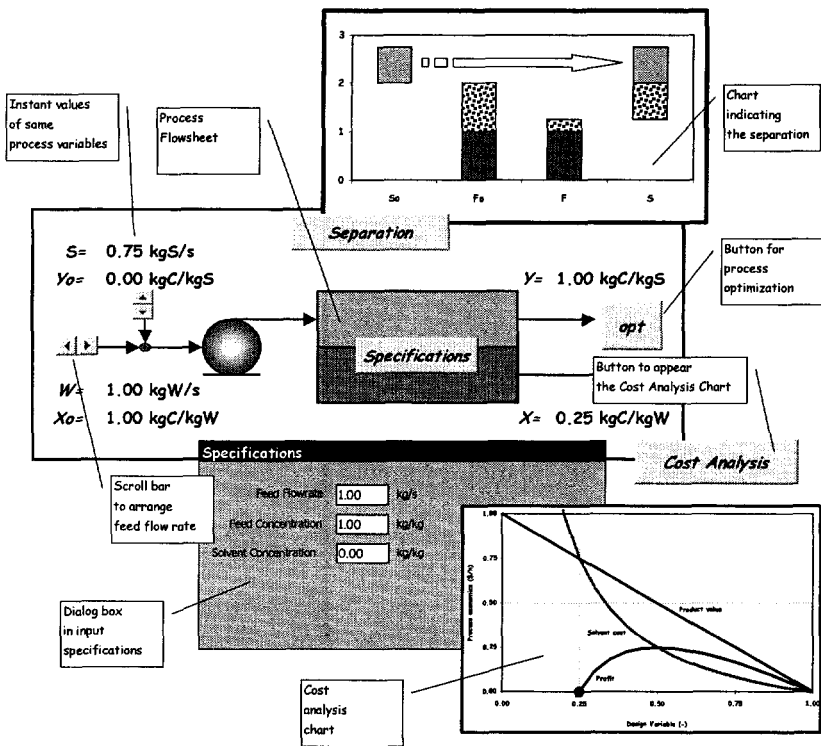


Figure 3.12 Comments on the graphics interface.

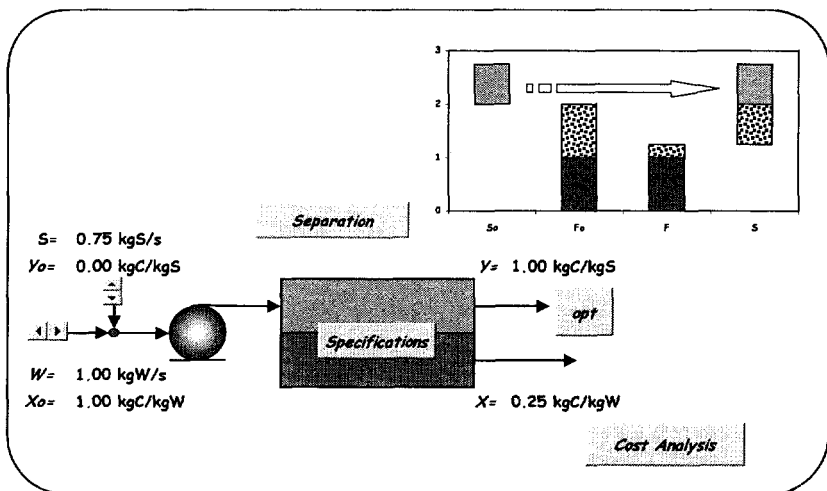


Figure 3.13 Graphics interface.

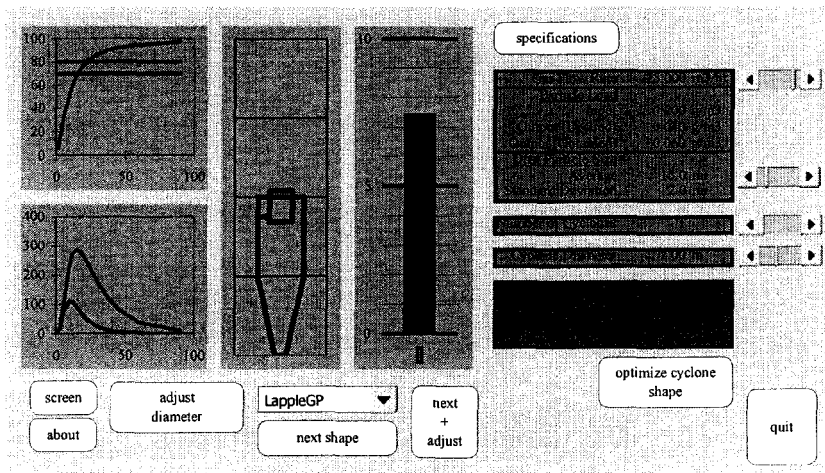


Figure 3.14 Cyclone simulator (Maroulis and Kremalis, 1995).

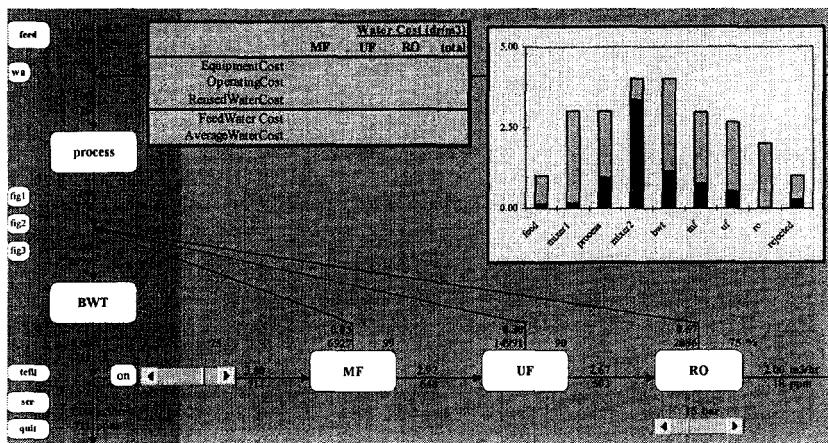


Figure 3.15 Combined UF/RO wastewater treatment system (Voros et al., 1999).

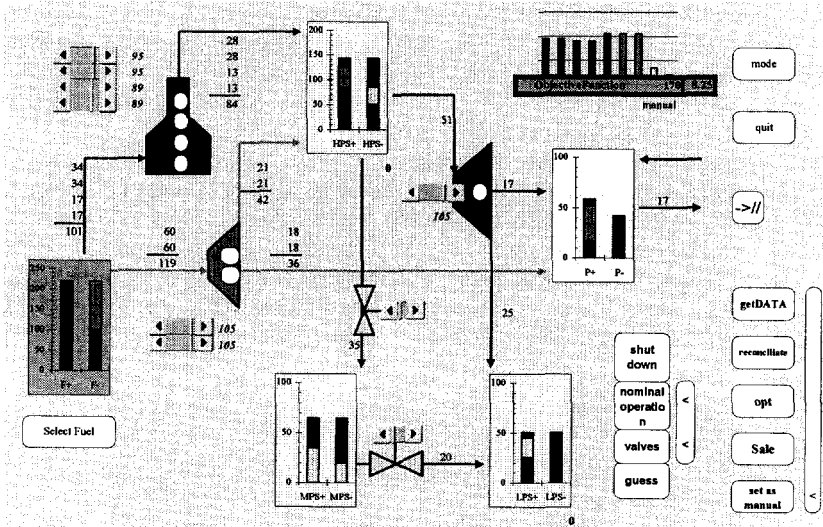


Figure 3.16 Utilities management system (Prokopakis and Maroulis, 1996).

NOMENCLATURE

| | | |
|-------|---------|-----------------------|
| C_p | \$/kgC | Compound cost |
| C_s | \$/kgS | Solvent cost |
| m | - | Equilibrium constant |
| P | \$/s | Profit |
| S | kg/s | Solvent flow rate |
| W | kg/s | Feed flow rate |
| X | kgC/kgW | Waste concentration |
| X_o | kgC/kgW | Feed concentration |
| Y | kgC/kgS | Product concentration |
| Y_o | kgC/kgS | Solvent concentration |

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4

Heating Processes

I. INTRODUCTION

Heating processes are used in most food processing operations as an important part of the manufacturing or preservation processes. The transfer of energy to or from food materials during processing is based on the engineering principles of Heat Transfer, Chemical Engineering, and Food Process Engineering, and the experience of manufacturers of process equipment.

This chapter reviews the heat transfer coefficients and correlations of importance to food systems, and it describes briefly some heat exchangers and heaters used in Food Processing. The basic heat transfer processes, used in industrial food processing, are described in the following chapters of this book: 5 (refrigeration and freezing), 6 (evaporation), 7 (drying), and 8 (thermal processing). Table 4.1 shows the basic heat transfer equipment used in food processing and food preservation operations.

In this chapter, the principles of heat transfer are summarized together with a brief description of food heat transfer processes and equipment not described in other chapters, such as heat exchangers, ovens, roasters, and some novel equipment, e.g., microwave and ohmic heating processes. Table 4.2 shows some special heat transfer processes and equipment, described in this chapter.

Food heating processes may involve chemical, biochemical, and biological changes of the food material that must be considered simultaneously with the physical heating process. The main purpose of special heating processes, such as cooking, baking, roasting, and frying, is to improve the eating quality of the food products.

Heat exchangers are used widely in the process industries to recover thermal energy from hot streams, while preheating cold streams.

The design and evaluation of heat exchangers and heating equipment requires quantitative data on the heat transfer coefficients, which are either estimated empirically or determined from empirical correlations. High heat transfer coefficients are desired for faster transfer of heat energy and more economical operation.

Table 4.1 Basic Heat Transfer Processes in Food Processing

| Equipment | Function |
|----------------|---|
| Cooling | Preservation of fresh foods |
| Freezing | Food preservation |
| Evaporation | Concentration of liquid foods |
| Drying | Drying / dehydration of foods |
| Blanching | Inactivation of enzymes |
| Pasteurization | Inactivation of enzymes and microbes |
| Sterilization | Inactivation of microbes and microbial spores |

Table 4.2 Special Heat Transfer Processes and Equipment Used in Food Processing

| Equipment / Processes | Function |
|-----------------------|--------------------------------------|
| Heat Exchangers | Heat Recovery |
| Ovens | Baking, cooking of foods |
| Roasters | Thermal treatment of foods |
| Frying | Deep-fat or dry frying of foods |
| Infrared Heating | Heating, drying of foods |
| Microwave Heating | Heating, thawing, or drying of foods |

II. HEAT TRANSFER COEFFICIENTS

1. General Considerations

The operation of heating equipment is based on the transfer of heat from a heating medium to a food material, usually by forced convection. Natural convection, contact, and radiation heating may also be involved in some applications. The cooling equipment operates on the same principles, but heat is transferred from the food material to a cooling medium, which can also be a cold food stream.

The heat transport within food materials is expressed by the two basic heat transport properties, thermal conductivity (λ , W/m K) and thermal diffusivity (α , m²/s), defined by the Fourier equations for one-dimensional (x) transfer:

$$q/A = \lambda (\Delta T/x) \quad (4-1)$$

$$\partial T/\partial t = \alpha (\partial^2 T/\partial x^2) \quad (4-2)$$

where q/A is the heat flux (W/m^2) and $\Delta T/x$ is the temperature gradient (K/m) in the x direction.

Data on the thermal transport properties of foods are presented in tables, databanks, and books, such as Rahman (1995) and Saravacos and Maroulis (2001).

The thermal diffusivity α is usually calculated from the more available thermal conductivity λ , using the equation:

$$\alpha = \lambda / \rho C_p \quad (4-3)$$

where ρ is the density (kg/m^3) and C_p is the specific heat (J/kg K) of the material.

Densities, specific heats and other physical properties of foods are found in food properties books, e.g., Rahman (1995). Some typical physical properties of foods, useful in the design and operation of heat exchangers and other food processing equipment are given in the Appendix of this book.

Heat exchangers and heat transfer equipment are characterized by the (film) heat transfer coefficient (h , $\text{W/m}^2\text{K}$) and the overall heat transfer coefficient (U , $\text{W/m}^2\text{K}$), defined by the equations:

$$q/A = h \Delta T \quad (4-4)$$

$$q/A = U \Delta T \quad (4-5)$$

In Equation (4-4), ΔT is the temperature difference between the bulk of the fluid and the heat transfer surface. Equation (4-5) is applied to heat transfer through the wall of a heat exchanger, and ΔT refers to the overall temperature difference between the two fluids, e.g., heating medium and food material. The units of ΔT are degrees Kelvin (K) or degrees Celsius ($^{\circ}\text{C}$).

The overall resistance to heat transfer ($1/U$) through a wall of thickness x and thermal conductivity λ is given by the equation:

$$1/U = 1/h_1 + x/\lambda + 1/h_2 \quad (4-6)$$

where h_1 and h_2 are the (film) heat transfer coefficients of the two sides (1, 2) of the heat transfer wall.

Equation (4-6) applies to clean flat heat transfer surfaces, and to tubular surfaces of diameter larger than 25 mm (1 inch). For tubes of smaller diameter, the inner and outer diameters must be considered (Saravacos and Kostaropoulos, 2002).

In practice, the heat transfer surfaces of the heat exchangers are not clean, due to fouling, i.e., deposits which increase significantly the thermal resistance. Thus, a fouling resistance term FR should be added to the overall resistance Equation (4-6):

$$1/U = 1/h_1 + x/\lambda + 1/h_2 + FR \quad (4-7)$$

Table 4.3 shows some typical heat transfer coefficients h in food heat exchange systems (Rahman, 1995; Saravacos and Maroulis, 2001).

Correlations of heat transfer data are useful for estimating the (film) heat transfer coefficient h in various processing equipment and operating conditions. These correlations contain, in general, dimensionless numbers, characteristic of the heat transfer mechanism, the flow conditions, and the thermophysical and transport properties of the fluids. Table 4.4 lists the most important dimensionless numbers used in heat transfer operations (Saravacos and Maroulis, 2001).

The Reynolds number ($Re = uL/\nu$) is used widely in almost all correlations. In this number, the velocity u is in (m/s), the length L is in (m) and the kinematic viscosity or momentum diffusivity ($\nu = \eta/\rho$) is in (m^2/s). The length L can be the internal diameter of the tube, the equivalent diameter of the non-circular duct, the diameter of a spherical particle or droplet, or the thickness of a falling film. In non-circular geometries, the characteristic length L is taken as twice the hydraulic diameter ($L = 2d_H$) of the flow channel, which is twice the hydraulic radius of the duct ($d_H = 2r_H$), defined as $r_H = A/WP$, where A is the cross sectional area and WP is the “wetted” perimeter.

Some dimensionless numbers are used also in mass transfer correlations, i.e., Bi , St , and j_M . In these numbers, the heat transfer coefficient h is replaced by the mass transfer coefficient k_c .

Table 4.5 shows some heat transfer correlations of general application. For natural convection, the parameters a and m characterize the various shapes of the equipment and the conditions of the fluid (McAdams, 1954; Perry and Green, 1997; Geankoplis, 1993; Saravacos and Maroulis, 2001).

The ratio of tube diameter to tube length (d/L) is important in the laminar flow ($Re < 200$), but it becomes negligible in the turbulent flow in long tubes ($L/d > 60$). For shorter tubes, the ratio (d/L) should be included in the correlation.

The viscosity ratio η/η_w refers to the different viscosity in the bulk of the fluid η and at the tube wall η_w . This ratio becomes important in highly viscous fluids, like oils, in which the viscosity drops sharply at the high wall temperatures, increasing the heat transfer coefficient.

Several other correlations have been proposed in the literature for different heat transfer in fluid systems, like flow outside tubes, flow in packed beds, etc.

The heat transfer coefficients of condensing vapors have been correlated to the geometry of the tubes and the properties of the liquid film or droplets. Very high heat transfer coefficients are obtained by drop-wise condensation.

Most liquid foods are non-Newtonian fluids, and their apparent viscosity η_a is related to the rheological constants (K , n) of the power-law model (Saravacos and Maroulis, 2001).

Most non-Newtonian liquid foods are pseudoplastic, i.e., $n < 1$, and their apparent viscosity decreases as the shear rate (velocity, agitation) is increased (shear-thinning fluids). Therefore, the heat transfer coefficient will increase at higher flow velocities, or speeds of agitation. Empirical correlations have been suggested in the literature.

The heat transfer coefficient of air and water in some important operations can be estimated from simplified dimensional equations, applicable to specific equipment geometries and system conditions (Perry and Green, 1997; Geankoplis, 1993).

Table 4.3 Typical Film Heat Transfer Coefficients

| Heat Transfer System | h , kW/m ² K |
|-----------------------|---------------------------|
| Air drying | 0.02 – 0.10 |
| Baking oven | 0.03 – 0.20 |
| Air cooling | 0.01 – 0.06 |
| Air freezing | 0.02 – 0.10 |
| Water in pipe flow | 5.0 |
| Falling film of water | 4.0 |
| Condensing steam | 10.0 |

Table 4.4 Dimensionless Numbers in Heat and Mass Transfer

| Number | | Applications |
|----------------------|---|-----------------|
| Reynolds | $Re = u L / \nu$ | Flow processes |
| Nusselt | $Nu = h L / \lambda$ | Heat transfer |
| Prandtl | $Pr = \nu / \alpha$ | Heat transfer |
| Grashof | $Gr = L^3 g (\Delta \rho / \rho) / \nu^2$ | Free convection |
| Graetz | $Gz = G A C_p / \lambda L$ | Heat transfer |
| Biot | $Bi = h L / \lambda$ | Heat transfer |
| Stanton | $St = h / G C_p$ | Heat transfer |
| Schmidt | $Sc = \nu / D$ | Mass transfer |
| Heat transfer factor | $j_H = St Pr^{2/3}$ | Heat transfer |
| Mass transfer factor | $j_M = St Sc^{2/3}$ | Mass transfer |

A , interfacial area (m²); L , length (m); α , thermal diffusivity (m²/s); C_p , specific heat (J/kg K); g , acceleration of gravity (9.81 m/s²); $G = u\rho$, mass flow rate kg/m²s; h , heat transfer coefficient (W/m²K); η , viscosity (Pa s); ρ , density (kg/m³); D , mass diffusivity (m²/s); $\nu = \eta/\rho$, kinematic viscosity (m²/s); u , velocity (m/s)

Table 4.5 General Heat Transfer Correlations

| Heat Transfer System | Correlation |
|------------------------------------|--|
| Natural convection | $Nu = a (Gr Pr)^m$ |
| Laminar inside tubes | $Nu = 1.86 [Re Pr (d/L)]^{1/3} (\eta/\eta_w)^{0.14}$ |
| Turbulent inside long tubes | $Nu = 0.023 Re^{0.8} Pr^{1/3} (\eta/\eta_w)^{0.14}$ |
| Parallel to flat plate (laminar) | $Nu = 0.664 Re^{0.5} Pr^{1/3}$ |
| Parallel to flat plate (turbulent) | $Nu = 0.0366 Re^{0.8} Pr^{1/3}$ |
| Flow past single sphere | $Nu = 2.0 + 0.60 Re^{0.5} Pr^{1/3}$ |

Dimensionless numbers defined in Table 4.4. a and m , parameters of natural convection characteristic of the system (Perry and Green, 1997); L , d length and diameter of tube. Long tubes ($L/d > 60$)

2. Heat and Mass Transfer Factors

The heat transfer factor of various heating processes j_H is related to the Reynolds number of the system Re by the empirical generalized correlation (Saravacos and Maroulis, 2001):

$$j_H = m Re^n \quad (4-8)$$

where m and n are characteristic constants of the equipment / material system.

Regression analysis of compiled literature data on various heating operations of foods has yielded the following generalized correlation:

$$j_H = 0.344 Re^{-0.423} \quad (4-9)$$

It is thus possible to estimate an approximate value of the heat transfer coefficient h for a given flow condition (Re), using the equation:

$$j_H = St_H Pr^{2/3} = [h / (u \rho C_p)] Pr^{2/3} \quad (4-10)$$

where $St_H = h / (u \rho C_p)$ Stanton number, and $Pr = (\eta C_p) / \lambda$, Prandtl number.

By analogy, the mass transfer factor j_M for mass transfer operations in food systems can be estimated from the empirical generalized equation:

$$j_M = 1.11 Re^{-0.54} \quad (4-11)$$

The mass transfer factor is related to the dimensionless numbers Schmidt Sc and Stanton St_M by the equation:

$$j_M = St_M Sc^{2/3} \quad (4-12)$$

where $Sc = (\eta / \rho D)$, $St_M = k_c / u$, and η (viscosity, Pa s), D (diffusivity, m²/s), k_c (mass transfer coefficient, m/s), λ (thermal conductivity, W/m K), u (velocity, m/s), and ρ (density, kg/m³).

Specific empirical correlations of the j_H and j_M factors as functions of Re for various food processes have been estimated by Saravacos and Maroulis (2001).

III. FOOD HEATING EQUIPMENT

Food heating equipment includes conventional heat exchangers used mainly for fluid foods, and special heating equipment, used mostly for solid and semi-solid food products.

1. Heat Exchangers

The design of heat exchangers in the process industries is described in general chemical engineering books, such as Perry and Green (1997), Walas (1988), and Sinnott (1996). Several types of heat exchangers are used in food processing, suitable for heating, cooling, sterilizing, baking, etc, of various food products. Special heat exchangers, used in basic food processes (Table 4.1), such as evaporation, drying, and refrigeration/freezing, are described in various chapters of this book. General aspects of heat exchangers, and special heating equipment (Table 4.2) are discussed in this chapter.

The heating rate (heat load) of a heat exchanger (q , kW), required to heat the product (m , kg/s) by a temperature difference (ΔT , K or °C), is estimated from the equation:

$$q = m C_p \Delta T \quad (4-13)$$

where C_p (kJ/kg K) is the specific heat of the product.

The design and operation of heat exchangers is based on the overall heat transfer coefficient U , which is defined by the generalized form of equation:

$$q/A = U \Delta T_L \quad (4-14)$$

where U is the overall heat transfer coefficient, A is the heat transfer area, and ΔT_L is the logarithmic mean temperature difference, defined by equation (4-15):

$$\Delta T_L = (\Delta T_1 - \Delta T_2) / \ln (\Delta T_1 / \Delta T_2) \quad (4-15)$$

where ΔT_1 and ΔT_2 are the temperature differences (K or °C) at the two ends of the exchanger.

When the two temperature differences (ΔT_1 , ΔT_2) are equal or close to each other, the arithmetic temperature difference $\Delta T_M = (\Delta T_1 + \Delta T_2) / 2$ should be used, instead of the logarithmic ΔT_L .

The overall heat transfer coefficient U is either calculated from the overall thermal resistance of the heat exchanger, including fouling (Equation 4-7) or taken from empirical data or measurements of a similar heat exchanger-product system.

The overall heat transfer coefficient U_i refers normally to the internal surface of the tube or heated surface, which represents the major thermal resistance, since the food material flows usually inside the tubes and it creates most of the fouling. The heating (or cooling) medium flows outside the tubes, and it is normally characterized by higher heat transfer coefficients and negligible fouling.

The fouling factor FR of Equation (4-7) or the corresponding “fouling coefficient” $h_f = 1/FR$ are difficult to predict accurately, and for this reason, empirical overall heat transfer coefficients U are used in practice. Reliable U values are obtained in pilot plant or industrial-scale measurements of heat transfer rates on similar equipment, processing similar products under similar operating conditions. Fouling of heat exchangers discussed by Bott (1995). O’ Donnell (2001) described the cleaning of heat exchangers. Modeling and simulation of fouling of milk heat exchangers was discussed by Georgiadis et al. (1998a, 1998b).

In most food heat exchangers, stainless steel is used, which has a comparatively low thermal conductivity ($\lambda = 15 \text{ W/m K}$), and the thermal resistance of the wall (x/λ) can become significant, particularly in thick-walled tubes or other heat transfer equipment (e.g., scraped surface heat exchangers).

Table 4.6 shows some typical overall heat transfer coefficients of various heat exchangers used in food processing, under normal operating conditions, i.e., moderate fouling (Saravacos and Kostaropoulos, 2002).

a. Tubular Heat Exchangers

The simplest heat exchanger is the double-pipe type, which consists of two concentric pipes with the cold stream flowing usually in the inner tube and the heating medium in the annulus. It operates in either co-current or counter-current flow.

The double-pipe heat exchangers have the disadvantage of limited heat transfer surface, which led to the shell and tube heat exchangers, used widely in the chemical process industries.

Figure 4.1 shows a diagram of a counter-current shell and tube heat exchanger, which consists of a bundle of parallel tubes enclosed in a shell. In heating applications, the cold stream flows through the tubes, while the heating medium (e.g., saturated steam) flows in the shell. The heat exchanger can be operated in co-current, counter-current, or cross-flow. Large heat transfer surfaces can be produced by assembling many tubes of small diameter in a shell. The tubes are fixed on two tubesheets, creating two heads at the ends of the heat exchanger. Two or more tube passes may be used to increase the residence time of the tube fluid in the heat exchanger. However, multiple passes in tubular heat exchangers develop high pressure drop and they are cleaned and sanitized with difficulty.

The shell and tube heat exchangers are less expensive than the other types of food heat exchangers, especially when high heat loads are transferred. They can be operated at higher temperatures and pressures (e.g., steam at 6 bar and 160°C) than the plate heat exchangers. However, they cannot be cleaned and sanitized easily, like the plate heat exchangers. A tubular heater exchanger used for heating fruit juices was described by Youngworth and Swientek (1987).

Table 4.6 Typical Overall Heat Transfer Coefficients U in Heat Exchangers

| Heat exchange system | U , kW/m ² K |
|--------------------------------|---------------------------|
| Steam / Water | 2.50 |
| Water / Water | 2.00 |
| Water / Liquid food | 1.50 |
| Water or steam / Viscous food | 1.00 |
| Water or steam / Vegetable oil | 0.80 |
| Water or steam / Air (gas) | 0.10 |
| Air (gas) / Air (gas) | 0.03 |

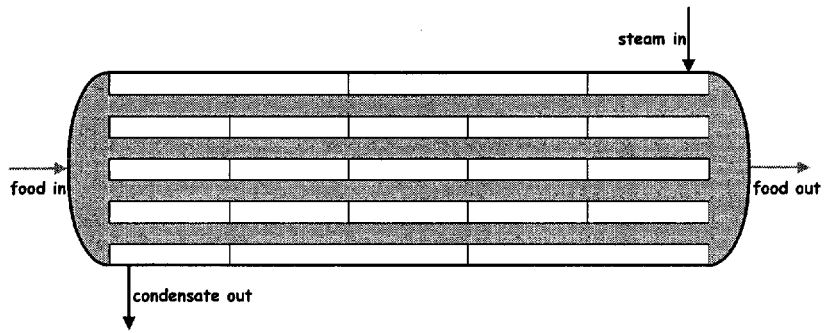


Figure 4.1 Shell and tube heat exchanger.

The design and applications of the shell and tube heat exchangers are discussed by Perry and Green (1997), Walas (1988), and Bhatia and Chermisinoff (1980). Detailed specifications on the dimensions of tubes and shells and on the construction of heat exchangers are described in technical publications of suppliers of heat exchange equipment, such as the Tubular Exchangers Manufacturers Association (*TEMA*, 1978).

The heat transfer coefficients tend to increase as the tube diameter is reduced, while the opposite effect is observed with the pressure drop. The heat transfer coefficient h , according to the empirical correlations, is proportional to the fluid velocity $u^{0.33}$ in the laminar flow, or to $u^{0.8}$ in the turbulent flow. The pressure drop Δp is proportional to the velocity u in the laminar flow, or about to the square of velocity u^2 in the turbulent flow. The optimum operation of a heat exchanger should combine the highest heat transfer coefficient with a minimum pressure drop. High pressure drop increases power requirement (kW) for pumping the fluid through the heat exchanger.

Concentric tube heat exchangers consist of double, triple or more concentric tubes with associated entry and exit ports for the product and the heating/cooling medium (Lewis and Heppell, 2000). Corrugated, instead of smooth, tubes may be used to increase turbulence and heat transfer rate. In three or more concentric tubes, the heating/cooling medium and the product flow in alternate annuli and give the largest heat transfer area. They may be used in the final heating stage of *UHT* sterilizers.

b. Plate Heat Exchangers

The plate heat exchangers are used widely in the food industry because of their high thermal efficiency, compactness, and hygienic and cleaning design. Higher heat transfer coefficients h are obtained, due to the geometry of plate heat exchangers, which promotes turbulent flow at low Reynolds numbers (Alfa-Laval, 1969, 1971)

They consist of closely spaced thin plates, clamped together in a frame. A thin gasket seals the plates round their edges. The liquid product and the heating / cooling medium flow alternately in co-current or counter-current fashion (Figure 4.2). The plates are corrugated (patterns of ridges) for increasing the rigidity of the plate and improving heat transfer by promoting turbulence.

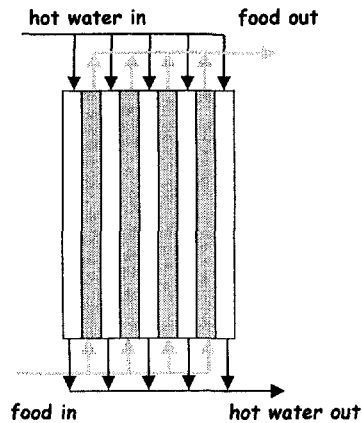


Figure 4.2 Plate heat exchanger.

Gaskets used for *PHE* include the elastomers nitrile-butadiene (*NBR*) and isobutylene-isoprene (*IIR*) for temperatures up to 135°C, and fluorocarbon polymers for higher temperatures (Carlson, 1992).

Dimensions and capacities of plate heat exchangers: (Muller and Steinhagen, 1997): Standard plates of effective heat transfer areas 0.02-4.75 m², plate thickness 0.6-0.9 mm, channel spacing (gaps) 2-6 mm, flow rates 0.5-5 000 m³/h, connection diameters 25-400 mm, up to 1 000 plates per pack, operating temperature -35 to 200°C, pressure up to 25 bar.

The heat transfer coefficient h in plate heat exchangers is correlated as Nusselt Nu number to the Reynolds Re and Prandtl Pr numbers by empirical equations of the form (Fryer, 1997):

$$Nu = a Re^m Pr^{1/3} \quad (4-16)$$

Typical values for the parameters of this correlation for $Re > 5$ are $a = 0.352$ and $m = 0.539$.

The Re number is based on the equivalent diameter of the flow channel, which is 4 times the hydraulic radius, defined as the ratio of the cross sectional area over the wetted perimeter.

In general, the heat transfer coefficients in plate heat exchangers (*PHE*) are higher than those predicted by the classical correlations (Table 4.5). This is caused by the geometry and flow patterns of the fluids on the heat transfer surfaces, which develop turbulence at low Reynolds numbers, e.g., $Re > 10$ (Saravacos and Kostaropoulos, 2002).

The design (sizing) of a plate heat exchanger to handle a heat load of (q , kW) is based on the calculation of the total heat transfer surface area A_t from the basic Equation (4-14). Typical overall heat transfer coefficients in *PHE* are 3 000 W/m² K (aqueous liquid foods of apparent viscosities up to 1 Pa s) and 800 W/m²K (vegetable oils). In plate heat exchangers, the log mean temperature difference is approximately equal to the arithmetic mean temperature difference between the heating medium and the product.

The total number of the required plates N_t is calculated from the equation (Alfa-Laval, 1971):

$$N_t = A_t / A_o = 2 N_p N_s \quad (4-17)$$

where A_o is the surface area of one plate, and N_p is the number of plate channels in series, and N_s is the number of plate passes in series.

The design pressure of *PHE* decreases from about 25 bar to 5 bar, as the operating temperature is increased from about 0°C to 200°C (limitation due to rubber gasket materials). A typical industrial *PHE* may have plates of heat transfer surface 0.4 m²/plate and total surface up to 100 m² (250 plates).

c. Scraped Surface Heat Exchangers

The scraped surface heat exchangers (*SSHE*) are used for heating or cooling highly viscous, fouling, and crystallizing liquid foods, such as margarine and ice cream. They are double-pipe heat exchangers with agitated scrapers in the inner (product) pipe, and the heating or cooling medium in the outer pipe (Figure 4.3).

The scrapers often consist of several knives, positioned in such way that, besides scraping, a screw pumping effect is achieved. The *SSHE* can be operated either horizontally or in vertical position.

The product is pumped through the *SSHE* with a positive displacement pump, and high pressure drops (Δp) up to 10 bar may develop during operation.

Pipe diameters commonly used are 15 cm (inside) and 20 cm (outside). The speed of an agitated scraper is in the range of 500-700 *RPM*. The *SSHE* are often used in series in double stacks (Perry and Green, 1997). Due to the mechanical construction and the moving parts, *SSHE* are more expensive than any other heat exchanger used in food processing.

Heat transfer in *SSHE* is accomplished by heating (or cooling) the product layer adjacent to the heat transfer wall, and scraping and mixing it with the bulk of the fluid food. Back-mixing of the fluid in the agitated pipe may reduce the heat transfer rate. There are no reliable empirical correlations to predict the (film) heat transfer coefficient in *SSHE*. The complex hydrodynamics of fluid food flow in *SSHE* is discussed by Halstrom et al. (1988).

The thermal design of *SSHE* is based on empirical (experimental) overall heat transfer coefficients U , with typical values in the range of 500 to 1000 W/m²K, which are relatively low, due to the high resistance of thick walls of the heat transfer pipe. In order to obtain a high heat flux, temperature differences ΔT between the heating medium and the product of about 25°C are used, which are much higher than the low ΔT used in plate heat exchangers (about 5°C).

Scraped surface heat exchangers are relatively expensive equipment, and they can be justified only for very viscous fluids, e.g., viscosities higher than 10 Pa s, and particulate suspensions. Less expensive tubular heat exchangers should be used whenever the food product can be pumped through a piping system, e.g., pseudoplastic tomato pulp or paste.

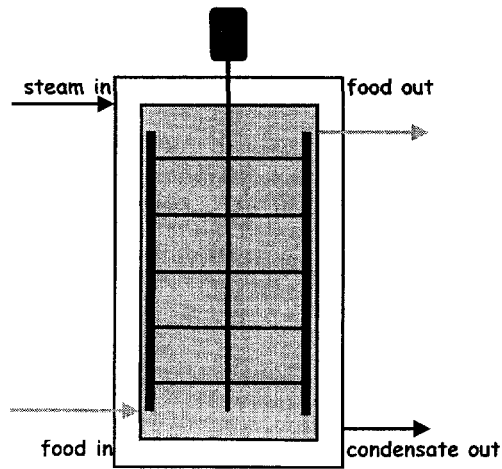


Figure 4.3 Scraped surface heat exchanger.

d. Agitated Kettles

Agitated kettles and agitated vessels are used for heating (or cooling) viscous and highly non-Newtonian fluid foods, which cannot be handled in normal (plate) heat exchangers. They are also used, in small scale, for open-evaporation of food pulps and viscous suspensions. Several types of agitators are used, such as propeller, paddle, blade, helical ribbon, and anchor. Figure 4.4 shows the diagram of a jacketted kettle with propeller agitator, used for heating viscous foods.

When propeller agitators are used, the propeller axis is inclined (about 15°) against the vertical central axis of the kettle, for reducing circulatory flow and swirling (Perry and Green, 1997).

The heat transfer coefficient h in the product side is calculated by the empirical correlation (Saravacos and Kostaropoulos, 2002):

$$Nu = a Re^{2/3} Pr^{1/3} (\eta/\eta_w)^{0.14} \quad (4-18)$$

where, the parameter a depends on the type of agitator and the nature of the fluid. In agitated vessels, the Re number is defined as $Re = (d_i^2 N \rho) / \eta$, where (d_i) is the diameter of the impeller, and N is the speed of rotation, $N = RPM/60$, 1/s. The Nusselt number is defined as $Nu = (h d_T) / \lambda$, where d_T is the tank (vessel) diameter.

Table 4.7 shows some a values for various types of agitators (Saravacos and Moyer, 1967; Hallstrom et al., 1988; Singh, 1992). The parameter a depends on the type of flow (Re), the geometry and location of the agitator (e.g., flat blade, curved blade, or retreating blade), and the presence of baffles in the vessel. Baffles have no significant effect on heat transfer in viscous pseudoplastic fluids.

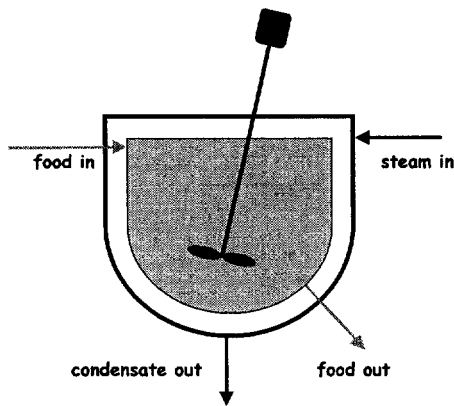


Figure 4.4 Agitated kettle.

Experimental data on heating fruit juices and purees (pulp) in an anchor-agitated kettle yielded overall heat transfer coefficients U in the range of 1.0 to 2.0 $\text{kW/m}^2\text{K}$ for clarified juices and sugar solutions, and 0.8 to 1.2 $\text{kW/m}^2\text{K}$ for fruit purees, when the agitation speed was increased from 30 to 100 RPM (Saravacos and Moyer, 1967). The U values of pseudoplastic fluids (e.g., fruit purees) increase at a faster rate with increasing agitation than the U of Newtonian fluids (e.g., clarified juices and sugar solutions), due to the lower apparent viscosity at higher shear rates (shear-thinning effect).

Tables 4.7 Parameters a of Empirical Equation (4-18) for Agitated Vessels

| Agitator type / Fluid product | a |
|------------------------------------|-------|
| Paddle, $\text{Re} < 4\,000$ | 0.415 |
| Paddle, $\text{Re} > 4\,000$ | 0.36 |
| Blade-flat, $\text{Re} < 400$ | 0.54 |
| Blade-flat, $\text{Re} > 400$ | 0.74 |
| * Helical ribbon $\text{Re} > 130$ | 0.238 |
| Scraping anchor / Newtonian | 0.55 |
| Scraping anchor / pseudoplastic | 1.474 |

* The second part of Equation (4-18) should be multiplied by $(i/d)^{-0.25}$ where i is the pitch and d is the diameter of the agitator.

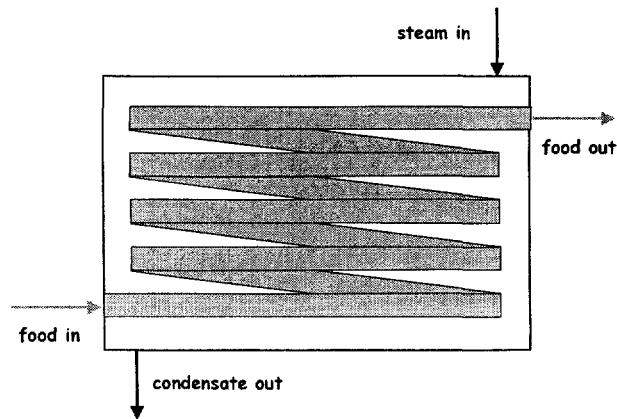


Figure 4.5 Spiral-tube heater.

e. Spiral-tube Heat Exchangers

Spiral-tube (helical) heat exchangers, with the product inside the tube and the heating medium (e.g., condensing steam) in the outside jacket (Figure 4.5), are used for the fast heating of liquids. High heat transfer coefficients can be achieved, since turbulence is obtained at lower Re numbers. Spiral tubes are suitable for fast heating of liquids in *UHT* aseptic sterilizers (Chapter 8).

Spiral (helical) tubes induce turbulence at lower Reynolds (Re) numbers than straight tubes, by reducing the ratio of maximum to average fluid velocity (u_{max}/u_{av}). The turbulence in spiral tubes is expressed by the Dean (Dn) number, which is defined as $Dn = Re(d/D)^{1/2}$, where d and D are the tube and coil diameters, respectively. Experimental values of $u_{max}/u_{av} = 1.61$ and $Dn = 168$ were obtained by Palazoglu et al. (2002).

High heat transfer coefficients are achieved in helical heat exchangers, due to improved turbulence and mixing of the fluid product. Coronel et al. (2000) obtained experimental heat transfer coefficients for water and pseudoplastic solutions in the range of 3500 to 9000 W/m^2K .

2. Direct Heating

Direct heaters are based on mixing of heating steam with the liquid food product, resulting in very fast heat transfer rate. The steam used for direct heating, e.g., of milk, must be clean, tasteless, and free of any dissolved gases and toxic components, coming from the steam boiler. The feed water to the boiler should be of culinary (potable) quality and contain no chemical additives.

Mixing of steam with the liquid food can be accomplished by two methods (Hallstrom et al., 1988): a) Steam injection, by injecting steam into the fluid food through small holes, and b) Steam infusion, by mixing the steam with films and droplets of the liquid food, sprayed in a special vessel (Figure 4.6).

Direct heating of foods minimizes fouling, by eliminating the heat transfer surface. The condensed water from the heating steam is usually removed by vacuum flashing, which also reduces the temperature and removes off-flavors and odors from the liquid product, as in *UHT* sterilization (Chapter 8).

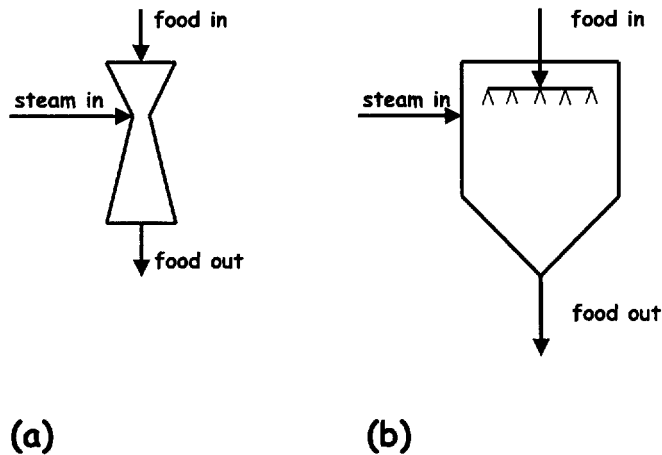


Figure 4.6 Direct heating of liquid food: (a) steam injection; (b) steam infusion.

3. Special Heating Equipment

a. Baking Ovens

Baking is used to prepare bread, biscuits, meats, vegetables, etc. for eating, by various heating processes. The term “roasting” has the same meaning, but it usually refers to the thermal treatment of meat, nuts, or coffee. Although ovens are synonymous to baking ovens (Fellows, 1990), their use in connection to coffee, cocoa, and nuts is mainly related to the development of color and aroma.

The ovens consist of either a compartment of several shelves, or a tunnel through which the product is baked on a conveyor belt (Hallstrom et al., 1988). They are usually heated by hot air, sometimes mixed with steam, using either natural convection or forced circulation. Radiation heating from the oven walls to the product may be also involved.

Air temperatures of 150-250°C are used in baking bread and meat. The wet product is heated by the hot air or hot combustion gases until the surface layer is dried, forming a crust at 100°C, while its interior remains at a lower temperature. The heat and mass transfer mechanisms of baking and roasting are similar to the mechanisms of air-drying (Chapter 7). The heat transfer coefficients in baking depend on the air velocity and temperature in the oven, varying in the range of 20-120 W/m²K (Hallstrom et al., 1988).

The heat transfer coefficient in convection ovens can be estimated from the following empirical correlation of the heat transfer factor (Saravacos and Maroulis, 2001), which is based on published experimental data:

$$j_H = 0.8 Re^{-0.4} \quad (4-19)$$

where the (dimensionless) heat transfer factor is defined as $j_H = h / u \rho C_p$, and h is the heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$), ρ is the density (kg/m^3), u is the velocity (m/s), and C_p is the specific heat ($\text{J}/\text{kg K}$) of the air (gases).

Heat transfer to the baking ovens can be: a) Direct heating by combustion gases from a clean gas fuel, such as natural gas or *LPG*, or from a *MW* power source; b) indirect heating from heated oven walls, from steam tubes, or from electrical resistances (Fellows, 1990). Direct heating is preferred, because it is faster and more efficient.

Small forced-circulation ovens are used for baking bread and other products, placed on racks/trolleys, or fixed/rotating shelves. Hot air (gas or oil burning) at $50\text{-}250^\circ\text{C}$ is circulated by a fan at velocities $1\text{-}10\text{ m/s}$. In some applications electrical heating may be used, or live steam may be injected in the oven to accelerate the heat transfer process.

In large installations, tunnel ovens are used (Figure 4.7), equipped with fans at different locations, which recirculate the hot air (Saravacos and Kostaropoulos, 2002). Forced circulation ovens start with a preheating cycle of about 1 hour, before loading with the product; baking (air) temperatures of $150\text{-}220^\circ\text{C}$ are used, and the motion of belts or chains is continued after heating is stopped, to avoid mechanical damage due to thermal expansion. A traveling band baking oven that utilizes various heating systems is described by Mermelstein (1999). The oven is 0.40 m wide and 15 m long, and it can be heated by gas burning (radiating), air convection, high velocity-impingement jets, and microwaves. Conveyorized proofing and baking systems, retailer and wholesaler ovens, and other baking equipment are described by Matz (1989).

Computer modeling and simulations can be utilized to optimize and control the operation of commercial ovens (Zanoni et al., 1997).

High heat transfer rates in ovens can be achieved by steam injection and steam condensation on the products. Combination of steam injection and air impingement improves heat transfer and increases product yield.

In the baking of biscuits, continuous ovens, with long metallic conveyor belts, are used. Such belts can be $40\text{-}45\text{ m}$ long, and they may be divided into 3 or more compartments, in which the temperature may rise progressively to $210\text{-}270^\circ\text{C}$, with total residence time $4\text{-}6\text{ min}$. Multi-deck traveling belts are used for reducing the length of continuous ovens, used in baking products, such as French toast, Swedish rolls, and several cakes.

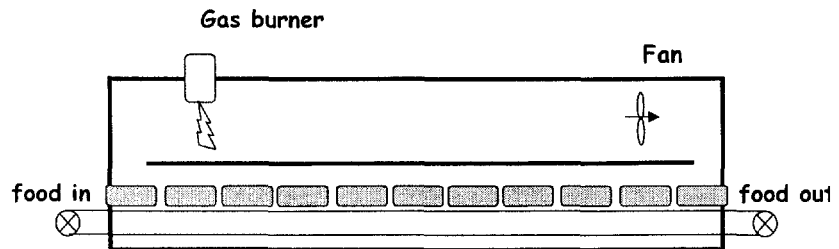


Figure 4.7 Continuous forced circulation baking oven.

b. Impingement Heating

Air-impingement heating is used for fast heating of food products, e.g., cookies, breakfast cereals, pizzas, and tortilla chips, in various ovens and dryers. Air impingement jets, at velocities 15-30 m/s, are directed vertically against the surface of food products, moving usually on a conveyor belt.

Heat transfer coefficients of 150-250 W/m²K are obtained by air impingement, compared to 30-50 W/m²K in air-convection heating (Karwe, 2002; Singh, 2002). Combinations of air impingement/microwaves or air impingement/convection/microwaves are used in commercial ovens to bake, cook, or thaw a variety of food products (Norris et al., 2002; Swackhamer, 2002).

c. Frying

Frying is a high-temperature treatment of foods by contact with a hot surface (pan) or by immersion in hot fat or oil. It is used primarily to improve the eating quality of foods. At the same time, most of the spoilage microorganisms are inactivated, and a surface crust is formed, which improves the preservation and the quality of the food product (Fellows, 1990). Crust formation removes the free water from the food material, creating empty pores, which are filled with oil. The technology of frying of food products is described by Moreira (1999), and Rossell (2001).

Continuous fryers are often used in commercial applications. The product is carried on conveyor belts or in baskets through the hot oil at temperatures 160-180°C. Film heat transfer coefficients in deep-fat (convection) frying are in the range of 200-300 W/m²K. The oil is heated by electrical resistances, gas, fuel oil, or steam. Fryers and associated baking equipment are described by Matz (1989).

Depending on the product, a 5-6 m long continuous fryer has a capacity of 1.5-2.0 ton/h. The residence time in a continuous fryer may vary in the range of 3-30 min.

d. Radiation Heating

Radiation heating refers mostly to heating by infrared radiation (wavelengths of 1-1000 μm). It is applied to some special cases of heating food products (Saravacos and Kostaropoulos, 2002).

Thermal radiation from a body is a strong function (4th power) of its absolute temperature. The emissivity e of a body is defined as $e = W / WB$, where W and WB are the radiated energies of the specific body and the black body ($e=1$), respectively. Polished metals have low emissivities (e.g., $e=0.05$), but for normal stainless steel $e=0.6$. Water and several food materials have emissivities higher than 0.9. Typical food emissivities are: dough 0.85, lean beef 0.74, and fat beef 0.78 (Fellows, 1990).

The heat exchange ($q_{1,2}$, W) between two “gray” (non-black) surfaces (1) and (2), which are kept at temperatures (T_1 , K) and (T_2 , K) is given by the equation (Perry and Green, 1997):

$$q_{1,2} = 5.675 A_1 \Phi_{1,2} [(T_1/100)^4 - (T_2/100)^4] \quad (4-20)$$

where, A_1 is the radiating surface, and $\Phi_{1,2}$ is the overall radiation exchange factor, defined by the equation:

$$1/\Phi_{1,2} = 1/F_{1,2} + (1/e_1 - 1) + (A_1/A_2)(1/e_2 - 1) \quad (4-21)$$

where $F_{1,2}$ is the “view factor”, i.e., how surface A_1 “sees” surface A_2 , and (e_1, e_2) are the emissivities of the two surfaces. The view factors are given in tables of the heat transfer literature (McAdams, 1954; Perry and Green, 1997).

Equation (4-20) can be written in the classical heat transfer form as:

$$q_{1,2} = h_r A_1 \Delta T \quad (4-22)$$

where, h_r is a radiation “heat transfer coefficient” (W/m^2K), and $(\Delta T = T_1 - T_2)$ is the temperature difference between the two surfaces.

The radiation heat transfer coefficient can be calculated by combining Equations (4-20) and (4-22).

When both heat convection and radiation are significant, the overall heat transfer rate can be calculated from the equation:

$$q_{1,2} = (h_c + h_r) A_1 \Delta T \quad (4-23)$$

where h_c is the convection heat transfer coefficient.

e. Infrared Heaters

The temperature of the infrared (*IR*) radiators determines the spectral distribution and the maximum emitted radiation flux. The penetration of *IR* radiation is limited. Short wavelength radiation ($\lambda < 1.25 \mu\text{m}$) is preferred in food processing, because it has higher penetrating power (Halstrom et al., 1988). Radiation heating should avoid the overheating and burning of the food surface.

Two types of *IR* radiators are used in food processing: a) Gas-heated, which give large wavelengths, and b) electrically heated, which include the tubular and ceramic heaters (long λ), and the quartz and halogen heaters (short λ). Some high-intensity radiators require water or air-cooling to avoid overheating.

Water vapor and carbon dioxide absorb part of the *IR* radiation, reducing the efficiency of *IR* heating of foods, e.g., during the end of the baking process, when the product may be sprayed with water.

IR radiation is used for the drying of vegetables, in baking/roasting, and in frying of foods. The main advantages are fast heating, high efficiency, and easy process control.

f. Heat Generation Processes

Heat generation processes are based on the conversion of electrical energy into heat within the food material. Electrical heating can be achieved either as a result of friction during molecular rotation of water and other molecules (microwave or dielectric heating) or by the electrical resistance of the material to the passage of electric current (ohmic heating).

g. Microwave and Dielectric Heating

The most common radiations used in electrical heating of foods are the microwaves (*MW*) at frequencies 915 or 2 450 MHz, and dielectric or radio-frequency (*RF*) at frequencies 3-30 MHz (1 Hz = 1/s).

The heat generation rate (Q , W) per unit volume of a material is given by the equation:

$$Q = 0.56 \times 10^{-10} E^2 \epsilon' \omega \tan \delta \quad (4-24)$$

where E is the electric field strength (V), ω is the frequency (1/s), ϵ' is the dielectric constant, and $\tan \delta$ is the loss tangent of the material, defined as $\tan \delta = \epsilon'' / \epsilon'$, where ϵ'' is the dielectric loss.

The dielectric constant ϵ' is a measure of the *MW* rate of penetration in the food. It indicates the ability of material (food) to store electrical energy. The dielectric loss ϵ'' is a measure of energy dissipation.

The dielectric properties are difficult to predict and they are measured experimentally. They depend strongly on the composition of the food material, and they usually decrease with increasing temperature, with liquid water having the highest values. Typical values of dielectric constants and dielectric losses (ϵ' , ϵ'') are (Datta et al., 1995): water (80, 20), fruit (55, 15), meat (40, 10), vegetable oils (2.5, 1.5), and ice (3.2, 0.003).

The very low dielectric constants of ice are of importance in the microwave thawing of frozen foods, since the liquid water tends to be heated much faster than the melting of ice, damaging the food quality. For this reason, microwave thawing should be controlled carefully (tempering of frozen foods).

In addition to the dielectric constants, the penetration depth of *MW* and *RF* radiation in the materials is of fundamental importance in heating applications. The penetration depth d , defined as the depth where the radiation intensity decays by 37 % ($1/e$) of its surface value, is given by the equation:

$$d = \lambda_0 (\epsilon')^{0.5} / 2 \pi \epsilon'' = \lambda_0 / 2 \pi (\epsilon')^{0.5} \tan \delta \quad (4-25)$$

where λ_0 is the wavelength of the radiation in the air (Schiffmann, 1987).

According to Equation (4-25), the penetration of radiation in a material increases with the wavelength. Thus, *MW* radiation at 915 MHz penetrates more than the 2 450 MHz radiation, i.e., 30 versus 10 mm.

Application of *MW* and *RF* to the heating of food products requires the knowledge of dielectric properties (ϵ' , ϵ'') and the thermal and transport properties (C_p , λ , α , D) of the food materials.

The electrical properties of foods, related to *MW* and *RF* heating, were investigated in the *COST 90 bis* cooperative project of the European Union (Jowitt et al., 1987). Applications of the electrical properties to food processing are discussed by Mudgett (1990, 1995), Kent (1987), Ohlsson (1987), and Datta et al. (1995).

Equipment used in *MW-RF* food processing is discussed by Decareau and Peterson (1986), Schiffmann (1987), Mudgett (1990), Reuter (1993), and Datta and Anantheswaran (2001). *MW-RF* equipment used in food dehydration is re-

viewed in Chapter 8. Magnetrons, waveguides, and applicators are used in various arrangements to produce and direct the radiation to the food product, which is usually moved on a conveyor belt. For pasteurization and cooking at temperatures below 100°C, open conveying systems are used with a hood to keep the water vapors (or steam) and prevent surface drying-out of the product. For high-temperature sterilization (110-130°C), closed systems are used. When food in plastic trays or pouches is sterilized, overpressure (air or steam) is required to prevent bursting of the packages. The presence of steam during *MW-RF* treatment reduces also the corner and edge effects and the cold spots of the food products.

Food applications of *MW-RF*, in addition to drying (Chapter 7), include tempering of frozen foods, precooking of meat and other foods, and pasteurization and sterilization of foods.

Tempering of frozen foods is one of the most important *MW-RF* applications (Mermelstein, 1999). The frozen product is heated to a temperature just below the freezing point and then allowed to fully thaw at low temperature, reducing sharply the normal thawing time. The process is applied to the packaged frozen food, without taking the packages apart, and it reduces significantly drip losses (more economical than conventional thawing).

MW-tempered frozen meat and fish (in bulk) can be processed fast by slicing, grinding, and forming into patties, which can be frozen again for storage.

Precooking of meat, poultry, and other foods by *MW-RF* is faster than conventional heating. The products are packaged in plastic or tray containers (Mermelstein, 1997).

Capacity and cost data of *MW* food installations were presented by Edgar (1986).

h. Ohmic Heating

Ohmic or electric resistance heating refers to heat generation within the food material by electrical current. It is a function of the electrical resistance of the food product and the applied electrical potential. It is particularly suitable for heating fluid particulate foods, in which the particles cannot be heated evenly with the food liquid during conventional (heat convection) heating. The food suspension is heated by passing through special electrodes.

Low-frequency alternating current (50 to 60 Hz) is supplied to the ohmic heating column, which consists of 4 or more electrodes. The electrodes are connected using stainless steel spacer tubes, lined with insulating plastics, e.g., polyvinylidene (APV, 2000).

The heating column is mounted in a vertical position with the flow of product in the upward direction. Each heating section has the same electrical impedance, and the interconnecting tubes increase in length towards the outlet, to account for the increasing ionic mobility by the increased temperature.

Commercial scale ohmic heating systems have been designed for power outputs of 75 kW and 750 kW, corresponding to product capacities of approximately 750 and 3 000 kg/h, respectively, for a temperature rise of water of 75°C.

IV. SIMPLIFIED DESIGN OF A HEAT EXCHANGER

A rough estimation of the required heat transfer area, and consequently a preliminary estimation of the equipment and operating cost, of a heat exchanger of any type can be obtained using the simplified approach described in this section.

The main assumptions for rough estimation are:

- The overall heat transfer coefficient is constant and independent on the geometry of the exchanger and the flow conditions.
- The pumping cost is negligible in comparison to the heating cost.

1. Process Description

A typical flowsheet for a steam heater is presented in Figure 4.8. The process stream with a flow rate F (kg/s) at temperature T_1 ($^{\circ}\text{C}$) is heated in a heat exchanger to temperature T_2 ($^{\circ}\text{C}$). Saturated vapor with a flow rate F_s (kg/s) at temperature T_s ($^{\circ}\text{C}$) is used as a heating medium. The condensate is assumed to be saturated liquid with the same flow rate and temperature.

The heat exchanger has a total heat transfer area A (m^2), through which heat Q (kW) is transferred using a driving force ΔT_m ($^{\circ}\text{C}$).

2. Process Model

A simplified mathematical model of the process presented in Figure 4.8 is summarized in Table 4.8. It includes 4 equations: The energy balance for the process stream (Equation E01), the energy balance for the utility stream (Equation E02), the definition of the heat transfer driving force (Equation E03) and the phenomenological equation of heat flow between the two streams (Equation E04). In Table 4.8 are also listed eight process variables and three technical data of the model. Consequently the degrees of freedom of the process are 4 (that is 8 variables minus 4 equations).

In a typical design problem presented in Table 4.9 the feed flow rate, the initial and the target temperatures of the process stream, as well as the steam temperature are known, that is the number of specifications are four, which spend four degrees of freedom and consequently no design variables are available for this problem. The solution algorithm is simple and is also presented in Table 4.9.

The total annualized cost of the process TAC is selected as the objective function to be optimized. It is calculated in Table 4.10. Equation (F01) calculates the equipment cost C_{eq} , (F02) the annual operating cost C_{op} , while Equation (F03) defines the total annualized cost TAC , which is a weighted sum between the equipment and operating cost. The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (F04). The cost data needed for the cost estimation are also summarized in Table 4.10.

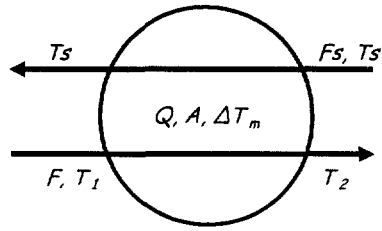


Figure 4.8 Flowsheet for a typical steam heater.

Table 4.8 Simplified Model of a Heat Exchanger

Process Equations

$$Q = FC_p(T_2 - T_1) \quad (\text{E01})$$

$$Q = F_s \Delta H_s \quad (\text{E02})$$

$$\Delta T_m = \frac{(T_s - T_1) - (T_s - T_2)}{\ln[(T_s - T_1)/(T_s - T_2)]} \quad (\text{E03})$$

$$Q = AU\Delta T_m \quad (\text{E04})$$

Process Variables

| | | | |
|--------------|----------------|-----------------------------|-----|
| F | kg/s | Feed flow rate | (1) |
| F_s | kg/s | Steam flow rate | (2) |
| T_1 | °C | Feed temperature | (3) |
| T_2 | °C | Target temperature | (4) |
| T_s | °C | Steam temperature | (5) |
| ΔT_m | °C | Mean temperature difference | (6) |
| Q | kW | Thermal load | (7) |
| A | m ² | Heat transfer area | (8) |

Technical Data

| | | | |
|--------------|---------------------|-----------------------------------|-----|
| U | kW/m ² K | Overall heat transfer coefficient | (1) |
| C_p | kJ/kg K | Specific heat | (2) |
| ΔH_s | kJ/kg | Latent heat of steam condensation | (3) |

Table 4.9 Solution of a Typical Design Problem**Process Specifications**

| | | | |
|-------|------|--------------------|-----|
| F | kg/s | Feed flow rate | (1) |
| T_1 | °C | Feed temperature | (2) |
| T_2 | °C | Target temperature | (3) |
| T_s | °C | Steam temperature | (4) |

Degrees-of-Freedom Analysis

| | |
|--------------------|----------|
| Process variables | 8 |
| Process equations | <u>4</u> |
| Degrees of freedom | 4 |
| Specifications | <u>4</u> |
| Design variables | 0 |

Solution Algorithm

(E01) $\rightarrow Q$

(E02) $\rightarrow F_s$

(E03) $\rightarrow \Delta T_m$

(E04) $\rightarrow A$

Table 4.10 Cost Analysis of Heat Exchanger**Equipment cost**

$$C_{eq} = C_{exc} A^{n_{exc}} \quad (\text{F01})$$

Annual operating cost

$$C_{op} = C_s Q t_y \quad (\text{F02})$$

Total annualized cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (\text{F03})$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (\text{F04})$$

Cost Data

| | | |
|-----------|-------------------|-------------------------------|
| C_{exc} | \$/m ² | Heat exchanger unit cost |
| n_{exc} | - | Heat exchanger scaling factor |
| C_s | \$/kWh | Cost of heating steam |
| t_y | hr/yr | Annual operating time |
| i_r | - | Interest rate |
| l_f | yr | Lifetime |

3. Application to Tomato Paste Heating

The previous design procedure is applied to tomato paste heating and the results are summarized in Table 4.11. Process specifications are compatible with the requirements of the process described in the application of Chapter 2.

The results presented in Table 4.11 are based on the assumption that the overall heat transfer coefficient is about $500 \text{ W/m}^2\text{K}$. In order to analyze the accuracy of the results a sensitivity analysis is presented in Figure 4.9.

Figure 4.9 suggests that the calculated heat transfer area is strongly related with the design value of the overall heat transfer coefficient, especially in the range of low heat transfer coefficients. Thus for safe design the minimum estimated value of overall heat transfer coefficient should be used.

However, a more detailed design procedure is needed in order to take into account the geometrical characteristics of the exchanger to obtain more accurate values for the overall heat transfer coefficient.

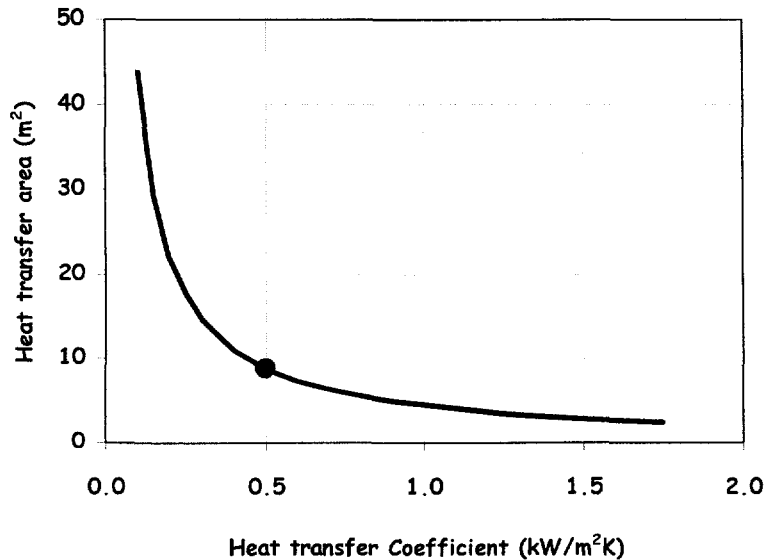


Figure 4.9 Flowsheet for a typical steam heater.

Table 4.11 Process Design Results**Process Specifications**

| | | |
|-------|----------|--------------------|
| F | = 1 kg/s | Feed flow rate |
| T_1 | = 50 °C | Feed temperature |
| T_2 | = 100 °C | Target temperature |
| T_s | = 120 °C | Steam temperature |

Technical Data

| | | |
|--------------|----------------------------|-----------------------------------|
| U | = 0.50 kW/m ² K | Overall heat transfer coefficient |
| C_p | = 3.50 kJ/kg K | Specific heat of fluid food |
| ΔH_s | = 2200 kJ/kg | Latent heat of steam condensation |

Model Solution

| | | |
|--------------|--------------------|-----------------------------|
| Q | = 175 kW | Thermal load |
| F_s | = 0.08 kg/s | Steam flow rate |
| ΔT_m | = 40 °C | Mean temperature difference |
| A | = 9 m ² | Heat transfer area |

Cost Data

| | | |
|-----------|---------------------------|-------------------------------|
| C_s | = 0.04 \$/kWh | Cost of heating steam |
| C_{exc} | = 3.00 k\$/m ² | Heat exchanger unit cost |
| n_{exc} | = 0.65 | Heat exchanger scaling factor |
| t_y | = 1400 hr/yr | Annual operating time |
| i_r | = 0.09 | Interest rate |
| l_f | = 5 yr | Lifetime |

Cost Estimation Results

| | | |
|----------|---------------|-------------------------|
| e | = 0.257 | Capital recovery factor |
| C_{eq} | = 12.3 k\$ | Equipment cost |
| C_{op} | = 8.6 k\$/yr | Operating cost |
| TAC | = 12.0 k\$/yr | Total annualized cost |

V. DETAILED DESIGN OF A SHELL AND TUBE HEAT EXCHANGER

In this section, a detailed design approach is described for a shell and tube heat exchanger. Equipment geometry is taken into account and consequently a detailed estimation of the heat transfer coefficient and the corresponding pressure loss can be obtained. A similar design procedure for plate heat exchangers is presented in the next Section.

1. Process Description

A typical flowsheet of a shell and tube steam heater is presented in Figure 4.10. The process stream with a flow rate F (kg/s) and a temperature T_1 ($^{\circ}\text{C}$) is heated in a heat exchanger to temperature T_2 ($^{\circ}\text{C}$). Saturated vapor with a flow rate F_s (kg/s) at temperature T_s ($^{\circ}\text{C}$) is used as a hot utility. The condensate is assumed saturated liquid with the same flow rate and temperature.

The heat exchanger has a total heat transfer area A (m^2) through which heat Q (kW) is transferred using a driving force ΔT_m ($^{\circ}\text{C}$).

The total heat transfer area A (m^2) is distributed in N (-) tubes of diameter d (m) and length L (m), arranged in n (-) passes into a shell of diameter D (m).

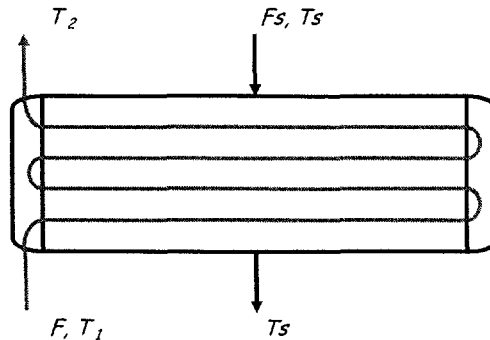


Figure 4.10 Flowsheet of a typical shell and tubes steam heater.

2. Process Model

A mathematical model of the process presented in Figure 4.10 is summarized in Table 4.12. The first four Equations (E01-E04) are similar to those of the simplified model in Table 4.8. The remainder equations essentially calculate the overall heat transfer coefficient versus the geometrical characteristics of the exchanger and the flow conditions. Thus, Equation (E05) calculates the overall (U) versus the surface heat transfer coefficients, inside (h_i) and outside (h_o) of the tubes. The thermal resistance of the wall and fouling are ignored. Equation (E06) estimates the inside surface heat transfer coefficient (h_i). Different equations, depended on the type of flow, are usually used (e.g., Levenspiel, 1986). Equation (E07) calculates the surface heat transfer coefficient for water vapor condensation outside of vertical tubes (h_o). Equation (E08) calculates the fluid velocity inside the tubes (u). Equations (E09 and E10) come from geometry, (E09) calculates the total heat transfer area (A) versus the number (N) and size (d, L) of tubes, (E10) calculates the required shell diameter (D) in order to include the tubes in triangular arrangement with pitch 25% greater than the tube diameter (Sinnott, 1996). Equation (E11) calculates the friction pressure loss (Δp) inside the tube when the Fanning friction coefficient f is estimated from Equation (E12); different equations are used for different types of flow. Finally Equation (E13) estimates the required pump electrical work (W).

Twenty process variables are involved in the above model of thirteen equations, which means that five degrees of freedom are available, and if we consider the same problem specifications to those of Table 4.9 three design variables are available for process optimization. Table 4.13 suggests a selection of design variables. The corresponding solution algorithm is presented in Figure 4.11. A trial-and-error procedure is used to obtain an accurate value of the overall heat transfer coefficient. Finally, Figure 4.12 summarizes the classical procedure for design heat exchangers.

The cost analysis method presented in Table 4.10 is modified to that of Table 4.14 in order to include the required electricity.

Concerning the design variables the following points should be considered:

- The fluid velocity inside the tubes (u) is a crucial design variable. When high values of velocity are selected high heat transfer coefficients are obtained but significant pressure loss occurs. Thus lower equipment cost and higher operating cost are resulted.
- The inside tube diameter (d) is also a crucial design variable. When small values of diameter are selected high heat transfer coefficients are obtained but significant pressure loss also occurs. Thus lower equipment cost and higher operating cost are resulted.
- The number of passes (n) is selected so as to use tubes of accepted length.

Table 4.12 Detailed Model of a Shell and Tube Heat Exchanger

$$Q = FC_p(T_2 - T_1) \quad (\text{E01})$$

$$Q = F_s \Delta H_s \quad (\text{E02})$$

$$\Delta T_m = \frac{(T_s - T_1) - (T_s - T_2)}{\ln[(T_s - T_1)/(T_s - T_2)]} \quad (\text{E03})$$

$$Q = AU\Delta T_m \quad (\text{E04})$$

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \quad (\text{E05})$$

$$\left. \begin{aligned} \left(\frac{h_i d}{\lambda}\right) &= 1.86 \left(\frac{\rho d u C_p \eta d}{\eta \lambda L}\right)^{1/3} \left(\frac{\eta}{\eta_w}\right)^{0.14}, \text{ for laminar flow} \\ \left(\frac{h_i d}{\lambda}\right) &= 0.023 \left(\frac{\rho d u}{\eta}\right)^{0.8} \left(\frac{C_p \eta}{\lambda}\right)^{1/3} \left(\frac{\eta}{\eta_w}\right)^{0.14}, \text{ for turbulent flow} \end{aligned} \right\} \quad (\text{E06})$$

$$h_o = 2750 \left(\frac{Nd}{F_s}\right)^{1/3} \quad (\text{E07})$$

$$u = \frac{F}{\rho \frac{N}{n} \left(\frac{\pi d^2}{4}\right)} \quad (\text{E08})$$

$$A = N\pi dL \quad (\text{E09})$$

$$N = 0.319 \left(\frac{D}{d}\right)^{2.142} \quad (\text{E10})$$

$$\Delta p = n \left[4f \left(\frac{L}{d}\right) + 2.5 \right] \frac{\rho u^2}{2} \quad (\text{E11})$$

$$\left. \begin{aligned} f &= 16 \left(\frac{\rho d u}{\eta}\right)^{-1}, \text{ for laminar flow} \\ f &= 0.08 \left(\frac{\rho d u}{\eta}\right)^{-1/4}, \text{ for turbulent flow} \end{aligned} \right\} \quad (\text{E12})$$

$$E = F \Delta p \quad (\text{E13})$$

Table 4.12 Continued**Process Variables**

| | | | |
|--------------|--------------------|---|------|
| F | kg/s | Feed flow rate | (1) |
| F_s | kg/s | Steam flow rate | (2) |
| T_1 | °C | Feed temperature | (3) |
| T_2 | °C | Target temperature | (4) |
| T_s | °C | Steam temperature | (5) |
| ΔT_m | °C | Mean temperature difference | (6) |
| Q | kW | Thermal load | (7) |
| A | m ² | Heat transfer area | (8) |
| E | kW | Electric power | (9) |
| U | W/m ² K | Overall heat transfer coefficient | (10) |
| h_i | W/m ² K | Surface heat transfer coefficient inside tubes | (11) |
| h_o | W/m ² K | Surface heat transfer coefficient outside tubes | (12) |
| f | - | Fanning friction coefficient | (13) |
| u | m/s | Fluid velocity in tubes | (14) |
| Δp | Pa | Friction pressure loss in tubes | (15) |
| d | m | Tube diameter | (16) |
| L | m | Tube length | (17) |
| N | - | Total number of tubes | (18) |
| n | - | Number of passes in tubes | (19) |
| D | m | Shell diameter | (20) |

Technical Data

| | | | |
|--------------|-------------------|-----------------------------------|-----|
| ρ | kg/m ³ | Fluid density | (1) |
| C_p | kJ/kg C | Fluid specific heat | (2) |
| λ | W/mK | Fluid thermal conductivity | (3) |
| η | Pa s | Fluid apparent viscosity | (4) |
| ΔH_s | kJ/kg | Latent heat of steam condensation | (5) |

Table 4.13 Solution of a Typical Design Problem**Process Specifications**

| | | | |
|-------|------|--------------------|-----|
| F | kg/s | Feed flow rate | (1) |
| T_1 | °C | Feed temperature | (2) |
| T_2 | °C | Target temperature | (3) |
| T_s | °C | Steam temperature | (4) |

Degrees-of-Freedom Analysis

| | |
|--------------------------|-----------|
| Process variables | 20 |
| <u>Process equations</u> | <u>13</u> |
| Degree of freedom | 7 |
| <u>Specifications</u> | <u>4</u> |
| Design variables | 3 |

Design Variables

| | | |
|-----|-----|---------------------------|
| u | m/s | Fluid velocity in tubes |
| d | m | Tube diameter |
| n | - | Number of passes in tubes |

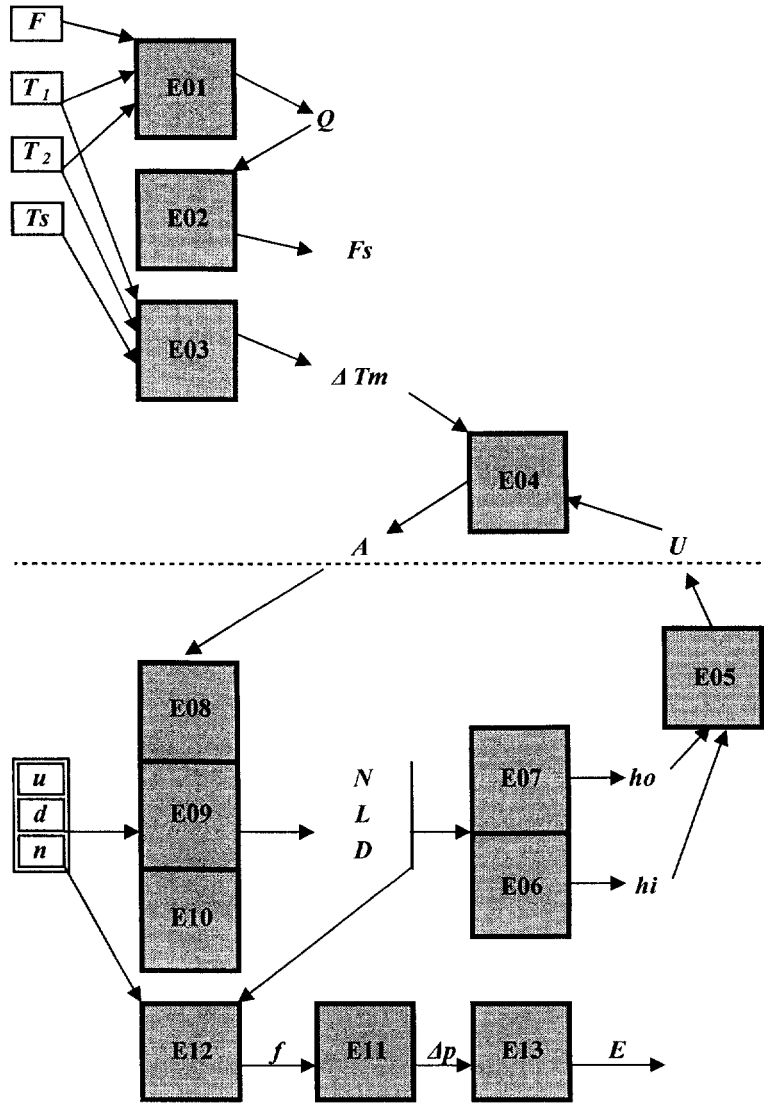


Figure 4.11 Solution algorithm of the model of Table 4.12.

| Heat Exchanger Design Procedure | |
|--|---|
| 1) | Define process specifications and calculate the thermal load |
| 2) | Estimate the thermophysical properties of liquids |
| 3) | Select the type of heat exchanger (e.g. Shell and tube) |
| 4) | Select an initial value for the overall heat transfer coefficient |
| 5) | Calculate the driving force (Logarithmic mean temperature difference) |
| 6) | Calculate the heat transfer area |
| 7) | Select geometry (e.g. Tube diameter, number of tube passes) |
| 8) | Estimate surface heat transfer coefficients |
| 9) | Estimate the corresponding overall heat transfer coefficient |
| 10) | If the new value is different from the initial go to step 4 |
| 11) | Calculate the friction pressure loss |
| 12) | If the calculated pressure loss is not acceptable go to step 7 or 3 |
| 13) | Optimize the heat exchanger go to step 7 for parametric optimization go to step 3 for structural optimization |

Figure 4.12 Generalized design procedure for heat exchangers.

Table 4.14 Cost Analysis**Equipment cost**

$$C_{eq} = C_{exc} A^{n_{exc}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_s Q + C_e E) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (F04)$$

Cost Data

| | | |
|-----------|-------------------|-------------------------------|
| C_{exc} | \$/m ² | Heat exchange unit cost |
| n_{exc} | - | Heat exchanger scaling factor |
| C_s | \$/kWh | Cost of heating steam |
| C_e | \$/kWh | Cost of electricity |
| t_y | hr/yr | Annual operating time |
| i_r | - | Interest rate |
| l_f | yr | Lifetime |

3. Application to Tomato Paste Heating

The previous design procedure is applied to the tomato paste heating process presented in Section IV.3, and the results are summarized in Table 4.15. The required heat transfer area, calculated from the detailed procedure, is estimated to 9 m² which is the same as in the simplified procedure, but now we are sure that the results are of much better accuracy. Besides, now we have more information concerning the geometric characteristics of the heat exchanger. Moreover, we now know the electricity requirements.

The effect of the design fluid velocity on the total annualized cost is presented in Figure 4.13. The optimum value is a trade off between the equipment and operating cost. The effect of other design variables on the total annualized cost TAC is presented in Figure 4.15 for the tube diameter d and in Figure 4.16 for the tube side number of passes n . Figure 4.14 shows the cost breakdown at the optimum point.

The effect of fluid velocity inside the tubes is further analyzed in Figures 4.17 and 4.18: When the design velocity is increased, the heat transfer coefficient increases (with decreasing rate), but the pressure loss is exponentially increased. Thus the optimum value revealed in Figure 4.13 is justified.

Table 4.15 Process Design Results**Process Specifications**

| | | |
|-------|----------|--------------------|
| F | = 1 kg/s | Feed flow rate |
| T_1 | = 50 °C | Feed temperature |
| T_2 | = 100 °C | Target temperature |
| T_s | = 120 °C | Steam temperature |

Technical Data

| | | |
|--------------|--------------------------|-----------------------------------|
| ρ | = 1130 kg/m ³ | Fluid density |
| λ | = 0.55 W/mK | Fluid thermal conductivity |
| η | = 0.27 Pa s | Fluid apparent viscosity |
| C_p | = 3.50 kJ/kg C | Specific heat |
| ΔH_s | = 2200 kJ/kg | Latent heat of steam condensation |

Design variables

| | | |
|-----|----------|---------------------------|
| u | = 1 m/s | Fluid velocity in tubes |
| d | = 0.01 m | Tube diameter |
| n | = 4 | Number of passes in tubes |

Model Solution

| | | |
|--------------|----------------------------|---|
| Q | = 175 kW | Thermal load |
| F_s | = 0.08 kg/s | Steam flow rate |
| ΔT_m | = 40 °C | Mean temperature difference |
| A | = 9 m ² | Heat transfer area |
| E | = 2.1 kW | Electric power |
| U | = 0.46 kW/m ² K | Overall heat transfer coefficient |
| h_i | = 0.51 kW/m ² K | Surface heat transfer coefficient inside the tubes |
| h_o | = 4.9 kW/m ² K | Surface heat transfer coefficient outside the tubes |
| f | = 1.27 | Fanning friction coefficient |
| Δp | = 23 bar | Friction pressure loss in tubes |
| L | = 6.70 m | Tube length |
| N | = 4 x 11 | Total number of tubes |
| D | = 0.20 m | Shell diameter |

Cost Data

| | | |
|-----------|---------------------------|-----------------------|
| C_s | = 0.04 \$/kWh | Cost of heating steam |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_{exc} | = 3.00 k\$/m ² | Heat exchanger |
| n_{exc} | = 0.65 | Heat exchanger |
| t_y | = 1400 hr/yr | Annual operating time |
| i_r | = 0.09 | Interest rate |
| l_f | = 5 yr | Lifetime |

Cost Estimation Results

| | | |
|----------|--------------|-------------------------|
| e | = 0.257 | Capital recovery factor |
| C_{eq} | = 12.9 k\$ | Equipment cost |
| C_{op} | = 8.9 k\$/yr | Operating cost |
| TAC | = 12.2 k/yr | Total annualized cost |

* Information concerning the detailed model of Table 4.15, not required for the simplified model of Table 4.8.

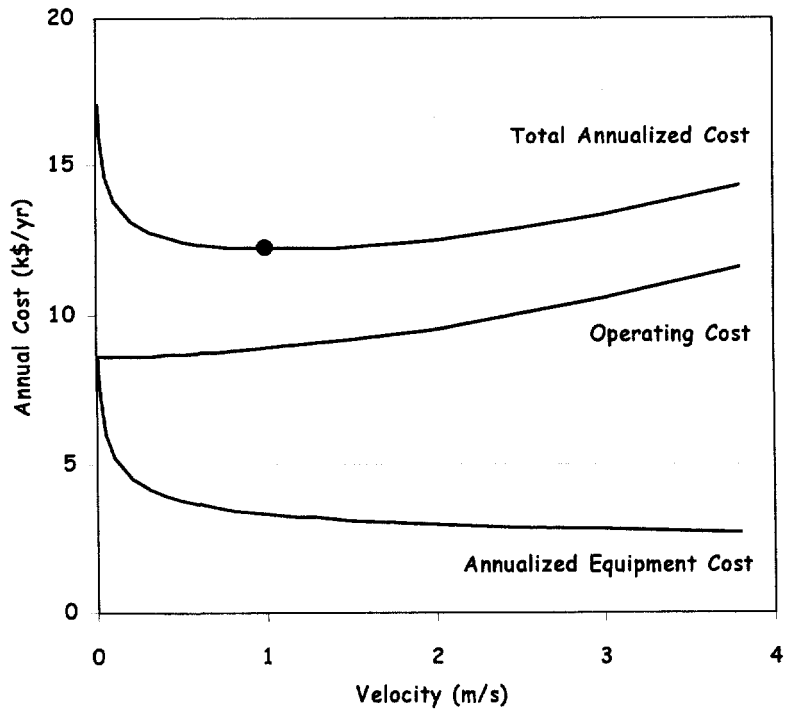


Figure 4.13 Effect of fluid velocity (design variable) on equipment, operating and total annualized cost (objective function). The bold point denotes the optimum.

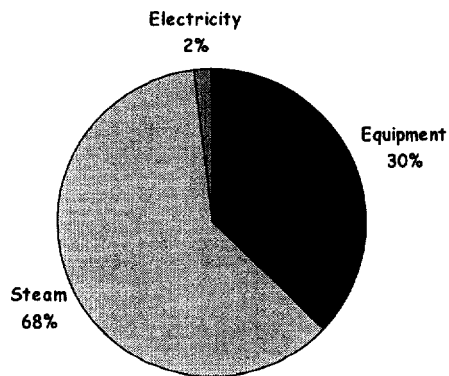


Figure 4.14 Cost breakdown at optimum.

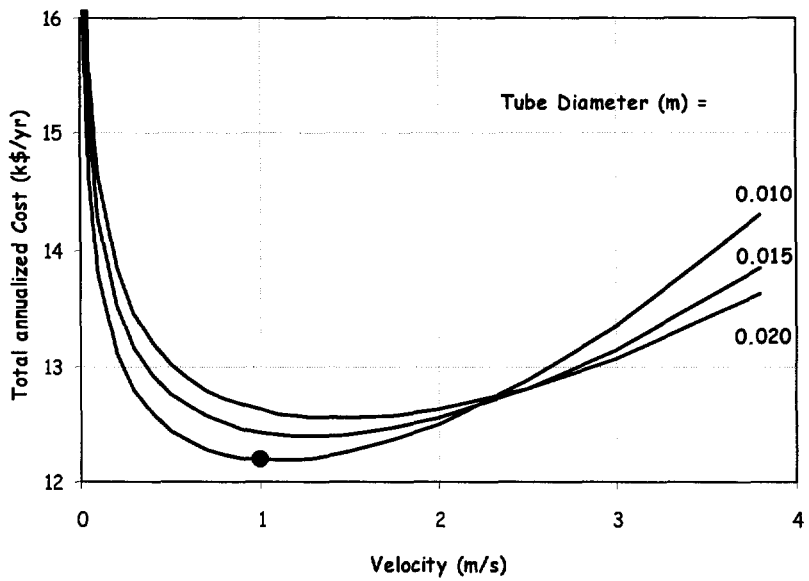


Figure 4.15 Effect of velocity and tube diameter on total annualized cost.

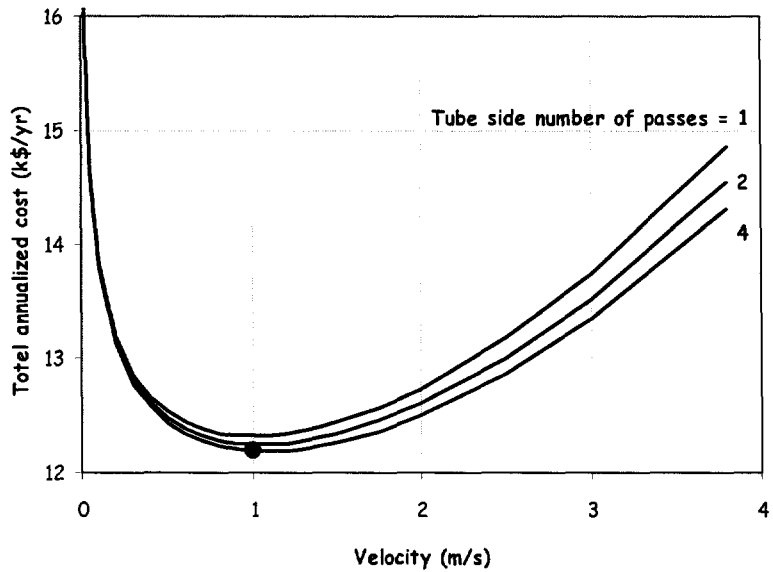


Figure 4.16 Effect of velocity and tube passes on total annualized cost.

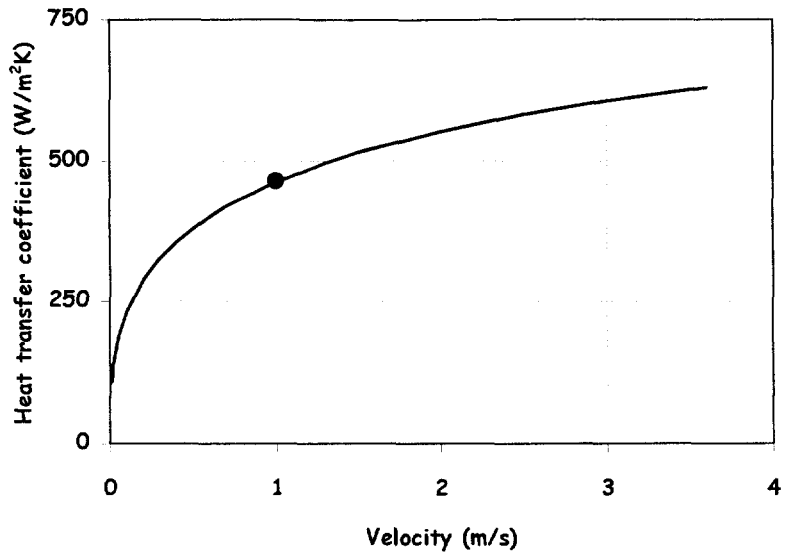


Figure 4.17 Effect of fluid velocity on the heat transfer coefficient.

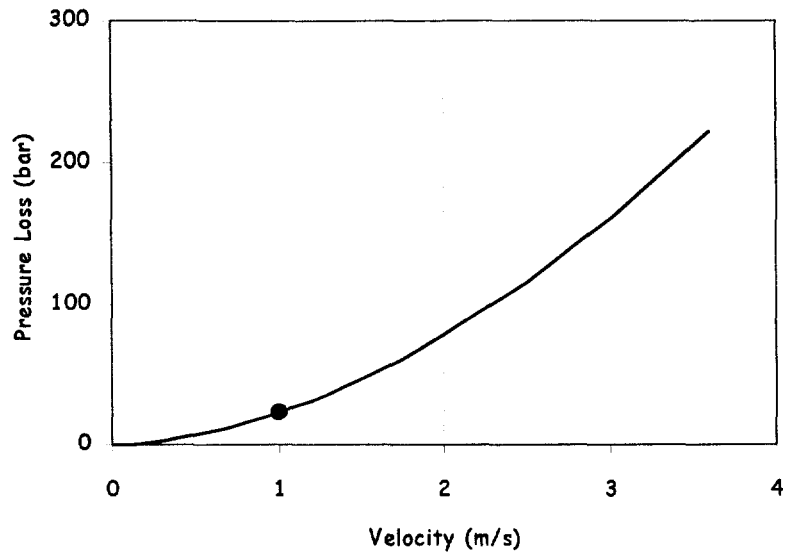


Figure 4.18 Effect of fluid velocity on friction pressure loss.

VI. DETAILED DESIGN OF A PLATE HEAT EXCHANGER

In this section, a detailed design approach is described for a plate heat exchanger. Equipment geometry is taken into account and consequently a detailed estimation of the heat transfer coefficient and the corresponding pressure loss can be obtained. Similar design procedures can also be developed for the other types of heat exchangers.

1. Process Description

A typical flowsheet of a plate heat exchanger is presented in Figure 4.19. The process stream at a flow rate F (kg/s) and a temperature T_1 ($^{\circ}\text{C}$) is heated up to a temperature T_2 ($^{\circ}\text{C}$). Hot water at a flow rate F_w (kg/s) at temperature T_{w1} ($^{\circ}\text{C}$) is used as heating medium, which exits at temperature T_{w2} ($^{\circ}\text{C}$).

The heat exchanger consists of N plates distributed in n groups in series. Each plate has width W (m) and length L (m). The plate spacing is b (m). The total heat transfer area is A (m^2) through which heat Q (kW) is transferred using a driving force ΔT_m ($^{\circ}\text{C}$).

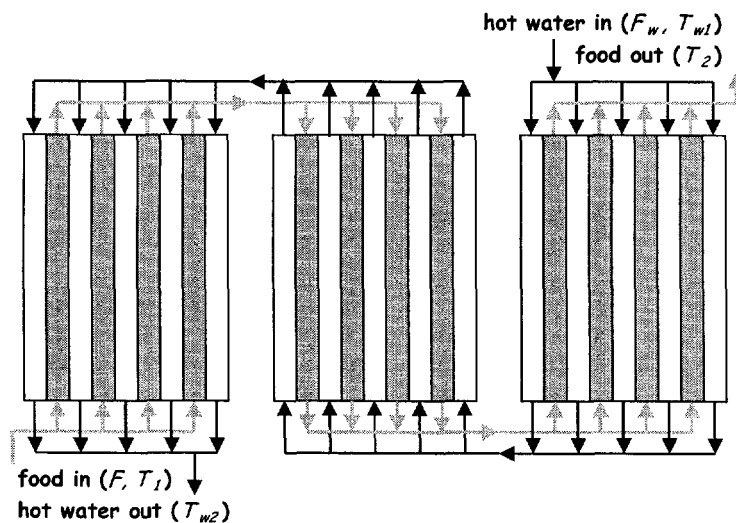


Figure 4.19 Flowsheet of a typical plate heat exchanger.

2. Process Model

A mathematical model of the process presented in Figure 4.19 is summarized in Table 4.16. The first four Equations (E01-E04) are similar to those of the simplified model in Table 4.8. The remainder equations essentially calculate the overall heat transfer coefficient versus the geometrical characteristics of the exchanger and the flow conditions. Equations (E05) calculates the total heat transfer area (A) versus the total number of plates (N) and size of each plate (W , L). Equations (E06) and (E07) calculate the fluid velocity of the process stream (u) and the utility stream (u_w), respectively. Equation (E10) calculates the overall heat transfer coefficient (U) versus the surface heat transfer coefficients of the process stream side (h_i) and the utility side (h_o). The same equation is used for both sides (Equations E08 and E09). The thermal resistance of the thin wall and fouling are ignored. Equations (E11) and (E12) calculate the friction pressure loss for the process stream side (Δp) and the utility stream side (Δp_w), respectively, when the corresponding Fanning friction coefficients f and f_w are estimated from Equations (E13) and (E14), which are valid for turbulent flow. Finally, Equations (E15) and (E16) estimate the required pump electrical work (E and E_w).

Twenty five process variables are involved in the above model of sixteen equations, which means that nine degrees of freedom are available, and if we consider five problem specifications, four design variables are available for process optimization. Table 4.17 suggests a selection of design variables. The corresponding solution algorithm is presented in Figure 4.20. A trial-and-error procedure is used to obtain an accurate value of the overall heat transfer coefficient.

The cost estimation presented in Table 4.14 can also be applied for the case of plate heat exchanger.

Concerning the design variables the following points should be considered:

- The fluid velocity of the process stream (which usually is the most viscous fluid) (u) is a crucial design variable. When high values of velocity are selected high heat transfer coefficients are obtained but significant pressure loss occurs. Thus lower equipment cost and higher operating cost result.
- The plate spacing (b) is also a crucial design variable. When small values are selected high heat transfer coefficients are obtained but significant pressure loss also occurs. Thus lower equipment cost and higher operating cost result.
- The number of groups in series (n) is selected so as to use plates of acceptable length.

Table 4.16 Detailed Model of a Plate Heat Exchanger

$$Q = FC_p (T_2 - T_1) \quad (\text{E01})$$

$$Q = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E02})$$

$$\Delta T_m = \frac{(T_{w2} - T_1) - (T_{w1} - T_2)}{\ln[(T_{w2} - T_1)/(T_{w1} - T_2)]} \quad (\text{E03})$$

$$Q = AU\Delta T_m \quad (\text{E04})$$

$$A = NWL \quad (\text{E05})$$

$$u = \frac{F}{\rho \frac{N}{n} Wb} \quad (\text{E06})$$

$$u_w = \frac{F_w}{\rho_w \frac{N}{n} Wb} \quad (\text{E07})$$

$$\left(\frac{h_i 2b}{\lambda}\right) = 0.023 \left(\frac{\rho 2bu}{\eta}\right)^{0.8} \left(\frac{C_p \eta}{\lambda}\right)^{1/3} \quad (\text{E08})$$

$$\left(\frac{h_o 2b}{\lambda_w}\right) = 0.023 \left(\frac{\rho_w 2bu_w}{\eta_w}\right)^{0.8} \left(\frac{C_{pw} \eta_w}{\lambda_w}\right)^{1/3} \quad (\text{E09})$$

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \quad (\text{E10})$$

$$\Delta p = n \left[4f \left(\frac{L}{2b}\right) + 2.5 \right] \frac{\rho u^2}{2} \quad (\text{E11})$$

$$\Delta p_w = n \left[4f_w \left(\frac{L}{2b}\right) + 2.5 \right] \frac{\rho_w u_w^2}{2} \quad (\text{E12})$$

$$f = 0.08 \left(\frac{\rho 2bu}{\eta}\right)^{-0.25} \quad (\text{E13})$$

$$f_w = 0.08 \left(\frac{\rho_w 2bu_w}{\eta_w}\right)^{-0.25} \quad (\text{E14})$$

$$E = F \Delta p \quad (\text{E15})$$

$$E_w = F_w \Delta p_w \quad (\text{E16})$$

Note: The hydraulic diameter is used in dimensionless groups, that is, $2b$.

Table 4.16 Continued**Process Variables**

| | | | |
|--------------|--------------------|--|------|
| F | kg/s | Feed flow rate (process stream) | (1) |
| F_w | kg/s | Heating water flow rate (utility stream) | (2) |
| T_1 | °C | Feed temperature | (3) |
| T_2 | °C | Target temperature | (4) |
| T_{w1} | °C | Heating water inlet temperature | (5) |
| T_{w2} | °C | Heating water outlet temperature | (6) |
| ΔT_m | °C | Mean temperature difference | (7) |
| Q | kW | Thermal load | (8) |
| A | m ² | Heat transfer area | (9) |
| E | kW | Electric power for process side pump | (10) |
| E_w | kW | Electric power for utility side pump | (11) |
| U | W/m ² K | Overall heat transfer coefficient | (12) |
| h_i | W/m ² K | Surface heat transfer coefficient in process stream side | (13) |
| h_o | W/m ² K | Surface heat transfer coefficient in utility stream side | (14) |
| f | - | Fanning friction coefficient in process stream side | (15) |
| f_w | - | Fanning friction coefficient in utility stream side | (16) |
| u | m/s | Process stream fluid velocity in process stream side | (17) |
| u_w | m/s | Utility stream fluid velocity in utility stream side | (18) |
| Δp | Pa | Friction pressure loss in process stream side | (19) |
| Δp_w | Pa | Friction pressure loss in utility stream side | (20) |
| b | m | Plate spacing | (21) |
| L | m | Plate length | (22) |
| N | - | Total number of plates | (23) |
| n | - | Number of groups in series | (24) |
| W | m | Plate width | (25) |

Technical Data

| | | | |
|-----------|-------------------|------------------------------------|-----|
| ρ | kg/m ³ | Fluid density | (1) |
| C_p | kJ/kg K | Fluid specific heat | (2) |
| λ | W/m K | Fluid thermal conductivity | (3) |
| η | Pa s | Fluid apparent viscosity | (4) |
| ρ_w | g/m ³ | Heating water density | (5) |
| C_{pw} | kJ/kg K | Heating water specific heat | (6) |
| η_w | W/m K | Heating water thermal conductivity | (7) |
| η_w | Pa s | Heating water apparent viscosity | (8) |

Table 4.17 Solution of a Typical Design Problem

| | | | |
|------------------------------------|-----------|----------------------------------|-----|
| Process Specifications | | | |
| F | kg/s | Feed flow rate | (1) |
| T_1 | °C | Feed temperature | (2) |
| T_2 | °C | Target temperature | (3) |
| T_{w1} | °C | Heating water inlet temperature | (4) |
| T_{w2} | °C | Heating water outlet temperature | (5) |
| Degrees-of-Freedom Analysis | | | |
| Process variables | 25 | | |
| <u>Process equations</u> | <u>16</u> | | |
| Degree of freedom | 9 | | |
| <u>Specifications</u> | <u>5</u> | | |
| Design variables | 4 | | |
| Design Variables | | | |
| u | m/s | Fluid velocity | |
| b | m | Plate spacing | |
| W | m | Plate width | |
| n | - | Number of groups in series | |

3. Application to Orange Juice Heating

The previous design procedure is applied to an orange juice heater/pasteurizer. During orange juice pasteurization, the juice is preheated in the regenerator from 20 to about 65°C and subsequently is heated in the heater/pasteurizer from 65 to 90°C. The pasteurizer is heated with hot water, which is heated by steam injection at 110°C, and exits at 90°C. The results are summarized in Table 4.18.

The effect of the design fluid velocity on the total annualized cost is presented in Figure 4.21. The optimum value is a trade off between the equipment and operating cost. The effect of other design variables on the total annualized cost TAC is presented in Figure 4.23 for the plate spacing b and in Figure 4.24 for the number of groups in parallel n . Figure 4.22 shows the cost breakdown at the optimum point.

The effect of fluid velocity in process stream side is further analyzed in Figures 4.25 and 4.26: When the design velocity is increased, the heat transfer coefficient increases (with decreasing rate), but the pressure loss is exponentially increased. Thus the optimum value revealed in Figure 4.21 is justified.

Note: Equation (4-16) predicts high heat transfer coefficients (no fouling and no wall resistance) similar to the values calculated from Equation (E08). In practice, fouling may be taken into consideration, using a smaller (e.g., by 10 %) overall heat transfer coefficient.

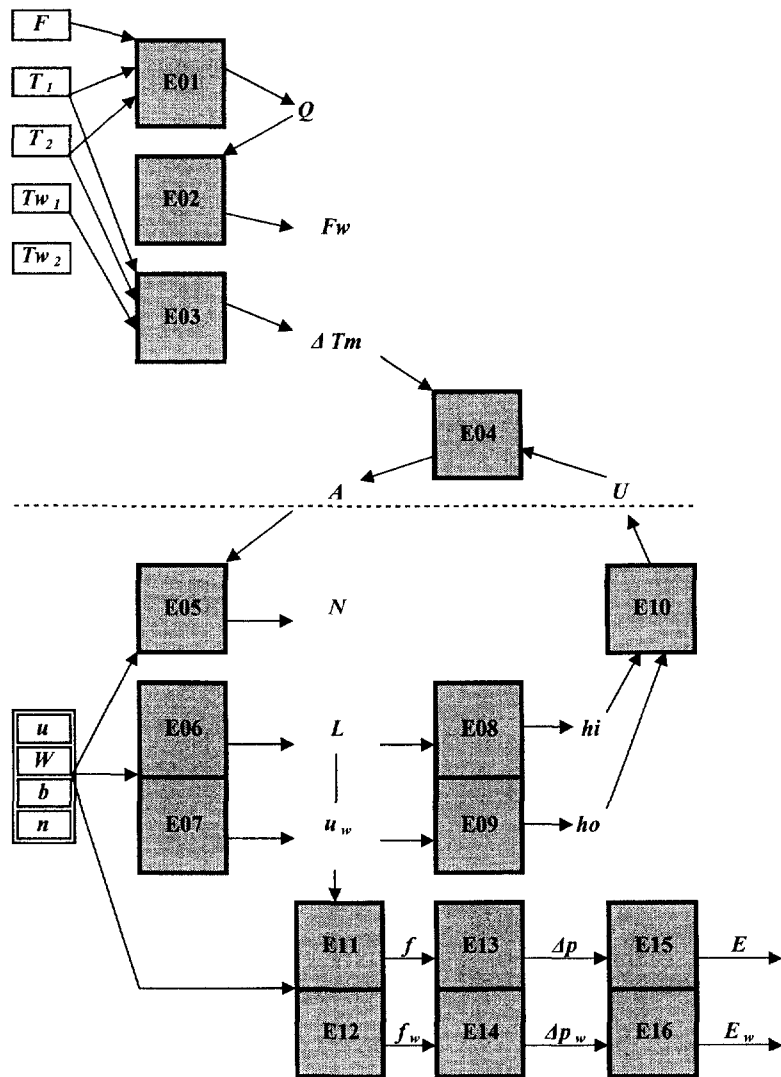


Figure 4.20 Solution algorithm of the model of Table 4.16.

Table 4.18 Process Design Results

Process Specifications

| | | |
|----------|----------|----------------------------------|
| F | = 5 kg/s | Feed flow rate |
| T_1 | = 65 °C | Feed temperature |
| T_2 | = 90 °C | Target temperature |
| T_{w1} | = 110 °C | Heating water inlet temperature |
| T_{w2} | = 95 °C | Heating water outlet temperature |

Table 4.18 Continued

| Technical Data | | |
|--------------------------------|----------------------------|--|
| ρ | = 1000 kg/m ³ | Fluid density |
| λ | = 0.65 W/mK | Fluid thermal conductivity |
| η | = 0.40 mPa s | Fluid apparent viscosity |
| C_p | = 3.86 kJ/kg K | Fluid specific heat |
| ρ_w | = 960 kg/m ³ | Heating water density |
| λ_w | = 0.65 W/m K | Heating water thermal conductivity |
| η_w | = 0.25 mPa s | Heating water apparent viscosity |
| C_{pw} | = 4.18 kJ/kg K | Heating water specific heat |
| Design variables | | |
| u | = 1 m/s | Fluid velocity of process stream |
| b | = 5 mm | Plate spacing |
| n | = 5 | Number of groups in series |
| Model Solution | | |
| Q | = 483 kW | Thermal load |
| F_w | = 7.7 kg/s | Heating water flow rate |
| ΔT_m | = 25 °C | Mean temperature difference |
| A | = 4.6 m ² | Heat transfer area |
| u_w | = 1.60 m/s | Velocity of utility stream |
| U | = 4.23 kW/m ² K | Overall heat transfer coefficient |
| h_i | = 6.6 W/m ² K | Surface heat transfer coefficient in process stream side |
| h_o | = 11.9 W/m ² K | Surface heat transfer coefficient in utility stream side |
| f | = 0.006 | Fanning friction coefficient in process stream side |
| f_w | = 0.005 | Fanning friction coefficient in utility stream side |
| Δp | = 0.1 bar | Friction pressure loss in process stream side |
| Δp_w | = 0.3 bar | Friction pressure loss in utility stream side |
| E | = 0.1 kW | Electric power of process stream pump |
| E_w | = 0.2 kW | Electric power of utility stream pump |
| L | = 1.15 m | Tube length |
| N | = 2x5 | Total number of plates |
| W | = 0.50 m | Plate width |
| Cost Data | | |
| C_s | = 0.05 \$/kWh | Cost of heating water |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_{exc} | = 4.00 k\$/m ² | Plate heat exchanger unit cost |
| n_{exc} | = 0.75 | Plate heat exchanger scaling factor |
| t_y | = 1400 hr/yr | Annual operating time |
| i_r | = 0.09 | Interest rate |
| l_f | = 5 yr | Lifetime |
| Cost Estimation Results | | |
| e | = 0.257 | Capital recovery factor |
| C_{eq} | = 12.6 k\$ | Equipment cost |
| C_{op} | = 33.8 k\$/yr | Operating cost |
| TAC | = 37.1 k/yr | Total annualized cost |

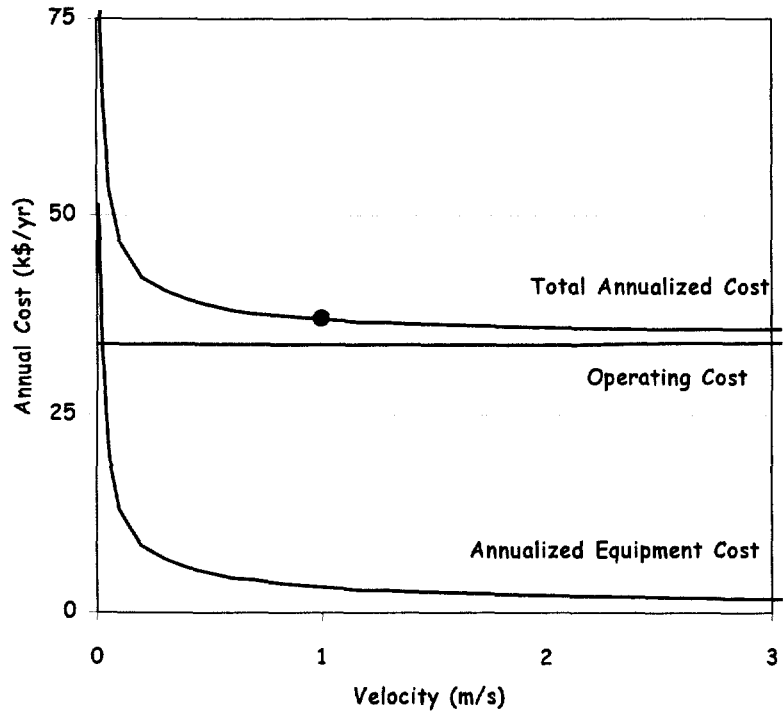


Figure 4.21 Effect of fluid velocity (design variable) on equipment, operating and total annualized cost (objective function). The bold point denotes the optimum.

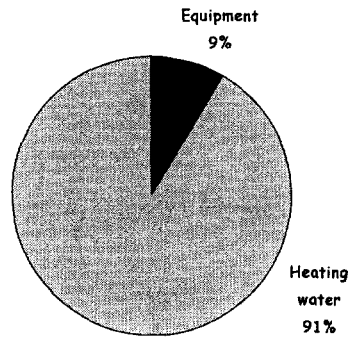


Figure 4.22 Cost breakdown at optimum.

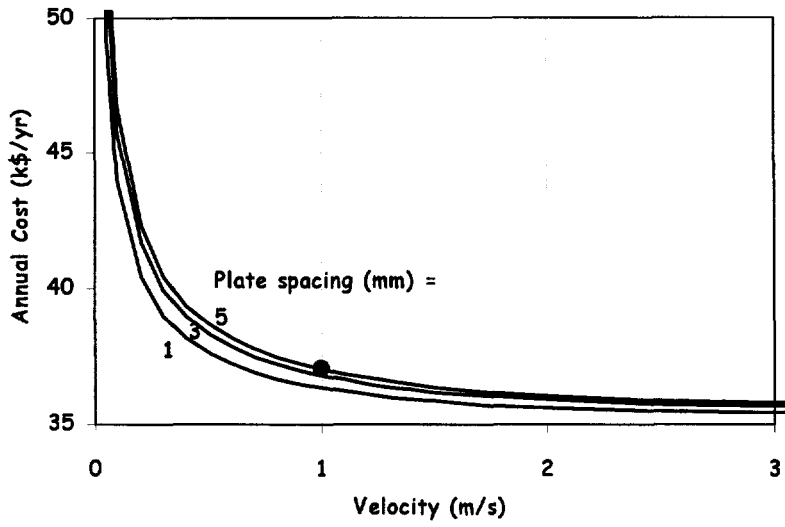


Figure 4.23 Effect of velocity and plate spacing on total annualized cost.

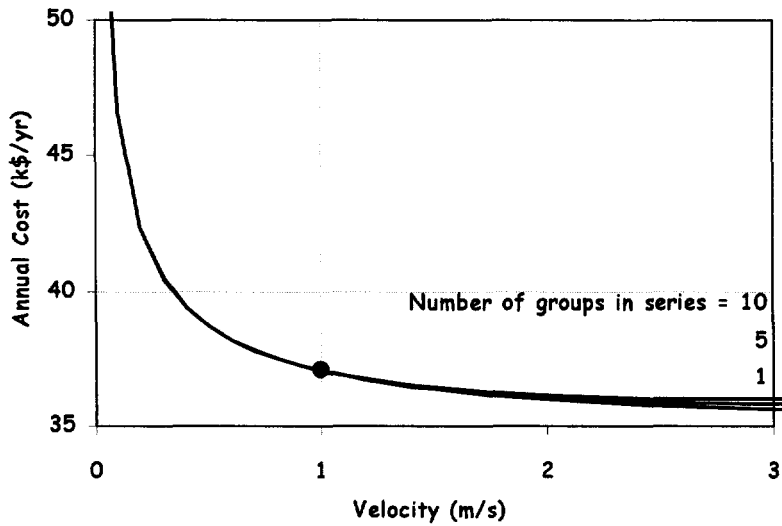


Figure 4.24 Effect of velocity and number of groups in series on total annualized cost.

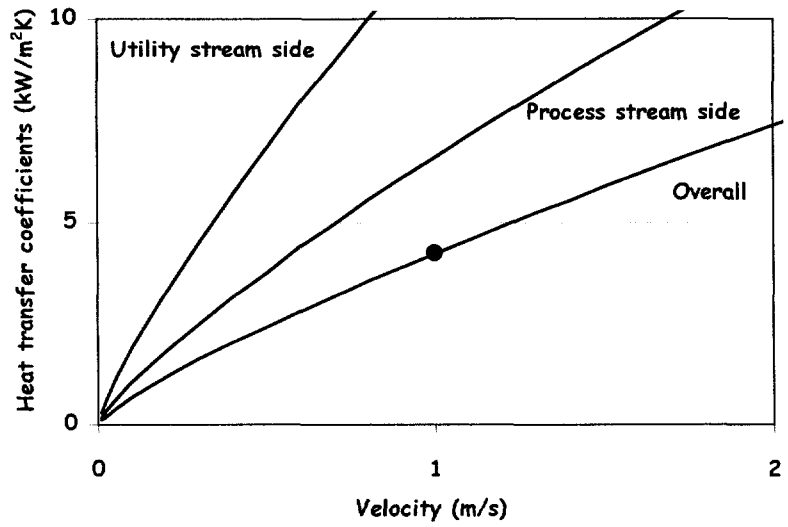


Figure 4.25 Effect of fluid velocity on heat transfer coefficient.

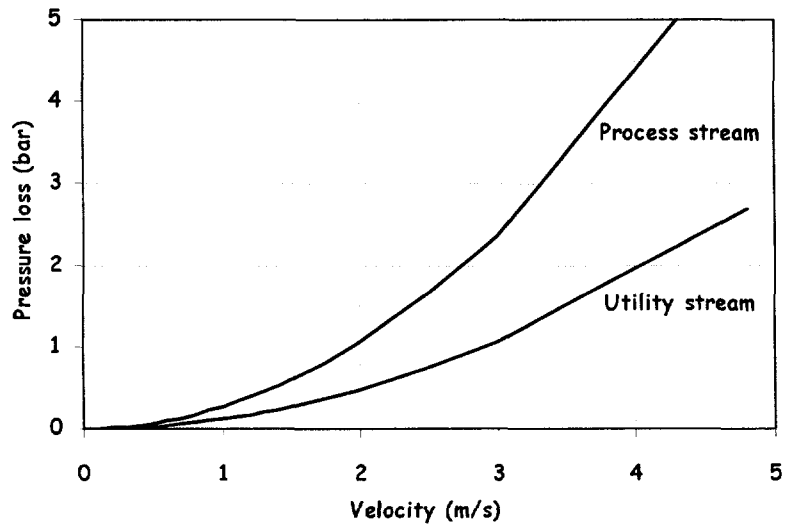


Figure 4.26 Effect of fluid velocity on friction pressure loss.

VII. HYGIENIC AND QUALITY CONSIDERATIONS

Heating processes and equipment, used in food processing, should conform to the general hygienic requirements, outlined in Chapter 2, and the food safety and quality requirements, discussed in Chapter 1.

Fouling of the heating surfaces of food processing equipment with food materials has undesirable effects on the heat transfer rate and the quality of the food product. Process and equipment design should minimize fouling and provide easy access to the heating surfaces for cleaning and sanitization. Food heat exchangers should be designed for continuous cleaning in place (CIP) operation

High temperatures of the heating medium should be avoided, since high wall temperature may cause severe fouling. High wall temperature may cause denaturation and fouling of heat-sensitive proteins. Thus, hot water is preferred to steam heating in milk pasteurization. Relatively small temperature difference (ΔT) should be preferred for both thermodynamic and food quality reasons.

Ovens and roasters operate at high temperatures and low humidities, conditions that destroy the vegetative cells of contaminating bacteria. However, they should be kept clean for good food quality.

Food heating equipment should be designed and operated to deliver the exact heat treatment (time-temperature) to the product, using adequate process instrumentation and control.

The eating quality of foods is very important in some heating processes, as cooking and baking.

NOMENCLATURE

| | |
|-----------|---|
| A | Heat transfer area |
| b | Plate spacing |
| C_e | Cost of electricity |
| C_{eq} | Equipment cost |
| C_{exc} | Unit cost of heat exchanger |
| C_{op} | Operating cost |
| C_p | Fluid specific heat |
| C_{pw} | Heating water specific heat |
| C_s | Cost of steam |
| D | Diffusivity |
| D | Shell diameter |
| d | Tube diameter |
| E | Electric field strength |
| E | Electric power for process side pump |
| e | Emissivity |
| e | Capital recovery factor |
| E_w | Electric power for utility side pump |
| f | Fanning friction coefficient in process stream side |
| F | Feed flow rate (process stream) |
| $F_{1,2}$ | View factor |
| F_s | Steam flow rate |
| f_w | Fanning friction coefficient in utility stream side |

| | |
|--------------|--|
| F_w | Heating water flow rate |
| h | Film heat transfer coefficient |
| h_c | Convection heat transfer coefficient |
| h_i | Surface heat transfer coefficient in process stream side |
| h_i | Surface heat transfer coefficient inside tubes |
| h_o | Surface heat transfer coefficient in utility stream side |
| h_o | Surface heat transfer coefficient outside tubes |
| h_r | Radiation heat transfer coefficient |
| i_r | Interest rate |
| j_H | Heat transfer factor |
| j_M | Mass transfer factor |
| k_c | Mass transfer coefficient |
| L | Length |
| l_f | Lifetime |
| m | Constant |
| n | Constant |
| n | Number of groups in series |
| n | Number of passes in tubes |
| N | Total number of plates |
| N | Total number of tubes |
| n_{exc} | Heat exchanger scaling factor |
| N_p | Number of plate channels in parallel |
| N_s | Number of plate passes in series |
| N_t | Total number of plates |
| Nu | Nusselt number |
| Pr | Prandtl number |
| q | Heat transfer rate |
| Q | Thermal load |
| Re | Reynolds number |
| Sc | Schmidt number |
| St | Stanton number |
| St_M | Stanton number for mass transfer |
| T | Temperature |
| t | Time |
| T_1 | Feed temperature |
| T_2 | Target temperature |
| $\tan\delta$ | Loss tangent |
| T_s | Steam temperature |
| T_{w1} | Heating water inlet temperature |
| T_{w2} | Heating water outlet temperature |
| t_y | Annual time of operation |
| u | Fluid velocity |
| u | Fluid velocity in tubes |
| U | Overall heat transfer coefficient |
| u | Process stream fluid velocity |
| u_w | Utility stream fluid velocity |
| W | Plate width |
| x | Wall thickness |

Greek

| | |
|-----------------|---|
| Δp | Friction pressure loss in process stream side |
| Δp_w | Friction pressure loss in utility stream side |
| ΔH_s | Latent heat of steam condensation |
| Δp | Tube inside friction pressure loss |
| ΔT_m | Mean temperature difference |
| ΔT | Temperature difference |
| ΔT_L | Logarithmic mean temperature difference |
| Φ | Overall radiation exchange factor |
| ε' | Dielectric constant |
| ε'' | Dielectric loss |
| η | Fluid apparent viscosity |
| η_w | Heating water apparent viscosity |
| λ | Fluid thermal conductivity |
| λ_o | Radiation wave length in the air |
| ρ | Fluid density |
| ρ_w | Heating water density |

Abbreviations

| | |
|-------------|--|
| <i>COST</i> | Cooperative Organization on Science and Technology |
| <i>FR</i> | Fouling resistance |
| <i>IIR</i> | Isobutylene-isoprene |
| <i>MW</i> | Micro Wave |
| <i>NBR</i> | Nitrile-butadiene |
| <i>PHE</i> | Plate Heat Exchanger |
| <i>RF</i> | Radio frequency |
| <i>SSHE</i> | Scraped Surface Heat Exchanger |
| <i>TAC</i> | Total Annualized Cost |
| <i>TEMA</i> | Tubular Exchanger Manufacturer Association |
| <i>UHT</i> | Ultra High Temperature |

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5

Refrigeration and Freezing

I. INTRODUCTION

Refrigeration is used for cooling/chilling or freezing of foods. Low temperatures are applied mainly for preservation (extension of storage life) of fresh or processed foods, by reducing the activity of microorganisms, enzymes, and chemical and biological reactions. Typical applications include preservation of fresh products, such as vegetables, through pre-cooling and chilling, cold storage of fruits, vegetables, meat, and fish, and freezing of meat and fish.

The basic methods for reducing temperature are: 1) mechanical compression, 2) ammonia absorption, 3) evaporation of cryogenic fluids, and 4) use of ice. Mechanical compression is the most important method, followed by evaporation of cryogenic fluids. Ice is used in fish preservation and in some smaller applications. Evaporation of free water under vacuum is sometimes applied to the cooling of leafy vegetables (Saravacos and Kostaropoulos, 2002).

Cooling of liquid and solid foods is accomplished using various processes and equipment, based on the principles and practices of heat transfer. Refrigerated foods are stored for several weeks at temperatures near 0 °C in insulated storage rooms or containers/trucks, sometimes in controlled atmosphere.

Freezing of foods is based on the rapid crystallization of free water into ice, and storage of the frozen foods for several months at temperatures near -18 °C. Frozen foods are usually packaged to prevent moisture loss and quality changes during storage. Thawing of frozen foods requires special attention to prevent undesirable changes in food quality.

Refrigeration is used extensively in the transportation, distribution, retailing, and home storage of perishable foods.

II. REFRIGERATION EQUIPMENT

1. Compression Refrigeration Cycles

The conventional mechanical refrigeration system (heat pump) consists basically of a compressor, a condenser, an expansion (throttling) valve and an evaporator. The system is integrated with the necessary piping, control and electrical components. Refrigeration is produced by compression/expansion of a suitable refrigerant, such as ammonia or fluorinated hydrocarbons (Freons).

Figure 5.1 shows the diagram of a typical mechanical refrigeration system. Figure 5.2 shows thermodynamic pressure-enthalpy diagram of the refrigeration system of Figure 5.1.

The processes of evaporation (4-1) and condensation (2-3) are isothermic (constant T), while the expansion (throttling) process (3-4) is isenthalpic (constant H). The compression process (1-2) is considered isentropic (constant S). The refrigeration load (heat removed, Q_e) is used to evaporate the liquid refrigerant in the evaporator. The vapors are compressed to a higher pressure, and they are condensed in the water- or air-cooled condenser.

The conventional refrigeration cycle of Figure 5.1 is used for temperatures down to $-20\text{ }^{\circ}\text{C}$. For lower temperatures a two-stage compression refrigeration cycle is used (Saravacos and Kostaropoulos, 2002). Typical temperatures for single-stage refrigeration cycle, evaporator $T_e = -10\text{ }^{\circ}\text{C}$ and condenser $T_c = 25\text{ }^{\circ}\text{C}$. For two-stage systems, $T_e = -40\text{ }^{\circ}\text{C}$ and $T_c = 25\text{ }^{\circ}\text{C}$.

2. Mechanical Compressors

There are 4 main types of mechanical compressors used in food refrigeration and freezing systems:

- Reciprocating compressors for small to medium capacity systems, 10-300 kW.
- Rotary compressors for very small to small capacities, 1-30 kW.
- Centrifugal compressors for large capacities, 200-700 kW.
- Screw compressors for medium to large capacities, 100-500 kW.

The volumetric capacity of compressors is expressed as displacement or swept volume, m^3/h of refrigerant. The volumetric capacity depends on the size of the compressor, the type of refrigerant, and the operating conditions (pressure-temperature), e.g., 10-10 000 m^3/h for ammonia.

The characteristics of mechanical compressors, evaporators and condensers, used in refrigeration systems, are described by Saravacos and Kostaropoulos (2002).

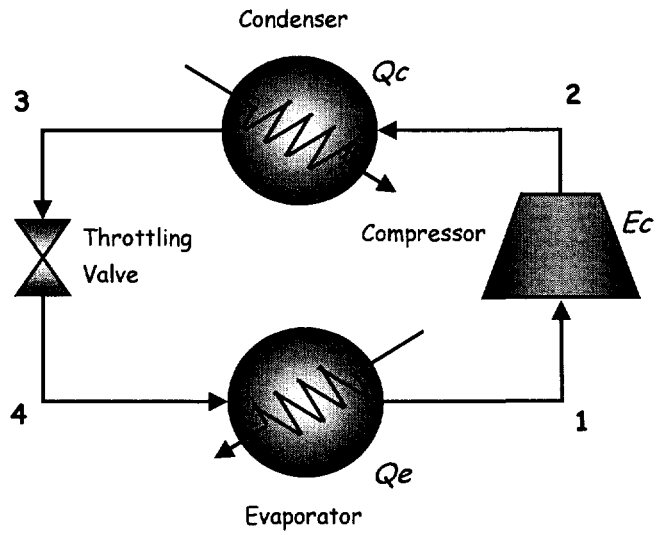


Figure 5.1 Compression refrigeration system.

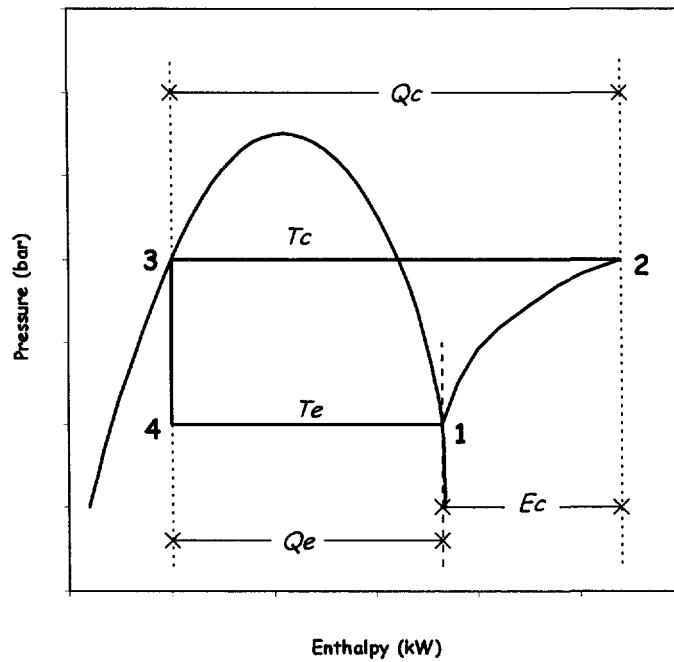


Figure 5.2 Pressure-enthalpy diagrams of single-stage refrigeration cycle.

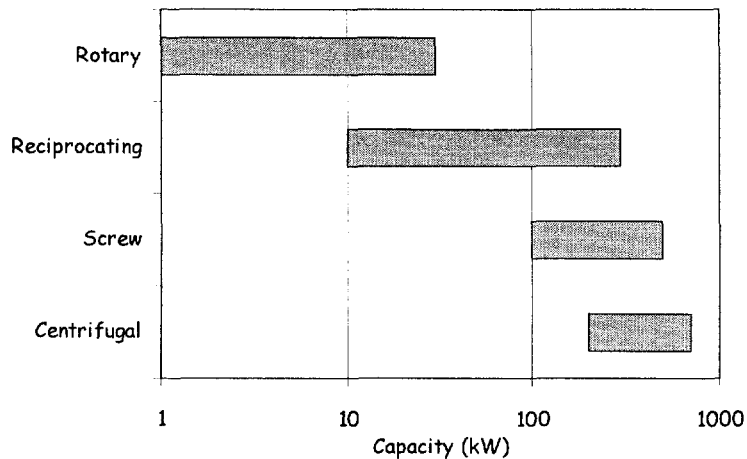


Figure 5.3 Capacity range of mechanical compressors.

3. Refrigeration Evaporators

Several types of evaporators are used in refrigeration systems, some of them similar to the heat exchangers discussed in Chapter 4:

- Tubular evaporators, bare or finned tubes
- Double-pipe evaporators
- Shell and tube evaporators, with the refrigerant in the shell side, and the cooled product in the tubes.
- Bath and tube evaporators, with the refrigerant (e.g., ammonia) in the tubes and the cooled product (e.g., brine for indirect refrigeration) in the bath.
- Double-wall evaporators
- Rotating coil evaporator
- Baudelot evaporators, consisting of horizontal tubes, containing the refrigerant, with a film of liquid product (e.g., fruit juice, wine) falling in the outside.

The overall heat transfer coefficients U in refrigeration evaporators depend on the refrigerant, the geometry of the heat exchanger, and the velocity of the cooled medium (air, water or brine). Typical values of U in the range of 0.2-1.5 kW/m²K.

4. Condensers

The following heat exchangers are used as condensers in refrigeration systems:

- Tubular, i.e., double pipe, shell and tube.
- Evaporative condensers with the refrigerant in horizontal tubes and water sprays in the outside and air blown to evaporate the water.
- Tower condensers, used in large capacity systems. They consist of a vertical shell and tube heat exchanger with water falling in the tubes and the refrigerant condensing in the outside. High overall heat transfer coefficients are obtained, e.g., 0.7-1.7 kW/ m²K.
- Air condensers, with air blown outside finned tubes and refrigerant condensing inside.

Overall heat transfers coefficients in refrigeration condensers in the range of 0.2-1.5 kW/ m²K.

5. Capacity Control

The capacity of refrigeration systems (kW) can be controlled by

- the refrigerant flow rate,
- the capacity of the compressor, and
- the capacity of the condenser (e.g., cooling water).

Control valves (thermostatic, pressure float, and capillary) are used to control the flow rate of the refrigerant. "On-off" operation of the compressor may be used to control the capacity of large refrigeration units.

6. Shortcut Design Procedure

A shortcut design procedure for vapor compression refrigeration cycles (heat pumps) was proposed by Shelton and Grossmann (1985). The procedure requires only the latent heat of vaporization and the specific heat of saturated liquid. The procedure avoids the complex and lengthy calculations that are required with equations of state, while yielding good approximations of the performance of vapor compression cycles employing pure refrigerants.

The proposed model consists of the three equations, which calculate the thermal load of the evaporator Q_e , the thermal load of the condenser Q_c , and the electrical power of the compressor E_c as follows:

$$Q_e = [\Delta H_r - C_{pr} (T_c - T_e)] F_r \quad (5-1)$$

$$Q_c = [\Delta H_r (T_c / T_e) - C_{pr} (T_c - T_e)] F_r \quad (5-2)$$

$$E_c = \Delta H_r [(T_c - T_e) / T_e] F_r \quad (5-3)$$

where

T_e = Refrigerant evaporating temperature (K)

T_c = Refrigerant condensing temperature (K)

F_r = Refrigerant flow rate (kg/s)

ΔH_r = Refrigerant latent heat of vaporization (kJ/kg)

C_{pr} = Refrigerant specific heat of saturated liquid (kJ/kg K)

A well known characteristic of a cycle is the coefficient of performance *COP* and is defined by the equation:

$$COP = \frac{Q_e}{E_c} \quad (5-4)$$

which is transformed to the following equation, when taking into account Equations (5-1), (5-2), and (5-3):

$$COP = \frac{T_e}{T_c - T_e} - \frac{C_{pr} T_e}{\Delta H_r} \quad (5-5)$$

The first term of the above expression is exactly the ideal Carnot coefficient of performance and is independent of the refrigerant being used. The second term contains the contribution due to the selected refrigerant.

It must be noted that a necessary condition for the selection of a refrigerant is the following:

$$T_p < T_e < T_c < T_{cp} \quad (5-6)$$

where

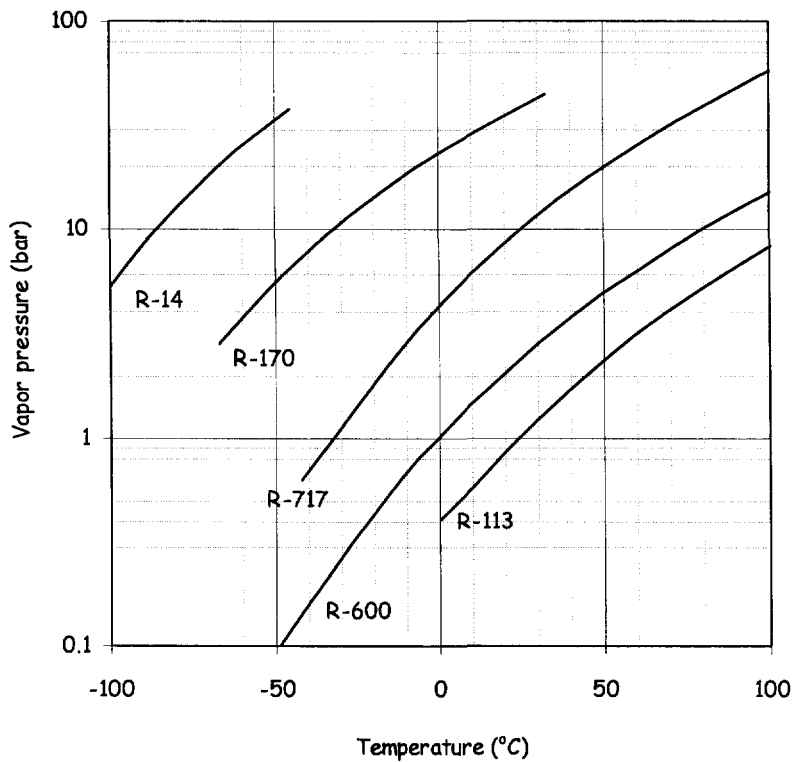
T_p = Refrigerant temperature at triple point (K)

T_{cp} = Refrigerant critical temperature (K)

Table 5.1 and Figure 5.4 summarize the required properties for some usual refrigerants. The fluorinated hydrocarbons (Freons), due to environmental concerns, will be replaced by other, non-polluting refrigerants.

Table 5.1 Useful Properties of Some Refrigerants

| Refrigerant | Composition | Latent heat of vaporization kJ/kg | Specific heat of liquid kJ/kgK | Triple point °C | Critical point °C |
|-------------|--------------------------|---|--------------------------------------|-----------------------|-------------------------|
| R-14 | Carbontetrafluoride | 136 | | -100 | -46 |
| R-170 | Ethane | 490 | 2.54 | -67 | 32 |
| R-717 | Ammonia | 1250 | 1.20 | -42 | 132 |
| R-600 | Butane | 386 | | -52 | 152 |
| R-113 | Trichlorotrifluoroethane | 180 | 0.62 | 1 | 198 |

**Figure 5.4** Vapor pressure versus temperature of some refrigerants.

III. COOLING OF FOODS

1. Introduction

Cooling or chilling of foods at temperatures close to 0 °C is applied to extend the shelf life of “fresh” foods immediately after harvesting or processing. The products are stored in insulated refrigerated storage rooms, trucks or ships.

The cooling load Q (kJ), i.e., the heat that is removed from the product upon cooling by ΔT (°C or K), is calculated from the equation:

$$Q = m C_p \Delta T \quad (5-7)$$

where m is the mass of the product (kg), and C_p is its specific heat (kJ/kg K).

The specific heat of a product can be estimated from the empirical “mixing rule” equation:

$$C_p = X_w C_{pw} + X_c C_{pc} + X_f C_{pf} + X_s C_{ps} \quad (5-8)$$

where X_w , X_c , X_f , X_s are the mass fractions, and C_{pw} , C_{pc} , C_{pf} , C_{ps} are the specific heats of the water, carbohydrate, fat, and salt components of the food product.

The enthalpy change of fruits and vegetables ΔH (kJ/kg) can be estimated from the empirical equation:

$$\Delta H = (1-X) \Delta H_L + X C_{PS} \Delta T \quad (5-9)$$

where ΔH_L is the enthalpy change of the juice, and X , C_{PS} are the mass fraction and the specific heat of the dry matter (non-juice dry solids), respectively (Saravacos and Kostaropoulos, 2002).

Typical values of specific heats of foods, used in refrigeration and freezing calculations, are the following:

- High-moisture foods above freezing ($T > 0$ °C): 3.5-3.9 kJ/kg K
- High-moisture foods below freezing ($T < 0$ °C): 1.8 -1.9 kJ/kg K
- Dehydrated foods: 1.3-2.1 kJ/kg K
- Fat ($T > 40$ °C): 1.7-2.2 kJ/kg K
- Fat ($T < 40$ °C): 1.5 kJ/kg K

The enthalpy of some important classes of food, such as fruits and vegetables, meats, and fats, are given in the Riedel diagrams as a function of moisture content and temperature. In the same diagrams, the percent unfrozen water can be estimated (Saravacos and Kostaropoulos, 2002; ASHRAE, 1993).

2. Cooling Equipment

a. Liquid Foods

The cooling of liquid foods is normally carried out in conventional heat exchangers, similar to those described in Chapter 4. Typical cooling units are:

- the plate heat exchanger,
- the shell and tube heat exchanger,
- the scraped surface heat exchanger,
- the shell and coil, and
- the jacketed vessels.

b. Solid Foods

i. Hydrocooling

Hydrocooling is realized by contacting solid foods with cold water, resulting in fast cooling of the product. The cooling process can be improved significantly by increasing the relative motion of solids/water, which results in high heat transfer coefficients at the water/solid interface. Hydrocooling is applied mainly to fresh fruits and vegetables, where cooling may be combined with washing of the product, prior to processing. The cooling water is recirculated through heat exchangers, and it may be chlorinated to prevent microbial growth. Hydrocooling equipment includes immersion and spraying, rotating (tumbling) drums, and tunnel systems.

ii. Air Cooling

Air cooling equipment for solid foods is similar to air (convective) dryers (Chapter 7). Low-temperature tunnels are similar to drying tunnels, with the food product moving on trucks or in conveyors.

iii. Vacuum Cooling

Leafy vegetables and liquid foods can be cooled rapidly by applying vacuum near to 6 mbar (near the freezing point of the product). Evaporation of free water results in very fast cooling of the product.

IV. FREEZING OF FOODS

1. Introduction

Freezing of food products involves crystallization of free water into ice and lowering the temperature to about $-18\text{ }^{\circ}\text{C}$, so that the product can be preserved for several months, without any appreciable microbial or enzymatic deterioration. The food freezing processes are discussed by Cleland (1990), Heldman (1992), Singh (1995), and Cleland and Valentas (1997).

The speed of freezing u , defined as the speed of movement of the “cold front” between unfrozen and frozen product, should be high enough (fast freezing), e.g., $u > 5\text{ cm/h}$, for economic operation and preservation of food quality.

The freezing point of food depends on its composition (moisture content), e.g., meat $-1.0\text{ }^{\circ}\text{C}$, fish $-2.0\text{ }^{\circ}\text{C}$, peaches $-1.4\text{ }^{\circ}\text{C}$, and cherries $-4.5\text{ }^{\circ}\text{C}$ (Saravacos and Kostaropoulos, 2002). The fraction of frozen water in frozen foods depends on the temperature and the moisture content, and the type of product, varying from 50 % (bread) to 90 % (fruit juice) at $-18\text{ }^{\circ}\text{C}$.

2. Freezing Time

The freezing time t_f is useful in design calculations of freezing processes and freezing equipment. It is estimated by the conventional Plank equation, which is derived from basic heat transfer and freezing analysis:

$$t_f = \frac{\rho \Delta H_L}{T_f - T_a} \left(\frac{2Pr}{h} + \frac{4Rr^2}{\lambda} \right) \quad (5-10)$$

where

ρ = density of the frozen product (kg/m^3)

ΔH_L = latent heat of crystallization of ice (kJ/kg)

T_f = freezing point ($^{\circ}\text{C}$)

T_a = temperature of freezing medium ($^{\circ}\text{C}$)

r = half-thickness (radius) of the product (m)

h = convective heat transfer coefficient ($\text{W/m}^2\text{K}$)

λ = thermal conductivity of the frozen product (W/m K)

P, R = shape factors:

sphere: $P = 1/16$ $R = 1/24$

infinite plate: $P = 1/2$ $R = 1/8$

infinite cylinder: $P = 1/4$ $R = 1/16$

The following modified Plank equation can be used for various shapes of solid foods (Cleland and Valentas, 1997):

$$t_f = \frac{1}{E} \left(\frac{\Delta H_1}{\Delta T_1} + \frac{\Delta H_2}{\Delta T_2} \right) \left(\frac{r}{h} + \frac{r^2}{2\lambda} \right) \quad (5-11)$$

where

$$\begin{aligned}
 E &= \text{shape factor related to heat transfer:} \\
 &\quad \text{sphere: } E = 3, \text{ infinite cylinder: } E = 2; \text{ and infinite plate (slab): } E = 1 \\
 \Delta T_1 &= 0.5 (T_i + T_{fm}) - T_a \\
 \Delta T_2 &= T_{fm} - T_a \\
 T_i &= \text{initial product temperature (}^\circ\text{C)} \\
 T_{fm} &= \text{initial freezing temperature (}^\circ\text{C)} \\
 T_{fm} &= \text{mean freezing temperature (}^\circ\text{C)} \\
 T_a &= \text{temperature of freezing medium (}^\circ\text{C)} \\
 \Delta H_1 &= \rho C_{pu} (T_i - T_{fm}) \text{ (kJ/m}^3\text{)} \\
 \Delta H_2 &= \rho \Delta H_L + \rho C_{pf} (T_{fm} - T_{fm}) \text{ (kJ/m}^3\text{)} \\
 T_{fm} &= 1.8 + 0.263 T_{fm} + 0.105 T_a \\
 C_{pu} &= \text{specific heat of unfrozen product (kJ/kg K)} \\
 C_{pf} &= \text{specific heat of frozen product (kJ/kg K)}
 \end{aligned}$$

The shape factor E (dimensionless) is estimated from the following empirical equations:

$$E = 1 + \frac{1 + \frac{2}{Bi}}{\beta_1^2 + \frac{2\beta_1}{Bi}} + \frac{1 + \frac{2}{Bi}}{\beta_2^2 + \frac{2\beta_2}{Bi}} \quad (5-12)$$

where

$$\begin{aligned}
 \beta_1 &= A/(\pi r^2) \\
 \beta_2 &= 3V(4\pi\beta_1 r^2) \\
 Bi &= (hr) / \lambda \\
 r &= \text{half thickness (radius)} \\
 V &= \text{volume} \\
 A &= \text{smallest cross sectional area of product (m}^2\text{)} \\
 h &= \text{convective heat transfer coefficient (W/m}^2\text{ K)} \\
 \lambda &= \text{thermal conductivity of frozen product (W/m K)}
 \end{aligned}$$

3. Heat Load

The total heat load Q (kJ) to be removed in a freezing process from a product of mass m includes lowering the product temperature from the ambient (initial) T_i to the mean freezing temperature T_o , removing the latent heat of crystallization ΔH_L , and lowering the product temperature to the temperature of the frozen product T_f (storage temperature), according to the equation:

$$Q = m [C_{pu} (T_i - T_o) + \Delta H_L + C_{pf} (T_o - T_f)] \quad (5-13)$$

where C_{pu} and C_{pf} are the specific heats of the unfrozen and frozen product, respectively.

4. Freezing Equipment

The freezing equipment is determined according to the type of product (liquid, solid), the freezing medium (air, cold surface, liquid), and the type of processing (batch, continuous).

a. Air Freezers

Forced convection cooling equipment is applied, using high-velocity fans to increase the heat transfer coefficient of air/product. Figures 5.5 and 5.6 show straight and curved (spiral) freezers. Tunnel freezers, similar to the tunnel coolers and tunnel dryers are often used. Figure 5.7 shows a continuous fluidized bed freezer, in which high heat transfer coefficients and short freezing times are obtained.

b. Cold Surface Freezers

Figure 5.8 shows a cold surface freezer, in which the food packages are pressed and frozen between cold plates.

c. Heat Exchanger Freezers

Heat exchangers, used in cooling viscous liquid foods, like the scraped surface, and the agitated vessel, can be used in freezing processes.

d. Cryogenic Liquids

Cryogenic liquids, such as liquid nitrogen and liquid carbon dioxide, can be used in freezing processes. They cause rapid freezing of the product, due to high heat transfer coefficients. Cryogenic freezing equipment is described by Saravacos and Kostaropoulos (2002).

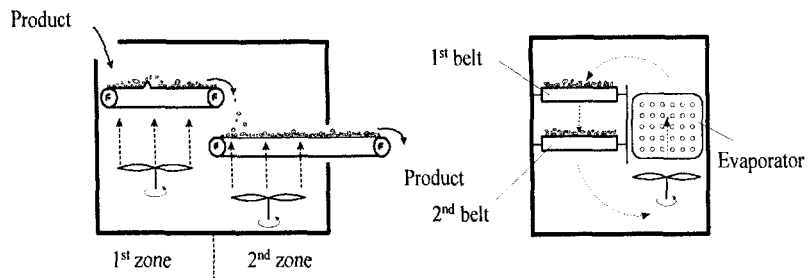


Figure 5.5 Straight-belt freezing equipment.

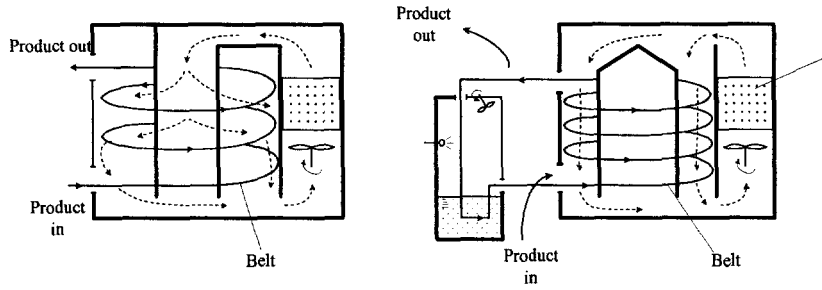


Figure 5.6 Spiral belt freezing equipment.

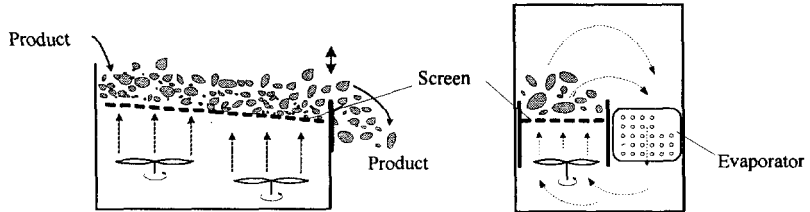


Figure 5.7 Fluidized bed freezing equipment.

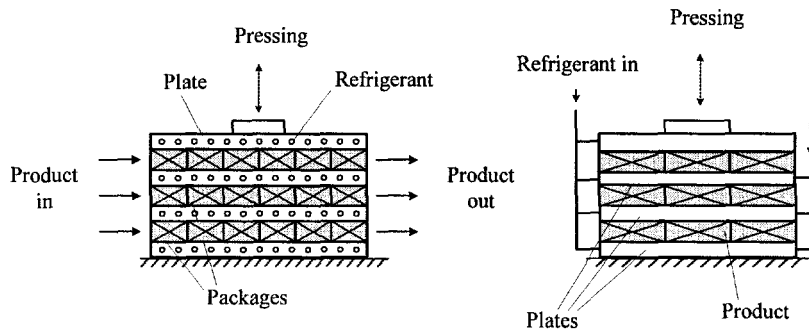


Figure 5.8 Horizontal plate freezing equipment.

V. THAWING EQUIPMENT

Frozen foods must be thawed before use by heating the product from the storage temperature (-18°C) to room or higher temperature. Thawing is important in food preparation and catering of large quantities of frozen foods. The following methods are used in commercial thawing (Hallstom et al., 1988):

- Convective (hot air) thawing, usually in air tunnels, which are similar to the freezing tunnels
- Hot water thawing
- Vacuum thawing
- Contact (hot surface) thawing
- Dielectric or microwave thawing.

VI. COLD STORAGE OF FOODS

Refrigerated and frozen foods are stored for short or long time in special insulated storage rooms, which are kept at a constant low temperature, using a mechanical refrigeration system. The refrigerated storage conditions for various foods are described by ASHRAE (1993), Rao (1992), and IIR (1967, 1972).

Food products are chilled rapidly near the freezing point, then frozen, and finally cooled to the low storage temperature. The refrigeration load of a storage room involves calculations for the removal of the following heat loads:

- Sensible heat of the fresh product,
- latent heat of ice crystallization,
- heat of respiration of the fresh product (W/ton),
- heat produced by the equipment (fans, blowers, etc.),
- heat due to leakage,
- heat produced by personnel, light, etc.

The refrigeration load for chilling and freezing is estimated per hour, while storage is estimated for 24 h operation.

In several applications, a secondary refrigeration system, e.g., ethylene glycol solution, is used to transfer the heat load from the room/product to the primary (compression) refrigeration system, which is usually installed in a separate room.

Controlled atmosphere can improve the storage life of fresh foods. Controlled atmosphere contains less oxygen and carbon dioxide.

VII. DESIGN OF A CONVEYOR BELT COOLER

In this section, a design approach is described for the conveyor belt cooler.

1. Process Description

A typical flowsheet of a conveyor belt cooler is presented in Figure 5.9. The feed with flow rate F (kg/s) and temperature T_1 ($^{\circ}\text{C}$) is distributed on the belt as it enters the cooler. The cooled product exits the cooler with the same flow rate F (kg/s) and temperature T_2 ($^{\circ}\text{C}$). The belt is moving at a velocity u_b (m/s) and requires an electrical power E_b (kW). The cooling air enters the cooler at a flow rate F_f (kg/s) and temperature T ($^{\circ}\text{C}$). An electrical power E_f (kW) is expended by the fan. The air conditions for design can be considered constant due to the high air recirculation.

The cooling air temperature is controlled in the evaporator of a heat pump. The refrigerant is evaporated at temperature T_e ($^{\circ}\text{C}$) in the evaporator of thermal load Q_e (kW), it is compressed in the compressor of power E_c (kW), then it is liquidified at temperature T_c ($^{\circ}\text{C}$) in the condenser of thermal load Q_c (kW) via cooling water, and finally it is expanded through a valve to the refrigerant evaporator.

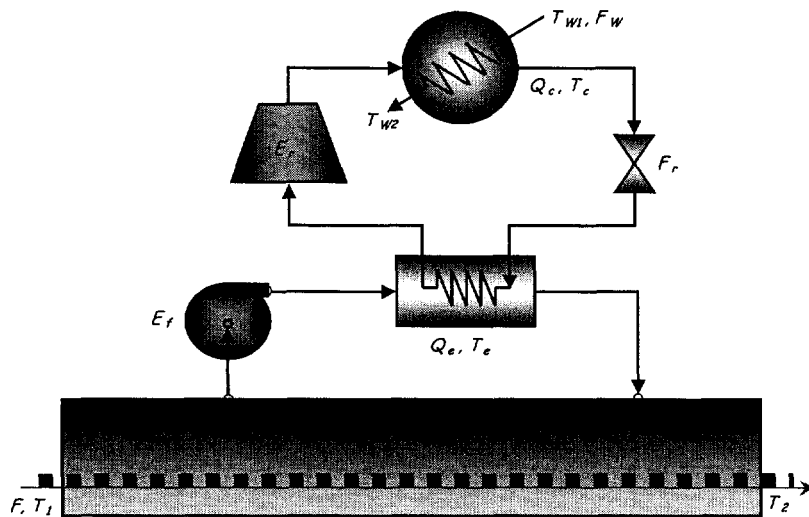


Figure 5.9 Schematic representation of a belt cooler.

2. Process Model

A mathematical model of the process presented in Figure 5.9 is summarized in Table 5.2.

Equation (E01) estimates the cooling time constant at cooling conditions, while Equation (E02) calculates the required cooling time. Equations (E01) and (E02) are based on the assumption that thermal diffusivity inside the food particles is the dominant heat transfer resistance.

Equations (E03) through (E05) constitute the refrigerant evaporator model. Equation (E03) calculates the required cooling load, which must be covered by the refrigerant evaporator. Equation (E04) calculates the heat which is absorbed by the refrigerant. Equation (E05) is the phenomenological equation for heat flow between the two streams in the refrigerant evaporator.

Similarly, Equations (E06) through (E08) constitute the refrigerant condenser model. Equation (E06) is the energy balance for the cooling water into the condenser, Equation (E07) the energy balance for the condensing refrigerant, while Equation (E08) describes the heat flow rate between these two streams.

Equation (E09) calculates the required power for the refrigerant compressor. It must be noted that Equations (E04), (E07) and (E09) come from the shortcut design procedure for refrigeration systems proposed by Shelton and Grossmann (1985).

Equations (E10) through (E15) are used for sizing the conveyor belt. Equation (E10) correlates the residence time with the mass holdup, while Equation (E11) the mass holdup with the volume holdup. Equation (E12) is the geometrical evaluation of the volume holdup on the belt. Equation (E13) calculates the required belt area, while Equation (E14) estimates the required belt velocity to obtain the desired residence time. Equation (E15) estimates the required electrical power to move the belt.

Equations (E16) through (E18) are used for sizing the fan. Equation (E16) calculates the friction pressure loss of air through the loaded belt. Equation (E17) correlates the air flow with the air velocity. Equation (E18) estimates the required electrical power to operate the fan.

Finally, Equation (E19) summarizes the required electrical power for the system.

Thirty one variables presented in Table 5.3 are involved in the model of nineteen equations presented in Table 5.2. The corresponding technical data are summarized in Table 5.4. The process specifications of a typical design problem, a degrees-of-freedom analysis, a suggestion for the design variables selection and the corresponding solution algorithm are presented in Table 5.5. The total annualized cost TAC presented in Table 5.6 is used as objective function in process optimization. The required cost data are summarized in Table 5.7.

Table 5.2 Belt Cooler Model**Cooling Time**

$$t_c = d^2 / (\pi^2 \alpha) \quad (\text{E01})$$

$$t = -t_c \ln[(T_1 - T) / (T_2 - T)] \quad (\text{E02})$$

Refrigerant Evaporator

$$Q_e = F C_{ps} (T_1 - T_2) \quad (\text{E03})$$

$$Q_e = [\Delta H_r - C_{pr} (T_c - T_d)] F_r \quad (\text{E04})$$

$$Q_e = A_e U_e (T - T_d) \quad (\text{E05})$$

Refrigerant Condenser

$$Q_c = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E06})$$

$$Q_c = [\Delta H_r (T_c / T_e) - C_{pr} (T_c - T_d)] F_r \quad (\text{E07})$$

$$Q_c = A_c U_c [(T_c - T_{w1}) - (T_c - T_{w2})] / \ln[(T_c - T_{w1}) / (T_c - T_{w2})] \quad (\text{E08})$$

Refrigerant Compressor

$$E_c = \Delta H_r [(T_c - T_d) / T_d] F_r \quad (\text{E09})$$

Belt

$$M = t F \quad (\text{E10})$$

$$M = (1 - e) \rho_s H \quad (\text{E11})$$

$$H = Z D L \quad (\text{E12})$$

$$A_b = L D \quad (\text{E13})$$

$$u_b = L / t \quad (\text{E14})$$

$$E_b = b L F \quad (\text{E15})$$

Fan

$$\Delta P = f (Z/d) \rho_a u^2 \quad (\text{E16})$$

$$F_a = u D L \quad (\text{E17})$$

$$E_f = \Delta P F_a \quad (\text{E18})$$

Electrical energy requirements

$$E = E_b + E_f + E_c \quad (\text{E19})$$

Table 5.3 Process Variables

| | | |
|------------|-------------|--|
| A_b | m^2 | Belt area |
| A_c | m^2 | Heat transfer area of the refrigerant condenser |
| A_e | m^2 | Heat transfer area of the refrigerant evaporator |
| D | m | Belt width |
| d | m | Particle size |
| E | kW | Total power requirement |
| E_b | kW | Belt driver power |
| E_c | kW | Refrigerant compressor power |
| E_f | kW | Fan power |
| F | kg/s | Material flow rate |
| F_a | m^3/s | Flow rate of the circulated air |
| F_r | kg/s | Refrigerant flow rate |
| F_w | kg/s | Cooling water flow rate |
| H | m^3 | Cooler volume holdup |
| L | m | Belt length |
| M | ton | Cooler mass holdup |
| Q_c | kW | Refrigerant condenser thermal load |
| Q_e | kW | Refrigerant evaporator thermal load |
| t | h | Cooling time |
| T | $^{\circ}C$ | Cooling air temperature |
| T_1 | $^{\circ}C$ | Initial temperature of food |
| T_2 | $^{\circ}C$ | Final temperature of food |
| t_c | h | Cooling time constant at cooling conditions |
| T_c | $^{\circ}C$ | Refrigerant condensing temperature |
| T_e | $^{\circ}C$ | Refrigerant evaporating temperature |
| T_{w1} | $^{\circ}C$ | Cooling water inlet temperature |
| T_{w2} | $^{\circ}C$ | Cooling water outlet temperature |
| u | m/s | Cooling air velocity |
| u_b | m/s | Belt velocity |
| Z | m | Loading depth |
| ΔP | bar | Pressure loss of air flowing through belt |

Table 5.4 Process Data

Density

| | | |
|----------|-------------------|------------------|
| ρ_a | kg/m ³ | Air density |
| ρ_s | kg/m ³ | Material density |

Specific heat

| | | |
|----------|---------|-------------|
| C_{pw} | kJ/kg K | Water |
| C_{pr} | kJ/kg K | Refrigerant |
| C_{pa} | kJ/kg K | Air |
| C_{ps} | kJ/kg K | Food |

Thermal diffusivity

| | | |
|----------|-------------------|------|
| α | m ² /s | Food |
|----------|-------------------|------|

Latent heat of evaporation

| | | |
|--------------|-------|-------------|
| ΔH_w | kJ/kg | Water |
| ΔH_r | kJ/kg | Refrigerant |

Heat transfer coefficients

| | | |
|-------|---------------------|---|
| U_e | kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_c | kW/m ² K | Heat transfer coefficient at the refrigerant condenser |

Empirical constants

| | | |
|---------------|---|---------------------------|
| b | - | Belt drive power equation |
| f | - | Pressure loss equation |
| ε | - | Void (empty) fraction |

Table 5.5 Process Solution**Process Specifications**

| | | |
|----------|-------|----------------------------------|
| F | ton/h | Material flow rate |
| T_1 | °C | Initial temperature of food |
| T_2 | °C | Final temperature of food |
| d | m | Food particle size |
| T_{w1} | °C | Cooling water inlet temperature |
| T_{w2} | °C | Cooling water outlet temperature |

Degrees-of-Freedom Analysis

| | | | |
|--------------------|----|--------------------|----------|
| Process variables | 31 | Degrees of freedom | 12 |
| Process equations | 19 | Specifications | <u>6</u> |
| Degrees of freedom | 12 | Design variables | 6 |

Design Variables

| | | |
|-------|-----|-------------------------------------|
| D | m | Belt width |
| T_c | °C | Refrigerant condensing temperature |
| T_e | °C | Refrigerant evaporating temperature |
| T | °C | Cooling air temperature |
| u | m/s | Cooling air velocity |
| Z | m | Loading depth |

Model Solution Algorithm

| | | |
|-------|---|------------|
| (E01) | → | t_c |
| (E02) | → | t |
| (E03) | → | Q_e |
| (E04) | → | F_r |
| (E05) | → | A_e |
| (E07) | → | Q_c |
| (E06) | → | F_w |
| (E08) | → | A_c |
| (E09) | → | E_c |
| (E10) | → | M |
| (E11) | → | H |
| (E12) | → | L |
| (E13) | → | A_b |
| (E14) | → | u_b |
| (E15) | → | E_b |
| (E16) | → | ΔP |
| (E17) | → | F_a |
| (E18) | → | E_f |
| (E19) | → | E |

Table 5.6 Cost Analysis**Equipment cost**

$$C_{eq} = C_{bel} A^{n_{bel}} + C_{exc} A_e^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{fan} E_f^{n_{fan}} + C_{com} E_c^{n_{com}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q_c + C_e (E_c + E_f + E_b)) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (F04)$$

Cost index

$$c = TAC / (F t_y) \quad (F05)$$

Table 5.7 Cost Data**Utility cost**

C_e \$/kWh Cost of electricity

C_w \$/kWh Cost of cooling water

Equipment unit cost

C_{bel} \$/m² Belt cooler

C_{exc} \$/m² Heat exchanger

C_{fan} \$/kW Fan

C_{com} \$/kW Compressor

Equipment size scaling factor

n_{bel} - Belt cooler

n_{exc} - Heat exchanger

n_{fan} - Fan

n_{com} - Compressor

Other

t_y h/yr Annual operating time

i_r - Interest rate

l_f yr Lifetime

3. Application to Strawberry Cooling

A typical application to strawberries cooling is described in this section. The process specifications and the required technical and economic data are summarized in Tables 5.6 through 5.10. The properties of strawberry were taken from Rahman (1995). The optimal values of the design variables are presented in Table 5.11, while the corresponding technical and economic results are summarized in Tables 5.12 and 5.13. Some of these results are also presented in Figures 5.10 through 5.12.

Table 5.8 Process Data**Density**

| | | |
|----------|--------------------------|------------|
| ρ_a | = 1.00 kg/m ³ | Air |
| ρ_s | = 1000 kg/m ³ | Strawberry |

Specific heat

| | | |
|----------|---------------|-----------------------------|
| C_{pw} | = 4.20 kJ/kgK | Water |
| C_{pr} | = 1.20 kJ/kgK | Refrigerant R-717 (Ammonia) |
| C_{pa} | = 1.00 kJ/kgK | Air |
| C_{ps} | = 3.80 kJ/kgK | Food |

Thermal diffusivity

| | | |
|----------|---|------|
| α | = 1.18 10 ⁻⁷ m ² /s | Food |
|----------|---|------|

Latent heat of evaporation

| | | |
|--------------|--------------|-----------------------------|
| ΔH_w | = 2500 kJ/kg | Water |
| ΔH_r | = 1250 kJ/kg | Refrigerant R-717 (Ammonia) |

Heat transfer coefficients

| | | |
|-------|----------------------------|---|
| U_e | = 0.25 kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_c | = 0.75 kW/m ² K | Heat transfer coefficient at the refrigerant condenser |

Empirical constants

| | | |
|---------------|--------|----------------------------|
| b | = 2.00 | Belt driver power equation |
| f | = 0.20 | Pressure loss equation |
| ε | = 0.45 | Void (empty) fraction |

Table 5.9 Cost Data**Utility cost**

| | | |
|-------|---------------|-----------------------|
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_w | = 0.01 \$/kWh | Cost of cooling water |

Equipment unit cost

| | | |
|-----------|-------------------------|----------------|
| C_{bel} | = 10 k\$/m ² | Belt cooler |
| C_{exc} | = 3 k\$/m ² | Heat exchanger |
| C_{fan} | = 1 k\$/kW | Fan |
| C_{com} | = 3 k\$/kW | Compressor |

Equipment size scaling factor

| | | |
|-----------|--------|----------------|
| n_{bel} | = 0.95 | Belt cooler |
| n_{exc} | = 0.65 | Heat exchanger |
| n_{fan} | = 0.75 | Fan |
| n_{com} | = 0.75 | Compressor |

Other

| | | |
|-------|------------|-----------------------|
| t_y | = 720 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 5.10 Process Specifications

| | | |
|----------|--------------|----------------------------------|
| F | = 0.50 ton/h | Material flow rate |
| T_1 | = 32 °C | Initial temperature of food |
| T_2 | = 2 °C | Final temperature of food |
| d | = 0.03 m | Particle size |
| T_{w1} | = 15 °C | Cooling water inlet temperature |
| T_{w2} | = 25 °C | Cooling water outlet temperature |

Table 5.11 Design Variables

| | | |
|-------|----------|-------------------------------------|
| D | = 1 m | Belt width |
| T_c | = 35 °C | Refrigerant condensing temperature |
| T_e | = -18 °C | Refrigerant evaporating temperature |
| T | = -8 °C | Cooling air temperature |
| u | = 5 m/s | Cooling air velocity |
| Z | = 0.10 m | Loading depth |

Table 5.12 Cost Analysis Results**Equipment cost**

| | | |
|------------|------------|----------------|
| C_{bel} | = 28.7 k\$ | Belt |
| C_{con} | = 4.4 k\$ | Condenser |
| C_{evap} | = 10.0 k\$ | Evaporator |
| C_{fan} | = 0.4 k\$ | Fan |
| C_{com} | = 7.6 k\$ | Compressor |
| C_{eq} | = 51.0 k\$ | Equipment cost |

Operating cost

| | | |
|----------|--------------|-----------------------|
| C_e | = 0.4 k\$/yr | Cost of electricity |
| C_w | = 0.1 k\$/yr | Cost of cooling water |
| C_{op} | = 0.5 k\$/yr | Operating cost |

Objective function

| | | |
|-----------|--------------|---------------------------|
| eC_{eq} | = 7.6 k\$/yr | Annualized equipment cost |
| C_{op} | = 0.5 k\$/yr | Operating cost |
| TAC | = 8.1 k\$/yr | Total annual cost |

Cost index

| | | |
|-----|---------------|-------------------|
| c | = 0.022 \$/kg | Cooling unit cost |
|-----|---------------|-------------------|

Table 5.13 Process Design Results**Belt geometry**

| | | |
|-------|----------------------|-----------------------|
| A_b | $= 3 \text{ m}^2$ | Belt area |
| D | $= 1 \text{ m}$ | Belt width |
| L | $= 3 \text{ m}$ | Belt length |
| Z | $= 0.10 \text{ m}$ | Loading depth |
| H | $= 0.30 \text{ m}^3$ | Freezer volume holdup |
| d | $= 0.03 \text{ m}$ | Particle size |
| M | $= 0.17 \text{ ton}$ | Cooler mass holdup |

Belt operation

| | | |
|-------|-------------------------------|--|
| F | $= 0.50 \text{ ton/h}$ | Material flow rate |
| T_1 | $= 32 \text{ }^\circ\text{C}$ | Initial temperature of food |
| T_2 | $= 2 \text{ }^\circ\text{C}$ | Final temperature of food |
| T | $= -8 \text{ }^\circ\text{C}$ | Cooling air temperature |
| t_c | $= 0.24 \text{ h}$ | Freezing time constant at cooling conditions |
| t | $= 0.33 \text{ h}$ | Freezing time |
| u_b | $= 9.1 \text{ m/h}$ | Belt velocity |
| E_b | $= 0.80 \text{ kW}$ | Belt drive power |

Refrigerant evaporator

| | | |
|-------|--------------------------------|--|
| T_e | $= -18 \text{ }^\circ\text{C}$ | Refrigerant evaporating temperature |
| Q_e | $= 15.8 \text{ kW}$ | Refrigerant evaporator thermal load |
| F_r | $= 0.01 \text{ kg/s}$ | Refrigerant flow rate |
| A_e | $= 6.3 \text{ m}^2$ | Heat transfer area of the refrigerant evaporator |

Refrigerant condenser

| | | |
|----------|-------------------------------|---|
| T_c | $= 35 \text{ }^\circ\text{C}$ | Refrigerant condensing temperature |
| Q_c | $= 19.3 \text{ kW}$ | Refrigerant condenser thermal load |
| A_c | $= 1.8 \text{ m}^2$ | Heat transfer area of the refrigerant condenser |
| F_w | $= 0.46 \text{ kg/s}$ | Cooling water flow rate |
| T_{w1} | $= 15 \text{ }^\circ\text{C}$ | Cooling water inlet temperature |
| T_{w2} | $= 25 \text{ }^\circ\text{C}$ | Cooling water outlet temperature |

Refrigerant compressor

| | | |
|-------|--------------------|------------------------------|
| E_c | $= 3.5 \text{ kW}$ | Refrigerant compressor power |
|-------|--------------------|------------------------------|

Fan

| | | |
|------------|-----------------------------|---|
| u | $= 5 \text{ m/s}$ | Cooling air velocity |
| ΔP | $= 17 \text{ Pa}$ | Pressure loss of air flowing through belt |
| F_a | $= 15 \text{ m}^3/\text{s}$ | Flow rate of the circulated air |
| E_f | $= 0.3 \text{ kW}$ | Fan power |
| E | $= 4.6 \text{ kW}$ | Total power requirement |

Figure 5.10 represents an enthalpy-temperature diagram for the system. It shows the temperature differences and the corresponding thermal loads for each process.

Figure 5.11 distributes the equipment and operating cost into the different processes, while Figure 5.12 represents the total cost distribution to the main processes.

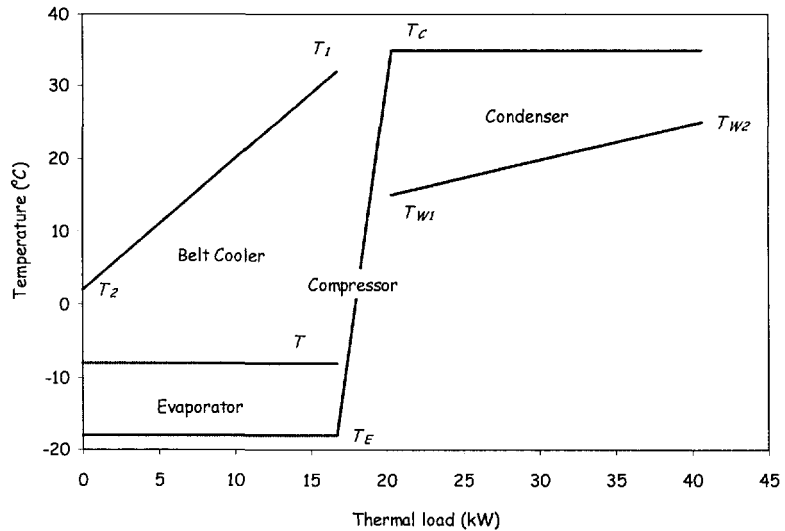


Figure 5.10 Enthalpy-temperature diagram.

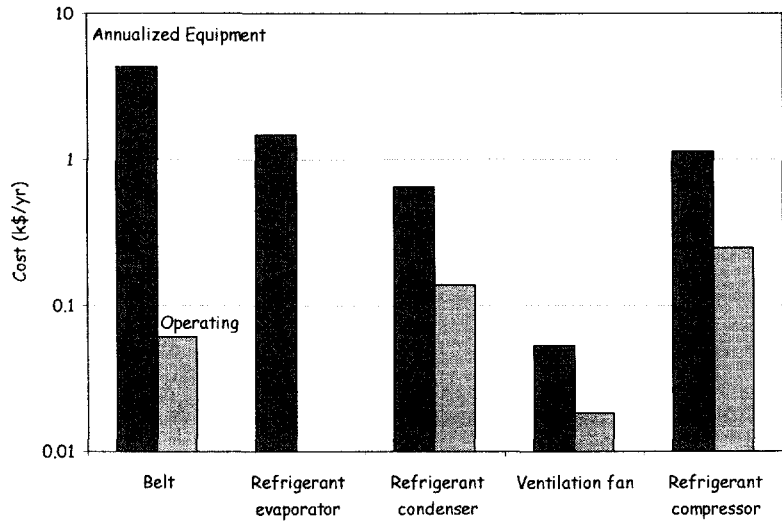


Figure 5.11 Equipment and operating cost.

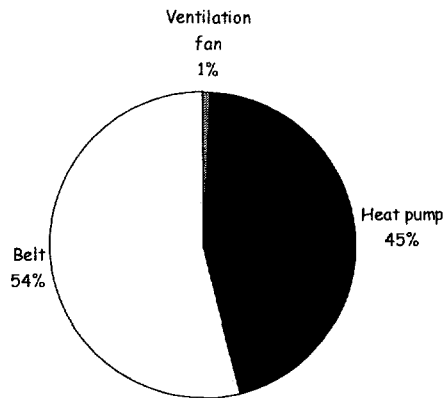


Figure 5.12 Cost breakdown.

VIII. DESIGN OF A COLD STORAGE ROOM

In this section, a design approach is described for a typical cold storage room.

1. Process Description

A flowsheet of a typical cold storage room is presented in Figure 5.13. The food material of mass M (ton) is kept in a well insulated room for some appropriate time t_y (hr/yr). The cold room is rectangular with dimensions D (m) \times Z (m) \times L (m) resulting in volume V (m³). The stored material is packed in boxes or/and pallettes so as a void (empty) fraction ε (-) is obtained to permit the cold air circulation.

The cold storage room is kept at temperature T (°C) by circulation of cold air with superficial velocity u (m/s) via a fan of power E_f (kW). The circulated air is cooled in the evaporator of a heat pump. The refrigerant is evaporated at temperature T_e (°C) in the evaporator of thermal load Q_e (kW), it is compressed in the compressor of power E_c (kW), then it is liquidified at temperature T_c (°C) in the condenser of thermal load Q_c (kW) via cooling water, and finally it is expanded through a valve to the refrigerant evaporator.

The refrigeration load required by the heat pump is obtained through energy balance of the room. The thermal energy which inputs into or outputs from the storage room is presented schematically in Figure 5.14. Thermal energy Q_L (kW) is transferred into the room through the insulated wall from the environment which is at higher temperature T_a (°C) than that of the cold room T (°C). Thermal energy Q_A (kW) is also introduced through the air renewal. The fan also adds a thermal load Q_f (kW). The food material produces thermal energy Q_R (kW) due to respiration and absorbs thermal energy Q_W (kW) due to water, which is evaporated during the storage. Finally, the heat pump removes thermal energy Q_e (kW) in order to keep the room temperature at the desired level T (°C).

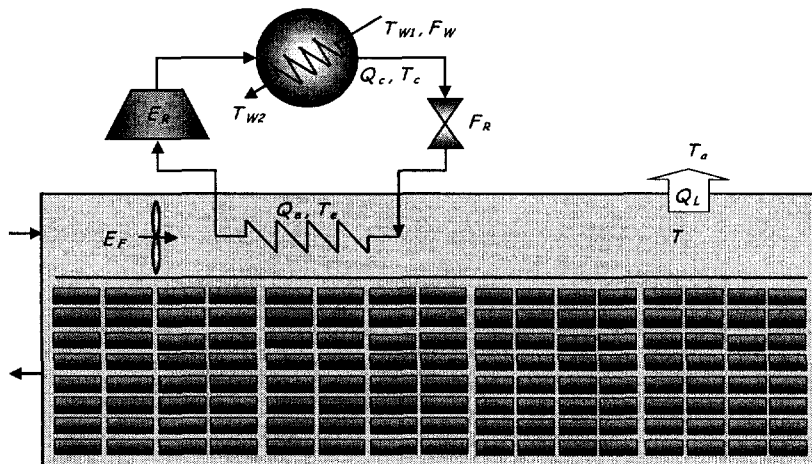


Figure 5.13 Schematic representation of a cold storage room.

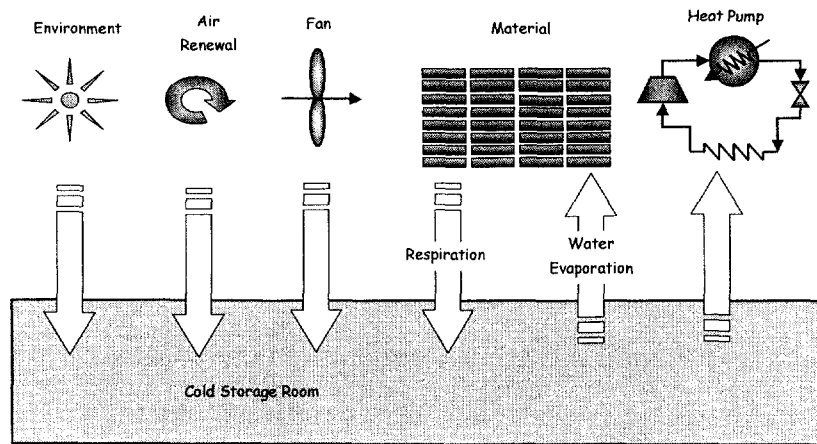


Figure 5.14 Schematic representation of the energy flow to and from a cold storage room.

2. Process Model

A mathematical model of the process presented in Figure 5.13 is summarized in Table 5.14.

Equations (E01) through (E03) are of geometrical nature. Equation (E01) calculates the total mass of the food of given density, which contains in the room of given volume, when the packing void fraction is given. Equation (E02) calculates the volume of the room from its dimensions, while Equation (E03) calculates the total surface of the room, through which thermal leakage occurs.

Equations (E04) through (E08) calculate the thermal flows to and from the room as they are described in Figure 5.14, that is Equation (E04) describes the thermal leakage to the surroundings, (E05) the heat input due to the air renewal, (E06) the heat due to the ventilation fan, (E07) the respiration heat, and (E08) the thermal energy which is absorbed by the water, which is evaporated.

Equations (E09) through (E011) constitute the refrigerant evaporator model. Equation (E09) calculates the required thermal load for the room, which must be covered by the refrigerant evaporator. Equation (E10) calculates the heat which is absorbed by the refrigerant. Equation (E11) is the phenomenological equation for heat flow between the two streams in the refrigerant evaporator.

Similarly, Equations (E12) through (E14) constitute the refrigerant condenser model. Equation (E12) is the energy balance for the cooling water into the condenser, Equation (E13) is the energy balance for the condensing refrigerant, while Equation (E14) describes the heat flow rate between these two streams.

Equation (E15) calculates the required power for the refrigerant compressor. It should be noted that Equations (E10), (E13) and (E15) come from the shortcut design procedure for refrigeration systems proposed by Shelton and Grossmann (1985).

Equations (E16) to (E18) are used to size the ventilation fan. Equation (E16) calculates the friction pressure loss of the air passing through the food material. Equation (E17) calculates the required air flow to guarantee the desired superficial air velocity. Equation (E18) calculates the required size of the ventilation fan.

Finally, Equation (E19) summarizes the required electrical power for the entire system.

Table 5.14 Cold Room Storage Model

Room Geometry

$$M = (1 - \varepsilon) \rho_s V \quad (\text{E01})$$

$$V = D Z L \quad (\text{E02})$$

$$A_L = 2 (ZL + LD + DZ) \quad (\text{E03})$$

Cold Storage Load

$$Q_L = A_L U_L (T - T_a) \quad (\text{E04})$$

$$Q_A = \rho_a C_{pa} (T_a - T) \varepsilon V / t_{ar} \quad (\text{E05})$$

$$Q_F = E_f \quad (\text{E06})$$

$$Q_R = r M \quad (\text{E07})$$

$$Q_W = w M \Delta H_w \quad (\text{E08})$$

Refrigerant Evaporator

$$Q_e = Q_L + Q_R + Q_A + Q_F - Q_W \quad (\text{E09})$$

$$Q_e = [\Delta H_r - C_{pr} (T_c - T_e)] F_r \quad (\text{E10})$$

$$Q_e = A_e U_e (T - T_e) \quad (\text{E11})$$

Refrigerant Condenser

$$Q_c = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E12})$$

$$Q_c = [\Delta H_r (T_c / T_e) - C_{pr} (T_c - T_e)] F_r \quad (\text{E13})$$

$$Q_c = A_c U_c [(T_c - T_{w1}) - (T_c - T_{w2})] / \ln[(T_c - T_{w1}) / (T_c - T_{w2})] \quad (\text{E14})$$

Refrigerant Compressor

$$E_c = \Delta H_r [(T_c - T_e) / T_e] F_r \quad (\text{E15})$$

Ventilation Fans

$$\Delta P = f (L/d) \rho_a u^2 \quad (\text{E16})$$

$$F_a = \rho_a u D Z \quad (\text{E17})$$

$$E_f = \Delta P F_a / \rho_a \quad (\text{E18})$$

Electrical energy requirements

$$E = E_c + E_f \quad (\text{E19})$$

Thirty one variables presented in Table 5.15 are involved in the model of nineteen equations presented in Table 5.14. The corresponding technical data are summarized in Table 5.16. The process specifications of a typical design problem, a degrees-of-freedom analysis, a suggestion for the design variables selection and the corresponding solution algorithm are presented in Table 5.17. The total annualized cost TAC presented in Table 5.18 is used as objective function in process optimization. The required cost data are summarized in Table 5.19.

Table 5.15 Process Variables

| | | |
|---------------|-------------|--|
| A_c | m^2 | Heat transfer area of the refrigerant condenser |
| A_e | m^2 | Heat transfer area of the refrigerant evaporator |
| A_L | m^2 | Heat loss area of the cold storage room |
| D | m | Cold room width |
| E | kW | Total power requirements |
| E_c | kW | Refrigerant compressor power |
| E_f | kW | Fan power |
| F_a | ton/h | Flow rate of the circulated air |
| F_r | ton/h | Refrigerant flow rate |
| F_w | ton/h | Cooling water flow rate |
| L | m | Cold room length |
| M | ton | Mass of the stored material |
| Q_A | kW | Input heat due to air renewal |
| Q_c | kW | Refrigerant condenser thermal load |
| Q_e | kW | Refrigerant evaporator thermal load |
| Q_f | kW | Input heat due to fan |
| Q_L | kW | Input heat from the environment |
| Q_R | kW | Heat of respiration |
| Q_W | kW | Heat removal due to water evaporation |
| T | $^{\circ}C$ | Cold room temperature |
| T_a | $^{\circ}C$ | Ambient temperature |
| t_{ar} | h | Required time for air renewal |
| T_c | $^{\circ}C$ | Refrigerant condensing temperature |
| T_e | $^{\circ}C$ | Refrigerant evaporating temperature |
| T_{w1} | $^{\circ}C$ | Cooling water inlet temperature |
| T_{w2} | $^{\circ}C$ | Cooling water outlet temperature |
| u | m/s | Superficial air velocity |
| V | m^3 | Cold room volume |
| Z | m | Cold room height |
| ΔP | bar | Pressure loss of air flowing through the room |
| ε | - | Packing void (empty) fraction |

Table 5.16 Process Data

| | | |
|--------------|-----------------------------------|---|
| | Density | |
| ρ_a | kg/m ³ | Air density |
| ρ_s | kg/m ³ | Material density |
| | Specific heat | |
| C_{pw} | kJ/kg K | Water |
| C_{pr} | kJ/kg K | Refrigerant |
| C_{pa} | kJ/kg K | Air |
| | Latent heat of evaporation | |
| ΔH_w | kJ/kg | Water |
| ΔH_r | kJ/kg | Refrigerant |
| | Heat transfer coefficients | |
| U_L | kW/m ² K | Overall heat loss coefficient through the wall |
| U_E | kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_C | kW/m ² K | Heat transfer coefficient at the refrigerant condenser |
| | Other constants | |
| w | kg/ton day | Water loss rate |
| r | W/ton | Heat of respiration |
| f | - | Pressure loss equation |

Concerning the design variables the following points should be considered:

- The packing void (empty) fraction depends on the food material arrangement in boxes and /or pallets. Consequently, a few values can be considered depending on the alternative packaging strategies.
- The air removal time period depends on the food material characteristics. The minimum accepted value for quality standards is usually selected. Higher values require more energy consumption.
- Similarly, the air velocity is selected at the minimum accepted level for quality reasons. Higher values require more energy consumption.
- The selection of room width and height has a minor effect on the objective function. It determines the shape of the room and, consequently, it depends on the available space.
- Refrigerant condensing and evaporating temperature affect the performance of the heat pump. The equipment and operating cost of a heat pump depend on the difference between condensing and evaporating temperatures. As this difference decreases the operating cost decreases but the equipment cost increases, due to the larger heat transfer area required for both condenser and evaporator. Thus, an optimum value generally exists.

Table 5.17 Process Solution

Process Specifications

| | | |
|----------|-----|----------------------------------|
| M | ton | Mass of the stored material |
| T_a | °C | Ambient temperature |
| T | °C | Cold room temperature |
| T_{w1} | °C | Cooling water inlet temperature |
| T_{w2} | °C | Cooling water outlet temperature |

Degrees-of-Freedom Analysis

| | | | |
|--------------------|-----------|--------------------|----------|
| Process variables | 31 | Degrees of freedom | 12 |
| Process equations | <u>19</u> | Specifications | <u>5</u> |
| Degrees of freedom | 12 | Design variables | 7 |

Design Variables

| | | |
|---------------|-----|-------------------------------------|
| ε | - | Packing void (empty) fraction |
| T_c | °C | Refrigerant condensing temperature |
| T_e | °C | Refrigerant evaporating temperature |
| t_{ar} | m/s | Required time for air removal |
| D | m | Cold room width |
| Z | m | Cold room height |
| u | m/s | Superficial air velocity |

Model Solution Algorithm

| | | |
|-------|---|------------|
| (E01) | → | V |
| (E02) | → | L |
| (E03) | → | A_L |
| (E04) | → | Q_L |
| (E05) | → | Q_A |
| (E16) | → | ΔP |
| (E17) | → | F_a |
| (E18) | → | E_f |
| (E06) | → | Q_F |
| (E07) | → | Q_R |
| (E08) | → | Q_W |
| (E09) | → | Q_e |
| (E10) | → | F_r |
| (E11) | → | A_e |
| (E13) | → | Q_c |
| (E12) | → | F_w |
| (E14) | → | A_c |
| (E15) | → | E_c |
| (E19) | → | E |

Table 5.18 Cost Analysis**Equipment cost**

$$C_{eq} = C_{rm} A_L^{n_{rm}} + C_{exc} A_e^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{fan} E_f^{n_{fan}} + C_{com} E_c^{n_{com}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q_c + C_e (E_c + E_f)) t_y + C_l DL \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (F04)$$

Cost index

$$c = TAC/M \quad (F05)$$

Table 5.19 Cost Data**Utility cost**

| | | |
|-------|----------------------|-----------------------|
| C_e | \$/kWh | Cost of electricity |
| C_w | \$/kWh | Cost of cooling water |
| C_l | \$/m ² yr | Cost of land |

Equipment unit cost

| | | |
|-----------|-------------------|----------------|
| C_{rm} | \$/m ² | Room |
| C_{exc} | \$/m ² | Heat exchanger |
| C_{fan} | \$/kW | Fan |
| C_{com} | \$/kW | Compressor |

Equipment size scaling factor

| | | |
|-----------|---|----------------|
| n_{rm} | - | Room |
| n_{exc} | - | Heat exchanger |
| n_{fan} | - | Fan |
| n_{com} | - | Compressor |

Other

| | | |
|-------|------|-----------------------|
| t_y | h/yr | Annual operating time |
| i_r | - | Interest rate |
| l_f | yr | Lifetime |

3. Application to Seasonal Cold Storage of Apples

Apples are preserved in cold room storage for several months. The process specifications and the required technical and economic data are summarized in Tables 5.20 through 5.22. The optimal values of the design variables are presented in Table 5.23, while the corresponding technical and economic results are summarized in Tables 5.24 and 5.25.

Figure 5.15 represents an enthalpy-temperature diagram for the heat pump of the system. It shows the temperature differences and the corresponding thermal loads for each individual process.

The heat energy balance of the cold room is further schematically represented in Figure 5.16.

The equipment and operating cost is also schematically represented in Figure 5.17, while in Figure 5.18 the cost is distributed to the processes of the system.

It must be noted that the required storage temperature is a crucial specification, which affects the cold storage room economics. Figure 5.19 represents the effect of the cold room temperature on the required refrigeration load and the refrigerant compressor power.

Table 5.20 Process Data

| | | |
|-----------------------------------|----------------------------|---|
| Density | | |
| ρ_a | = 1.00 kg/m ³ | Air density |
| ρ_s | = 850 kg/m ³ | Material density |
| Specific heat | | |
| C_{pw} | = 4.20 kJ/kg K | Water |
| C_{pr} | = 1.20 kJ/kg K | Refrigerant R-717 (Ammonia) |
| C_{pa} | = 1.00 kJ/kg K | Air |
| Latent heat of evaporation | | |
| ΔH_w | = 2500 kJ/kg | Water |
| ΔH_r | = 1250 kJ/kg | Refrigerant R-717 (Ammonia) |
| Heat transfer coefficients | | |
| U_L | = 0.15 W/m ² K | Overall heat loss coefficient through the wall |
| U_E | = 0.25 kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_C | = 0.75 kW/m ² K | Heat transfer coefficient at the refrigerant condenser |
| Other constants | | |
| w | = 0.50 kg/ton day | Water loss rate |
| r | = 14.7 W/ton | Heat of respiration |
| f | = 0.20 | Pressure loss equation |

Table 5.21 Cost Data**Utility cost**

| | | |
|-------|-----------------------------|-----------------------|
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_w | = 0.01 \$/kWh | Cost of cooling water |
| C_l | = 5.00 \$/m ² yr | Cost of land |

Equipment unit cost

| | | |
|-----------|---------------------------|----------------|
| C_{rm} | = 0.05 k\$/m ² | Room |
| C_{exc} | = 3.00 k\$/m ² | Heat exchanger |
| C_{fan} | = 1.00 k\$/kW | Fan |
| C_{com} | = 3.00 k\$/kW | Compressor |

Equipment size scaling factor

| | | |
|-----------|--------|----------------|
| n_{rm} | = 0.95 | Room |
| n_{exc} | = 0.65 | Heat exchanger |
| n_{fan} | = 0.75 | Fan |
| n_{com} | = 0.75 | Compressor |

Other

| | | |
|-------|-------------|-----------------------|
| t_y | = 3600 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 5.22 Process Specifications

| | | |
|----------|-----------|----------------------------------|
| M | = 360 ton | Mass of the stored material |
| T_a | = 35 °C | Ambient temperature |
| T | = 2 °C | Cold room temperature |
| T_{W1} | = 15 °C | Cooling water inlet temperature |
| T_{W2} | = 25 °C | Cooling water outlet temperature |

Table 5.23 Design Variables

| | | |
|---------------|---------|-------------------------------------|
| ε | = 0.85 | Packing void (empty) fraction |
| T_c | = 35 °C | Refrigerant condensing temperature |
| T_e | = -8 °C | Refrigerant evaporating temperature |
| t_{ar} | = 6 hr | Required time for air removal |
| D | = 15 m | Cold room width |
| Z | = 5 m | Cold room height |
| u | = 1 m/s | Superficial air velocity |

Table 5.24 Process Design Results

Cold chamber

| | | |
|---------------|-----------------------|---|
| M | = 360 ton | Mass of the stored material |
| ε | = 0.85 | Packaging empty void |
| D | = 15 m | Cold room width |
| L | = 38 m | Cold room length |
| Z | = 5 m | Cold room height |
| V | = 2824 m ³ | Cold room volume |
| A_L | = 1656 m ² | Heat loss area of the cold storage room |
| T | = 2 °C | Cold room temperature |
| T_a | = 35 °C | Ambient temperature |
| t_{ar} | = 6 h | Required time for air renewal |

Thermal loads

| | | |
|-------|----------|---------------------------------------|
| Q_L | = 8.2 kW | Input heat from the environment |
| Q_A | = 3.7 kW | Input heat due to air renewal |
| Q_F | = 5.6 kW | Input heat due to fan |
| Q_R | = 5.3 kW | Heat of respiration |
| Q_W | = 5.2 kW | Heat removal due to water evaporation |

Refrigerant evaporator

| | | |
|-------|----------------------|--|
| T_e | = -8 °C | Refrigerant evaporating temperature |
| Q_e | = 17.6 kW | Refrigerant evaporator thermal load |
| F_R | = 0.01 kg/s | Refrigerant flow rate |
| A_e | = 7.0 m ² | Heat transfer area of the refrigerant evaporator |

Refrigerant condenser

| | | |
|----------|----------------------|---|
| T_c | = 35 °C | Refrigerant condensing temperature |
| Q_c | = 20.6 kW | Refrigerant condenser thermal load |
| A_c | = 1.9 m ² | Heat transfer area of the refrigerant condenser |
| F_w | = 0.49 kg/s | Cooling water flow rate |
| T_{w1} | = 15 °C | Cooling water inlet temperature |
| T_{w2} | = 25 °C | Cooling water outlet temperature |

Refrigerant compressor

| | | |
|-------|----------|------------------------------|
| E_c | = 3.0 kW | Refrigerant compressor power |
|-------|----------|------------------------------|

Ventilation fan

| | | |
|------------|-----------|---|
| u | = 1 m/s | Superficial air velocity |
| ΔP | = 75 Pa | Pressure loss of air flowing through the room |
| F_a | = 75 kg/s | Flow rate of the circulated air |
| E_f | = 5.6 kW | Fan power |

Total electrical power

| | | |
|-----|----------|--------------------------|
| E | = 8.6 kW | Total power requirements |
|-----|----------|--------------------------|

Table 5.25 Cost Analysis Results**Equipment cost**

| | | |
|------------|------------------|----------------|
| C_{rm} | = 57.1 k\$ | Room |
| C_{con} | = 4.5 k\$ | Condenser |
| C_{evap} | = 10.7 k\$ | Evaporator |
| C_{fan} | = 3.7 k\$ | Fan |
| C_{com} | = <u>6.8</u> k\$ | Compressor |
| C_{eq} | = 82.8 k\$ | Equipment cost |

Operating cost

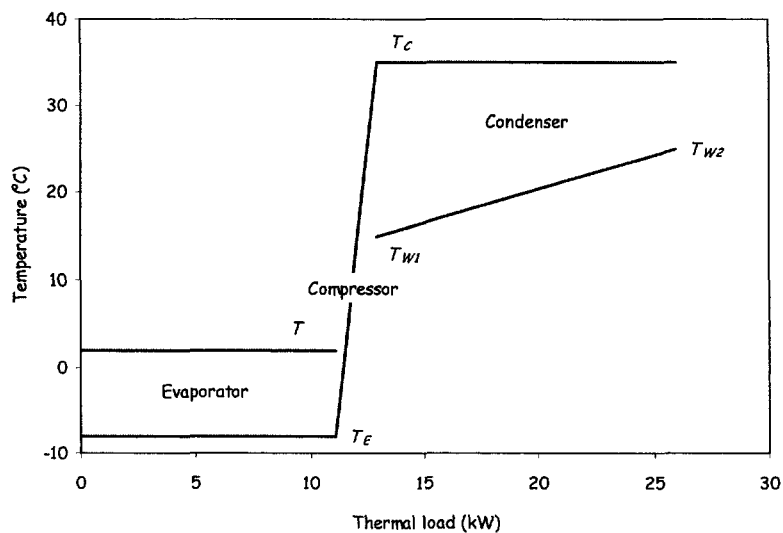
| | | |
|----------|---------------------|-----------------------|
| C_l | = 2.8 k\$/yr | Cost of land |
| C_e | = 3.2 k\$/yr | Cost of electricity |
| C_w | = <u>0.7</u> k\$/yr | Cost of cooling water |
| C_{op} | = 6.7 k\$/yr | Operating cost |

Objective function

| | | |
|-----------|---------------------|---------------------------|
| eC_{eq} | = 12.3 k\$/yr | Annualized equipment cost |
| C_{op} | = <u>6.7</u> k\$/yr | Operating cost |
| TAC | = 19.0 k\$/yr | Total annual cost |

Cost index

| | | |
|-----|---------------|------------------------|
| c | = 0.057 \$/kg | Cold storage unit cost |
|-----|---------------|------------------------|

**Figure 5.15** Temperature – Enthalpy diagram.

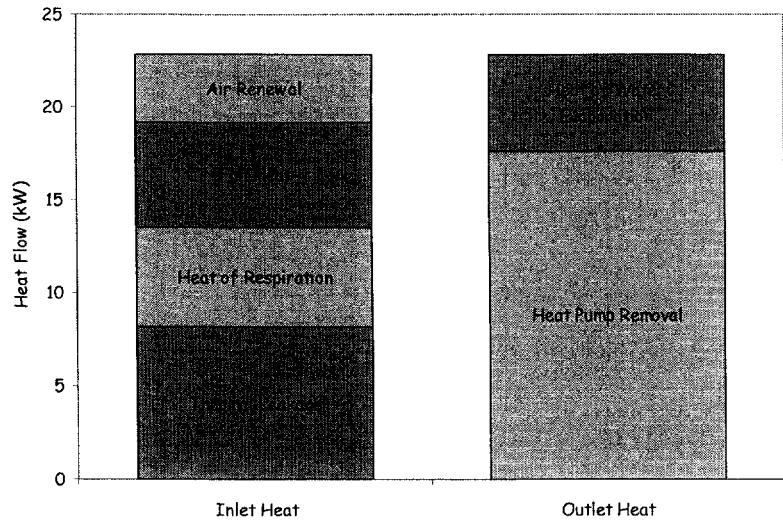


Figure 5.16 Cold storage room energy balance.

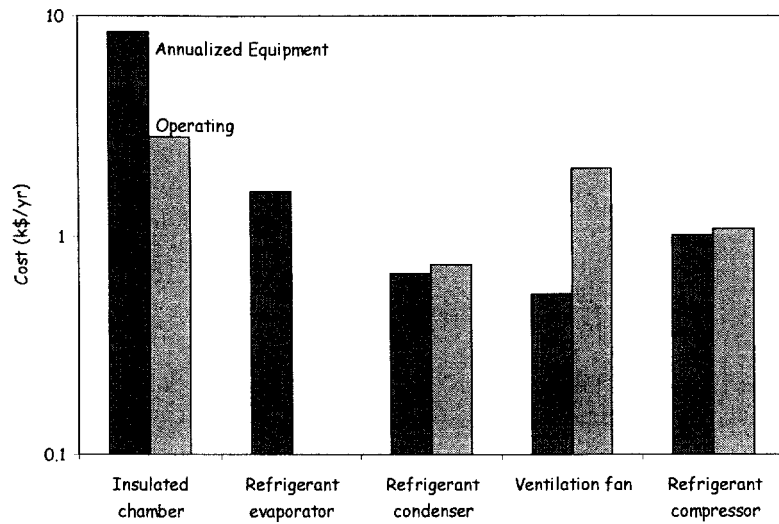


Figure 5.17 Equipment and operating cost.

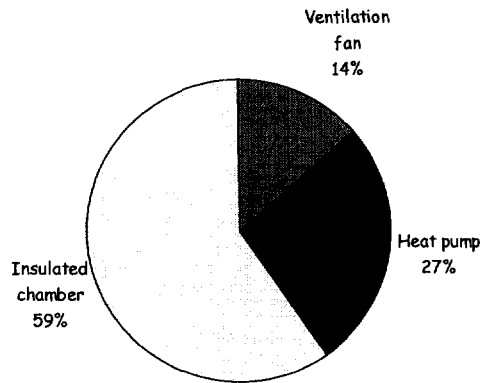


Figure 5.18 Cost breakdown.

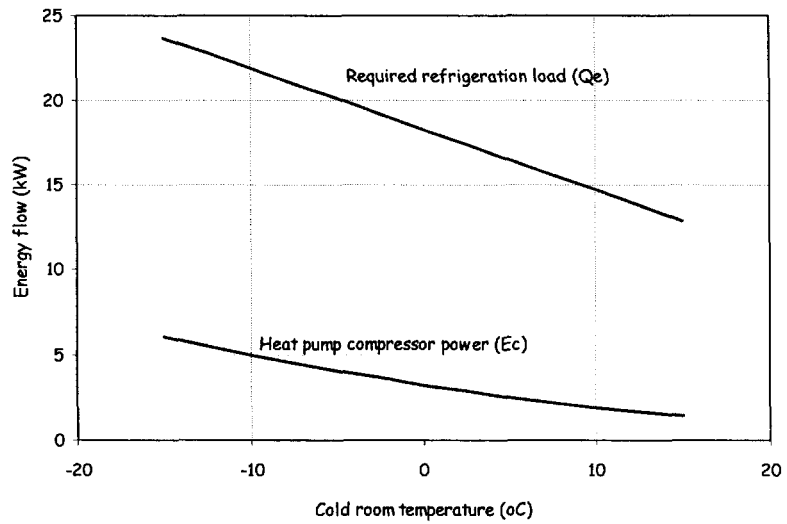


Figure 5.19 Effect of cold room temperature on the required refrigeration load and compressor power.

IX. DESIGN OF A FLUIDIZED BED FREEZER

A design approach to freezing of a particulate food is described in this section, using a fluidized bed system.

1. Process Description

A typical flowsheet of a fluidized bed freezer is presented in Figure 5.20. The feed at flow rate F (kg/s) and temperature T_1 ($^{\circ}\text{C}$) is distributed on a metallic screen as it enters the freezer. The frozen product exits the freezer at the same flow rate F (kg/s) and temperature T_2 ($^{\circ}\text{C}$). The fluidized particles move slowly due to gravity at a velocity u_b (m/s). The cooling air enters the freezer at a flow rate F_a (kg/s) and temperature T ($^{\circ}\text{C}$). An electrical power E_f (kW) is expended by the fan. The air conditions for design can be considered constant due to the high air recirculation.

The cooling air temperature is controlled in the evaporator of a heat pump. The refrigerant is evaporated at temperature T_e ($^{\circ}\text{C}$) in the evaporator of thermal load Q_e (kW), it is compressed in the compressor of power E_c (kW), then it is liquidified at temperature T_c ($^{\circ}\text{C}$) in the condenser of thermal load Q_c (kW) via cooling water, and finally it is expanded through a valve to the refrigerant evaporator.

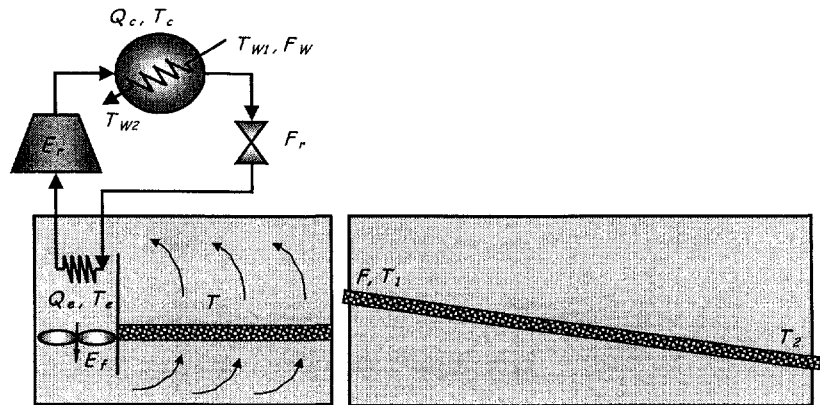


Figure 5.20 Schematic representation of a fluidized bed freezer.

2. Process Model

A mathematical model of the process presented in Figure 5.20 is summarized in Table 5.26.

Equation (E01) estimates the heat transfer coefficient (Saravacos and Maroulis, 2001), while Equation (E02) calculates the required freezing time (see the conventional Plank Equation 5-10).

Equations (E03) through (E05) constitute the refrigerant evaporator model. Equation (E03) calculates the required cooling load, which must be covered by the refrigerant evaporator. Equation (E04) calculates the heat which is absorbed by the refrigerant. Equation (E05) is the phenomenological equation for heat flow between the two streams in the refrigerant evaporator.

Similarly, Equations (E06) through (E08) constitute the refrigerant condenser model. Equation (E06) is the energy balance for the cooling water into the condenser, Equation (E07) the energy balance for the condensing refrigerant, while Equation (E08) describes the heat flow rate between these two streams.

Equation (E09) calculates the required power for the refrigerant compressor. It must be noted that Equations (E04), (E07) and (E09) come from the shortcut design procedure for refrigeration systems proposed by Shelton and Grossmann (1985).

Equations (E10) through (E15) are used for sizing the fluidized bed. Equation (E10) correlates the residence time with the mass holdup, while Equation (E11) relates the mass holdup with the volume hold up. Equation (E12) is the geometrical evaluation of the volume hold up on the bed. Equation (E13) calculates the required bed area, while Equation (E14) estimates the required bed velocity to obtain the desired residence time. Equation (E15) estimates the superficial moving velocity of the fluidized particles versus the bed tilt.

Equations (E16) through (E18) are used for sizing the fan. Equation (E16) calculates the friction pressure loss of air through the loaded bed. Equation (E17) correlates the air flow with the air velocity. Equation (E18) estimates the required electrical power to operate the fan.

Finally, Equation (E19) summarizes the required electrical power for the system.

Thirty-one variables presented in Table 5.27 are involved in the model of nineteen equations presented in Table 5.26. The corresponding technical data are summarized in Table 5.28. The process specifications of a typical design problem, a degrees-of-freedom analysis, a suggestion for the design variables selection and the corresponding solution algorithm are presented in Table 5.29. The total annualized cost *TAC* presented in Table 5.30 is used as objective function in process optimization. The required cost data are summarized in Table 5.31.

Table 5.26 Fluidized-Bed Freezer Model**Freezing Time**

$$\left(\frac{h}{u \rho_a C_{pa}} \right) = a \left(\frac{u \rho_a d}{\eta} \right)^n \quad (\text{E01})$$

$$t = \frac{\rho_s \Delta H_L}{T_f - T} \left(\frac{P d}{h} + \frac{R d^2}{\lambda} \right) \quad (\text{E02})$$

Refrigerant Evaporator

$$Q_e = F C_{ps} (T_f - T_d) + F \Delta H_L + F C_{ps} (T_f - T_2) \quad (\text{E03})$$

$$Q_e = [\Delta H_r - C_{pr} (T_c - T_d)] F_r \quad (\text{E04})$$

$$Q_e = A_e U_e (T - T_d) \quad (\text{E05})$$

Refrigerant Condenser

$$Q_c = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E06})$$

$$Q_c = [\Delta H_r (T_c / T_e) - C_{pr} (T_c - T_d)] F_r \quad (\text{E07})$$

$$Q_c = A_c U_c [(T_c - T_{w1}) - (T_c - T_{w2})] / \ln[(T_c - T_{w1}) / (T_c - T_{w2})] \quad (\text{E08})$$

Refrigerant Compressor

$$E_c = \Delta H_r [(T_c - T_d) / T_d] F_r \quad (\text{E09})$$

Bed

$$M = t F \quad (\text{E10})$$

$$M = (1 - \varepsilon) \rho_s H \quad (\text{E11})$$

$$H = Z D L \quad (\text{E12})$$

$$A_b = L D \quad (\text{E13})$$

$$u_b = L / t \quad (\text{E14})$$

$$u_b = b s \quad (\text{E15})$$

Fan

$$\Delta P = \frac{1 - \varepsilon}{\varepsilon^3} \left[150 \frac{(1 - \varepsilon)}{\text{Re}} + 1.75 \right] \frac{Z}{d} \rho_a u^2 \quad (\text{E16})$$

$$F_a = u D L \quad (\text{E17})$$

$$E_f = \Delta P F_a \quad (\text{E18})$$

Electrical energy requirements

$$E = E_b + E_f + E_c \quad (\text{E19})$$

Table 5.27 Process Variables

| | | |
|------------|-------------|--|
| A_b | m^2 | Bed area |
| A_c | m^2 | Heat transfer area of the refrigerant condenser |
| A_e | m^2 | Heat transfer area of the refrigerant evaporator |
| D | m | Bed width |
| d | m | Particle size |
| E | kW | Total power requirement |
| E_c | kW | Refrigerant compressor power |
| E_f | kW | Fan power |
| F | kg/s | Material flow rate |
| F_a | m^3/s | Flow rate of the circulated air |
| F_r | kg/s | Refrigerant flow rate |
| F_w | kg/s | Cooling water flow rate |
| H | m^3 | Freezer volume holdup |
| h | W/m^2K | Heat transfer coefficient |
| L | m | Bed length |
| M | ton | Freezer mass holdup |
| Q_c | kW | Refrigerant condenser thermal load |
| Q_e | kW | Refrigerant evaporator thermal load |
| s | kW | Bed tilt |
| t | h | Freezing time |
| T | $^{\circ}C$ | Cooling air temperature |
| T_1 | $^{\circ}C$ | Initial temperature of food |
| T_2 | $^{\circ}C$ | Final temperature of food |
| T_c | $^{\circ}C$ | Refrigerant condensing temperature |
| T_e | $^{\circ}C$ | Refrigerant evaporating temperature |
| T_{w1} | $^{\circ}C$ | Cooling water inlet temperature |
| T_{w2} | $^{\circ}C$ | Cooling water outlet temperature |
| u | m/s | Cooling air velocity |
| u_b | m/s | Bed velocity |
| Z | m | Loading depth |
| ΔP | bar | Pressure loss of air flowing through bed |

Table 5.28 Process Data

Density

| | | |
|----------|-------------------|----------|
| ρ_a | kg/m ³ | Air |
| ρ_s | kg/m ³ | Material |

Specific heat

| | | |
|----------|---------|-------------|
| C_{pw} | kJ/kg K | Water |
| C_{pr} | kJ/kg K | Refrigerant |
| C_{pa} | kJ/kg K | Air |
| C_{ps} | kJ/kg K | Food |

Thermal conductivity

| | | |
|-----------|-------|-----|
| λ | W/m K | Ice |
|-----------|-------|-----|

Viscosity

| | | |
|--------|------|-----|
| η | Pa s | Air |
|--------|------|-----|

Latent heat of evaporation

| | | |
|--------------|-------|-------------|
| ΔH_w | kJ/kg | Water |
| ΔH_r | kJ/kg | Refrigerant |

Ice crystallization

| | | |
|--------------|-------|-------------|
| ΔH_L | kJ/kg | Latent heat |
| T_f | °C | Temperature |

Heat transfer coefficients

| | | |
|-------|---------------------|---|
| U_e | kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_c | kW/m ² K | Heat transfer coefficient at the refrigerant condenser |

Empirical constants

| | | |
|---------------|---|----------------------------|
| b | - | Particle velocity equation |
| f | - | Pressure loss equation |
| ε | - | Void (empty) fraction |
| a | - | Heat transfer equation |
| n | - | Heat transfer equation |
| P | - | Shape factor |
| R | - | Shape factor |

Table 5.29 Process Solution**Process Specifications**

| | | |
|----------|-------|----------------------------------|
| F | ton/h | Material flow rate |
| T_1 | °C | Initial temperature of food |
| T_2 | °C | Final temperature of food |
| d | m | Food particle size |
| T_{w1} | °C | Cooling water inlet temperature |
| T_{w2} | °C | Cooling water outlet temperature |

Degrees-of-Freedom Analysis

| | | | |
|--------------------|-----------|--------------------|----------|
| Process variables | 31 | Degrees of freedom | 12 |
| Process equations | <u>19</u> | Specifications | <u>6</u> |
| Degrees of freedom | 12 | Design variables | 6 |

Design Variables

| | | |
|-------|-----|-------------------------------------|
| D | m | Bed width |
| T_c | °C | Refrigerant condensing temperature |
| T_e | °C | Refrigerant evaporating temperature |
| T | °C | Cooling air temperature |
| u | m/s | Cooling air velocity |
| Z | m | Loading depth |

Model Solution Algorithm

| | | |
|-------|---|------------|
| (E01) | → | h |
| (E02) | → | t |
| (E03) | → | Q_e |
| (E04) | → | F_r |
| (E05) | → | A_e |
| (E07) | → | Q_c |
| (E06) | → | F_w |
| (E08) | → | A_c |
| (E09) | → | E_c |
| (E10) | → | M |
| (E11) | → | H |
| (E12) | → | L |
| (E13) | → | A_b |
| (E14) | → | u_b |
| (E15) | → | s |
| (E16) | → | ΔP |
| (E17) | → | F_a |
| (E18) | → | E_f |
| (E19) | → | E |

Table 5.30 Cost Analysis**Equipment cost**

$$C_{eq} = C_{bed} A^{n_{bed}} + C_{exc} A_e^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{fan} E_f^{n_{fan}} + C_{com} E_c^{n_{com}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q_c + C_e (E_c + E_f)) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (F04)$$

Cost index

$$c = TAC / (F t_y) \quad (F05)$$

Table 5.31 Cost Data**Utility cost**

C_e \$/kWh Cost of electricity

C_w \$/kWh Cost of cooling water

Equipment unit cost

C_{bed} \$/m² Bed dryer

C_{exc} \$/m² Heat exchanger

C_{fan} \$/kW Fan

C_{com} \$/kW Compressor

Equipment size scaling factor

n_{bed} - Bed freezer

n_{exc} - Heat exchanger

n_{fan} - Fan

n_{com} - Compressor

Other

t_y h/yr Annual operating time

i_r - Interest rate

l_f yr Lifetime

3. Application to Green Pea Freezing

A typical application to green peas freezing is described in this section. The process specifications and the required technical and economic data are summarized in Tables 5.30 through 5.33. The optimal values of the design variables are presented in Table 5.34, while the corresponding technical and economic results are summarized in Tables 5.35 and 5.36. Some of these results are also presented in Figures 5.21 through 5.23.

Table 5.32 Technical and Cost Data**Density**

| | | |
|----------|--------------------------|------|
| ρ_a | = 1.00 kg/m ³ | Air |
| ρ_s | = 1000 kg/m ³ | Peas |

Specific heat

| | | |
|----------|----------------|-----------------------------|
| C_{pw} | = 4.20 kJ/kg K | Water |
| C_{pr} | = 1.20 kJ/kg K | Refrigerant R-717 (Ammonia) |
| C_{pa} | = 1.00 kJ/kg K | Air |
| C_{ps} | = 4.00 kJ/kg K | Peas |

Thermal conductivity

| | | |
|-----------|-------------|-----|
| λ | = 1.5 W/m K | Ice |
|-----------|-------------|-----|

Viscosity

| | | |
|--------|---------------|-----|
| η | = 0.017 mPa s | Air |
|--------|---------------|-----|

Ice crystallization

| | | |
|--------------|-------------|-------------|
| ΔH_L | = 334 kJ/kg | Latent heat |
| T_f | = -1.5°C | Temperature |

Latent heat of evaporation

| | | |
|--------------|--------------|-----------------------------|
| ΔH_w | = 2500 kJ/kg | Water |
| ΔH_r | = 1250 kJ/kg | Refrigerant R-717 (Ammonia) |

Heat transfer coefficients

| | | |
|-------|----------------------------|---|
| U_e | = 0.25 kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_c | = 0.75 kW/m ² K | Heat transfer coefficient at the refrigerant condenser |

Empirical constants

| | | |
|---------------|----------|----------------------------|
| b | = 10 | Particle velocity equation |
| ε | = 0.50 | Void (empty) fraction |
| a | = 0.143 | Heat transfer equation |
| n | = -0.455 | Heat transfer equation |
| P | = 0.0625 | Shape factor |
| R | = 0.0417 | Shape factor |

Utility cost

| | | |
|-------|---------------|-----------------------|
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_w | = 0.01 \$/kWh | Cost of cooling water |

Equipment unit cost

| | | |
|-----------|-------------------------|----------------|
| C_{bed} | = 10 k\$/m ² | Bed freezer |
| C_{exc} | = 3 k\$/m ² | Heat exchanger |
| C_{fan} | = 1 k\$/kW | Fan |
| C_{com} | = 3 k\$/kW | Compressor |

Equipment size scaling factor

| | | |
|-----------|--------|----------------|
| n_{bed} | = 0.95 | Bed freezer |
| n_{exc} | = 0.65 | Heat exchanger |
| n_{fan} | = 0.75 | Fan |
| n_{com} | = 0.75 | Compressor |

Other

| | | |
|-------|------------|-----------------------|
| t_y | = 720 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 5.33 Process Specifications

| | | |
|----------|--------------|----------------------------------|
| F | = 0.50 ton/h | Material flow rate |
| T_1 | = 10 °C | Initial temperature of food |
| T_2 | = -18°C | Final temperature of food |
| d | = 8 mm | Particle size |
| T_{w1} | = 15 °C | Cooling water inlet temperature |
| T_{w2} | = 25 °C | Cooling water outlet temperature |

Table 5.34 Design Variables

| | | |
|-------|----------|-------------------------------------|
| D | = 1 m | Bed width |
| T_c | = 35 °C | Refrigerant condensing temperature |
| T_e | = -35 °C | Refrigerant evaporating temperature |
| T | = -25 °C | Cooling air temperature |
| u | = 5 m/s | Cooling air velocity |
| Z | = 10 cm | Loading depth |

Table 5.35 Cost Analysis Results**Equipment cost**

| | | |
|------------|-------------------|----------------|
| C_{bed} | = 13.3 k\$ | Bed |
| C_{con} | = 11.1 k\$ | Condenser |
| C_{evap} | = 24.2 k\$ | Evaporator |
| C_{fan} | = 8.8 k\$ | Fan |
| C_{com} | = <u>27.9</u> k\$ | Compressor |
| C_{eq} | = 85.3 k\$ | Equipment cost |

Operating cost

| | | |
|----------|---------------------|-----------------------|
| C_e | = 2.7 k\$/yr | Cost of electricity |
| C_w | = <u>0.6</u> k\$/yr | Cost of cooling water |
| C_{op} | = 3.3 k\$/yr | Operating cost |

Objective function

| | | |
|-----------|---------------------|---------------------------|
| eC_{eq} | = 12.7 k\$/yr | Annualized equipment cost |
| C_{op} | = <u>3.3</u> k\$/yr | Operating cost |
| TAC | = 16.0 k\$/yr | Total annual cost |

Cost index

| | | |
|-----|---------------|--------------------|
| c | = 0.044 \$/kg | Freezing unit cost |
|-----|---------------|--------------------|

Table 5.36 Process design results**Bed geometry**

| | | |
|-------|-----------------------|-----------------------|
| A_b | = 1.35 m ² | Bed area |
| D | = 1 m | Bed width |
| L | = 1.35 m | Bed length |
| Z | = 0.10 m | Loading depth |
| H | = 0.14 m ³ | Freezer volume holdup |
| d | = 8 mm | Particle size |
| M | = 0.07 ton | Freezer mass holdup |

Bed operation

| | | |
|-------|---------------------------|-----------------------------|
| F | = 0.5 ton/h | Material flow rate |
| T_1 | = 10 °C | Initial temperature of food |
| T_2 | = -18 °C | Final temperature of food |
| T | = -25 °C | Cooling air temperature |
| h | = 20.9 W/m ² K | Heat transfer coefficient |
| t | = 0.14 h | Freezing time |
| u_b | = 10 m/h | Bed velocity |
| s | = 2.8 % | Bed tilt |

Refrigerant evaporator

| | | |
|-------|-----------------------|--|
| T_e | = -35 °C | Refrigerant evaporating temperature |
| Q_e | = 61.9 kW | Refrigerant evaporator thermal load |
| F_r | = 0.05 kg/s | Refrigerant flow rate |
| A_e | = 24.8 m ² | Heat transfer area of the refrigerant evaporator |

Refrigerant condenser

| | | |
|----------|----------------------|---|
| T_c | = 35 °C | Refrigerant condensing temperature |
| Q_c | = 81.5 kW | Refrigerant condenser thermal load |
| A_c | = 7.5 m ² | Heat transfer area of the refrigerant condenser |
| F_w | = 1.94 kg/s | Cooling water flow rate |
| T_{w1} | = 15 °C | Cooling water inlet temperature |
| T_{w2} | = 25 °C | Cooling water outlet temperature |

Refrigerant compressor

| | | |
|-------|-----------|------------------------------|
| E_c | = 19.5 kW | Refrigerant compressor power |
|-------|-----------|------------------------------|

Fan

| | | |
|------------|-------------------------|--|
| u | = 5 m/s | Cooling air velocity |
| ΔP | = 2.67 kPa | Pressure loss of air flowing through bed |
| F_a | = 6.8 m ³ /s | Flow rate of the circulated air |
| E_f | = 18.1 kW | Fan power |
| E | = 37.6 kW | Total power requirement |

Figure 5.21 represents an enthalpy-temperature diagram for the system. It shows the temperature differences and the corresponding thermal loads for each process.

Figure 5.22 distributes the equipment and operating cost into the different processes, while Figure 5.23 represents the total cost distribution to the main processes.

It must be noted that the freezing time was calculated assuming a uniform spherical size of the peas (8 mm). The calculations should be modified for a non uniform size distribution of the peas.

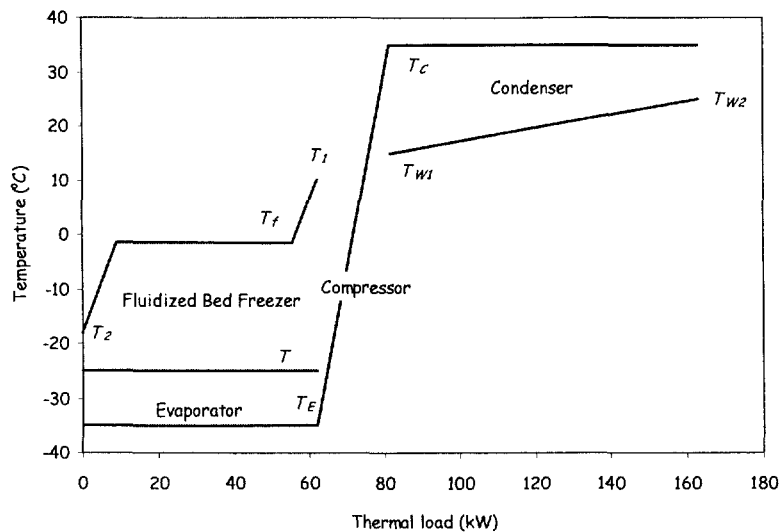


Figure 5.21 Enthalpy-temperature diagram.

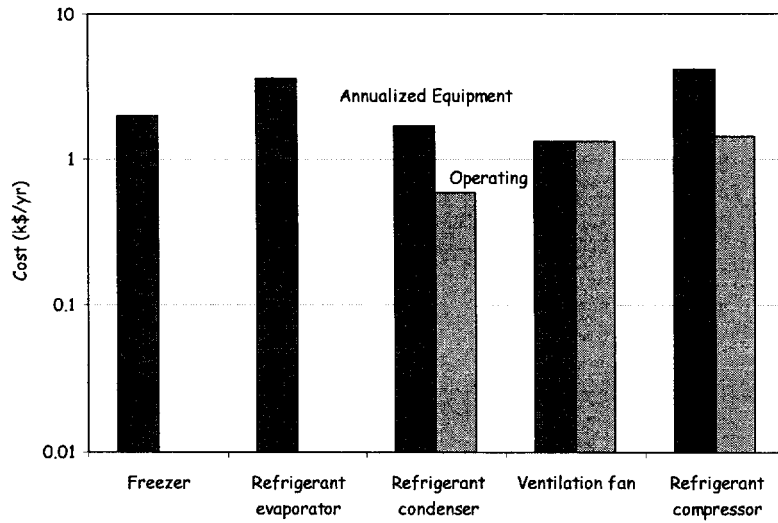


Figure 5.22 Equipment and operating cost.

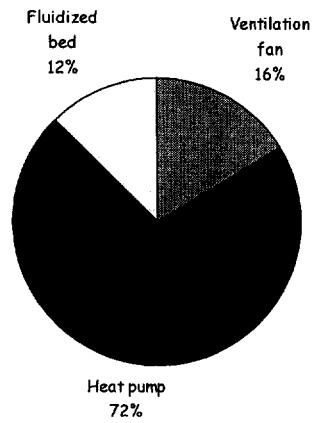


Figure 5.23 Cost breakdown.

NOMENCLATURE

| | | |
|-----------|-------------------|--|
| A_b | m^2 | Belt area |
| A_c | m^2 | Heat transfer area of the refrigerant condenser |
| A_e | m^2 | Heat transfer area of the refrigerant evaporator |
| A_L | m^2 | Heat loss area of the cold storage room |
| b | - | Belt drive power equation |
| C_{bet} | $\$/m^2$ | Belt freezer |
| C_{com} | $\$/kW$ | Compressor |
| C_e | $\$/kWh$ | Cost of electricity |
| C_{exc} | $\$/m^2$ | Heat exchanger |
| C_{fan} | $\$/kW$ | Fan |
| C_l | $\$/m^2\text{yr}$ | Cost of land |
| C_{pa} | kJ/kg K | Air |
| C_{pr} | kJ/kg K | Refrigerant |
| C_{ps} | kJ/kg K | Food |
| C_{pw} | kJ/kg K | Water |
| C_{rm} | $\$/m^2$ | Room |
| C_w | $\$/kWh$ | Cost of cooling water |
| D | m | Belt width |
| D | m | Cold room width |
| d | m | Particle size |
| E | kW | Total power requirements |
| E_b | kW | Belt driver power |
| E_c | kW | Refrigerant compressor power |
| E_f | kW | Fan power |
| f | - | Pressure loss equation |
| F | kg/s | Material flow rate |
| F_a | m^3/s | Flow rate of the circulated air |
| F_a | ton/h | Flow rate of the circulated air |
| F_r | kg/s | Refrigerant flow rate |
| F_r | ton/h | Refrigerant flow rate |
| F_w | kg/s | Cooling water flow rate |
| F_w | ton/h | Cooling water flow rate |
| H | m^3 | Freezer volume holdup |
| i_r | - | Interest rate |
| L | m | Belt length |
| L | m | Cold room length |
| l_f | yr | Lifetime |
| M | ton | Freezer mass holdup |
| M | ton | Mass of the stored material |
| n_{bet} | - | Belt freezer |
| n_{com} | - | Compressor |
| n_{exc} | - | Heat exchanger |
| n_{fan} | - | Fan |
| n_{rm} | - | Room |
| Q_A | kW | Input heat due to air renewal |
| Q_c | kW | Refrigerant condenser thermal load |

| | | |
|---------------|---------------------|---|
| Q_e | kW | Refrigerant evaporator thermal load |
| Q_F | kW | Input heat due to fan |
| Q_L | kW | Input heat from the environment |
| Q_R | kW | Heat of respiration |
| Q_W | kW | Heat removal due to water evaporation |
| r | W/ton | Heat of respiration |
| t | h | Freezing time |
| T | °C | Cold room temperature |
| T | °C | Cooling air temperature |
| T_1 | °C | Initial temperature of food |
| T_2 | °C | Final temperature of food |
| T_a | °C | Ambient temperature |
| t_{ar} | h | Required time for air renewal |
| t_c | h | Freezing time constant at freezing conditions |
| T_c | °C | Refrigerant condensing temperature |
| T_e | °C | Refrigerant evaporating temperature |
| T_{w1} | °C | Cooling water inlet temperature |
| T_{w2} | °C | Cooling water outlet temperature |
| t_y | h/yr | Annual operating time |
| u | m/s | Cooling air velocity |
| u | m/s | Superficial air velocity |
| u_b | m/s | Belt velocity |
| U_c | kW/m ² K | Heat transfer coefficient at the refrigerant condenser |
| U_C | kW/m ² K | Heat transfer coefficient at the refrigerant condenser |
| U_e | kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_E | kW/m ² K | Heat transfer coefficient at the refrigerant evaporator |
| U_L | kW/m ² K | Overall heat loss coefficient through the wall |
| V | m ³ | Cold room volume |
| w | kg/ton day | Water loss rate |
| Z | m | Cold room height |
| Z | m | Loading depth |
| ΔH_r | kJ/kg | Refrigerant |
| ΔH_w | kJ/kg | Water |
| ΔP | bar | Pressure loss of air flowing through belt |
| ΔP | bar | Pressure loss of air flowing through the room |
| α | m ² /s | Food |
| ε | - | Packing void (empty) fraction |
| ε | - | Void (empty) fraction |
| ρ_a | kg/m ³ | Air density |
| ρ_s | kg/m ³ | Material density |

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6

Evaporation

I. INTRODUCTION

Thermal evaporation is used in food processing to remove most of the water of liquid foods, such as fruit and vegetable juices, milk, and coffee extracts, resulting in a concentrated product, which may be used as such or processed further, e.g., by drying. Evaporation is also used in the refining of sugar and table salt, and in concentrating liquid wastes from processing plants.

Concentration of liquid foods can be accomplished by newer non-thermal processes, such as reverse osmosis (Chapter 10) and freeze-concentration (Saravacos and Kostaropoulos, 2002), which are not cost-effective at the present time.

The engineering design of evaporators is based on the efficient transfer of heat from the heating medium (usually steam) to the liquid product, the effective vapor/liquid separation, and the utilization of energy (Minton 1986). Technical data are provided by manufacturers of evaporation equipment (Alfa-Laval, 1972; APV, 1987). Thermal energy (steam) constitutes the major operating cost, and various energy-saving systems have been devised and applied to modern evaporation plants.

Evaporation should not damage the quality of heat-sensitive food products, and the evaporation equipment should conform to the hygienic requirements of cleaning and good manufacturing practices (Chapters 1 and 2).

II. HEAT TRANSFER IN EVAPORATORS

1. Introduction

Material and energy balances around the evaporator determine the mass and heat flows in the system.

Heat transfer at the evaporation surface is related directly to the thermo-physical and transport properties, and the flow pattern of the liquid. Physical prop-

erties of the food liquids of importance to evaporation, are the density, viscosity, thermal conductivity, specific heat, and boiling point elevation (Chen, 1993; Rahman, 1995). The transport properties of foods are discussed by Saravacos and Maroulis (2001). The surface tension of water at 25°C is 73 dyn/cm or 73 mJ/m² (Reid et al., 1987). The surface tension of liquid foods is lower, e.g., 50 dyn/cm, due to the food surface-active components.

Heat transfer at the wall/liquid interface is the most important transfer operation in evaporation, since the thermal resistances of the wall (stainless steel) and the heating medium (saturated steam) are generally smaller. For example, heating medium (steam)-side heat transfer coefficients $h_s=10$ kW/m²K can be obtained on clean surfaces. The thermal resistance of the tube wall in an evaporator depends on the wall thickness x and its thermal conductivity λ ; For a 1½ inch (38mm) stainless steel tube, $x = 3$ mm, and $\lambda=15$ W/m K; therefore $x/\lambda = 0.003/15 = 0.0002$ m²K/W, and $h_w = 1/0.0002 = 5$ kW/m²K. The heat transfer coefficient of liquid foods h varies widely, depending mainly on its apparent viscosity, e.g., 5kW/m²K (clarified juice) and 0.5 kW/m²K (tomato paste).

The boiling point elevation *BPE*, caused by solute/water interaction, is undesirable in evaporation, since it requires a higher temperature of the heating medium to affect the same driving force (temperature difference). It is particularly high (e.g., 30°C) in concentrated aqueous solutions of salts, but it is relatively low in liquid foods, and in most cases it can be neglected in heat transfer calculations. The *BPE* becomes important in concentrated solutions of sugars and other low molecular components, e.g., fruit juices. High molecular components dissolved or dispersed in water, such as starch, pectins, and proteins give negligible *BPE*.

For sugar solutions, such as fruit juices, the following empirical equation can be used to estimate *BPE* (Chen and Hernandez, 1997):

$$BPE = 0.33 \exp(4X) \quad (6-1)$$

where X is the mass fraction of the sugar. Thus, the *BPE* of a fruit juice will increase, during evaporation, from about 0.7°C (20°Brix) to 4.4°C (65°Brix).

2. Heat Transfer Coefficients

The heat transfer Q (W) at the heating surface is given by the general equation:

$$Q = U A \Delta T \quad (6-2)$$

where U is the overall heat transfer coefficient (W/m²K), A is the heating surface (m²) and ΔT is the temperature difference between the heating medium (steam, vapors) and the boiling medium (°C or K). The overall heat transfer coefficient U is usually determined experimentally or is taken from operating similar industrial or pilot plant evaporators. Theoretical prediction of U is difficult because of the fouling resistance at the heating surface, which cannot be quantified accurately. However, heat transfer analysis is useful in evaluating the thermal resistances of evaporation systems.

The overall thermal resistance of a heating system for evaporation is given by the equation:

$$1/U = 1/h_s + x/\lambda + 1/h + FR \quad (6-3)$$

where h_s and h are, respectively, the heat transfer coefficients at the heating (steam) and evaporation sides, x/λ is the thermal resistance of the evaporator wall, and FR is the fouling resistance. Equation (6-3) refers to plane heat transfer surfaces, and it can be applied approximately to tubes of relatively large diameter, e.g., 50 mm (2 inches). For small-diameter tubes the thermal resistances must be corrected by the ratio of outside to inside diameters.

The evaporation heat transfer coefficient h depends strongly on the apparent viscosity of the liquid foods. Most food suspensions and juices are pseudoplastic materials (Saravacos and Maroulis, 2001), and their apparent viscosity decreases substantially at high shear rates (high velocities or agitation rates), resulting in improved evaporation rates (Saravacos, 1974). Temperature has a strong negative effect on the viscosity of sugar solutions and clarified juices, but a rather small effect on fruit suspensions or pulps (Saravacos, 1970).

Fouling in food evaporators, and other heat transfer equipment, is caused mainly by the adsorption and denaturation of food biomolecules, such as proteins, pectins and starch, on the heated surface.

The use of fouling resistances FR or fouling factors in Equation (6-3) gives only approximate values for a specific evaporator and product. Practically, it is more accurate to use reliable values of the overall heat transfer coefficient U obtained under appropriate conditions. Typical values of U for food evaporators are shown in Table 6.1 (Saravacos and Kostaropoulos, 2002).

Empirical correlations of fouling resistance to the operating time of food evaporators for a specific application (e.g., sugar evaporators) are useful for determining the optimum cleaning cycle, i.e., how often will the evaporator be cleaned by interrupting its operation.

Table 6.1 Typical Values of Overall Heat Transfer Coefficients (U)

| Type of evaporator | Liquid food | U , kW/m ² K |
|-----------------------|----------------------------|---------------------------|
| Falling film, tubular | Fruit juices 12 - 65 °Brix | 2.00 - 0.60 |
| Falling film, plate | Milk 10 - 30 % TS | 2.50 - 1.50 |
| Rising film, tubular | Milk 10 - 35 % TS | 2.00 - 1.20 |
| Forced circulation | Sugar syrups 15 - 65 °Brix | 2.50 - 1.50 |
| Agitated film | Fruit / vegetable pulp | 1.50 - 0.70 |

Data from Saravacos and Kostaropoulos, 2002

III. FOOD EVAPORATORS

Various types of evaporation equipment are described by Minton (1986), and Saravacos and Kostaropoulos (2002). An evaporation system consists of a group of essential components, which must be designed for coordinated effective and economic operation (Hahn, 1986).

In modern food processing, two basic types are used widely, i.e., the falling (or rising) film, and the forced circulation evaporators. Film evaporators are relatively simpler and less expensive than the forced circulation units, and they have shorter residence times (Moore and Hessler, 1963; Moore and Pinkel, 1968). The falling film evaporators are less expensive than the rising film units. Film evaporators are usually long tube vertical (*LTV*) units with tube diameters 30-50 mm and lengths of about 10 m (outdoor installation). Shorter, plate film evaporators are used for indoor installation. They are similar to the plate heat exchangers, discussed in Chapter 4.

The forced circulation evaporators are used for food suspensions and pulps, which would severely foul the film units. For highly fouling foods, expensive agitated film evaporators may be used.

1. Falling Film Evaporators

In falling film evaporators (see Figure 6.1), the liquid film falls by gravity in the vertical evaporation surface (inside the tube or plate), while heat is transferred through the wall by condensing steam. The mixture of liquid/vapors (*L/V*) from the bottom of the tube or plate enters a vapor/liquid separator, from which concentrated liquid is pumped out and water vapors are directed to a condenser (Saravacos and Kostaropoulos, 2002).

The Reynolds number of the falling film is given by the simplified equation (Perry and Green, 1997):

$$Re = 4 \Gamma / \eta \quad (6-4)$$

where Γ is the liquid flowrate per unit length (irrigation rate) and η is the viscosity of the liquid.

For water at 80°C at the minimum flow rate of $\Gamma = 0.08$ kg/m s, the Reynolds number will be $Re = 4 \times 0.08 / 0.000356 = 900$, i.e., the flow is laminar.

Higher heat transfer coefficients h are obtained in the turbulent flow regime, i.e., at $Re > 2100$. The following empirical equation can be used to estimate the heat transfer coefficient of water films in the turbulent flow regime:

$$h = 9150 \Gamma^{1/3} \quad (6-5)$$

The heat transfer coefficient of falling liquid films in turbulent flow ($Re > 1000$) is given by the empirical equation:

$$h = 0.01(4\Gamma/\eta)^{1/3}(C_p\eta/\lambda)^{1/3}(\lambda^3\rho^2g/\eta^2)^{1/3} \quad (6-6)$$

where ρ is the liquid density, λ is the thermal conductivity, and $g = 9.81$ m/s².

The simplified equation for water films (6-5) yields the following heat transfer coefficient, for $\Gamma = 0.5$ kg/m s (turbulent flow), $h_f = 9150 \times 0.5^{1/3} = 7.3$ kW/m²K. Assuming heat transfer coefficients of the heating steam $h_s = 10$ kW/m²K, and metal (stainless steel) wall $h_w = 5$ kW/m²K, it follows that (Equation 6-3, clean surfaces) $U = 2.3$ kW/m²K, which is very close to the U values used in practice (Table 6.1).

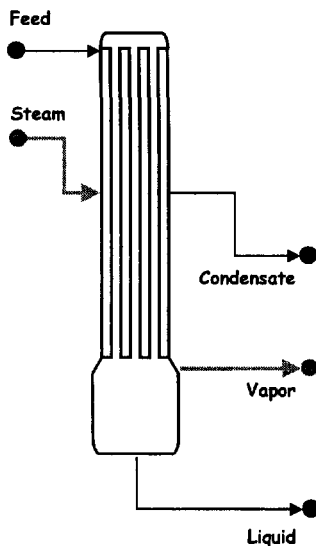


Figure 6.1 Diagram of a long tube vertical falling film evaporator.

The rising (climbing) film evaporators find fewer applications than the falling film systems, because of the longer residence time and higher operating temperatures, which may be detrimental to the quality of heat-sensitive food liquids, like fruit juices. However, the rising film systems do not require special feed distributors, and they do not foul as severely the heat transfer surfaces as the falling film units.

Falling film evaporators are used extensively for the concentration of fruit juices and experimental data of heat transfer coefficients are useful for design and evaluation of the industrial units, such as the data reported by Saravacos et al., (1970) for apple and grape juices. The overall heat transfer coefficient of clarified Concord grape juice increased from about 1.3 to 2 kW/m²K, as the boiling temperature was raised from 20 to 100°C. Clarified juices are Newtonian fluids and the activation energy for flow increases sharply at higher concentrations, e.g., 50 kJ/mol at 60°Brix (Saravacos and Maroulis, 2001).

The U decreases significantly as the juice is concentrated. Thus, the U decreases from about 1.9 to 1.2 W/m²K as the % soluble solids is increased from 20 to 65 °Brix. Lower U values were obtained in the evaporation of unfiltered grape juice, e.g., 1.35 to 0.65 kW/m²K due to fouling at the evaporator surface with particles and organic components (tartrates), which precipitate at high concentrations. Similar U values were obtained with filtered (depectinized) apple juice (2.0 to 1.15 kW/m²K for 10 to 65 °Brix), and cloudy (unfiltered) apple juice (1.48 to 0.74 kW/m²K for 10 to 60 °Brix).

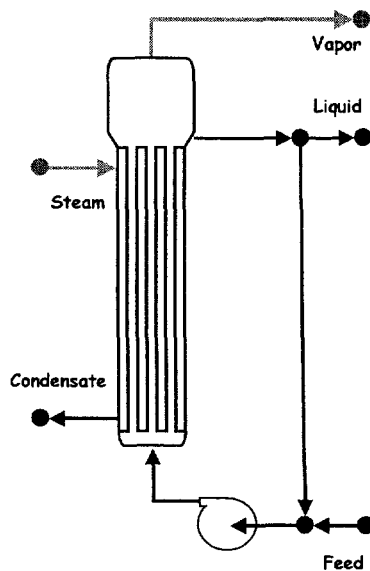


Figure 6.2 Diagram of forced circulation evaporator.

2. Forced Circulation Evaporators

A forced circulation evaporator is shown schematically in Figure 6.2. The liquid is recirculated by a centrifugal pump through an external (or internal) heat exchanger at high velocity (3-5 m/s), heated by condensing steam (or water vapors). The liquid does not boil within the heat exchanger tubes, due to high pressure drop, and it is flashed into the liquid/vapor separator, operated under vacuum. High heat transfer coefficients are obtained, due to high liquid velocities, and fouling of the tubes is minimized.

The heater (heat exchanger) consists of several tubes of relatively small diameter (e.g., 25 mm) and medium length (e.g., 3-5 m). The residence time can be several minutes, and heat-resistant food liquids, such as sugar solutions and tomato juice and pulp are evaporated effectively.

3. Agitated Film Evaporators

Agitated film evaporators are used in concentrating very viscous and fouling liquid foods, or suspensions of particulates, which cannot be handled in normal tubular or plate evaporators. The main evaporator body (vertical or horizontal) contains a rotor (scraping or hinged), which agitates the viscous fluid, improving heat transfer and preventing fouling (Figure 6.3). In the vertical units, the vapor/liquid separator and the motor can be placed either at the top or at the bottom of the unit. The evaporation surface of agitated film evaporators is limited, e.g., up to 10 m², due to mechanical limitation (a single tube with an agitator). The overall heat transfer coefficient U of an agitated film evaporator is generally high, depending on the rotational speed of the blades, the feed rate, and the concentration (°Brix) of the liquid product.

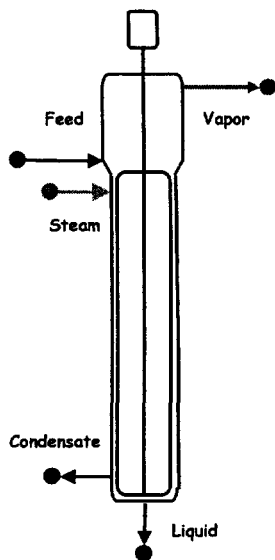


Figure 6.3 Diagram of agitated film evaporator with top vapor/liquid separator.

High overall heat transfer coefficients U of 2.00 to 3.00 kW/m²K are obtained with very viscous fluids (Bhatia, 1983). U values of 2.70 to 2.10 kW/m²K were obtained in the concentration of sucrose solutions at 100°C from 10 to 60°Brix, using a pilot agitated film evaporator (Marinos-Kouris and Saravacos, 1974).

Other types of food evaporators and evaporator components, such as vapor/liquid separators, condensers, and vacuum systems are discussed by Minton (1986), and Saravacos and Kostaropoulos (2002).

Testing and evaluation of evaporators can be performed using a procedure developed by the American Institute of Chemical Engineers (AIChE, 1979).

IV. ENERGY-SAVING EVAPORATORS

Evaporation is an energy-intensive unit operation of food processing, and energy saving and utilization systems have been developed and applied to overcome the increasing energy costs (ERDA, 1977). In addition to evaporation, concentration of liquid foods can be achieved by other methods, requiring less energy (Schwartzberg, 1977).

The energy required for evaporation, expressed in the form of saturated steam, is used mainly to vaporize the water from the liquid food material. Theoretically, the evaporation of 1 kg of water requires slightly more than 1 kg of saturated steam, used as the heating medium, since the enthalpy (heat) of vaporization of water decreases as the pressure is increased. For example, evaporation of water at 100°C requires 2.26 MJ/kg and if saturated steam of 2 bar (absolute) pressure is used as the heating medium, the energy given up by its condensation will be 2.20

MJ/kg (Haar et al., 1984). Thus, the steam economy (SE) in this operation will be $SE=2.20/2.26=0.97$ kg water evaporated/kg steam (single effect operation). In this example, the liquid is assumed to enter and leave the evaporation unit at the boiling point.

The steam economy of evaporators can be increased substantially, using multiple effect and vapor recompression (Saravacos and Kostaropoulos, 2002). Table 6.2 shows typical steam economies of industrial evaporators (Kessler, 1986; Hartel, 1992; Chen and Hernandez, 1997).

In comparing the energy requirements of evaporator systems, the total energy use should be considered, including steam and electrical power. As a common basis the primary energy (i.e., fuel consumption) can be used. For this purpose the thermal energy (steam) is multiplied by 1.2, while a conversion factor of 3 is used for the electrical energy (Kessler, 1986). Examples of energy requirements, based on the single effect evaporator (100%) are 3-effect (40%), thermal recompression/3-effect (25%), and mechanical compression/3-effect (8%).

1. Multiple-Effect Evaporators

The multiple-effect evaporation system is based on the repeated use of the water vapors from one evaporation unit (effect) to heat the next effect, which operates at a lower pressure. Thus, 1 kg of steam can evaporate more water, depending on the number of effects and the operating pressures.

For heat-sensitive liquid foods, the temperature in the first effect should not be higher than 100°C, while the temperature in the last effect should not be lower than about 50°C, in order to use cooling water at ambient temperature in the condenser of the last vapors. The first effect is usually heated with low-pressure steam, e.g., at 120°C (2 bar). Assuming a temperature difference (ΔT) of 10°C in each effect, the maximum number of effects in a food evaporation system would be about $N=70/10=7$. In practice, up to 5 effects are used in food evaporators. Computer methods can be used in analyzing the design and operation of multiple effect evaporation systems (Holland and Liapis, 1983).

Figure 6.4 shows a triple-effect rising-film evaporator with forward feed operation (co-current flow of heating medium (steam/vapors) and liquid). Forward feed evaporators are preferred because the feed (low concentration, low viscosity) is evaporated more efficiently at high temperatures, without serious fouling.

Table 6.2 Steam Economies (SE) of Evaporator Systems

| Evaporator system | SE , kg water/kg steam |
|---------------------------------|--------------------------|
| Single effect | 0.90 - 0.98 |
| Double effect | 1.70 - 1.90 |
| Triple effect | 2.40 - 2.80 |
| Six effect | 4.60 - 4.90 |
| Thermocompressor, triple effect | 4 - 8 |
| Mechanical vapor recompression | 10 - 30 |

The steam economy (SE) in a multiple effect system is higher than 1, but less than the number of effects N :

$$SE = \sum m_{vi}/m_s \quad (i=1, N) \quad (6-7)$$

where m_s is the steam consumption (kg/s) and m_{vi} is the evaporation rate (kg/s) of the i effect.

The boiling point elevation BPE has a negative effect on the operation of a multiple effect evaporation system. In such a case, the vapors coming out of the vapor/liquid separator will be superheated by BPE degrees, but they will be condensed in the heater of the next effect at saturation temperature, losing the BPE superheat as available driving force ΔT . For most liquid foods, the BPE is usually small and it can be neglected, except in very concentrated sugar solutions and juices (last effects).

In the evaporation of large quantities of aqueous non-food solutions (e.g., water desalination) the number of effects in multiple effect systems may be much higher than in food evaporators, because higher temperatures in the first effect can be used, and the temperature difference ΔT per effect can be smaller. Thus, multiple-effect evaporator systems with 8-12 effects may be used, achieving significant reduction of the operating cost. In very large desalination plants (e.g., from sea water) the multiple stage flash (MSF) evaporation system is used with a large number of stages (25-40) and a small temperature difference (e.g., $\Delta T=2^\circ\text{C}$) per effect.

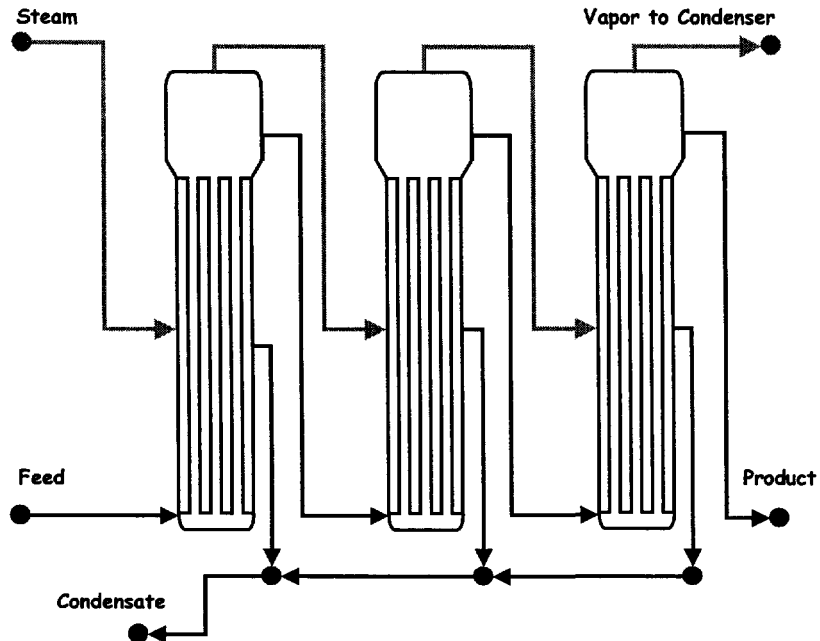


Figure 6.4 Schematic diagram of a triple-effect, forward-feed evaporator.

2. Vapor Recompression Evaporators

Steam economies higher than those of multiple effect systems can be obtained by vapor recompression evaporators, in which the vapors from the evaporation unit are compressed and re-used as a heating medium. Recompression is achieved either by thermal or mechanical compressors (Figure 6.5). Thermal recompression is economical when high-pressure steam is available at low cost, while mechanical vapor recompression (*MVR*) evaporators are preferred when electrical power is available at low cost. Investment cost for mechanical compressors is much higher than thermal compressors, but the operating cost is lower. Higher steam economies are obtained with mechanical recompression (Table 6.2).

In the thermal recompression system, the vapors are compressed in a steam ejector by high pressure steam (about 7 bar), and the mixture is used as a heating medium of the same evaporator unit (effect). Part of the vapors must be removed in a condenser to satisfy the material balance around the unit.

In the mechanical recompression system, the vapors are compressed mechanically and they are used as the heating medium of the evaporator unit. A small amount of heating steam is added to the system to make up the condensate formed during compression of water vapors. Centrifugal compressors are used to compress the water vapors by a ratio of 1.4-2.0, increasing the temperature difference ΔT by 5 to 20°C. More economical operation is obtained with turbo fans, which operate at a lower compression ratio, e.g., 1.2, which corresponds to a ΔT of about 5°C. Low ΔT 's can be applied to falling film evaporators, operating with no appreciable boiling point elevation and low pressure drop in the tube. Sizing and cost data for both thermocompressor and *MVR* systems are given by Minton (1986).

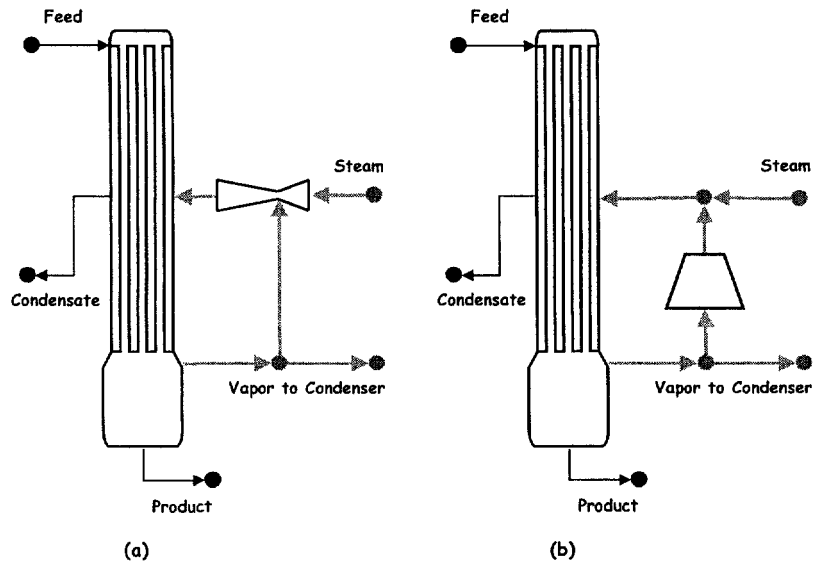


Figure 6.5 Vapor recompression evaporators. (a) thermal; (b) mechanical.

Combined multiple-effect and vapor recompression systems (thermal or mechanical), offering high steam economies and economic operation, are often used in the concentration of liquid foods.

Figure 6.6a shows a 3-effect evaporator system in which the first effect is replaced with a thermal recompression unit. Part of the vapors coming out of the vapor/liquid separator of the first effect is compressed in a steam ejector and used for heating the same effect, while the remaining vapors are used as a heating medium of the second effect. Total energy consumption of such a system can be about 25% of the energy required in a single effect system.

The design of mechanical vapor recompression evaporators is based on the design of the mechanical compressors (Minton, 1986). High energy economies can be achieved in single effect units, using fan compressors of small pressure differential, corresponding to about $\Delta T = 5^{\circ}\text{C}$. Combined systems of mechanical vapor recompression and multiple-effect evaporators may require only 8-10% of the total energy of a simple evaporator. Figure 6.6b shows a 3-effect evaporator with a mechanical compressor, which compresses the vapors of the last effect to heat the first effect.

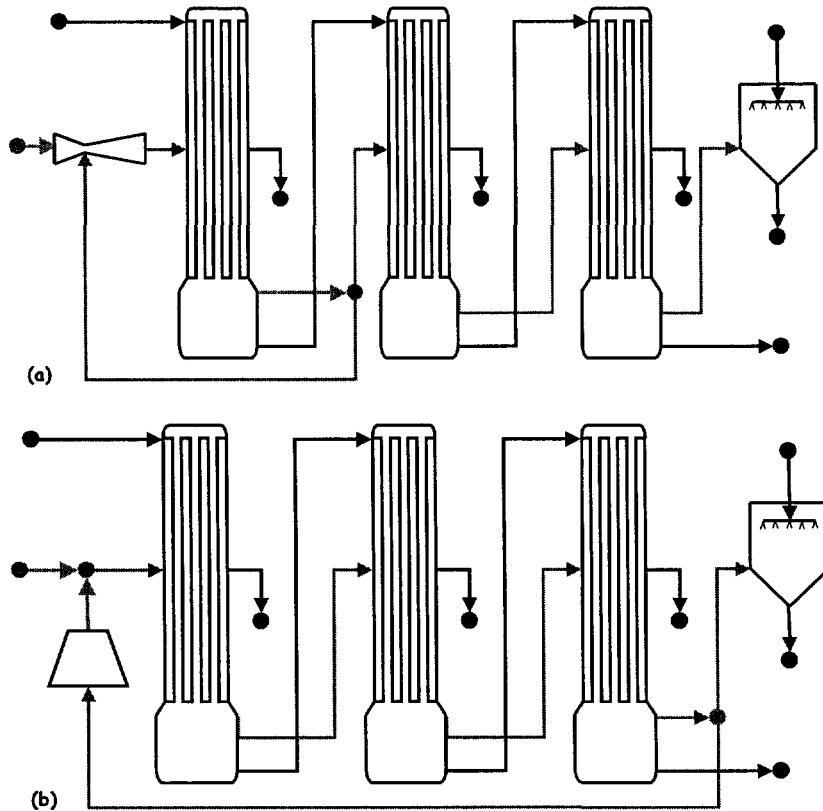


Figure 6.6 Combined recompression and triple-effect evaporators: (a) thermal; (b) mechanical.

V. DESIGN OF A TRIPLE-EFFECT EVAPORATOR

In this section, a detailed design approach is described for triple-effect forced-circulation evaporator.

1. Process Description

A typical flowsheet of a triple-effect forced-circulation evaporator is presented in Figure 6.7. The process stream with a flow rate L_0 (kg/s), a temperature T_0 (°C) and concentration X_0 (kg/kg wb) enters the first stage in which part of the feed is evaporated at temperature T_1 (°C) with a flow rate V_1 (kg/s). The first stage is heated by utility steam at temperature T_s (°C) and flow rate V_s (kg/s). The second stage is heated by the vapor produced in the first stage and so on. The liquid outlet from the third stage is the concentrated product. The vapor produced in the third stage is condensed in a direct contact condenser, which uses cooling water with flow rate L_w (kg/s) and temperature T_w (°C).

2. Process Model

A mathematical model of the process presented in Figure 6.7 is summarized in Table 6.3.

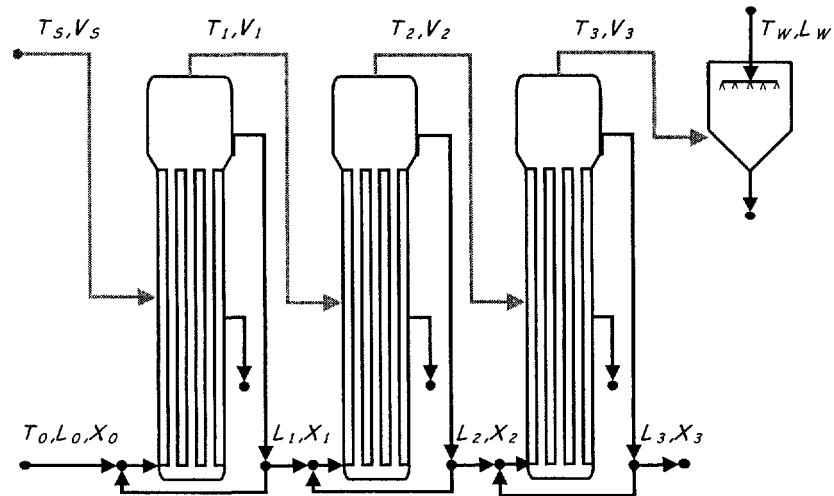


Figure 6.7 Flowsheet of a triple-effect forced-circulation evaporator.

The first seven Equations (E01-E07) refer to the first evaporator: Equation (E01) is the material balance for the solids and Equation (E02) is the total material balance. Equation (E03) is the energy balance for the process stream, while Equation (E04) is the energy balance for the heating steam. Equation (E05) is the well known phenomenological equation for heat transfer between the two streams. Equation (E06) calculates the overall heat transfer coefficient as a function of flow conditions and the properties of the liquids. Equation (E07) calculates the boiling point elevation. Equations (E08-E14) are the corresponding equations for the second evaporator, while Equations (E15-21) pertain to the third evaporator. Equations (E22-E24) refer to the direct contact condenser: Equation (E22) refers to condensing stream, and Equation (E23) to the cooling water. Equation (E24) calculates the volume of the direct contact condenser versus the liquid residence time. The same equation is used for sizing the vapor-liquid separation vessels for each evaporator (Equations E25, E26, and E27). The number of tubes for each evaporator and the corresponding shell diameter are calculated from Equations (E28-E33). Equation (E34) calculates the friction pressure loss inside the tube when the Fanning friction coefficient is estimated from Equation (E35). Equation (E36) calculates the total fluid flow in order to obtain the appropriate fluid velocity, while Equation (E37) estimates the required pump electrical work. Equations (E34-E37) refer to the first evaporator, (E38-E41) to the second, and (E42-E45) to the third. Finally Equation (E46) defines the steam economy for the evaporating system.

Table 6.4 summarizes the estimates of the properties used in the process model of Table 6.3. Equation (E01) suggests that a series structural model is used for the estimation of the density ρ of the solutions versus the densities of water ρ_w and solids ρ_s . On the other hand, a parallel structural model is used for the specific heat C_p (Equation E02). The effect of temperature on the density and specific heat is negligible. More information on properties structural models is presented by Saravacos and Maroulis (2001). Equation (E03) calculates the thermal conductivity λ versus the concentration and the temperature of the mixture. Equation (E04) defines the apparent viscosity η of non-Newtonian fluids inside tubes. The corresponding consistency coefficient K and the flow behavior index n versus the fluid concentration and temperature are estimated from Equations (E05) and (E06), respectively (Saravacos and Maroulis, 2001). The latent heat for water evaporation ΔH_s versus the temperature is estimated by the Equation (E07). Equation (E08) defines the overall heat transfer coefficient (U) as a function of heat transfer coefficient inside the tubes (h), the heat transfer coefficient in steam side (h_s), the heat transfer coefficient through the wall (h_w) and the heat transfer coefficient through the fouling (h_f). Equations (E09) and (E10) calculate the crucial technical transport properties of heat transfer coefficient inside the tubes (h) and the Fanning friction coefficient (f), respectively. Finally, Equation (E11) calculates the BPE as a function of concentration.

It must be noted that the detailed properties calculation described in Table 6.4 is necessary for accurate design of the evaporators because the fluid properties and mainly the viscosity are varied considerably, as water is removed during the evaporation process.

Table 6.3 Detailed Model of a Triple-Effect Evaporator

1st Effect

$$L_0 X_0 = L_1 X_1 \quad (\text{E01})$$

$$L_0 = L_1 + V_1 \quad (\text{E02})$$

$$Q_1 = L_0 C_p (X_1) (T_1 - T_0) + V_1 \Delta H (T_1) \quad (\text{E03})$$

$$Q_1 = V_s \Delta H (T_s) \quad (\text{E04})$$

$$Q_1 = A_1 U_1 (T_s - T_1) \quad (\text{E05})$$

$$U_1 = U (X_1, T_1, u_1, d, L) \quad (\text{E06})$$

$$\delta T_1 = BPE (X_1) \quad (\text{E07})$$

2nd Effect

$$L_1 X_1 = L_2 X_2 \quad (\text{E08})$$

$$L_1 = L_2 + V_2 \quad (\text{E09})$$

$$Q_2 = L_1 C_p (X_2) (T_2 - T_1) + V_2 \Delta H (T_2) \quad (\text{E10})$$

$$Q_2 = V_1 \Delta H (T_1 - \delta T_1) \quad (\text{E11})$$

$$Q_2 = A_2 U_2 [(T_1 - \delta T_1) - T_2] \quad (\text{E12})$$

$$U_2 = U (X_2, T_2, u_2, d, L) \quad (\text{E13})$$

$$\delta T_2 = BPE (X_2) \quad (\text{E14})$$

3rd Effect

$$L_2 X_2 = L_3 X_3 \quad (\text{E15})$$

$$L_2 = L_3 + V_3 \quad (\text{E16})$$

$$Q_3 = L_2 C_p (X_3) (T_3 - T_2) + V_3 \Delta H (T_3) \quad (\text{E17})$$

$$Q_3 = V_2 \Delta H (T_2 - \delta T_2) \quad (\text{E18})$$

$$Q_3 = A_3 U_3 [(T_2 - \delta T_2) - T_3] \quad (\text{E19})$$

$$U_3 = U (X_3, T_3, u_3, d, L) \quad (\text{E20})$$

$$\delta T_3 = BPE (X_3) \quad (\text{E21})$$

Condenser

$$Q_c = V_3 \Delta H (T_3 - \delta T_3) \quad (\text{E22})$$

$$Q_c = L_w C_{pw} [(T_3 - \delta T_3) - T_w] \quad (\text{E23})$$

$$V_c = f_c \frac{L_w + V_3}{\rho_w} \tau_c \quad (\text{E24})$$

1st Vessel

$$V_{R1} = f_v \frac{F_1 - V_1}{\rho(X_1, T_1)} \tau \quad (\text{E25})$$

2nd Vessel

$$V_{R2} = f_v \frac{F_2 - V_2}{\rho(X_2, T_2)} \tau \quad (\text{E26})$$

3rd Vessel

$$V_{R3} = f_v \frac{F_3 - V_3}{\rho(X_3, T_3)} \tau \quad (\text{E27})$$

1st Exchanger Geometry

$$A_1 = N_1 \pi d L \quad (\text{E28})$$

$$D_1 = d \left(\frac{N_1}{0.32} \right)^{0.47} \quad (\text{E29})$$

2nd Exchanger Geometry

$$A_2 = N_2 \pi d L \quad (\text{E30})$$

$$D_2 = d \left(\frac{N_2}{0.32} \right)^{0.47} \quad (\text{E31})$$

3rd Exchanger Geometry

$$A_3 = N_3 \pi d L \quad (\text{E32})$$

$$D_3 = d \left(\frac{N_3}{0.32} \right)^{0.47} \quad (\text{E33})$$

Table 6.3 Continued1st Pump

$$\Delta p_1 = 4f_1 \left(\frac{L}{d} \right) \frac{\rho(X_1, T_1) u_1^2}{2} \quad (\text{E34})$$

$$f_1 = f(X_1, T_1, u_1, d, L) \quad (\text{E35})$$

$$F_1 = u_1 N_1 \left(\frac{\pi d^2}{4} \right) \quad (\text{E36})$$

$$E_1 = F_1 \Delta p_1 \quad (\text{E37})$$

2nd Pump

$$\Delta p_2 = 4f_2 \left(\frac{L}{d} \right) \frac{\rho(X_2, T_2) u_2^2}{2} \quad (\text{E38})$$

$$f_2 = f(X_2, T_2, u_2, d, L) \quad (\text{E39})$$

$$F_2 = u_2 N_2 \left(\frac{\pi d^2}{4} \right) \quad (\text{E40})$$

$$E_2 = F_2 \Delta p_2 \quad (\text{E41})$$

3rd Pump

$$\Delta p_3 = 4f_3 \left(\frac{L}{d} \right) \frac{\rho(X_3, T_3) u_3^2}{2} \quad (\text{E42})$$

$$f_3 = f(X_3, T_3, u_3, d, L) \quad (\text{E43})$$

$$F_3 = u_3 N_3 \left(\frac{\pi d^2}{4} \right) \quad (\text{E44})$$

$$E_3 = F_3 \Delta p_3 \quad (\text{E45})$$

Steam Economy

$$SE = \frac{V_1 + V_2 + V_3}{V_s} \quad (\text{E46})$$

Table 6.4 Thermophysical and Transport Properties

Density

$$\rho(X, T) = \frac{1}{\frac{X}{\rho_s} + \frac{(1-X)}{\rho_w}} \quad (\text{E01})$$

Specific Heat

$$C_p(X) = XC_{ps} + (1-X)C_{pw} \quad (\text{E02})$$

Thermal Conductivity

$$\lambda(X, T) = X\lambda_s \exp\left[-\frac{E_s}{R}\left(\frac{1}{T} - \frac{1}{T_r}\right)\right] + (1-X)\lambda_w \exp\left[-\frac{E_w}{R}\left(\frac{1}{T} - \frac{1}{T_r}\right)\right] \quad (\text{E03})$$

Apparent Viscosity

$$\eta(X, T, u, d) = K\left(\frac{8u}{d}\right)^{n-1} \quad (\text{E04})$$

$$K = K_o \exp(BX) \exp\left[\frac{E}{R}\left(\frac{1}{T} - \frac{1}{T_o}\right)\right] \quad (\text{E05})$$

$$n = n_o - bX \quad (\text{E06})$$

Latent heat

$$\Delta H_s(T) = \Delta H_o - (C_{pL} - C_{pV})T \quad (\text{E07})$$

Overall heat transfer coefficient

$$U(X, T, u, d, L) = \frac{1}{\frac{1}{h} + \frac{1}{h_s} + \frac{1}{h_w} + \frac{1}{h_f}} \quad (\text{E08})$$

where

$$\left. \begin{aligned} \left(\frac{hd}{\lambda}\right) &= 1.86 \left(\frac{\rho du}{\eta} \frac{C_p \eta}{\lambda} \frac{d}{L}\right)^{1/3} \left(\frac{\eta}{\eta_w}\right)^{0.14}, \text{ for laminar flow} \\ \left(\frac{hd}{\lambda}\right) &= 0.023 \left(\frac{\rho du}{\eta}\right)^{0.8} \left(\frac{C_p \eta}{\lambda}\right)^{1/3} \left(\frac{\eta}{\eta_w}\right)^{0.14}, \text{ for turbulent flow} \end{aligned} \right\} \quad (\text{E09})$$

Friction coefficient

$$\left. \begin{aligned} f &= 16 \left(\frac{\rho du}{\eta}\right)^{-1}, \text{ for laminar flow} \\ f &= 0.08 \left(\frac{\rho du}{\eta}\right)^{-1/4}, \text{ for turbulent flow} \end{aligned} \right\} \quad (\text{E10})$$

Boiling point elevation

$$BPE(X) = \delta \exp(\zeta X) \quad (\text{E11})$$

Sixty variables presented in Table 6.5 are involved in the model of forty-six equations presented in Table 6.3, which means that fourteen degrees of freedom are available, and if we consider a typical problem with six specifications presented in Table 6.6 eight design variables are available for process optimization. Table 6.6 suggests a selection of design variables and the corresponding solution algorithm is presented in Table 6.7. The total annualized cost (*TAC*) presented in Table 6.8 is used as objective function in process optimization.

Concerning the design variables the following points should be considered:

- Standard size for tubes are often used. Tube diameter d varies between 2 and 5 cm, while tube length L varies between 5 and 10 m.
- The fluid velocities inside the tubes u_i are crucial design variables. When high values of velocities are selected, high heat transfer coefficients are obtained but significant pressure losses occur. Thus, lower equipment costs and higher operating costs are obtained.
- The evaporating temperatures T_i vary in the range between the heating steam temperature T_s and the cooling water temperature T_w .

When a multiple effect system is designed, it is a common practice to use similar effects. That is, when equal heat transfer areas are required, two more equations are added and consequently two degrees of freedom are spent. Thus, two temperatures are used as trial variables instead of design variables. Furthermore, if equal pumps are required, two more equations are added and consequently two degrees of freedom are further spent. Thus, two velocities are used as trial variables instead of design variables. This case is also presented in Table 6.6.

Table 6.5 Process Variables

| | | | |
|----------|----------|--|------|
| A_1 | m^2 | Heat transfer area of the 1 st evaporator | (01) |
| A_2 | m^2 | Heat transfer area of the 2 nd evaporator | (02) |
| A_3 | m^2 | Heat transfer area of the 3 rd evaporator | (03) |
| d | m | Tube diameter | (04) |
| D_1 | m | Shell diameter of the 1 st evaporator | (05) |
| D_2 | m | Shell diameter of the 2 nd evaporator | (06) |
| D_3 | m | Shell diameter of the 3 rd evaporator | (07) |
| E_1 | kW | Electric power of the 1 st pump | (08) |
| E_2 | kW | Electric power of the 2 nd pump | (09) |
| E_3 | kW | Electric power of the 3 rd pump | (10) |
| f_1 | - | Fanning friction coefficient at the 1 st evaporator effect | (11) |
| f_2 | - | Fanning friction coefficient at the 2 nd effect | (12) |
| f_3 | - | Fanning friction coefficient at the 3 rd effect | (13) |
| F_1 | m^3/s | Flow rate at the exchanger of the 1 st evaporator effect | (14) |
| F_2 | m^3/s | Flow rate at the exchanger of the 2 nd effect | (15) |
| F_3 | m^3/s | Flow rate at the exchanger of the 3 rd effect | (16) |
| L | m | Tube length | (17) |
| L_0 | kg/s | Feed flow rate | (18) |
| L_1 | kg/s | Liquid outlet from the 1 st evaporator effect | (19) |
| L_2 | kg/s | Liquid outlet from the 2 nd effect | (20) |
| L_3 | kg/s | Product flow rate | (21) |
| L_w | kg/s | Water flow rate | (22) |
| N_1 | - | Total number of tubes of the 1 st evaporator effect | (23) |
| N_2 | - | Total number of tubes of the 2 nd effect | (24) |
| N_3 | - | Total number of tubes of the 3 rd effect | (25) |
| Q_1 | kW | Thermal load of the 1 st evaporator effect | (26) |
| Q_2 | kW | Thermal load of the 2 nd effect | (27) |
| Q_3 | kW | Thermal load of the 3 rd effect | (28) |
| Q_c | kW | Thermal load of the condenser | (29) |
| T_0 | °C | Feed temperature | (30) |
| T_1 | °C | Evaporating temperature at 1 st evaporator effect | (31) |
| T_2 | °C | Evaporating temperature at 2 nd effect | (32) |
| T_3 | °C | Evaporating temperature at 3 rd effect | (33) |
| T_s | °C | Steam temperature | (34) |
| T_w | °C | Water temperature | (35) |
| u_1 | m/s | Fluid velocity in tubes (1 st effect) | (36) |
| u_2 | m/s | Fluid velocity in tubes (2 nd effect) | (37) |
| u_3 | m/s | Fluid velocity in tubes (3 rd effect) | (38) |
| U_1 | W/m^2K | Overall heat transfer coefficient at the 1 st evaporator effect | (39) |
| U_2 | W/m^2K | Overall heat transfer coefficient at the 2 nd effect | (40) |
| U_3 | W/m^2K | Overall heat transfer coefficient at the 3 rd effect | (41) |
| V_{R1} | m^3 | Liquid-vapor separation vessel of the 1 st evaporator effect | (42) |
| V_{R2} | m^3 | Liquid-vapor separation vessel of the 2 nd effect | (43) |
| V_{R3} | m^3 | Liquid-vapor separation vessel of the 3 rd effect | (44) |
| V_c | m^3 | Cooling tower volume | (45) |

Table 6.5 Continued

| | | | |
|--------------|----------|--|------|
| V_1 | kg/s | Vapor outlet from the 1 st evaporator effect | (46) |
| V_2 | kg/s | Vapor outlet from the 2 nd effect | (47) |
| V_3 | kg/s | Vapor outlet from the 3 rd effect | (48) |
| V_s | kg/s | Steam flow rate | (49) |
| X_0 | kg/kg wb | Feed concentration | (50) |
| X_1 | kg/kg wb | Concentration at the outlet of the 1 st evaporator effect | (51) |
| X_2 | kg/kg wb | Concentration at the outlet of the 2 nd effect | (52) |
| X_3 | kg/kg wb | Target concentration | (53) |
| Δp_1 | Pa | Friction pressure loss in tubes of the 1 st evaporator effect | (54) |
| Δp_2 | Pa | Friction pressure loss in tubes of the 2 nd effect | (55) |
| Δp_3 | Pa | Friction pressure loss in tubes of the 3 rd effect | (56) |
| δT_1 | °C | Boiling point elevation at 1 st evaporator effect | (57) |
| δT_2 | °C | Boiling point elevation at 2 nd effect | (58) |
| δT_3 | °C | Boiling point elevation at 3 rd effect | (59) |
| SE | - | Steam economy | (60) |

Technical Data

| | | | |
|--------------|---------------------|---|------|
| B | - | Parameter in consistency coefficient equation | (01) |
| b | - | Parameter in flow behavior index equation | (02) |
| C_{ps} | kJ/kg K | Specific heat of solids | (03) |
| C_{pv} | kJ/kg K | Specific heat of water vapor | (04) |
| C_{pw} | kJ/kg K | Specific heat of liquid water | (05) |
| E | kJ/mol | Activation energy for water in viscosity equation | (06) |
| E_s | kJ/mol | Activation energy for solids in thermal conductivity | (07) |
| E_w | kJ/mol | Activation energy for water in thermal conductivity | (08) |
| h_f | kW/m ² K | Fouling heat transfer coefficient | (09) |
| h_s | kW/m ² K | Steam side heat transfer coefficient | (10) |
| h_w | kW/m ² K | Wall heat transfer coefficient | (11) |
| K_o | Pa s ⁿ | Parameter in consistency coefficient equation | (12) |
| n_o | - | Parameter in flow behavior index equation | (13) |
| R | kJ/mol K | Perfect gas constant | (14) |
| T_o | K | Reference temperature for viscosity equation | (15) |
| T_r | K | Reference temperature for thermal conductivity equation | (16) |
| ΔH_o | kJ/kg | Latent heat of condensation of water vapor at 0°C | (17) |
| λ_s | W/m K | Solids thermal conductivity | (18) |
| λ_w | W/m K | Water thermal conductivity | (19) |
| ρ_s | kg/m ³ | Solids density | (20) |
| ρ_w | kg/m ³ | Water density | (21) |
| τ_c | s | Condenser residence time | (22) |
| τ_e | s | Separator residence time | (23) |
| δ | °C | Parameter in boiling point elevation equation | (24) |
| ζ | - | Parameter in boiling point elevation equation | (25) |

Table 6.6 Solution of a Typical Design Problem**Process Specifications**

| | | | |
|-------|----------|----------------------|-----|
| L_0 | kg/s | Feed flow rate | (1) |
| T_0 | °C | Feed temperature | (2) |
| X_0 | kg/kg wb | Feed concentration | (3) |
| X | kg/kg wb | Target concentration | (4) |
| T_s | °C | Steam temperature | (5) |
| T_w | °C | Water temperature | (6) |

Degrees-of-Freedom Analysis

| | |
|--------------------------|-----------|
| Process variables | 60 |
| <u>Process equations</u> | <u>46</u> |
| Degrees of freedom | 14 |
| <u>Specifications</u> | <u>6</u> |
| Design variables | 8 |

Design Variables

| | | | |
|-------|-----|---|-----|
| d | m | Tube diameter | (1) |
| L | m | Tube length | (2) |
| u_1 | m/s | Fluid velocity in tubes (1 st effect) | (3) |
| u_2 | m/s | Fluid velocity in tubes (2 nd effect) | (4) |
| u_3 | m/s | Fluid velocity in tubes (3 rd effect) | (5) |
| T_1 | °C | Evaporation temperature at 1 st effect | (6) |
| T_2 | °C | Evaporation temperature at 2 nd effect | (7) |
| T_3 | °C | Evaporation temperature at 3 rd effect | (8) |

Degrees-of-Freedom Analysis (Equal exchangers and equal pumps)

| | |
|--------------------------|-----------|
| Process variables | 60 |
| <u>Process equations</u> | <u>50</u> |
| Degree of freedom | 10 |
| <u>Specifications</u> | <u>6</u> |
| Design variables | 4 |

Design Variables (Equal exchangers and equal pumps)

| | | | |
|-------|-----|---|-----|
| d | m | Tube diameter | (1) |
| L | m | Tube length | (2) |
| u_3 | m/s | Fluid velocity in tubes (3 rd effect) | (3) |
| T_3 | °C | Evaporation temperature at 3 rd effect | (4) |

Table 6.7 Solution of a Typical Design Problem

| | X_1, X_2 | Assume initial trial values |
|-------|--------------------------|---|
| (E01) | $\rightarrow L_1$ | |
| (E02) | $\rightarrow V_1$ | |
| (E03) | $\rightarrow Q_1$ | |
| (E04) | $\rightarrow V_s$ | |
| (E07) | $\rightarrow \delta T_1$ | |
| (E11) | $\rightarrow Q_2$ | |
| (E10) | $\rightarrow V_2$ | |
| (E09) | $\rightarrow L_2$ | |
| (E08) | $\rightarrow X_2$ | If X_2 is different than the trial value of X_2 correct the value of X_2 and go to (E10) |
| (E14) | $\rightarrow \delta T_2$ | |
| (E18) | $\rightarrow Q_3$ | |
| (E15) | $\rightarrow V_3$ | |
| (E16) | $\rightarrow L_3$ | |
| (E13) | $\rightarrow X_3$ | If X_3 is different than specification correct the value of X_1 and go to (E01) |
| (E06) | $\rightarrow U_1$ | |
| (E05) | $\rightarrow A_1$ | |
| (E19) | $\rightarrow U_2$ | |
| (E12) | $\rightarrow A_2$ | |
| (E20) | $\rightarrow U_3$ | |
| (E17) | $\rightarrow A_3$ | |
| (E21) | $\rightarrow \delta T_3$ | |
| (E22) | $\rightarrow Q_c$ | |
| (E23) | $\rightarrow L_w$ | |
| (E24) | $\rightarrow V_c$ | |
| (E25) | $\rightarrow V_{R1}$ | (E26) $\rightarrow V_{R2}$ (E27) $\rightarrow V_{R3}$ |
| (E28) | $\rightarrow N_1$ | (E29) $\rightarrow D_1$ |
| (E30) | $\rightarrow N_2$ | (E31) $\rightarrow D_2$ |
| (E32) | $\rightarrow N_3$ | (E33) $\rightarrow D_1$ |
| (E35) | $\rightarrow f_1$ | (E34) $\rightarrow \Delta p_1$ |
| (E36) | $\rightarrow F_1$ | |
| (E37) | $\rightarrow E_1$ | |
| (E39) | $\rightarrow f_2$ | (E38) $\rightarrow \Delta p_2$ |
| (E40) | $\rightarrow F_2$ | |
| (E41) | $\rightarrow E_2$ | |
| (E43) | $\rightarrow f_3$ | (E42) $\rightarrow \Delta p_3$ |
| (E44) | $\rightarrow F_3$ | |
| (E45) | $\rightarrow E_3$ | |
| (E46) | $\rightarrow SE$ | |

Table 6.8 Cost Analysis**Equipment cost**

$$C_{eq} = \sum_j \left(C_{exc} A_j^{n_{exc}} + C_{ves} V_{Rj}^{n_{ves}} + C_{pum} E_j^{n_{pum}} \right) + C_{con} V_c^{n_{con}} \quad (F01)$$

Annual operating cost

$$C_{op} = \left(C_s V_s + C_w L_w + C_e \sum_j E_j \right) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (F04)$$

Cost Data

| | | |
|-----------|-------------------|-------------------------------|
| C_{exc} | \$/m ² | Heat exchange unit cost |
| n_{exc} | - | Heat exchanger scaling factor |
| C_{ves} | \$/m ³ | Vessel unit cost |
| n_{ves} | - | Vessel scaling factor |
| C_{pum} | \$/kW | Pump unit cost |
| n_{pum} | - | Pump scaling factor |
| C_{con} | \$/m ³ | Condenser unit cost |
| n_{con} | - | Condenser scaling factor |
| C_w | \$/kWh | Cost of water |
| C_s | \$/kWh | Cost of steam |
| C_e | \$/kWh | Cost of electricity |
| t_y | h/yr | Annual operating time |
| i_r | - | Interest rate |
| l_f | yr | Lifetime |

3. Application to Tomato-Paste Concentration

The previous design procedure is applied to the tomato paste concentration process presented in Section VI of Chapter 2. Typical process specifications for the design problem are presented in Table 6.9. The necessary technical and cost data are also summarized in Table 6.9. Most of these technical data are related to the properties defined in Table 6.4. As it can be shown through Figures 6.8 and 6.9, most of the tomato paste properties are strongly dependent on temperature and concentration. It should be noted that the boiling point elevation of tomato juice is relatively small and it can be neglected.

Table 6.9 Process Specifications and Process Technical and Cost Data**Process Specifications**

| | | |
|-------|----------|----------------------|
| L_0 | = 5 kg/s | Feed flow rate |
| T_0 | = 20°C | Feed temperature |
| X_0 | = 6% wb | Feed concentration |
| X | = 32% | Target concentration |
| T_s | = 120°C | Steam temperature |
| T_w | = 20°C | Water temperature |

Technical Data

| | | |
|--------------|--------------------------|---|
| B | = 0.15 | Parameter in consistency coefficient equation |
| b | = 0.0028 | Parameter in flow behavior index equation |
| C_{ps} | = 0.50 kJ/kg K | Specific heat of solids |
| C_{pv} | = 1.88 kJ/kg K | Specific heat of water vapor |
| C_{pw} | = 4.18 kJ/kg K | Specific heat of liquid water |
| E | = 15.8 kJ/mol | Activation energy for water in viscosity equation |
| E_s | = 5 kJ/mol | Activation energy for solids in thermal conductivity |
| E_w | = 0.17 kJ/mol | Activation energy for water in thermal conductivity |
| h_f | = 10 kW/m ² K | Fouling heat transfer coefficient |
| h_s | = 10 kW/m ² K | Steam side heat transfer coefficient |
| h_w | = 5 kW/m ² K | Wall heat transfer coefficient |
| K_o | = 1.27 Pa s ⁿ | Parameter in consistency coefficient equation |
| n_o | = 0.40 | Parameter in flow behavior index equation |
| R | = 8.31 kJ/kmolK | Perfect gas constant |
| T_o | = 25°C | Reference temperature for viscosity equation |
| T_r | = 60°C | Reference temperature for thermal conductivity equation |
| ΔH_o | = 2500 kJ/kg | Latent heat of condensation of water vapor at 0°C |
| λ_s | = 0.22 W/m K | Solids thermal conductivity |
| λ_w | = 0.68 W/m K | Water thermal conductivity |
| ρ_s | = 1600 kg/m ³ | Solids density |
| ρ_w | = 1000 kg/m ³ | Water density |
| τ_c | = 5 min | Condenser residence time |
| τ_e | = 5 s | Separator residence time |

Cost Data

| | | |
|-----------|--------------------------|-------------------------------|
| C_{exc} | = 3.00 \$/m ² | Heat exchange unit cost |
| n_{exc} | = 0.65 | Heat exchanger scaling factor |
| C_{ves} | = 2.00 \$/m ³ | Vessel unit cost |
| N_{ves} | = 0.65 | Vessel scaling factor |
| C_{pum} | = 0.50 \$/kW | Pump unit cost |
| N_{pum} | = 0.65 | Pump scaling factor |
| C_{con} | = 5.00 \$/m ³ | Condenser unit cost |
| N_{con} | = 0.65 | Condenser scaling factor |
| C_w | = 0.15 \$/tn | Cost of water |
| C_s | = 30.0 \$/tn | Cost of steam |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| t_y | = 1500 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Life-time |

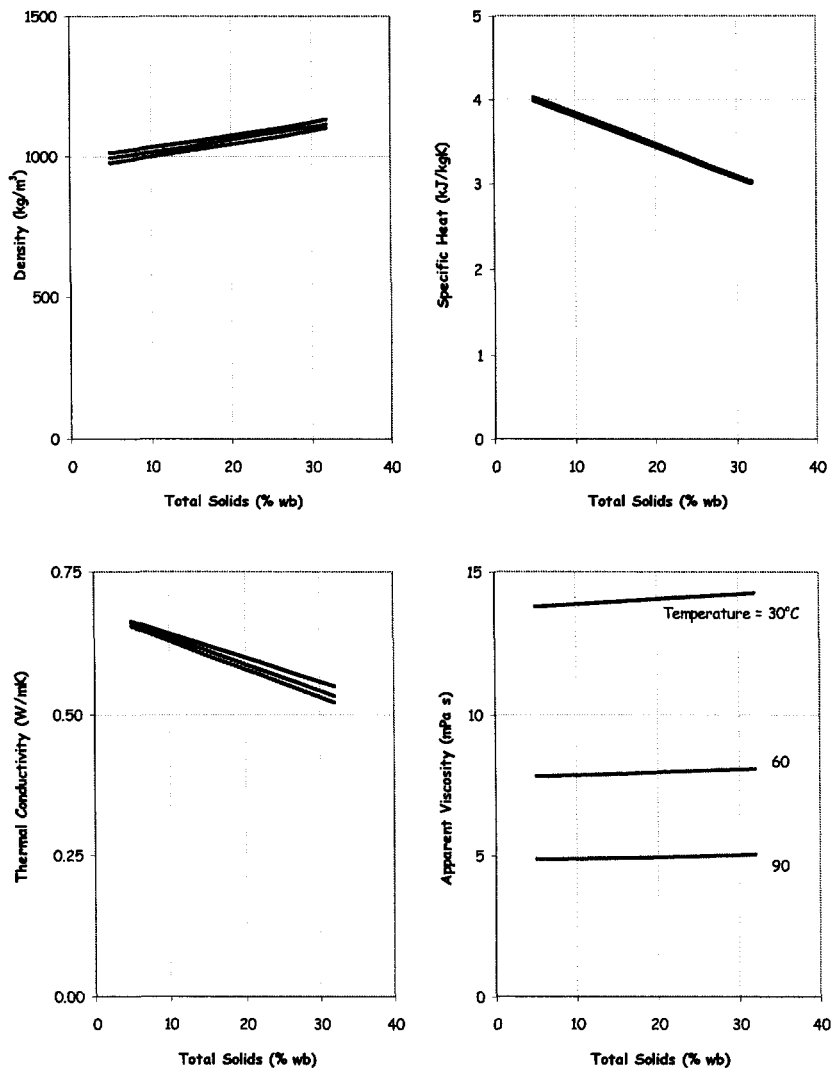


Figure 6.8 Thermophysical properties of tomato paste versus concentration at various temperatures. The data for apparent viscosity refer to tube diameter 2.5 cm and fluid velocity 5 m/s.

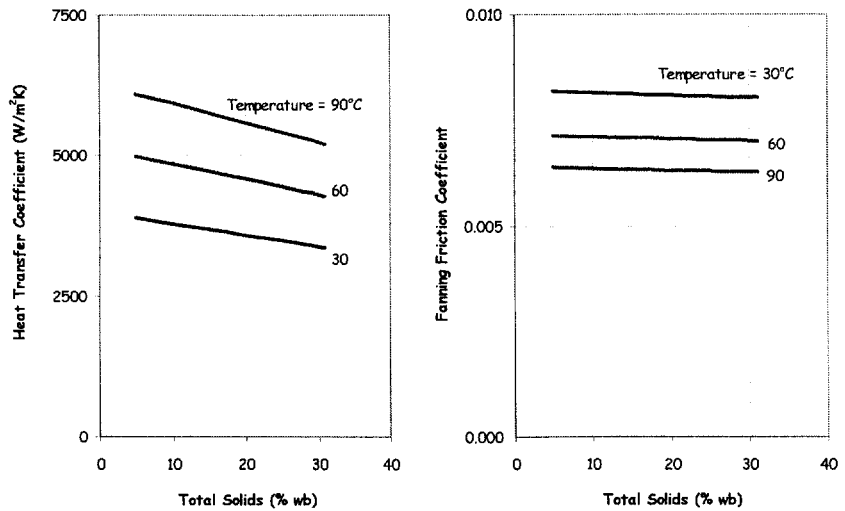


Figure 6.9 Heat transfer coefficient and Fanning friction coefficient for tomato paste versus concentration at various temperatures. The data refer to tube diameter 2.5 cm and fluid velocity 5 m/s.

The results of the design procedure presented in the previous section, with the assumption of equal heat transfer area and equal pumps, are presented in Tables 6.10 and 6.11. Some of these results are also presented graphically in Figures 6.10, 6.11, 6.12, and 6.13. Figure 6.10 represents an enthalpy-temperature diagram for the system. It shows the temperature difference and thermal load distribution in each evaporating stage. Figure 6.11 reveals the liquid temperature and concentration evolution versus the thermal load. Figure 6.12 compares the thermal and electrical loads at the different stages. Finally, Figure 6.13 represents the utility cost distribution.

Table 6.10 Process Design Results

| | | |
|--------------|--|---|
| d | = 2.5 cm | Tube diameter |
| L | = 5.00 m | Tube length |
| u_i | = 5.51, 5.34, 5.00 m/s | Fluid velocity in tubes |
| T_i | = 93.1, 74.2, 50.0°C | Evaporating temperatures |
| A_i | = 85 m ² | Heat transfer area of each evaporator |
| N | = 217 | Total number of tubes in each evaporator |
| D | = 0.96 m | Shell diameter |
| X_i | = 8.0, 12.4, 32.0%wb | Concentration at the outlet of the i^{th} effect |
| L_i | = 3.77, 2.41, 0.94 kg/s | Liquid outlet from the i^{th} effect |
| V_i | = 1.23, 1.36, 1.48 kg/s | Vapor outlet from the i^{th} effect |
| V_s | = 1.90 kg/s | Steam flow rate |
| U_i | = 1.84, 1.75, 1.54 kW/m ² K | Overall heat transfer coefficients |
| Q_i | = 4.23, 2.81, 3.16 MW | Thermal load of the i^{th} effect |
| F_i | = 0.6, 0.6, 0.5 m ³ /s | Flow rate in the exchanger of the i^{th} effect |
| Δp_i | = 0.75, 0.77, 0.83 bar | Friction pressure loss in tubes of the i^{th} effect |
| E_i | = 44 kW | Electric power of the pumps |
| V_{Ri} | = 6, 6, 5 m ³ | Liquid-vapor separation vessels |
| L_w | = 28 kg/s | Water flow rate |
| Q_c | = 3.52 MW | Thermal load of the condenser |
| V_c | = 18 m ³ | Cooling tower volume |
| SE | = 2.14 | Steam economy |

Table 6.11 Process Economics

| Process | Size | Utility | Equipment cost k\$ | Operating cost k\$/yr |
|-----------------------|-----------------------|-------------|-----------------------|--------------------------|
| Exchangers | 3 x 85 m ² | Steam | 162 | 308 |
| Separation vessels | 3 x 6 m ³ | - | 18 | - |
| Pumps | 3 x 44 kW | Electricity | 18 | 20 |
| Condenser | 18 m ³ | Water | 13 | 23 |
| Total | | | 211 | 350 |
| Total Annualized Cost | | | | 382 |

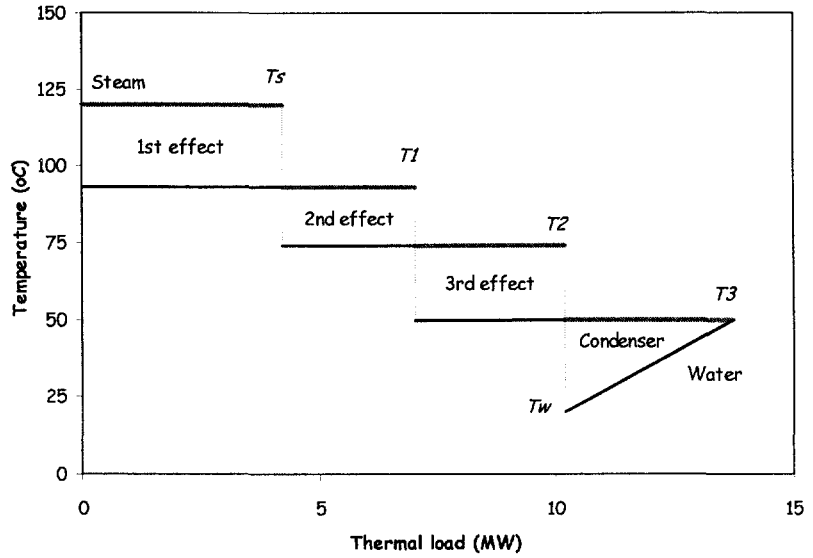


Figure 6.10 Evaporator enthalpy-temperature diagram.

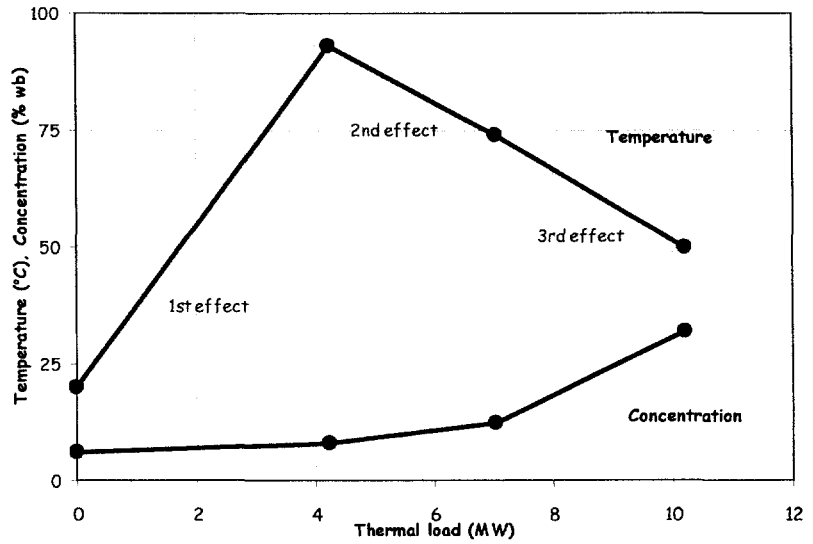


Figure 6.11 Liquid temperature and concentration versus thermal load.

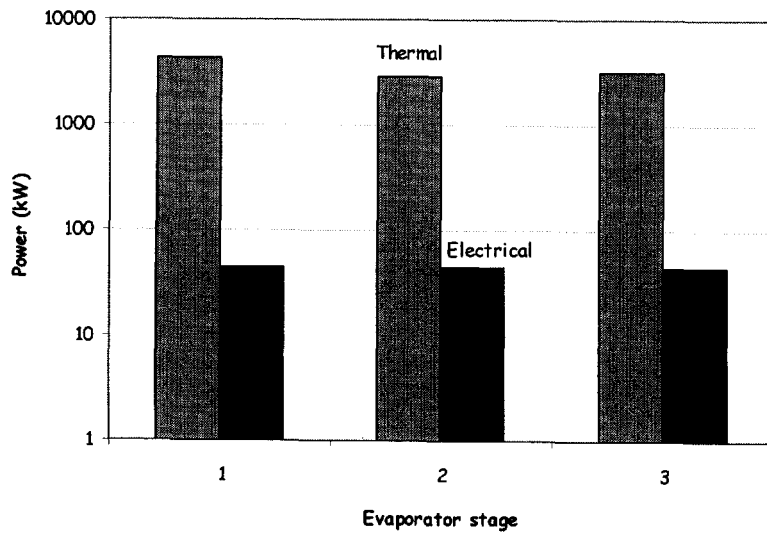


Figure 6.12 Thermal and electrical load distribution at evaporator stages.

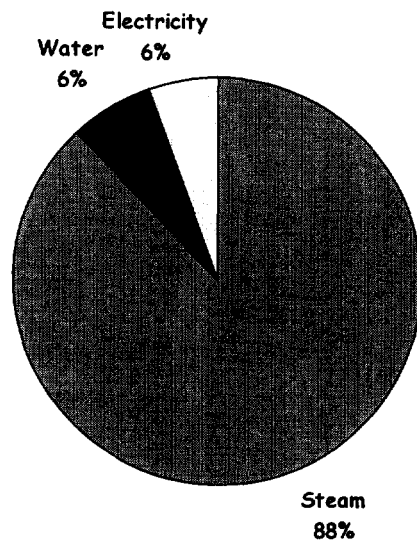


Figure 6.13 Utilities cost breakdown.

VI. DESIGN OF A VAPOR RECOMPRESSION EVAPORATOR

In this section, a detailed design approach is described for a mechanical vapor recombination evaporator.

1. Process Description

A typical forced-circulation evaporator with mechanical vapor recombination is presented in Figure 6.14. The process stream with a flow rate L_0 (kg/s), temperature T_0 (°C) and concentration X_0 (kg/kg wb) enters the evaporator and exits with a flow rate L (kg/s), temperature T (°C) and concentration X (kg/kg wb). All the vapors produced in the evaporator V (kg/s) are compressed mechanically and used as heating medium of the evaporator. A small amount of utility steam is used to make up the vapors condensed in the compressor and pipes.

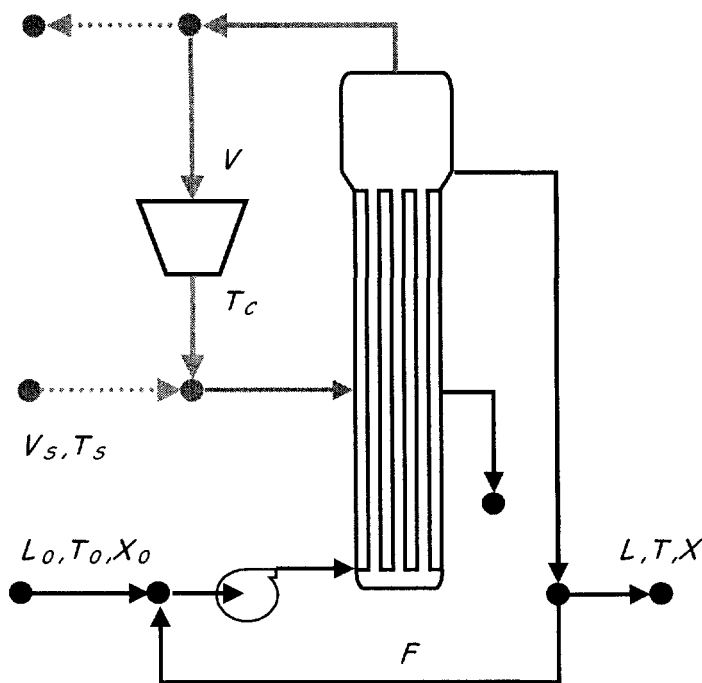


Figure 6.14 Flowsheet of a typical mechanical vapor recombination evaporator.

2. Process Model

A mathematical model of the process presented in Figure 6.14 is summarized in Table 6.12. Equation (E01) is the material balance for the solids and Equation (E02) is the total material balance. Equation (E03) is the energy balance for the process stream, while Equation (E04) is the energy balance for the heating steam. Equation (E05) is the well known phenomenological equation for heat transfer between the two streams. Equation (E06) calculates the overall heat transfer coefficient as a function of flow conditions and the properties of the liquids. The number of tubes in the heat exchanger and the corresponding shell diameter are calculated from Equations (E07) and (E08). Equation (E10) calculates the friction pressure loss inside the tube when the Fanning friction coefficient is estimated from Equation (E09). Equation (E11) calculates the total fluid flow in order to obtain the appropriate fluid velocity, while Equation (E12) estimates the required pump electrical work. Equation (E13) is used for sizing the vapor-liquid separation vessel as a function of liquid residence time. Equations (E14) through (E17) refer to the vapor compressor. Equations (E14) and (E15) are of the Antoine type and calculate the vapor pressures at the heating steam temperature and the temperature of the vapor produced in the evaporator, respectively. Equation (E16) suggests isentropic compression, while Equation (E17) calculates the required electrical work of the compressor. Finally, Equation (E18) defines the steam economy for the evaporating system. Most of the thermophysical and transport properties of the liquids, presented in Table 6.4 are also used in this model.

Twenty-seven variables, presented in Table 6.13, are involved in the model of eighteen equations presented in Table 6.12, which means that nine degrees of freedom are available, and if we consider a typical problem with five specifications, presented in Table 6.14, four design variables are available for process optimization. Table 6.14 also suggests a selection of design variables and a corresponding solution algorithm. No trial variables are required. The total annualized cost (*TAC*) presented in Table 6.15, is used as objective function in process optimization.

Table 6.12 Detailed Model of a Mechanical Vapor Recompression Evaporator

Exchanger

$$L_0 X_0 = LX \quad (\text{E01})$$

$$L_0 = L + V \quad (\text{E02})$$

$$Q = L_0 C_p (X_0) (T - T_0) + V \Delta H(T) \quad (\text{E03})$$

$$Q = V_s \Delta H(T_s) + V [\Delta H(T_s) + C_{pv} (T_c - T_s)] \quad (\text{E04})$$

$$Q = AU(T_s - T) \quad (\text{E05})$$

$$U = U(X, T, u, d, L) \quad (\text{E06})$$

Exchanger Geometry

$$A = N\pi dL \quad (\text{E07})$$

$$D = d \left(\frac{N}{0.32} \right)^{0.47} \quad (\text{E08})$$

Pump

$$f = f(X, T, u, d, L) \quad (\text{E09})$$

$$\Delta p = 4f \left(\frac{L}{d} \right) \frac{\rho(X, T) u^2}{2} \quad (\text{E10})$$

$$F = uN \left(\frac{\pi d^2}{4} \right) \quad (\text{E11})$$

$$E_p = F \Delta p \quad (\text{E12})$$

Vapor liquid separation vessel

$$V_R = f_v \frac{F - V}{\rho(X, T)} \tau_e \quad (\text{E13})$$

Compressor

$$P_s = \exp[a_1 - a_2 / (a_3 + T_s)] \quad (\text{E14})$$

$$P = \exp[a_1 - a_2 / (a_3 + T)] \quad (\text{E15})$$

$$\left(\frac{T_c}{T} \right) = \left(\frac{P_s}{P} \right)^k \quad (\text{E16})$$

$$E_c = V C_{pv} (T_c - T) \quad (\text{E17})$$

Steam Economy

$$SE = \frac{1.2Q}{3.0E_c} \quad (\text{E18})$$

Table 6.13 Process Variables

| | | | |
|------------|-------------|---|------|
| A | m^2 | Heat transfer area of the evaporator | (01) |
| d | m | Tube diameter | (02) |
| D | m | Shell diameter of the evaporator | (03) |
| E_p | kW | Electric power of the pump | (04) |
| E_c | kW | Electric power of the compressor | (05) |
| f | - | Fanning friction coefficient in the evaporator | (06) |
| F | m^3/s | Flow rate in the exchanger of the evaporator | (07) |
| L | m | Tube length | (08) |
| L_0 | kg/s | Feed flow rate | (09) |
| L | kg/s | Product flow rate | (10) |
| N | - | Total number of tubes of the evaporator | (11) |
| P | bar | Operating pressure | (12) |
| P_s | bar | Steam pressure | (13) |
| Q | kW | Thermal load of the evaporator | (14) |
| T_0 | $^{\circ}C$ | Feed temperature | (15) |
| T | $^{\circ}C$ | Evaporating temperature in evaporator | (16) |
| T_c | $^{\circ}C$ | Temperature at the compressor outlet | (17) |
| T_s | $^{\circ}C$ | Steam temperature | (18) |
| u | m/s | Fluid velocity in tubes | (19) |
| U | W/m^2K | Overall heat transfer coefficient in the evaporator | (20) |
| V_R | m^3 | Liquid-vapor separation vessel | (21) |
| V | kg/s | Vapor outlet from the evaporator | (22) |
| V_s | kg/s | Steam flow rate | (23) |
| X_0 | kg/kg wb | Feed concentration | (24) |
| X | kg/kg wb | Target concentration | (25) |
| Δp | Pa | Friction pressure loss in tubes of the evaporator | (26) |
| SE | - | Steam economy | (27) |

Table 6.14 Solution of a Typical Design Problem**Process Specifications**

| | | | |
|-------|----------|----------------------|-----|
| L_0 | kg/s | Feed flow rate | (1) |
| T_0 | °C | Feed temperature | (2) |
| X_0 | kg/kg wb | Feed concentration | (3) |
| X | kg/kg wb | Target concentration | (4) |
| T_s | °C | Steam temperature | (5) |

Degrees-of-Freedom Analysis

| | |
|--------------------|----|
| Process variables | 27 |
| Process equations | 18 |
| Degrees of freedom | 9 |
| Specifications | 5 |
| Design variables | 4 |

Design Variables

| | | | |
|-----|-----|-------------------------|-----|
| d | m | Tube diameter | (1) |
| L | m | Tube length | (2) |
| u | m/s | Fluid velocity in tubes | (3) |
| T | °C | Evaporating temperature | (4) |

Solution Algorithm

| | | |
|-------|---|------------|
| (E01) | → | L |
| (E02) | → | V |
| (E03) | → | Q |
| (E04) | → | V_s |
| (E05) | → | A |
| (E06) | → | U |
| (E07) | → | N |
| (E08) | → | D |
| (E09) | → | f |
| (E10) | → | Δp |
| (E11) | → | F |
| (E12) | → | E_p |
| (E13) | → | V_R |
| (E14) | → | P_s |
| (E15) | → | P |
| (E16) | → | T_c |
| (E17) | → | E_c |
| (E18) | → | SE |

Table 6.15 Cost Analysis**Equipment cost**

$$C_{eq} = C_{exc} A^{n_{exc}} + C_{ves} V_R^{n_{ves}} + C_{pum} E_p^{n_{pum}} + C_{com} E_c^{n_{com}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_s V_s + C_e E_p + C_e E_c) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (F04)$$

Cost Data

| | | |
|-----------|-------------------|-------------------------------|
| C_{exc} | \$/m ² | Heat exchange unit cost |
| n_{exc} | - | Heat exchanger scaling factor |
| C_{ves} | \$/m ³ | Vessel unit cost |
| n_{ves} | - | Vessel scaling factor |
| C_{pum} | \$/kW | Pump unit cost |
| n_{pum} | - | Pump scaling factor |
| C_{com} | \$/kW | Compressor unit cost |
| n_{com} | - | Compressor scaling factor |
| C_s | \$/kWh | Cost of steam |
| C_e | \$/kWh | Cost of electricity |
| t_y | hr/yr | Annual operating time |
| i_r | - | Interest rate |
| l_f | yr | Life-time |

3. Application to Tomato-Paste Concentration

The previous design procedure is applied to the tomato paste concentration process presented in Section VI of Chapter 2. The process specifications for the design, and the required technical and cost data are the same as those of the problem of Section V.3. as they are summarized in Table 6.9. A smaller feed rate of 2 kg/s is considered in this application.

The results of the design procedure discussed in this section are presented in Tables 6.16 and 6.17.

A comparison in process economics between vapor-recompression evaporator and triple-effect evaporator is presented in Figures 6.15 and 6.16. Considerable energy savings are obtained in vapor recompression evaporator but equipment cost is higher due to the compressor.

Table 6.16 Process Design Results

| | | |
|------------|---------------------------|---|
| L_o | = 2 kg/s | Feed flow rate |
| d | = 2.5 cm | Tube diameter |
| L | = 5 m | Tube length |
| u | = 5 m/s | Fluid velocity in tubes |
| T | = 100°C | Evaporating temperature |
| A | = 125 m ² | Heat transfer area |
| N | = 318 | Total number of tubes |
| D | = 0.64 m | Shell diameter |
| L | = 0.38 kg/s | Liquid outlet from the evaporator |
| V | = 1.62 kg/s | Vapor outlet from the evaporator |
| V_s | = 0.24 kg/s | Steam flow rate |
| U | = 1730 W/m ² K | Overall heat transfer coefficients |
| Q | = 4.32 MW | Thermal load of the evaporator |
| f | = 0.006 | Fanning friction coefficients |
| F | = 887 kg/s | Flow rate in the exchanger of the evaporator |
| Δp | = 0.68 bar | Friction pressure loss in tubes of the evaporator |
| E_p | = 53 kW | Electric power of the pump |
| E_c | = 246 kW | Electric power of the compressor |
| V_R | = 8 m ³ | Liquid-vapor separation vessel |
| SE | = 7.03 | Steam economy |

Table 6.17 Process Economics

| Process | Size | Utility | Equipment cost k\$ | Operating cost k\$/yr |
|-----------------------|--------------------|-------------|-----------------------|--------------------------|
| Exchanger | 125 m ² | Steam | 69 | 38 |
| Separation vessels | 8 m ³ | - | 23 | - |
| Pump | 53 kW | Electricity | 7 | 8 |
| Compressor | 246 kW | Electricity | 186 | 37 |
| Total | | | 285 | 83 |
| Total Annualized Cost | | | | 125 |

4. Application to Milk Concentration

The previous design procedure is also applied to milk concentration. The process specifications for the design, and the required technical and cost data, are summarized in Table 6.18, while results obtained are presented in Tables 6.19 and 6.20. Data on the viscosity of milk were taken from Velez-Ruiz and Barbosa-Canovas (1988).

Note: Heat-sensitive food liquids, such as fruit juices, are evaporated efficiently in falling-film evaporators equipped with mechanical vapor recompression. In these systems, centrifugal “fan” compressors, operating at low ΔT (e.g., 5°C), yield very high steam economy.

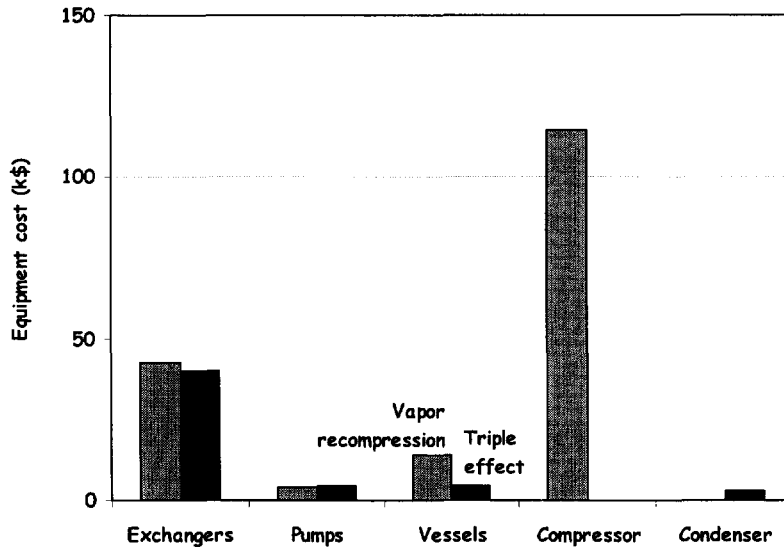


Figure 6.15 Comparison between vapor-recompression evaporator and triple-effect evaporator. Equipment cost. The figure refers to evaporating capacity of 1kg/s.

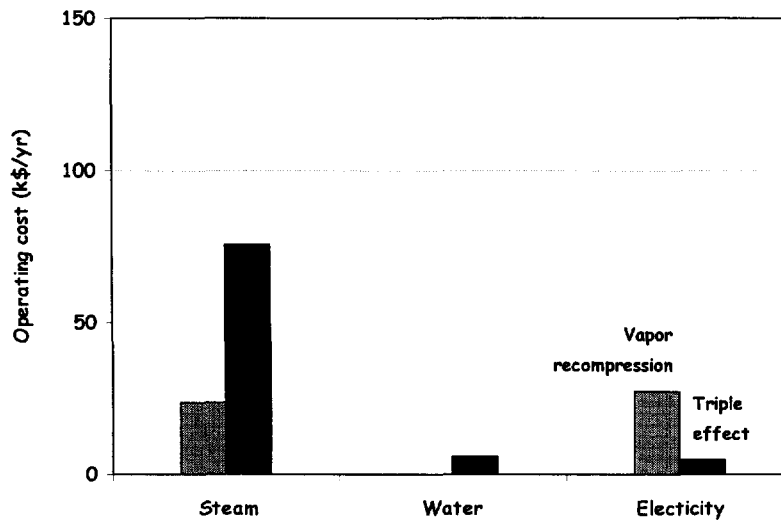


Figure 6.16 Comparison between vapor-recompression evaporator and triple-effect evaporator. Operating cost. The figure refers to evaporating capacity of 1kg/s.

Table 6.18 Process Specifications and Process Technical and Cost Data**Process Specifications**

| | | |
|-------|----------|----------------------------------|
| L_0 | = 2 kg/s | Feed flow rate |
| T_0 | = 20°C | Feed temperature |
| X_0 | = 12% wb | Feed concentration |
| X | = 30% | Target concentration |
| T_s | = 100°C | Steam (water vapors) temperature |

Technical Data

| | | |
|--------------|--------------------------|---|
| B | = 0.25 | Parameter in consistency coefficient equation |
| b | = 0.0025 | Parameter in flow behavior index equation |
| C_{ps} | = 0.50 kJ/kg K | Specific heat of solids |
| C_{pv} | = 1.88 kJ/kg K | Specific heat of water vapor |
| C_{pw} | = 4.18 kJ/kg K | Specific heat of liquid water |
| E | = 25 kJ/mol | Activation energy for water in viscosity equation |
| E_s | = 5 kJ/mol | Activation energy for solids in thermal conductivity |
| E_w | = 0.17 kJ/mol | Activation energy for water in thermal conductivity |
| h_f | = 10 kW/m ² K | Fouling heat transfer coefficient |
| h_s | = 10 kW/m ² K | Steam side heat transfer coefficient |
| h_w | = 5 kW/m ² K | Wall heat transfer coefficient |
| K_o | = 0.01 Pa s ⁿ | Parameter in consistency coefficient equation |
| n_o | = 1.00 | Parameter in flow behavior index equation |
| R | = 8.31 kJ/kmolK | Perfect gas constant |
| T_o | = 25°C | Reference temperature for viscosity equation |
| T_r | = 60°C | Reference temperature for thermal conductivity equation |
| ΔH_o | = 2500 kJ/kg | Latent heat of steam condensation at 0°C |
| λ_s | = 0.22 W/m K | Solids thermal conductivity |
| λ_w | = 0.68 W/m K | Water thermal conductivity |
| ρ_s | = 1500 kg/m ³ | Solids density |
| ρ_w | = 1000 kg/m ³ | Water density |
| τ_e | = 5 s | Separator residence time |

Cost Data

| | | |
|-----------|--------------------------|-------------------------------|
| C_{exc} | = 3.00 \$/m ² | Heat exchange unit cost |
| n_{exc} | = 0.65 | Heat exchanger scaling factor |
| C_{ves} | = 2.00 \$/m ³ | Vessel unit cost |
| n_{ves} | = 0.65 | Vessel scaling factor |
| C_{pum} | = 0.50 \$/kW | Pump unit cost |
| n_{pum} | = 0.65 | Pump scaling factor |
| C_{com} | = 3.00 \$/kW | Condenser unit cost |
| n_{com} | = 0.75 | Condenser scaling factor |
| C_s | = 30.0 \$/tn | Cost of steam |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| t_y | = 1500 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Life-time |

Table 6.19 Process Design Results

| | | |
|------------|---------------------------|---|
| d | = 2.5 cm | Tube diameter |
| L | = 5 m | Tube length |
| u | = 5 m/s | Fluid velocity in tubes |
| T | = 80°C | Evaporating temperature |
| A | = 84 m ² | Heat transfer area |
| N | = 215 | Total number of tubes |
| D | = 0.53 m | Shell diameter |
| L | = 0.80 kg/s | Liquid outlet from the evaporator |
| V | = 1.20 kg/s | Vapor outlet from the evaporator |
| V_s | = 0.16 kg/s | Steam flow rate |
| U | = 1915 W/m ² K | Overall heat transfer coefficients |
| Q | = 3.23 MW | Thermal load of the evaporator |
| f | = 0.005 | Fanning friction coefficients |
| F | = 620 kg/s | Flow rate at the exchanger of the evaporator |
| Δp | = 0.59 bar | Friction pressure loss in tubes of the evaporator |
| E_p | = 31 kW | Electric power of the pump |
| E_c | = 196 kW | Electric power of the compressor |
| V_R | = 5 m ³ | Liquid-vapor separation vessel |
| SE | = 6.58 | Steam economy |

Table 6.20 Process Economics

| Process | Size | Utility | Equipment cost k\$ | Operating cost k\$/yr |
|-----------------------|-------------------|-------------|-----------------------|--------------------------|
| Exchanger | 84 m ² | Steam | 54 | 25 |
| Separation vessels | 5 m ³ | - | 18 | - |
| Pump | 31 kW | Electricity | 5 | 5 |
| Compressor | 196 kW | Electricity | 157 | 29 |
| Total | | | 233 | 59 |
| Total Annualized Cost | | | | 94 |

VII. FOOD QUALITY CONSIDERATIONS

Evaporation should minimize the undesirable changes in quality and nutritive value of heat-sensitive food materials, such as losses of organoleptic quality (flavor, aroma and color), damage to vitamins and proteins, and production of undesirable compounds (browning, caramelization). Losses of aroma components during evaporation of fruit juices and aqueous extracts can be restored by essence recovery in the distillate (Chapter 9), or by add-back of fresh juice to the concentrate.

Evaporation at low temperatures (vacuum operation) can prevent heat damage to food quality, but it has the disadvantage of lower heat transfer coefficients and longer residence time in the evaporator. High temperature-short time systems, such as the thermally accelerated short time evaporator (TASTE), are used for the concentration of citrus juices (Chen and Hernandez, 1997).

Fouling of evaporation surfaces is a serious problem in the evaporation of food suspensions and pulps, because it reduces heat transfer and evaporation rate, and it may produce undesirable changes in food quality and in the hygienic operation of the equipment. Fouling is minimized by proper evaporator design and operation. High fluid velocities are desirable, particularly with non-Newtonian (pseudoplastic) foods, since the apparent viscosity decreases at high shear rates. Mechanical agitation cleans the heating surface and reduces the apparent viscosity of such fluids.

High temperature evaporation may have beneficial effects in some foods, such as tomato products (paste, ketchup), in which the “cooked” flavor is desirable, and the losses in nutrients is not very important.

Hygienic (sanitary) guidelines and codes for food processing equipment (Chapter 2) are of particular importance to the design, operation, and cleaning of food evaporators. The most important hygienic problem is the fouling of evaporation surfaces, caused by deposition of food colloids and other food components. Fouling is an empirical function of the operation time, and periodic cleaning is necessary (Minton, 1986). The operation cycle may be long (a week) for sugar evaporators, and shorter for dairy units (a day), due to the precipitation of milk heat-sensitive proteins. CIP cleaning systems can be used for all the parts of food-contact equipment.

NOMENCLATURE

| | |
|----------|---|
| A | Heat transfer area |
| A_1 | Heat transfer area of the 1 st evaporator effect |
| A_2 | Heat transfer area of the 2 nd effect |
| A_3 | Heat transfer area of the 3 rd effect |
| B | Parameter in consistency coefficient equation |
| b | Parameter in flow behavior index equation |
| C_{ps} | Fluid specific heat |
| C_{pv} | Fluid specific heat |
| C_{pw} | Fluid specific heat |
| d | Tube diameter |
| D_1 | Shell diameter of the 1 st evaporator effect |
| D_2 | Shell diameter of the 2 nd effect |
| D_3 | Shell diameter of the 3 rd effect |
| E | Activation energy for water in viscosity equation |
| E_1 | Electric power of the 1 st pump |
| E_2 | Electric power of the 2 nd pump |
| E_3 | Electric power of the 3 rd pump |
| E_s | Activation energy for solids in thermal conductivity |
| E_w | Activation energy for water in thermal conductivity |

| | |
|----------|--|
| f_1 | Fanning friction coefficient at the 1 st evaporator effect |
| f_2 | Fanning friction coefficient at the 2 nd effect |
| f_3 | Fanning friction coefficient at the 3 rd effect |
| F_1 | Flow rate at the exchanger of the 1 st evaporator effect |
| F_2 | Flow rate at the exchanger of the 2 nd effect |
| F_3 | Flow rate at the exchanger of the 3 rd effect |
| h | Surface heat transfer coefficient in process stream side |
| h_f | Fouling heat transfer coefficient |
| h_s | Steam side heat transfer coefficient |
| h_s | Surface heat transfer coefficient in steam side |
| h_w | Wall heat transfer coefficient |
| K_o | Parameter in consistency coefficient equation |
| L | Tube length |
| L_0 | Feed flow rate |
| L_1 | Liquid outlet from the 1 st evaporator effect |
| L_2 | Liquid outlet from the 2 nd effect |
| L_3 | Product flow rate |
| L_w | Water flow rate |
| m_s | Steam consumption |
| m_{vi} | Evaporation rate of the i effect |
| N_1 | Total number of tubes of the 1 st evaporator effect |
| N_2 | Total number of tubes of the 2 nd effect |
| N_3 | Total number of tubes of the 3 rd effect |
| n_o | Parameter in flow behavior index equation |
| Q | Thermal load |
| Q_1 | Thermal load of the 1 st evaporator effect |
| Q_2 | Thermal load of the 2 nd effect |
| Q_3 | Thermal load of the 3 rd effect |
| Q_c | Thermal load of the condenser |
| R | Perfect gas constant |
| Re | Reynolds number |
| SE | Steam economy |
| T_0 | Feed temperature |
| T_1 | Evaporating temperature at 1 st evaporator effect |
| T_2 | Evaporating temperature at 2 nd effect |
| T_3 | Evaporating temperature at 3 rd effect |
| T_o | Reference temperature for viscosity equation |
| T_r | Reference temperature for thermal conductivity equation |
| T_s | Steam temperature |
| T_w | Water temperature |
| U | Overall heat transfer coefficient |
| U_1 | Overall heat transfer coefficient at the 1 st evaporator effect |
| U_2 | Overall heat transfer coefficient at the 2 nd effect |
| U_3 | Overall heat transfer coefficient at the 3 rd effect |
| u_1 | Fluid velocity in tubes (1 st effect) |
| u_2 | Fluid velocity in tubes (2 nd effect) |
| u_3 | Fluid velocity in tubes (3 rd effect) |
| V_1 | Vapor outlet from the 1 st evaporator effect |

| | |
|----------|---|
| V_2 | Vapor outlet from the 2 nd effect |
| V_3 | Vapor outlet from the 3 rd effect |
| V_c | Cooling tower volume |
| V_{R1} | Liquid-vapor separation vessel of the 1 st evaporator effect |
| V_{R2} | Liquid-vapor separation vessel of the 2 nd effect |
| V_{R3} | Liquid-vapor separation vessel of the 3 rd effect |
| V_s | Steam flow rate |
| wb | Wet basis |
| x | Wall thickness |
| X_0 | Feed concentration |
| X_1 | Concentration at the outlet of the 1 st evaporator effect |
| X_2 | Concentration at the outlet of the 2 nd effect |
| X_3 | Target concentration |

Greek

| | |
|--------------|--|
| Γ | Liquid flow rate per unit length (irrigation rate) |
| ΔH_o | Latent heat of condensation of water vapor at 0°C |
| Δp_1 | Friction pressure loss in tubes of the 1 st evaporator effect |
| Δp_2 | Friction pressure loss in tubes of the 2 nd effect |
| Δp_3 | Friction pressure loss in tubes of the 3 rd effect |
| ΔT | Temperature difference |
| δ | Parameter in boiling point elevation equation |
| δT_1 | Boiling point elevation at 1 st evaporator effect |
| δT_2 | Boiling point elevation at 2 nd effect |
| δT_3 | Boiling point elevation at 3 rd effect |
| ζ | Parameter in boiling point elevation equation |
| η | Fluid apparent viscosity |
| λ_s | Fluid thermal conductivity |
| λ_w | Fluid thermal conductivity |
| λ | Fluid thermal conductivity |
| ρ_s | Fluid density |
| ρ_w | Fluid density |
| τ_c | Condenser residence time |
| τ_e | Separator residence time |

Abbreviations

| | |
|------------|-------------------------|
| <i>BPE</i> | Boiling Point Elevation |
| <i>FR</i> | Fouling resistance |
| <i>SE</i> | Steam economy |
| <i>TAC</i> | Total Annualized Cost |
| <i>wb</i> | Wet basis |

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7

Dehydration

I. INTRODUCTION

Food dehydration is a traditional method of food preservation, which is also used for the production of special foods and food ingredients, and for the utilization or disposal of food plant wastes. In addition to the basic process engineering requirements, food dehydration must meet the strict standards for food quality and food hygiene and safety.

Removal of water from the food materials is usually accomplished by thermal evaporation, which is an energy-intensive process, due to the high latent heat of vaporization of water, e.g., 2.26 MJ/kg at 100 °C.

Part of the water in some “wet” food materials (e.g., solid food wastes) can be removed by inexpensive non-thermal processes, such as mechanical pressing/expression and filtration. Osmotic dehydration, requiring less energy than thermal drying, can be used for partial removal of water from foods.

Dehydration of liquid foods, such as milk and soluble coffee, is normally preceded by efficient thermal evaporation, which requires considerably less energy (Chapter 6). Membrane separation processes, such as ultrafiltration and reverse osmosis, requiring less energy, can be used for pre-concentration of liquid foods.

Diverse drying processes and equipment are used in food processing, due to the difficulty of handling and processing solid materials, and the special requirements for the various food products. In addition, economics (investment and operation) is an important factor, especially for seasonal and low cost food products, such as fruits and vegetables. The equipment ranges from crude solar dryers to sophisticated spray dryers or freeze-dryers. In addition to process and equipment design, food dehydration involves processing technology for various food products, discussed in specialized books, such as van Arsdel et al. (1973), Barbosa-Canovas and Vega-Mercado (1996), Baker (1997), Greensmith (1998), and Saravacos and Kostaropoulos (2002). The modeling, simulation and design of drying processes was reviewed by Maroulis and Saravacos (2002).

II. GENERAL CONSIDERATIONS

The design of industrial dryers in food processing is based mainly on practical experience, since handling and processing of solid and semi-solid food materials cannot be described adequately by mathematical models and computer simulations. Recent advances in the application of Transport Phenomena, Particle Technology, and Computer Technology to Food Engineering can improve markedly the design and operation of food dryers.

Food dehydration is a heat and mass transfer process, involving handling of solids and particles. The drying process must be cost-effective, and preserve or improve the quality of the product. Particular attention must be given to the safety, nutritional quality and consumer acceptance (convenience) of the dehydrated food products.

1. Heat and Mass Transfer

Heat and mass transport within the food materials (internal transport) controls the drying rate of most food. The internal transport properties, i.e., mass diffusivity (moisture and solutes) and thermal conductivity/diffusivity strongly affect the drying rate, and they should be known or determined experimentally for each food material. A detailed discussion on the transport properties of foods, with particular emphasis on the drying processes is presented by Saravacos and Maroulis (2001).

In general, both mass and thermal transport properties are affected strongly by the physical structure (porosity) of the material, and to a lesser degree by the temperature and the moisture content. Detailed data on the thermal transport properties are presented by Rahman (1995).

Interphase (surface) heat and mass transfer are important in the early stages of drying, when the external drying conditions (air velocity, temperature and humidity) have a decisive effect on the drying rate.

The surface heat transfer coefficient h (W/m^2K) in a drying operation is defined by the equation:

$$Q/A = h \Delta T \quad (7-1)$$

where Q/A (W/m^2) is the heat flux and ΔT (K) is the temperature difference between the heating medium and the heated surface of the material.

The surface mass transfer coefficient h_M (kg/m^2s) is defined by an analogous equation:

$$J = h_M \Delta Y \quad (7-2)$$

where J (kg/m^2s) is the mass transfer rate and ΔY (kg/kg db) is the difference of moisture content between the material's surface and the bulk of the drying medium (air).

The mass transfer coefficient k_c (m/s), also used in the literature, is based on the driving force of concentration difference ΔC (kg/m^3), according to the equation:

$$J = k_c \Delta C \quad (7-3)$$

The two mass transfer coefficients are interrelated by the equation:

$$h_M = k_c \rho \quad (7-4)$$

where ρ (kg/m³) is the density of the transfer medium, i.e., the air.

For air-moisture systems of normal drying conditions (atmospheric pressure, temperature less than 100°C) the air density is approximately 1 kg/m³ and, therefore, the two coefficients are numerically equal:

$$h_M \text{ (kg/m}^2\text{s)} \approx k_c \text{ (m/s)} \quad (7-5)$$

The interphase transfer coefficients are affected by the air velocity and temperature and the geometry of the transfer system. Approximate values of the coefficients are given in the literature (Rahman, 1995; Perry and Green, 1997). The transfer coefficients are correlated by empirical equations for various systems, using the known dimensionless numbers Reynolds ($Re = u\rho d/\eta$), Prandtl ($Pr = C_p\eta/\lambda$) Nusselt ($Nu = hd/\lambda$), Schmidt ($Sc = \eta/\rho D$), and Sherwood ($Sh = k_c d/D$).

The thermophysical and transport properties used in these numbers are the density ρ (kg/m³), specific heat C_p (J/kg K), velocity u (m/s), (equivalent) diameter d (m), thermal conductivity λ (W/m K), viscosity η (Pa s), mass diffusivity D (m²/s), heat transfer coefficient h (W/m²K), and mass transfer coefficient k_c (m/s).

The Colburn analogy is employed to relate the heat and mass transfer coefficients, using the dimensionless heat and mass transfer factors (j_H and j_M), defined by the following equation (Geankoplis, 1993):

$$j_H = j_M \quad (7-6)$$

where $j_H = St_H Pr^{2/3}$ and $j_M = St_M Sc^{2/3}$, where the Stanton numbers for heat and mass transfer, St_H and St_M are defined as: $St_H = Nu/RePr = h/u\rho C_p$ and $St_M = Sh/ReSc = h_M/u\rho = k_c/u$.

The Colburn heat and mass transfer analogy can be applied to air/moisture systems at atmospheric pressure for comparing the two transfer coefficients (Geankoplis, 1993):

$$h/h_M = C_p \text{ or } h/k_c = \rho C_p \quad (7-7)$$

For air at atmospheric pressure $\rho = 1$ kg/m³ and $C_p = 1000$ J/kgK. Therefore, the two coefficients are numerically identical, if expressed in appropriate units:

$$h \text{ (W/m}^2\text{K)} \approx h_M \text{ (g/m}^2\text{s)} \approx k_c \text{ (mm/s)} \quad (7-8)$$

The Colburn analogies can be simplified by assuming that the heat and mass transfer factors (j_H and j_M) are functions of the Reynolds number only (Saravacos and Maroulis, 2001). Regression analysis of several literature data has yielded the following two empirical equations:

$$j_H = aRe^n \text{ and } j_M = bRe^m \quad (7-9)$$

where a , b , m , and n are empirical constants. For convective (air) drying, Equations (7-9) become:

$$j_H = 1.04Re^{-0.45} \text{ and } j_M = 23.5Re^{-0.88} \quad (7-10)$$

2. Design of Industrial Dryers

The design of industrial dryers is based largely on empirical knowledge, while modeling and simulation can generalize and improve the design procedure. The diversity of drying processes and dried products has resulted in a multitude of drying equipment, specific for each class of products. Thus, selection of a particular dryer becomes as important as the engineering design of the equipment.

The principles and details of dryer design are outlined in Process Engineering books, such as Bhatia (1983), Lee (1983), Strumillo and Kudra (1986), Walas (1988), Van't Land (1991), Mujumdar (1995), and Perry and Green (1997). Food dryers are described and discussed in the books of Van Arsdell et al. (1973), Barbosa-Canovas and Vega-Mercado (1996), Baker (1997), Greensmith (1998), and Saravacos and Kostaropoulos (2002). Crapiste and Rotstein (1997) analyzed the design and performance of food dryers, and Tsotsas (1998) discussed a recent approach to the design of convective dryers.

The thermophysical, transport, and equilibrium (isotherms) properties of the material are very important in specifying the proper dryer and drying conditions. Mechanical properties of solids and solid particles are important in handling and processing of the various materials. Drying rates (kinetics of moisture removal) are useful for preliminary estimation of the drying time. The estimated time is taken as the approximate mean residence time in the dryer, an important parameter in dryer design.

The specifications of an industrial dryer should be listed in an equipment specification form, and should include the properties of the wet and dried material, the temperature sensitivity and water activity of the product, the capacity and evaporation duty (kg/h), the energy supply and cost, the environmental impact, and the relationship to the other plant operations.

There are about 40 classes and over 100 subclasses of dryers, which are classified by different methods, based on type of operation (batch, continuous), type of feed (liquid, suspension, paste, granules, fibrous solids, porous solids, dense solids, and sheets), heating method (convection, contact, radiation, dielectric), product sensitivity (vacuum, low temperature). The size of dryers can be small (up to 50 kg/h), medium (50-1000 kg/h), or large (above 1000 kg/h), Walas (1988).

The capacity of the dryers can be expressed also as kilograms of water evaporated per unit surface and unit time, which is very high in rotary dryers (about 50 kg/m²h) and low in tray dryers (about 1 kg/m²h).

The cost of drying is an important factor in dryer design, especially for large volume products of relatively low value. Energy (fuel) is the major cost in drying operations (62%), followed by capital and labor costs (Lee, 1983).

The major energy use is for the evaporation of water (moisture), which varies considerably for the different dryers, e.g., from 3 MJ/kg water (spray dryers) to 6 MJ/kg (tray dryers). The energy efficiency of the dryers (ratio of the heat of evaporation to the heat input to the dryer) depends strongly on the type of dryer. It is higher in contact than in convective drying, e.g., (40-80%) versus (20-40%). Rotary dryers are more efficient than tray, fluid bed, and spray dryers (Walas, 1988).

Four categories of industrial dryer problems have been identified by Kemp and Gardiner (2000): a) Underperforming (corrected by performing material and energy balances, by drying kinetics, and by applying moisture equilibria); b) materials handling; c) product quality; and d) mechanical breakdown.

Table 7.1 shows selected operating characteristics of various types of dryers, used in food processing (Walas, 1988; Perry and Green, 1997; Crapiste and Rotstein, 1997, Saravacos and Kostaropoulos, 2002). These data are further plotted in Figures 7.1 and 7.2. Figure 7.1 represents the range of the crucial operating conditions, that is, the residence time and the drying temperature. The various dryers are distributed along the diagonal of the graph, which means that the higher the temperature, the shorter the residence time. The resulting evaporation capacities are summarized in Figure 7.2.

Table 7.1 Characteristics of Food Dryers

| Dryer type | Product form | Product Temperature °C | Evaporation Capacity kg/m ² h | Residence time |
|-----------------|------------------|---------------------------|---|----------------|
| Bin or Silo | Pieces, Grains | 30 - 50 | - | 1 - 3 days |
| Tray | Pieces | 40 - 60 | 0.2 - 2 | 3 - 10 h |
| Tunnel | Pieces | 50 - 80 | 5 - 15 | 0.5 - 3 h |
| Conveyor belt | Pieces | 50 - 80 | 5 - 15 | 0.5 - 3 h |
| Rotary | Grains, granules | 60 - 100 | 30 - 100 | 0.2 - 1 h |
| Drum | Sheet | 80 - 110 | 5 - 30 | 10 - 30 s |
| Fluid bed | Grains, granules | 60 - 100 | 30 - 90 | 2 - 20 min |
| Pneumatic flash | Grains, granules | 60 - 120 | 10 - 100* | 2 - 20 s |
| Spray | Powder | 60 - 130 | 1 - 30* | 10 - 60 s |
| Vacuum/Freeze | Pieces | 10 - 20 | 1 - 7 | 5 - 24 h |

* kg/m³h. Pieces >5 mm, grains-granules 0. -5 mm, powders < 0.5 mm

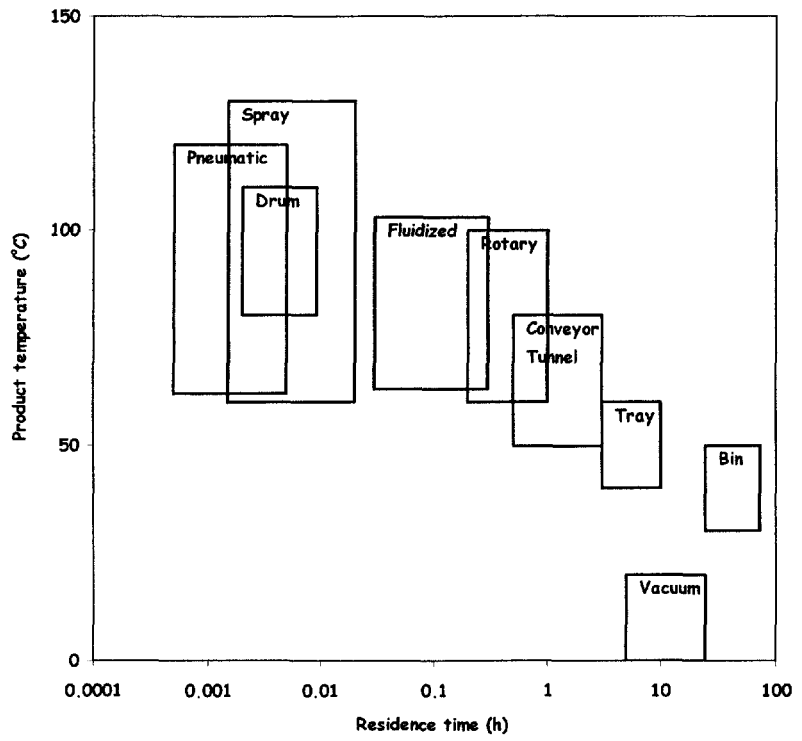


Figure 7.1 Operating conditions of various types of dryers. Data from Table 7.1.

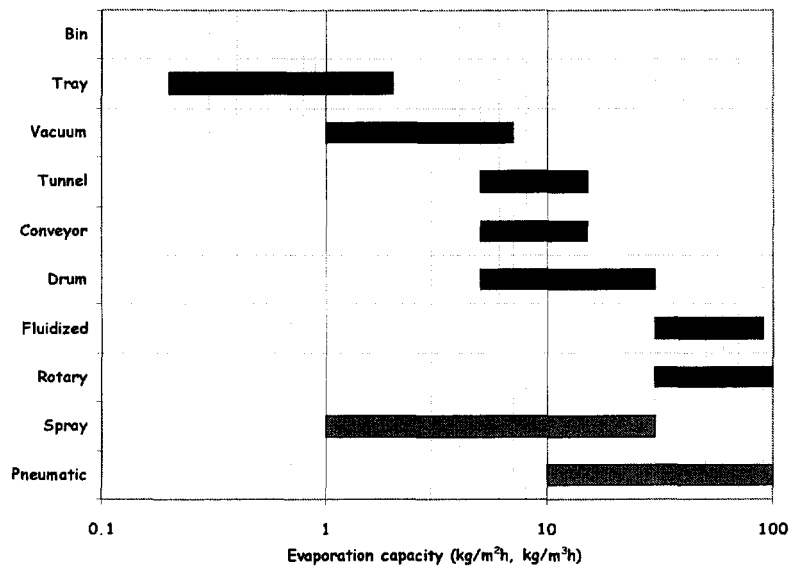


Figure 7.2 Evaporation capacity of various types of dryers. Data from Table 7.1.

III. DRYING EQUIPMENT

1. Selection of Industrial Dryers

Traditional methods of dryer selection are qualitative and they do not cover the wide range of available industrial dryers. Selection of the optimal dryer type and size should satisfy all process and product requirements at minimum cost. Food dryers should satisfy the strict requirements for food quality and food hygiene and safety.

Recent progress in computer applications has resulted in a number of selection procedures, which facilitate the selection of the proper dryer for each particular application.

Kemp (1999) has developed an expert system for dryer selection (DRYSEL), which has been tested in various industrial case studies. Dryer selection depends on a) the equipment, b) the material, and c) the overall process flowsheet. The algorithm includes the following steps: 1. Define the problem and supply the needed material and flowsheet data. 2. Make basic choices of feed/product form, operation/heating, single- or multi-stage. 3. Evaluate merit factors for individual dryers and perform approximate size estimation. 4. Study subtypes and refinements of the selected dryers. 5. Assess all remaining possible dryers and make final decision.

Batch dryers are used for solids throughputs (flows) below 50 kg/h, while continuous operation is preferred above 1000 kg/h.

The program may suggest several promising dryers, out of whom the final choice is made, based on the specific product.

A practical database (DRYERBASE), listing commercially available dryers and dryer manufacturers, was developed by Tsotsas (2000). The dryers are classified into 25 main types, which are sorted by dryer type and dryer manufacturer (supplier). The 10 most important industrial dryers of the general list are: tray, tunnel, belt, drum, rotary, bin, flash, fluid bed, spray, and tumbler dryers.

The mode of operation can be batch, continuous, or combined. Heating can be convective or contact, and operating pressure atmospheric or vacuum. The feed can be liquids, pastes, powders (< 0.5 mm), grains (0.5-5 mm), and pieces (> 5 mm).

Definitions and terms used in drying, and a list of drying equipment suppliers, were published by the German Society of Process Equipment VDMA (1998a, 1999b). This publication lists, in a matrix form, 35 suppliers and 65 drying systems, which include field of application, classification of dryers, and dryer construction (convection, contact, radiation, electrical, combined energy, and vacuum).

The European activities in drying are summarized in the web sites Hyprotech (2002) [www.hyprotech.com/poman/efce] and Drying (2002) [www.drying.net]. A dryer manufacturers database was prepared by Pakowski (2002).

2. Typical Food Dryers

The various dryers used in food processing are discussed by Saravacos and Kostaropoulos (2002). Details of the different dryers are described in bulletins of equipment manufacturers and suppliers. Some typical diagrams of food dryers are shown in this section of the book.

a. Bin (Silo) Dryers

The bin or silo dryers (Figure 7.3) are used for partial drying of large quantities of grains (wheat, corn, etc.) from moisture contents of about 25% (harvest) to about 18% (storage). Hot air at 40-60°C is blown from the bottom of the fixed bed through the grains for several hours. Continuous tower dryers are more effective, using higher air temperatures (e.g., 80°C), while the grain is moving slowly from the top to the bottom.

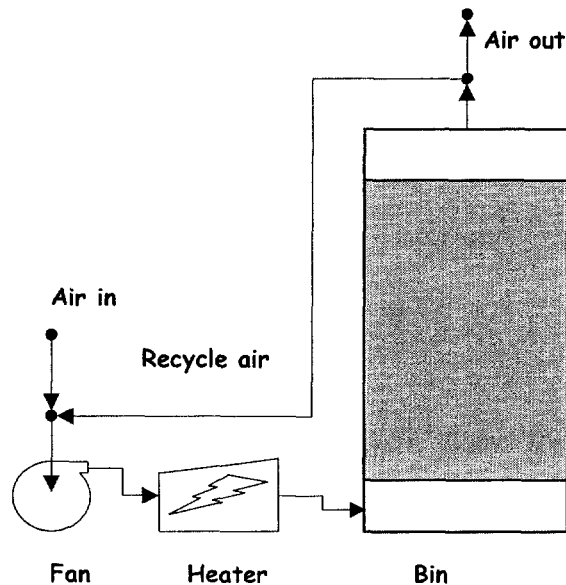


Figure 7.3 Batch bin (silo) dryer.

b. Tray Dryers

Tray dryers are relatively small batch units for drying small quantities of product (Figure 7.4). A heat exchanger is used to heat the air, which is usually recirculated to conserve thermal energy. The product in the form of pieces, particles, or pastes is placed in metallic trays, which are re-used after the drying operation.

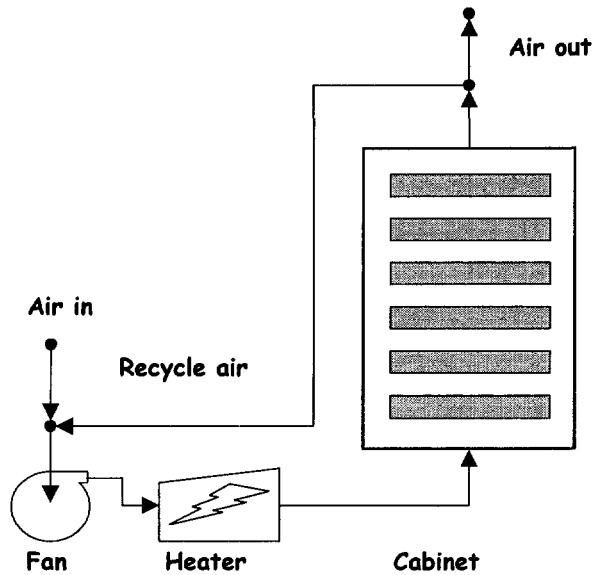


Figure 7.4 Cabinet (tray) dryer.

c. Tunnel (Truck) Dryers

Tunnel dryers are relatively inexpensive constructions, with the product trays (pieces, pastes) loaded on trucks, which move slowly co-current or counter-current to the hot air (Figure 7.5). Recirculation of the air improves thermal efficiency. The system runs semi-continuously, and the trays are loaded and unloaded manually.

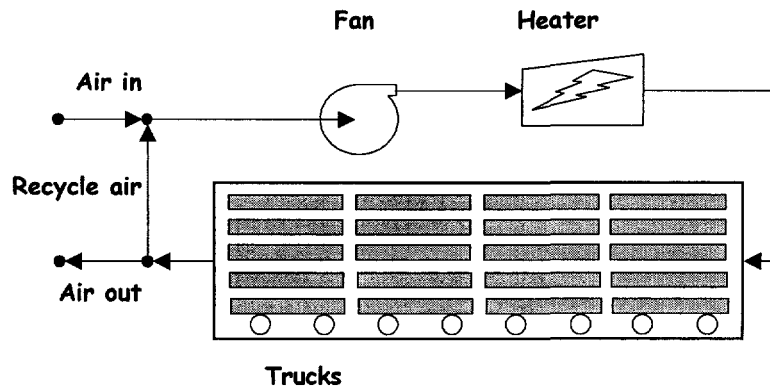


Figure 7.5 Tunnel (truck) dryer.

d. Belt Dryers

Belt (conveyor) dryers are used widely in food processing for continuous drying of food pieces (Figure 7.6). The product, in the form of pieces (e.g., fruits and vegetables), is dried on a long perforated conveyor belt, which moves slowly in order to stay in the dryer for the required drying time. The air is heated to the desired temperature in heat exchangers or is mixed with the combustion gases of “clean” fuels, and it is directed against the product in up- or down-flow. Long residence times are obtained using multi-belt dryers (e.g., 3 belts), which run in opposite directions.

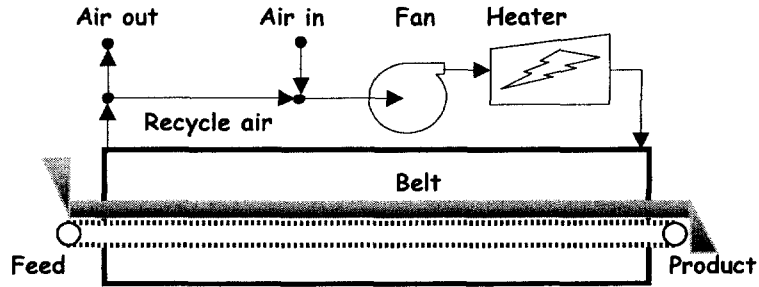


Figure 7.6 Single-belt conveyor dryer.

e. Rotary Dryers

The rotary dryers consist of an inclined long cylinder rotating slowly, while the material (grains, granules, powders) flows with the tumbling (cascading) action of the internal flights (Figure 7.7). The air is heated either by heat exchangers or by mixing with combustion gases of “clean” fuels, e.g., natural gas. Rotary dryers are less expensive than belt dryers, but they cannot handle large food pieces, which may be damaged by mechanical abrasion during tumbling.

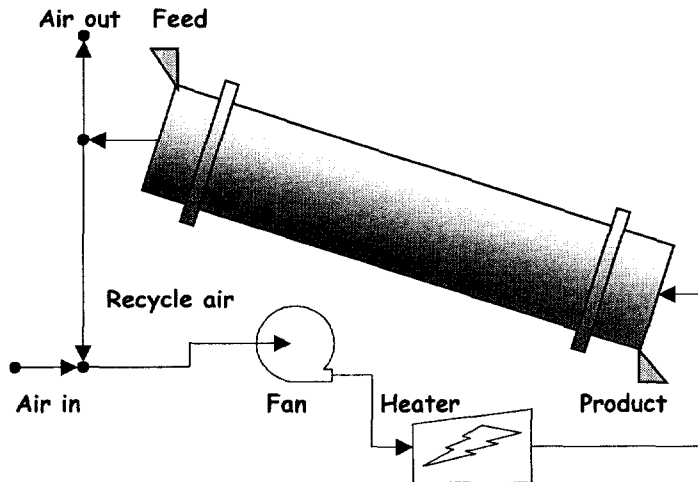


Figure 7.7 Rotary dryer.

f. Fluidized Bed Dryers

Fluidized bed dryers are used for fast drying of food pieces and particles that can be suspended in a stream of hot air (Figure 7.8). High drying rates are obtained due to high heat and mass transfer.

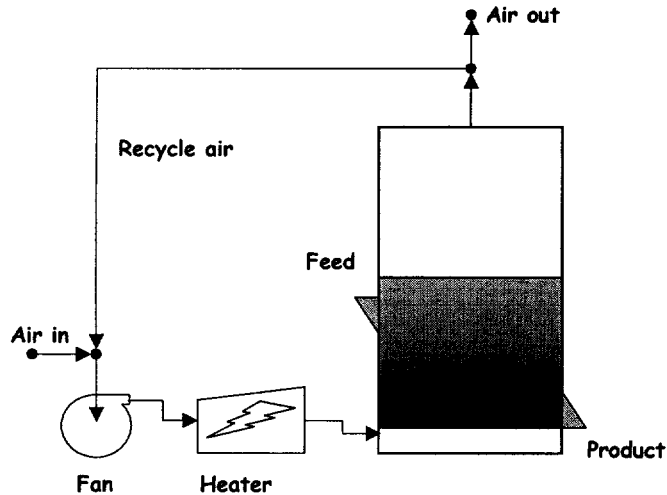


Figure 7.8 Fluidized bed dryer.

g. Pneumatic (Flash) Dryers

Pneumatic (flash) dryers are used for fast and efficient drying of food particles that can be transported by a stream of the heating air (Figure 7.9). The residence time in pneumatic dryers is much shorter than in fluidized bed units.

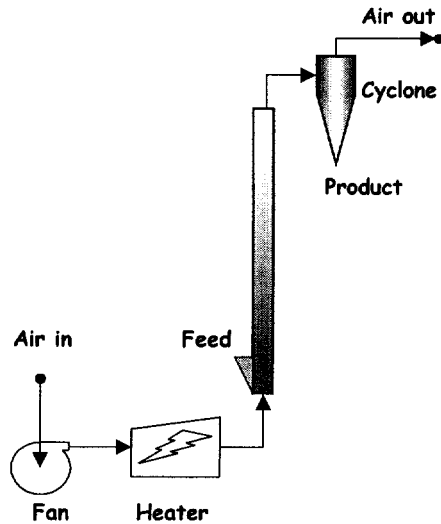


Figure 7.9 Pneumatic dryer.

h. Spray Dryers

Spray dryers are used to dehydrate fast liquid foods or food suspensions into dry powders or agglomerates. The liquid feed is atomized in special valves and the droplets are dried by hot air as they fall in a large chamber (Figure 7.10). The flow of hot air can be co-current or counter-current to the flow (fall) of the droplets and dried particles. The dryers are equipped with cyclone collectors and bag filters to collect the small particles from the exhaust air/gases, and prevent air pollution. Spray dryers are usually combined with agglomeration equipment, which produces food agglomerates of desirable quality (Masters, 1991; Filkova and Mujumdar, 1995).

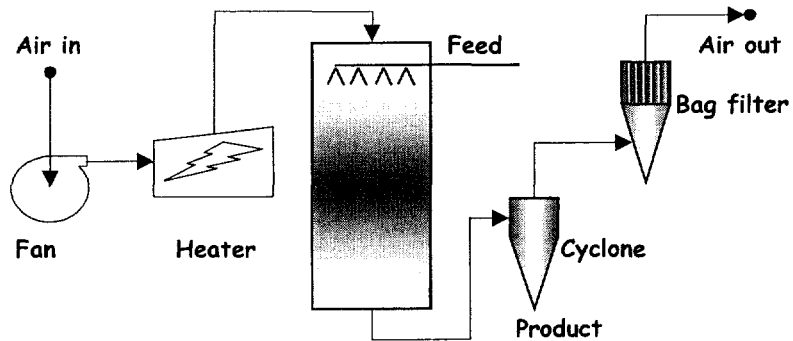


Figure 7.10 Spray dryer (co-current flow).

i. Drum dryers

Drum dryers are used to dehydrate concentrated liquid foods/suspensions and food pastes. They consist of one or two slowly rotating drums, heated internally by steam, with the product dried on the cylindrical surface (Figure 7.11). They are more efficient thermally than convective (air) dryers and they are operated either at atmospheric pressure or in vacuum.

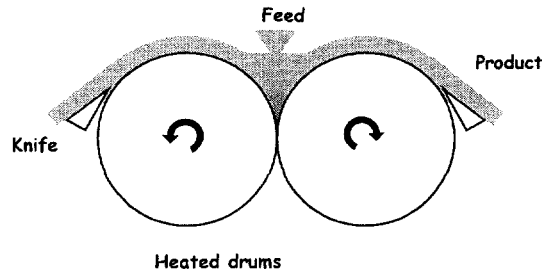


Figure 7.11 Double drum dryer.

k. Vacuum Dryers

Vacuum dryers are used for the dehydration of heat-sensitive food products, such as fruit juices. They operate at pressures of about 10 mbar and temperatures around 10°C (drying from the liquid state). They require vacuum pumping and low-temperature condensing equipment. Heat transfer is by contact to a heated shelf, infrared radiation or microwaves. The product is dried either in trays or in a belt (Figure 7.12). Both batch and continuous operating systems are used.

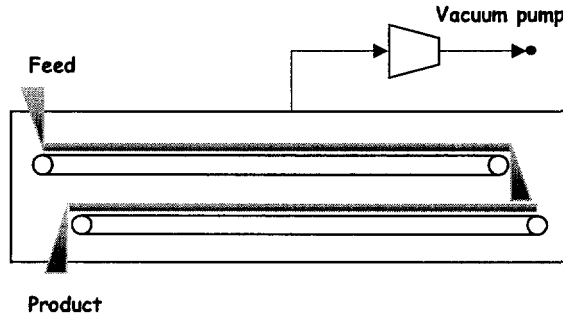


Figure 7.12 Continuous (belt) vacuum dryer.

l. Freeze Dryers

Freeze dryers are the most expensive drying equipment, and they are justified economically only for drying certain expensive food products of unique quality. They operate at pressures below 1 mbar and temperatures below -10°C (drying from the frozen state). The pre-frozen product is placed in trays (Figure 7.13), and heating is by contact, infrared radiation or microwaves. Usually batch freeze dryers are used, but there are some semi-continuous systems for large-scale operation (Liapis and Bruttini, 1995; Oetjen, 1999).

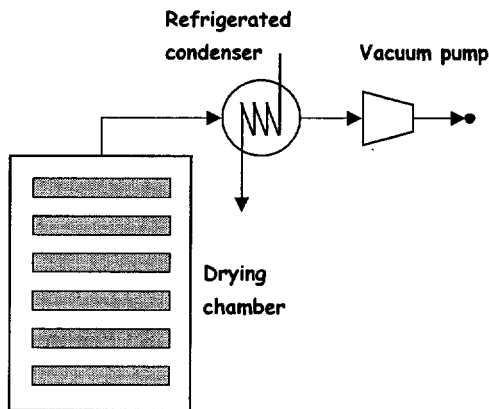


Figure 7.13 Batch freeze-dryer.

IV. DRYING PRINCIPLES

Most of the drying processes are carried out in an air stream (convective drying), and drying calculations are based on the properties of air/water vapor mixtures (Psychrometrics), and the drying rate of the material. Specialized drying processes, such as contact drying and vacuum or freeze drying, require special treatment of heat and mass transfer, but the analysis of drying rate is the same. The physical and engineering principles of drying are discussed by Pakowski and Mujumdar (1995).

1. Psychrometrics

The properties of air/water vapor mixtures are of fundamental importance to drying processes. Psychrometrics deals with the determination of the thermodynamic properties of moist air, based on some given data, and utilization of these properties in process analysis and design. The set of equations describing the thermodynamic relationships between these properties constitutes the psychrometric model. A simple but powerful model for process design purposes can be built as follows:

Moist air is considered to be a mixture of independent perfect gases, dry air and water vapor, each assumed to obey the perfect gas equation of state:

$$P_A V = (m_A / M_A) R T \quad (7-11)$$

$$P_V V = (m_V / M_W) R T \quad (7-12)$$

where P_A and P_V are the partial pressure of dry air and water vapor, V is the total mixture volume, m_A and m_V are masses of air and water vapor, M_A and M_W the molecular weights of air and water, R is the universal gas constant, and T is the absolute temperature.

The total pressure P of the mixture obeys the Dalton law:

$$P = P_A + P_V \quad (7-13)$$

The absolute humidity of the air Y_V and the water activity a_w (relative humidity) are defined as follow:

$$Y_V = m_V / m_A \quad (7-14)$$

$$a_w = P_V / P_S \quad (7-15)$$

where P_S (bar) is the vapor pressure of water at saturation which is calculated versus the temperature T (K) using the Antoine equation (Reid et al., 1987):

$$P_S = \exp [a_1 - a_2 / (a_3 + T)] \quad (7-16)$$

where a_1 , a_2 , and a_3 are empirical adjustable constants.

Equations (7-11) to (7-16) can be combined in the following psychrometric equation:

$$Y_V = m a_w P_S / (P - a_w P_S) \quad (7-17)$$

where m is the air / water molecular weight ratio:

$$m = M_W / M_A \quad (7-18)$$

The humidity of saturation Y_S is obtained from Equation (7-17) when $a_w = 1$ and $Y_V = Y_S$:

$$Y_S = m P_S / (P - P_S) \quad (7-19)$$

The boiling temperature T_b is obtained from Equation (7-16) when $P_S = P$ and $T = T_b$:

$$T_b = -a_3 + a_2 / (a_1 - \ln P) \quad (7-20)$$

The dew point temperature T_d is obtained by combined Equations (7-16) and (7-17) when $Y_S = Y$ and $T = T_d$:

$$T_d = a_2 / (a_1 - \ln[Y P / (m + Y)]) - a_3 \quad (7-21)$$

The dew point pressure P_d is obtained by combined Equations (7-16) and (7-17) when $Y_S = Y$ and $P = P_d$:

$$P_d = \exp [a_1 - a_2 / (a_3 + T)] \quad (7-22)$$

The enthalpy of the moist air H is calculated using the equation:

$$H = C_{PA} T + Y_V (\Delta H_0 + C_{PV} T) + Y_L C_{PL} T \quad (7-23)$$

where C_{PA} , C_{PL} , and C_{PV} are the specific heats of air, water, and vapor, respectively, ΔH_0 is the latent heat of evaporation of water at 0°C, Y_V is the mixture content in water vapor, and Y_L is the mixture content in liquid water.

Generally, the total humidity Y refers to both water vapor Y_V and liquid water Y_L as so:

$$Y_V = \min (Y, Y_S) \quad (7-24)$$

$$Y_L = Y - Y_V \quad (7-25)$$

Equation (7-24) means that (at given temperature) air cannot take up more than a certain amount of vapor. Liquid droplets then precipitate due to over-saturation, and this is called the cloud or fog state.

It is obvious from Equations (7-24) and (7-25) that if the mixture

- is under-saturated ($Y < Y_S$) then $Y_V = Y$ and $Y_L = 0$
- is saturated ($Y = Y_S$) then $Y_V = Y_S$ and $Y_L = 0$
- is over-saturated ($Y > Y_S$) then $Y_V = Y_S$ and $Y_L = Y - Y_V$

The adiabatic saturation temperature, which (for water/air mixtures) is equal to the wet-bulb temperature T_w , can be obtained from the following equation:

$$(Y_V - Y_w) / (T - T_w) = -C_p / \Delta H_S \quad (7-26)$$

where P_w and Y_w are the vapor pressure and absolute humidity at saturation at temperature T_w :

$$P_w = \exp [a_1 - a_2 / (a_3 + T_w)] \quad (7-27)$$

$$Y_w = m P_w / (P - P_w) \quad (7-28)$$

The specific heat of moist air C_p and the latent heat of water vaporization ΔH_S in Equation (7-16) can be estimated using the following equations:

$$C_p = C_{pA} + Y_V C_{pV} \quad (7-29)$$

$$\Delta H_S = \Delta H_0 - (C_{pL} - C_{pV}) T \quad (7-30)$$

The proposed psychrometric model consists of Equations (7-17) through (7-30) and is summarized in Table 7.2. The involved variables are also summarized in Table 7.3, while the appropriate air-water constants are presented in Table 7.4.

A degree-of-freedom analysis, presented in Table 7.5, results in three free variables. Thus three variables are required to describe the state of the mixture. If three variables are known all other variables can be obtained using the psychrometric model. The most common problem is described in Table 7.6.

A psychrometric chart is a graphical representation of the thermodynamic properties of moist air. Its distinctive features are of practical value in solving engineering problems. The choice of coordinates for a psychrometric chart is arbitrary and consequently different psychrometric charts exist.

Four different psychrometric charts are presented in Figures 7.14, 7.15, 7.16, and 7.17. The corresponding calculation procedures are presented in Table 7.7.

- Humidity versus temperature for various water activities at constant pressure
 $Y = F(T, a_w, P)$, see Figure 7.14.
- Humidity at saturation versus temperature for various pressures
 $Y_S = F(T, P)$, see Figure 7.15
- Enthalpy versus temperature for various humidities at constant pressure
 $H = F(T, Y, P)$, see Figure 7.16

- Enthalpy versus humidity for various water activities at constant pressure
 $H = F(Y, a_w, P)$, see Figure 7.17

Figure 7.18 represents some characteristic humidities (Y) and temperatures (T) of a mixture in the most common psychrometric chart.

Table 7.2 Psychrometric Model

$$P_S = \exp [a_1 - a_2 / (a_3 + T)] \quad (\text{E01})$$

$$Y_S = m P_S / (P - P_S) \quad (\text{E02})$$

$$Y_V = \min (Y, Y_S) \quad (\text{E03})$$

$$Y_L = Y - Y_V \quad (\text{E04})$$

$$Y_V = m a_w P_S / (P - a_w P_S) \quad (\text{E05})$$

$$H = C_{PA} T + Y_V (\Delta H_0 + C_{PV} T) + Y_L C_{PL} T \quad (\text{E06})$$

$$T_b = -a_3 + a_2 / (a_1 - \ln P) \quad (\text{E07})$$

$$T_d = a_2 / (a_1 - \ln [Y P / (m + Y)]) - a_3 \quad (\text{E08})$$

$$P_d = \exp [a_1 - a_2 / (a_3 + T)] \quad (\text{E09})$$

$$P_w = \exp [a_1 - a_2 / (a_3 + T_w)] \quad (\text{E10})$$

$$Y_w = m P_w / (P - P_w) \quad (\text{E11})$$

$$C_P = C_{PA} + Y_V C_{PV} \quad (\text{E12})$$

$$\Delta H_S = \Delta H_0 - (C_{PL} - C_{PV}) T \quad (\text{E13})$$

$$(Y_V - Y_W) / (T - T_W) = -C_P / \Delta H_S \quad (\text{E14})$$

Table 7.3 Variables Involved in the Psychrometric Model

| | | |
|--------------|-----------|---|
| P | bar | Pressure |
| P_d | bar | Dew pressure |
| P_s | bar | Vapor pressure at temperature T |
| P_w | bar | Vapor pressure at temperature T_w |
| T | °C | Temperature |
| T_b | °C | Boiling temperature |
| T_d | °C | Dew temperature |
| T_w | °C | Wet bulb temperature |
| Y | kg/kg db | Total humidity (liquid + vapor) |
| Y_L | kg/kg db | Humidity in liquid |
| Y_V | kg/kg db | Humidity in vapor |
| Y_s | kg/kg db | Saturation humidity at temperature T |
| Y_w | kg/kg db | Saturation humidity at temperature T_w |
| a_w | - | Water activity |
| H | kJ/kg db | Enthalpy of humid air |
| C_p | kJ/kgK db | Specific heat of humid air |
| ΔH_S | kJ/kg | Latent heat of condensation of water vapor at temperature T |

Table 7.4 Psychrometric Data for Air-Water Vapor Mixture

| | | |
|--------------|------------------------|---|
| R | = 8.31 kJ/kmol K | Ideal gas constant |
| m | = 0.622 | Air/water molecular weight ratio |
| C_{pA} | = 1.00 kJ/kg K | Specific heat of air |
| C_{pV} | = 1.90 kJ/kg K | Specific heat of water vapor |
| C_{pW} | = 4.20 kJ/kg K | Specific heat of liquid water |
| ΔH_0 | = 2.50 MJ/kg | Latent heat of water evaporation at 0°C |
| a_1 | = 1.19 10 ¹ | Antoine equation constants for water |
| a_2 | = 3.99 10 ³ | |
| a_3 | = 2.34 10 ² | |

Table 7.5 Degrees-of-Freedom Analysis

| | |
|-------------------------|-----------|
| Psychrometric variables | 17 |
| <u>Model equations</u> | <u>14</u> |
| Degrees of freedom | 3 |

Table 7.6 Most Common Psychrometric CalculationsGiven variables: P , T , Y .

| | | | | |
|-------|---|--------------|-----------------|---|
| (E01) | → | P_s | | |
| (E02) | → | Y_s | | |
| (E03) | → | Y_V | | |
| (E04) | → | Y_L | | |
| (E05) | → | a_w | | |
| (E06) | → | H | | |
| (E07) | → | T_b | | |
| (E08) | → | T_d | | |
| (E09) | → | P_d | | |
| | | T_w | trial value | ← |
| (E10) | → | P_w | | |
| (E11) | → | Y_w | | |
| (E12) | → | C_p | | |
| (E13) | → | ΔH_S | | |
| (E14) | → | T_w | corrected value | → |

Table 7.7 Construction Procedure of Psychrometric Charts

$$Y_V = F(T, a_w, P)$$

| | | |
|-------|---|-------|
| (E01) | → | P_s |
| (E05) | → | Y_V |

$$Y_s = F(T, P)$$

| | | |
|-------|---|-------|
| (E01) | → | P_s |
| (E02) | → | Y_s |

$$H = F(T, Y, P)$$

| | | |
|-------|---|-------|
| (E01) | → | P_s |
| (E02) | → | Y_s |
| (E03) | → | Y_V |
| (E04) | → | Y_L |
| (E06) | → | H |

$$H = F(Y_V, a_w, P)$$

| | | |
|-------|---|-------|
| (E05) | → | P_s |
| (E01) | → | T |
| (E06) | → | H |

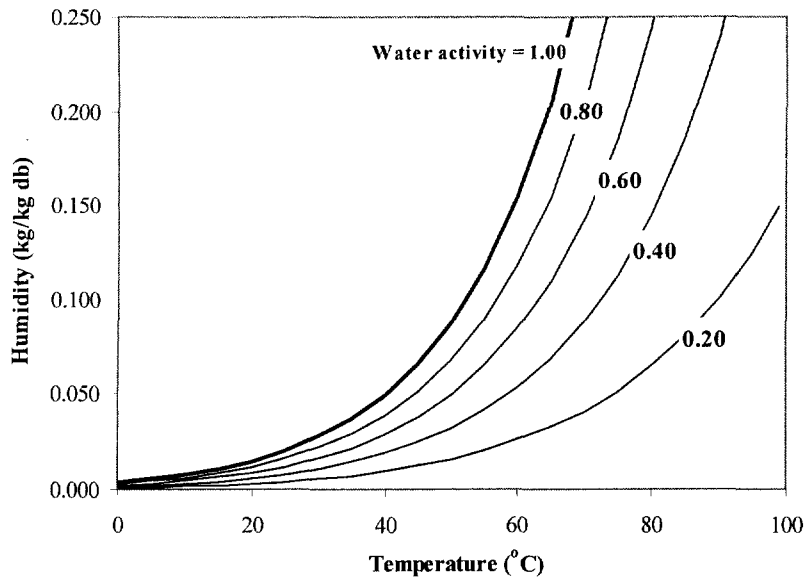


Figure 7.14 Psychrometric Chart: Humidity versus temperature at various water activities. Pressure = 1 bar.

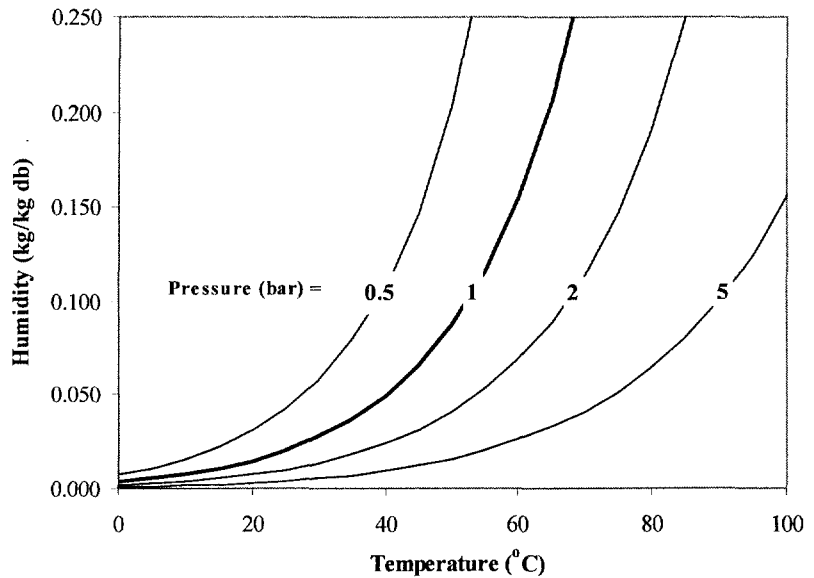


Figure 7.15 Psychrometric Chart: Humidity at saturation versus temperature at various pressures. Water activity = 1.

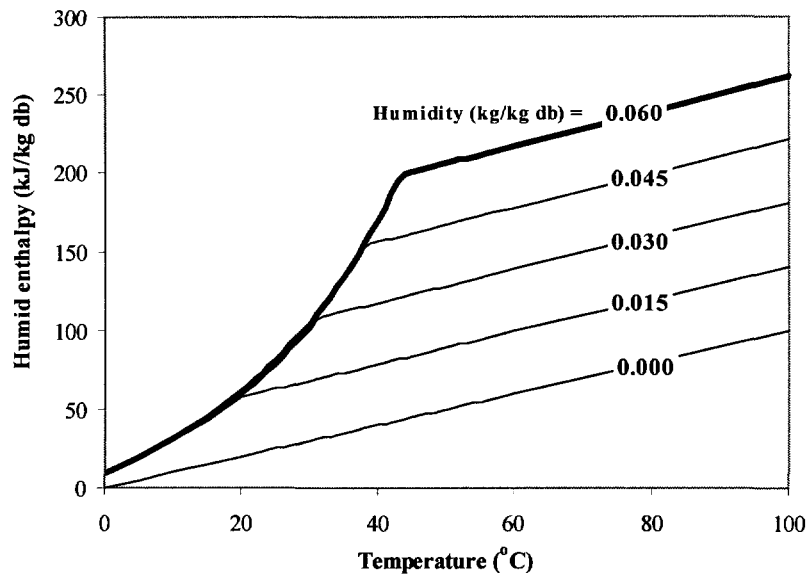


Figure 7.16 Humid air specific enthalpy versus temperature at various humidities. Pressure = 1 bar.

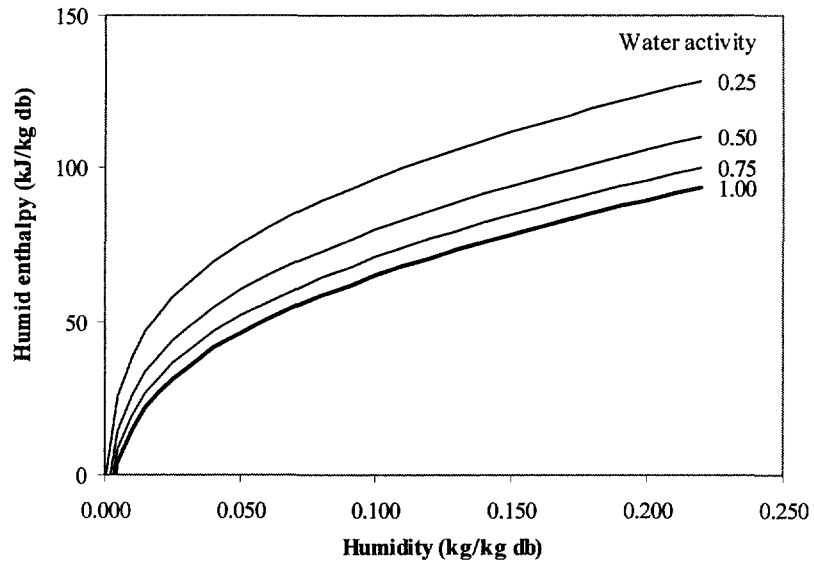


Figure 7.17 Humid air specific enthalpy versus humidity at various water activities. Pressure = 1 bar.

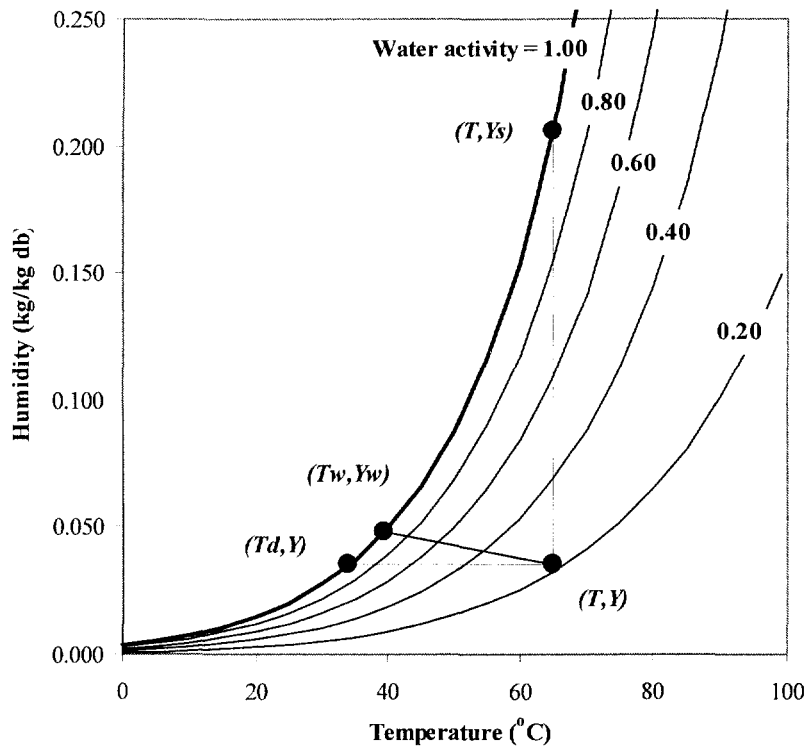


Figure 7.18 Psychrometric chart and characteristic temperatures and humidities. The corresponding numeric values are summarized in Table 7.8.

Table 7.8 Numerical Values of the Characteristic Variables Presented in Figure 7.18

| | | |
|-------|------------------|--|
| P | = 1 bar | Pressure |
| T | = 65 °C | Temperature |
| Y | = 0.035 kg/kg db | Humidity |
| a_w | = 0.214 | Water activity |
| T_d | = 34.0 °C | Dew temperature |
| T_w | = 39.4 °C | Wet bulb temperature |
| Y_s | = 0.206 kg/kg db | Saturation humidity at temperature T |
| Y_w | = 0.048 kg/kg db | Saturation humidity at temperature T_w |

2. Drying Kinetics

The drying rate of a product depends on the material characteristics and the drying air conditions. The drying rate may indicate initially a constant rate period, during which free water is removed. However, food materials usually contain mostly bound water which is removed during the falling rate period.

Independently of the physical drying mechanism, in most cases, a first-order kinetic model is adequate to describe the falling rate drying period for foods:

$$dX/dt = - (1/t_c) (X - X_e) \quad (7-31)$$

where dX/dt is the drying rate, X is the material moisture content at time t , t_c is the drying time constant and X_e is the equilibrium material moisture content.

The drying time constant t_c is used in this book instead of the well known drying constant K :

$$t_c = 1 / K \quad (7-32)$$

The drying time constant is consistent with the process control terminology for first-order dynamic systems and its value has physical meaning denoting the required time to remove about 2/3 of the total moisture.

The drying constant depends on drying air conditions and material characteristics. Various empirical or semi-theoretical models have been proposed in the literature (Marinos-Kouris and Maroulis, 1995). A simple power law model seems adequate for process design purposes. It incorporates the effect of material size d and the drying air conditions, that is air velocity u , temperature T and humidity Y :

$$t_c = c_0 d^{c_1} u^{c_2} T^{c_3} Y^{c_4} \quad (7-33)$$

where c_0, c_1, c_2, c_3, c_4 are adjustable empirical constants, depending on the material characteristics and drying equipment geometry.

The equilibrium material moisture content X_e depends on air temperature T and water activity a_w . Various empirical or semi-theoretical models have been proposed in the literature, but a modified Oswin model seems to be the most appropriate in process design calculations (Iglesias and Chirife, 1983):

$$X_e = b_1 \exp[b_2 / (273 + T)] [a_w / (1 - a_w)]^{b_3} \quad (7-34)$$

where b_1, b_2 , and b_3 are adjustable empirical constants, depending on the material characteristics.

When drying conditions are kept constant, Equation (7-31) is integrated as follows:

$$t = -t_c \ln[(X - X_e) / (X_0 - X_e)] \quad (7-35)$$

Equation (7-35) calculates the required drying time t to remove the moisture from the initial X_0 to the final X material moisture content.

The drying time constant t_c or the drying (rate) constant K can be estimated approximately from the effective moisture diffusivity D in the material, using the simplified Equation (7-36), Saravacos and Kostaropoulos (2002):

$$t_c = 1/K = d^2/\pi^2 D \quad (7-36)$$

where d is the thickness or radius of the material. It is assumed that water is transported by molecular diffusion.

In conclusion, the drying kinetics for most of the foods can be expressed adequately, for process design purposes, using the first-order kinetic model summarized in Table 7.9. It means that eight adjustable parameters are required for each food material, drying conditions and equipment geometry. Some values for a few vegetables are summarized in Table 7.10. Figures 7.19 and 7.20 shows graphically the effect of material and drying conditions on the equilibrium material moisture content and on the drying time constant. Finally, Figure 7.21 reveals the material moisture content versus drying time for carrot under various drying conditions.

Table 7.9 Drying Kinetics

$$t = -t_c \ln[(X - X_e)/(X_o - X_e)] \quad (E01)$$

$$X_e = b_1 \exp[b_2/(273+T)] [a_w/(1-a_w)]^{b_3} \quad (E02)$$

$$t_c = c_0 d^{c_1} u^{c_2} T^{c_3} Y^{c_4} \quad (E03)$$

Table 7.10 Equilibrium and Drying Kinetics Data of Some Vegetables

| | potato | carrot | pepper | onion |
|-------------------------------------|----------|----------|----------|----------|
| <i>Equilibrium moisture content</i> | | | | |
| <i>adjustable constants</i> | | | | |
| b_1 | 0.000327 | 0.000735 | 0.000401 | 0.000651 |
| b_2 | 1840 | 1750 | 1980 | 1830 |
| b_3 | 0.400 | 0.401 | 0.389 | 0.377 |
| <i>Drying time</i> | | | | |
| <i>adjustable constants</i> | | | | |
| c_0 | 0.581 | 0.500 | 0.546 | 0.534 |
| c_1 | 1.49 | 1.40 | 1.07 | 1.26 |
| c_2 | -1.85 | -1.65 | -2.55 | -3.11 |
| c_3 | -0.345 | -0.250 | -0.442 | -0.251 |
| c_4 | 0.151 | 0.120 | 0.087 | 0.160 |

Potatoes and carrots refer to cubes, pepper and onion to strips.

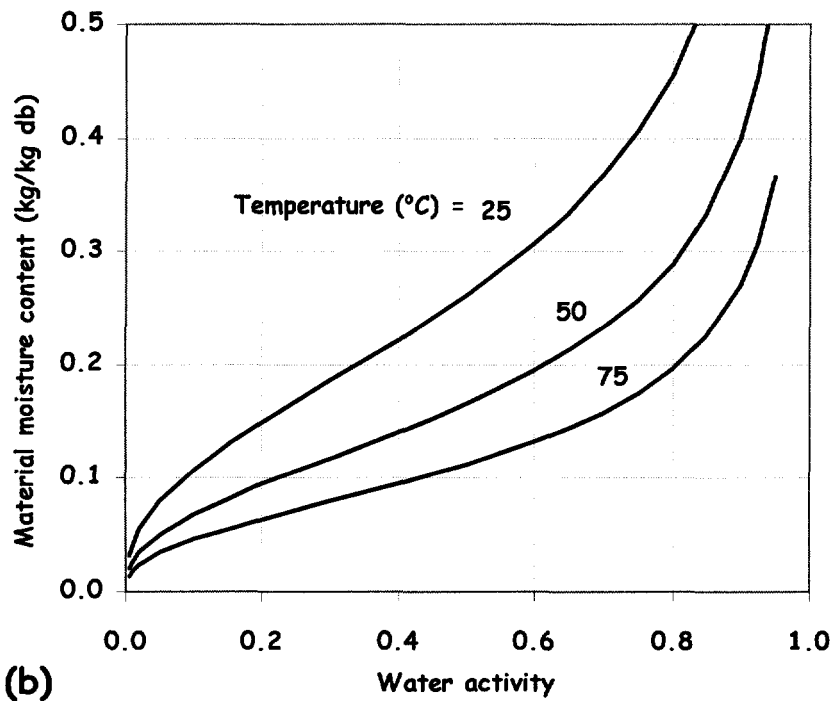
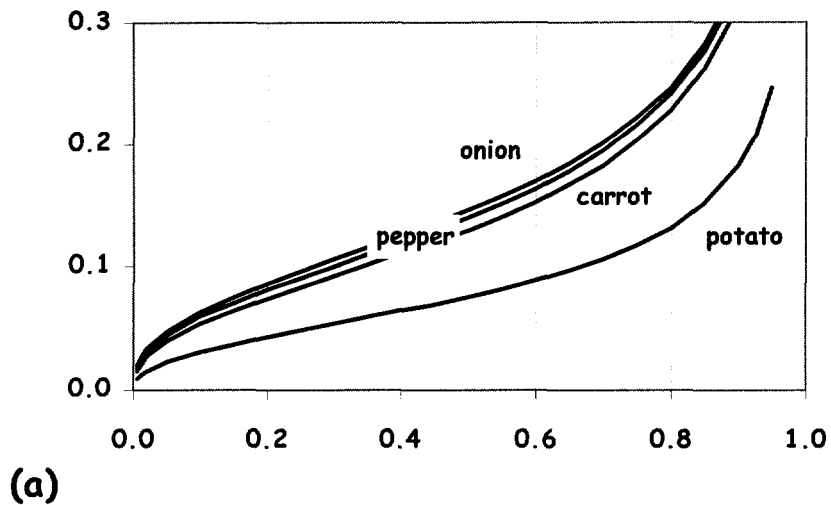
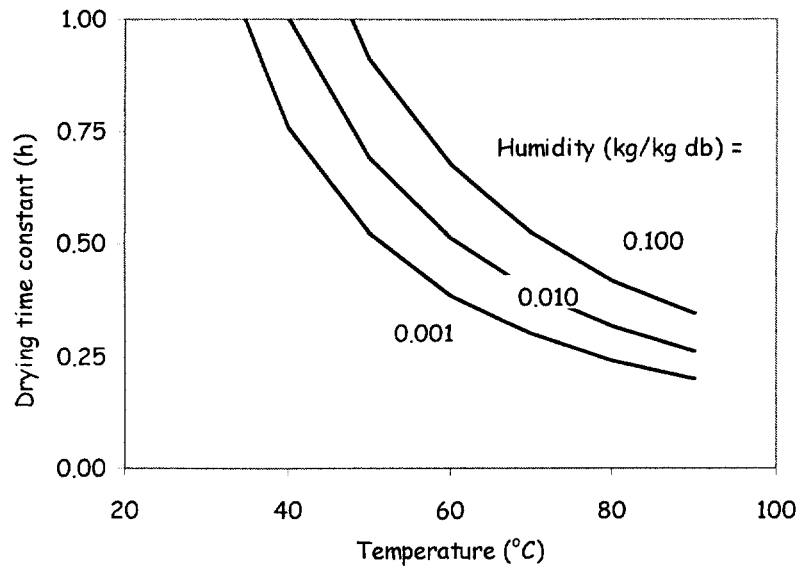
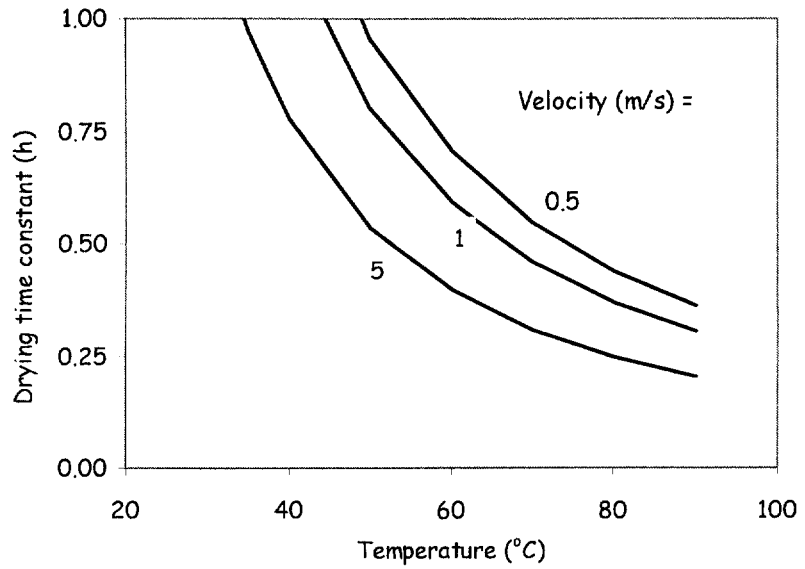


Figure 7.19 Carrot isotherms: (a) Equilibrium material moisture content versus water activity for some vegetables at 65°C. (b) Equilibrium material moisture content versus water activity at various temperatures for carrot.



(a)



(b)

Figure 7.20 Drying kinetics of carrot: Drying constant versus temperature for various (a) air humidities (b) air velocities.

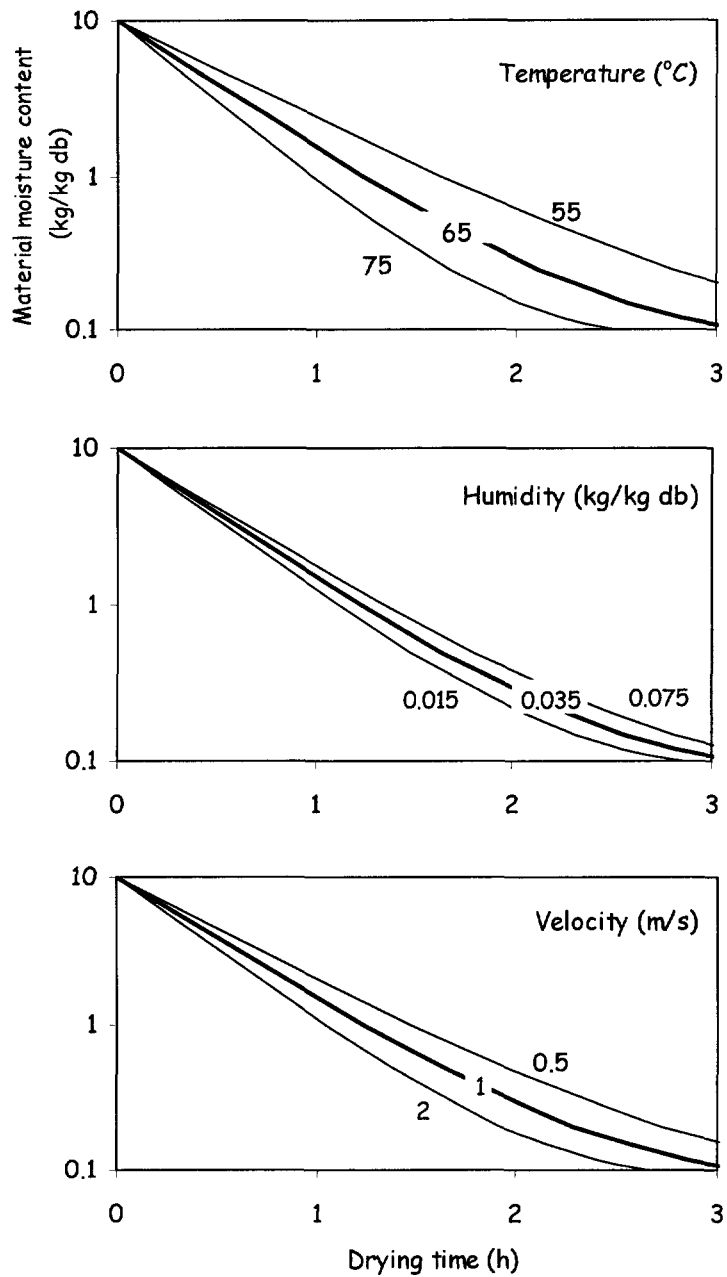


Figure 7.21 Drying curves for carrot at various air conditions. Base case refers to 65°C, 0.035kg/kgdb and 1m/s.

3. Material and Heat Balances

Figure 7.22 is a schematic representation of a typical air dryer. It denotes the input and output material and energy flows which are needed to formulate the overall material and heat balances. Wet material enters in the dryer with a flow rate F , moisture x_o and temperature T_o and exits with the same flow rate F in dry basis, moisture X and temperature T . Fresh air enters the dryer with a flow rate F_a , moisture Y_o and temperature T_o and exits with the same flow rate F_a in dry basis, moisture Y and temperature T . It is assumed that both streams, solid and air, leave the dryer in thermal equilibrium, that is at the same temperature. Q is the thermal energy input to the dryer.

The moisture balance of the dryer is expressed using the following equations, which suggest that the moisture removed from the solid W is held up by the air:

$$W = F (X_o - X) \quad (7-37)$$

$$W = F_a (Y - Y_o) \quad (7-38)$$

The energy balance of the dryer suggests that the thermal energy input to the dryer Q is used to heat the solid material Q_{sh} and the fresh air Q_{ah} and to evaporate the moisture Q_{we} :

$$Q = Q_{we} + Q_{sh} + Q_{ah} \quad (7-39)$$

$$Q_{we} = F (X_o - X) [\Delta H_o - (C_{PL} - C_{PV}) T] \quad (7-40)$$

$$Q_{sh} = F [C_{PS} + X_o C_{PL}] (T - T_o) \quad (7-41)$$

$$Q_{ah} = F_a [C_{PA} + Y_o C_{PV}] (T - T_o) \quad (7-42)$$

It can be proved that the same thermal load Q can be obtained by the alternative formulation of the energy balance:

$$Q = H - H_o + h - h_o \quad (7-43)$$

$$H = C_{PA} T + Y (\Delta H_o + C_{PV} T) \quad (7-44)$$

$$H_o = C_{PA} T_o + Y_o (\Delta H_o + C_{PV} T_o) \quad (7-45)$$

$$h = (C_{PS} + X C_{PL}) T \quad (7-46)$$

$$h_o = (C_{PS} + X_o C_{PL}) T_o \quad (7-47)$$

where H_o , H , h_o , h are the enthalpy of the air and solid streams at the dryer inlet and outlet, respectively.

The thermal efficiency of the dryer is defined as follows:

$$n = Q_{we} / Q \quad (7-48)$$

It should be noted that temperature T is in °C and that the basis of enthalpy (H , h) calculations is 0°C.

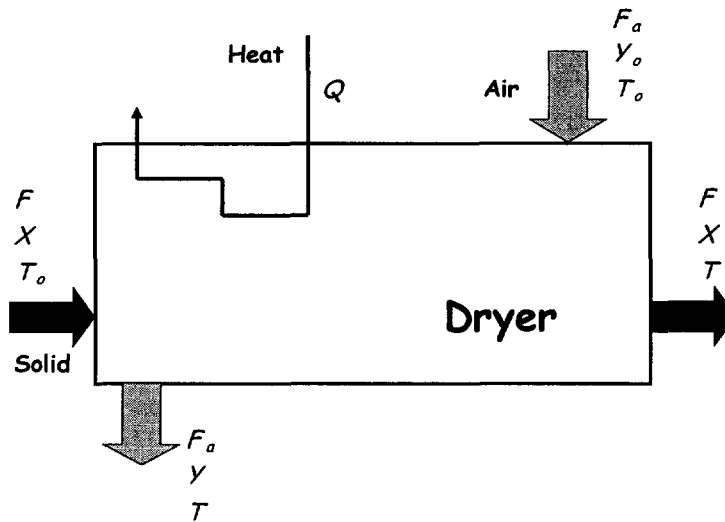


Figure 7.22 Schematic representation of a typical air (convective) dryer.

Table 7.11 Material and Heat Balances

Material balance

$$W = F (X_o - X) \quad (\text{E1})$$

$$W = F_a (Y - Y_o) \quad (\text{E2})$$

Energy (heat) balance

$$Q = Q_{we} + Q_{sh} + Q_{ah} \quad (\text{E3})$$

$$Q_{we} = F (X_o - X) [\Delta H_o - (C_{PL} - C_{PV}) T] \quad (\text{E4})$$

$$Q_{sh} = F [C_{PS} + X_o C_{PL}] (T - T_o) \quad (\text{E5})$$

$$Q_{ah} = F_a [C_{PA} + Y_o C_{PV}] (T - T_o) \quad (\text{E6})$$

Thermal efficiency

$$n = Q_{we} / Q \quad (\text{E7})$$

4. Sizing

The size of a dryer should guarantee that the residence time of the product will be equal to the required drying time. The residence time in a continuous dryer is a crucial point in process design. Generally, the average residence time can be obtained from the equation:

$$t = M / [F (1+X_o)] \quad (7-49)$$

where M is the mass holdup of the dryer, which is correlated with the volume holdup H of the dryer as follows:

$$M = (1-\varepsilon) \rho_s H \quad (7-50)$$

where ε is the empty void in solid dispersion and ρ_s is the solids density.

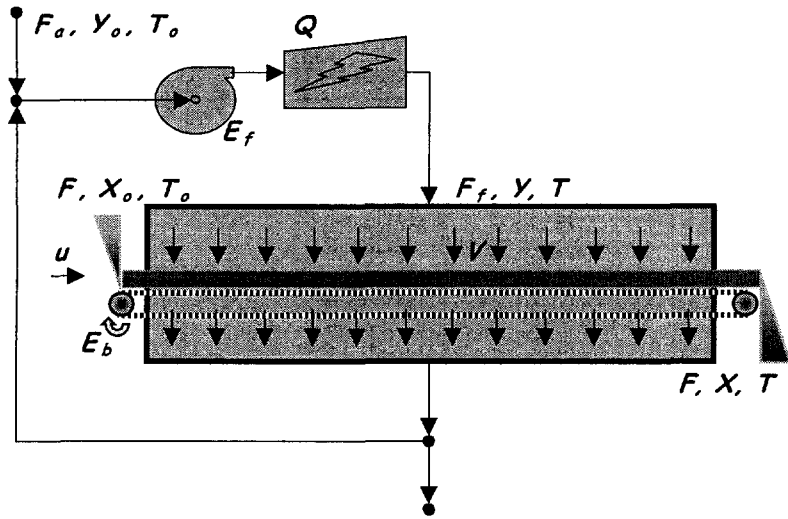
The volume holdup of the dryer H is directly correlated to the dryer size and geometry. The calculation of the volume holdup H of a dryer is different for different dryer types. The conveyor belt dryer as well as the rotary dryer are described in the following paragraphs.

V. DESIGN OF A CONVEYOR BELT DRYER

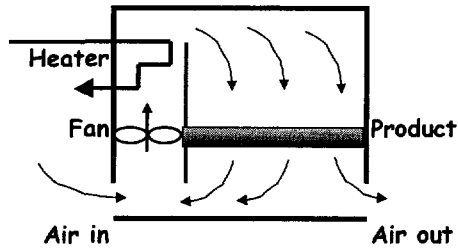
In this section a design approach is described for the conveyor belt dryer (Kira-noudis et al., 1994; Marinos-Kouris et al., 1996).

1. Process Description

A typical flowsheet of a conveyor belt dryer is presented in Figure 7.23. The wet feed with flow rate F (kg/s db), temperature T_o ($^{\circ}$ C) and humidity X_o (kg/kg db) is distributed on the belt as it enters the dryer. The dried product exits the dryer with the same flow rate in dry basis F (kg/s db), temperature T ($^{\circ}$ C) and humidity X (kg/kg db). The belt is moving at a velocity u (m/s) and requires an electrical power E_b (kW). The drying air enters the dryer at a flow rate F_f (kg/s db), temperature T ($^{\circ}$ C) and humidity Y (kg/kg db). The drying air temperature is controlled in the heater, and the drying air humidity is controlled through the flow rate of the fresh air F_o (kg/s db). An electrical power E_f (kW) is expended by the fan and a thermal power Q (kW) is expended by the heater. The air conditions for design can be considered constant due to the high air recirculation, which is shown in the dryer cross-sectional diagram of Figure 7.23b.



(a)



(b)

Figure 7.23 Schematic representation of a belt dryer: (a) Typical flowsheet; (b) Cross-sectional view.

2. Process Model

A mathematical model of the process presented in Figure 7.23 is summarized in Table 7.12. Equation (E01) calculates the vapor pressure at drying temperature, while Equation (E02) is the psychrometric equation described in Section IV.1 of this chapter. Equations (E01) and (E02) are used to calculate the water activity at drying conditions, that is temperature T and air humidity Y . Equation (E03) calculates the equilibrium material moisture content at drying conditions, while Equation (E04) estimates the drying time constant at drying conditions. Both Equations (E03) and (E04) are used in Equation (E05), which calculates the required drying time. Equations (E06) and (E07) constitute the moisture balance at the dryer. Equation (E06) refers to solid, and (E07) to air. The thermal energy requirements for drying are summarized in Equations (E08) through (E11). Equation (E08) refers to water evaporation, (E09) to solids heating, (E10) to rejected air heating, and (E11) refers to the total energy required by the heater. Equation (E12) is used for sizing the heater. Equations (E13) through (E17) are used for sizing the belt. Equation (E13) correlates the residence time with the mass holdup, and Equation (E14) the mass holdup with the volume hold up. These equations are valid for all dryer types as described in Section IV.4 in this chapter. Equation (E15) is the geometrical distribution of the volume hold up on the belt. Equation (E16) calculates the required belt area, and Equation (E17) the required belt velocity to obtain the desired residence time. Equations (E18) through (E20) are used for sizing the fan. Equation (E18) calculates the pressure loss of air through the loaded belt. Equation (E19) correlates the air flow with the air velocity. Equation (E20) estimates the required electrical power to operate the fan. Equation (E21) estimates the required electrical power to move the belt. Equation (E22) calculates the required total electrical power. Finally, Equations (E23) and (E24) define two crucial dryer performance indices. Equation (E23) defines the dryer thermal performance, while (E24) calculates the evaporating capacity per unit belt area.

Thirty seven variables presented in Table 7.13 are involved in the model of twenty four equations presented in Table 7.12. The corresponding technical data are summarized in Table 7.14. The process specifications of a typical design problem is presented in Table 7.15, while a degrees-of-freedom analysis is shown in Table 7.16, which results in four design variables. Table 7.17 suggests a selection of design variables and the corresponding solution algorithm is presented in Table 7.18. The total annualized cost TAC presented in Table 7.19 is used as objective function in process optimization. The required cost data are summarized in Table 7.20.

Table 7.12 Belt Dryer Model**Psychrometric equations**

$$P_S = \exp [a_1 - a_2 / (a_3 + T)] \quad (\text{E01})$$

$$Y = m a_w P_S / (P - a_w P_S) \quad (\text{E02})$$

Drying kinetics

$$X_e = b_1 \exp[b_2 / (273 + T)] [a_w / (1 - a_w)]^{b_3} \quad (\text{E03})$$

$$t_c = c_0 a^{c_1} u^{c_2} T^{c_3} Y^{c_4} \quad (\text{E04})$$

$$t = -t_c \ln[(X - X_e) / (X_o - X_e)] \quad (\text{E05})$$

Material balance

$$W = F (X_o - X) \quad (\text{E06})$$

$$W = F_a (Y - Y_o) \quad (\text{E07})$$

Thermal energy requirements

$$Q_{we} = F (X_o - X) [\Delta H_o - (C_{PL} - C_{PV}) T] \quad (\text{E08})$$

$$Q_{sh} = F [C_{PS} + X_o C_{PL}] (T - T_o) \quad (\text{E09})$$

$$Q_{ah} = F_a [C_{PA} + Y_o C_{PV}] (T - T_o) \quad (\text{E10})$$

$$Q = Q_{we} + Q_{sh} + Q_{ah} \quad (\text{E11})$$

Air heater

$$Q = A_s U_s (T_s - T) \quad (\text{E12})$$

Belt dryer

$$M = t F (1 + X_o) \quad (\text{E13})$$

$$M = (1 - \varepsilon) \rho_s H \quad (\text{E14})$$

$$H = Z_o D L \quad (\text{E15})$$

$$A_b = L D \quad (\text{E16})$$

$$u_b = L / t \quad (\text{E17})$$

Fan

$$\Delta P = f_l Z_o V^2 \quad (\text{E18})$$

$$F_f = \rho_a V D L \quad (\text{E19})$$

$$E_f = \Delta P F_f / \rho_a \quad (\text{E20})$$

Belt driver

$$E_b = e_l L (1 + X_o) F \quad (\text{E21})$$

Electrical energy requirements

$$E = E_b + E_f \quad (\text{E22})$$

Performance indices

$$n = Q_{we} / Q \quad (\text{E23})$$

$$r = W / A_b \quad (\text{E24})$$

Table 7.13 Process Variables

| | | |
|------------------------|---------------------|---|
| Drying air | | |
| F_a | ton/h | Fresh air flow rate |
| F_f | ton/h | Recycle air flow rate |
| T | °C | Drying air temperature |
| Y | kg/kg db | Drying air humidity |
| V | m/s | Drying air velocity |
| P | bar | Drying pressure |
| T_o | °C | Ambient temperature |
| Y_o | kg/kg db | Ambient humidity |
| P_s | bar | Vapor pressure at drying conditions |
| a_w | - | Water activity at drying conditions |
| Material | | |
| F | ton/h | Material flow rate |
| X_o | kg/kg db | Initial moisture content |
| X | kg/kg db | Final moisture content |
| X_e | kg/kg db | Equilibrium moisture content at drying conditions |
| d | m | Particle size |
| t_c | h | Drying time constant at drying conditions |
| t | h | Drying time |
| Dryer | | |
| W | ton/h | Drying rate |
| L | m | Dryer length |
| D | m | Dryer width |
| M | ton | Dryer mass holdup |
| H | m ³ | Dryer volume holdup |
| A_b | m ² | Belt area |
| A_s | m ² | Air heater transfer area |
| u_b | m/s | Belt velocity |
| Z_o | m | Loading depth |
| ΔP | bar | Pressure loss of air flowing through belt |
| Thermal Load | | |
| Q_{we} | kW | Water vaporization |
| Q_{sh} | kW | Solid heating |
| Q_{ah} | kW | Air heating |
| Q | kW | Total thermal load |
| T_s | °C | Steam temperature |
| Electrical Load | | |
| E_b | kW | Belt driver |
| E_f | kW | Fan |
| E | kW | Total power requirement |
| Performance | | |
| n | - | Thermal efficiency |
| r | kg/h m ² | Specific rate of evaporation |

Table 7.14 Process Data

| | |
|---------------------------|---|
| | Density (kg/m³) |
| ρ_w | Water |
| ρ_a | Air |
| ρ_s | Dry material |
| | Specific heat (kJ/kgK) |
| C_{PL} | Water |
| C_{PV} | Water vapor |
| C_{PA} | Air |
| C_{PS} | Dry material |
| | Latent heat (kJ/kg) |
| ΔH_o | Steam condensation at 0°C |
| | Other |
| U_s | Heat transfer coefficient at air heater (kW/m ² K) |
| ε | Void (empty) fraction of loading |
| | Empirical constants |
| a_1, a_2, a_3 | Antoine equation for vapor pressure of water |
| b_1, b_2, b_3 | Oswin equation for material isotherms |
| c_0, c_1, c_2, c_3, c_4 | Drying kinetics equation |
| e_j | Belt driver power equation |
| f_l | Pressure loss equation |

Concerning the design variables, the following points should be considered: One variable refers to geometry: Standard commercial belt widths are usually used. The remainder three variables consist the drying air conditions: Drying air humidity, temperature and velocity.

When high values of temperature are used, the drying time and consequently the dryer size decrease, but the energy losses of the rejected air stream increase. Thus, lower equipment cost and higher operating cost are obtained. It must also be noted that most of the food components are damaged at high temperatures.

When lower values of humidity are used, the drying time and consequently the dryer size decreases, but the required fresh air increases and consequently the energy losses of the rejected air stream increase. Thus, lower equipment cost and higher operating cost are obtained.

When higher values of air velocity are used, the drying time and consequently the dryer size decrease, but the required fan power increases. Thus, lower equipment cost and higher operating cost are obtained.

Table 7.15 Process Specifications

| | | |
|-------|----------|-----------------------------------|
| F | ton/h db | Feed flow rate |
| X_o | kg/kg db | Initial material moisture content |
| X | kg/kg db | Final material moisture content |
| d | m | Material characteristic size |
| T_o | °C | Ambient temperature |
| Y_o | kg/kg db | Ambient humidity |
| Z_o | m | Loading depth |
| P | bar | Ambient pressure |
| T_s | °C | Heating steam temperature |

Table 7.16 Degrees-of-Freedom Analysis

| | | | |
|--------------------|----|--------------------|----|
| Process variables | 37 | Degrees of freedom | 13 |
| Process equations | 24 | Specifications | 9 |
| Degrees of freedom | 13 | Design variables | 4 |

Table 7.17 Design Variables

| | | |
|-----|----------|------------------------|
| Y | kg/kg db | Drying air humidity |
| T | °C | Drying air temperature |
| V | m/s | Drying air velocity |
| D | m | Belt width |

Table 7.18 Model Solution Algorithm

| | | | | |
|-------|---|------------|--------------------|------------------|
| (E01) | → | P_s | | |
| (E02) | → | a_w | | |
| (E03) | → | X_c | | |
| (E04) | → | t_c | | |
| (E05) | → | t | | |
| (E06) | → | W | | |
| (E07) | → | F_a | | |
| (E08) | → | Q_{we} , | (E09) → Q_{sh} , | (E10) → Q_{ah} |
| (E11) | → | Q | | |
| (E12) | → | A_s | | |
| (E13) | → | M | | |
| (E14) | → | H | | |
| (E15) | → | L | | |
| (E16) | → | A_b | | |
| (E17) | → | u_b | | |
| (E18) | → | ΔP | | |
| (E19) | → | F_f | | |
| (E20) | → | E_f , | (E21) → E_b | |
| (E22) | → | E | | |
| (E23) | → | n | | |
| (E24) | → | r | | |

Table 7.19 Cost Analysis**Equipment cost**

$$C_{eq} = C_{bel} A^{n_{bel}} + C_{exc} A_s^{n_{exc}} + C_{fan} E_f^{n_{fan}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_s Q + C_e E) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (F04)$$

Table 7.20 Cost Data**Utility cost**

C_e \$/kWh Cost of electricity
 C_s \$/kWh Cost of heating steam

Equipment unit cost

C_{bel} \$/m² Belt dryer
 C_{exc} \$/m² Heat exchanger
 C_{fan} \$/kW Fan

Equipment size scaling factor

n_{bel} - Belt dryer
 n_{exc} - Heat exchanger
 n_{fan} - Fan

Other

t_y h/yr Annual operating time
 i_r - Interest rate
 l_f yr Lifetime

3. Application to Carrot Dehydration

Dehydrated fruits are usually used in mixtures with cereals, while dehydrated vegetables are used in mixtures with rice in instant soups. Belt dryers are used in multi-product dehydration plants. A typical application to carrot dehydration is described in this section. The process specifications, the required technical and economic data are summarized in Tables 7.21 through 7.23. The optimal values of the design variables are presented in Table 7.24, while the corresponding technical and economic results are summarized in Tables 7.25 and 7.26.

The results are further investigated through a sensitivity analysis of the design variables on the cost (Figure 7.24), the dryer size (Figure 7.25) and the required thermal load (Figure 7.26).

Table 7.21 Process Data**Density (kg/m³)**

| | | |
|----------|--------|--------------|
| ρ_w | = 1000 | Water |
| ρ_a | = 1 | Air |
| ρ_s | = 1750 | Dry material |

Specific heat (kJ/kgK)

| | | |
|----------|--------|--------------|
| C_{PL} | = 4.20 | Water |
| C_{PV} | = 1.90 | Water vapor |
| C_{PA} | = 1.00 | Air |
| C_{PS} | = 2.00 | Dry material |

Latent heat (MJ/kg)

| | | |
|--------------|--------|---------------------------|
| ΔH_o | = 2.50 | Steam condensation at 0°C |
|--------------|--------|---------------------------|

Other

| | | |
|---------------|--------|----------------------------------|
| U_s | = 0.10 | Air heater (kW/m ² K) |
| ε | = 0.40 | Void (empty) fraction (-) |

Empirical constants

| | | |
|-------|-------------------------|--|
| a_1 | = 1.19 10 ¹ | Antoine equation for vapor pressure of water |
| a_2 | = 3.99 10 ³ | |
| a_3 | = 2.34 10 ² | |
| b_1 | = 7.35 10 ⁻⁴ | Oswin equation for material isotherms |
| b_2 | = 1.75 10 ³ | |
| b_3 | = 4.00 10 ⁻¹ | |
| c_0 | = 0.50 | Drying kinetics equation |
| c_1 | = 1.40 | |
| c_2 | = -1.65 | |
| c_3 | = -0.25 | |
| c_4 | = 0.12 | |
| e_1 | = 2.00 | Belt driver power equation |
| f_1 | = 2.00 | Pressure loss equation |

Table 7.22 Cost Data**Utility cost**

| | | |
|-------|---------------|-----------------------|
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_s | = 0.05 \$/kWh | Cost of heating steam |

Equipment unit cost

| | | |
|-----------|---------------------------|----------------|
| C_{bel} | = 25.0 k\$/m ² | Belt dryer |
| C_{exc} | = 2.00 k\$/m ² | Heat exchanger |
| C_{fan} | = 1.00 k\$/kW | Fan |

Equipment size scaling factor

| | | |
|-----------|--------|----------------|
| n_{bel} | = 0.95 | Belt dryer |
| n_{exc} | = 0.65 | Heat exchanger |
| n_{fan} | = 0.75 | Fan |

Other

| | | |
|-------|-------------|-----------------------|
| t_y | = 4000 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 5 yr | Lifetime |

Table 7.23 Process Specifications

| | | |
|-------|-----------------|-----------------------------------|
| F | = 0.10 ton/h db | Feed flow rate |
| X_o | = 10.0 kg/kg db | Initial material moisture content |
| X | = 0.10 kg/kg db | Final material moisture content |
| d | = 0.01 m | Material characteristic size |
| T_o | = 25.0 °C | Ambient temperature |
| Y_o | = 0.01 kg/kg db | Ambient humidity |
| Z_o | = 0.20 m | Loading depth |
| P | = 1.00 bar | Ambient pressure |
| T_s | = 160. °C | Heating steam temperature |

Table 7.24 Design Variables

| | | |
|-----|------------------|------------------------|
| Y | = 0.035 kg/kg db | Drying air humidity |
| T | = 65.0 °C | Drying air temperature |
| V | = 1.50 m/s | Drying air velocity |
| D | = 2.00 m | Belt width |

Table 7.25 Cost Analysis Results**Equipment cost**

| | | |
|-----------|-----------------|----------------|
| C_{bel} | = 900 k\$ | Belt dryer |
| C_{exc} | = 45 k\$ | Heat exchanger |
| C_{fan} | = <u>21</u> k\$ | Fan |
| C_{eq} | = 966 k\$ | Equipment cost |

Operating cost

| | | |
|----------|---------------------|-----------------------|
| C_e | = 43 k\$/yr | Cost of electricity |
| C_s | = <u>229</u> k\$/yr | Cost of heating steam |
| C_{op} | = 271 k\$/yr | Operating cost |

Objective function

| | | |
|-----------|---------------------|---------------------------|
| eC_{eq} | = 242 k\$/yr | Annualized equipment cost |
| C_{op} | = <u>271</u> k\$/yr | Operating cost |
| TAC | = 513 k\$/yr | Total annual cost |

Note: In the absence of experimental drying data, approximate process design can be based on an estimated drying time t_c from Equation (7-36). The water diffusivity in carrots at 65°C is taken from the literature as $D=3 \times 10^{-9} \text{ m}^2/\text{s}$ (Saravacos and Maroulis, 2001). For an average material size $d=1$ cm, Equation (7-36) yields $t_c=0.94$ h, which is about 15% higher than the value used in this example (0.81 h). The material size is critical to the design calculations, since the drying time constant is proportional to its square.

Table 7.26 Process Design Results**Drying air**

| | | |
|-------|------------------|-------------------------------------|
| F_a | = 40 ton/h | Fresh air flow rate |
| F_f | = 235 ton/h | Recycle air flow rate |
| T | = 65 °C | Drying air temperature |
| Y | = 0.035 kg/kg db | Drying air humidity |
| V | = 1.50 m/s | Drying air velocity |
| P | = 1 bar | Drying pressure |
| T_o | = 25 °C | Ambient temperature |
| Y_o | = 0.010 kg/kg db | Ambient humidity |
| P_s | = 0.25 bar | Vapor pressure at drying conditions |
| a_w | = 0.21 | Water activity at drying conditions |

Material

| | | |
|-------|-----------------|------------------------------|
| F | = 0.10 ton/h db | Material flow rate |
| X_o | = 10. kg/kg db | Initial moisture content |
| X | = 0.1 kg/kg db | Final moisture content |
| X_e | = 0.08 kg/kg db | Equilibrium moisture content |
| d | = 0.01 m | Particle size |
| t_c | = 0.81 h | Drying time constant |
| t | = 4.94 h | Drying time |

Dryer

| | | |
|-------|-----------------------|--------------------------|
| W | = 1 ton/h | Drying rate |
| L | = 21.7 m | Dryer length |
| D | = 2.00 m | Dryer width |
| M | = 5.43 ton | Dryer mass holdup |
| H | = 8.70 m ³ | Dryer volume holdup |
| A | = 43 m ² | Belt area |
| A_s | = 120 m ² | Air heater transfer area |
| u_b | = 4.4 m/h | Belt velocity |
| Z_o | = 0.20 m | Loading depth |

Thermal Load

| | | |
|----------|-----------|--------------------|
| Q_{we} | = 0.65 MW | Water vaporization |
| Q_{sh} | = 0.05 MW | Solid heating |
| Q_{ah} | = 0.45 MW | Air heating |
| Q | = 1.14 MW | Total thermal load |
| T_s | = 160 °C | Steam temperature |

Electrical Load

| | | |
|-------|----------|-------------------------|
| E_b | = 48 kW | Belt drive |
| E_f | = 59 kW | Fan |
| E | = 107 kW | Total power requirement |

Performance

| | | |
|-----|----------------------------|------------------------------|
| n | = 0.57 | Thermal efficiency |
| r | = 22.8 kg/h m ² | Specific rate of evaporation |

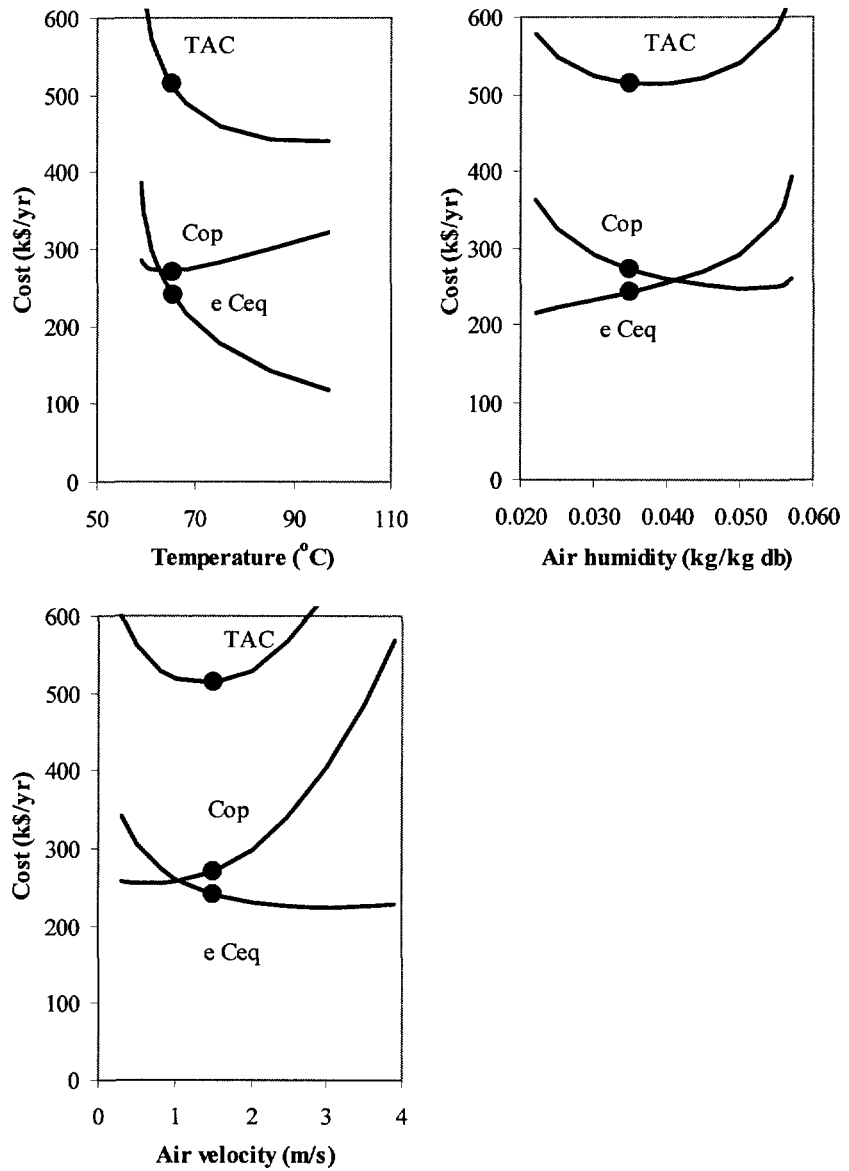


Figure 7.24 Sensitivity analysis of belt dryer: The effect of the design variables on the cost.

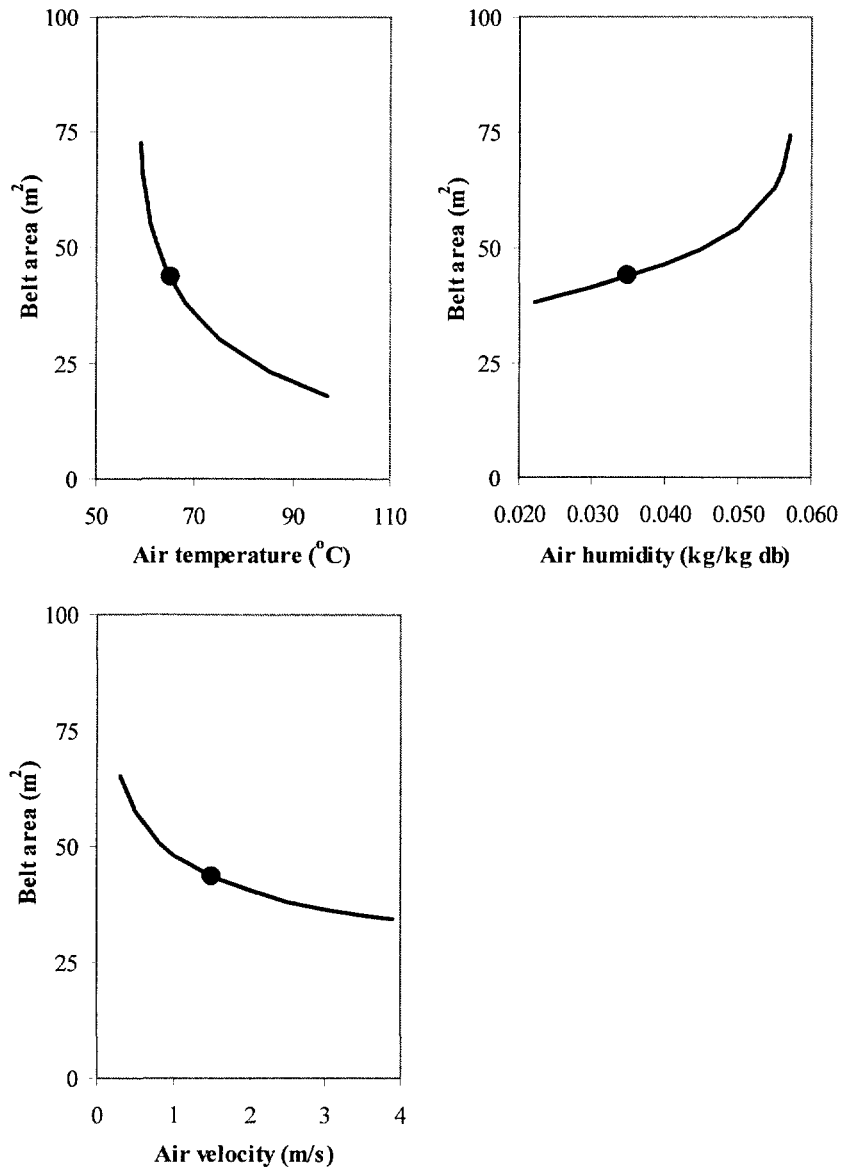


Figure 7.25 Sensitivity analysis of belt dryer: The effect of the design variables on the required belt area.

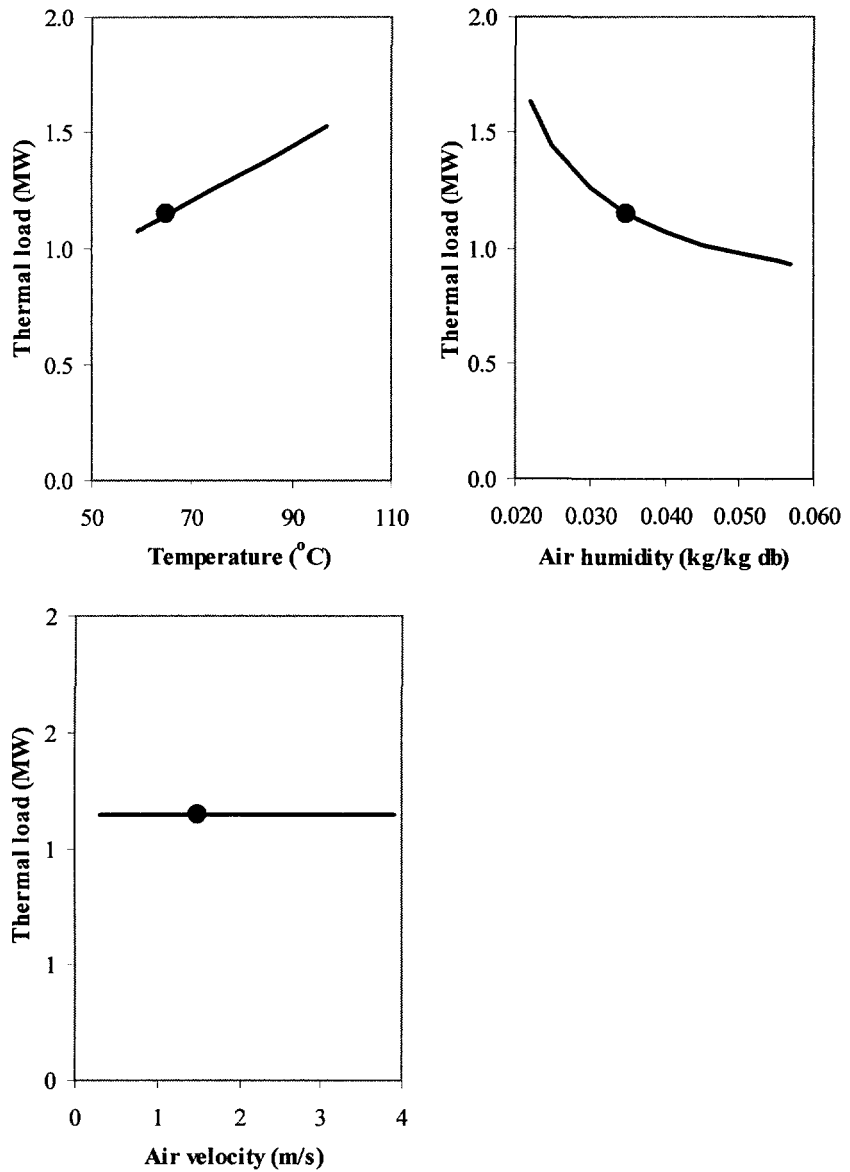


Figure 7.26 Sensitivity analysis of belt dryer: The effect of the design variables on the required thermal load.

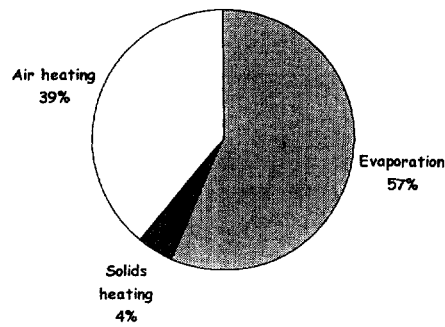


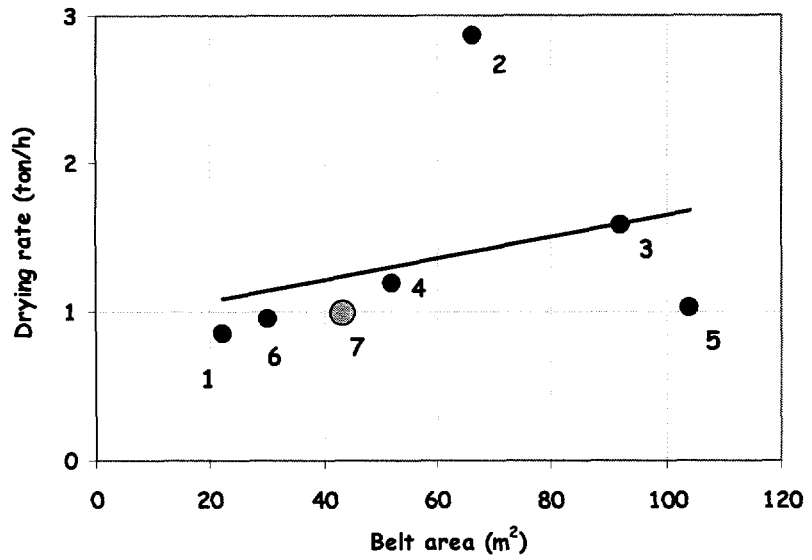
Figure 7.27 Thermal load breakdown of belt dryer.

The results of belt dryer design for carrot dehydrations are compared in figures 7.28 and 7.29 with other belt dryer dehydration processes presented in Table 7.27.

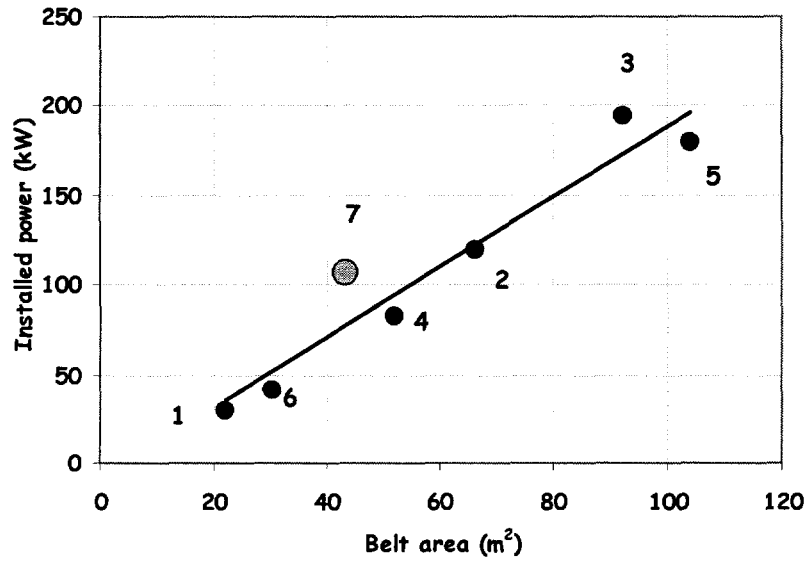
Table 7.27 Comparison Between Various Belt Dryer Designs

| Case study number | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|--------------------------|-------------------|-------------|--------------|---------------------|---------|--------------------|---------------|
| | Inorganic pigment | Corn starch | Fiber staple | Charcoal briquettes | Gelatin | Inorganic chemical | Present study |
| Material flow rate | | | | | | | |
| F ton/h db | 0.71 | 4.54 | 1.72 | 5.44 | 0.30 | 0.86 | 0.10 |
| Belt area | | | | | | | |
| A m ² | 22.1 | 66.4 | 92.2 | 52 | 104 | 30.2 | 43.5 |
| Depth of loading | | | | | | | |
| Z ₀ m | 0.03 | 0.04 | | 0.16 | 0.05 | 0.04 | 0.2 |
| Air temperature | | | | | | | |
| T °C | 120 | 130 | 110 | 130 | 42 | 100 | 65 |
| Particle size | | | | | | | |
| d cm | 0.64 | | | 5.00 | 0.20 | 0.64 | 1.00 |
| Initial moisture content | | | | | | | |
| X ₀ kg/kg db | 120 | 85 | 110 | 37 | 300 | 111 | 1000 |
| Final moisture content | | | | | | | |
| X kg/kg db | 0.5 | 13.6 | 9.0 | 5.3 | 11.1 | 1.0 | 10.0 |
| Drying time | | | | | | | |
| t h | 0.58 | 0.40 | 0.18 | 1.75 | 3.20 | 1.20 | 4.94 |
| Air velocity | | | | | | | |
| V m/s | 1.27 | 1.12 | 0.66 | 1.12 | 1.27 | 1.27 | 1.50 |
| Drying rate | | | | | | | |
| W ton/h | 0.85 | 2.86 | 1.58 | 1.20 | 1.03 | 0.95 | 0.99 |
| Thermal load | | | | | | | |
| Q MW | 1.09 | 3.73 | 1.78 | | 1.90 | 1.43 | 1.14 |
| Installed power | | | | | | | |
| E kW | 30 | 119 | 194 | 82 | 179 | 41 | 107 |

Data from Perry and Green (1997)

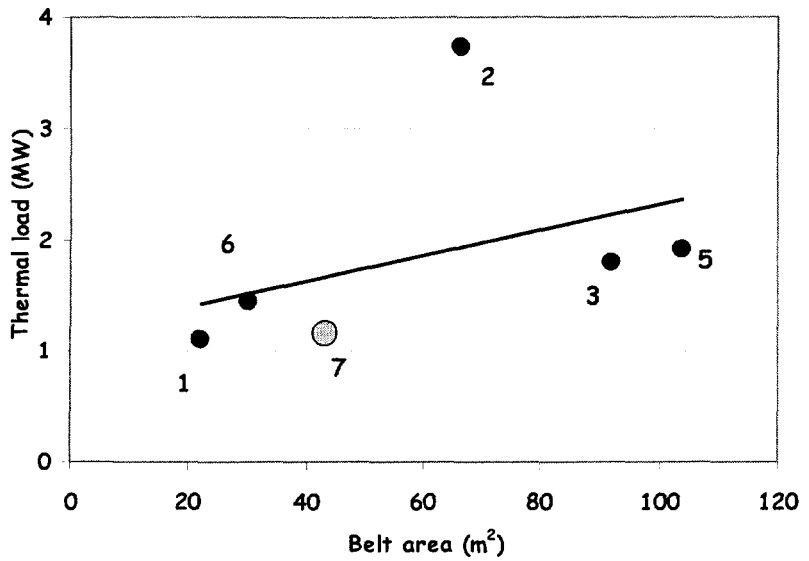


(a)

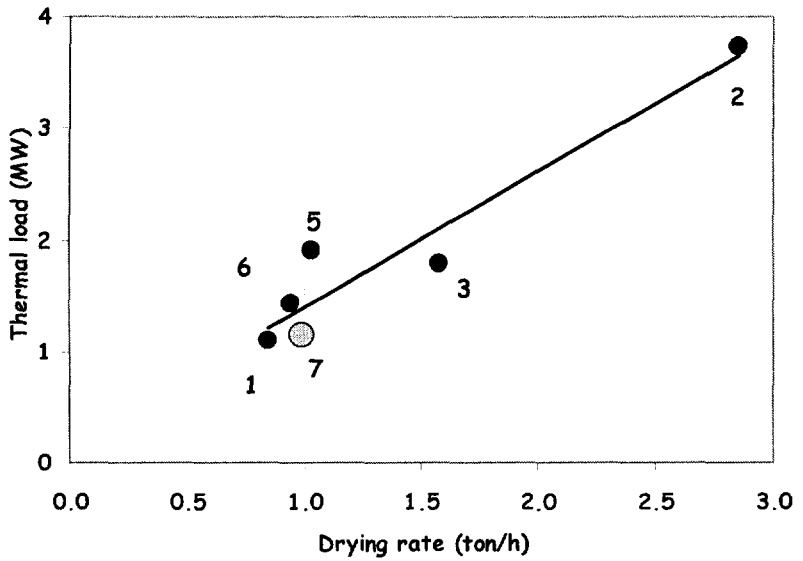


(b)

Figure 7.28 Comparison between various belt dryer designs: Drying rate (a) and installed power (b) versus belt area. Data from Table 7.27.



(a)



(b)

Figure 7.29 Comparison between various belt dryer designs: Thermal load versus belt area (a) and drying rate (b). Data from Table 7.27.

VI. DESIGN OF ROTARY DRYER

In this section, a design approach is described for the rotary dryer (Kiranoudis et al., 1994; Marinos-Kouris et al., 1996).

1. Process Description

A typical flowsheet of a rotary dryer is presented in Figure 7.30. The wet feed at flow rate F (kg/s db), temperature T_o ($^{\circ}\text{C}$) and humidity X_o (kg/kg db) enters the dryer. The dried product exits at the same flow rate in dry basis F (kg/s db), temperature T ($^{\circ}\text{C}$) and humidity X (kg/kg db). The dryer is rotated at a velocity N (RPM) and needs an electrical power E_r (kW). Drying air enters the dryer with a flow rate F_f (kg/s db), temperature T ($^{\circ}\text{C}$) and humidity Y (kg/kg db). The drying air temperature is controlled in the heater and the drying air humidity is controlled via the flow rate of the fresh air F_o (kg/s db). An electrical power E_f (kW) is expended by the fan and a thermal power Q (kW) is expended by the heater.

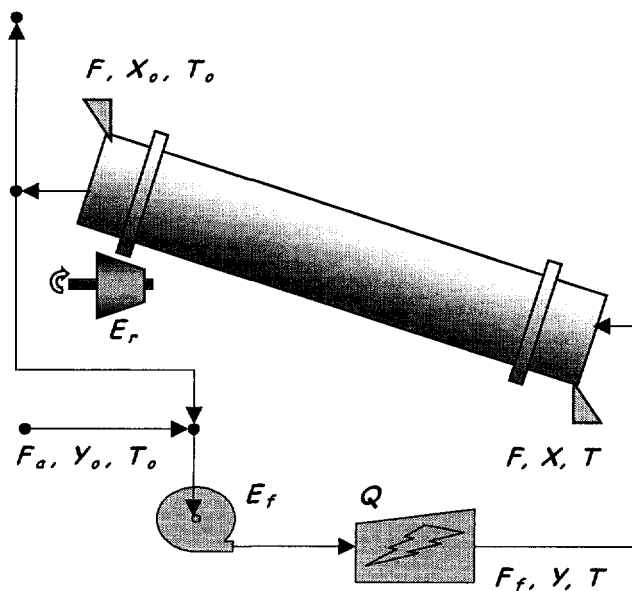


Figure 7.30 Schematic representation of a rotary dryer.

2. Process Model

A mathematical model of the process presented in Figure 7.30 is summarized in Table 7.28. Equations (E01) through (E14) are similar to these for belt dryer and they are described in the previous section. Equation (E15) calculates the volume hold up versus the geometrical characteristics of the dryer. Equation (E16) calculates the required volume of the dryer, while Equation (E17) estimates the required rotation velocity to obtain the desired residence time. Equations (E18) through (E20) are used for sizing the fan. Equation (E18) calculates the pressure loss of air through the dryer. Equation (E19) correlates the air flow with the air velocity. Equation (E20) estimates the required electrical power to operate the fan. Equation (E21) calculates the dryer weight. Equation (E22) estimates the required electrical power to rotate the dryer. Equation (E23) calculates the required total electrical power. Finally Equations (E24) and (E25) define two crucial dryer performance indices. Equation (E24) defines the dryer thermal performance, while (E25) calculates the evaporating capacity per unit dryer volume.

Forty-one variables presented in Table 7.29 are involved in the model of twenty-five equations presented in Table 7.28. The corresponding technical data are summarized in Table 7.30. The process specifications of a typical design problem is presented in Table 7.31, while a degrees-of-freedom analysis is shown in Table 7.32, which results in seven design variables. Table 7.33 suggests a selection of design variables and the corresponding solution algorithm is presented in Table 7.34. The total annualized cost TAC presented in Table 7.35 is used as objective function in process optimization. The required cost data are summarized in Table 7.36.

Concerning the design variables the following points should be considered: Four variables refer to geometrical characteristics of the dryer: Dryer diameter, number of flights, flight hold up, slope. Standard geometrical configurations for flights are usually applied.

The remainder three variables constitute the drying air conditions: Drying air humidity, temperature and velocity.

When high values of temperature are used, the drying time and consequently the dryer size decrease, but the energy losses in the rejected air stream increase. Thus, lower equipment cost and higher operating cost are obtained. It must also be noted that most of the food components are damaged at high temperatures.

When lower values of humidity are used, the drying time and consequently the dryer size decrease, but the required fresh air increases and, consequently, the energy losses in the rejected air stream increase. Thus, lower equipment cost and higher operating cost are obtained.

When higher values of air velocity are used, the drying time and consequently the dryer size decrease, but the required fan power increases. Thus, lower equipment cost and higher operating cost are obtained.

Table 7.28 Rotary Dryer Model**Psychrometric equations**

$$P_S = \exp [a_1 - a_2 / (a_3 + T)] \quad (\text{E01})$$

$$Y = m a_w P_S / (P - a_w P_S) \quad (\text{E02})$$

Drying kinetics

$$X_c = b_1 \exp[b_2 / (273 + T)] [a_w / (1 - a_w)]^{b_3} \quad (\text{E03})$$

$$t_c = c_0 d^1 u^2 T^3 Y^{c_4} \quad (\text{E04})$$

$$t = -t_c \ln[(X - X_c) / (X_o - X_c)] \quad (\text{E05})$$

Material balance

$$W = F (X_o - X) \quad (\text{E06})$$

$$W = F_a (Y - Y_o) \quad (\text{E07})$$

Thermal energy requirements

$$Q_{we} = F (X_o - X) [\Delta H_o - (C_{PL} - C_{PV}) T] \quad (\text{E08})$$

$$Q_{sh} = F [C_{PS} + X_o C_{PL}] (T - T_o) \quad (\text{E09})$$

$$Q_{ah} = F_a [C_{PA} + Y_o C_{PV}] (T - T_o) \quad (\text{E10})$$

$$Q = Q_{we} + Q_{sh} + Q_{ah} \quad (\text{E11})$$

Air heater

$$Q = A_s U_s (T_s - T) \quad (\text{E12})$$

Rotary dryer

$$M = t F (1 + X_o) \quad (\text{E13})$$

$$M = (1 - \epsilon) \rho_p H \quad (\text{E14})$$

$$H_r = 1/2 (n_f + 1) h_o L \quad (\text{E15})$$

$$V_r = (\pi D^2 / 4) L \quad (\text{E16})$$

$$t = g_1 L / N D s \quad (\text{E17})$$

Fan

$$\Delta P = f_1 L V^2 \quad (\text{E18})$$

$$F_f = \rho_a V \pi D^2 / 4 \quad (\text{E19})$$

$$E_f = \Delta P F_f / \rho_a \quad (\text{E20})$$

Rotation

$$M_d = (2\pi D^2 / 4 + \pi D L) dx \rho_m \quad (\text{E21})$$

$$E_r = e_1 N D (M + M_d) \quad (\text{E22})$$

Electrical energy requirements

$$E = E_r + E_f \quad (\text{E23})$$

Performance indices

$$n = Q_{we} / Q \quad (\text{E24})$$

$$r = W / V_r \quad (\text{E25})$$

Table 7.29 Process Variables**Drying air**

| | | |
|-------|----------|-------------------------------------|
| F_a | ton/h | Fresh air flow rate |
| F_f | ton/h | Recycle air flow rate |
| T | °C | Drying air temperature |
| Y | kg/kg db | Drying air humidity |
| V | m/s | Drying air velocity |
| P | bar | Drying pressure |
| T_o | °C | Ambient temperature |
| Y_o | kg/kg db | Ambient humidity |
| P_s | bar | Vapor pressure at drying conditions |
| a_w | - | Water activity at drying conditions |

Material

| | | |
|-------|----------|------------------------------|
| F | ton/h db | Material flow rate |
| X_o | kg/kg db | Initial moisture content |
| X | kg/kg db | Final moisture content |
| X_e | kg/kg db | Equilibrium moisture content |
| d | m | Particle size |
| t | h | Drying time |
| t_c | h | Drying time constant |

Dryer

| | | |
|------------|-------------------|---|
| W | ton/h | Evaporating capacity |
| L | m | Dryer length |
| D | m | Dryer diameter |
| M | ton | Dryer mass holdup |
| H | m ³ | Dryer volume holdup |
| V_r | m ³ | Dryer volume |
| A_s | m ² | Air heater transfer area |
| N | RPM | Rotating velocity |
| s | - | Dryer tilt (slope) |
| Δx | m | Dryer wall thickness |
| ΔP | bar | Pressure loss of air flowing through belt |
| M_d | ton | Dryer weight |
| n_f | - | Number of flights |
| h_o | m ³ /m | Solids holdup in a flight |

Thermal Load

| | | |
|----------|----|--------------------|
| Q_{we} | kW | Water vaporization |
| Q_{sh} | kW | Solid heating |
| Q_{ah} | kW | Air heating |
| Q | kW | Total thermal load |
| T_s | °C | Steam temperature |

Electrical Load

| | | |
|-------|----|-------------------------|
| E_r | kW | Rotating driver |
| E_f | kW | Fan |
| E | kW | Total power requirement |

Performance

| | | |
|-----|---------------------|------------------------------|
| n | - | Thermal efficiency |
| r | kg/h m ² | Specific rate of evaporation |

Table 7.30 Process Data

| | |
|---|--|
| Density (kg/m³) | |
| ρ_w | Water |
| ρ_a | Air |
| ρ_s | Dry material |
| ρ_m | Construction material |
| Specific heat (kJ/kg K) | |
| C_{PL} | Water |
| C_{PV} | Water vapor |
| C_{PA} | Air |
| C_{PS} | Dry material |
| Latent heat (kJ/kg) | |
| ΔH_o | Steam condensation at 0°C |
| Heat transfer coefficients (kW/m²K) | |
| U_s | Air heater |
| Empirical constants | |
| a_1, a_2, a_3 | Antoine equation for vapor pressure of water |
| b_1, b_2, b_3 | Oswin equation for material isotherms |
| c_0, c_1, c_2, c_3, c_4 | Drying kinetics equation |
| e_1 | Belt driver power equation |
| f_1 | Pressure loss equation |
| g_1 | Residence time equation |

Table 7.31 Process Specifications

| | | |
|------------|----------|-----------------------------------|
| F | ton/h db | Feed flow rate |
| X_o | kg/kg db | Initial material moisture content |
| X | kg/kg db | Final material moisture content |
| d | m | Material characteristic size |
| T_o | °C | Ambient temperature |
| Y_o | kg/kg db | Ambient humidity |
| Δx | m | Dryer wall thickness |
| P | bar | Ambient pressure |
| T_s | °C | Heating steam temperature |

Table 7.32 Degrees-of-Freedom Analysis

| | | | |
|--------------------|-----------|--------------------|----------|
| Process variables | 41 | Degrees of freedom | 16 |
| Process equations | <u>25</u> | Specifications | <u>9</u> |
| Degrees of freedom | 16 | Design variables | 7 |

Table 7.33 Design Variables

| | | |
|-------|-------------------|---------------------------|
| Y | kg/kg db | Drying air humidity |
| T | °C | Drying air temperature |
| V | m/s | Drying air velocity |
| D | m | Dryer diameter |
| n_f | - | Number of flights |
| h_o | m ³ /m | Solids holdup in a flight |
| s | - | Dryer slope |

Table 7.34 Model Solution Algorithm

| | | |
|-------|---|------------|
| (E01) | → | P_s |
| (E02) | → | a_w |
| (E03) | → | X_e |
| (E04) | → | t_c |
| (E05) | → | t |
| (E06) | → | W |
| (E07) | → | F_a |
| (E08) | → | Q_{we} |
| (E09) | → | Q_{sh} |
| (E10) | → | Q_{ah} |
| (E11) | → | Q |
| (E12) | → | A_s |
| (E13) | → | M |
| (E14) | → | H |
| (E15) | → | L |
| (E16) | → | V_r |
| (E17) | → | N |
| (E18) | → | ΔP |
| (E19) | → | F_f |
| (E20) | → | E_f |
| (E21) | → | M_d |
| (E22) | → | E_r |
| (E23) | → | E |
| (E24) | → | n |
| (E25) | → | r |

Table 7.35 Cost Analysis**Equipment cost**

$$C_{eq} = C_{rot} A^{n_{rot}} + C_{exc} A_s^{n_{exc}} + C_{fan} E_f^{n_{fan}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_s Q + C_e E) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (F04)$$

Table 7.36 Cost Data

| Utility cost | | |
|-------------------------------|-------------------|-----------------------|
| C_e | \$/kWh | Cost of electricity |
| C_s | \$/kWh | Cost of heating steam |
| Equipment unit cost | | |
| C_{rot} | \$/m ³ | Rotary dryer |
| C_{exc} | \$/m ² | Heat exchanger |
| C_{fan} | \$/kW | Fan |
| Equipment size scaling factor | | |
| N_{rot} | - | Rotary dryer |
| n_{exc} | - | Heat exchanger |
| n_{fan} | - | Fan |
| Other | | |
| t_y | h/yr | Annual operating time |
| i_r | - | Interest rate |
| l_f | yr | Lifetime |

3. Application to Carrot Dehydration

The dehydration process solved in the previous section for the belt dryer is solved again in this section for the rotary dryer. The process specifications, and the required technical and economic data, are summarized in Tables 7.37 through 7.39. The optimal values of the design variables are presented in Table 7.40, while the corresponding technical and economic results are summarized in Tables 7.41 and 7.42.

The results are further evaluated through a sensitivity analysis of the design variables on the cost (Figure 7.31), the dryer size (Figure 7.32) and the required thermal load (Figure 7.33).

4. Comparison of Belt and Rotary Dryers

The cost of rotary dryers is significantly lower than that of belt dryers, mainly due to the lower equipment cost (about one half of the cost of the belt unit).

Conveyor belt dryers are used in the dehydration of sensitive food pieces, that must retain their size and shape after drying. Hard food pieces, such as carrots, which can stand the abrasive action of tumbling, can be dehydrated more economically in rotary dryers. The rotary units are used effectively in the dehydration of food particles and powders that move easily through the dryer. They are economical for drying solid food wastes, such as by-products of fruit and vegetable processing, and direct low-cost heating by fuel combustion can be applied.

Table 7.37 Process Data

| | | |
|---|-------------------------|--|
| Density (kg/m³) | | |
| ρ_w | = 1000 | Water |
| ρ_a | = 1 | Air |
| ρ_s | = 1750 | Dry material |
| ρ_m | = 8000 | Construction material |
| Specific heat (kJ/kg K) | | |
| C_{PL} | = 4.20 | Water |
| C_{PV} | = 1.90 | Water vapor |
| C_{PA} | = 1.00 | Air |
| C_{PS} | = 2.00 | Dry material |
| Latent heat (MJ/kg) | | |
| ΔH_o | = 2.50 | Steam condensation at 0°C |
| Heat transfer coefficients (kW/m²K) | | |
| U_s | = 0.10 | Air heater |
| Empirical constants | | |
| a_1 | = 1.19 10 ¹ | Antoine equation for vapor pressure of water |
| a_2 | = 3.99 10 ³ | |
| a_3 | = 2.34 10 ² | |
| b_1 | = 7.35 10 ⁻⁴ | Oswin equation for material isotherms |
| b_2 | = 1.75 10 ³ | |
| b_3 | = 4.00 10 ⁻¹ | |
| c_0 | = 0.50 | Drying kinetics equation |
| c_1 | = 1.40 | |
| c_2 | = -1.65 | |
| c_3 | = -0.25 | |
| c_4 | = 0.12 | |
| e_1 | = 1.00 | Belt driver power equation |
| f_1 | = 2.50 | Pressure loss equation |
| g_1 | = 0.25 | Residence time equation |

Table 7.38 Cost Data

| | | |
|--------------------------------------|---------------------------|-----------------------|
| Utility cost | | |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_s | = 0.05 \$/kWh | Cost of heating steam |
| Equipment unit cost | | |
| C_{rot} | = 15.0 k\$/m ³ | Rotary dryer |
| C_{exc} | = 2.00 k\$/m ² | Heat exchanger |
| C_{fan} | = 1.00 k\$/kW | Fan |
| Equipment size scaling factor | | |
| N_{rot} | = 0.85 | Rotary dryer |
| n_{exc} | = 0.65 | Heat exchanger |
| n_{fan} | = 0.75 | Fan |
| Other | | |
| t_y | = 4000 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 7.39 Process Specifications

| | | |
|------------|-----------------|--------------------------------------|
| F | = 0.10 ton/h db | Feed flow rate |
| X_o | = 10.0 kg/kg db | Initial material moisture content |
| X | = 0.10 kg/kg db | Final material moisture content |
| d | = 0.01 m | Material characteristic size (tubes) |
| T_o | = 25.0 °C | Ambient temperature |
| Y_o | = 0.01 kg/kg db | Ambient humidity |
| Δx | = 0.01 m | Dryer wall thickness |
| P | = 1.00 bar | Ambient pressure |
| T_s | = 160 °C | Heating steam temperature |

Table 7.40 Design Variables

| | | |
|-------|---------------------------|---------------------------|
| Y | = 0.045 kg/kg db | Drying air humidity |
| T | = 65.0 °C | Drying air temperature |
| V | = 1.50 m/s | Drying air velocity |
| D | = 2.00 m | Dryer diameter |
| n_f | = 20 | Number of flights |
| h_o | = 0.045 m ³ /m | Solids holdup in a flight |
| s | = 4 % | Dryer slope |

Table 7.41 Cost Analysis Results

| | | |
|---------------------------|---------------------|---------------------------|
| Equipment cost | | |
| C_{rot} | = 519 k\$ | Rotary dryer |
| C_{exc} | = 42 k\$ | Heat exchanger |
| C_{fan} | = <u>12</u> k\$ | Fan |
| C_{eq} | = 572 k\$ | Equipment cost |
| Operating cost | | |
| C_e | = 38 k\$/yr | Cost of electricity |
| C_s | = <u>203</u> k\$/yr | Cost of heating steam |
| C_{op} | = 241 k\$/yr | Operating cost |
| Objective function | | |
| eC_{eq} | = 143 k\$/yr | Annualized equipment cost |
| C_{op} | = <u>241</u> k\$/yr | Operating cost |
| TAC | = 384 k\$/yr | Total annual cost |

Table 7.42 Process Design Results

| | |
|---------------------------------|-------------------------------------|
| Drying air | |
| F_a = 28.3 ton/h | Fresh air flow rate |
| F_f = 28.3 ton/h | Drying air flow rate |
| T = 65 °C | Drying air temperature |
| Y = 0.045 kg/kg db | Drying air humidity |
| V = 1.60 m/s | Drying air velocity |
| P = 1 bar | Drying pressure |
| T_o = 25 °C | Ambient temperature |
| Y_o = 0.010 kg/kg db | Ambient humidity |
| P_s = 0.25 bar | Vapor pressure at drying conditions |
| a_w = 0.27 | Water activity at drying conditions |
| Material | |
| F = 0.10 ton/h db | Material flow rate |
| X_o = 10. kg/kg db | Initial moisture content |
| X = 0.1 kg/kg db | Final moisture content |
| X_e = 0.08 kg/kg db | Equilibrium moisture content |
| d = 0.01 m | Particle size |
| t = 5.50 h | Drying time |
| t_c = 0.82 h | Drying time constant |
| Dryer | |
| W = 1 ton/h | Evaporating capacity |
| L = 13.2 m | Dryer length |
| D = 2.50 m | Dryer diameter |
| M = 6.05 ton | Dryer mass holdup |
| H = 9.70 m ³ | Dryer volume holdup |
| V_r = 64.7 m ³ | Dryer volume |
| A_s = 107 m ² | Air heater transfer area |
| N = 3.59 RPM | Rotating velocity |
| s = 4% | Dryer tilt |
| Δx = 0.01 m | Dryer wall thickness |
| M_d = 9.06 ton | Dryer weight |
| n_f = 25 | Number of flights |
| h_o = 0.057 m ³ /m | Solids holdup in a flight |
| Thermal Load | |
| Q_{we} = 0.65 MW | Water vaporization |
| Q_{sh} = 0.05 MW | Solid heating |
| Q_{ah} = 0.32 MW | Air heating |
| Q = 1.02 MW | Total thermal load |
| Electrical Load | |
| E_r = 68 kW | Belt driver |
| E_f = 26 kW | Fan |
| E = 94 kW | Total power requirement |
| Performance | |
| n = 0.64 | Thermal efficiency |
| r = 15.3 kg/h m ³ | Specific rate of evaporation |

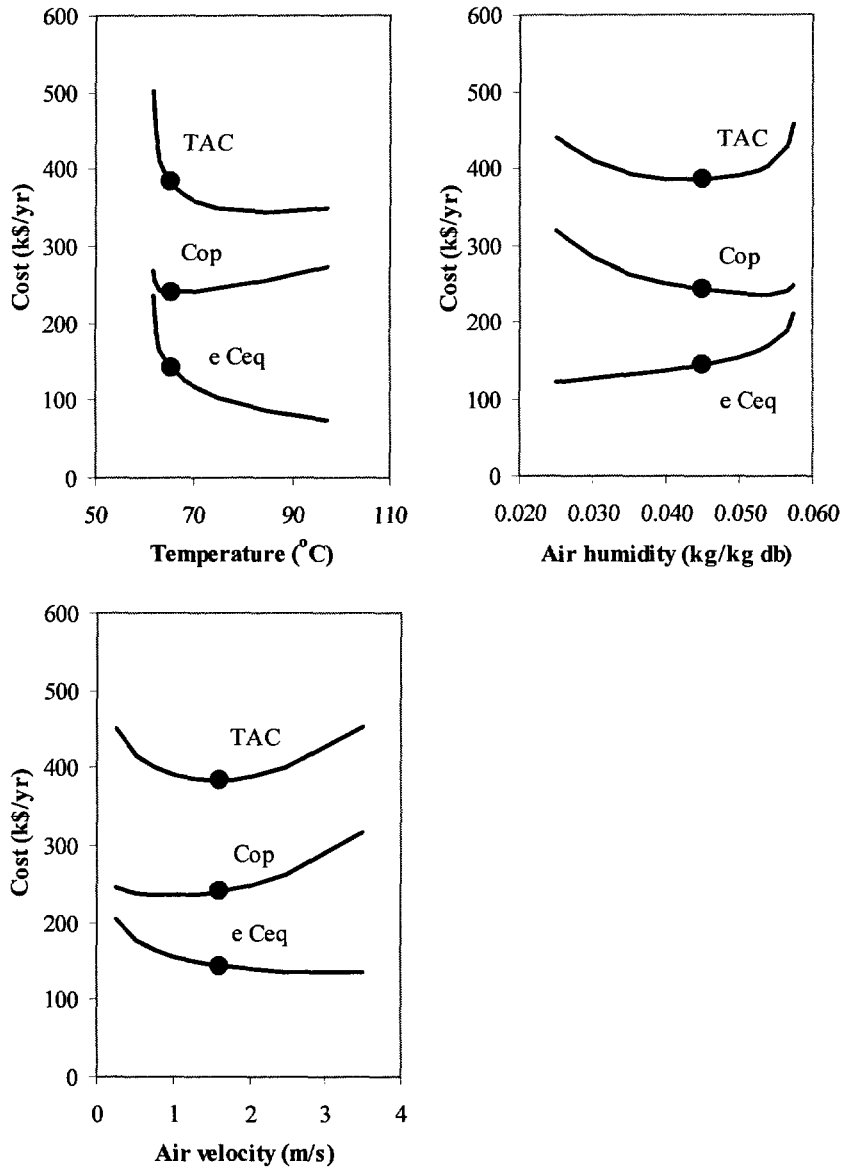


Figure 7.31 Sensitivity analysis of rotary dryer: The effect of the design variables on the cost.

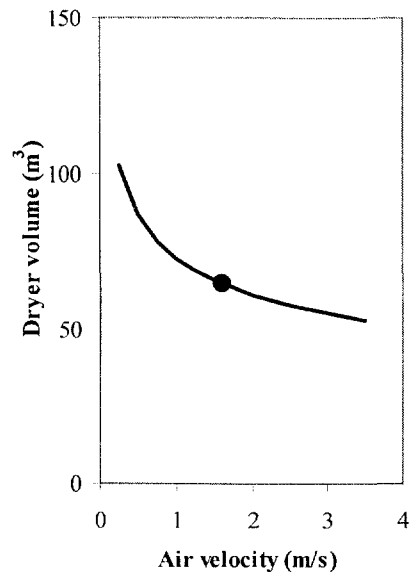
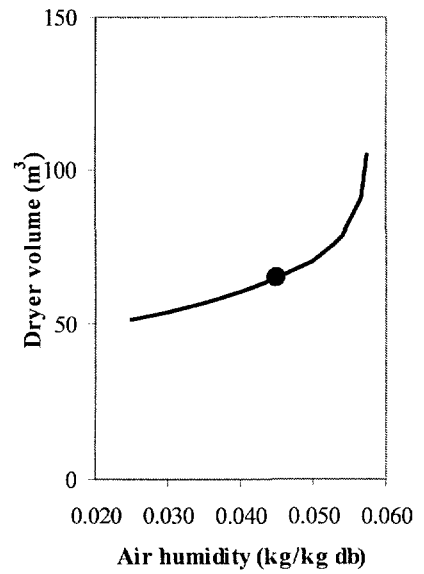
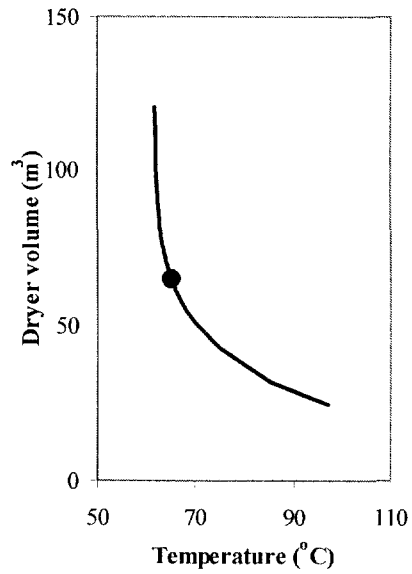


Figure 7.32 Sensitivity analysis of rotary dryer: The effect of the design variables on the required dryer volume.

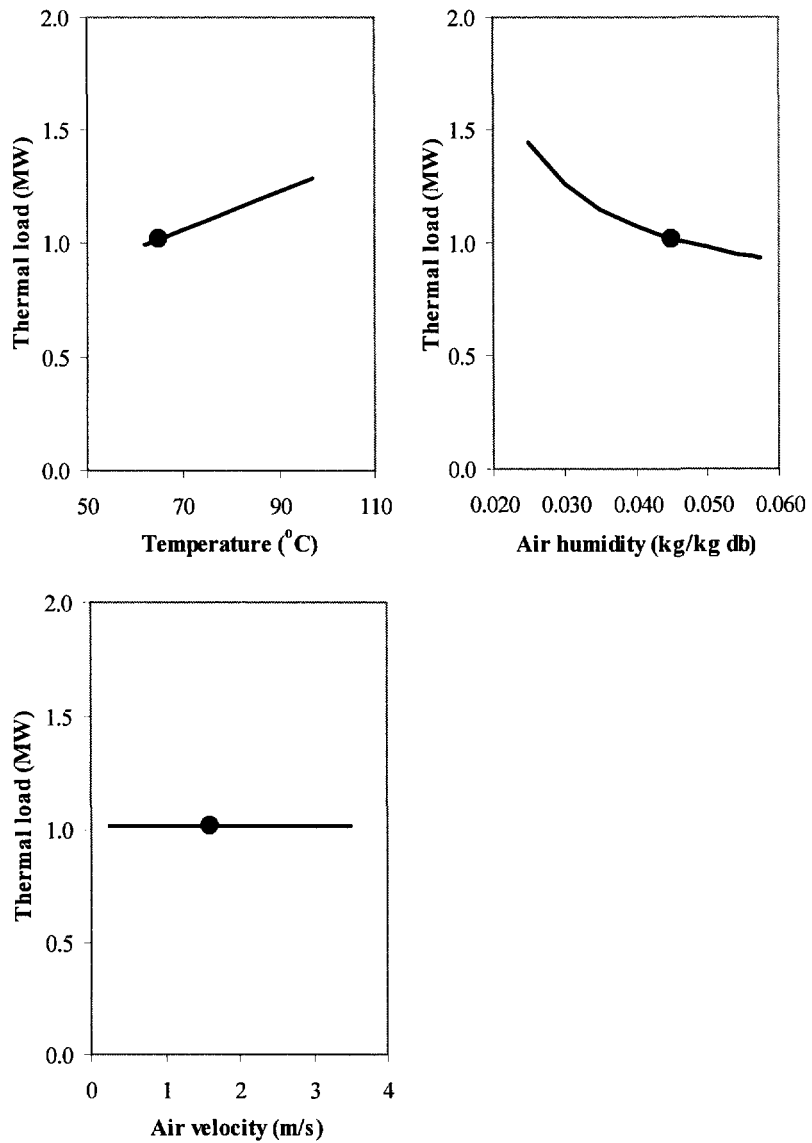


Figure 7.33 Sensitivity analysis of rotary dryer: The effect of the design variables on the required thermal load.

NOMENCLATURE

| | | |
|-----------|-------------|-------------------------------------|
| a_i | - | Antoine equation constants |
| A_b | m^2 | Belt area |
| A_s | m^2 | Air heater transfer area |
| a_w | - | Water activity |
| b_i | - | Oswin equation constants |
| c_i | - | Drying kinetics equation constants |
| C_{bel} | $\$/m^2$ | Belt dryer unit cost |
| C_e | $\$/kWh$ | Cost of electricity |
| C_{exc} | $\$/m^2$ | Heat exchanger unit cost |
| C_{fan} | $\$/kW$ | Fan unit cost |
| C_p | kJ/kgK db | Specific heat of humid air |
| C_{pA} | $kJ/kg K$ | Specific heat of air |
| C_{pL} | $kJ/kg K$ | Specific heat of water |
| C_{pS} | $kJ/kg K$ | Specific heat of dry material |
| C_{pV} | $kJ/kg K$ | Specific heat of water vapor |
| C_{pW} | $kJ/kg K$ | Specific heat of liquid water |
| C_{rot} | $\$/m^3$ | Rotary dryer unit cost |
| C_s | $\$/kWh$ | Cost of heating steam |
| D | m | Dryer width |
| d | m | Particle size |
| db | - | Dry basis |
| E | kW | Total power requirement |
| e | - | Capital recovery factor |
| e_1 | - | Belt driver power equation constant |
| E_b | kW | Belt driver power |
| E_f | kW | Fan power |
| E_r | kW | Rotating driver power |
| F | ton/h db | Material flow rate |
| f_l | - | Pressure loss equation power |
| F_a | ton/h | Fresh air flow rate |
| F_f | ton/h | Recycle air flow rate |
| g_l | - | Residence time equation power |
| H | kJ/kg db | Enthalpy of humid air |
| H | m^3 | Dryer volume holdup |
| h_o | m^3/m | Solids holdup in a flight |
| i_r | - | Interest rate |
| L | m | Dryer length |
| l_f | yr | Lifetime |
| m | - | Air/water molecular weight ratio |
| M | ton | Dryer mass holdup |
| M_d | ton | Dryer weight |
| n | - | Thermal efficiency |
| N | RPM | Rotating velocity |
| n_{bel} | - | Belt dryer scaling factor |
| n_{exc} | - | Heat exchanger |
| n_f | - | Number of flights |

| | | |
|---------------|---------------------|---|
| n_{fan} | - | Fan scaling factor |
| n_{rot} | - | Rotary dryer scaling factor |
| P | bar | Pressure |
| P_d | bar | Dew pressure |
| P_s | bar | Vapor pressure at drying conditions |
| P_s | bar | Vapor pressure at temperature T |
| P_w | bar | Vapor pressure at temperature T_w |
| Q | kW | Total thermal load |
| Q_{ah} | kW | Air heating thermal load |
| Q_{sh} | kW | Solid heating thermal load |
| Q_{we} | kW | Water vaporization thermal load |
| R | kJ/kmol K | Ideal gas constant |
| r | kg/h m ² | Specific rate of evaporation |
| s | - | Dryer tilt |
| t | h | Drying time |
| T | °C | Drying air temperature |
| T | °C | Temperature |
| T_b | °C | Boiling temperature |
| t_c | h | Drying time constant |
| T_d | °C | Dew temperature |
| T_o | °C | Ambient temperature |
| T_s | °C | Steam temperature |
| T_w | °C | Wet bulb temperature |
| t_y | h/yr | Annual operating time |
| u_b | m/s | Belt velocity |
| U_s | kW/m ² K | Heat transfer coefficient at air heater |
| V | m/s | Drying air velocity |
| V_r | m ³ | Dryer volume |
| W | ton/h | Drying rate |
| W | ton/h | Evaporating capacity |
| wb | - | Wet basis |
| X | kg/kg db | Final moisture content |
| X_e | kg/kg db | Equilibrium moisture content |
| X_o | kg/kg db | Initial moisture content |
| Y | kg/kg db | Drying air humidity |
| Y | kg/kg db | Total humidity (liquid + vapor) |
| Y_L | kg/kg db | Humidity in liquid |
| Y_o | kg/kg db | Ambient humidity |
| Y_s | kg/kg db | Saturation humidity at temperature T |
| Y_V | kg/kg db | Humidity in vapor |
| Y_w | kg/kg db | Saturation humidity at temperature T_w |
| Z_o | m | Loading depth |
| ΔH_0 | kJ/kg | Latent heat of water evaporation at 0°C |
| ΔH_S | kJ/kg | Latent heat of water evaporation at temperature T |
| ΔP | bar | Pressure loss of air |
| Δx | m | Dryer wall thickness |
| ε | - | Void (empty) fraction of loading |
| ρ_a | kg/m ³ | Air density |

| | | |
|----------|-------------------|-------------------------------|
| ρ_m | kg/m ³ | Construction material density |
| ρ_s | kg/m ³ | Dry material density |
| ρ_w | kg/m ³ | Water density |

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8

Thermal Processing of Foods

I. INTRODUCTION

Thermal Processing is primarily concerned with the application of heat to destroy (inactivate) microorganisms and enzymes, which can cause spoilage of foods and health hazards to the consumers (Saravacos and Kostaropoulos, 2002). Thermal processing involves heating of foods at various time-temperature combinations, which define the three main thermal processes, i.e., blanching, pasteurization and sterilization. The objective of thermal processing is the long-time and safe preservation of sensitive foods, preferably at ambient (room) temperatures. Traditionally, thermal processing has been applied to the canning of foods, packaged in metallic containers, and preserved for long periods (longer than 6 months).

Thermal processing includes, in addition to canning, the following food processing operations:

- **Blanching** (heat inactivation of spoilage enzyme in vegetables prior to further processing),
- **Pasteurization** (inactivation of pathogenic and spoilage microorganisms and enzymes), and
- **Aseptic processing** (long-term preservation of foods with minimum heat damage).

High-temperature processing may have, in addition to preservation, some other desirable effects on foods, like improvement of eating quality (cooking), softening of some hard foods, and destruction of some undesirable components, like the trypsin inhibitor in legumes.

Thermal treatments (heating), aimed mainly at improvement of eating and other qualities of foods, like baking, cooking, and frying, are discussed in Chapter 4 of this book. Low temperature preservation (refrigeration and freezing) is described in Chapter 5.

The major problem of thermal processing is the significant damage to the nutritional (vitamins, proteins) and organoleptic (sensory) quality (taste, color, and

texture) of foods, particularly those exposed to high temperature for a relatively long time. This problem is resolved by using high temperature-short time processing methods, aseptic packaging, and special food containers.

Thermal processing is accomplished either by

- in-container sterilization (e.g., traditional canning), or
- continuous flow processing (e.g., pasteurization or aseptic processing).

Thermal processing is also used in the bulk storage of fluid foods, usually combined with refrigeration. Gentle thermal processing is used recently in the *minimal processing of foods*.

The design of thermal processing equipment is based on

- the heat inactivation of unwanted enzymes and microorganisms, and
- the heat transfer from the heating medium to the food product.

II. KINETICS OF THERMAL INACTIVATION

The kinetics of thermal inactivation deals with the destruction of enzymes and microorganisms involved in the spoilage of foods, and in the protection of Public Health. The same principles are applied in the effect of thermal processing on the nutritional and sensory quality of processed foods.

1. Inactivation of Microorganisms and Enzymes

The heat resistance of the microorganisms runs from high (spore-forming bacteria) to relatively low (vegetative cells). Some enzymes can be as heat resistant as the spore-forming bacteria, e.g., catalase and lipoxidase (Teixeira, 1992; Heldman and Hartel, 1997; Ramaswamy and Singh, 1997).

The kinetics of thermal inactivation usually follow a first-order chemical reaction, although the mechanism may be more complex, resulting in non-log-linear curves (IFT, 2003). At a given temperature, the rate of inactivation of a population

(N) of microorganisms is given by the equation:

$$\frac{dN}{dt} = -kN \quad (8-1)$$

where k is the reaction constant, and t is the time.

Integration of equation (8-1) yields:

$$\ln \frac{N}{N_o} = -kt \quad (8-2)$$

where N_o and N are the initial and final numbers of microorganisms after time of heating t at the given temperature. The material is assumed to heat instantaneously to the processing temperature and to cool rapidly, after remaining constant for the specified time t .

Equation (8-2) is usually applied in the following more practical form:

$$\log \frac{N}{N_o} = - \frac{t}{D} \quad (8-3)$$

where

$$D = 2.30 / k \quad (8-4)$$

is the decimal reduction time, usually in minutes.

The decimal reduction time is the time required for 90% reduction (one log cycle) of a population. It is determined as the inverse of the negative slope of a semi-log survivor plot of $\log(N/N_o)$ versus time t .

The inactivation (destruction) of microorganisms is determined by measuring the survivors, i.e., the microorganisms which can grow in a standard microbiological medium. The decimal reduction time characterizes the heat resistance of a microorganism at a given temperature.

2. Effect of Temperature

Equation (8-3) can be used for the heat inactivation of spoilage enzymes, and the determination of the corresponding decimal reduction time at a given temperature.

The effect of temperature on the decimal reduction time is usually described according to the equation:

$$\log \frac{D}{D_r} = - \frac{T - T_r}{z} \quad (8-5)$$

where D and D_r are the decimal reduction times at temperature T and T_r , respectively, T_r is a reference temperature usually 121°C (250°F), and z is the thermal resistance factor in °C.

The thermal resistance factor is the temperature rise required to reduce the decimal reduction time by 90% (one log cycle). For most microorganisms of interest to thermal processing, the thermal resistance factor is about 10°C.

The effect of temperature on the decimal reduction time (D) can be also expressed by the Arrhenius equation and the Q_{10} ratio.

The Arrhenius equation for the decimal reduction time is:

$$\ln \frac{D}{D_r} = -\frac{E}{R} \left(\frac{1}{T} - \frac{1}{T_r} \right) \quad (8-6)$$

where E is the energy of activation (kJ/mol), and R is the gas constant ($R=8.31$ J/mol K). T and T_r are the temperatures (in K) corresponding to D and D_r .

By combining Equations (8-5) and (8-6), the following equation is obtained, which relates the energy of activation E to the thermal resistance factor z :

$$E = 2.30 R T^2 / z \quad (8-7)$$

Activation energies in the range of 210 to 476 kJ/mol are reported in the literature for the thermal inactivation of various microorganisms (Lewis and Heppell, 2000).

The Q_{10} ratio represents the increase of reaction rate by an increase of the temperature by 10 °C which in the case of decimal reduction time is defined as:

$$Q_{10} = D_T / D_{T-10} \quad (8-8)$$

By combining Equations (8-5) and (8-8), the following relation is obtained:

$$\log Q_{10} = 10 / z \quad (8-9)$$

Thus, for the usual value of $z = 10$, the ratio $Q_{10} = 10$, i.e., the decimal reduction time D decreases by 10 times, when the temperature is increased by 10°C. It should be noted that for most chemical reactions the ratio Q_{10} is about 2, i.e., the inactivation of microorganisms is about 5 times faster than a chemical reaction.

3. Commercial Sterility

The thermal death time F required to obtain the specified inactivation for commercial sterility is a multiple of the decimal reduction time:

$$F = m D \quad (8-10)$$

where m is the reduction exponent defined as (Brennan et al., 1990):

$$\log \left(\frac{N}{N_o} \right)_{cs} = -m \quad (8-11)$$

where $(N/N_o)_{cs}$ is the required reduction of the microorganisms population for commercial sterility.

Different m -values are considered for various microorganisms.

4. Inactivation at Varying Temperature

The preceding considerations have been based on the presumption that the product is heated instantaneously to the treatment temperature, held at this temperature for the whole of the treatment time and then cooled instantaneously. But in practice the rates of heating and cooling are slower and the destruction of microorganisms starts before the maximum temperature has been reached and it continues while the product is cooled. This fact is taken into account by integrating the effect of temperature:

$$\log\left(\frac{N}{N_0}\right) = -\int_0^t \frac{dt}{D} \quad \text{where} \quad D = D_r 10^{\frac{T_r - T}{z}} \quad (8-12)$$

The integration can be obtained numerically when the time varying temperature is known.

5. Thermal Damage to Food Components

Although some thermal processes may cause desirable changes to foods, like cooking and improvement of eating quality of foods, most heat-induced chemical and biochemical changes are undesirable, e.g., non-enzymatic browning, and vitamin, taste, texture, and color deterioration.

Most thermal damage reactions are described by first-order kinetics, similar to the inactivation of microorganisms and enzymes (Heldman and Hartel, 1997). The rate of thermal damage to food components is much slower than the thermal inactivation of the heat-resistant microorganisms and enzymes, i.e., significantly higher D and z values (and lower Arrhenius activation energies).

6. Thermal Destruction Data

Table 8.1 shows some typical values of the decimal reduction time D at a reference temperature T_R , the thermal resistance factor z , and the reduction exponent m . Figure 8.1 uses these data to present the effect of temperature on decimal reduction time. Characteristic D and z values for damage to food components are presented in Table 8.2 and Figure 8.2. Figure 8.3 compares the D -values between the above categories.

Table 8.1 Thermal Destruction Data of Spoilage Microorganisms

| | Organism | T_R , °C | D , min | z , °C | m , -- | Product |
|---|------------------------|------------|-----------|----------|----------|-------------------------------------|
| 1 | <i>C. botulinum</i> | 121 | 0.30 | 11 | 12 | Low Acid Foods (pH>4.5) |
| 2 | <i>C. sporogenes</i> | 121 | 1.50 | 11 | 5 | Meats |
| 3 | <i>B. stearotherm.</i> | 121 | 5.00 | 10 | 5 | Vegetables and Milk |
| 4 | <i>C. thermosacch.</i> | 121 | 4.00 | 11 | 5 | Vegetables |
| 5 | <i>B. subtilis</i> | 121 | 0.40 | 7 | 6 | Milk Products |
| 6 | <i>B. coagulans</i> | 121 | 0.07 | 10 | 5 | Foods of pH=4.2-4.5, e.g., tomatoes |
| 7 | <i>C. pasteurianum</i> | 100 | 0.50 | 8 | 5 | Foods of pH=4.2-4.5, e.g., pears |

T_R : Reference temperature; D : Decimal reduction time; z : Thermal resistance factor; m : Reduction exponent

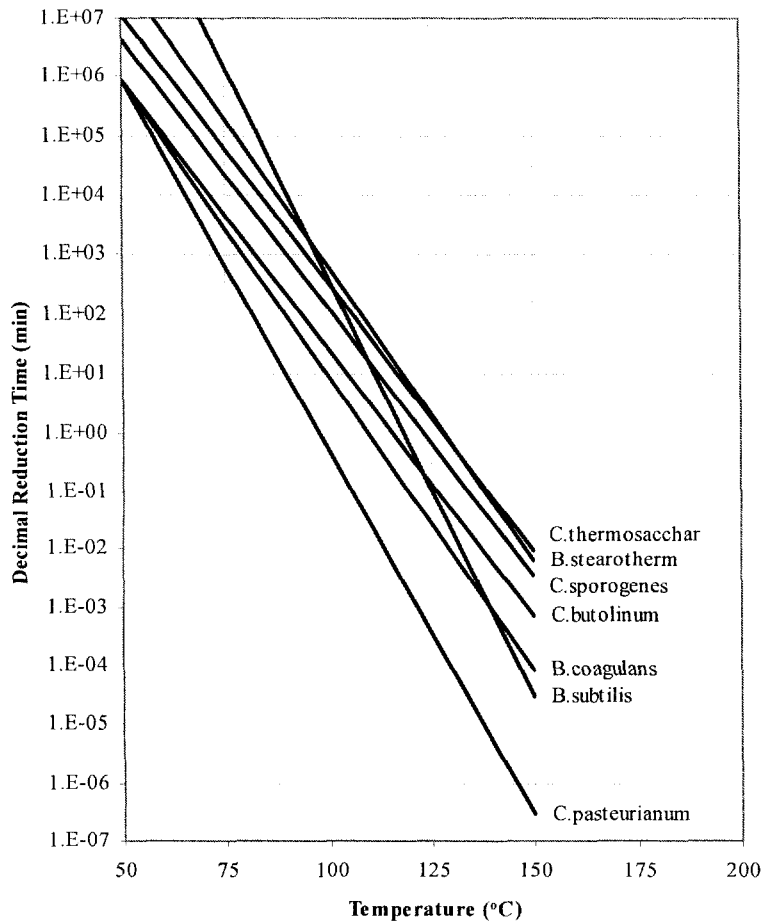
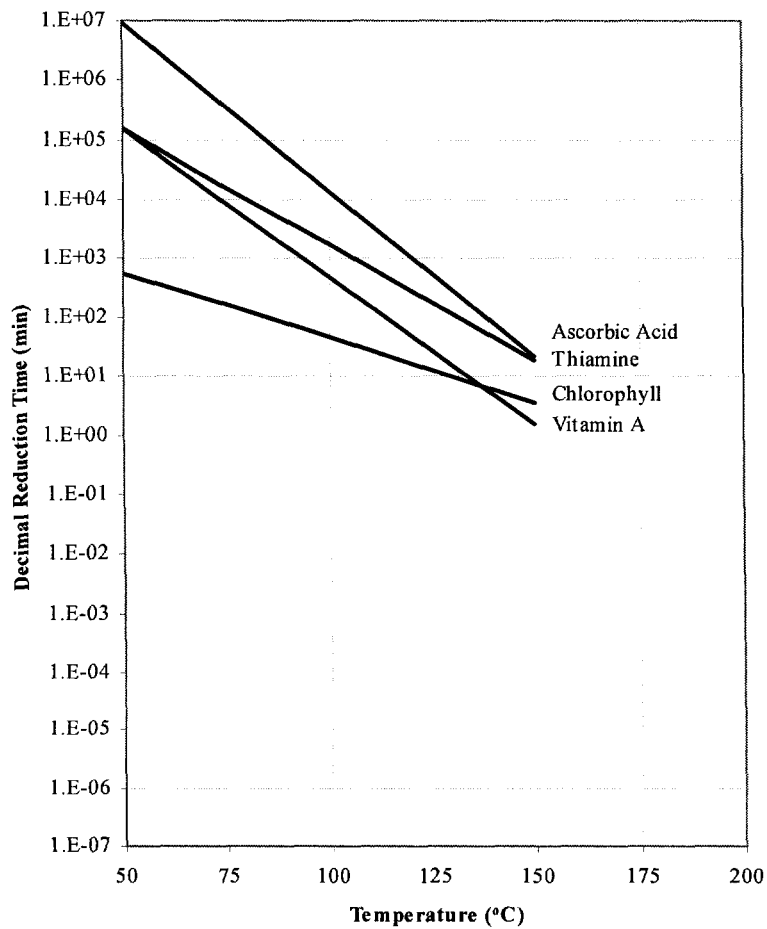
**Figure 8.1** Thermal destruction data of spoilage microorganisms.

Table 8.2 Thermal Damage Data of Food Components

| | Food Component | T_R | D | z |
|---|----------------|-------|------|------|
| 1 | Ascorbic Acid | 121 | 931 | 17.8 |
| 2 | Thiamine | 121 | 254 | 25.4 |
| 3 | Vitamin A | 121 | 43.5 | 20.0 |
| 4 | Chlorophyll | 121 | 15.4 | 45.0 |

T_R : Reference temperature; D : Decimal reduction time; z : Thermal resistance factor

**Figure 8.2** Thermal damage data of food components.

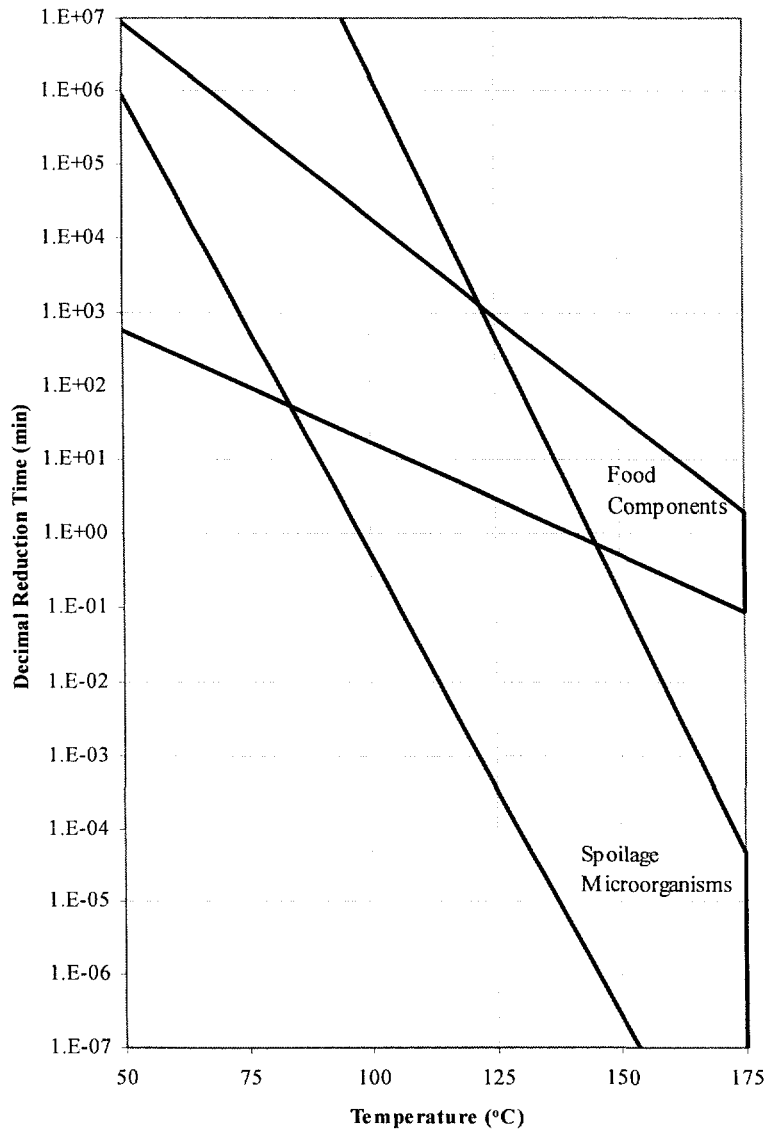


Figure 8.3 Thermal damage data: Comparison between spoilage microorganisms and food components.

7. Application to Milk

The conditions of milk pasteurization or sterilization, and the regulations, which vary from country to country, are summarized in Table 8.3 (Kessler, 1981; Lewis and Heppel, 2000). Milk pasteurization destroys pathogenic microorganisms, but heat resistant organisms survive. Sterilization destroys all heat-resistant microorganisms and microbial spores. Table 8.4 summarizes the thermal resistance data for the microorganisms in raw milk, which are grouped in three categories. In the same table thermal data for three critical food components of the milk are also presented.

Table 8.3 Heating Methods for Milk

| | | | |
|---|------------|--|-----------|
| Pasteurization | | | |
| Batch Heating | 62-65 °C | | 30 min |
| Short time heating (HTST) | 71 °C | | 15-40 s |
| High temperature heating (Flash Heating) | 85-90 °C | | 1-4 s |
| Sterilization | | | |
| Sterilization | 109-115 °C | | 20-40 min |
| Ultra high temperature heat treatment (UHT) | 135-150 °C | | 6-2 s |

Table 8.4 Thermal Resistance Data of Microorganisms and Food Components in Milk

| | T_R , °C | D , min | z , °C | m , - |
|------------------------|------------|-----------|----------|---------|
| Microorganisms | | | | |
| Pathogenic | 75 | 0.01 | 9.5 | 12 |
| Mesophilic | 120 | 0.15 | 10.0 | 9 |
| Thermophilic | 120 | 0.50 | 10.5 | 9 |
| Food Components | | | | |
| Thiamine | 120 | 120 | 30 | |
| Lysine | 120 | 960 | 20 | |
| Ascorbic acid | 120 | 960 | 18 | |

T_R : Reference temperature; D : Decimal reduction time; z : Thermal resistance factor

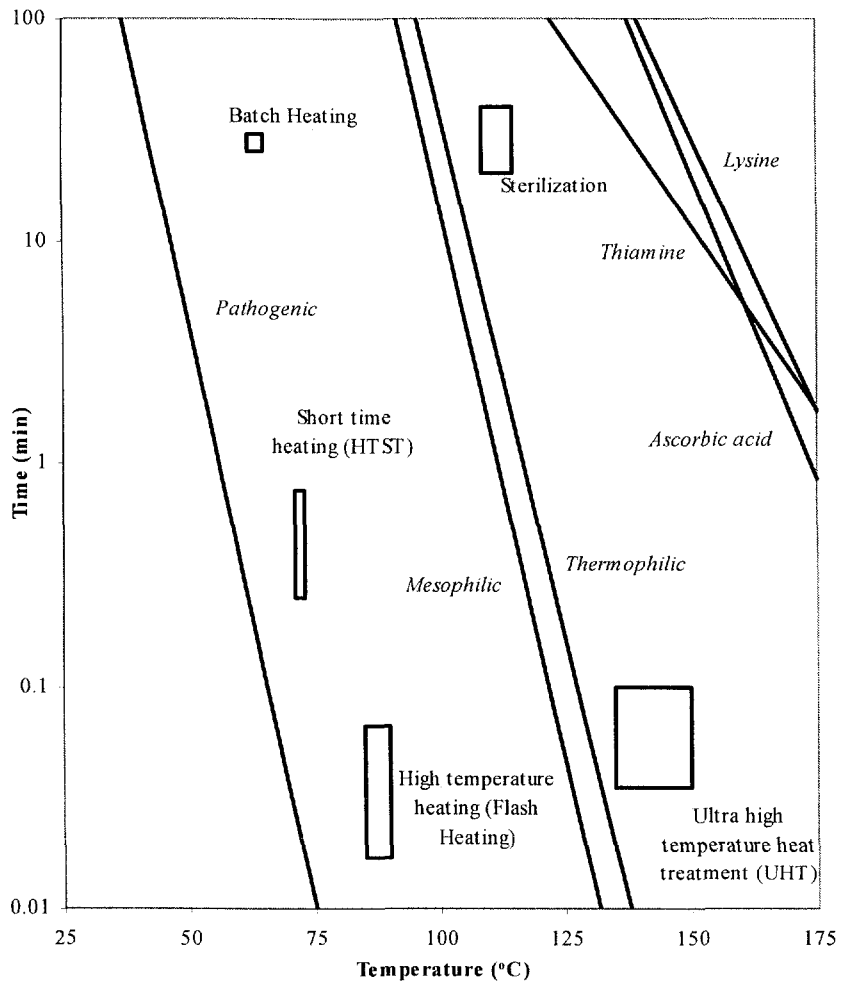


Figure 8.4 Temperature-time data for milk pasteurization or sterilization (D-values).

Figure 8.4 shows graphically the data of Tables 8.3 and 8.4 in a temperature-time diagram. The straight lines represent the decimal reduction time for each microorganism or food component, while rectangular or squares represent the operating region for each practical pasteurization/sterilization process. A more informative diagram is obtained in Figure 8.5, in which the straight lines represent the thermal death time for microorganisms (the corresponding m -values are given in Table 8.4) and the required time to inactivate the food components by 2%.

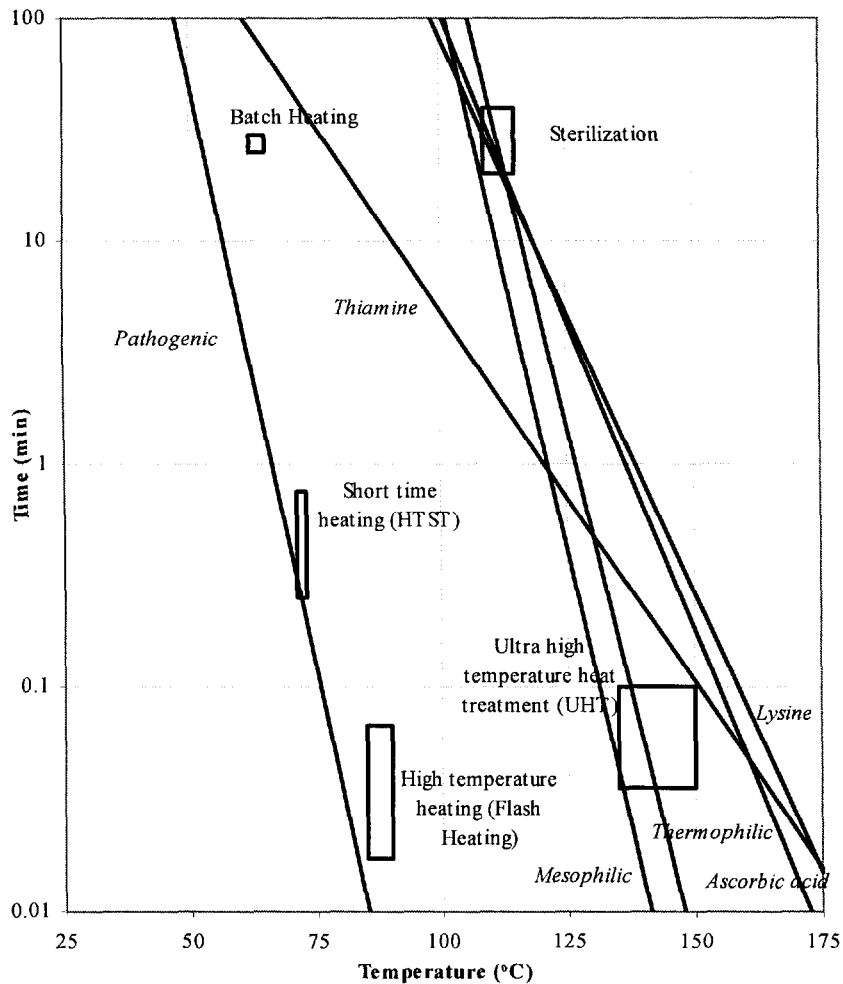


Figure 8.5 Temperature-time data for milk pasteurization or sterilization. Microorganisms, D-values; food components, 2% inactivation.

It is clear that the pasteurizing process completely destroys all the pathogenic microorganisms but most of the mesophilic and thermophilic organisms survive. Sterilization processes adequately destroy all the mesophilic and thermophilic organisms. In all processes the thermal damage of the examined food components are less than 2%. These figures can be more informative, if more data concerning other food components are added, and different percent losses of nutrients are considered.

8. In-Container Thermal Processing

In-container sterilization is applied to the sterilization and pasteurization of solid, semi-solid and liquid foods, packaged in metallic or plastic containers. Calculation of thermal process time is based on the thermal destruction of microorganisms and enzymes and the minimization of undesired thermal damage to heat-sensitive food components. Thermal process design deals primarily with the sterilization of solid non-acid foods ($\text{pH} > 4.5$), packaged in cylindrical metallic containers (cans), heated by unsteady-state conduction. The objective of sterilization is the thermal inactivation of spoilage and toxin-producing bacteria, which may cause food spoilage and food safety problems. Sterilization and pasteurization of liquid foods is relatively easier, due to the faster heat transfer by natural and forced convection.

The principles of thermal inactivation spoilage microorganisms and thermal damage to food components are described in the previous section. Heat transfer by conduction in food containers requires special treatment, and experimental measurements are traditionally required to determine the thermal process data.

The heat penetration in a food container is determined by measuring the temperature T at its geometric center. Heating is provided usually by saturated steam at constant temperature T_h , as shown in Figure 8.6. The food container is usually a metallic can of diameter $2r$ and height $2L$, surrounded by saturated steam of constant temperature T_h . The heating data (T, t) for the food container are plotted on a semi-log diagram, as shown in Figure 8.7. From the heat penetration diagram of Figure 8.7 the following heating parameters are determined:

$$\text{- heating parameter: } f = -1/S \quad (8-13)$$

$$\text{- heating lag factor: } j = (T_h - T_o) / (T_h - T_{po}) \quad (8-14)$$

where S is the slope of the heat penetration straight line (see Figure 8.7), T_o is the initial temperature and T_{po} the pseudo-initial temperature, obtained by extrapolation of the straight line (see Figure 8.7).

It has been generally recognized that it is convenient to treat experimental heating or cooling data using the f and j concept advocated by Ball (1957) and use it as a data fitting tool. Alternatively, the heating parameter f can be estimated analytically from the thermal diffusivity a and the dimensions of the cylindrical container ($2r, 2L$), (Teixeira, 1992):

$$f = \frac{0.398}{\left(\frac{1}{r^2} + \frac{0.427}{L^2} \right) a} \quad (8-15)$$

The thermal diffusivity of food can be estimated from the equation:

$$a = \frac{\lambda}{\rho C_p} \quad (8-16)$$

where λ is the thermal conductivity, ρ the density, and C_p the specific heat (Saravacos and Maroulis, 2001; Rao and Rizvi, 1995; Rahman, 1995).

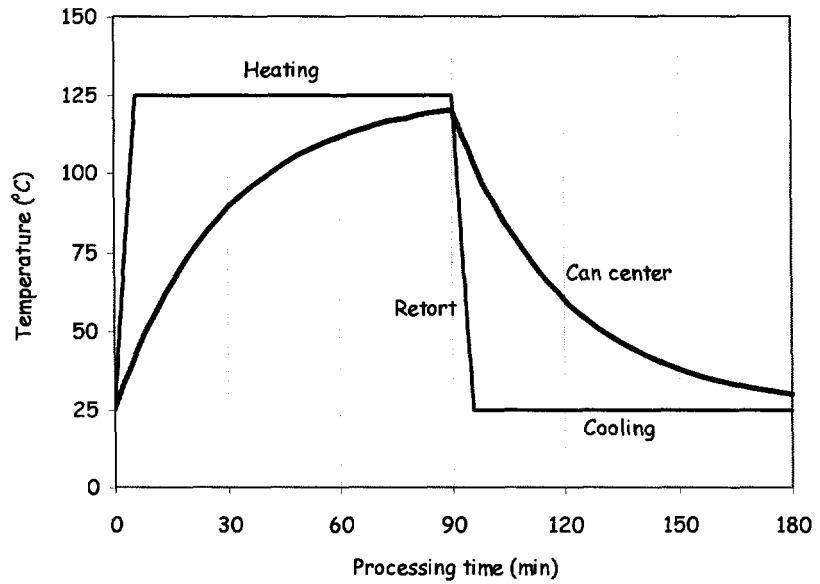


Figure 8.6 Heat penetration profile of in-container heating and cooling foods.

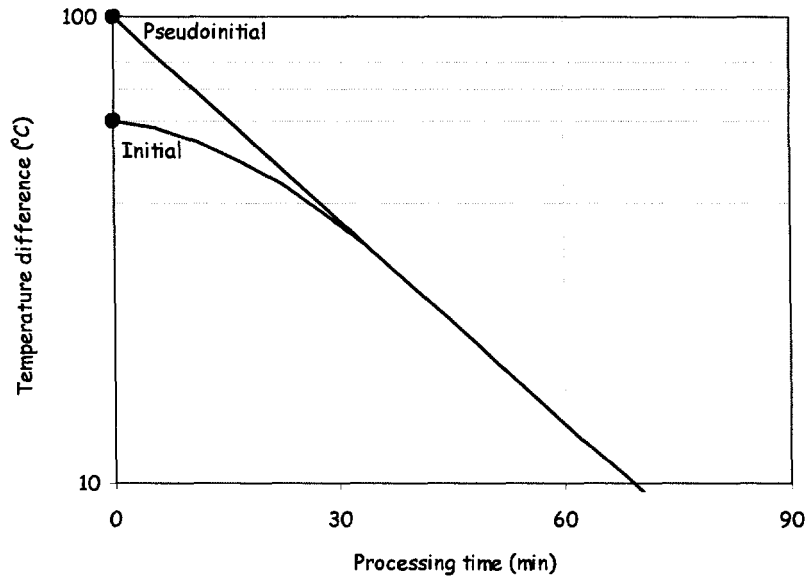


Figure 8.7 Typical heat penetration in-container curve.

The thermal process time to inactivate a target microorganism to the desired level, can be estimated using the concept of paragraph II.4 in this Section, where the time varied temperature at the center of a cylindrical container can be estimated by the analytical equation:

$$\frac{T - T_h}{T_o - T_h} = \exp\left(-\frac{t}{f}\right) \quad (8-17)$$

where T is the temperature at the center of the can at time t , T_h is the temperature of the retort and T_o the initial temperature at the can center. This equation suggests that the heating process follows a first-order kinetic model, where the time constant is equal to the heating factor (Equation 8-15).

The total heating process time t_h (from the time the steam is turned on) is calculated from the equation:

$$t_h = t + 0.58 \text{ CUT} \quad (8-18)$$

where CUT is the “come-up” time, required to bring the sterilizer to the process temperature T_h .

In conventional thermal process design, the process time is calculated using either the general (graphical, Bigelow) or the formula (Ball) methods (Teixeira, 1992; Stoforos, 1993, Downing, 1995; Ramaswamy and Singh, 1997).

In the graphical method, the thermal process time t_p is calculated by graphical (or numerical) integration of heat penetration (T , t) data at the center of the container:

$$t_p = \sum_0^t L \Delta t \quad (8-19)$$

where L is the lethality at temperature T defined as:

$$\log(L) = (T-121)/z \quad (8-20)$$

The Ball formula method calculates the process time t_B as a function of the heating parameters (f , j), the initial temperature T_o , and the temperature difference $g = T_h - T_{max}$, where T_h is the heating temperature and T_{max} is the temperature at the container center at the end of heating:

$$t_B = f \log\left(\frac{j(T_h - T_o)}{g}\right) \quad (8-21)$$

The parameter g is estimated from literature tables or diagrams as a function of the heating factor f , the heating lag factor j and the sterilization time for the microorganisms U at the heating temperature T_h , which is related to the sterilization time F at 121°C:

$$U = F 10^{\frac{T_h - 121}{z}} \quad (8-22)$$

The original Ball formula is based on the lethality at the center of the container. It assumes that cooling has a sterilizing effect and the cooling lag factor is $j_c = 1.41$, which may be different from the heating lag factor j_h . Recent modifications of the formula method consider varying heating/cooling factors, and the lethal effects in the entire food container (Weng and Park, 1997; Afaghi, 1999).

The volume average microorganism survival and quality degradation in the food container can be estimated by the integrated sterility and “cook”-value Equations (1-11) and (1-13) of Chapter 1.

In-container thermal processing of convection-heated foods (liquids or liquid/particle suspensions) requires considerably less time than conduction heating, due to the faster heat transfer rate. Agitation of food containers (horizontal or end-over-end) will increase the heat transfer rate (Saravacos and Kostaropoulos, 2002).

The fastest heating (and shortest processing time) can be achieved when liquid food is mixed thoroughly so that its temperature becomes uniform throughout the container. In this case the heat penetration into the container is given by the equation:

$$\frac{T - T_h}{T_o - T_h} = \exp\left(-\frac{UA}{\rho V C_p} t\right) \quad (8-23)$$

From Equation 8-17, it follows that the heating factor f becomes:

$$f = \frac{\rho V C_p}{UA} \quad (8-24)$$

where V and A are the volume (m^3) and external surface (m^2) of the container, respectively, ρ and C_p are, respectively, the density (kg/m^3) and specific heat (kJ/kgK) of the product, and U is the overall heat transfer coefficient (W/m^2K).

The coefficient U depends mainly on the wall/product heat transfer coefficient, since the thermal resistances of the metallic wall and steam/wall side are negligible.

9. In-Container Sterilizers

Traditional canning of packaged foods is carried out in batch or continuous sterilizers, which range from still retorts to continuous hydrostatic sterilizers. The design and operation of these sterilizers has developed through the years from practical experience of food and equipment manufacturers (Downing, 1995; Saravacos and Kostaropoulos, 2002).

a. Batch Sterilizers

Batch sterilizers (retorts) are used in small to medium size canning operations. The traditional still retort consists of a vertical or horizontal vessel, equipped with the necessary doors, heating and cooling pipes and valves, and control/recording instrumentation (Figure 8.8).

The food cans are loaded in crates of wagons, which are hoisted or moved into the retorts, the lids or doors of which are closed hermetically. The heating medium is usually saturated steam that is introduced at the bottom of the retort through a steam spreader. The retorts are vented from the occluded air. The total heating time (from the moment steam is turned on until turning off) includes the calculated thermal process time plus 58% of the “come-up” time (i.e., the time required for the retort to reach the temperature of the saturated steam). Hot water can also be used as a heating medium.

The sterilizer cans are cooled to near ambient temperature using cold water running through the retort. Overriding air pressure is required in thermal processing and cooling of large cans, flat cans, glass containers, and plastic pouches. The air pressure prevents the bulging and deformation of metallic containers and the popping of lids of glass jars.

Crateless retorts can be operated in semi-continuous manner by using a series of retorts, in which the cans are loaded, processed, unloaded, and cooled in a water canal below.

Batch rotary sterilizers, e.g., the FMC orbitort, are used for the processing of convection heated canned foods, containing food liquids or food particles and syrup or juice. Horizontal or end-over-end rotation of the cans heating and cooling substantially increases the heat transfer rate and reduces the thermal process time.

Improved batch thermal processing equipment includes circulating water, track flow, and flame sterilizers (Saravacos and Kostaropoulos, 2002).

b. Continuous Sterilizers

Continuous sterilizers are advantageous in large canning operations for engineering and economic reasons. The sterilizers are fed continuously with the food cans which are preheated, sterilized and cooled in two sections of the installation. Figure 8.38 shows the diagram of a continuous cooker/cooler (FMC), equipped with special pressure locks for feeding the cooker and discharging the cans from the cooler (pressure vessels). The cans are moved progressively on rotating spiral reels, improving the heat transfer rate of continuous heated foods.

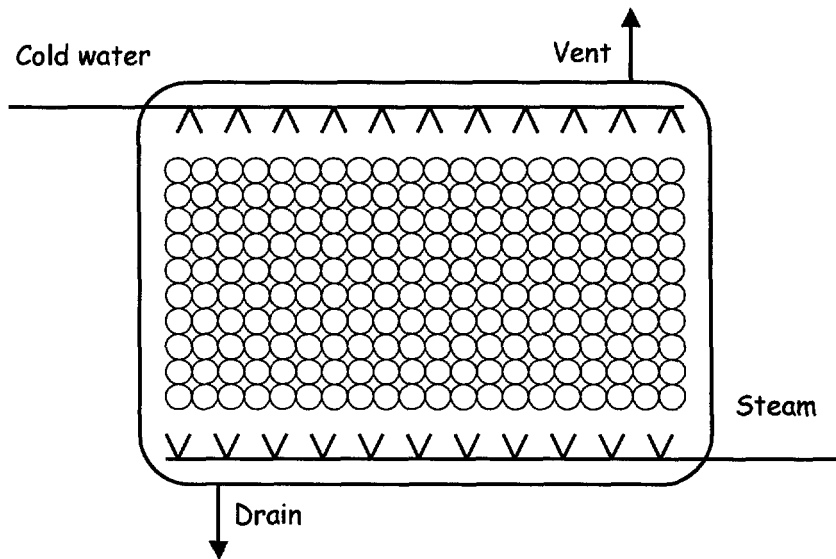


Figure 8.8 Still retort.

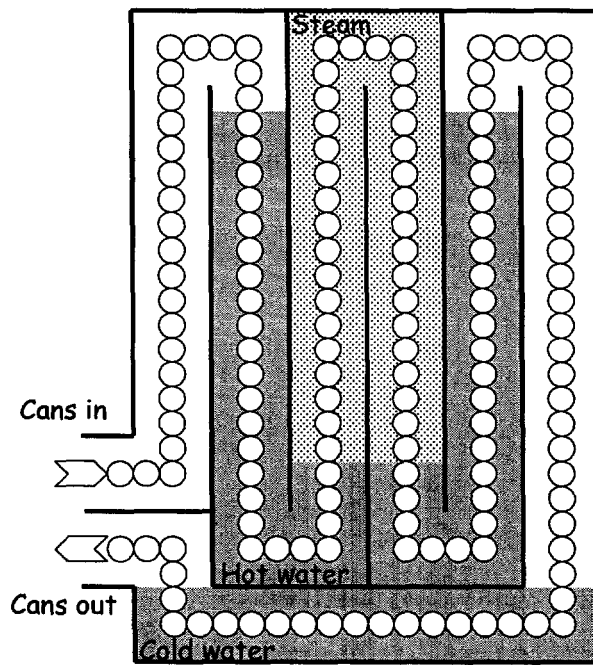


Figure 8.9 Hydrostatic sterilizer.

Hydrostatic sterilizers (Figure 8.9) are used in very large canning operations for processing thousands of cans per hour. They can operate at pressures above atmospheric ($T > 100^{\circ}\text{C}$), which are maintained by water columns (hydrostatic pressure), eliminating the need for closed vessels and pressure locks. They are tall installations, e.g., a pressure of 1 bar requires a water column of about 10m.

The hydrostatic sterilizers consist basically 4 chambers: (a) the hydrostatic feed leg, (b) the sterilization chamber, (c) the hydrostatic discharge leg, and (d) the cooling canal.

The cans are transported through the sterilizer by one or more chain conveyors, operated in parallel. The residence time in the sterilization chamber is defined by the calculated thermal process time of the particular can size.

III. CONTINUOUS FLOW PASTEURIZATION

1. Process Description

The essential requirements in the pasteurization process of fluid food is to bring the temperature up to an appropriate level and hold it for enough time to obtain commercial sterilization for all pathogenic microorganisms. The product is assumed to be well mixed and flow at a uniform (average) velocity in the heat exchangers and the holding tube.

A traditional pasteurization process is presented in Figure 8.10. It uses hot water for heating and a heat pump for cooling. The incoming liquid food flows first through a regenerator and then through a hot water heater. The food is kept into a holding tube for some time. Thereafter the food flows through the other side of the regenerator and finally through the evaporator of a heat pump.

The following types of heat exchangers are commonly used :

- Plate heat exchangers
- Tubular heat exchangers
- Shell and tubes heat exchangers
- Spiral heat exchangers

2. Process Model

The mathematical model of the process presented in Figure 8.10 is summarized in Table 8.5. The variables of the model are summarized in Table 8.6, while the appropriate technical data are presented in Table 8.7.

Equations (E01), (E02), and (E03) in Table 8.5 describe a simple mathematical model for the regenerator. Equation (E01) refers to raw stream, which is preheated, Equation (E02) to pasteurized stream, which is precooled, and Equation (E03) determines the heat flow between the above streams.

Similarly, Equations (E04), (E05), and (E06) in Table 8.5 describe a simple mathematical model for the hot water heater. Equation (E04) refers to raw stream, which is heated to the desired temperature, Equation (E05) to hot water, and Equation (E06) determines the heat flow between the two streams.

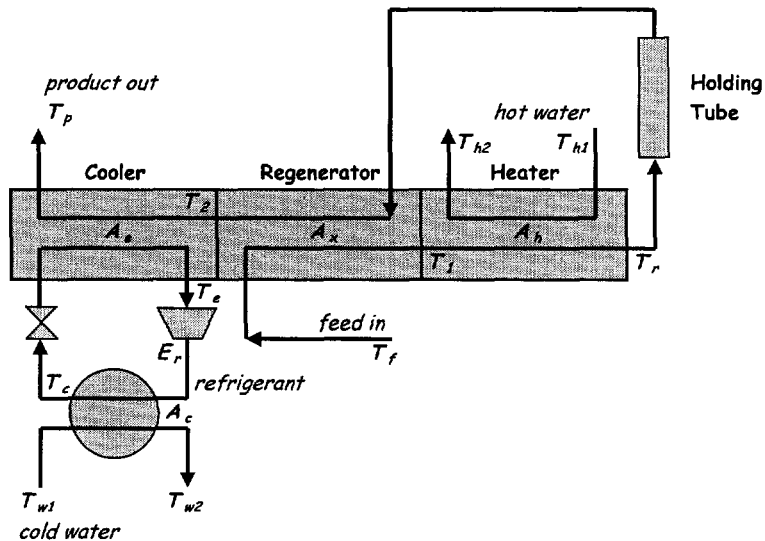


Figure 8.10 Flow sheet for continuous flow pasteurization of foods.

The Equations (E07), (E08), and (E09) refer to the refrigerant evaporator, (E10), (E11) and (E12) to refrigerant condenser and (E13) to refrigerant compressor. Equations (E08) for the refrigerant evaporator, (E10) for the refrigerant condenser and (E13) for the refrigerant compressor come from the shortcut model of refrigeration loops described by Shelton and Grossmann (1985).

Equations (E14), (E15), (E16), (E17), and (E18) calculate the residence time of the processing food in all process equipment, i.e., the holder, the regenerator (though which the food passes twice), the cooler, and the heater. These data compose the time-temperature treatment which leads to the accurate calculation of the microorganism survivors and food component damage. Equation (E14) refers to a holding tube, while Equations (E15), (E16), and (E17) refer to plate heat exchangers (other heat exchanger types may modify these equations). Equation (E18) is the total processing time.

Equation (E19) calculates the ratio of survivors to initial population of a microorganism. It is applied N_{m0} times, one time for each microorganism considered in the analysis. The integration can be calculated numerically, using the time-temperature function obtained from the aforementioned equations.

Similarly, Equation (E20) calculates the ratio of the final to initial concentration of a food component. It is applied N_{fc} times, one time for each food component considered in the analysis.

Two kinds of model constraints are presented in Table 8.8. Food quality constraints, that is Equations (C01) and (C02), and heat transfer constraints, that is Equations (C03), (C04), (C05), (C06), and (C07).

Food quality constraints express the desired specifications for the final product. They are crucial in the design, since they define the feasible region for time-temperature treatment. Equation (C01) is applied N_{mo} times, one time for each microorganism considered in the analysis. It defines the minimum accepted destruction for each microorganism. Equation (C02) is applied N_{fc} times, one time for each food component considered in the analysis. It defines the maximum accepted damage for each food component.

Heat transfer constraints are obvious. They define the minimum accepted temperature difference in a heat exchanger. They are useful in optimization procedure, during which they keep the search into the feasible region.

Process specifications of a typical design problem are presented in Table 8.9, while a degrees-of-freedom analysis is presented in Table 8.10. The proposed mathematical model consists of $(18+N_{mo}+N_{fc})$ equations, involving $(30+N_{mo}+N_{fc})$ variables. Consequently, the degrees of freedom of the system are 12. Typical process specifications spend 7 degrees of freedom and consequently the remaining design variables are 5.

A design variables selection is presented in Table 8.11, while a solution algorithm for the specific problem is summarized in Table 8.12. It is concluded that the resulting solution is simple and no recycle calculations are needed.

The total annual cost of the process TAC is selected as the objective function to be optimized. It is calculated in Table 8.13. Equation (F01) calculates the equipment cost C_{eq} , (E02) the annual operating cost C_{op} , while Equation (F03) defines the total annual cost TAC , which is a weighted sum between the equipment and operating cost. The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (F04). The economic data needed for the cost estimation are summarized in Table 8.14.

In the selection of design variables and their effect on the objective function, the following points should be noted:

- T_i determines the distribution of the heating load between the regenerator (which has no operating cost) and the hot water heater (which has a cost). In other words, T_i determines the extent of heat recovery. T_i varies between T_j (no heat recovery) and T_r (maximum heat recovery, which requires infinite heat transfer area). The optimum is a tradeoff between equipment and operating cost.
- T_c and T_e are crucial in the heat pump design. The range (T_c-T_e) determines the size of the compressor and consequently the equipment and operating cost of the heat pump. It is obvious that the optimum values are obtained when the constraints (C03) and (C04) are valid as equalities.
- T_r and τ_r are the principal design variables in pasteurization processes. They must satisfy the quality constraints of Table 8.8 and optimize the objective function of the process.

Table 8.5 Process Model: Equations**Regenerator**

$$Q_x = F C_{pm} (T_r - T_j) \quad (\text{E01})$$

$$Q_x = F C_{pm} (T_r - T_2) \quad (\text{E02})$$

$$Q_x = U_x A_x [(T_2 - T_j) - (T_r - T_1)] / \ln[(T_2 - T_j) / (T_r - T_1)] \quad (\text{E03})$$

Heater

$$Q_h = F C_{pm} (T_r - T_1) \quad (\text{E04})$$

$$Q_h = F_h C_{pw} (T_{h1} - T_{h2}) \quad (\text{E05})$$

$$Q_h = U_h A_h [(T_{h2} - T_1) - (T_{h2} - T_r)] / \ln[(T_{h2} - T_1) - (T_{h2} - T_r)] \quad (\text{E06})$$

Cooler (Heat Pump Evaporator)

$$Q_e = F C_{pm} (T_2 - T_p) \quad (\text{E07})$$

$$Q_e = [\Delta H_r - C_{pr} (T_c - T_d)] F_r \quad (\text{E08})$$

$$Q_e = U_e A_e [(T_p - T_d) - (T_2 - T_d)] / \ln[(T_p - T_d) / (T_2 - T_d)] \quad (\text{E09})$$

Heat Pump Condenser

$$Q_c = [\Delta H_r (T_c / T_d) - C_{pr} (T_c - T_d)] F_r \quad (\text{E10})$$

$$Q_c = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E11})$$

$$Q_c = U_c A_c [(T_{w1} - T_c) - (T_{w2} - T_d)] / \ln[(T_{w1} - T_c) / (T_{w2} - T_d)] \quad (\text{E12})$$

Heat Pump Compressor

$$E_r = \Delta H_r (T_c / T_e - 1) F_r \quad (\text{E13})$$

Residence Time

$$\tau_r = \rho_m V_r / F \quad (\text{E14})$$

$$\tau_x = \rho_m A_x b / F \quad (\text{E15})$$

$$\tau_h = \rho_m A_h b / F \quad (\text{E16})$$

$$\tau_e = \rho_m A_e b / F \quad (\text{E17})$$

$$\tau = \tau_r + 2\tau_x + \tau_h + \tau_e \quad (\text{E18})$$

Inactivation of Microorganisms

$$\log \left(\frac{N_j}{N_{oj}} \right) = - \int_0^{\tau} \frac{dt}{D_j} \quad \text{where} \quad D_j = D_{Rj} 10^{\frac{T_{Rj} - T}{z_j}} \quad \forall \quad j = 1, 2, \dots, N_{mo} \quad (\text{E19})$$

Thermal Damage of Food Components

$$\log \left(\frac{C_i}{C_{oi}} \right) = - \int_0^{\tau} \frac{dt}{D_i} \quad \text{where} \quad D_i = D_{Ri} 10^{\frac{T_{Ri} - T}{z_i}} \quad \forall \quad i = 1, 2, \dots, N_{fc} \quad (\text{E20})$$

Table 8.6 Process Variables

| | |
|------------|--|
| | Flow rate (kg/s) |
| F | Process liquid |
| F_h | Hot water |
| F_w | Cold water |
| F_r | Refrigerant |
| | Temperature ($^{\circ}\text{C}$) |
| T_f | Feed |
| T_1 | Between regenerator and heater |
| T_r | Holding tube |
| T_2 | Between regenerator and cooler |
| T_p | Product |
| T_e | Refrigerant evaporation |
| T_c | Refrigerant condensation |
| T_{h1} | Hot water inlet |
| T_{h2} | Hot water outlet |
| T_{w1} | Cold water inlet |
| T_{w2} | Cold water outlet |
| | Thermal Load (kW) |
| Q_x | Regenerator |
| Q_h | Heater |
| Q_e | Refrigerant evaporator |
| Q_c | Refrigerant condenser |
| | Heat Transfer Area (m^2) |
| A_x | Regenerator |
| A_h | Heater |
| A_e | Refrigerant evaporator |
| A_c | Refrigerant condenser |
| | Residence Time (min) |
| τ | Total residence time |
| τ_r | Holding tube |
| τ_x | Regenerator |
| τ_h | Heater |
| τ_e | Cooler (Refrigerant evaporator) |
| | Other |
| V_r | Holding vessel (m^3) |
| E_r | Refrigerant compressor (kW) |
| | Population of microorganisms (-) |
| N/N_{oj} | Survivors to initial population ratio of microorganism j , $j = 1, 2, \dots, N_{mo}$ |
| | Concentration of food components (-) |
| C/C_{oi} | Final to initial concentration of food component i , $i = 1, 2, \dots, N_c$ |

Table 8.7 Process Data

| | |
|-----------------------------------|--|
| Liquid food | |
| C_{pm} | Specific heat of liquid food |
| ρ_m | Density |
| Water | |
| C_{pw} | Specific heat of water |
| Refrigerant | |
| C_{pr} | Specific heat of refrigerant |
| ΔH_r | Latent heat of refrigerant condensation |
| Heat transfer coefficients | |
| b | Distance available for flow between two plates of a plate heat exchanger |
| U_x | Regenerator |
| U_h | Heater |
| U_e | Refrigerant evaporator |
| U_c | Refrigerant condenser |

Table 8.8 Process Model: Constraints**Food quality**

$$\frac{N_j}{N_{oj}} \leq 10^{-m_j} \quad \forall \quad j = 1, 2, \dots, N_{mo} \quad (C01)$$

$$\frac{C_{oi} - C_i}{C_{oi}} \leq l_i \quad \forall \quad i = 1, 2, \dots, N_{fc} \quad (C02)$$

where

m_j = Minimum accepted reduction exponent of the j microorganism

l_i = Maximum accepted damage for the i food component

Heat transfer in heat exchangers

$$T_c - T_{w2} \geq \Delta T_{\min} \quad (C03)$$

$$T_p - T_e \geq \Delta T_{\min} \quad (C04)$$

$$T_r - T_1 \geq \Delta T_{\min} \quad (C05)$$

$$T_{h1} - T_r \geq \Delta T_{\min} \quad (C06)$$

$$T_{h2} - T_1 \geq \Delta T_{\min} \quad (C07)$$

where ΔT_{\min} Minimum accepted temperature difference at heat exchangers

Table 8.9 Process Specifications

| | |
|--------------------|-------------------------------|
| Liquid food | |
| F | Feed flow rate |
| T_f | Feed temperature |
| T_p | Product temperature |
| Utilities | |
| T_{h1} | Hot water inlet temperature |
| T_{h2} | Hot water outlet temperature |
| T_{w1} | Cold water inlet temperature |
| T_{w2} | Cold water outlet temperature |

Table 8.10 Degrees-of-Freedom Analysis

| | |
|--------------------|------------------------|
| Process variables | $30 + N_{mo} + N_{fc}$ |
| Process equations | $18 + N_{mo} + N_{fc}$ |
| Degrees of freedom | 12 |
| Degrees of freedom | 12 |
| Specifications | 7 |
| Design variables | 5 |

Table 8.11 Design Variables

| | |
|----------|--|
| T_l | Temperature at the outlet of the regenerator |
| T_r | Temperature in the holding vessel |
| T_e | Evaporation temperature of the refrigerant |
| T_c | Condensation temperature of the refrigerant |
| τ_r | Residence time in the holding tube |

Table 8.12 Model Solution Algorithm

| | | |
|-------|---|----------|
| (E01) | → | Q_x |
| (E02) | → | T_2 |
| (E03) | → | A_x |
| (E04) | → | Q_h |
| (E05) | → | F_h |
| (E06) | → | A_h |
| (E07) | → | Q_c |
| (E08) | → | F_r |
| (E09) | → | A_e |
| (E10) | → | Q_c |
| (E11) | → | F_w |
| (E12) | → | A_c |
| (E13) | → | E_r |
| (E14) | → | V_r |
| (E14) | → | τ_x |
| (E16) | → | τ_h |
| (E17) | → | τ_e |
| (E18) | → | τ |
| (E19) | → | N_j |
| (E20) | → | C_j |

Table 8.13 Cost Analysis**Equipment cost**

$$C_{eq} = C_{exc} A_x^{n_{exc}} + C_{exc} A_h^{n_{exc}} + C_{exc} A_e^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{ves} V_r^{n_{vesc}} + C_{com} E_r^{n_{com}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q_w + C_h Q_h + C_e E_r) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (F04)$$

Table 8.14 Cost Data

| Utility cost | |
|--------------------------------------|--------------------------------------|
| C_e | Cost of electricity (\$/kWh) |
| C_w | Cost of cold water (\$/kWh) |
| C_h | Cost of hot water (\$/kWh) |
| Equipment unit cost | |
| C_{ves} | Holding tube (k\$/m ³) |
| C_{exc} | Heat exchanger (k\$/m ²) |
| C_{com} | Compressor (k\$/kW) |
| Equipment size scaling factor | |
| n_{ves} | Holding tube (-) |
| n_{exc} | Heat exchanger (-) |
| n_{com} | Compressor (-) |
| Other | |
| t_y | Annual operating time (hr/yr) |
| i_r | Interest rate (-) |
| l_f | Lifetime (yr) |

3. Application to Milk

Design a continuous flow thermal processing system (Figure 8.10) for milk pasteurization, according to the specifications presented in Table 8.15. Sterilization requires the inactivation of all microorganisms in milk while pasteurization targets only destruction of pathogenic microorganisms. The necessary technical and economic data are summarized in Table 8.16.

The optimum values of the design variables and the corresponding values of the economic magnitudes are presented in Table 8.17.

The corresponding optimal enthalpy-temperature graph for all the process equipment is shown in Figure 8.11.

A detailed analysis concerning (a) the equipment size and utility flow and (b) the equipment and operating cost is presented in Figure 8.12.

Figure 8.13 shows explicitly the temperature of the milk versus the processing time while the corresponding surviving fraction of the pathogenic microorganisms versus the processing time is presented in Figure 8.14. The surviving fraction of all the food components is equal to one, that is practically no thermal damage happens to useful food components.

Figure 8.15 shows a sensitivity analysis of one of the design variables on the optimum. $\Delta T = T_r - T_l$ is used in the x-axis instead of T_l itself. The total cost is further analyzed to installation and operating cost.

In Figure 8.16, the obtained optimal operating cost is further analyzed to the required resource utilities.

Table 8.15 Process Specifications for Milk Pasteurization

Liquid food

| | | |
|-------|----------|---------------------|
| F | = 5 kg/s | Feed flow rate |
| T_f | = 5 °C | Feed temperature |
| T_p | = 3 °C | Product temperature |

Utilities

| | | |
|----------|----------|-------------------------------|
| T_{h1} | = 100 °C | Hot water inlet temperature |
| T_{h2} | = 80 °C | Hot water outlet temperature |
| T_{w1} | = 15 °C | Cold water inlet temperature |
| T_{w2} | = 35 °C | Cold water outlet temperature |

Quality

Complete inactivation of pathogenic microorganisms
Damage of food components less than 2%

Table 8.16 Technical and Economic Data

| | | |
|--------------------------------------|----------------------------|---|
| Liquid food | | |
| C_{pm} | = 3.78 kJ/kg K | Specific heat |
| ρ_m | = 1025 kg/m ³ | Density |
| Water | | |
| C_{pw} | = 4.18 kJ/kg K | Specific heat |
| Refrigerant | | |
| C_{pr} | = 2.50 kJ/kg K | Specific heat |
| ΔH_r | = 500 kJ/kg | Latent heat |
| Heat transfer coefficients | | |
| b | = 0.005 m | Distance between two plates of a plate heat exchanger |
| U_x | = 2.00 kW/m ² K | Regenerator |
| U_h | = 2.00 kW/m ² K | Heater |
| U_e | = 2.50 kW/m ² K | Refrigerant evaporator |
| U_c | = 2.50 kW/m ² K | Refrigerant condenser |
| Utility cost | | |
| C_e | = 0.10 \$/kWh | Electricity |
| C_h | = 0.04 \$/kWh | Hot water |
| C_w | = 0.02 \$/kWh | Cold water |
| Equipment unit cost | | |
| C_{ves} | = 3.00 k\$/m ³ | Holding tube |
| C_{exc} | = 5.00 k\$/m ³ | Heat exchanger |
| C_{com} | = 2.00 k\$/kW | Compressor |
| Equipment size scaling factor | | |
| n_{ves} | = 0.70 | Holding tube |
| n_{exc} | = 0.75 | Heat exchanger |
| n_{com} | = 0.85 | Compressor |
| Other | | |
| t_y | = 1500 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 8.17 Optimum Values of the Design Variables

Design variables

| | | |
|-------|------|-----|
| T_l | 60 | °C |
| T_e | -7 | °C |
| T_c | 45 | °C |
| T_r | 70 | °C |
| t_r | 0.30 | min |

Cost estimation

| | | |
|----------|-----|--------|
| C_{eq} | 215 | k\$ |
| C_{op} | 29 | k\$/yr |
| TAC | 61 | k\$/yr |

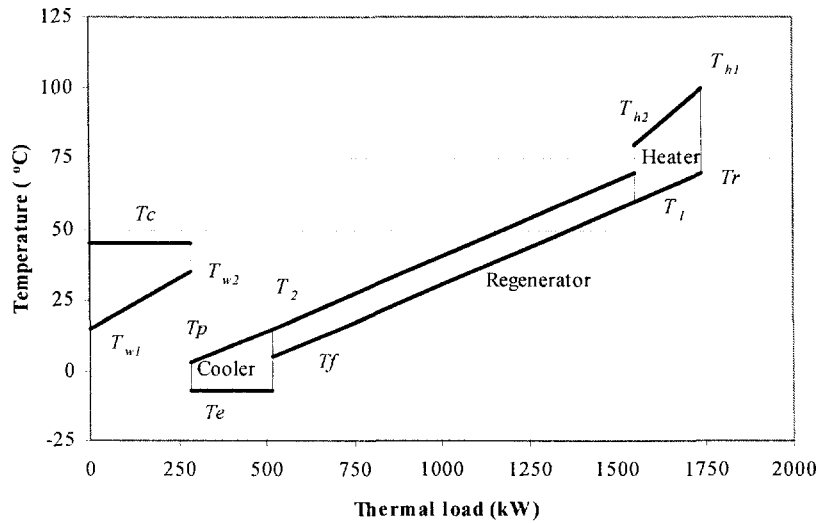


Figure 8.11 Enthalpy-temperature diagram for pasteurization.

| Equipment | Capacity | Load | Installation cost |
|--------------|---------------------|---------|-------------------|
| Regenerator | 52.0 m ² | 1040 kW | 97 k\$ |
| Heater | 3.8 m ² | 189 kW | 14 k\$ |
| Cooler | 6.0 m ² | 227 kW | 19 k\$ |
| Condenser | 6.3 m ² | 287 kW | 20 k\$ |
| Holding tube | 0.1 m ³ | 0 kW | 1 k\$ |
| Compressor | 60 kW | 60 kW | 65 k\$ |
| | | | 215 k\$ |

| Equipment | Utility | Utility flow | Operating cost |
|--------------|-------------|--------------|----------------|
| Regenerator | - | | |
| Heater | Hot water | 8 tn/hr | 11 k\$/yr |
| Cooler | - | | |
| Condenser | Cold water | 10 tn/hr | 9 k\$/yr |
| Holding tube | - | | |
| Compressor | Electricity | 60 kW | 9 k\$/yr |
| | | | 29 k\$/yr |

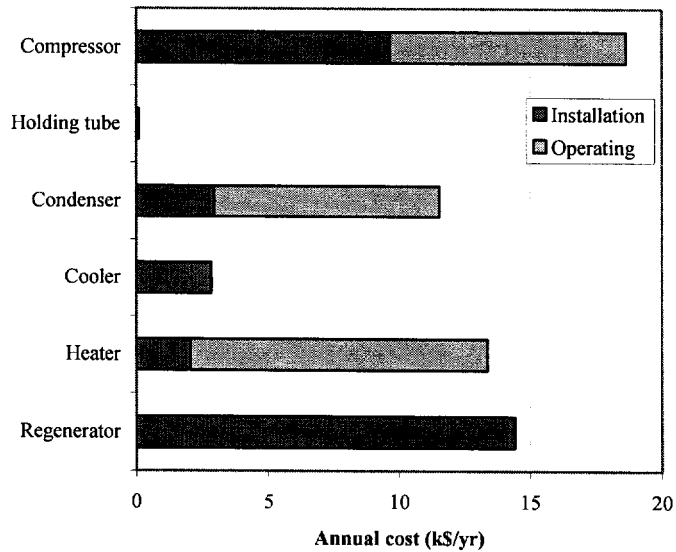


Figure 8.12 Optimum design for milk pasteurization.

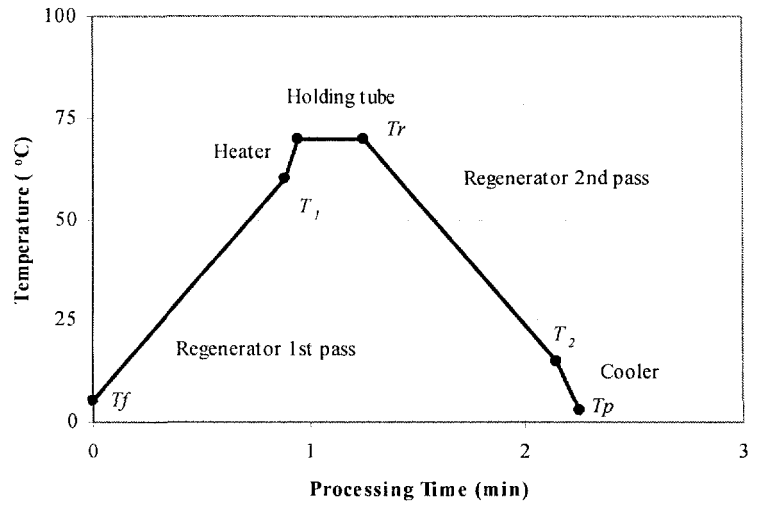


Figure 8.13 Milk temperature versus processing time.

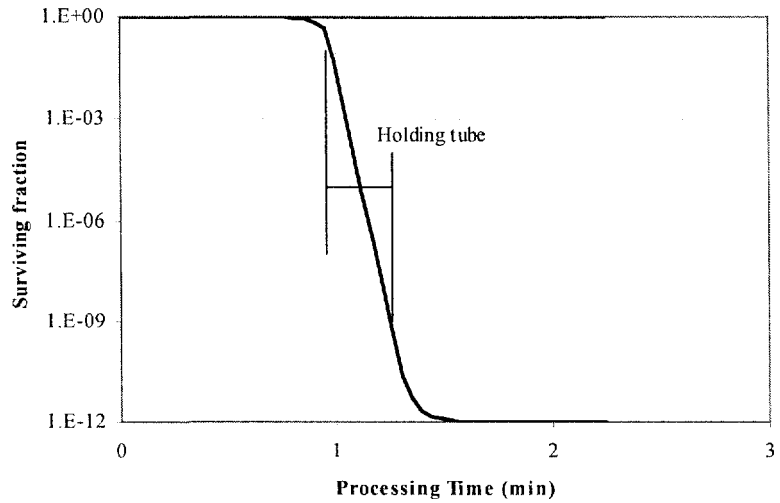


Figure 8.14 Inactivation of microorganisms versus processing time.

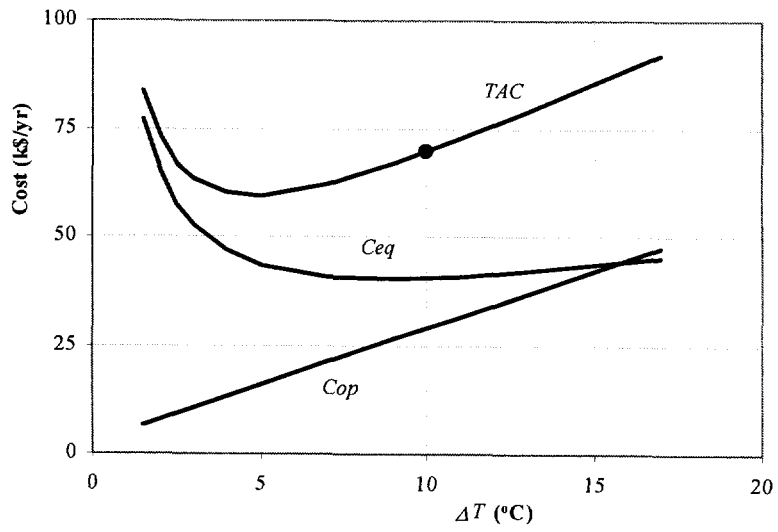


Figure 8.15 Effect of T_1 on the equipment (C_{eq}), operating (C_{op}) and total (TAC) cost.

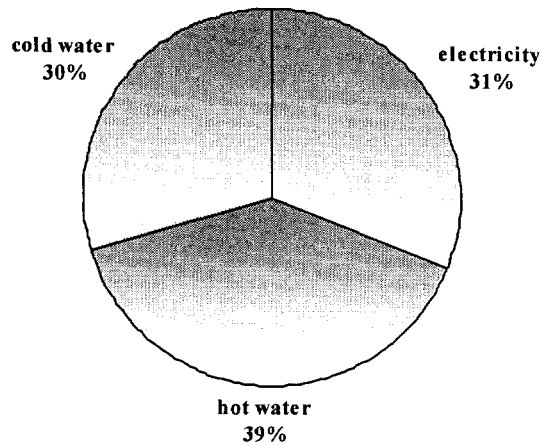
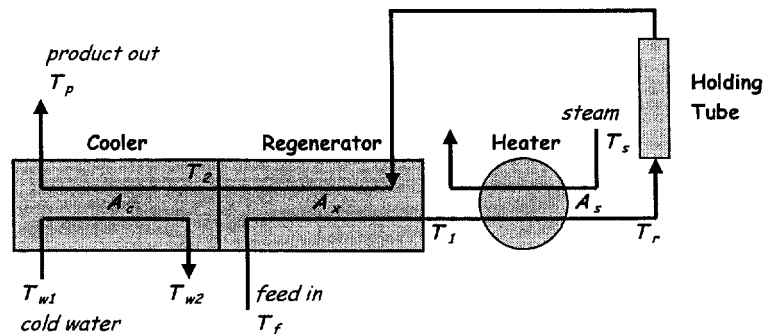


Figure 8.16 Optimal operating cost breakdown.

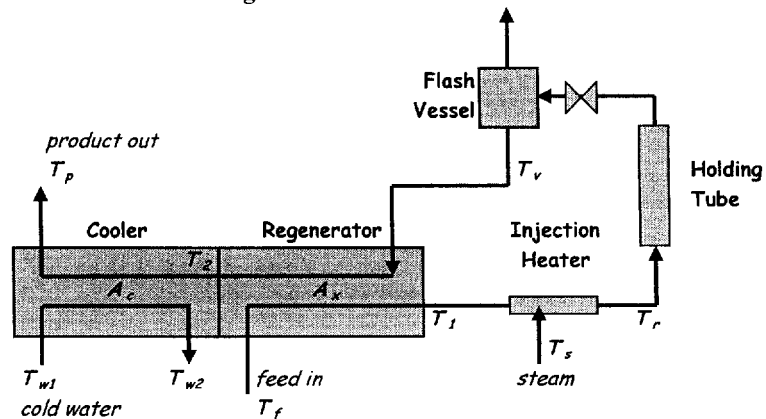
IV. CONTINUOUS FLOW STERILIZATION: INDIRECT STEAM HEATING

1. Process Description

The essential requirements in the sterilization process of fluid food is to bring the temperature up to an appropriate level and hold it for enough time to obtain commercial sterility. Sterilization is conducted at higher temperature than pasteurization and consequently steam heating is necessary. Either indirect steam heating (Figure 8.17a) or injection steam heating (Figure 8.17b) can be used. The sterilized food can be kept at room temperature and consequently cooling by cold water is adequate. The incoming liquid food flows first through a regenerator and then through a heater. The food is kept in a holding tube for the required process time. Thereafter, the food flows through the other side of the regenerator and finally through the cold water cooler. The product is assumed to be well mixed and flow at a uniform (average) velocity in the heat exchangers and the holding tube.



a. Indirect steam heating



b. Injection steam heating

Figure 8.17 Flow sheet for continuous flow thermal processing of foods.

2. Process Model

The mathematical model of the process presented in Figure 8.17a is summarized in Table 8.18. The variables of the model are summarized in Table 8.19, while the appropriate technical data are presented in Table 8.20.

Equations (E01), (E02), and (E03) in Table 8.18 describe a simple mathematical model for the regenerator. Equation (E01) refers to the raw stream, which is preheated, Equation (E02) refers to the sterilized stream, which is precooled, and Equation (E03) determines the heat flow between the above streams.

Similarly, Equations (E04), (E05), and (E06) in Table 8.18 describe a simple mathematical model for the steam heater. Equation (E04) refers to raw stream, which is heated to the desired temperature, Equation (E05) refers to the steam, and Equation (E06) determines the heat flow between the two streams.

Equations (E07), (E08), and (E09) describe a simple mathematical model for a water cooler. Equation (E07) refers to the process stream, which is cooled to the desired temperature, Equation (E08) refers to the cooling water, and Equation (E09) determines the heat flow between the two streams.

Equations (E10), (E11), (E12), (E13), and (E14) calculate the residence time of the processed food in all process equipment, i.e., the holder, the regenerator (though which the food passes twice), the heater, and the cooler. These data compose the time-temperature treatment which leads to the accurate calculation of the microorganism survivors and food component damage. Equation (E10) refers to the holding tube, Equations (E11) and (E13) refer to the plate heat exchangers, while Equation (E12) refers to the shell and tube heater. Equation (E14) is the total processing time.

Equation (E15) calculates the ratio of survivors to initial population of a microorganism. It is applied N_{mo} times, one time for each microorganism considered in the analysis. The integration can be calculated numerically, using the time-temperature function obtained from the aforementioned equations.

Similarly, Equation (E16) calculates the ratio of the final to initial concentration of a food component. It is applied N_{fc} times, one time for each food component considered in the analysis.

Two kinds of model constraints are presented in Table 8.21. Food quality constraints, that is Equations (C01) and (C02), and heat transfer constraints, that is Equations (C03), (C04), (C05), (C06), and (C07).

Food quality constraints express the desired specifications for the final product. They are crucial in the design, since they define the feasible region for time-temperature treatment. Equation (C01) is applied N_{mo} times, one time for each microorganism considered in the analysis. It defines the minimum accepted destruction for each microorganism. Equation (C02) is applied N_{fc} times, one time for each food component considered in the analysis. It defines the maximum accepted damage for each food component.

Heat transfer constraints are obvious. They define the minimum accepted temperature difference in a heat exchanger. They are useful in optimization procedure, during which they keep the search within the feasible region.

Process specifications of a typical design problem are presented in Table 8.22, while a degrees-of-freedom analysis is presented in Table 8.23. The proposed mathematical model consists of $(14+N_{mo}+N_{jc})$ equations, involving $(23+N_{mo}+N_{jc})$ variables. Consequently, the degrees of freedom of the system are 9. Typical process specifications spend 6 degrees of freedom and consequently the remaining design variables are 3.

A design variables selection is presented in Table 8.24, while a solution algorithm for the specific problem is summarized in Table 8.25. It is concluded that the resulting solution is simple and no recycle calculations are needed.

The total annual cost of the process TAC is selected as the objective function to be optimized. It is calculated in Table 8.26. Equation (F01) calculates the equipment cost C_{eq} , (F02) the annual operating cost C_{op} , while Equation (F03) defines the total annual cost TAC , which is a weighted sum between the equipment and operating cost. The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (F04). The economic data needed for the cost estimation are summarized in Table 8.27.

In the selection of design variables and their effect on the objective function, the following points should be noted:

- T_I determines the distribution of the heating load between the regenerator (which has no operating cost) and the steam heater (which consumes heating steam). In other words, T_I determines the extent of heat recovery. T_I varies between T_f (no heat recovery) and T_r (maximum heat recovery, which requires infinite heat transfer area). The optimum is a tradeoff between equipment and operating cost.
- T_r and τ_r are the principal design variables in sterilization processes. They must satisfy the quality constraints of Table 8.21 and optimize the objective function of the process.

Table 8.18 Process Model: Equations**Regenerator**

$$Q_x = F C_{pm} (T_r - T_p) \quad (\text{E01})$$

$$Q_x = F C_{pm} (T_r - T_2) \quad (\text{E02})$$

$$Q_x = U_x A_x [(T_2 - T_p) - (T_r - T_1)] / \ln[(T_2 - T_p) / (T_r - T_1)] \quad (\text{E03})$$

Steam Heater

$$Q_s = F C_{pm} (T_r - T_1) \quad (\text{E04})$$

$$Q_s = F_s \Delta H_s \quad (\text{E05})$$

$$Q_s = U_s A_s [(T_s - T_{1s}) - (T_s - T_{2s})] / \ln[(T_s - T_{1s}) / (T_s - T_{2s})] \quad (\text{E06})$$

Cooler

$$Q_c = F C_{pm} (T_2 - T_p) \quad (\text{E07})$$

$$Q_c = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E08})$$

$$Q_c = U_c A_c [(T_p - T_{w1}) - (T_r - T_{w2})] / \ln[(T_p - T_{w1}) / (T_r - T_{w2})] \quad (\text{E09})$$

Residence Time

$$\tau_r = \rho_m V_r / F \quad (\text{E10})$$

$$\tau_x = \rho_m A_x b / F \quad (\text{E11})$$

$$\tau_s = \rho_m A_s d / 4F \quad (\text{E12})$$

$$\tau_c = \rho_m A_c b / F \quad (\text{E13})$$

$$\tau = \tau_r + 2\tau_x + \tau_s + \tau_c \quad (\text{E14})$$

Inactivation of Microorganisms

$$\log\left(\frac{N_j}{N_{oj}}\right) = -\int_0^{\tau} \frac{dt}{D_j} \quad \text{where } D_j = D_{Rj} 10^{\frac{T_{Rj}-T}{z_j}} \quad \forall j = 1, 2, \dots, N_{mo} \quad (\text{E15})$$

Thermal Damage of Food Components

$$\log\left(\frac{C_i}{C_{oi}}\right) = -\int_0^{\tau} \frac{dt}{D_i} \quad \text{where } D_i = D_{Ri} 10^{\frac{T_{Ri}-T}{z_i}} \quad \forall i = 1, 2, \dots, N_{fc} \quad (\text{E16})$$

Table 8.19 Process Variables

| | |
|--------------|--|
| | Flow rate (kg/s) |
| F | Process liquid |
| F_s | Steam |
| F_w | Cooling water |
| | Temperature (°C) |
| T_f | Feed |
| T_1 | Between regenerator and heater |
| T_r | Holding tube |
| T_2 | Between regenerator and cooler |
| T_p | Product |
| T_s | Steam |
| T_{wi} | Cooling water inlet |
| T_{w2} | Cooling water outlet |
| | Thermal Load (kW) |
| Q_x | Regenerator |
| Q_s | Steam heater |
| Q_c | Cooler |
| | Heat Transfer Area (m²) |
| A_x | Regenerator |
| A_s | Steam heater |
| A_c | Cooler |
| | Residence Time (min) |
| τ | Total residence time |
| τ_r | Holding tube |
| τ_x | Regenerator |
| τ_s | Steam heater |
| τ_c | Cooler |
| | Other |
| V_r | Volume of holding tube (m ³) |
| | Population of microorganisms (-) |
| N_j/N_{0j} | Survivors to initial population ratio of microorganism j , $j = 1, 2, \dots, N_{mo}$ |
| | Concentration of food components (-) |
| C_i/C_{0i} | Final to initial concentration of food component i , $i = 1, 2, \dots, N_{fc}$ |

Table 8.20 Process Data

| | |
|-----------------------------------|--|
| Liquid food | |
| C_{pm} | Specific heat of liquid food |
| ρ_m | Density |
| Heating Steam | |
| ΔH_s | Latent heat of steam condensation |
| Cooling Water | |
| C_{pw} | Specific heat of cooling water |
| Heat transfer coefficients | |
| b | Distance available for flow between two plates of a plate heat exchanger |
| d | Tube diameter of a shell and tube heat exchanger |
| U_x | Regenerator |
| U_s | Steam Heater |
| U_c | Cooler |

Table 8.21 Process Model: Constraints**Food quality**

$$\frac{N_j}{N_{oj}} \leq 10^{-m_j} \quad \forall \quad j = 1, 2, \dots, N_{mo} \quad (\text{C01})$$

$$\frac{C_{oi} - C_i}{C_{oi}} \leq l_i \quad \forall \quad i = 1, 2, \dots, N_{fc} \quad (\text{C02})$$

where

- m_j Minimum accepted reduction exponent of the j microorganism
 l_i Maximum accepted damage for the i food component

Heat transfer in heat exchangers

$$T_p - T_{w1} \geq \Delta T_{\min} \quad (\text{C03})$$

$$T_2 - T_{w2} \geq \Delta T_{\min} \quad (\text{C04})$$

$$T_2 - T_f \geq \Delta T_{\min} \quad (\text{C05})$$

$$T_r - T_1 \geq \Delta T_{\min} \quad (\text{C06})$$

$$T_s - T_r \geq \Delta T_{\min} \quad (\text{C07})$$

where

- ΔT_{\min} Minimum accepted temperature difference at heat exchangers
-

Table 8.22 Process Specifications

Liquid food

| | |
|-------|---------------------|
| F | Feed flow rate |
| T_f | Feed temperature |
| T_p | Product temperature |

Utilities

| | |
|----------|----------------------------------|
| T_s | Heating steam temperature |
| T_{w1} | Cooling water inlet temperature |
| T_{w2} | Cooling water outlet temperature |

Table 8.23 Degrees-of-Freedom Analysis

| | |
|--------------------|------------------------|
| Process variables | $23 + N_{mo} + N_{fc}$ |
| Process equations | $14 + N_{mo} + N_{fc}$ |
| Degrees of freedom | 9 |
| Degrees of freedom | 9 |
| Specifications | 6 |
| Design variables | 3 |

Table 8.24 Design Variables

| | |
|----------|--|
| T_l | Temperature at the outlet of the regenerator |
| T_r | Temperature in the holding tube |
| τ_r | Residence time in the holding tube |

Table 8.25 Model Solution Algorithm

| | | |
|-------|---|----------|
| (E01) | → | Q_x |
| (E02) | → | T_2 |
| (E03) | → | A_x |
| (E04) | → | Q_s |
| (E05) | → | F_s |
| (E06) | → | A_s |
| (E07) | → | Q_c |
| (E08) | → | F_w |
| (E09) | → | A_c |
| (E10) | → | V_r |
| (E11) | → | τ_x |
| (E12) | → | τ_s |
| (E13) | → | τ_c |
| (E14) | → | τ |
| (E15) | → | N_j |
| (E16) | → | C_j |

Table 8.26 Cost Analysis**Equipment cost**

$$C_{eq} = C_{exc} A_x^{n_{exc}} + C_{exc} A_s^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{ves} V_r^{n_{vesc}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q_w + C_s Q_s) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (F04)$$

Table 8.27 Cost Data

| | |
|--------------------------------------|--------------------------------------|
| Utility cost | |
| C_w | Cost of cooling water (\$/kWh) |
| C_s | Cost of heating steam (\$/kWh) |
| Equipment unit cost | |
| C_{ves} | Holding tube (k\$/m ³) |
| C_{exc} | Heat exchanger (k\$/m ²) |
| Equipment size scaling factor | |
| n_{ves} | Holding tube (-) |
| n_{exc} | Heat exchanger (-) |
| Other | |
| t_y | Annual operating time (hr/yr) |
| i_r | Interest rate (-) |
| l_f | Lifetime (yr) |

3. Application to Milk

Design a continuous flow thermal processing system (Figure 8.17a) for milk sterilization using indirect steam heating, according to the specifications presented in Table 8.28. Sterilization requires the inactivation of all microorganisms in milk while pasteurization targets only destruction of pathogenic microorganisms. The necessary technical and economic data are summarized in Table 8.29.

The optimum values of the design variables and the corresponding values of the economic magnitudes are presented in Table 8.30.

The corresponding optimal enthalpy-temperature graph for all the process equipment is shown in Figure 8.18.

Figure 8.19 shows explicitly the temperature of the milk versus the processing time. The corresponding surviving fraction of the microorganisms versus the processing time is presented in Figure 8.20, while the surviving fraction of some food components is shown in Figure 8.21.

Finally, the required equipment size and utility flows as well as the corresponding cost, are summarized in Figure 8.22.

Table 8.28 Process Specifications for Milk Sterilization

Liquid food

| | | |
|-------|----------|---------------------|
| F | = 5 kg/s | Feed flow rate |
| T_f | = 5 °C | Feed temperature |
| T_p | = 25 °C | Product temperature |

Utilities

| | | |
|----------|----------|-------------------------------|
| T_s | = 160 °C | Steam temperature |
| T_{w1} | = 15 °C | Cold water inlet temperature |
| T_{w2} | = 35 °C | Cold water outlet temperature |

Quality

Complete inactivation of all microorganisms
Damage of food components less than 2%

Table 8.29 Technical and Economic Data

| | | |
|--------------------------------------|----------------------------|---|
| Liquid food | | |
| C_{pm} | = 3.78 kJ/kg K | Specific heat |
| ρ_m | = 1025 kg/m ³ | Density |
| Water | | |
| C_{pw} | = 4.18 kJ/kg K | Specific heat |
| Steam (160°C) | | |
| C_{ps} | = 1.90 kJ/kg K | Specific heat |
| ΔH_s | = 2000 kJ/kg | Latent heat |
| Heat transfer coefficients | | |
| b | = 0.005 m | Distance between two plates of a plate heat exchanger |
| d | = 0.020 m | Tube diameter of a shell and tubes heat exchanger |
| U_x | = 2.00 kW/m ² K | Regenerator |
| U_s | = 2.50 kW/m ² K | Heater |
| U_c | = 2.00 kW/m ² K | Cooler |
| Utility cost | | |
| C_e | = 0.10 \$/kWh | Electricity |
| C_s | = 0.05 \$/kWh | Steam |
| C_w | = 0.02 \$/kWh | Cold water |
| Equipment unit cost | | |
| C_{ves} | = 3.00 k\$/m ³ | Holding tube or flash vessel |
| C_{exc} | = 5.00 k\$/m ³ | Heat exchanger |
| Equipment size scaling factor | | |
| n_{ves} | = 0.70 | Holding tube or flash vessel |
| n_{exc} | = 0.75 | Heat exchanger |
| Other | | |
| t_y | = 1500 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 8.30 Optimum Values of the Design VariablesDesign variables

| | | |
|----------|------|-----|
| T_l | 113 | °C |
| T_r | 133 | °C |
| τ_r | 0.23 | min |

Cost estimation

| | | |
|----------|-----|--------|
| C_{eq} | 110 | k\$ |
| C_{op} | 28 | k\$/yr |
| TAC | 45 | k\$/yr |

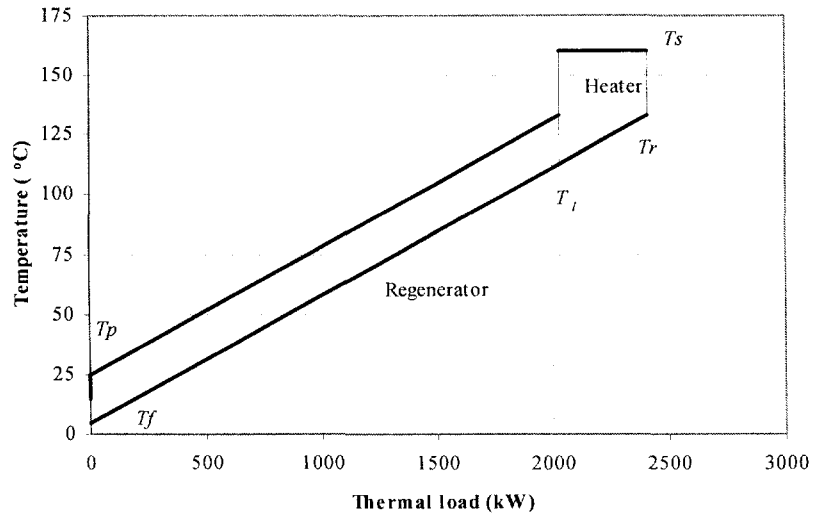


Figure 8.18 Enthalpy-temperature diagram for sterilization using indirect steam heating.

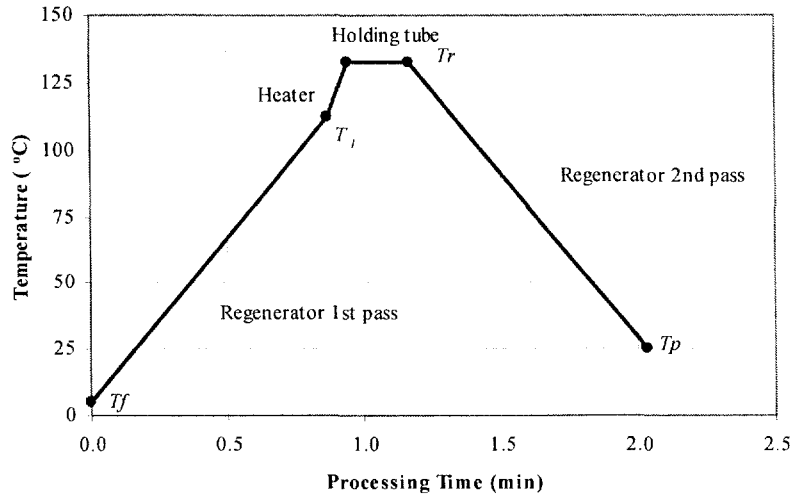


Figure 8.19 Milk temperature versus processing time.

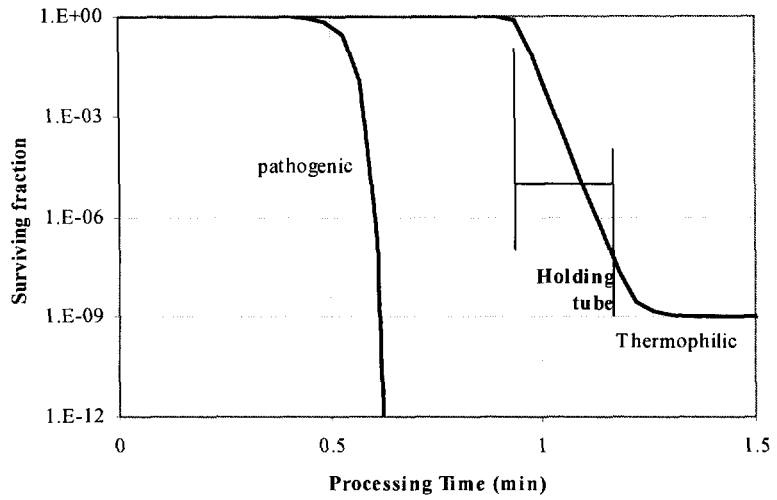


Figure 8.20 Inactivation of microorganisms versus processing time.

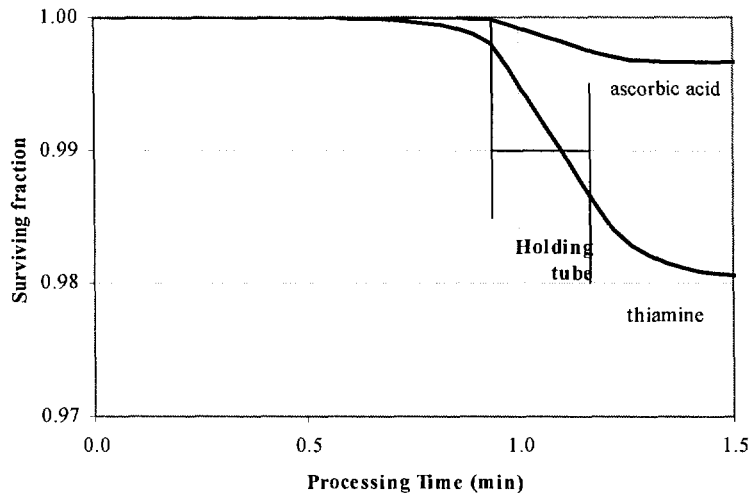


Figure 8.21 Destruction of food components versus processing time.

| Equipment | Capacity | Load | Installation cost |
|--------------|---------------------|---------|-------------------|
| Regenerator | 50.9 m ² | 2035 kW | 95.2 k\$ |
| Heater | 4.1 m ² | 378 kW | 14.5 k\$ |
| Cooler | 0.0 m ² | 0 kW | 0.0 k\$ |
| Holding tube | 0.1 m ³ | 0 kW | 0.4 k\$ |
| | | | <u>110.2 k\$</u> |

| Equipment | Utility | Utility flow | Operating cost |
|--------------|------------|--------------|--------------------|
| Regenerator | - | | |
| Heater | Hot water | 0.54 tn/hr | 28.4 k\$/y1 |
| Cooler | Cold water | 0.00 tn/hr | 0.0 k\$/y1 |
| Holding tube | - | | |
| | | | <u>28.4 k\$/y1</u> |

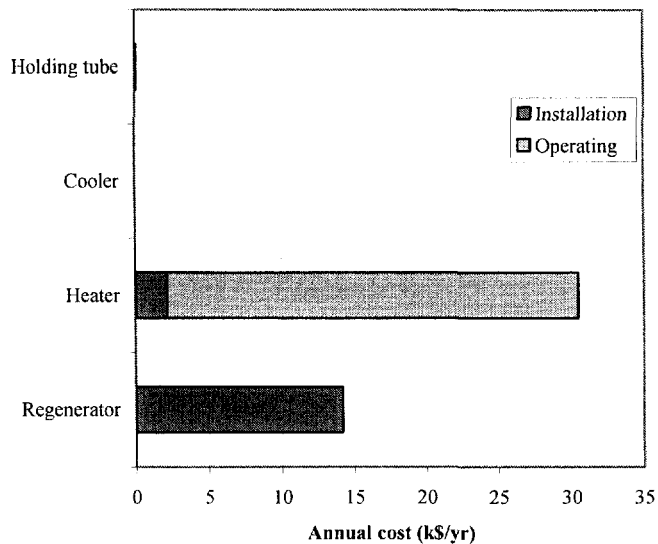


Figure 8.22 Equipment and operating cost.

V. CONTINUOUS FLOW STERILIZATION: INJECTION STEAM HEATING

1. Process Description

The flow sheet of a continuous flow sterilization process using injection steam heating is described in Section 8.IV.1 and in Figure 8.17b.

2. Process Model

The mathematical model of the process presented in Figure 8.17b is summarized in Table 8.31. The variables of the model are summarized in Table 8.32, while the appropriate technical data are presented in Table 8.33.

Equations (E01), (E02), and (E03) in Table 8.31 describe a simple mathematical model for the regenerator. Equation (E01) refers to the raw stream, which is preheated, Equation (E02) refers to the sterilized stream, which is precooled, and Equation (E03) determines the heat flow between the above streams.

Equations (E04), (E05) describe the heat balance in the steam injector. Equation (E04) refers to the raw stream, which is heated to the desired temperature, and Equation (E05) to the steam.

Equation (E06) is a heat balance in the flash vessel.

Equations (E07), (E08), and (E09) describe a simple mathematical model for a water cooler. Equation (E07) refers to the process stream, which is cooled to the desired temperature, Equation (E08) refers to the cooling water, and Equation (E09) determines the heat flow between the two streams.

Equations (E10), (E11), (E12), (E13), and (E14) calculate the residence time of the processed food in all process equipment, i.e., the holder, the regenerator (though which the food passes twice), the flash vessel, and the cooler. These data compose the time-temperature treatment which leads to the accurate calculation of the microorganism survivors and food component damage. Equation (E10) refers to the holding tube, Equations (E11) and (E13) refer to the plate heat exchangers, and Equation (E12) to the flash vessel. Equation (E14) is the total processing time.

Equation (E15) calculates the ratio of survivors to initial population of a microorganism. It is applied N_{mo} times, one time for each microorganism considered in the analysis. The integration can be calculated numerically, using the time-temperature function obtained from the aforementioned equations.

Similarly, Equation (E16) calculates the ratio of the final to initial concentration of a food component. It is applied N_c times, one time for each food component considered in the analysis.

Two kinds of model constraints are presented in Table 8.34. Food quality constraints, that is Equations (C01) and (C02), and heat transfer constraints, that is Equations (C03), (C04), (C05), and (C06).

Food quality constraints express the desired specifications for the final product. They are crucial in the design, since they define the feasible region for time-temperature treatment. Equation (C01) is applied N_{mo} times, one time for each microorganism considered in the analysis. It defines the minimum accepted destruction for each microorganism. Equation (C02) is applied N_{fc} times, one time for each food component considered in the analysis. It defines the maximum accepted damage for each food component.

Heat transfer constraints are obvious. They define the minimum accepted temperature difference in a heat exchanger. They are useful in optimization procedure, during which they keep the search into the feasible region.

Process specifications of a typical design problem are presented in Table 8.35, while a degrees-of-freedom analysis is presented in Table 8.36. The proposed mathematical model consists of $(14+N_{mo}+N_{fc})$ equations, involving $(24+N_{mo}+N_{fc})$ variables. Consequently, the degrees of freedom of the system are 10. Typical process specifications spend 6 degrees of freedom and consequently the remaining design variables are 4.

A design variables selection is presented in Table 8.37, while a solution algorithm for the specific problem is summarized in Table 8.38. It is concluded that the resulting solution is simple and no recycle calculations are needed.

The total annual cost of the process TAC is selected as the objective function to be optimized. It is calculated in Table 8.39. Equation (F01) calculates the equipment cost C_{eq} , (F02) the annual operating cost C_{op} , while Equation (F03) defines the total annual cost TAC , which is a weighted sum between the equipment and operating cost. The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (F04). The economic data needed for the cost estimation are summarized in Table 8.40.

In the selection of design variables and their effect on the objective function, the following points should be noted:

- T_i determines the distribution of the heating load between the regenerator (which has no operating cost) and the steam heater (which consumes heating steam). In other words, T_i determines the extent of heat recovery. T_i varies between T_f (no heat recovery) and T_r (maximum heat recovery, which requires infinite heat transfer area). The optimum is a tradeoff between equipment and operating cost.
- T_r and τ_r are the principal design variables in sterilization processes. They must satisfy the quality constraints of Table 8.34 and optimize the objective function of the process.

Table 8.31 Process Model: Equations**Regenerator**

$$Q_x = F C_{pm} (T_r - T_j) \quad (\text{E01})$$

$$Q_x = F C_{pm} (T_r - T_2) \quad (\text{E02})$$

$$Q_x = U_x A_x [(T_2 - T_j) - (T_r - T_1)] / \ln[(T_2 - T_j) / (T_r - T_1)] \quad (\text{E03})$$

Steam Injector

$$Q_s = F C_{pm} (T_r - T_1) \quad (\text{E04})$$

$$Q_s = F_s dH_s + F_s C_{pw} (T_s - T_r) \quad (\text{E05})$$

Flash Vessel

$$F C_{pm} (T_r - T_v) + F_s C_{pw} (T_r - T_v) = F_s [dH_s + (C_{pw} - C_{ps})(T_r - T_v)] \quad (\text{E06})$$

Cooler

$$Q_c = F C_{pm} (T_2 - T_p) \quad (\text{E07})$$

$$Q_c = F_w C_{pw} (T_w2 - T_w1) \quad (\text{E08})$$

$$Q_c = U_c A_c [(T_p - T_w1) - (T_2 - T_w2)] / \ln[(T_p - T_w1) / (T_2 - T_w2)] \quad (\text{E09})$$

Residence Time

$$\tau_r = \rho_m V_r / F \quad (\text{E10})$$

$$\tau_x = \rho_m A_x b / F \quad (\text{E11})$$

$$\tau_f = \rho_m V_f (1 - f_v) / F \quad (\text{E12})$$

$$\tau_c = \rho_m A_c b / F \quad (\text{E13})$$

$$\tau = \tau_r + 2\tau_x + \tau_f + \tau_c \quad (\text{E14})$$

Inactivation of Microorganisms

$$\log\left(\frac{N_j}{N_{oj}}\right) = - \int_0^{\tau} \frac{dt}{D_j} \quad \text{where} \quad D_j = D_{Rj} 10^{\frac{T_{Rj} - T}{z_j}} \quad \forall \quad j = 1, 2, \dots, N_{mo} \quad (\text{E15})$$

Thermal Damage of Food Components

$$\log\left(\frac{C_i}{C_{oi}}\right) = - \int_0^{\tau} \frac{dt}{D_i} \quad \text{where} \quad D_i = D_{Ri} 10^{\frac{T_{Ri} - T}{z_i}} \quad \forall \quad i = 1, 2, \dots, N_{fc} \quad (\text{E16})$$

Table 8.32 Process Variables

| | |
|------------|--|
| | Flow rate (kg/s) |
| F | Process liquid |
| F_s | Steam |
| F_w | Cooling water |
| | Temperature (°C) |
| T_f | Feed |
| T_1 | Between regenerator and heater |
| T_r | Holding tube |
| T_f | Flash vessel |
| T_2 | Between regenerator and cooler |
| T_p | Product |
| T_s | Steam |
| T_{w1} | Cooling water inlet |
| T_{w2} | Cooling water outlet |
| | Thermal Load (kW) |
| Q_x | Regenerator |
| Q_s | Steam heater |
| Q_c | Cooler |
| | Heat Transfer Area (m²) |
| A_x | Regenerator |
| A_c | Cooler |
| | Residence Time (min) |
| τ | Total residence time |
| τ_r | Holding tube |
| τ_x | Regenerator |
| τ_f | Flash vessel |
| τ_c | Cooler |
| | Other |
| V_r | Holding tube (m ³) |
| V_f | Flash vessel (m ³) |
| | Population of microorganisms (-) |
| N/N_{oj} | Survivors to initial population ratio of microorganism j , $j = 1, 2, \dots, N_{mo}$ |
| | Concentration of food components (-) |
| C/C_{oi} | Final to initial concentration of food component i , $i = 1, 2, \dots, N_{fc}$ |

Table 8.33 Process Data

| | |
|-----------------------------------|--|
| Liquid food | |
| C_{pm} | Specific heat of liquid food |
| ρ_m | Density |
| Heating Steam | |
| ΔH_s | Latent heat of steam condensation |
| Cooling Water | |
| C_{pw} | Specific heat of cooling water |
| Heat transfer coefficients | |
| b | Distance available for flow between two plates of a plate heat exchanger |
| f_v | Vapor volume fraction in flash vessel |
| U_x | Regenerator |
| U_c | Cooler |

Table 8.34 Process Model: Constraints**Food quality**

$$\frac{N_j}{N_{oj}} \leq 10^{-m_j} \quad \forall \quad j = 1, 2, \dots, N_{mo} \quad (\text{C01})$$

$$\frac{C_{oi} - C_i}{C_{oi}} \leq l_i \quad \forall \quad i = 1, 2, \dots, N_{fc} \quad (\text{C02})$$

where

- m_j = Minimum accepted reduction exponent of the j microorganism
 l_i = Maximum accepted damage for the i food component

Heat transfer in heat exchangers

$$T_p - T_{w1} \geq \Delta T_{\min} \quad (\text{C03})$$

$$T_2 - T_{w2} \geq \Delta T_{\min} \quad (\text{C04})$$

$$T_2 - T_f \geq \Delta T_{\min} \quad (\text{C05})$$

$$T_v - T_1 \geq \Delta T_{\min} \quad (\text{C06})$$

where

ΔT_{\min} = Minimum accepted temperature difference at heat exchangers

Table 8.35 Process Specifications

| | |
|--------------------|----------------------------------|
| Liquid food | |
| F | Feed flow rate |
| T_f | Feed temperature |
| T_p | Product temperature |
| Utilities | |
| T_s | Heating steam temperature |
| T_{w1} | Cooling water inlet temperature |
| T_{w2} | Cooling water outlet temperature |

Table 8.36 Degrees-of-Freedom Analysis

| | |
|--------------------|------------------------|
| Process variables | $24 + N_{mo} + N_{fc}$ |
| Process equations | $14 + N_{mo} + N_{fc}$ |
| Degrees of freedom | 10 |
| Degrees of freedom | 10 |
| Specifications | 6 |
| Design variables | 4 |

Table 8.37 Design Variables

| | |
|----------|--|
| T_i | Temperature at the outlet of the regenerator |
| T_r | Temperature in the holding tube |
| τ_r | Residence time in the holding tube |
| τ_f | Residence time in the flash vessel |

Table 8.38 Model Solution Algorithm

| | | |
|-------|---|----------|
| (E01) | → | Q_x |
| (E02) | → | T_2 |
| (E03) | → | A_x |
| (E04) | → | Q_s |
| (E05) | → | F_s |
| (E06) | → | T_v |
| (E07) | → | Q_c |
| (E08) | → | F_w |
| (E09) | → | A_c |
| (E10) | → | V_r |
| (E11) | → | τ_x |
| (E12) | → | V_f |
| (E13) | → | τ_c |
| (E14) | → | τ |
| (E15) | → | N_j |
| (E16) | → | C_j |

Table 8.39 Cost Analysis**Equipment cost**

$$C_{eq} = C_{exc} A_x^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{ves} V_r^{n_{ves}} + C_{ves} V_f^{n_{ves}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q_w + C_s Q_s) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (F04)$$

Table 8.40 Cost Data

| | |
|--------------------------------------|--|
| Utility cost | |
| C_w | Cost of cooling water (\$/kWh) |
| C_s | Cost of heating steam (\$/kWh) |
| Equipment unit cost | |
| C_{ves} | Holding tube, flash vessel (k\$/m ³) |
| C_{exc} | Heat exchanger (k\$/m ²) |
| Equipment size scaling factor | |
| n_{ves} | Holding tube, flash vessel (-) |
| n_{exc} | Heat exchanger (-) |
| Other | |
| t_y | Annual operating time (hr/yr) |
| i_r | Interest rate (-) |
| l_f | Lifetime (yr) |

3. Application to Milk

Design a continuous flow thermal processing system (Figure 8.17b) for milk sterilization using injection steam heating, according to the specifications presented in Table 8.28. The necessary technical and economic data are summarized in Table 8.29.

The optimum values of the design variables and the corresponding values of the economic magnitudes are presented in Table 8.41.

The corresponding optimal enthalpy-temperature graph for all the process equipment is shown in Figure 8.23.

Figure 8.24 shows explicitly the temperature of the milk versus the processing time. The corresponding surviving fraction of the microorganisms versus the processing time is presented in Figure 8.25, while the surviving fraction of the food components is shown in Figure 8.26.

Finally, the required equipment size and utility flows as well as the corresponding cost are summarized in Figure 8.27.

In this example, no cooler with cold water is needed, since the sterilized product exits the regenerator at the specified temperature of 25°C. In this case cooling is accomplished by the cold feed (2°C).

Table 8.41 Optimum Values of the Design Variables

Design variables

| | | |
|----------|------|-----|
| T_l | 115 | °C |
| T_r | 135 | °C |
| τ_r | 0.16 | min |
| τ_f | 0.16 | min |

Cost estimation

| | | |
|----------|----|--------|
| C_{eq} | 98 | k\$ |
| C_{op} | 28 | k\$/yr |
| TAC | 43 | k\$/yr |

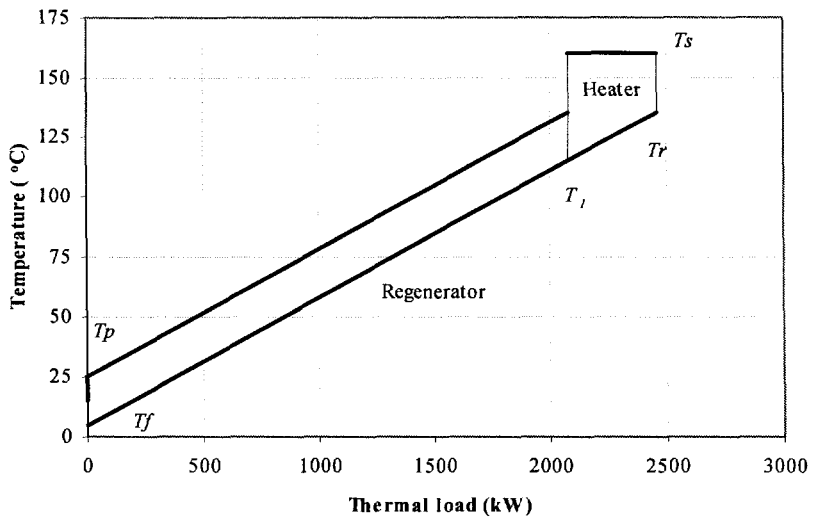


Figure 8.23 Enthalpy-temperature diagram for sterilization using indirect steam heating.

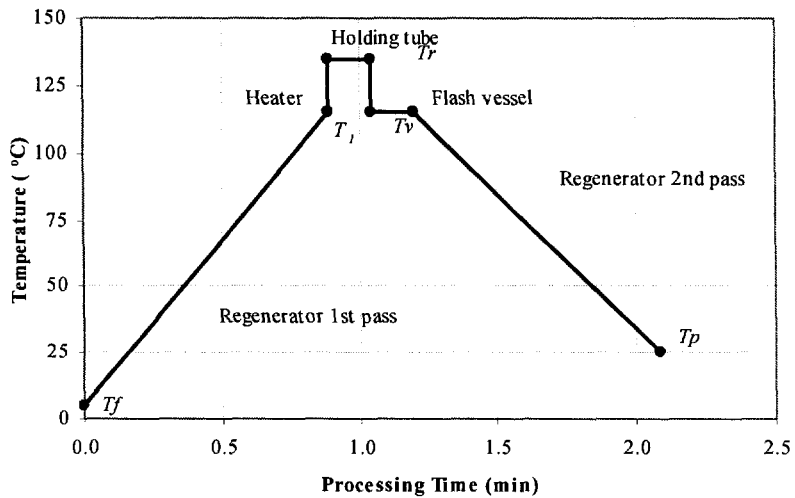


Figure 8.24 Milk temperature versus processing time.

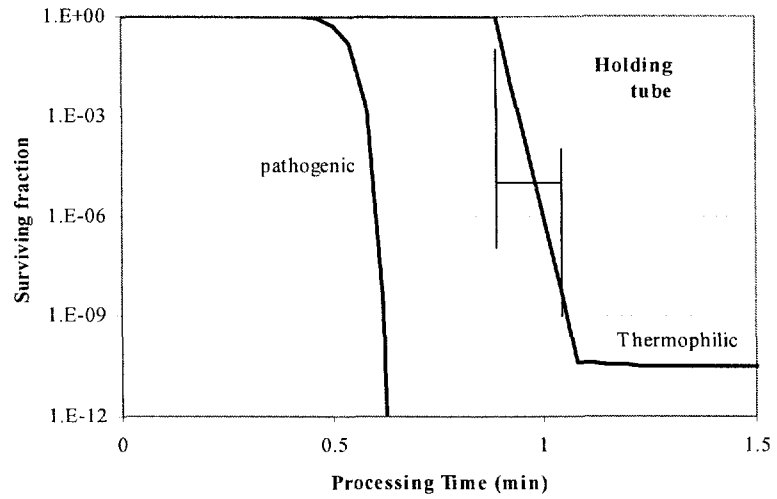


Figure 8.25 Inactivation of microorganisms versus processing time.

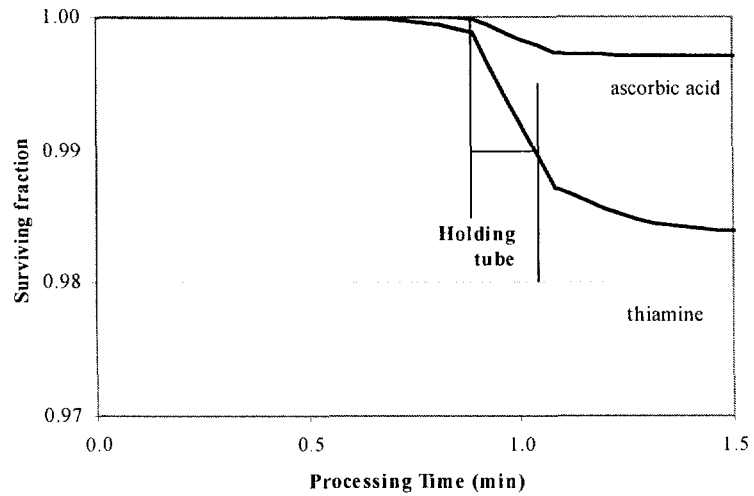


Figure 8.26 Destruction of food components versus processing time.

| Equipment | Capacity | Load | Installation cost |
|--------------|---------------------|---------|-------------------|
| Regenerator | 52.0 m ² | 2079 kW | 96.8 k\$ |
| Heater | 4.4 m ² | 378 kW | 0.0 k\$ |
| Cooler | 0.0 m ² | 0 kW | 0.0 k\$ |
| Holding tube | 0.05 m ³ | 0 kW | 0.3 k\$ |
| Flash vessel | 0.09 m ³ | 0 kW | 0.6 k\$ |
| | | | <u>97.7 k\$</u> |

| Equipment | Utility | Utility flow | Operating cost |
|--------------|------------|--------------|--------------------|
| Regenerator | - | | |
| Heater | Hot water | 0.54 tn/hr | 28.4 k\$/y1 |
| Cooler | Cold water | 0.00 tn/hr | 0.0 k\$/y1 |
| Holding tube | - | | |
| Flash vessel | - | | |
| | | | <u>28.4 k\$/y1</u> |

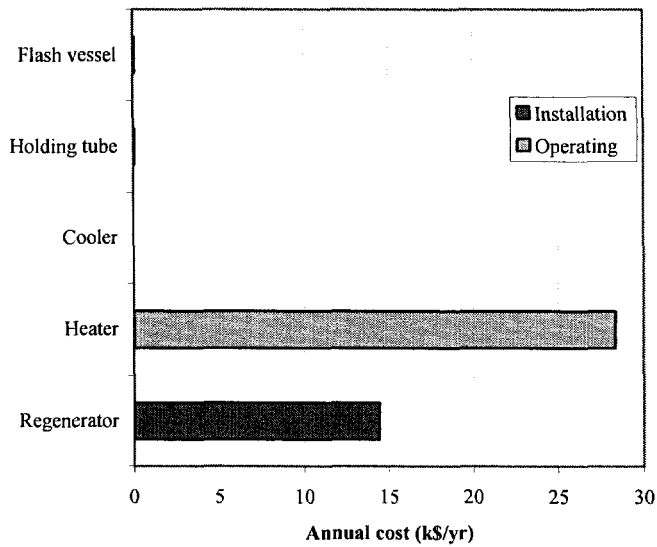


Figure 8.27 Equipment and operating cost.

A comparison between pasteurization and sterilization is presented in Table 8.42 and Figures 8.28, 8.29 and 8.30.

Table 8.42 compares the economics between the various methods of thermal processing of milk. The cost of pasteurization is higher because of the required heat pump (refrigeration).

Figure 8.28 compares the processing temperature versus time. The sharper changes in injection heating result in more effective microorganism inactivation (Figure 8.29) and less food components destruction (Figure 8.30).

Table 8.42 Comparison Between Various Methods of Thermal Processing

| | Pasteurization | Sterilization indirect | Sterilization injection | |
|--------------|----------------|------------------------|-------------------------|--------|
| Installation | 215 | 110 | 98 | k\$ |
| Operating | 29 | 28 | 28 | k\$/yr |
| Total Annual | 61 | 45 | 43 | k\$/yr |
| | 2.26 | 1.66 | 1.59 | \$/tn |

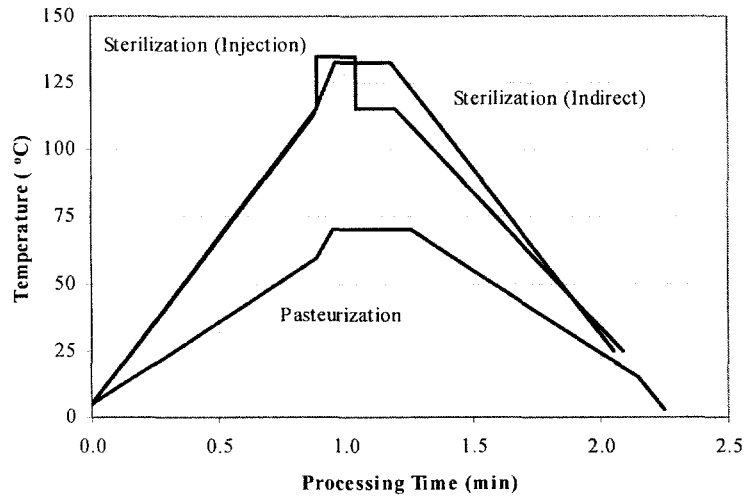


Figure 8.28 Milk temperature versus processing time.

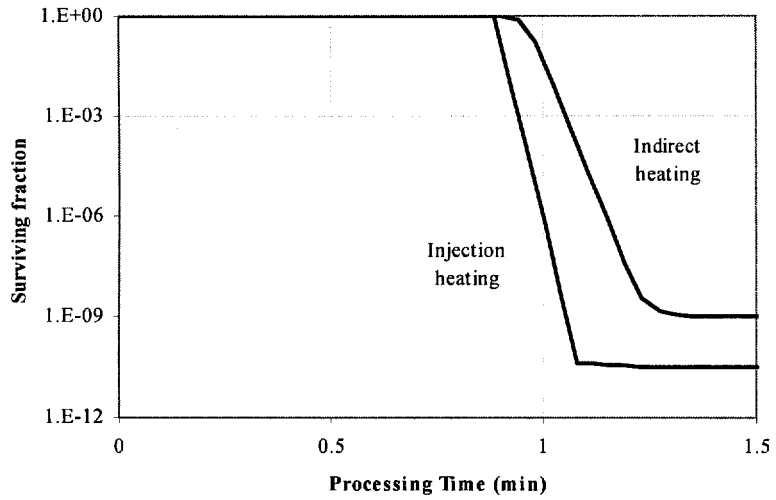


Figure 8.29 Inactivation of thermophilic microorganisms versus processing time.

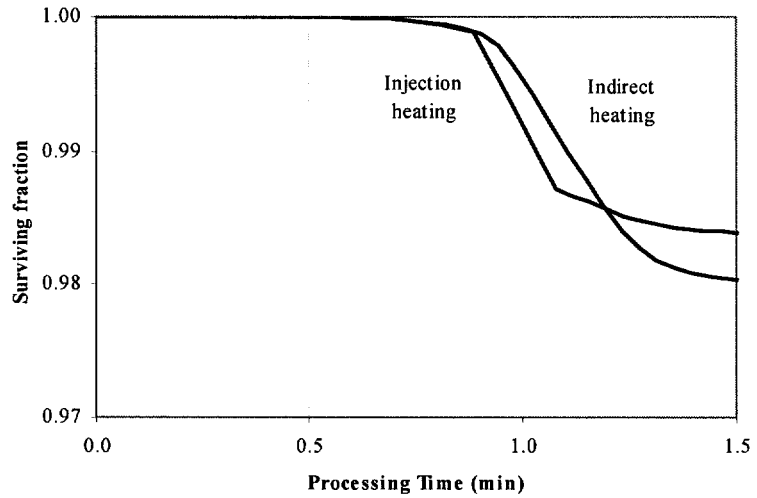


Figure 8.30 Destruction of thiamine versus processing time.

VI. CONTINUOUS FLOW STERILIZATION OF VISCOUS AND PARTICULATE FLUID FOODS

1. Process Description

The essential requirement in the continuous flow sterilization process design of viscous and particulate fluid foods is to calculate the temperature history at the center of the largest particle, and estimate the resulting inactivation of microorganisms.

Because of the high viscosity of the fluid food, scraped surface heat exchangers (SSHE) are preferred. Furthermore, inlet-outlet heat regeneration is not practically possible in this system.

Thus, the flowsheet of a continuous flow sterilization process of viscous and particulate fluid food is shown in Figure 8.31. The fluid food flows initially through a steam heater, then it is kept in a holding tube for the required time and thereafter it flows through a cold water cooler (Skjoldebrand and Ohlsson, 1993). The product is assumed to be well mixed and flow at a uniform (average) velocity in the heat exchangers and the holding tube.

2. Process Model

The mathematical model of the process presented in Figure 8.31 is summarized in Table 8.43. The variables of the model are selected in Table 8.44, while the appropriate technical data are presented in Table 8.45.

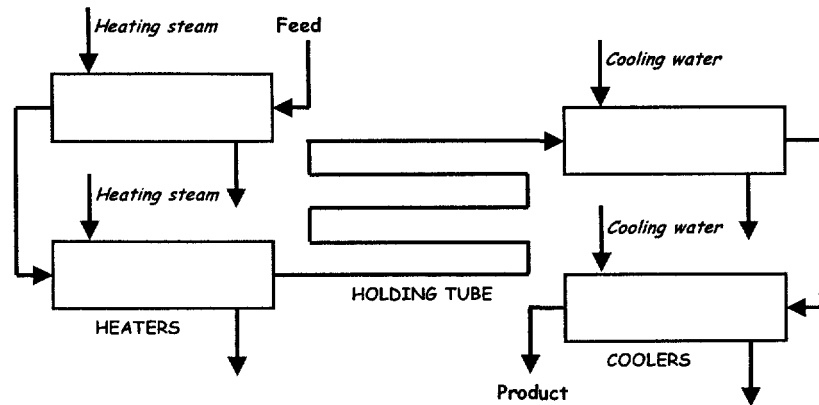


Figure 8.31 Flow sheet for continuous flow thermal processing of viscous and particulate fluid foods.

Equations (E01), (E02), and (E03) in Table 8.43 describe a simple mathematical model for the scraped surface heat exchanger. Equation (E01) refers to the process stream, which is heated, Equation (E02) to the heating steam, and Equation (E03) determines the heat flow between the above streams.

Equations (E04) is the heat balance of the process stream in the holding tube, assuming that the heat losses to the surroundings are calculated from Equation (E05).

Equations (E06), (E07), and (E08) describe a simple mathematical model for the water cooler. Equation (E06) refers to the process stream, which is cooled to the desired temperature, Equation (E07) refers to the cooling water, and Equation (E08) determines the heat flow between the two streams.

Equations (E09) to (E14) are simple geometrical equations to calculate the transfer area and the volume of the heater, holder, and cooler, respectively.

Equations (E15) through (E18) calculate the residence time of the processed food in all process equipment, i.e., the heater (E15), the holder (E16), and the cooler (E17). Equation (E18) is the total processing time. These data compose the time-temperature history of the liquid food. Based on the time-temperature history of the liquid fluid, the corresponding time-temperature history of the center of the largest particle is calculated by numerical solution of the differential equation.

Equation (E19) calculates the ratio of survivors to initial population of the microorganisms of the center of the largest particle. The integration can be calculated numerically, using the time-temperature function obtained from the aforementioned equations. The particle/fluid heat transfer coefficient (h) is calculated from the empirical equation for flow past single spheres of Table 4.5, assuming $Nu=2$ (boundary condition). Food quality constraints are expressed by the inequality (E20) in Table 8.43.

Process specifications of a typical design problem are presented in Table 8.46, while a degrees-of-freedom analysis is presented in Table 8.47. The proposed mathematical model consists of 19 equations, involving 28 variables. Consequently, the degrees of freedom of the system are 9. Typical process specifications spend 7 degrees of freedom and consequently the remaining design variables are 2. A design variables selection is presented in Table 8.48, while a solution algorithm for the specific problem is summarized in Table 8.49.

The total annualized cost of the process TAC is selected as the objective function to be optimized. It is calculated in Table 8.50. Equation (F01) calculates the equipment cost C_{eq} , and (F02) estimates the annual operating cost C_{op} , while Equation (F03) defines the total annual cost TAC , which is a weighted sum between the equipment and operating cost. The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (F04). The economic data needed for the cost estimation are summarized in Table 8.51.

In the selection of design variables and their effect on the objective function, the following points should be noted:

- T_r and τ_r are the principal design variables in sterilization processes. They must satisfy the quality constraints (Equation E20 in Table 8.43) and optimize the objective function of the process (Equation F03 in Table 8.50).

Table 8.43 Process Model: Equations**Heater**

$$Q_s = F C_{pm} (T_r - T_f) \quad (\text{E01})$$

$$Q_s = F_s \Delta H_s \quad (\text{E02})$$

$$Q_s = U_s A_s [(T_s - T_f) - (T_s - T_r)] / \ln[(T_s - T_f) / (T_s - T_r)] \quad (\text{E03})$$

Holder

$$Q_r = F C_{pm} (T_r - T_r') \quad (\text{E04})$$

$$Q_r = U_r A_r (T_r - T_a) \quad (\text{E05})$$

Cooler

$$Q_c = F C_{pm} (T_r' - T_p) \quad (\text{E06})$$

$$Q_c = F_w C_{pw} (T_{w2} - T_{w1}) \quad (\text{E07})$$

$$Q_c = U_c A_c [(T_p - T_{w1}) - (T_r' - T_{w2})] / \ln[(T_p - T_{w1}) / (T_r' - T_{w2})] \quad (\text{E08})$$

Geometry

$$A_s = \pi D_s L_s \quad (\text{E09})$$

$$A_r = \pi D_r L_r \quad (\text{E10})$$

$$A_c = \pi D_c L_c \quad (\text{E11})$$

$$V_s = \pi D_s^2 L_s / 4 \quad (\text{E12})$$

$$V_r = \pi D_r^2 L_r / 4 \quad (\text{E13})$$

$$V_c = \pi D_c^2 L_c / 4 \quad (\text{E14})$$

Residence Time

$$\tau_s = \rho_m V_s / F \quad (\text{E15})$$

$$\tau_r = \rho_m V_r / F \quad (\text{E16})$$

$$\tau_c = \rho_m V_c / F \quad (\text{E17})$$

$$\tau = \tau_s + \tau_r + \tau_c \quad (\text{E18})$$

Inactivation of Microorganisms at the center of the largest particle

$$\log\left(\frac{N}{N_o}\right) = -\int_0^{\tau} \frac{dt}{D} \quad \text{where} \quad D = D_R 10^{-\frac{T_R - T_{pc}(t)}{z}} \quad (\text{E19})$$

and $T_{pc}(t)$ is calculated from the differential equation:

$$T_{pc}(t) = T_p(t, x = 0)$$

where the Fourier Equation for heat conduction into the spherical particle

$$\frac{\partial T_p(t, x)}{\partial t} = \alpha \frac{\partial^2 T_p(t, x)}{\partial x^2}, \quad \alpha = \frac{\lambda_p}{\rho_p C_p}$$

is solved numerically with the following boundary conditions

$$\frac{\partial T_p(t, x = d/2)}{\partial x} = hA(T_p(t, x) - T(t)), \quad Nu = \frac{hd}{\lambda_p} = 2$$

$$\frac{\partial T_p(t, x = 0)}{\partial x} = 0$$

and the following initial condition

$$T_p(t = 0, x) = T_f$$

Quality Constraint

$$\frac{N}{N_o} \leq 10^{-m} \quad (\text{E20})$$

Table 8.44 Process Variables

| | |
|----------|---|
| | Flow rate (kg/s) |
| F | Process liquid |
| F_s | Heating steam |
| F_w | Cooling water |
| d | Largest particle diameter (design size) |
| | Temperature (°C) |
| T_f | Feed |
| T_r | Holding tube inlet |
| T_r' | Holding tube outlet |
| T_p | Product |
| T_s | Steam |
| T_{w1} | Cooling water inlet |
| T_{w2} | Cooling water outlet |
| | Thermal Load (kW) |
| Q_x | Holding tube losses |
| Q_s | Steam heater |
| Q_c | Cooler |
| | Equipment Length (m) |
| L_r | Holding tube |
| L_s | Steam heater |
| L_c | Cooler |
| | Heat Transfer Area (m²) |
| A_r | Holding tube |
| A_s | Steam heater |
| A_c | Cooler |
| | Equipment Volume (m³) |
| V_r | Holding tube |
| V_s | Steam heater |
| V_c | Cooler |
| | Residence Time (min) |
| τ | Total residence time |
| τ_r | Holding tube |
| τ_s | Steam heater |
| τ_c | Cooler |
| | Population of microorganisms (-) |
| N/N_o | Survivors to initial population ratio of microorganisms |

Table 8.45 Process Data

Liquid food

| | |
|-------------|-------------------------------|
| C_{pm} | Specific heat of liquid food |
| ρ_m | Density |
| ρ_p | Particle density |
| C_{pp} | Particle specific heat |
| λ_p | Particle thermal conductivity |

Heating Steam

| | |
|--------------|-----------------------------------|
| ΔH_s | Latent heat of steam condensation |
|--------------|-----------------------------------|

Cooling Water

| | |
|----------|--------------------------------|
| C_{pw} | Specific heat of cooling water |
|----------|--------------------------------|

Geometrical

| | |
|-------|-----------------------------|
| D_r | Holding tube inner diameter |
| D_s | Steam Heater inner diameter |
| D_c | Cooler inner diameter |

Heat transfer coefficients

| | |
|-------|-------------|
| U_s | Heater |
| U_r | Holder loss |
| U_c | Cooler |

Table 8.46 Process Specifications

Liquid food

| | |
|-------|---|
| F | Feed flow rate |
| T_f | Feed temperature |
| T_p | Product temperature |
| d | Largest particle diameter (Design size) |

Utilities

| | |
|----------|----------------------------------|
| T_s | Heating steam temperature |
| T_{w1} | Cooling water inlet temperature |
| T_{w2} | Cooling water outlet temperature |

Table 8.47 Degrees-of-Freedom Analysis

| | |
|--------------------|----|
| Process variables | 28 |
| Process equations | 19 |
| Degrees of freedom | 9 |
| Degrees of freedom | 9 |
| Specifications | 7 |
| Design variables | 2 |

Table 8.48 Design Variables

| | |
|----------|------------------------------------|
| T_r | Temperature in the holding tube |
| τ_r | Residence time in the holding tube |

Table 8.49 Model Solution Algorithm

| | |
|-----------------------------------|---|
| Process Specifications | $F, T_f, T_p, T_s, T_w1, T_w2$ (See Table 8.46) |
| Design Variable | T_r, τ_r (See Table 8.48) |
| Equations (See Table 8.43) | |
| (E01) | $\rightarrow Q_s$ |
| (E02) | $\rightarrow F_s$ |
| (E03) | $\rightarrow A_s$ |
| (E05) | $\rightarrow Q_r$ |
| (E04) | $\rightarrow T_r'$ |
| (E06) | $\rightarrow Q_c$ |
| (E07) | $\rightarrow F_w$ |
| (E08) | $\rightarrow A_c$ |
| (E09) | $\rightarrow L_s$ |
| (E10) | $\rightarrow L_r$ |
| (E11) | $\rightarrow L_c$ |
| (E12) | $\rightarrow V_s$ |
| (E13) | $\rightarrow V_r$ |
| (E14) | $\rightarrow V_c$ |
| (E15) | $\rightarrow \tau_s$ |
| (E16) | $\rightarrow \tau_r$ |
| (E17) | $\rightarrow \tau_c$ |
| (E18) | $\rightarrow \tau$ |
| (E19) | $\rightarrow N/N_o$ |

Table 8.50 Cost Analysis**Equipment cost**

$$C_{eq} = C_{exc} A_s^{n_{exc}} + C_{exc} A_c^{n_{exc}} + C_{ves} V_r^{n_{ves}} \quad (\text{F01})$$

Annual operating cost

$$C_{op} = (C_w Q_w + C_s Q_s) t_y \quad (\text{F02})$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (\text{F03})$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (\text{F04})$$

Table 8.51 Cost Data**Utility cost**

C_w Cost of cooling water (\$/kWh)

C_s Cost of heating steam (\$/kWh)

Equipment unit cost

C_{ves} Holding tube (k\$/m³)

C_{exc} Heat exchanger (k\$/m²)

Equipment size scaling factor

n_{ves} Holding tube (-)

n_{exc} Heat exchanger (-)

Other

t_y Annual operating time (hr/yr)

i_r Interest rate (-)

l_f Lifetime (yr)

3. Application to Model Potato Soup

Design a continuous flow thermal processing system (Figure 8.31) for a potato soup sterilization, according to the specifications presented in Table 8.52. The required technical and economic data are summarized in Table 8.53.

The process design results are summarized in Table 8.54 and in Figures 8.32 and 8.33. Figure 8.32 represents the processing time-temperature history for the fluid, the largest particle surface and center. This figure refers to all equipment: heater, holder and cooler. The surrounding temperature is also indicated (heating steam, ambient, cooling water). Based on this scheme, the microorganism inactivation for the bulk fluid and the largest particle center are calculated and shown in Figure 8.33. The holding time was selected to make the quality constraints to be valid (surviving fraction $< 10^{-9}$). The microorganism inactivation in the fluid has been completed in the heater, while the inactivation in the largest particle center is continued over in the cooler.

The effect of the treatment on the particles of different size is represented in Figures 8.34 and 8.35. Obviously, smaller particles are sterilized adequately, but particles larger than 15 mm are not (see Figure 8.35 for 20 mm particles).

Finally, Figures 8.36 and 8.37 indicate what happens when the design sterilization temperature is not obtained, let's say $T_r=125^\circ\text{C}$ instead of 135°C . Obviously the results are poor (when the holding time remains the same). For example, the surviving fraction in the particle center is worse than 10^{-3} .

It must be noted that the design calculations of this section and example are based on particles of spherical shape. When other shapes (e.g., cubes) are considered, the heat penetration into the particle center will be different, due to the edge effects. As a consequence the thermal processing time will be different. In such cases, the appropriate mathematical model of conduction heating of the particle center should be used.

Table 8.52 Process Specifications for Potato Soup Sterilization

| | | |
|--------------------|--------------|---|
| Liquid food | | |
| F | = 0.275 kg/s | Feed flow rate |
| T_f | = 25 °C | Feed temperature |
| T_p | = 25 °C | Product temperature |
| d | = 0.015 m | Largest particle diameter (Design size) |
| Utilities | | |
| T_s | = 160 °C | Steam temperature |
| T_{w1} | = 15 °C | Cold water inlet temperature |
| T_{w2} | = 35 °C | Cold water outlet temperature |
| Quality | | |
| m | = 9 | Commercial inactivation of microorganisms |

Table 8.53 Technical and Economic Data

| | | |
|--------------------------------------|----------------------------|-------------------------------|
| Liquid food | | |
| C_{pm} | = 3.78 kJ/kg K | Specific heat |
| ρ_m | = 1025 kg/m ³ | Fluid density |
| ρ_p | = 1075 kg/m ³ | Particle density |
| C_{pw} | = 3.60 kJ/kg K | Particle specific heat |
| λ_p | = 0.55 W/m | Particle thermal conductivity |
| Heating steam | | |
| ΔH_s | = 2000 kJ/kg | Latent heat |
| Cooling water | | |
| C_{pw} | = 4.18 kJ/kg K | Specific heat |
| Geometrical | | |
| D_r | = 0.05 m | Holding tube inner diameter |
| D_s | = 0.10 m | Steam heater inner diameter |
| D_c | = 0.10 m | Cooler inner diameter |
| Heat transfer coefficients | | |
| U_s | = 1.25 kW/m ² K | Heater |
| U_r | = 0.01 kW/m ² K | Holder loss |
| U_c | = 0.75 kW/m ² K | Cooler |
| h | = 67 W/m ² K | Particle/fluid |
| Utility cost | | |
| C_s | = 0.05 \$/kWh | Steam |
| C_w | = 0.02 \$/kWh | Cold water |
| Equipment unit cost | | |
| C_{ves} | = 3.00 k\$/m ³ | Holding tube or flash vessel |
| C_{exc} | = 5.00 k\$/m ³ | Heat exchanger |
| Equipment size scaling factor | | |
| n_{ves} | = 0.70 | Holding tube or flash vessel |
| n_{exc} | = 0.75 | Heat exchanger |
| Other | | |
| t_y | = 1500 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 8.54 Process Design Results for Potato Soup Sterilization**Design variables**

| | |
|----------------------|----------|
| Holder temperature | 135 °C |
| Holding time | 1.5 min |
| Total residence time | 12.5 min |

Main Process Design Results

| | <u>Heater</u> | <u>Holding</u> | <u>Cooler</u> |
|----------------------|---------------|----------------|---------------|
| Diameter (m) | 0.10 | 0.05 | 0.10 |
| Length (m) | 5.1 | 12.4 | 17.7 |
| Cost (k\$) | 7.09 | 0.22 | 18.1 |
| Velocity (m/s) | 0.034 | 0.138 | 0.034 |
| Residence time (min) | 2.4 | 1.5 | 8.6 |
| Utility | steam | - | water |
| Thermal load (kW) | 133 | - | 131 |
| Cost (k\$/yr) | 9.96 | - | 3.92 |

Cost summary

| | |
|-----------------------|------------------|
| Installation Cost | 25 k\$ |
| Operating cost | 14 k\$/yr |
| Total annualized cost | <u>18 k\$/yr</u> |

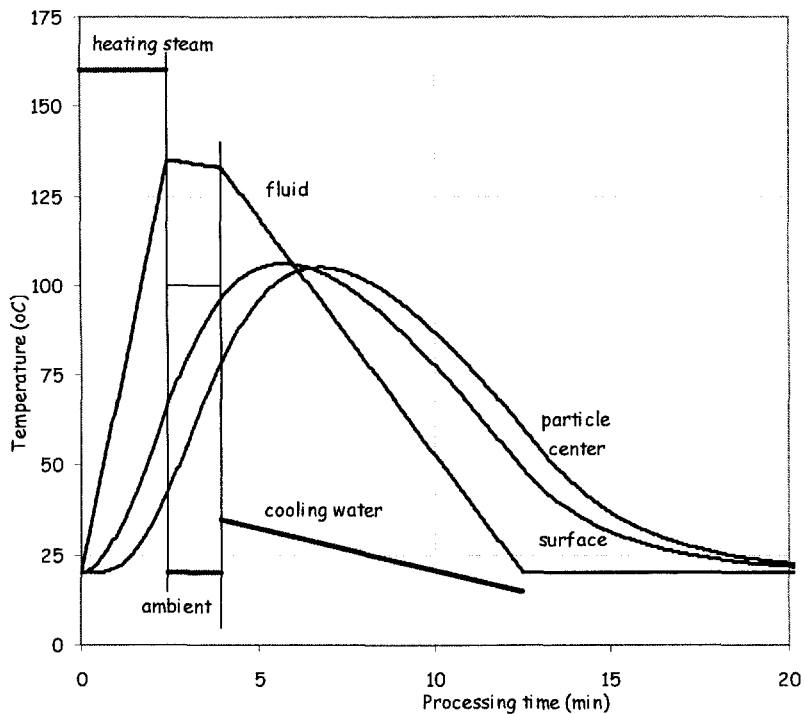


Figure 8.32 Temperature versus processing time (spherical particles, 15 mm).

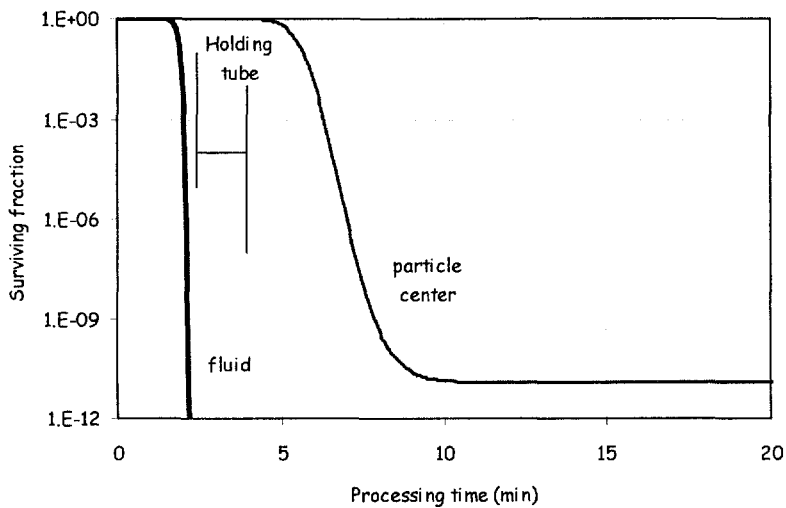


Figure 8.33 Inactivation of microorganisms versus processing time (spherical particles, 15 mm).

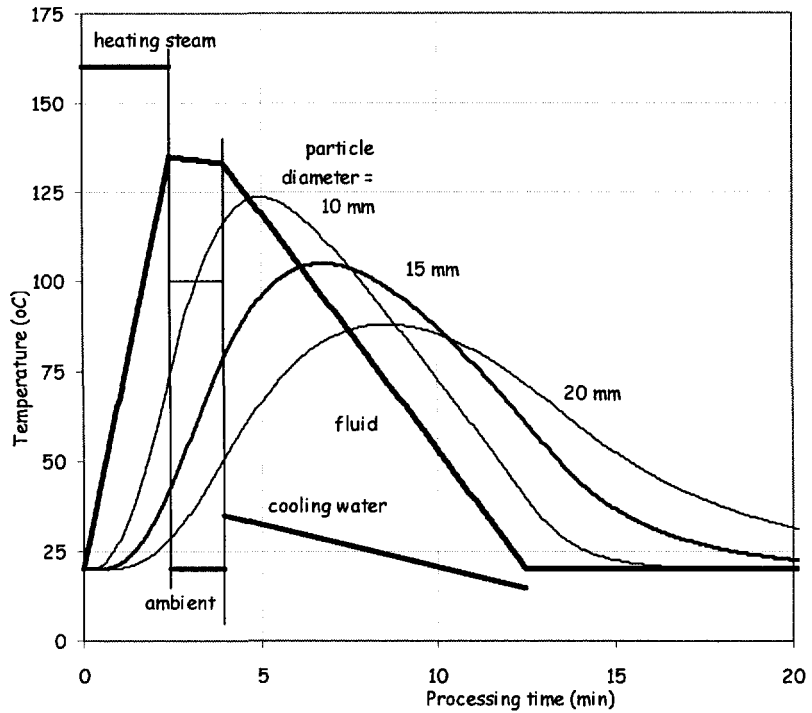


Figure 8.34 Temperature versus processing time (center of spherical particles).

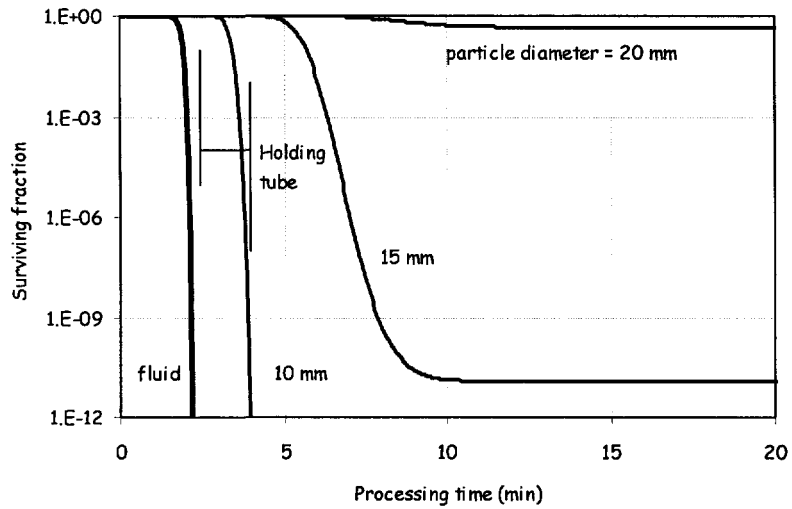


Figure 8.35 Inactivation of microorganisms versus processing time (center of spherical particles).

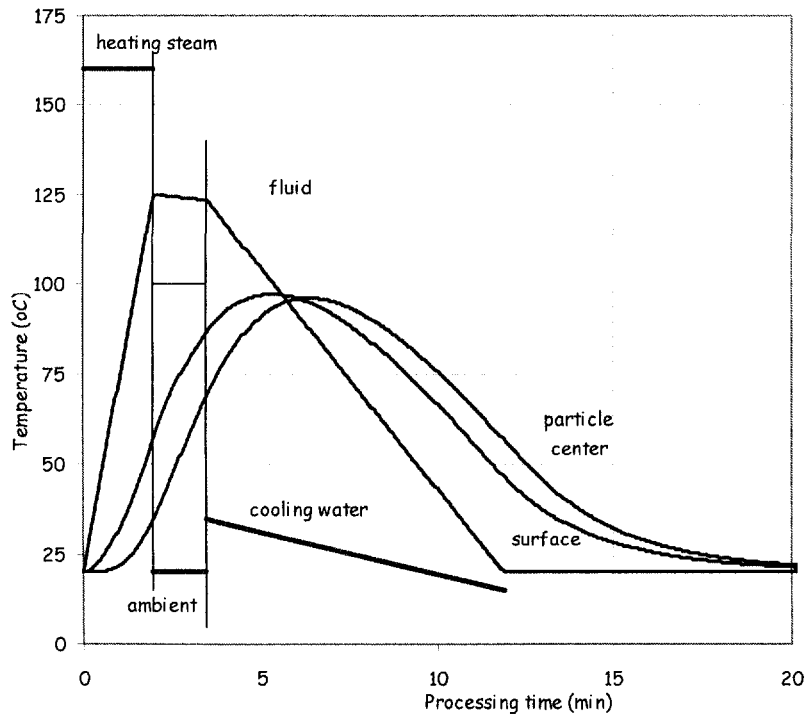


Figure 8.36 Temperature versus processing time (spherical particles, 15 mm). Maximum fluid (holder) temperature 125°C.

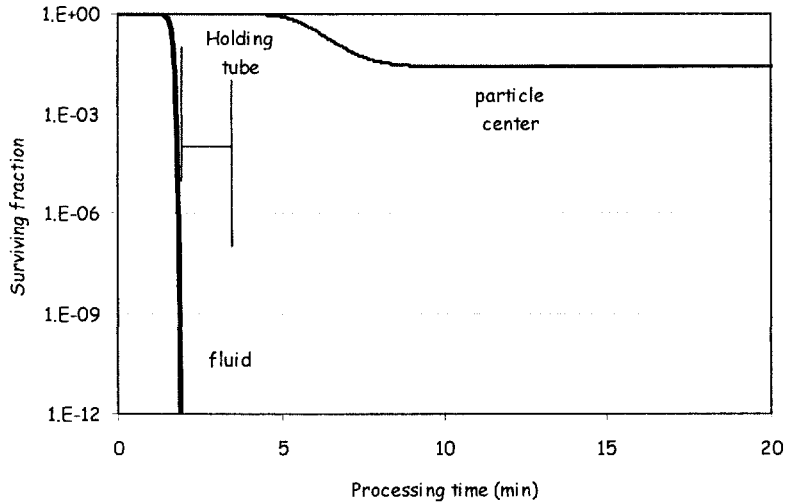


Figure 8.37 Inactivation of microorganisms versus processing time (spherical particles, 15 mm). Maximum fluid (holder) temperature 125°C.

VII. IN-CONTAINER THERMAL PROCESSING

The design of in-container sterilization process is discussed here in connection with a continuous rotary cooker/cooler, which is used in large canning operations. The calculation is based on the lethality at the lowest heating point (center) of the container. Volume-average sterilization values for microorganisms and food quality can be obtained by the integrated Equations (1-11) and (1-13) of Chapter 1.

1. Process Description

The rotary cooker/coolers consist of two horizontal pressure shells (vessels) equipped with a rotating spiral reel mechanism, in which the cans move progressively through the prescribed cycles of heating, holding and cooling. Special pressure feeders (pressure locks) transport the cans into the pressure cookers, and from the cookers to the atmospheric water coolers (Figure 8.38). The rotary system improves significantly the heat transfer to the canned product and reduces the processing time. Steam is applied at the bottom of the cooker, which is equipped with the proper venting, bleeding, and temperature recording/controlling accessories. The rotary cooler uses cold water (2/3 full), in counter flow with the cans. The cans enter and leave the pressure cooker through pressure locks. The pressure cooker is heated with live saturated steam, and cooling is achieved with cold water in the cooler, which operates normally at atmospheric pressure.

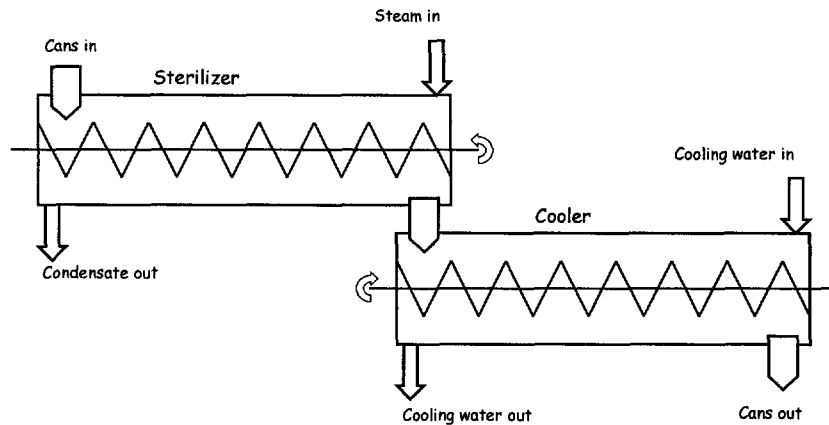


Figure 8.38 Flow sheet for rotary cooker/cooler in-container sterilizer.

2. Process Model

The mathematical model of the process presented in Figure 8.38 is summarized in Table 8.55. The variables of the model are selected in Table 8.56, while the appropriate technical data are presented in Table 8.57. It is assumed that the sterilizer operates at constant temperature (no “come-up” time), heating and cooling are by conduction with equal parameters f , and there is no heating or cooling lag ($j=1$).

Equation (E01) calculates the heating and cooling parameter for the can versus the food properties and the can dimensions. Equations (E02) and (E03) calculate the temperature at the center of the can for the sterilizer and the cooler, respectively, versus the corresponding residence times. Equations (E04) and (E05) estimate the required capacities for the sterilizer and the cooler, respectively. Equation (E06) calculates the mass of the food in a can. Equation (E07) calculates the heating load in the sterilizer, which is equal to the cooling load in the cooler. Equation (E08) calculates the ratio of survivors to initial population of the microorganisms in the center of the can. Food quality constraints are expressed by Equation (E09).

Process specifications of a typical design problem are presented in Table 8.58, while a degrees-of-freedom analysis is presented in Table 8.59. The proposed mathematical model consists of 9 equations, involving 15 variables. Consequently, the degrees of freedom of the system are 6. Typical process specifications spend 6 degrees of freedom and consequently no design variable is available for this design problem. A solution algorithm for the specific problem is summarized in Table 8.60.

The total annualized cost of the process TAC is selected as the objective function. It is calculated in Table 8.61. Equation (F01) calculates the equipment cost C_{eq} , (F02) the annual operating cost C_{op} , while Equation (F03) defines the total annual cost TAC , which is a weighted sum between the equipment and operating cost. The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (F04). The economic data needed for the cost estimation are summarized in Table 8.62.

3. Application to Corn Canning

Design a continuous flow cooker/cooler system (Figure 8.38) for corn canning, according to the specifications presented in Table 8.63. The necessary technical and economic data are summarized in Table 8.64.

The process design results are summarized in Table 8.65 and in Figures 8.39 and 8.40. Figure 8.39 represents the processing time-temperature history for the center of food in the can. The figure refers to both equipment heater and cooler. The surrounding temperature is also indicated (heating steam, cooling water). Based on this scheme the microorganisms inactivation for the center of the can is calculated and shown in Figure 8.40.

In-container sterilization results in larger thermal destruction of heat-sensitive food components, such as thiamine and ascorbic acid, than continuous flow processing (Figure 8.41). This is caused by the longer exposure of the food product to the high processing temperatures.

Table 8.55 Process Model: Equations

$$f = \frac{0.398}{\left(\frac{1}{r^2} + \frac{0.427}{L^2}\right)\left(\frac{\lambda}{\rho C_p}\right)} \quad (\text{E01})$$

$$\frac{T_m - T_s}{T_a - T_s} = \exp\left(-\frac{t_s}{f}\right) \quad (\text{E02})$$

$$\frac{T_a - T_w}{T_m - T_w} = \exp\left(-\frac{t_c}{f}\right) \quad (\text{E03})$$

$$N_s = F t_s \quad (\text{E04})$$

$$N_c = F t_c \quad (\text{E05})$$

$$m = \rho (\pi r^2) (2L) \quad (\text{E06})$$

$$Q = M F C_p (T_m - T_w) \quad (\text{E07})$$

Inactivation of Microorganisms at the center of the can

$$\log\left(\frac{N}{N_o}\right) = - \int_0^{t_s+t_c} \frac{dt}{D} \quad \text{where} \quad D = D_R 10^{\frac{T_R - T_c(t)}{z}} \quad (\text{E08})$$

$$\text{and } T_c(t) = \begin{cases} T_s + (T_a - T_s) \exp\left(-\frac{t}{f}\right) & \text{when } 0 < t < t_s \\ T_w + (T_m - T_w) \exp\left(-\frac{t}{f}\right) & \text{when } t_s < t < t_s + t_c \end{cases}$$

Quality Constraint

$$\frac{N}{N_o} \leq 10^{-m} \quad (\text{E09})$$

Table 8.56 Process Variables

| | | |
|---------|---------|---|
| F | cans/hr | Feed rate |
| f | hr | Heating/cooling parameter (time constant) |
| L | m | Can half height |
| M | kg | Product mass of a can |
| N/N_0 | – | Survivors to initial population ratio of microorganisms |
| N_c | cans | Cooler capacity |
| N_s | cans | Sterilizer capacity |
| Q | kW | Heating/cooling load |
| r | m | Can radius |
| T_a | °C | Ambient temperature |
| t_c | hr | Cooler residence time |
| T_m | °C | Maximum temperature at the center of the can |
| t_s | hr | Sterilizer residence time |
| T_s | °C | Steam temperature |
| T_w | °C | Cooling water temperature |

Table 8.57 Process Data

| | | |
|-----------|-------------------|------------------------------|
| C_p | kJ/kg K | Specific heat of food |
| ρ | kg/m ³ | Density of food |
| λ | W/m K | Thermal conductivity of food |
| D | min | Decimal reduction time |
| z | °C | Thermal resistance factor |
| m | – | Reduction exponent |
| T_R | °C | Reference temperature |

Table 8.58 Process Specifications

| | | |
|-------|---------|---------------------------|
| F | cans/hr | Feed rate |
| L | m | Can half height |
| r | m | Can radius |
| T_s | °C | Steam temperature |
| T_w | °C | Cooling water temperature |
| T_a | °C | Ambient temperature |

Table 8.59 Degrees-of-Freedom Analysis

| | |
|--------------------|----|
| Process variables | 15 |
| Process equations | 9 |
| Degrees of freedom | 6 |
| Degrees of freedom | 6 |
| Specifications | 6 |
| Design variables | 0 |

Table 8.60 Model Solution Algorithm

| | | |
|-------|---|---|
| T_m | = | trial value |
| (E01) | → | f |
| (E02) | → | t_s |
| (E03) | → | t_c |
| (E05) | → | N_s |
| (E04) | → | N_c |
| (E06) | → | M |
| (E07) | → | Q |
| (E08) | → | N/N_o |
| (E09) | → | If not valid, suggest a new value for T_m |

Table 8.61 Cost Analysis**Equipment cost**

$$C_{eq} = C_{sc} N_s^{n_{sc}} + C_{sc} A_c^{n_{sc}} \quad (F01)$$

Annual operating cost

$$C_{op} = (C_w Q + C_s Q) t_y \quad (F02)$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (F03)$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1} \quad (F04)$$

Table 8.62 Cost Data

| | |
|----------|---------------------------------------|
| C_w | Cost of cooling water (\$/kWh) |
| C_s | Cost of heating steam (\$/kWh) |
| C_{sc} | Sterilizer/cooler unit cost (k\$/can) |
| n_{sc} | Sterilizer/cooler scale factor (-) |
| t_y | Annual operating time (hr/yr) |
| i_r | Interest rate (-) |
| l_f | Lifetime (yr) |

Table 8.63 Process Specifications for Corn Canning

| | | |
|-------|----------------|---------------------------|
| F | = 1000 cans/hr | Feed rate |
| $2L$ | = 0.12 m | Can height |
| r | = 0.05 m | Can radius |
| T_s | = 120 °C | Steam temperature |
| T_w | = 15 °C | Cooling water temperature |
| T_a | = 25 °C | Ambient temperature |

Table 8.64 Technical and Economic Data

| | | |
|-----------|--------------------------|--------------------------------|
| C_p | = 3.50 kJ/kg K | Specific heat of food |
| ρ | = 1050 kg/m ³ | Density of food |
| λ | = 0.60 W/m K | Thermal conductivity of food |
| D | = 0.30 min | Decimal reduction time |
| z | = 11 °C | Thermal resistance factor |
| m | = 12 | Reduction exponent |
| T_R | = 121 °C | Reference temperature |
| C_w | = 0.01 \$/kWh | Cost of cooling water |
| C_s | = 0.04 \$/kWh | Cost of heating steam |
| C_{sc} | = 1 k\$/can | Sterilizer/cooler unit cost |
| n_{sc} | = 0.95 | Sterilizer/cooler scale factor |
| t_y | = 1500 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Table 8.65 Process Design Results

| | | |
|---------------------------|---------------------|---|
| F | = 1000 cans/hr | Feed rate |
| $2r$ | = 0.10 m | Can diameter |
| $2L$ | = 0.11 m | Can height |
| M | = 0.901 kg | Product mass of a can |
| f | = 75.1 min | Heating/cooling parameter (time constant) |
| t_s | = 87 min | Sterilizer residence time |
| t_c | = 63 min | Cooler residence time |
| Q | = 78 kW | Heating/cooling load |
| N_c | = 1000 cans | Cooler capacity |
| N_s | = 1500 cans | Sterilizer capacity |
| T_s | = 120 °C | Steam temperature |
| T_w | = 15 °C | Cooling water temperature |
| T_a | = 25 °C | Ambient temperature |
| T_m | = 114 °C | Maximum temperature at the center of the can |
| N/N_o | = 10^{-12} | Survivors to initial population ratio of microorganisms |
| Equipment cost | | |
| C_{st} | = 1041 k\$ | Sterilizer |
| C_{cl} | = <u>708</u> k\$ | Cooler |
| C_{eq} | = 1749 k\$ | Equipment cost |
| Operating cost | | |
| C_e | = 5.9 k\$/yr | Cost of electricity |
| C_w | = <u>1.2</u> k\$/yr | Cost of cooling water |
| C_{op} | = 7.1 k\$/yr | Operating cost |
| Objective function | | |
| TAC | = 268 k\$/yr | Total annualized cost |

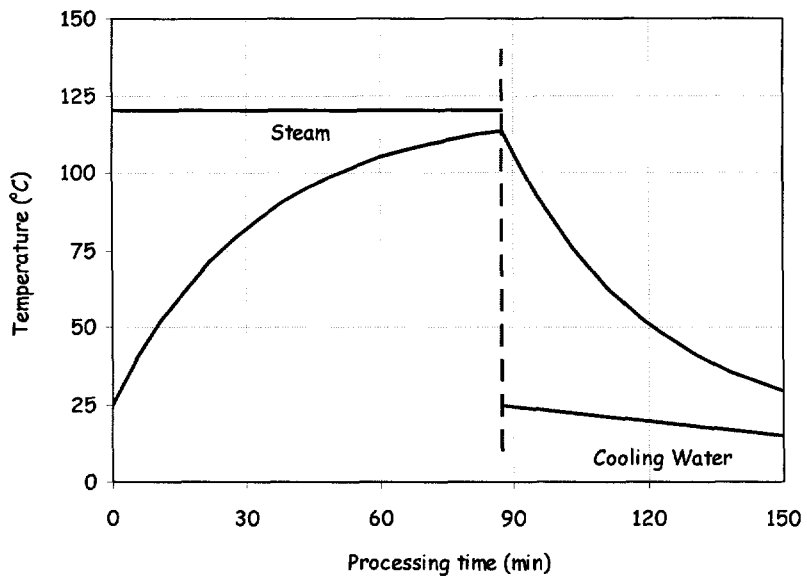


Figure 8.39 Temperature versus in-container processing time.

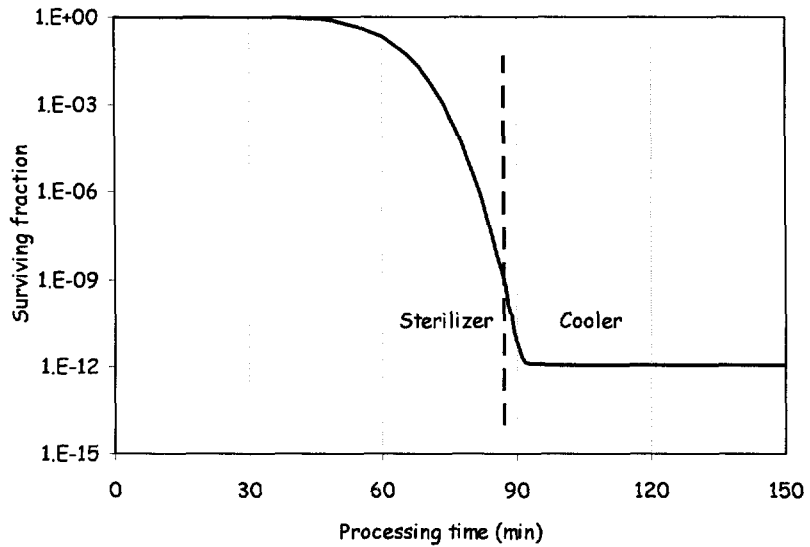


Figure 8.40 Inactivation of microorganisms versus in-container processing time.

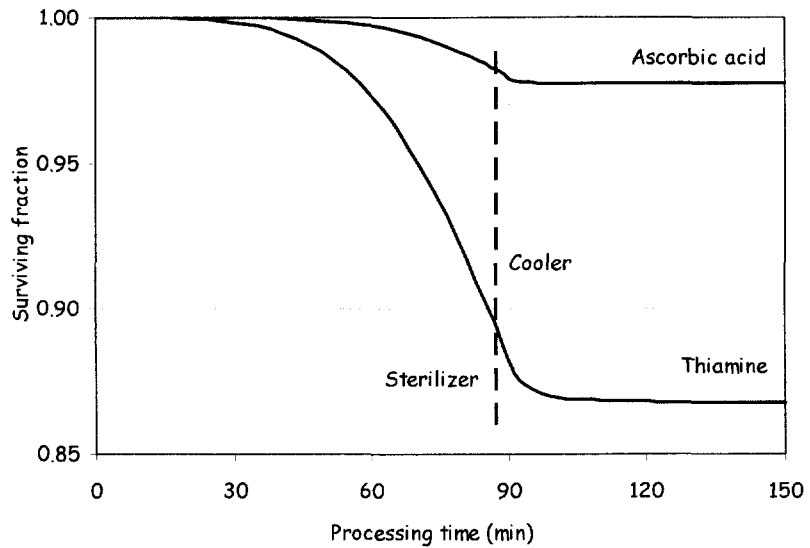


Figure 8.41 Destruction of food components versus in-container processing time.

Similar results are obtained by conventional calculation of the thermal process time t_B , using the Ball formula method assuming conduction heating with constant thermal diffusivity. For a can 10cm x 11cm containing corn, as calculated previously, $f=75$ min. Assuming $j=1$, the lethality at retort temperature $T_h=120^\circ\text{C}$ will be (Equation 8-22) $U = 3.6 \times 1.26 = 4.54$ min (since $F = 0.3 \times 12 = 3.6$ min). From literature tables for $f/U=16.5$ and $j=1$, $g=10^\circ\text{C}$ and consequently $T_{max}=110^\circ\text{C}$. Thus, thermal processing time $t_B=75$ min (Equation 8-21).

The Ball formula assumes a parabolic (T, t) curve when changing from heating to cooling, with a constant cooling lag factor $j=1.41$, which is different from the heating lag factor ($j=1$ in this example). The computer model used in this example assumes a sharp temperature change at the container with no lag ($j_h=j_c=1$). As a result, the sterilizing effect during cooling is underestimated, and the process time t_s becomes higher (by 16%) than the Ball process time t_B . However, if the cooling lag factor is taken as $j_c=1.41$, the computer model predicts a process time very close to the Ball process time.

Assuming convection heating and complete mixing with an overall heat transfer coefficient $U=100\text{W/m}^2\text{K}$ the f value is estimated to be $f=24.2$ min (Equation 8-24), i.e., about 1/3 of the f value for the conduction-heated product ($f=75$ min). Intermediate values will be obtained for lower U values (less agitation, viscous liquid/particles).

NOMENCLATURE

| | | |
|--------------|------------|---|
| A_c | m^2 | Cooler heat transfer area |
| A_c | m^2 | Refrigerant condenser heat transfer area |
| A_e | m^2 | Refrigerant evaporator heat transfer area |
| A_h | m^2 | Heater heat transfer area |
| A_r | m^2 | Holding tube heat transfer area |
| A_s | m^2 | Steam heater heat transfer area |
| A_x | m^2 | Regenerator heat transfer area |
| b | m | Distance available for flow between two plates of a plate heat exchanger |
| C_{com} | k\$/kW | Compressor unit cost |
| C_e | \$/kWh | Cost of electricity |
| C_{exc} | k\$/ m^2 | Heat exchanger unit cost |
| C_h | \$/kWh | Cost of hot water |
| C_i/C_{oi} | – | Final to initial concentration of food component i , $i=1,2, \dots, N_{fc}$ |
| C_p | kJ/kg K | Specific heat of food |
| C_{pm} | kJ/kg K | Specific heat of liquid food |
| C_{pp} | kJ/kg K | Particle specific heat |
| C_{pr} | kJ/kg K | Specific heat of refrigerant |
| C_{pw} | kJ/kg K | Specific heat of water |
| C_s | \$/kWh | Cost of heating steam |
| C_{sc} | k\$/can | Sterilizer/cooler unit cost |
| C_{ves} | k\$/ m^3 | Holding tube, flash vessel unit cost |
| C_w | \$/kWh | Cost of cooling water |
| d | m | Largest particle diameter (design size) |
| D | min | Decimal reduction time |
| D_R | min | Decimal reduction time at temperature T_R |
| d | m | Tube diameter of a shell and tube heat exchanger |
| D_c | m | Cooler inner diameter |
| ΔH_r | kJ/kg | Latent heat of refrigerant condensation |
| D_r | m | Holding tube inner diameter |
| D_s | m | Steam Heater inner diameter |
| E_r | kW | Refrigerant compressor power |
| F | cans/hr | Feed rate |
| f | hr | Heating/cooling parameter (time constant) |
| F | kg/s | Process liquid flow rate |
| F_h | kg/s | Hot water flow rate |
| F_r | kg/s | Refrigerant flow rate |
| F_s | kg/s | Heating steam flow rate |
| f_v | – | Vapor volume fraction at flash vessel |
| F_w | kg/s | Cooling water flow rate |
| i_r | – | Interest rate |
| L | m | Can half height |
| L | m | Length, tube or heat exchanger |
| L_c | m | Cooler length |
| l_f | yr | Lifetime |
| L_r | m | Holding tube length |

| | | |
|--------------|---------------------|--|
| L_s | | Steam heater length |
| m | – | Reduction exponent |
| M | kg | Product mass in a can |
| N/N_o | – | Survivors to initial population ratio of microorganisms |
| N_c | cans | Cooler capacity |
| n_{com} | – | Compressor scaling factor |
| n_{exc} | – | Heat exchanger scaling factor |
| N_j/N_{oj} | | Survivors to initial population ratio of microorganism j , $j=1,2, \dots, N_{mo}$ |
| N_s | cans | Sterilizer capacity |
| n_{sc} | – | Sterilizer/cooler scaling factor |
| n_{ves} | – | Holding tube, flash vessel scaling factor |
| Q | kW | Heating/cooling load |
| Q_c | kW | Cooler thermal load |
| Q_c | kW | Refrigerant condenser thermal load |
| Q_e | kW | Refrigerant evaporator thermal load |
| Q_h | kW | Heater thermal load |
| Q_s | kW | Steam heater thermal load |
| Q_x | kW | Holding tube thermal losses |
| Q_x | kW | Regenerator thermal load |
| r | m | Can radius |
| T_1 | °C | Temperature between regenerator and heater |
| T_2 | °C | Temperature between regenerator and cooler |
| T_a | °C | Ambient temperature |
| t_c | hr | Cooler residence time |
| T_c | °C | Refrigerant condensation temperature |
| T_e | °C | Refrigerant evaporation temperature |
| T_f | °C | Flash vessel temperature |
| T_{hl} | °C | Hot water inlet temperature |
| T_{h2} | °C | Hot water outlet temperature |
| T_m | °C | Maximum temperature at the center of the can |
| T_p | °C | Product temperature |
| T_r | °C | Holding tube inlet temperature |
| T_R | °C | Reference temperature |
| T_r' | °C | Holding tube outlet temperature |
| t_s | hr | Sterilizer residence time |
| T_s | °C | Steam temperature |
| T_w | °C | Cooling water temperature |
| T_{w1} | °C | Cooling water inlet temperature |
| T_{w2} | °C | Cooling water outlet temperature |
| t_y | hr/yr | Annual operating time |
| U | kW/m ² K | Overall heat transfer coefficient |
| U_c | kW/m ² K | Cooler overall heat transfer coefficient |
| U_c | kW/m ² K | Refrigerant condenser overall heat transfer coefficient |
| U_e | kW/m ² K | Refrigerant evaporator overall heat transfer coefficient |
| U_h | kW/m ² K | Heater overall heat transfer coefficient |
| U_r | kW/m ² K | Holder loss heat transfer coefficient |
| U_s | kW/m ² K | Heater overall heat transfer coefficient |

| | | |
|--------------|---------------------------|--|
| U_s | $\text{kW/m}^2 \text{ K}$ | Steam heater overall heat transfer coefficient |
| U_x | $\text{kW/m}^2 \text{ K}$ | Regenerator overall heat transfer coefficient |
| V | m^3 | Volume |
| V_c | m^3 | Cooler volume |
| V_r | m^3 | Holding tube volume |
| V_f | m^3 | Flash vessel volume |
| V_s | m^3 | Steam heater volume |
| z | $^\circ\text{C}$ | Thermal resistance factor |
| ΔH_s | kJ/kg | Latent heat of steam condensation |
| λ | W/mK | Thermal conductivity of food |
| λ_p | W/mK | Particle thermal conductivity |
| ρ | kg/m^3 | Density of food |
| ρ_m | kg/m^3 | Density of food |
| ρ_p | kg/m^3 | Particle density |
| τ | min | Total residence time |
| τ_c | min | Cooler residence time |
| τ_e | min | Cooler (Refrigerant evaporator) residence time |
| τ_f | min | Flash vessel residence time |
| τ_h | min | Heater residence time |
| τ_r | min | Holding tube residence time |
| τ_s | min | Steam heater residence time |
| τ_x | min | Regenerator residence time |

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9

Mass Transfer Processes

I. INTRODUCTION

Mass transfer processes are used in various physical separations of food components from liquids or solids, for recovering valuable products, or for removing undesirable components. The controlling transport mechanism is mass transfer at the molecular level (Sherwood et al., 1975; King, 1982). They do not include mechanical separations, which are based on differences in size, shape, and density of solid particles or pieces.

Mass transfer processes (operations) used in food processing are:

- **Distillation**
used in recovering volatile components from liquids
- **Solvent extraction or leaching**
used in recovering solutes from liquids or solids
- **Adsorption and ion exchange**
used for removing undesirable components from fluids
- **Crystallization**
used for separating dissolved sugars or salts from solutions

Distillation and crystallization involve heat transfer, but mass transfer is the rate-controlling process. Evaporation (Chapter 6) and drying (Chapter 7), involving both heat and mass transfer, are treated separately, since heat transfer is usually the rate-controlling mechanism.

In addition to the above mass transfer operations, there are some novel separation processes applied to food processing, notably membrane separations (e.g., ultrafiltration and reverse osmosis) and supercritical fluid (carbon dioxide) extraction. Membrane separations are treated in Chapter 10.

Mass transfer operations are based on two fundamental physical processes, i.e., phase equilibria and mass transfer. Both processes are controlled by molecular forces and they are evaluated and predicted by molecular dynamics, experimental measurements, or empirical correlations (Reid et al., 1987).

The design of distillation and extraction is described in more detail in this chapter, since they constitute the most important mass transfer operations used in food processing. The other mass transfer processes are analyzed in standard chemical engineering books, such as Perry and Green (1997). The equipment used in food mass transfer processes is described by Saravacos and Kostaropoulos (2002).

1. Phase Equilibria

Phase equilibria define the final concentrations of the components in two phases, if left long enough to reach thermodynamic equilibrium. They are essential in the design of mass transfer separations. Thus, vapor/liquid equilibria are used in the analysis of distillation, and liquid/liquid or liquid/solid equilibria are needed in solvent extraction and leaching processes.

Phase equilibria are calculated by equating the activities of a component in the two phases. In vapor/liquid equilibria, the activity of a component is equal to its partial pressure in a given phase. Empirical correlations have been developed for the estimation of activity coefficients in complex mixtures, using computer techniques.

2. Mass Transfer

Mass transfer in gases is fast, due to the molecular motion, and the diffusivity of a given molecule can be predicted fairly well by molecular dynamics and thermodynamics (Reid et al., 1987).

Mass transfer in liquid foods is slow, and it is controlled by molecular diffusion or other transport mechanisms. The diffusivity of molecular species in the liquid phase is affected by the molecular size and the intermolecular interactions of the molecules involved. (Cussler, 1997). Mass transfer in liquids can be enhanced by mechanical mixing or high flow velocity (turbulence).

The effective diffusivity of water and various solutes in solid and semi-solid foods depends on the molecular, micro-, and macro-structure of the food matrix (Aquilera and Stanley, 1999; Saravacos and Maroulis, 2001). Empirical models, based on regression analysis of experimental data, have been suggested to estimate the mass diffusivity of small molecules in various food materials.

3. Design of Equipment

Due to phase equilibrium limitations, mass transfer operations may require more than one contact stages. The number of required theoretical separation stages is calculated by graphical or analytical methods, which were developed mainly for distillation and liquid/liquid extraction, as discussed in this chapter. Similar methods are used in the design of gas absorption processes (Saravacos and Kostaropoulos, 2002).

The design of adsorption and ion exchange processes is based on empirical data and correlations (Walas, 1988; Perry and Green, 1997). Crystallization processes are more complex operations, involving phase equilibrium (solubility), mass transfer, heat transfer, and population dynamics (Mullin, 1993; Hartel, 2001).

The equipment used in food mass transfer operations is basically similar to the established chemical engineering equipment, with particular attention to hygienic (sanitary) design, heat sensitivity of the food product, and equipment corrosion (presence of water in the food).

II. DISTILLATION

1. Introduction

Distillation is the recovery of valuable components from a liquid mixture by vaporization and condensation, usually in several stages. Distillation usually combines stripping (removal) with rectifying (enrichment) of volatile components in a column, consisting of both stripping and rectifying sections.

Distillation systems, used in separating complex mixtures in the chemical and petrochemical industries, consist of a number of continuous distillation columns, arranged in complex configurations. The food processing industry uses limited distillation processes of medium to small size, some of them batch-operated.

The most common applications of distillation in food and agricultural industry are the ethanol recovery from fermentation solutions and the recovery of aroma or volatile components from fruit juices and fluid foods.

A variety of design procedures for distillation columns, ranging from short-cut methods to rigorous tray-by-tray calculations, are described in specialized books (Van Winkle, 1967; Billet, 1973; King, 1982) and the AIChE (2000) CD-ROM. A short-cut calculation of distillation design, using computer spreadsheets, is discussed by Jevric and Fayed (2002).

For the purpose of this book, a systematic step-by-step design procedure, based on the FUG (Fenske-Underwood-Gilliland) analytical method, is presented. The proposed procedure is suitable for spreadsheet calculations and it is applied to the design of an ethanol recovery distillation unit from fermentation solutions.

2. Process Description

The separation of volatile components of liquid mixtures is usually achieved in a series of equilibrium stages, operated countercurrently in distillation columns. Single stage separators can separate partially a component, because of equilibrium limitations. Single stage or flash units are used to separate some components from food liquids, e.g., off-flavors from milk.

Most stripping (removal) and rectifying (enrichment) of volatile components in a mixture is carried out in columns, using various types of trays (plates), each one of which acts as a vapor/liquid equilibrium stage. Since thermodynamic equilibrium is not possible to be reached in a tray liquid/vapor contactor, the number of trays, for a given separation, is always greater than the number of theoretical stages.

Most of the industrial distillation columns are operated as continuous units, although there are some batch columns, used in small-scale operations. Figure 9.1 shows diagrammatically a simple continuous distillation column. The unit consists of a long vertical column, containing the required number of trays, made up of the stripping (lower) and the fractionating (upper) sections. The trays (perforated, bubble cups or valves) allow the counter flow of liquid and vapor, after thorough mixing to approach equilibrium. The column is equipped with a reboiler at the bottom, which produces the required vapor flow upwards, and a condenser at the top, which supplies the required liquid flow downwards.

Feed F is introduced near the middle of the column, while a distillate D is received from the top and a residue B is obtained from the bottom. Steam S is used to heat the liquid in the reboiler and cooling water W is used in the condenser. The column is designed to separate a component from the feed of concentration X_F to a distillate X_D and a residue X_B .

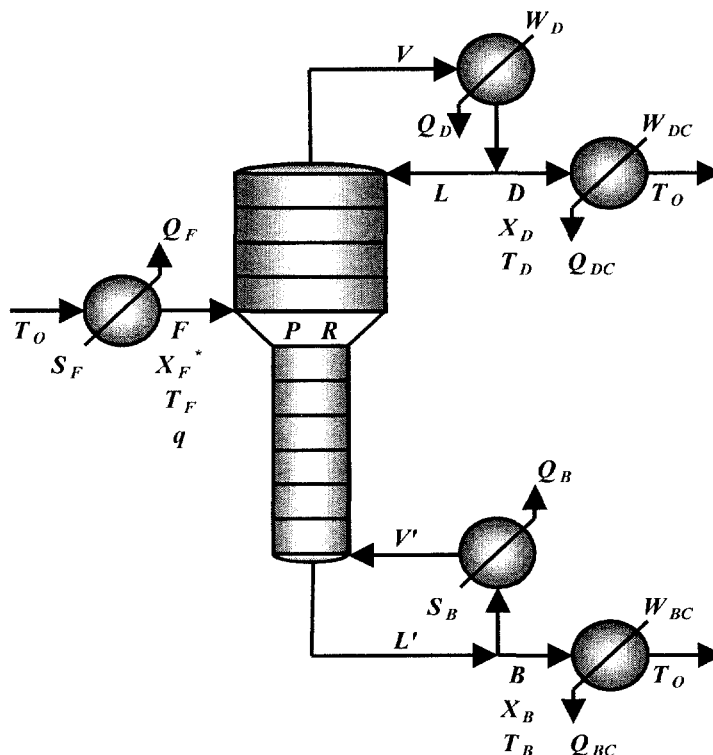


Figure 9.1 Diagram of a distillation process.

In most distillation columns, a total condenser is used, i.e., all the vapors coming out of the first (top) tray are condensed, and the liquid condensate is split into two streams, the distillate product and the reflux, which is returned to the column.

3. Process Model

a. Vapor-Liquid Equilibrium

Vapor-liquid equilibrium data are required for the design of the distillation columns. The basic laws and definitions concerning vapor-liquid equilibrium are summarized in Table 9.1.

Aqueous food mixtures (solutions) of volatile components are usually non-ideal, and the partial pressure is given by Equation (E01), where γ_i is the activity coefficient of the i component in the mixture, x_i is the mole fraction, and P_i^o is the vapor pressure at the given temperature T . The vapor pressure P_i^o is estimated from the Antoine Equation as a function of the temperature T (Equation E02).

The vapor phase in food systems can be considered ideal and the Dalton law is applicable (Equation E03), where y_i is the mole fraction in the vapor phase and P is the total pressure.

The assumption of an ideal vapor phase is reasonable, since most food-processing operations are carried out at atmospheric pressure or in vacuum. Non-ideal gas phases characterize high-pressure operations ($P > 10$ bar).

At equilibrium, the partial pressure of a component i is the same in both phases, as shown by Equation (E04).

The partition coefficient K_i of component i between the two phases is defined by Equation (E05). Taking into account the equilibrium condition of Equation (E04), Equation (E05) is transformed to Equation (E06). It should be pointed out that, for a given system, K_i is directly proportional to the activity coefficient γ_i .

For ideal solutions $\gamma_i = 1$ (Equation E07).

The activity coefficients for non-ideal mixtures are functions of concentration of the liquid mixtures (Equation E07). Empirical correlations, used to correlate the activity coefficients, include the Margules (two-parameter), the van Laar (two-parameter), and the Wilson (n-parameter) equations. The Wilson equation and its modifications are suited for computer calculations. A generalized correlation for multicomponent equilibrium data is the UNIQUAC (Universal Quasi-Chemical) equation. The UNIFAC (Universal Function Activity Contribution) method is based on the contributions of certain structural parameters of the components of the liquid mixture, like relative volume and surface area, which are given in thermodynamic tables (Reid et al., 1987; Fredeslund et al., 1977; Gmehling et al., 1984).

The main conclusion from Table 9.1 is that the partition coefficients describe the vapor-liquid equilibrium. They are function of pressure, temperature and eventually (for non-ideal mixtures) liquid concentration.

$$K_i(P, T, \underline{x}) = \frac{\gamma_i(P, T, \underline{x})P_i^o(T)}{P}$$

Table 9.1 Vapor-Liquid Equilibrium**Raoult Law**

$$P_i = \gamma_i x_i P_i^o \quad (\text{E01})$$

Antoine Equation

$$P_i^o = A_{1i} - \frac{A_{2i}}{A_{3i} + T} \quad (\text{E02})$$

Dalton Law

$$P_i = y_i P \quad (\text{E03})$$

Equilibrium

$$y_i P = \gamma_i x_i P_i^o \quad (\text{E04})$$

Partition Coefficient

$$K_i = \frac{y_i}{x_i} \quad (\text{E05})$$

$$K_i = \frac{\gamma_i P_i^o}{P} \quad (\text{E06})$$

Activity Coefficient

$$\gamma_i = \begin{cases} 1 & \text{for ideal mixtures} \\ \gamma_i(P, T, \underline{x}) & \text{for non-ideal} \end{cases} \quad (\text{E07})$$

In distillation design, partition coefficients are used to calculate

- the bubble and dew point temperatures
- the relative volatility
- the feed quality

Bubble and dew point temperatures at feed, top, and bottom are useful in distillation column design. They are calculated using the equations summarized in Table 9.2.

Relative volatility is crucial in theoretical stages calculation. Once we have estimated the bubble points of the feed, overhead and bottom streams at the operating pressure of the column, we can calculate the relative volatility of the components with respect to the heavy key component at feed, top and bottom. It is common practice to use as a design value of the relative volatility the geometric mean of the feed, top and bottom values. As an alternative for safe design, the smallest of these values may be used (see Table 9.3).

Table 9.2 Bubble and Dew Point Calculations**Top**

$$X_D K_{LC}(P, T_D) + (1-X_D) K_{HC}(P, T_D) = 1 \quad (\text{E01})$$

$$X_D / K_{LC}(P, T_D') + (1-X_D) / K_{HC}(P, T_D') = 1 \quad (\text{E02})$$

Bottom

$$X_B K_{LC}(P, T_B) + (1-X_B) K_{HC}(P, T_B) = 1 \quad (\text{E03})$$

$$X_B / K_{LC}(P, T_B') + (1-X_B) / K_{HC}(P, T_B') = 1 \quad (\text{E04})$$

Feed

$$X_F K_{LC}(P, T_F) + (1-X_F) K_{HC}(P, T_F) = 1 \quad (\text{E05})$$

$$X_F / K_{LC}(P, T_F') + (1-X_F) / K_{HC}(P, T_F') = 1 \quad (\text{E06})$$

Table 9.3 Relative Volatility Calculation

$$a = \min(a_D, a_F, a_B) \quad (\text{E01})$$

$$a_D = \frac{K_{LC}(P, T_D)}{K_{HC}(P, T_D)} \quad (\text{E02})$$

$$a_F = \frac{K_{LC}(P, T_F)}{K_{HC}(P, T_F)} \quad (\text{E03})$$

$$a_B = \frac{K_{LC}(P, T_B)}{K_{HC}(P, T_B)} \quad (\text{E04})$$

Feed quality strongly affects the distillation design. It can be obtained by applying the well-known flash-equations presented in Table 9.4 to the feed stream.

Equation (E01) is used to calculate the feed quality (q) when the feed temperature is known or, inversely, to calculate the feed temperature when the feed quality is known. Equations (E02) and (E03) estimate the concentrations of the liquid and vapor phases, which are needed in enthalpy calculations.

Table 9.4 Feed Quality (Flash Calculation)

$$\frac{X_F}{(1-q) + \frac{1}{K_{LC}(P, T_F^*) - 1}} + \frac{1 - X_F}{(1-q) + \frac{1}{K_{HC}(P, T_F^*) - 1}} = 0 \quad (\text{E01})$$

$$X_{FL} = \frac{X_F}{(1-q)(K_{LK}(P, T_F^*) - 1) + 1} \quad (\text{E02})$$

$$X_{FV} = K_{LK}(P, T_F^*) X_{FL} \quad (\text{E03})$$

b. Material and Heat Balances

The material balances of the distillation column system are summarized in Table 9.5. Equation (E01) is the total material balance for the column unit and (E02) the balance of the volatile component in the total column unit. Equation (E03) is the definition of the reflux ratio R and Equation (E05) the definition of the feed quality q . Equation (E04) is the total material balance at the top splitter and (E06) at the bottom splitter.

The heat balances at the distillation column are summarized in Table 9.6. They are used to calculate the thermal duties of the auxiliary heat exchangers. The stream enthalpies are estimated by assuming that the heat of mixing is negligible, and the enthalpy of any component is described by using three constant characteristic quantities, that is (a) the latent heat of vaporization at 0°C, (b) the average specific heat of liquid component and (c) the average specific heat of vapor component.

Table 9.5 Material Balances

$$F = B + D \quad (\text{E01})$$

$$F X_F = B X_B + D X_D \quad (\text{E02})$$

$$R = L / D \quad (\text{E03})$$

$$V = L + D \quad (\text{E04})$$

$$q = (L' - L) / F \quad (\text{E05})$$

$$L' = B + V' \quad (\text{E06})$$

Table 9.6 Heat Balances**Feed Heater**

$$Q_F = F (H_F - H_{F0}) \quad (\text{E01})$$

$$H_{F0} = X_F C_{PL}^{LC} T_o + (1 - X_F) C_{PL}^{HC} T_o \quad (\text{E02})$$

$$H_F = q H_{FL} + (1 - q) H_{FV} \quad (\text{E03})$$

$$H_{FL} = X_{FL} C_{PL}^{LC} T_F^* + (1 - X_{FL}) C_{PL}^{HC} T_F^* \quad (\text{E04})$$

$$H_{FV} = X_{FV} (\Delta H_o^{LC} + C_{PV}^{LC} T_F^*) + (1 - X_{FV}) (\Delta H_o^{HC} + C_{PV}^{HC} T_F^*) \quad (\text{E05})$$

Bottom Reboiler

$$Q_B = V' (H_{B'} - H_B) \quad (\text{E06})$$

$$H_{B'} = X_B (\Delta H_o^{LC} + C_{PV}^{LC} T_B') + (1 - X_B) (\Delta H_o^{HC} + C_{PV}^{HC} T_B') \quad (\text{E07})$$

$$H_B = X_B C_{PL}^{LC} T_B + (1 - X_B) C_{PL}^{HC} T_B \quad (\text{E08})$$

Bottom Product Cooler

$$Q_{BC} = B (H_B - H_{B0}) \quad (\text{E09})$$

$$H_{B0} = X_B C_{PL}^{LC} T_o + (1 - X_B) C_{PL}^{HC} T_o \quad (\text{E10})$$

Overhead Condenser

$$Q_D = V (H_{D'} - H_D) \quad (\text{E11})$$

$$H_{D'} = X_D (\Delta H_o^{LC} + C_{PV}^{LC} T_D') + (1 - X_D) (\Delta H_o^{HC} + C_{PV}^{HC} T_D') \quad (\text{E12})$$

$$H_D = X_D C_{PL}^{LC} T_D + (1 - X_D) C_{PL}^{HC} T_D \quad (\text{E13})$$

Overhead Product Cooler

$$Q_{DC} = D (H_D - H_{D0}) \quad (\text{E14})$$

$$H_{D0} = X_D C_{PL}^{LC} T_o + (1 - X_D) C_{PL}^{HC} T_o \quad (\text{E15})$$

c. Column Size

One of the most commonly used procedures for obtaining quick estimates of the number of theoretical stages required for a distillation separation is called the Fenske-Underwood-Gilliland (FUG) procedure.

Gilliland developed an empirical graphical correlation for the number of theoretical stages N in terms of the minimum number of stages at total reflux N_{min} , the minimum reflux ratio R_{min} , and the actual reflux ratio R (Table 9.7, Equation E05). Hence we can calculate N as a function of R after we have estimated N_{min} and R_{min} .

For constant a systems, Fenske derived an expression for the minimum number of theoretical stages at total reflux. The result is Equation (E02) in Table 9.7, where the separation factor SF is defined by Equation (E01). The resulting N_{min} is sensitive to relative volatility and consequently good estimates are needed. Smaller values of a overestimate the column size but afford safety margins.

Underwood Equations (E03 and E04) are the most used for estimating the R_{min} . The parameter θ is calculated from (E03) and used in (E04). Its value must be in the range (1, a).

The overall plate efficiency is defined by Equation (E02) in Table 9.8. A simple, but accurate, technique for estimating overall plate efficiency is to use O'Connell's correlation (Table 9.8, Equation E01). The column height is estimated by the geometric Equation (E03).

The estimation of the column diameter is based on the assumption that the vapor velocity in the column should be about 60% of the flooding velocity, that is Equation (E06) for the rectifying section and (E09) for the stripping section. The flooding velocity can be estimated from the Fair data, to which Equation (E05) has been fitted. Equation (E04) calculates the vapor molar density using the state equation of perfect gases.

Finally the feed-point location can be estimated from the Kirkbride empirical equation (E10).

Table 9.7 The FUG Procedure for Calculation of Theoretical Number of Stages

Fenske

$$SF = \frac{X_D}{1 - X_D} \frac{1 - X_B}{X_B} \quad (E01)$$

$$N_{min} = \frac{\ln(SF)}{\ln(a)} \quad (E02)$$

Underwood

$$\frac{aX_F}{a - \theta} + \frac{1 - X_F}{1 - \theta} = 1 - q \quad (E03)$$

$$R_{min} = \frac{aX_D}{a - \theta} + \frac{1 - X_D}{1 - \theta} - 1 \quad (E04)$$

Gilliland

$$\frac{N - N_{min}}{N + 1} = 0.75 \left[1 - \left(\frac{R - R_{min}}{R + 1} \right)^{0.57} \right] \quad (E05)$$

Table 9.8 Column Sizing**Column Height**

$$E_o = \frac{0.50}{(0.30a)^{0.25}} \quad (\text{E01})$$

$$N_{act} = \frac{N}{E_o} \quad (\text{E02})$$

$$H = H_0 + N_{act} H_1 \quad (\text{E03})$$

Column Diameter (Rectifying Section)

$$\rho_V = \frac{P}{RT_D} \quad (\text{E04})$$

$$u_F \sqrt{\rho_V} = 0.47 H_1^{0.74} \quad (\text{E05})$$

$$\frac{\pi D^2}{4} = \frac{V}{0.6 \rho_V u_F} \quad (\text{E06})$$

Column Diameter (Stripping Section)

$$\rho_V' = \frac{P}{RT_F} \quad (\text{E07})$$

$$u_F' \sqrt{\rho_V'} = 0.47 H_1^{0.74} \quad (\text{E08})$$

$$\frac{\pi D^2}{4} = \frac{V}{0.6 \rho_V' u_F'} \quad (\text{E09})$$

Feed-point Location

$$\ln\left(\frac{N_D}{N_B}\right) = 0.206 \ln\left[\frac{B}{D} \left(\frac{1-X_F}{X_F}\right) \left(\frac{X_B}{1-X_D}\right)^2\right] \quad (\text{E10})$$

$$N_{act} = N_D + N_B \quad (\text{E11})$$

d. Column Auxiliaries Sizing

All the distillation column auxiliaries are essentially heat exchangers and their sizing is based on the well known equation for heat flow in heat exchangers. The logarithmic mean temperature difference is considered the driving force. These equations are summarized in Table 9.9.

The corresponding utility flows are calculated in Table 9.10, using heat balances for the utilities streams. These flows could be reduced by energy integration methods.

Table 9.9 Heat Exchangers Sizing**Feed Heater**

$$Q_F = U_F A_F [(T_S - T_{Fo}) - (T_S - T_{Fi})] / \ln[(T_S - T_{Fo}) / (T_S - T_{Fi})] \quad (\text{E01})$$

Bottom Reboiler

$$Q_B = U_B A_B (T_S - T_B) \quad (\text{E02})$$

Bottom Product Cooler

$$Q_{BC} = U_{BC} A_{BC} [(T_B - T_{w2}) - (T_o - T_{wi})] / \ln[(T_B - T_{w2}) / (T_o - T_{wi})] \quad (\text{E03})$$

Overhead Condenser

$$Q_D = U_D A_D [(T_D - T_{w2}) - (T_D - T_{wi})] / \ln[(T_D - T_{w2}) / (T_D - T_{wi})] \quad (\text{E04})$$

Overhead Product Cooler

$$Q_{DC} = U_{DC} A_{DC} [(T_D - T_{w2}) - (T_o - T_{wi})] / \ln[(T_D - T_{w2}) / (T_o - T_{wi})] \quad (\text{E05})$$

Table 9.10 Utility Flow Rates**Feed Heater**

$$Q_F = S_F \Delta H_S \quad (\text{E01})$$

Bottom Reboiler

$$Q_B = S_B \Delta H_S \quad (\text{E02})$$

Bottom Product Cooler

$$Q_{BC} = W_B C_{PW} (T_{w2} - T_{wi}) \quad (\text{E03})$$

Overhead Condenser

$$Q_D = W_D C_{PW} (T_{w2} - T_{wi}) \quad (\text{E04})$$

Overhead Product Cooler

$$Q_{DC} = W_D C_{PW} (T_{w2} - T_{wi}) \quad (\text{E05})$$

e. Costing

The total annualized cost of the process TAC is selected as the objective function to be optimized. Equation (E14) in Table 9.11 defines the total annualized cost TAC , which is a weighted sum between the equipment C_{eq} and operating cost C_{op} . The weighting factor is the capital recovery factor e , which is defined by the well-known Equation (E15). Equation (E01) calculates the equipment cost C_{eq} , and (E13) the annual operating cost C_{op} .

The column cost is estimated by using the Guthrie's equations. Equation (E02) calculates the cost of the column shell and Equation (E03) the cost of the trays (plates). Equations (E04) through (E08) calculate the cost of the auxiliary exchangers. The correction factors for pressure (Equation E11) and for temperature (E10) presented in Table 9.11 come from fitting exponential equations to data presented by Biegler et al. (1997).

Finally, Equation (E12) updates the Chemical Engineering Plant Index (CEP).

Table 9.11 Cost Estimation**Equipment cost**

$$C_{eq} = C_{Sh} + C_{trs} + C_F + C_B + C_D + C_{BC} + C_{DC} \quad (E01)$$

Column

$$C_{Sh} = f C_{1Sh} (DH^{0.8} + D'H'^{0.8}) \quad (E02)$$

$$C_{trs} = f C_{1trs} (D^{1.5}H + D'^{1.5}H') \quad (E03)$$

Exchangers

$$C_F = f C_{1exc} A_F^{0.65} \quad (E04)$$

$$C_B = f C_{1exc} A_B^{0.65} \quad (E05)$$

$$C_D = f C_{1exc} A_D^{0.65} \quad (E06)$$

$$C_{BC} = f C_{1exc} A_{BC}^{0.65} \quad (E07)$$

$$C_{DC} = f C_{1exc} A_{DC}^{0.65} \quad (E08)$$

Correction Factors

$$f = f_T f_P \left(\frac{CEP}{500} \right) \quad (E09)$$

$$f_T = 0.85 \exp\left(\frac{T}{1000}\right) \quad (E10)$$

$$f_P = 0.98 \exp\left(\frac{P}{50}\right) \quad (E11)$$

$$\left(\frac{CEP}{500} \right) = \exp[0.02(t - 2000)] \quad (E12)$$

Annual operating cost

$$C_{op} = [C_S(Q_F + Q_B) + C_W(Q_D + Q_{DC} + Q_{BC})] t_y \quad (E13)$$

Total annualized cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (E14)$$

Capital recovery factor

$$e = \frac{i_r(1+i_r)^{t_f}}{(1+i_r)^{t_f} - 1} \quad (E15)$$

4. Process Design

In a typical distillation design problem, the feed stream is known and the concentrations of top and bottom streams are fixed by product purity specifications. Usually, the utility characteristics are also known, that is the temperature of the heating steam and the temperature operating range of the cooling water. These process design specifications are summarized in Table 9.12.

A degrees-of-freedom analysis is presented in Table 9.13. The mathematical model described in Tables 9.2 through 9.10 consists of 60 equations, in which 73 variables are incorporated, resulting in 13 degrees of freedom. Process specifications fix 8 variables, and consequently 5 design variables are available. The variables summarized in Table 9.14 are usually selected as design variables.

Table 9.15 summarizes a systematic design procedure for distillation columns (compositions are expressed as mole fractions).

Table 9.12 Process Specifications

| Process streams | |
|------------------------|----------------------------------|
| F | Feed flow rate (molar) |
| X_F | Feed composition |
| X_D | Overhead composition |
| X_B | Bottom composition |
| Utilities | |
| T_S | Steam temperature |
| T_{w1} | Cooling water inlet temperature |
| T_{w2} | Cooling water outlet temperature |
| T_o | Ambient temperature |

Table 9.13 Degrees-of-Freedom Analysis

| | |
|--------------------------|-----------|
| Process variables | 73 |
| <u>Process equations</u> | <u>60</u> |
| Degrees of freedom | 13 |
| Degrees of freedom | 13 |
| <u>Specifications</u> | <u>8</u> |
| Design variables | 5 |

Table 9.14 Design Variables

| | |
|-------|---------------------------------|
| P | Operating pressure |
| R | Reflux ratio |
| q | Feed quality |
| H_1 | Plate spacing |
| H_0 | Space at the ends of the column |

Step 1

Clarify process specifications presented in Table 9.12.

Step 2

Gather the appropriate technical and costing data from the literature, according to Tables 9.16 and 9.17. Experimental or empirical data on vapor/liquid equilibria (e.g., relative volatility) may be needed for non-ideal systems, such as aqueous systems (foods).

Step 3

Assign initial values to the design variables, according to the guidelines in Table 9.18.

Step 4

Solve the distillation column model presented in Tables 9.2 through 9.10, according to the solution algorithm presented in Table 9.19.

Step 5

Estimate the process equipment and operating cost, according to the cost model in Table 9.11 and the algorithm in Table 9.20.

Step 6

Estimate the optimal values for the design variables using an optimization algorithm.

Step 7

Analyze the sensitivity of the results with respect to process specifications and process data (technical and costing).

Table 9.16 Technical Data

| | |
|------------------|---|
| | Latent heat of vaporization at reference temperature (0°C) |
| ΔH_{os} | Water |
| ΔH_{oLC} | Light component |
| ΔH_{oHC} | Heavy component |
| | Average specific heat of liquids |
| C_{PLW} | Water |
| C_{PLL} | Light component |
| C_{PLH} | Heavy component |
| | Average specific heat of vapors |
| C_{PVW} | Water |
| C_{PVL} | Light component |
| C_{PVH} | Heavy component |
| | Antoine constants |
| A_{1LC} | Light component |
| A_{2LC} | |
| A_{3LC} | |
| A_{1HC} | Heavy component |
| A_{2HC} | |
| A_{3HC} | |
| | Heat transfer coefficients |
| U_F | Feed heater |
| U_B | Bottom reboiler |
| U_{BC} | Bottom product cooler |
| U_D | Overhead condenser |
| U_{DC} | Overhead product cooler |

Table 9.17 Cost Data

| | |
|--------------|--------------------------------------|
| | Utility cost |
| C_W | Cost of cooling water (\$/kWh) |
| C_S | Cost of steam (\$/kWh) |
| | Equipment unit cost |
| C_{1shell} | Column shell (k\$/m ²) |
| C_{1trays} | Column trays (k\$/m ²) |
| C_{1exc} | Heat exchanger (k\$/m ²) |
| | Other |
| t_y | Annual operating time (h/yr) |
| i_r | Interest rate (-) |
| l_f | Lifetime (yr) |
| t | Calendar year for CEP estimation |

Table 9.18 Guidelines for Assigning Initial Values to Design Variables

Column pressure (P)

The operating pressure P for a distillation column is normally fixed by the economic desirability of using a condenser supplied with the available cooling water, and a reboiler supplied with the available heating steam. That is:

(a) The bubble point at the condenser T_D should be higher than the operating range of cooling water (T_{W1} , T_{W2}) plus the minimum acceptable temperature difference ΔT_{min} for operating the condenser.

$$\text{Thus, } T_D - T_{W2} > \Delta T_{min}.$$

(b) The available steam temperature T_S should be higher than the dew point at the reboiler T_B' plus the minimum acceptable temperature difference dT_{min} for operating the reboiler.

$$\text{Thus, } T_S - T_B' > \Delta T_{min}.$$

Feed quality (q)

Feed quality q is defined as the fraction of liquid phase in the feed stream:

| | |
|-------------|----------------------|
| $q > 1$ | Subcooled liquid |
| $q = 1$ | Saturated liquid |
| $0 < q < 1$ | Vapor-liquid mixture |
| $q = 0$ | Saturated vapor |
| $q < 0$ | Superheated vapor |

Feed quality affects strongly the operating curves of the column, the thermal load distribution between the reboiler and the feed heater, the column diameter at stripping and rectifying section, etc.

Reflux ratio (R)

As the reflux ratio is increased, the number of theoretical stages (and trays) required for a given separation decreases, and the capital cost of the column decreases. However, increasing the reflux ratio will increase the vapor rate in the column, which corresponds to more expensive condensers and reboilers, along with higher cooling water and heating steam costs. Therefore, there is an optimum reflux ratio for any specified separation.

Experience has shown that the value of the optimum reflux ratio normally falls in the range $1.05 < R/R_{min} < 1.50$. The slope of the cost curve is very steep below the optimum, but relatively flat above it. Hence, it is common practice to use the rule-of-thumb value $R/R_{min} = 1.20$

Tray spacing (H_1 , H_0)

It is common practice to use $\frac{1}{4}$ -1m (1-3 ft) space for each tray, and $1\frac{1}{2}$ -3m (5-10 ft) space at the top and the bottom of the column.

It is undesirable to build very tall and skinny columns because they will bend. A design guideline often used is that the column height should be less than about 60 m (175 ft), but a better design guideline is that the height to diameter ratio should be less than 20 - 30.

Table 9.19 Solution Algorithm for Process Model**Bubble and Dew Point Temperature (Table 9.2)**

(E01) $\rightarrow T_D$

(E02) $\rightarrow T_D'$

(E03) $\rightarrow T_B$

(E04) $\rightarrow T_B'$

(E05) $\rightarrow T_F$

(E06) $\rightarrow T_F'$

Design Value for Relative Volatility (Table 9.3)

(E04) $\rightarrow a_B$

(E03) $\rightarrow a_F$

(E02) $\rightarrow a_D$

(E01) $\rightarrow a$

Feed Quality (Table 9.4)

(E01) $\rightarrow T_F^*$

(E02) $\rightarrow X_{FL}$

(E03) $\rightarrow X_{FV}$

Mass Balances (Table 9.5)

(E01, E02) $\rightarrow D$

(E01) $\rightarrow B$

(E03) $\rightarrow L$

(E04) $\rightarrow V$

(E05) $\rightarrow L'$

(E06) $\rightarrow V'$

Heat Balances (Table 9.6)

(E05) $\rightarrow H_{FV}$

(E04) $\rightarrow H_{FL}$

(E03) $\rightarrow H_F$

(E02) $\rightarrow H_{Fo}$

(E01) $\rightarrow Q_F$

(E08) $\rightarrow H_B$

(E07) $\rightarrow H_B'$

(E06) $\rightarrow Q_B$

(E10) $\rightarrow H_{Bo}$

(E09) $\rightarrow Q_{BC}$

(E13) $\rightarrow H_D$

(E12) $\rightarrow H_D'$

(E11) $\rightarrow Q_D$

(E15) $\rightarrow H_{Do}$

(E14) $\rightarrow Q_{Do}$

Number of Theoretical Stages Estimation. The FUG Method (Table 9.7)

(E01) $\rightarrow SF$

(E02) $\rightarrow N_{min}$

(E03) $\rightarrow \theta$

(E04) $\rightarrow R_{min}$

(E05) $\rightarrow N$

Table 9.19 Continued**Column Sizing (Table 9.8)**

| | | |
|------------|---|-----------|
| (E01) | → | E_o |
| (E02) | → | N_{act} |
| (E03) | → | H |
| (E04) | → | ρ_V |
| (E05) | → | u_F |
| (E06) | → | D |
| (E07) | → | ρ_V' |
| (E08) | → | u_F' |
| (E09) | → | D' |
| (E10, E11) | → | N_D |
| (E11) | → | N_B |

Heat Exchangers Sizing (Table 9.9)

| | | |
|-------|---|----------|
| (E01) | → | A_F |
| (E02) | → | A_B |
| (E03) | → | A_{BC} |
| (E04) | → | A_D |
| (E05) | → | A_{DC} |

Utility Flow Rates Estimation (Table 9.10)

| | | |
|-------|---|----------|
| (E01) | → | S_F |
| (E02) | → | S_B |
| (E03) | → | W_B |
| (E04) | → | W_D |
| (E05) | → | W_{DC} |

Table 9.20 Solution Algorithm for the Cost Model in Table 9.11

| | | |
|-------|---|-----------|
| (E15) | → | e |
| (E12) | → | CEP |
| (E11) | → | f_P |
| (E10) | → | f_T |
| (E09) | → | f |
| (E02) | → | C_{Sh} |
| (E03) | → | C_{irs} |
| (E04) | → | C_F |
| (E05) | → | C_B |
| (E06) | → | C_D |
| (E07) | → | C_{BC} |
| (E08) | → | C_{DC} |
| (E01) | → | C_{eq} |
| (E13) | → | C_{op} |
| (E14) | → | TAC |

5. Application to Ethanol Distillation

Distillation is used in the production of brandy and other alcoholic beverages from wine and other ethanol-fermented liquids. Traditional brass (copper) stills of small capacity are used to separate the ethanol and other volatiles in simple one-stage distillations. Partial fractionation of ethanol is obtained by partial reflux in the piping of the still, obtaining distillates of about 50% ethanol by volume.

Fractional distillation of alcoholic beverages in medium- to large-scale operations is carried out in batch or continuous distillation columns made of stainless steel, producing ethanol of high concentration, up to 95% by volume.

Ethanol distillation for the production of alcoholic beverages and spirits is governed by strict regulations, concerned with taxation, such as the U.S. Bureau of Alcohol and Cigarettes.

The step-by-step design procedure, described in Table 9.15, is applied to design a distillation column to recover the ethanol from a fermentation solution, containing 5% of ethanol. The composition of the distillate and the bottom products will be 75% and 1% ethanol respectively (concentrations in molar basis). The feed to the column will be 0.143 kmol/s, that is 10 ton/h.

a. Conventional Column

Distillation of ethanol from aqueous solutions can be carried out in conventional columns, including a reboiler (Figure 9.1), or in special columns, without a reboiler.

The results of steps 1 to 3 (Table 9.15) for a conventional column are summarized in Tables 9.21 through 9.23. Table 9.21 shows the process specifications of both process streams and utilities. Table 9.22 contains the applied design parameters. Table 9.23 contains the technical data (physicochemical property values). Table 9.24 exhibits the cost data taken into account for costing the process most significant results from the solution of the model are presented in Table 9.25 and Figures 9.2 through 9.6. In Table 9.25 the first column refers to saturated liquid (feed quality $q=1$) and the second one to saturated vapor (feed quality $q=0$). In Figures 9.2 to 9.5 a comparison between the two cases of feed quality can be made. Figures 9.2 to 9.4 show in a quantitative way the enthalpy, thermal load, heat transfer area requirements. Finally, Table 9.26 and Figure 9.5 give the results of the cost analysis of ethanol distillation.

Table 9.21 Process Specifications for Ethanol Distillation

| Process streams | | |
|------------------------|------------------|----------------------------------|
| F | = 0.143 kmol/s | Feed flow rate |
| X_F | = 0.05 kmol/kmol | Feed composition |
| X_D | = 0.75 kmol/kmol | Overhead composition |
| X_B | = 0.01 kmol/kmol | Bottom composition |
| Utilities | | |
| T_S | = 160 °C | Steam temperature |
| T_{w1} | = 15 °C | Cooling water inlet temperature |
| T_{w2} | = 45 °C | Cooling water outlet temperature |
| T_o | = 20 °C | Ambient temperature |

Table 9.22 Design Variables for Ethanol Distillation

| | | |
|-------------|----------|----------------------|
| P | = 1 bar | Operating pressure |
| R/R_{min} | = 1.20 | Reflux ratio |
| q | = 0.50 | Feed quality |
| H_j | = 0.75 m | Plate spacing |
| H_o/H_j | = 3 | Space at column ends |

Table 9.23 Technical Data for Ethanol Distillation

| Latent heat of vaporization at reference temperature (0°C) | | |
|---|-----------------|-----------------|
| ΔH_{os} | = 44.8 MJ/kmol | Water |
| ΔH_{oLC} | = 40.5 MJ/kmol | Light component |
| ΔH_{oHC} | = 44.8 MJ/kmol | Heavy component |
| Average specific heat of liquids | | |
| C_{PLW} | = 75.3 kJ/kmolK | Water |
| C_{PLL} | = 96.7 kJ/kmolK | Light component |
| C_{PLH} | = 75.3 kJ/kmolK | Heavy component |
| Average specific heat of vapors | | |
| C_{PVW} | = 34.2 kJ/kmolK | Water |
| C_{PVL} | = 73.9 kJ/kmolK | Light component |
| C_{PVH} | = 34.2 kJ/kmolK | Heavy component |
| Antoine constants | | |
| A_{1LC} | = 1.23E+01 | Light component |
| A_{2LC} | = 3.80E+03 | |
| A_{3LC} | = 2.31E+02 | |
| A_{1HC} | = 1.19E+01 | Heavy component |
| A_{2HC} | = 3.99E+03 | |
| A_{3HC} | = 2.34E+02 | |
| Molecular weight | | |
| M_{LC} | = 46.1 kg/kmol | Light component |
| M_{HC} | = 18.0 kg/kmol | Heavy component |

Table 9.23 Continued

| Heat transfer coefficients | | |
|----------------------------|----------------------------|--|
| U_B | = 1.50 kW/m ² K | Bottom reboiler (Evaporation/Condensation) |
| U_F | = 1.25 kW/m ² K | Feed heater (Evaporation/Heating-Condensation) |
| U_D | = 1.00 kW/m ² K | Overhead condenser (Condensation/Heating) |
| U_{BC} | = 0.75 kW/m ² K | Bottom product cooler (Cooling/Heating) |
| U_{DC} | = 0.75 kW/m ² K | Overhead product cooler (Cooling/Heating) |

Table 9.24 Cost Data for Ethanol Distillation

| Utility cost | | |
|---------------------|-------------------------|---|
| C_W | = 0.004 \$/kWh | Cost of cooling water |
| C_S | = 0.040 \$/kWh | Cost of steam |
| Equipment unit cost | | |
| C_{1sh} | = 10 k\$/m ² | Column shell |
| C_{1tr} | = 1 k\$/m ² | Column trays |
| C_{1exc} | = 5 k\$/m ² | Heat exchanger |
| Other | | |
| t_y | = 2000 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 5 yr | Lifetime |
| t | = 2005 | Calendar year for <i>CEP</i> estimation |

Table 9.25 Results of Model Solution for $q=1$ (1st Column) and $q=0$ (2nd Column)

| Flow rate (kmol/s) | | | |
|-------------------------|---------|-------|------------------------------|
| F | = 0.143 | | Feed |
| D | = 0.008 | | Top product |
| B | = 0.135 | | Bottom product |
| V | = 0.114 | 0.247 | Vapor at rectifying section |
| V' | = 0.114 | 0.104 | Vapor at stripping section |
| L | = 0.107 | 0.239 | Liquid at rectifying section |
| L' | = 0.250 | 0.239 | Liquid at stripping section |
| Composition (kmol/kmol) | | | |
| X_F | = 0.05 | | Feed |
| X_D | = 0.75 | | Top product |
| X_B | = 0.01 | | Bottom product |
| X_{FV} | = 0.11 | 0.05 | Feed |
| X_{FL} | = 0.05 | 0.02 | Feed |

Table 9.25 Continued

| Temperature (°C) | | | |
|---|-----------|------|---|
| T_F^* | = 98.2 | 99.1 | Feed |
| T_F | = 98.2 | | Feed bubble point |
| T_F' | = 99.1 | | Feed dew point |
| T_D | = 81.9 | | Top bubble point |
| T_D' | = 85.2 | | Top dew point |
| T_B | = 99.5 | | Bottom bubble point |
| T_B' | = 99.7 | | Bottom dew point |
| Thermal load (MW) | | | |
| Q_F | = 0.85 | 6.67 | Feed heater |
| Q_B | = 4.66 | 4.22 | Bottom reboiler |
| Q_{BC} | = 0.81 | | Bottom product cooler |
| Q_D | = 4.51 | 9.71 | Overhead condenser |
| Q_{DC} | = 0.04 | | Overhead product cooler |
| Heat transfer area (m²) | | | |
| A_F | = 11 | 88 | Feed heater |
| A_B | = 51 | 46 | Bottom reboiler |
| A_{BC} | = 31 | | Bottom product cooler |
| A_D | = 89 | 193 | Overhead condenser |
| A_{DC} | = 4 | | Overhead product cooler |
| Column geometric characteristics | | | |
| N | = 15.0 | 14.8 | Total number of theoretical plates |
| N_{min} | = 7.01 | | Minimum number of theoretical plates |
| N_{act} | = 27 | | Actual number of plates |
| N_D | = 13 | | Number of plates at rectifying section |
| N_B | = 14 | | Number of plates at stripping section |
| D | = 2.5 | 3.7 | Column diameter at rectifying section |
| D' | = 2.5 | 2.4 | Column diameter at stripping section |
| H | = 23 | 23 | Column height |
| Column operating characteristics | | | |
| P | = 1 bar | | Operating pressure |
| q | = 1 | 0 | Feed quality |
| R | = 13.8 | 30.9 | Reflux ratio |
| R_{min} | = 11.5 | 25.8 | Minimum reflux ratio |
| u_f | = 1.1 m/s | | Flooding velocity at the rectifying section |
| u_f' | = 1.1 m/s | | Flooding velocity at the stripping section |
| SF | = 297 | | Separation factor |
| α | = 2.30 | | Design relative volatility |
| α_F | = 3.50 | | Relative volatility at feed |
| α_D | = 2.30 | | Relative volatility at top |
| α_B | = 5.60 | | Relative volatility at bottom |
| E_o | = 0.55 | | Overall plate efficiency |

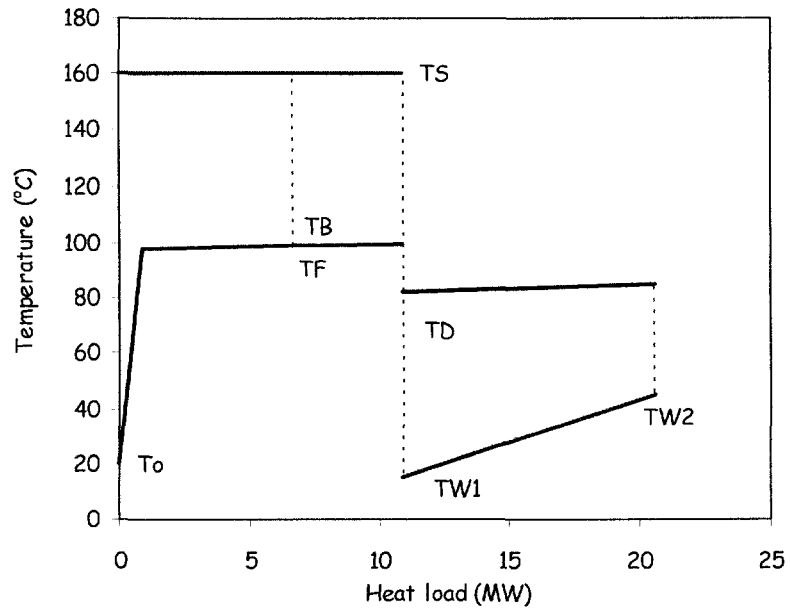
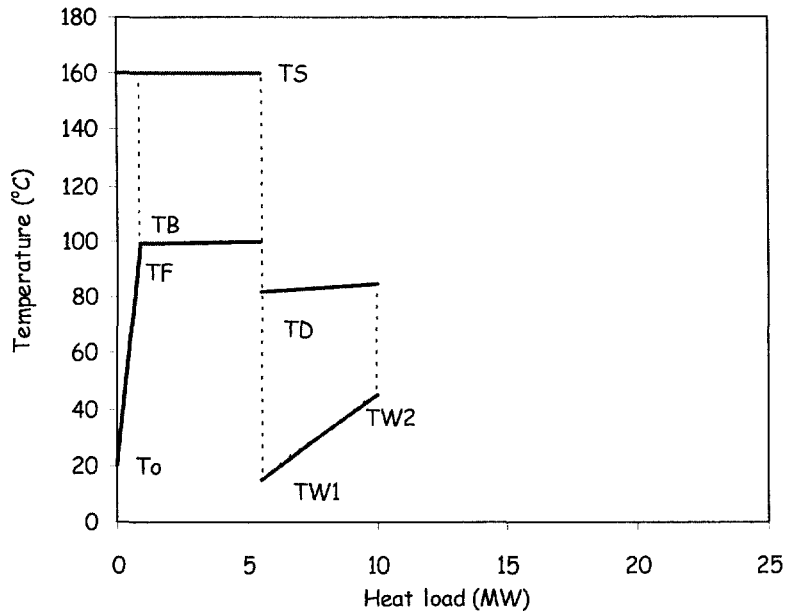


Figure 9.2 Enthalpy-temperature diagram for feed heater, reboiler, and condenser, for $q=1$ (upper) and $q=0$ (lower), respectively.

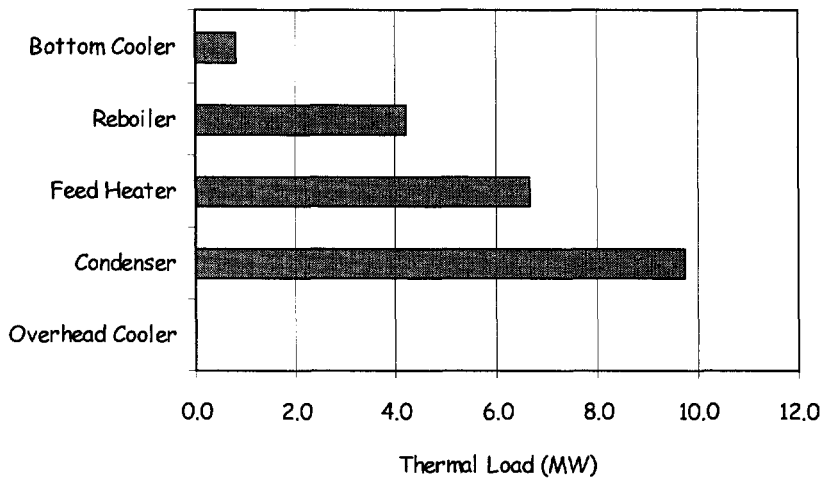
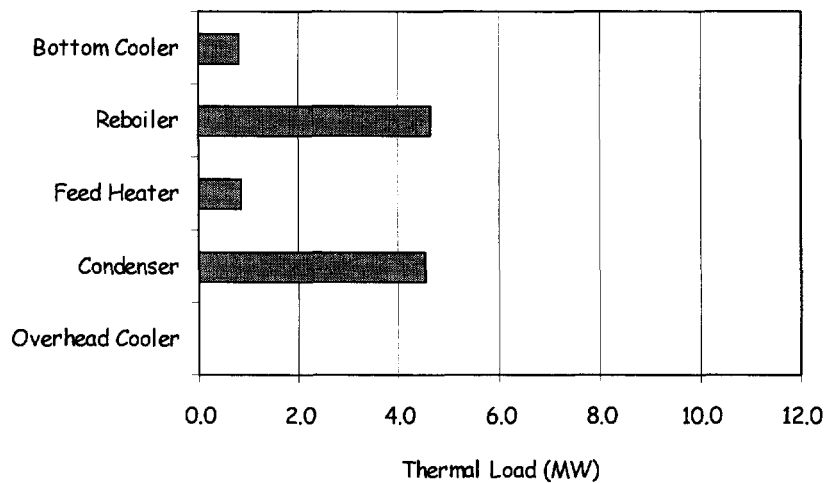


Figure 9.3 Thermal duties of the distillation unit, for $q=1$ (upper) and $q=0$ (lower), respectively.

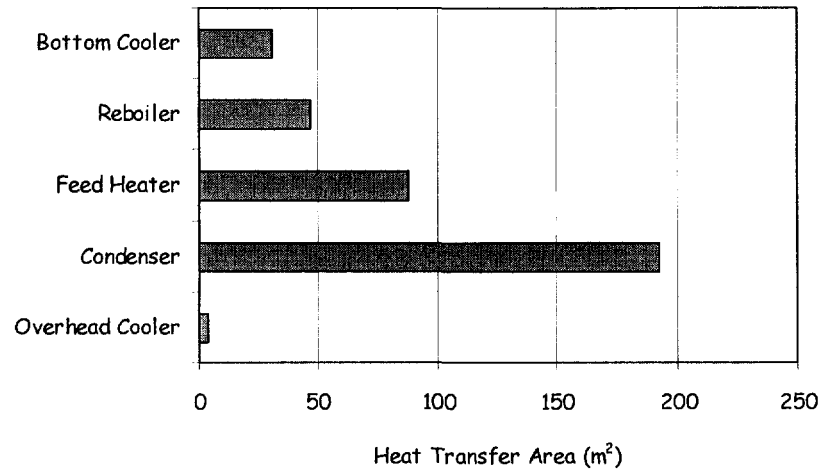
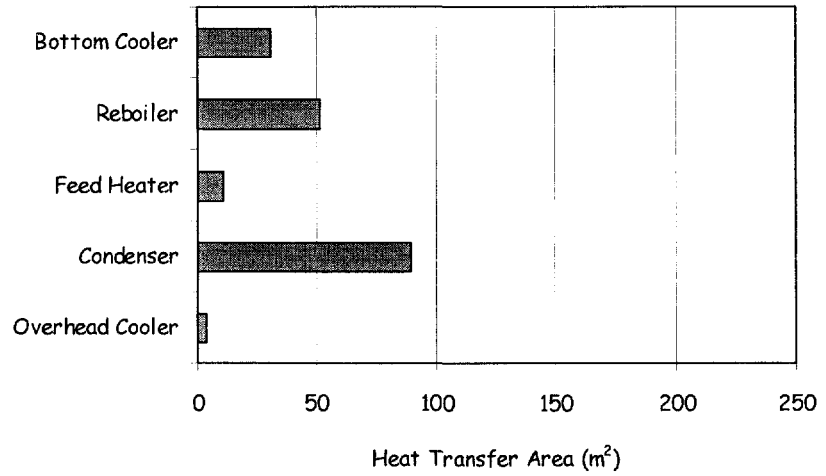


Figure 9.4 Required heat transfer areas for $q=1$ (upper) and $q=0$ (lower), respectively.

Table 9.26 Cost Analysis (Saturated Liquid Feed, $q=1$)

| Equipment | Size | Cost | Utility | Load |
|---------------------------|---------------------|-------------------|---------------|---------|
| <u>Heat Exchangers</u> | | | | |
| Bottom cooler | 30.9 m ² | 51 k\$ | Cooling water | 0.81 MW |
| Reboiler | 51.3 m ² | 107 k\$ | Steam | 4.66 MW |
| Feed heater | 11.1 m ² | 39 k\$ | Steam | 0.85 MW |
| Condenser | 89.4 m ² | 102 k\$ | Cooling water | 4.51 MW |
| Overhead cooler | 89.4 m ² | 13 k\$ | Cooling water | 0.04 MW |
| | | <u>313 k\$</u> | | |
| <u>Column</u> | | | | |
| Shell | 2.5m dia x 23m | 440 k\$ | | |
| Trays | 27 trays | 107 k\$ | | |
| | | <u>547 k\$</u> | | |
| <u>Utilities</u> | | | | |
| Cooling water | | 43 k\$/yr | | |
| Heating steam | | 441 k\$/yr | | |
| Operating cost | | <u>484 k\$/yr</u> | | |
| <u>Objective function</u> | | | | |
| Equipment cost | | 860 k\$ | | |
| Operating cost | | 484 k\$/yr | | |
| Total annualized cost | | <u>699 k\$/yr</u> | | |

It should be noted that all the above results (both technical and economical) are very sensitive to the design value of relative volatility. Ethanol/water is a non-ideal solution and the bottom product has a much higher relative volatility than the top. Empirical values of the relative volatilities a were obtained from experimental measurements and the literature (Saravacos and Kostaropoulos, 2002): $a_F = 3.5$, $a_D = 2.3$, $a_B = 5.6$, that is the relative volatility varies between 2.3 and 5.6. The geometric average of these values is often used as a design value. In this text the minimum value is considered for safe design. The effect of relative volatility on the actual number of trays, the required thermal load, and the resulting total annualized cost is presented in Figures 9.6 through 9.8, respectively. Figure 9.9 compares the vapor/liquid equilibria for different relative volatility values with the experimental data.

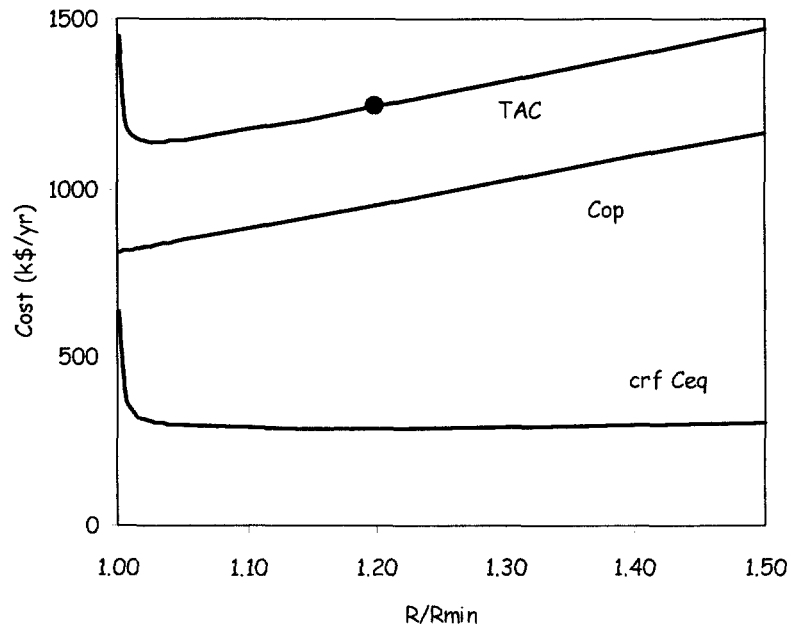
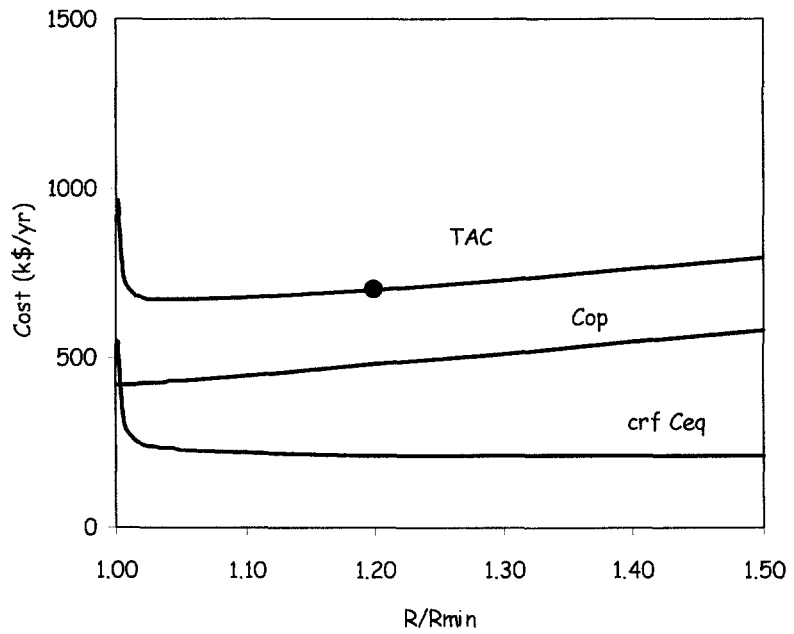


Figure 9.5 Effect of R/R_{min} on the total annualized cost TAC , equipment cost C_{eq} and operating cost Cop for $q=1$ (upper) and $q=0$ (lower), respectively.

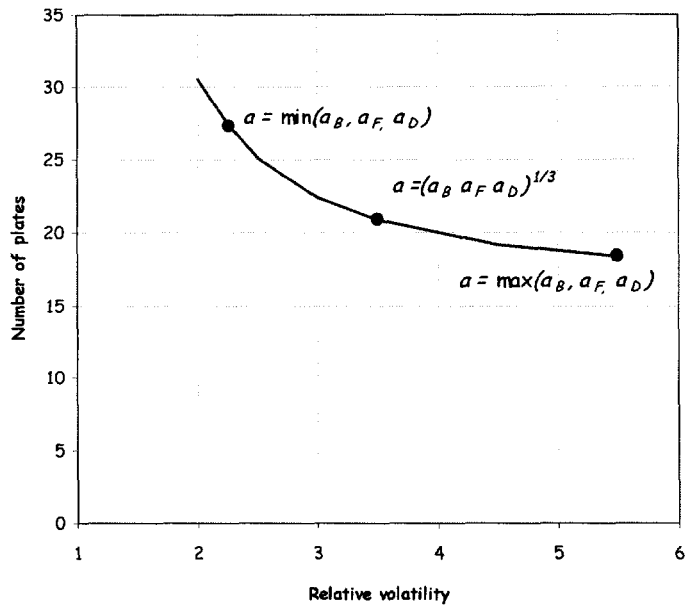


Figure 9.6 Effect of relative volatility on the required number of trays ($q=1$).

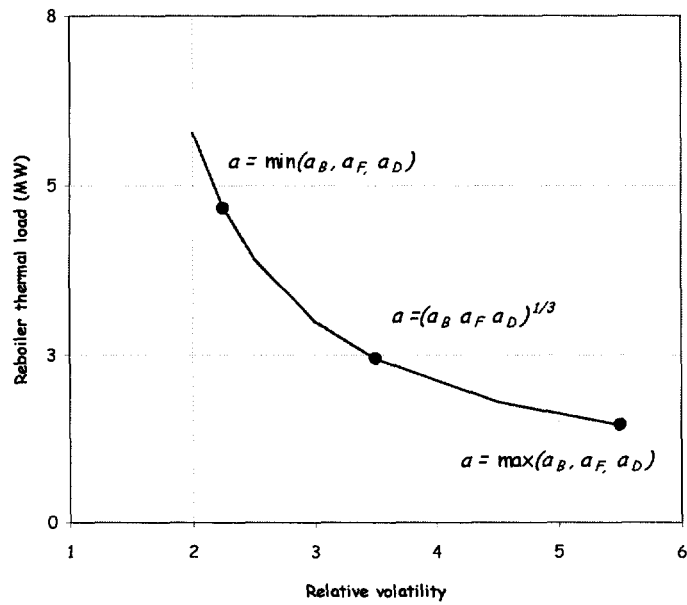


Figure 9.7 Effect of relative volatility on the required thermal load ($q=1$).

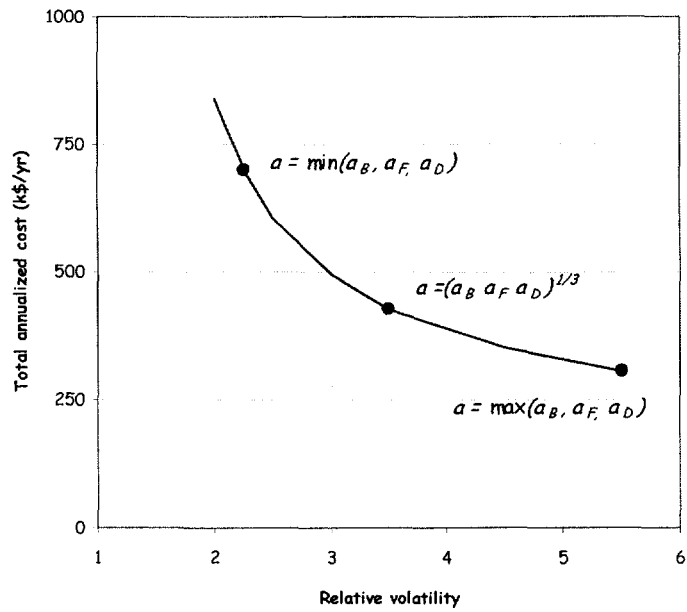


Figure 9.8 Effect of relative volatility on the total annualized cost ($q=1$).

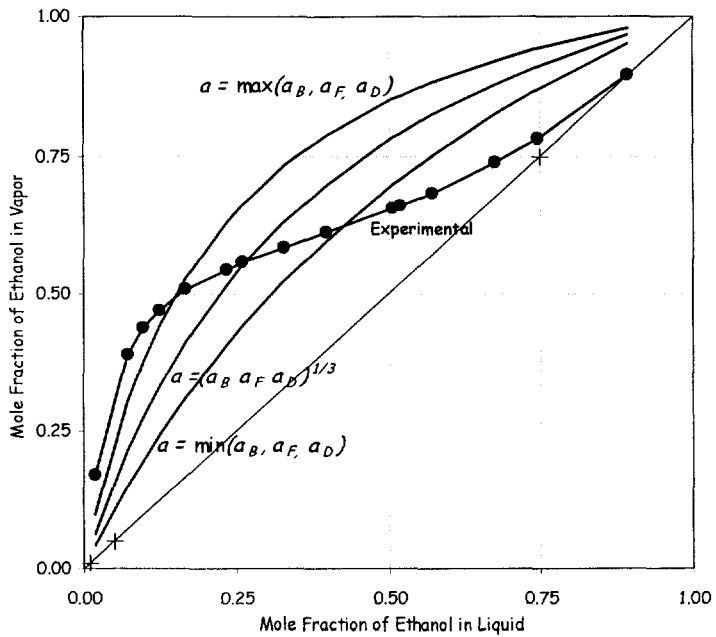


Figure 9.9 Vapor/liquid equilibrium of ethanol/water (atmospheric pressure).

b. Column without Reboiler

Since the bottom product (water) of ethanol distillation has no value, a simpler column can be used, without a reboiler, using direct heating with live steam (Figure 9.10). A side stream of fusel oils (higher alcohols) is usually removed from ethanol distillation columns (Saravacos and Kostaropoulos, 2002). The column with direct (live) steam heating is more economical than the conventional reboiler column.

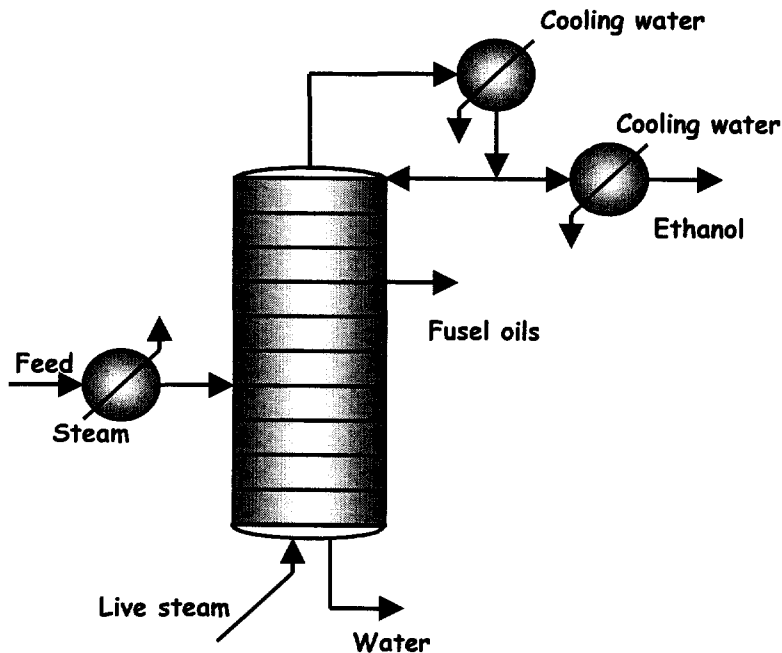


Figure 9.10 Distillation column for ethanol from aqueous solution, direct steam heating.

6. Aroma Recovery

Recovery of aroma or volatile components by distillation is important in the processing of fruit juices and other fluid foods, and it is applied in connection with evaporation and dehydration processes.

The fruit aromas consist of volatile organic compounds, such as esters, alcohols, and oxygenated compounds, which are lost or changed during evaporation, drying, and other physical and thermal processes. Most of these compounds are partially soluble in water and, therefore, they have high activity coefficients and high relative volatilities. Concentration of fruit juices by evaporation, or drying of aroma-containing foods results in considerable losses of most aroma components from the concentrated or dehydrated product (Karlsson and Tragardh, 1997; Saravacos and Kostaropoulos, 2002).

Aroma recovery can be accomplished by stripping/distillation processes, supercritical fluid extraction, pervaporation (membrane separation), solvent extraction and solid/fluid adsorption. The design of aroma recovery systems is based on vapor/liquid equilibria of aroma components/water and on distillation theory and practice (Roger and Turkot, 1965; Moyer and Saravacos, 1968; Bruin, 1969; Sulc, 1984; AIChE, 2000).

A conventional essence (aroma) recovery system, used in food processing, is shown in Figure 9.11. The fluid food, e.g., fruit juice, is stripped of the volatile aroma in e.g., a falling film evaporator, heated with saturated steam. The vapors from the vapor/liquid separator are introduced into the distillation column, while the stripped juice is concentrated further by evaporation. The distillation column separates the mixture into a distillate and a bottom product. The distillate is condensed with cold water, and cooled in a refrigerated condenser. The concentration of the volatile components in the aqueous aroma solution is about 100-200 times higher than the original concentration in the juice (Moyer and Saravacos, 1968).

The volatile aroma components in liquid foods, i.e., in aqueous systems, are generally non-ideal solutions with high relative volatilities (Sancho et al., 1997). They are present at very low concentrations (less than 0.1%), and they are considered as not interacting with each other during distillation. The presence of soluble sugars increases sharply the volatility of the aroma components in the food product (Saravacos and Kostaropoulos, 2002).

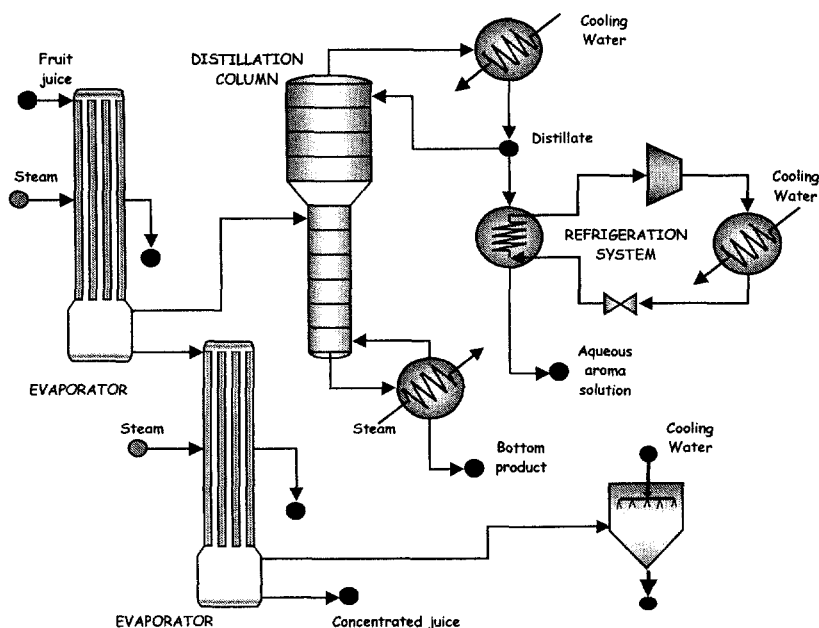


Figure 9.11 Simplified diagram of an essence recovery system at atmospheric pressure.

The design of distillation equipment is based on the separation of a typical “key” component from water, which is difficult to remove from the aqueous solution, i.e., it has a relative volatility lower than the other aroma components. Thus, by recovering this “key” component, most of the aroma components will be recovered in the distillate. As an example, the characteristic flavor component methyl anthranilate of Concord grapes with a relative volatility $\alpha = 3.4$ is used as a “key” aroma component in designing an aroma recovery system for grape juice (Moyer and Saravacos, 1968). The design of such a column would be similar to the design of ethanol distillation (for $\alpha = 2.3$), described in detail in this chapter. The column efficiency E_o in water-rich distillations is relatively low (50-60%), due to the difficulty of mixing water/vapors effectively. Typical columns of aroma recovery contain about 15 trays of 1 m diameter and 7.5 m high (distance between trays about 0.5 m). In small to medium-size operations, packed distillation columns are used, instead of tray columns.

Very volatile aroma components, such as esters, are easily stripped by evaporating a relatively small portion of the juice, e.g., 10-25% in apple juice. Higher boiling aroma components, such as methyl anthranilate and some carboxylic acids may require removal of about 40% water for substantial stripping (Saravacos and Moyer, 1969).

Vacuum stripping of aromas from heat-sensitive food liquids, such as orange juice, requires a special vacuum pump, which can collect most of the volatiles in the liquid ring of the pump (Bomben et al., 1973). An inert gas stream (nitrogen or air) may be injected in the bottom of the stripping column, and condensing of the stripped volatiles from the carrier gas requires low-temperature condensers and cold traps.

7. Spinning Cone Stripping Column

The spinning cone stripping column (SCSC) is a special distillation column, which is used for recovering aromas and removing undesirable volatile components from food liquids. It is operated at low temperatures with effective vapor/liquid mixing and high mass transfer rates, without heat damaging the quality of heat-sensitive products. The SCSC unit consists of a column, about 1 m diameter and 3 m high, with alternating stationary and rotating truncated cones, which act as contacting stages for liquid and vapors flowing countercurrently (Figure 9.12). The liquid is fed at the top of the column and live steam is injected at the bottom, acting as a stripping medium. The volatile components are condensed in refrigerated condensers at the top, and the stripped product is obtained at the bottom (Casimir and Craig, 1990).

The capacity of a standard (1m x 4m) SCSC unit is 10 000 L/h, and the residence time is less than 1 min. The moving cones are rotated at 250-550 RPM (Saravacos and Kostaropoulos, 2002).

The SCSC unit can be used for the removal of undesirable volatiles from fruit juices, e.g., sulfur dioxide from grape juice, and it is suited for the preparation of alcohol-free (nonalcoholic) wines, which contain less than 0.5 % ethanol. The SCSP de-alcoholization process involves first the recovery of the volatile aromas (esters etc) in a low-temperature column ($T < 30^{\circ}\text{C}$), and second the stripping of ethanol at a higher temperature (about 35°C), resulting in a stripped wine. The

volatile aromas, recovered in the first stripping are returned to the stripped wine, obtaining a full-flavored nonalcoholic product.

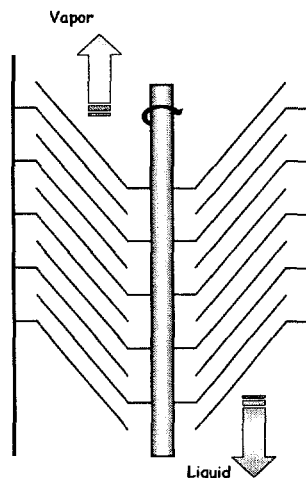


Figure 9.12 Principle of spinning cone stripping column.

III. EXTRACTION

Extraction processes include solvent extraction of liquids or solids, and leaching of solids.

The design of extraction is similar to the procedures used in distillation. Empirical correlations are used for liquid/liquid equilibria, and experimental equilibrium (solubility) data are essential for solid/liquid systems. Approximate methods are used for the estimation of equilibrium stages, and the extraction/leaching equipment is more specialized than the generalized distillation systems.

1. Phase Equilibria

Liquid/liquid extraction is based on the thermodynamic equilibrium between two partially soluble phases, the extract and the raffinate, between which the solute is distributed (Prausnitz et al., 1999). The equilibrium is expressed by the empirical equation:

$$Y = K X \quad (9-1)$$

The concentrations of the solute in the extract Y and raffinate X phases are usually expressed as mass fractions of the mixture. The partition or distribution coefficient K is analogous to the volatility of vapor/liquid equilibria, and it is related to the activity coefficients of the solute in the raffinate γ_r , and the extract γ_e :

$$K = \gamma_r / \gamma_e \quad (9-2)$$

The activity coefficients of various liquid systems are correlated in various empirical relations, such as the Wilson, the UNIQUAC, and the UNIFAC equations (Reid et al., 1987). Liquid/liquid mixtures, used in solvent extraction, are non-ideal systems, facilitating the separation in a relatively small number of stages.

The liquid/liquid equilibrium data are presented in 3 types of diagrams: the right-triangular, the modified McCabe-Thiele, and the Ponchon-Savarit diagrams (Walas, 1985; Saravacos and Kostaropoulos, 2002).

The orthogonal triangle diagram shows the liquid/liquid system with the solvent S , the residue A , and the solute B at the three corners. Normally, the solvent and residue are partially soluble, and the equilibrium line is represented by a curve within the triangle.

The modified McCabe-Thiele diagram for liquid/liquid extraction uses as coordinates the mass fractions of the solute B , in the extract Y and the raffinate X , defined by the equations:

$$Y = B / (A + B) \text{ extract phase} \quad (9-3)$$

$$X = B / (A + B) \text{ raffinate phase}$$

where A is the amount of the residue.

The two phases (extract and raffinate) have a common equilibrium point, called the "plait" point.

The Ponchon-Savarit diagram considers the amount of solvent S in the system, in addition to components A and B . The role of the solvent in extraction is analogous to the enthalpy (heat content) in distillation processes.

The ordinates of the Ponchon-Savarit diagram for liquid/liquid equilibria are:

$$Z = S / (A + B) \text{ for the extract} \quad (9-4)$$

$$z = S / (A + B) \text{ for the raffinate}$$

Equilibrium ("tie") lines connect the solvent V and the raffinate L phases.

Solid/liquid equilibria are determined experimentally, assuming that the solute B is completely soluble in the solvent S , and that it is not sorbed on the inert solid. Under these conditions, equilibrium is expressed by the simple equation:

$$Y_e = X \quad (9-5)$$

where Y_e is the mass fraction of the solute B in the extract, and X is the concentration of solute in the liquid remaining in the pores of the solids, after reaching equilibrium.

The solubility of solids in liquids is discussed by Reid et al. (1987).

The solid/liquid equilibrium data can be plotted on modified McCabe-Thiele and Ponchon-Savarit diagrams, in a similar manner with the liquid/liquid equilibria. In the modified McCabe-Thiele, the coordinates are the mass fractions of the solute B in the extract Y and the residue X , respectively, as defined by Equations 9-3.

The ordinates of the modified Ponchon-Savarit diagram for solid/liquid equilibria are the concentrations of the inert solid A in the solid (underflow) and in the extract (overflow), defined by the equations:

$$z = A / (B + S) \text{ underflow}$$

$$Z = A / (B + S) \text{ overflow} \quad (9-6)$$

where A , B , S are the amounts (kg) of inert solid, solute, and solvent, respectively. For the normal extraction case of an inert solid that is completely insoluble in the solvent, the overflow line coincides with the X -axis, i.e., $Z=0$. The equilibrium "tie" lines are vertical lines, which facilitate the graphical construction of the equilibrium stages between the equilibrium and the operating lines.

2. Equilibrium Stages

Due to equilibrium limitations, more than one equilibrium contact stage is needed in an extraction separation process, similar to distillation. While in liquid/liquid extraction the separation stages can be arranged as a continuous column, special contact equipment is used in solid/liquid extraction and leaching, such as batteries of static beds, operated semi-continuously.

The number of theoretical stages in liquid/liquid (L/L) extraction is determined using the modified McCabe-Thiele or Ponchon-Savarit diagrams. The equilibrium line is plotted, using literature or experimental data, and the operating line is constructed from material balances of the given system, in analogy to distillation. In extraction, the solvent V and raffinate L flows are taken as mass flow rates (kg/h) and the concentrations (Y , X) as mass fractions. In dilute (low-concentration) solutions, straight equilibrium and operating lines facilitate the graphical construction. The equilibrium line ($Y = K X$) is plotted from the origin of the coordinates with a slope of K . The operating line representing the overall material balance in a countercurrent column takes the form of a straight line with a slope of (L/V):

$$Y_1 - Y_{n+1} = (L/V) (X_0 - X_n) \quad (9-7)$$

where (X_0 , X_n) are the solute concentrations in the raffinate, and (Y_1 , Y_{n+1}) are the solute concentrations of the extract at the entrance (stage 1) and exit (stage n) of the column, respectively.

In multistage separations of dilute solutions, the number of equilibrium stages N can be determined analytically by the Kremser equation (Perry and Green, 1997):

$$N = \log[R(1-E) + E] / \log(E) \quad (9-8)$$

where $E = (K V) / L$, extraction factor,
and $R = (X_n - Y_{n+1} / K) / (X_0 - Y_1 / K)$, extraction ratio

The L/L extractors have lower efficiencies (20-30%) than the distillation columns, because of difficulties in mixing the liquid phases, i.e., inefficient mass transfer. The efficiency can be improved by thorough mixing, using agitation or vibration.

In solid/liquid extraction (leaching), a series (battery) of vessels can be considered a continuous countercurrent equilibrium process, similar in operation to a countercurrent liquid/liquid column. The extract V flows counter-currently to the liquid (usually water) residue L , while the inert solids remain fixed in the beds. The solvent gradually removes the solute from the liquid residue, reaching equilibrium at each stage (sufficient residence time, mechanical mixing).

The number of equilibrium stages can be determined graphically, using the modified McCabe-Thiele or Ponchon-Savarit diagrams. Graphical construction of the stages is facilitated by the assumption that the equilibrium line coincides with the diagonal line ($Y = X$).

For low concentrations of the solute in the inert solid and in the extract, a straight operating line is obtained, assuming constant flows of liquid underflow X and extract V . The number of equilibrium stages N can be calculated from the Kremser equation (9-8). The efficiency of the solid/liquid extractors is very high, about 90-95%, due to thorough mixing and long residence time in each stage.

3. Mass Transfer Considerations

Solvent extraction is generally controlled by mass transfer of the solutes from the material to the transfer interface and then to the solvent. Mass transfer within liquid and solid materials is usually assumed to take place through molecular diffusion, and the transport rate is expressed by the effective diffusivity D , which is an overall transport coefficient, based on the diffusion (Fick) equation (Saravacos and Maroulis, 2001).

Diffusivity data of solutes in liquids are presented by Cussler (1997) and Reid et al. (1987). Diffusivity D in liquids is related to the molecular (particle) size of the solute and the viscosity of the mixture (Stokes-Einstein equation), or the molecular solute/solvent interactions (Wilke-Chang equation). Typical values of D of solutes in dilute water solutions (25°C) are: sodium chloride, $12 \times 10^{-10} \text{ m}^2/\text{s}$; sucrose, $5 \times 10^{-10} \text{ m}^2/\text{s}$; and lactalbumin, $0.7 \times 10^{-10} \text{ m}^2/\text{s}$.

Diffusivity data in solids are more variable, depending strongly on the microscopic and macroscopic structure (homogeneous, porous, fibrous, etc.) of the material. Prediction and correlation of D in solids is difficult, and experimental data are required for each material at given conditions (Saravacos and Maroulis, 2001). Data on D of importance to leaching are given by Schwartzberg and Chao (1982) and Schwartzberg (1987), e.g., oil (soybean flakes)/hexane, $1 \times 10^{-10} \text{ m}^2/\text{s}$, and coffee solubles (coffee beans)/water, $1 \times 10^{-10} \text{ m}^2/\text{s}$. The D of solutes increases when the porosity of the solid is increased. Large molecules, such as lipids and proteins have smaller diffusivities, compared to, e.g., sugars.

The transport rate (diffusivity) of the solutes can be increased considerably by various mechanical and hydrothermal pre-treatments of the solid foods, e.g., slicing, flaking and steam injection. Slicing of beets and fruits, and flaking reduce the thickness (diffusion path) of the material, without serious damage of the cells, which would release cell components into the solution and make difficult the solid/liquid separation. Heating by steam or water modifies the cellular structure, increasing the diffusivity in oil extraction.

Recovery of oil from oilseeds containing above 22% oil is accomplished by mechanical expression, followed by leaching with an organic solvent, such as hexane, chlorinated hydrocarbons, and alcohols (Hui, 1996). Mechanical pressing can remove up to 90% of the oil in the seeds. Direct solvent leaching is applied to oilseeds containing less than 22% oil, e.g., soybeans (Aguilera and Stanley, 1999).

Sugar beets are cut into long, thin slices and heated to 50-60°C to denature the proteins and increase the diffusivity of sucrose, without leaching of non-sugar components.

Water extraction of soluble solids from roasted coffee beans is facilitated by the high porosity of the beans, developed during thermal treatment (roasting). In the processing of decaffeinated soluble coffee, the diffusion of caffeine in water is controlling the process.

4. Extraction Equipment

Multistage countercurrent extraction in liquid/liquid (*L/L*) systems is performed in extraction columns, which are similar to the stripping section of a standard distillation column. Equipment used in liquid/liquid extraction is discussed by Walas (1988), Schweitzer (1988), Sinnott (1996), and Perry and Green (1997). Most of this *L/L* extraction equipment is used in the chemical and petroleum industries, with some applications to food processing. The columns are analyzed either as plate (tray) or packed tower systems.

Industrial *L/L* solvent extraction columns include the sieve tray, the rotating disc (RTC), the Oldshue (paddle agitation), the Scheidel (packing and agitation), the pulsed columns, the Graesser rotary raining cup extractor, and the centrifugal extractor (Walas, 1988; Saravacos and Kostaropoulos, 2002).

Leaching of solutes from solids is usually carried out in static (fixed) beds of particulate material, prepared to the appropriate size and physicochemical condition to facilitate mass (diffusion) transport of the solute. The bed of material, resting on a false bottom, is enclosed in vertical cylindrical vessels, which can stand high pressure operation of volatile solvents. The solvent is fed to a distributor at the top, while the extract can be recirculated through the solids, before it is removed from the system.

The operation of the static-bed semi-continuous systems is described by Schwartzberg (1997) and Perry and Green (1997). In the leaching of sugar beets with water, 10-14 extraction cells are used, each with a volume of 4-12m³ and height to diameter ratio of 1.5. Leaching time is 60-100 min and 110 kg of sugar solution 12-16°Brix is produced per 100 kg beets. In the processing of instant coffee, 5-8 extraction columns (cylindrical vessels 0.76 m diameter and 5.3 m height) are used with cycle time ½ to 1 h. Hot water at 175°C is used in ratios 7/2 to 5/1 to solids, and the extracts contain 25-30% soluble solids.

Figure 9.13 shows a semi-continuous battery of extraction columns (cells) for soluble coffee (Schwartzberg, 1997). Six extraction columns (percolators) are filled with ground coffee of appropriate size to facilitate diffusion. The columns are operated under pressure (about 20 bar), and they are equipped with special valves for filling and removing solid particles. Hot water at about 175°C is fed to column (1), which contains the most-spent coffee grounds, and the lean extract is

reheated and fed to column (2) and so on. Finally, rich extract is removed from the last column (6), which contains unextracted fresh grounds. The final extract, containing about 25-30% soluble solids, is concentrated in an evaporator and spray-dried into granular soluble coffee. When exhausted, column (1) is emptied and refilled with fresh grounds, while the hot water is fed to column (2), and the extract from column (6) is fed to column (1). The cycle is repeated by emptying and refilling column (2), and so on.

Special leaching equipment for solids, operated continuously or semi-continuously includes the following:

- The Bollmann bucket elevator with perforated buckets carrying each about 40 kg of particulate material in a solvent bath, contained in a closed vessel.
- The Hildebrand extractor, in which the solids are transported with screw conveyors through three sections in countercurrent flow with the solvent/extract.
- The Bonotto multi-tray tower extractor, in which the trays rotate, while the solid particles are scraped and discharged from tray to tray.
- The Rotocel extractor, which consists of about 18 wedge-shaped cells in a rotating shell, enclosed in a stationary tank. Fresh solvent is charged to the last cell and the drained solutions are pumped countercurrently to each cell in series. The Rotocel is used in the leaching of sugar beets and oilseeds.
- The Screw extractor is used for leaching sugar beets, where the beet pieces move countercurrently at a slope against a hot water solution.

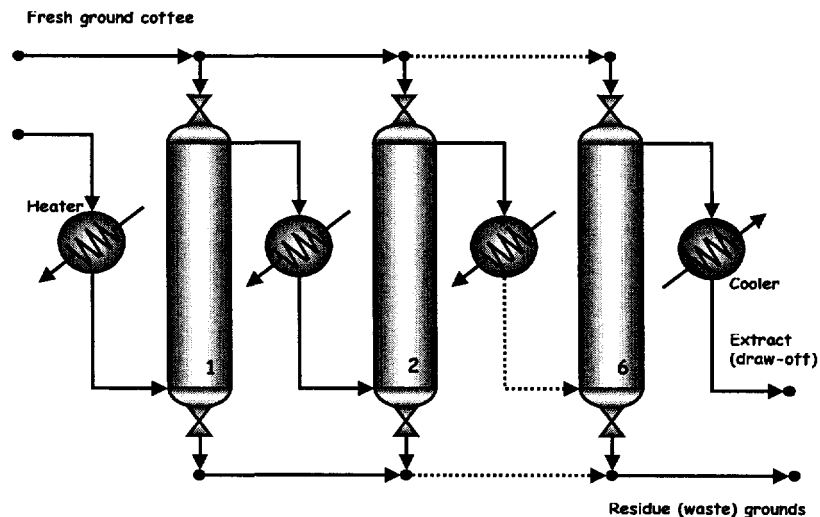


Figure 9.13 Semi-continuous battery of extraction columns for soluble coffee.

5. Design of Crosscurrent Flow Solid Extraction

In this section a design procedure for a three-stage crosscurrent-flow solids-extraction (leaching) system is described and applied to oil recovery from soybean meal.

a. Process Description

Solids extractors are often used for recovery of oil from vegetable seeds (Treybal, 1981). The solids are extracted in a series of vessels and are pushed from one to the next in the cascade by paddles, while the solvent flows countercurrent or crosscurrent (Figure 9.14).

It is assumed that the system consists of

- two phases: the overflow stream R and the underflow stream F
- and three components: the solvent S , the solids A and the solute B

The overflow stream is characterized by the following variables (Equation 9-6):

R (kg S +kg B)/h Solution flow rate
 X (kg B)/(kg S +kg B) Solute concentration

The underflow stream is characterized by the following variables:

F (kg S +kg B)/h Solution flow rate
 Y (kg B)/(kg S +kg B) Solute concentration
 z (kg A)/(kg S +kg B) Solids concentration

It is assumed that the inert solids in the overflow are completely insoluble in the solvent ($z=0$).

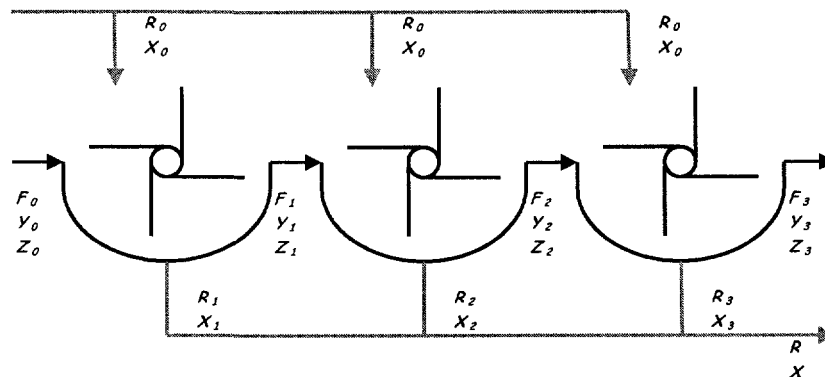


Figure 9.14 Diagram of a three-stage crosscurrent-flow solids-leaching system.

b. Process Model

A mathematical model for the process presented in Figure 9.14 is summarized in Table 9.27. Equations (E01) to (E05) refer to first stage: Equations (E01), (E02), and (E03) are the material balances for solids, solute and solution (solvent plus solute), respectively. Equation (E04) calculates the solution retained in the solids versus the composition of the solution. Equation (E05) suggests that, at equilibrium, the solute concentration in overflow and underflow solution should be equal.

Equations (E06) to (E10) refer to second stage, while Equations (E11) to (E15) refer to third stage. Equations (E16) and (E17) refer to overflow mixing; that is, (E016) is the solution material balance and (E17) the solute material balance.

Equations (E18) and (E19) calculate the required volume for each stage to guarantee the appropriate residence time to obtain equilibrium. Equation (E20) calculates the required electrical power for the paddle drive.

Twenty-five variables presented in Table 9.28 are involved in the model of twenty equations presented in Table 9.27, which means that five degrees of freedom are available, and if we consider a typical problem with four specifications presented in Table 9.29 one design variable is available for process optimization. Table 9.29 suggests a selection of design variable and the corresponding solution algorithm. The total annualized cost *TAC* presented in Table 9.30 is used as objective function in process optimization.

Table 9.27 Process Model

| | |
|--|-------|
| 1 st stage | |
| $F_0 Z_0 = F_1 Z_1$ | (E01) |
| $F_0 Y_0 + R_0 X_0 = F_1 Y_1 + R_1 X_1$ | (E02) |
| $F_0 + R_0 = F_1 + R_1$ | (E03) |
| $z_1 = a X_1^2 + b X_1 + c$ | (E04) |
| $Y_1 = X_1$ | (E05) |
| 2 nd stage | |
| $F_1 Z_1 = F_2 Z_2$ | (E06) |
| $F_1 Y_1 + R_1 X_1 = F_2 Y_2 + R_2 X_2$ | (E07) |
| $F_1 + R_1 = F_2 + R_2$ | (E08) |
| $z_2 = a X_2^2 + b X_2 + c$ | (E09) |
| $Y_2 = X_2$ | (E10) |
| 3 rd stage | |
| $F_2 Z_2 = F_3 Z_3$ | (E11) |
| $F_2 Y_2 + R_2 X_2 = F_3 Y_3 + R_3 X_3$ | (E12) |
| $F_2 + R_2 = F_3 + R_3$ | (E13) |
| $z_3 = a X_3^2 + b X_3 + c$ | (E14) |
| $Y_3 = X_3$ | (E15) |
| Overflow mix | |
| $R = R_1 + R_2 + R_3$ | (E16) |
| $R X = R_1 X_1 + R_2 X_2 + R_3 X_3$ | (E17) |
| Extractor size | |
| $Q = \max_{j=1,2,3} [R_j + F_j (1+z_j)]$ | (E18) |
| $V = f_o (Q/\rho) t$ | (E19) |
| Electricity | |
| $E = f_p F_0 z_0$ | (E20) |

Table 9.28 Process Variables and Technical Data**Process variables**

| | | |
|-------|-----------------|--|
| E | kW | Electric power for paddle drive |
| F_0 | (kgS+kgB)/h | Solution input flow rate at underflow stream |
| F_1 | (kgS+kgB)/h | Solution flow rate at underflow stream (1 st stage) |
| F_2 | (kgS+kgB)/h | Solution flow rate at underflow stream (2 nd stage) |
| F_3 | (kgS+kgB)/h | Solution flow rate at underflow stream (3 rd stage) |
| Q | ton/h | Maximum material flow from and to extractor vessel |
| R | (kgS+kgB)/h | Solution flow rate of overflow stream at output |
| R_0 | (kgS+kgB)/h | Solution flow rate of fresh solvent for each stage |
| R_1 | (kgS+kgB)/h | Solution flow rate at overflow stream (1 st stage) |
| R_2 | (kgS+kgB)/h | Solution flow rate at overflow stream (2 nd stage) |
| R_3 | (kgS+kgB)/h | Solution flow rate at overflow stream (3 rd stage) |
| V | m ³ | Extractor vessel volume |
| X | (kgB)/(kgS+kgB) | Solute concentration of overflow stream at output |
| X_0 | (kgB)/(kgS+kgB) | Solute concentration of fresh solvent for each stage |
| X_1 | (kgB)/(kgS+kgB) | Solute concentration at overflow stream (1 st stage) |
| X_2 | (kgB)/(kgS+kgB) | Solute concentration at overflow stream (2 nd stage) |
| X_3 | (kgB)/(kgS+kgB) | Solute concentration at overflow stream (3 rd stage) |
| Y_0 | (kgB)/(kgS+kgB) | Solute concentration at input underflow stream |
| Y_1 | (kgB)/(kgS+kgB) | Solute concentration at underflow stream (1 st stage) |
| Y_2 | (kgB)/(kgS+kgB) | Solute concentration at underflow stream (2 nd stage) |
| Y_3 | (kgB)/(kgS+kgB) | Solute concentration at underflow stream (3 rd stage) |
| z_0 | (kgA)/(kgS+kgB) | Solids concentration at input underflow stream |
| z_1 | (kgA)/(kgS+kgB) | Solids concentration at underflow stream (1 st stage) |
| z_2 | (kgA)/(kgS+kgB) | Solids concentration at underflow stream (2 nd stage) |
| z_3 | (kgA)/(kgS+kgB) | Solids concentration at underflow stream (3 rd stage) |

Technical data

| | | |
|--------|-------------------|---|
| f_o | – | Overdesign coefficient for extractor vessel |
| f_p | – | Coefficient in paddle power equation |
| a | – | Coefficient in solid/liquid equilibrium |
| b | – | Coefficient in solid/liquid equilibrium |
| c | – | Coefficient in solid/liquid equilibrium |
| t | (h) | Required residence time to obtain equilibrium |
| ρ | kg/m ³ | Pulp density |

Table 9.29 Solution of a Typical Design Problem**Process Specifications**

| | | |
|-------|-----------------|--|
| F_0 | (kgS+kgB)/h | Solution input flow rate in underflow stream |
| Y_0 | (kgB)/(kgS+kgB) | Solute concentration at input underflow stream |
| z_0 | (kgA)/(kgS+kgB) | Solids concentration at input underflow stream |
| X_0 | (kgB)/(kgS+kgB) | Solute concentration of fresh solvent for each stage |

Degrees-of-Freedom Analysis

| | |
|--------------------|----|
| Process variables | 25 |
| Process equations | 20 |
| Degrees of freedom | 5 |
| Specifications | 4 |
| Design variables | 1 |

Design Variables

| | | |
|-------|-------------|--|
| R_0 | (kgS+kgB)/h | Solution flow rate of fresh solvent for each stage |
|-------|-------------|--|

Solution Algorithm

| | | |
|--------------|---|-------|
| (E01), (E02) | → | X_1 |
| (E05) | → | Y_1 |
| (E04) | → | Z_1 |
| (E01) | → | F_1 |
| (E03) | → | R_1 |
| (E06), (E07) | → | X_2 |
| (E10) | → | Y_2 |
| (E09) | → | Z_2 |
| (E06) | → | F_2 |
| (E08) | → | R_2 |
| (E11), (E12) | → | X_2 |
| (E15) | → | Y_2 |
| (E14) | → | Z_2 |
| (E11) | → | F_2 |
| (E13) | → | R_2 |
| (E16) | → | R |
| (E17) | → | X |
| (E18) | → | Q |
| (E19) | → | V |
| (E20) | → | E_p |

Table 9.30 Cost Analysis**Equipment cost**

$$C_{eq} = 3C_v V^{n_v} + 3C_p E_p^{n_p}$$

Annual operating cost

$$C_{op} = (C_s R_o + C_e E_p) t_y$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op}$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1}$$

Cost Data

| | | |
|-------|-------------------|---------------------------------|
| C_v | \$/m ³ | Extractor vessel unit cost |
| n_v | – | Extractor vessel scaling factor |
| C_p | \$/kW | Paddle unit cost |
| n_p | – | Paddle scaling factor |
| C_s | \$/kWh | Cost of solvent |
| C_e | \$/kWh | Cost of electricity |
| t_y | h/yr | Annual operating time |
| i_r | – | Interest rate |
| l_f | yr | Lifetime |

c. Application to Oil Recovery from Soybean Meal

The previous design procedure is applied to oil recovery from soybean meal (Toledo, 1991). Typical process specifications for the design problem are presented in Table 9.31. The necessary technical and cost data are also summarized in Table 9.31 and in Figure 9.15. The process design results are presented in Table 9.32 and in Figures 9.16 through 9.21.

Figure 9.16 presents the operating points for each stage at the underflow curve, while Figure 9.17 analyzes the effect of design variable (solution flow rate) on oil recovery, separately for each stage. It must be noted that almost all the oil could be recovered using 3 stages and large amount of fresh solvent. At the selected fresh solvent flow rate more than 75% recovery can be obtained. The effect of the design variable on some characteristic concentrations at output streams is presented in Figure 9.18, while Figure 9.19 presents the effect on the process economics. Due to the large value of solvent, the total annualized cost is close to the annual operating cost.

Finally, Figures 9.20 and 9.21 reveal the material transport due to the extraction (leaching) process. The performance of the 1st stage (usually called the mixing stage) is a little different than the others. Due to the first contact of solids with the solvent, a large amount of the solvent is retained in the solids. During the next stages the solute is transferred from the solid phase to the solvent phase.

Table 9.31 Process Specifications and Process Technical and Cost Data**Process Specifications**

| | | |
|-------|--------------------|--|
| F_0 | = 0.20 ton(S+B)/h | Solution input flow rate at underflow stream |
| Y_0 | = 1.00 kgB/kg(S+B) | Solute concentration at input underflow stream |
| z_0 | = 5.00 kgA/kg(S+B) | Solids concentration at input underflow stream |
| X_0 | = 0.00 kgB/kg(S+B) | Solute concentration of fresh solvent for each stage |

Technical Data

| | | |
|--------|--------------------------|---|
| f_o | = 1.50 | Overdesign coefficient for extractor vessel |
| f_p | = 1.00 | Coefficient in paddle power equation |
| a | = -0.40 | Coefficient in solid/liquid equilibrium |
| b | = -0.30 | Coefficient in solid/liquid equilibrium |
| c | = 2.00 | Coefficient in solid/liquid equilibrium |
| t | = 0.50 h | Required residence time to obtain equilibrium |
| ρ | = 1000 kg/m ³ | Pulp density |

Cost Data

| | | |
|-------|------------------------|---------------------------------|
| C_v | = 5 k\$/m ³ | Extractor vessel unit cost |
| n_v | = 1 | Extractor vessel scaling factor |
| C_p | = 1 k\$/kW | Puddle unit cost |
| n_p | = 1 | Puddle scaling factor |
| C_s | = 0.05 \$/kg | Cost of solvent |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| t_y | = 5000 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

Design Variable

| | | |
|-------|-------------------|--|
| R_0 | = 0.50 ton(S+B)/h | Solution flow rate of fresh solvent for each stage |
|-------|-------------------|--|

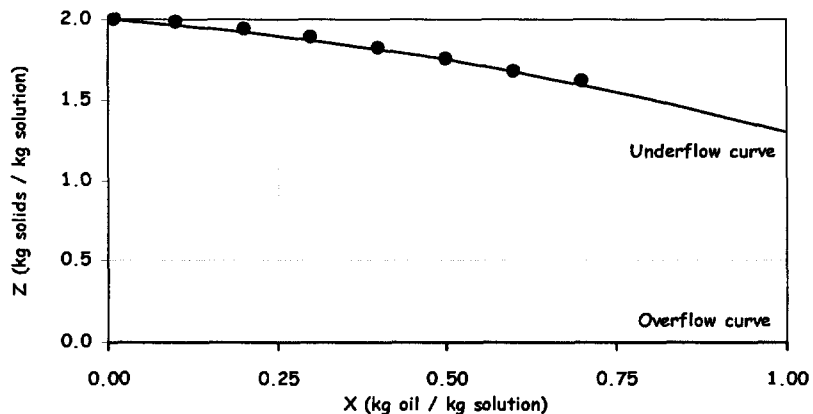
**Figure 9.15** Underflow equilibrium data (Ponchon-Savarit) for the soybean/hexane system.

Table 9.32 Process Design Results**Overflow**

| | | |
|-------|------------------------|--|
| R_0 | = 0.50 ton(S+B)/h | Solution flow rate of fresh solvent for each stage |
| R_1 | = 0.17 ton(S+B)/h | Solution flow rate at 1 st stage |
| R_2 | = 0.51 ton(S+B)/h | 2 nd stage |
| R_3 | = 0.51 ton(S+B)/h | 3 rd stage |
| R | = 1.19 ton(S+B)/h | output |
| X_0 | = 0.00 (kgC)/(kgS+kgB) | Solute concentration of fresh solvent |
| X_1 | = 0.29 (kgC)/(kgS+kgB) | Solute concentration at 1 st stage |
| X_2 | = 0.22 (kgC)/(kgS+kgB) | 2 nd stage |
| X_3 | = 0.11 (kgC)/(kgS+kgB) | 3 rd stage |
| X | = 0.18 (kgC)/(kgS+kgB) | output |

Underflow

| | | |
|-------|------------------------|---|
| F_0 | = 0.20 ton(S+B)/h | Solution flow rate at input |
| F_1 | = 0.53 ton(S+B)/h | 1 st stage |
| F_2 | = 0.52 ton(S+B)/h | 2 nd stage |
| F_3 | = 0.51 ton(S+B)/h | 3 rd stage |
| Y_0 | = 1.00 (kgB)/(kgS+kgB) | Solute concentration at input underflow |
| Y_1 | = 0.29 (kgB)/(kgS+kgB) | 1 st stage |
| Y_2 | = 0.22 (kgB)/(kgS+kgB) | 2 nd stage |
| Y_3 | = 0.11 (kgB)/(kgS+kgB) | 3 rd stage |
| z_0 | = 5.00 (kgA)/(kgS+kgB) | Solids concentration at input underflow |
| z_1 | = 1.88 (kgA)/(kgS+kgB) | 1 st stage |
| z_2 | = 1.92 (kgA)/(kgS+kgB) | 2 nd stage |
| z_3 | = 1.96 (kgA)/(kgS+kgB) | 3 rd stage |

Equipment

| | | |
|-------|-----------------------|---------------------------------|
| E_p | = 1 kW | Electric power for paddle drive |
| V | = 1.52 m ³ | Extractor vessel volume |

Cost

| | | |
|----------|--------------|------------------|
| C_{eq} | = 26 k\$ | Equipment |
| C_{op} | = 376 k\$/yr | Operating |
| TAC | = 379 k\$/yr | Total annualized |

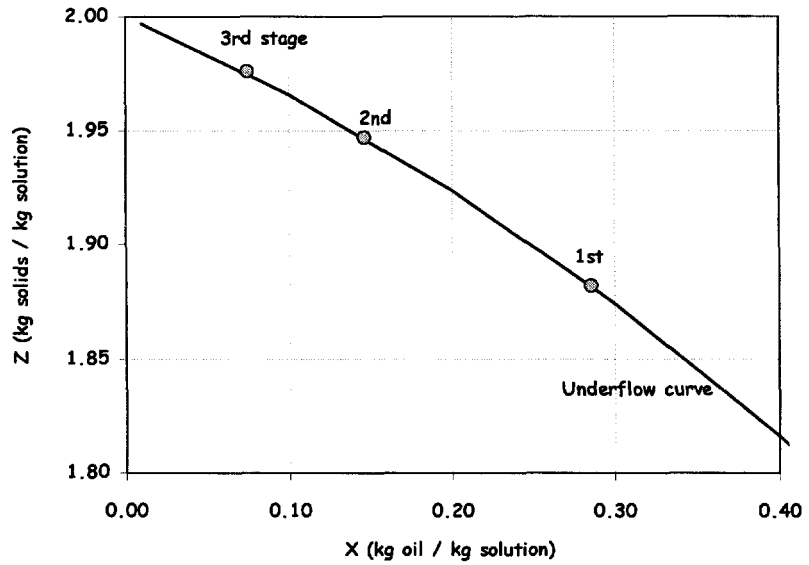


Figure 9.16 Operating points for each stage at the underflow curve.

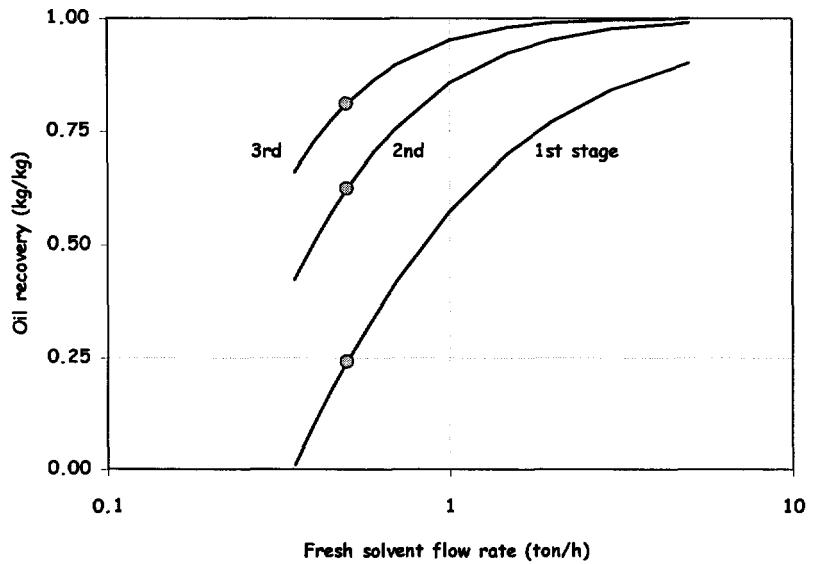


Figure 9.17 Oil recovery versus solvent flow rate for 1, 2 and 3 stages.

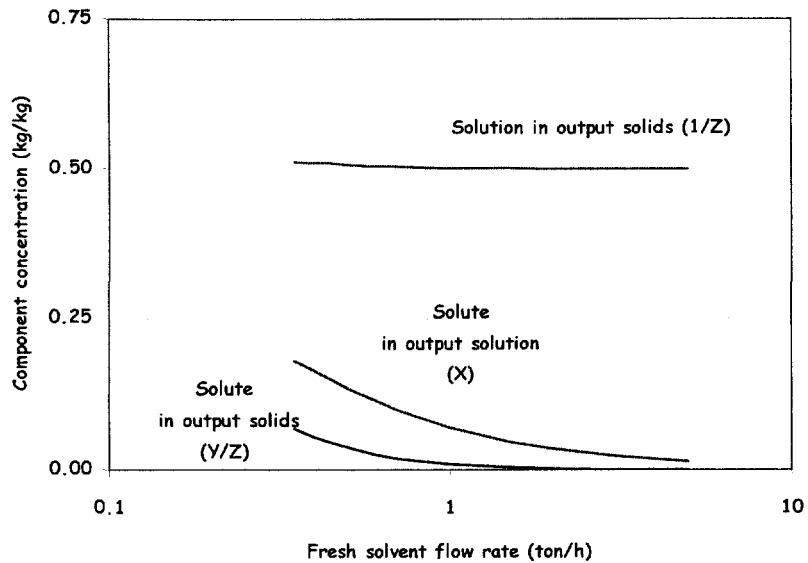


Figure 9.18 Effect of solvent flow rate on the output concentrations.

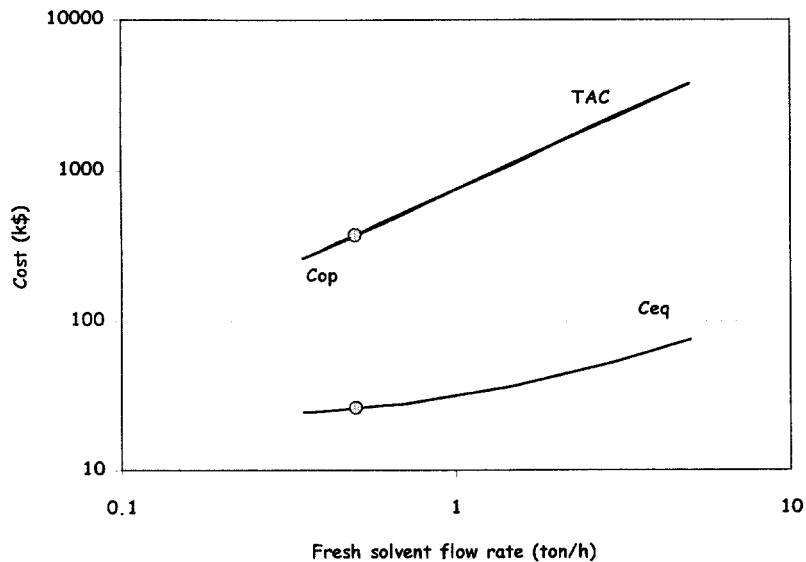


Figure 9.19 Effect of solvent flow rate on the cost.

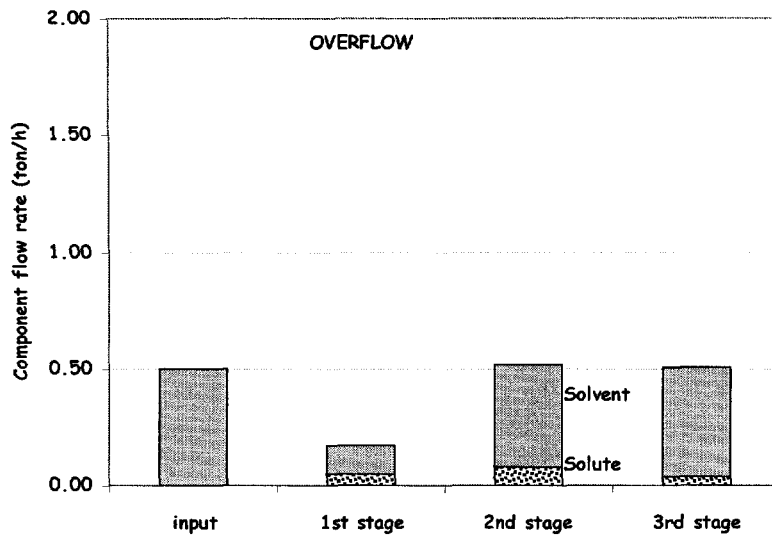


Figure 9.20 Diagram of a three-stage crosscurrent-flow solids-leaching system.

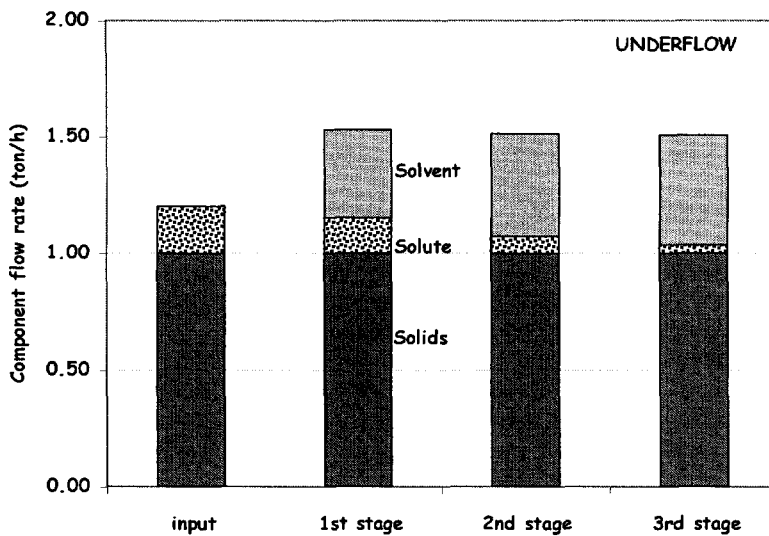


Figure 9.21 Diagram of a three-stage crosscurrent-flow solids-leaching system.

NOMENCLATURE

| | |
|-----------|---|
| a | Coefficient in solid/liquid equilibrium |
| A_B | Bottom reboiler transfer area |
| A_{BC} | Bottom product cooler transfer area |
| A_D | Overhead condenser transfer area |
| A_{DC} | Overhead product cooler transfer area |
| A_F | Feed heater transfer area |
| B | Bottom product or residual flow rate |
| b | Coefficient in solid/liquid equilibrium |
| c | Coefficient in solid/liquid equilibrium |
| C | Cost |
| D | Top product flow rate |
| D | Tower diameter at rectifying section |
| D' | Tower diameter at stripping section |
| E | Electric power for paddle drive |
| E_o | Overall plate efficiency |
| F | Feed flow rate |
| F_o | Solution input flow rate at underflow stream |
| F_1 | Solution flow rate at underflow stream (1 st stage) |
| F_2 | Solution flow rate at underflow stream (2 nd stage) |
| F_3 | Solution flow rate at underflow stream (3 rd stage) |
| f_o | Overdesign coefficient for extractor vessel |
| f_p | Coefficient in paddle power equation |
| H | Column height |
| H_o | Space at the ends of the column for vapor disengagement and liquid sump |
| H_1 | Plate spacing |
| H_B | Enthalpy of bottom liquid before the bottom cooler |
| $H_{B'}$ | Enthalpy of vapor after the bottom reboiler |
| H_{Bo} | Enthalpy of bottom liquid after the bottom cooler |
| H_D | Enthalpy of overhead liquid before the overhead cooler |
| $H_{D'}$ | Enthalpy of vapor before the overhead condenser |
| H_{Do} | Enthalpy of overhead liquid after the overhead cooler |
| H_F | Enthalpy of vapor-liquid in equilibrium after the feed heater |
| H_{Fo} | Enthalpy of liquid before the feed heater |
| K | Partition coefficient |
| L | Liquid at rectifying section flow rate |
| L' | Liquid at stripping section flow rate |
| N | Total number of theoretical stages |
| N_{act} | Actual number of plates |
| N_B | Number of stages at rectifying section |
| N_D | Number of stages at rectifying section |
| N_{min} | Minimum number of theoretical stages |
| P | Operating pressure |
| q | Feed quality |
| Q | Maximum material flow from and to extractor vessel |
| Q_B | Bottom reboiler thermal load |

| | |
|-----------|---|
| Q_{BC} | Bottom product cooler thermal load |
| Q_D | Overhead condenser thermal load |
| Q_{DC} | Overhead product cooler thermal load |
| Q_F | Feed heater thermal load |
| R | Reflux ratio |
| R | Solution flow rate of overflow stream at output |
| R_0 | Solution flow rate of fresh solvent for each stage |
| R_1 | Solution flow rate at overflow stream (1 st stage) |
| R_2 | Solution flow rate at overflow stream (2 nd stage) |
| R_3 | Solution flow rate at overflow stream (3 rd stage) |
| R_{min} | Minimum reflux ratio |
| S_B | Steam at bottom reboiler flow rate |
| SF | Separation factor |
| S_F | Steam at feed heater flow rate |
| t | Required residence time to obtain equilibrium |
| t | Time |
| T_B' | Bottom dew point temperature |
| T_D' | Top dew point temperature |
| T_F' | Feed dew point temperature |
| T_B | Bottom bubble point temperature |
| T_D | Top bubble point temperature |
| T_F | Feed bubble point temperature |
| T_F^* | Feed temperature |
| T_o | Ambient temperature |
| T_S | Heating steam temperature |
| T_{w1} | Cooling water inlet temperature |
| T_{w2} | Cooling water outlet temperature |
| U | Overall heat transfer coefficient |
| u_f' | Flooding velocity at the stripping section |
| u_f | Flooding velocity at the rectifying section |
| V | Extractor vessel volume |
| V | Vapor at rectifying section flow rate |
| V' | Vapor at stripping section flow rate |
| W_{BC} | Water at bottom product cooler flow rate |
| W_D | Water at overhead condenser flow rate |
| W_{DC} | Water at overhead product cooler flow rate |
| X | Molar fraction of a component in the liquid phase |
| X | Solute concentration of overflow stream at output |
| X_0 | Solute concentration of fresh solvent for each stage |
| X_1 | Solute concentration at overflow stream (1 st stage) |
| X_2 | Solute concentration at overflow stream (2 nd stage) |
| X_3 | Solute concentration at overflow stream (3 rd stage) |
| X_B | Bottom product molar fraction |
| X_D | Top product molar fraction |
| X_F | Feed molar fraction |
| X_{FL} | Feed molar fraction in liquid phase |
| X_{FV} | Feed molar fraction in vapor phase |
| Y_0 | Solute concentration at input underflow stream |

| | |
|-------|--|
| Y_1 | Solute concentration at underflow stream (1 st stage) |
| Y_2 | Solute concentration at underflow stream (2 nd stage) |
| Y_3 | Solute concentration at underflow stream (3 rd stage) |
| z_0 | Solids concentration at input underflow stream |
| z_1 | Solids concentration at underflow stream (1 st stage) |
| z_2 | Solids concentration at underflow stream (2 nd stage) |
| z_3 | Solids concentration at underflow stream (3 rd stage) |

Greek

| | |
|------------|---|
| α | Design relative volatility |
| α_B | Relative volatility at bottom |
| α_D | Relative volatility at top |
| α_F | Relative volatility at feed |
| γ | Activity coefficient |
| θ | Parameter in Underwood equation |
| ρ | Pulp density |
| ρ_V | Vapor molar density at the rectifying section |
| ρ_V' | Vapor molar density at the stripping section |

Abbreviations

| | |
|-------------|----------------------------------|
| <i>CEP</i> | Chemical engineering plant index |
| <i>e</i> | Capital recovery factor |
| <i>HTST</i> | High temperature short time |
| <i>TAC</i> | Total annualized cost |

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10

Membrane Separation Processes

I. INTRODUCTION

Membrane separations are based on the preferential permeation or rejection of various solute molecules in semi-permeable polymer or inorganic membranes. The driving force for mass transfer through the thin membranes is mechanical pressure. Membrane separations are low-energy operations, and they are less expensive than the conventional thermal separations, i.e., evaporation and distillation. Typical examples of membrane separations are ultrafiltration (recovery of proteins), reverse osmosis (desalination of water), and pervaporation (separation of ethanol).

Membrane separations are classified as shown in Figure 10.1 according to the size of the molecule or the particle being separated from a solution or suspension. Table 10.1 shows the main membrane separation processes which are applied to food-processing industry (Perry and Green, 1997; Cheryan, 1998; Saravacos and Kostaropoulos, 2002).

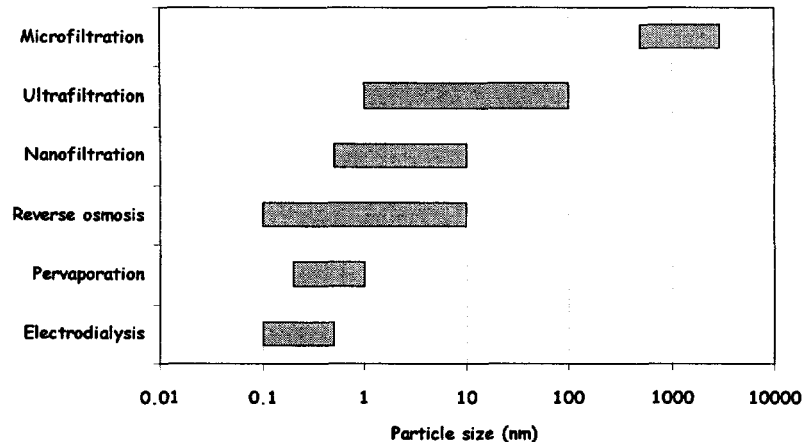


Figure 10.1 The membrane separation spectrum.

Table 10.1 Membrane Separations in Food Systems

| Separation Process | Particle Size | Typical Application |
|--------------------|-----------------------|--------------------------------|
| Microfiltration | 0.5 - 3 μm | Removal of fat, microorganisms |
| Ultrafiltration | 1 - 100 nm | Recovery of proteins |
| Nanofiltration | 0.5 - 10 nm | Whey desalting |
| Reverse osmosis | 0.1 - 10 nm | Water desalting |
| Pervaporation | 0.2 - 1.0 nm | Separation of ethanol |
| Electrodialysis | 0.1 - 0.5 nm | Water desalting |

II. PRINCIPLES OF MEMBRANE SEPARATIONS

1. Mass Transfer Considerations

The fundamental physicochemical and transport properties of food components, related to membrane separations are discussed by Matsuura and Sourirajan (1995).

The solute transfer through membranes depends on the physical structure of the membrane, which is usually a synthetic polymer or a porous ceramic material. In reverse osmosis, the membranes are macroscopically nonporous and homogeneous materials, through which solutes are transported by a molecular (Fickian) mechanism. In ultrafiltration and microfiltration membranes, solute transfer is mainly by viscous flow. In most food applications the concentration of solute molecules is low and the permeability equation is applicable (Saravacos and Maroulis, 2001).

The transport rate of a solute through a membrane of thickness z (m) is given by the permeability equation, which is a combination of the Fick and Henry laws:

$$J = (DS/z) (\Delta P) \quad (10-1)$$

where J ($\text{kg}/\text{m}^2\text{s}$) is the mass flux, ΔP (Pa) is the pressure drop, and (DS) is the permeability of the membrane ($\text{kg}/\text{m s Pa}$), and $DS = D/H$, where D (m^2/s) is the diffusivity of the solute in the membrane, S ($\text{kg}/\text{m}^3\text{Pa}$) is the solubility of the solute in the membrane, and H ($\text{Pa m}^3/\text{kg}$) is the Henry constant. The pressure drop is defined as $\Delta P = (P_f - P_p)$, and P_f, P_p are the feed and product pressures. The flux has the SI units of (m/s), but practical units, such as $\text{L}/\text{m}^2\text{h}$, $\text{gal}/\text{ft}^2\text{day}$ or ft/day are used. The following conversion factors should be used to convert the various flux units: $1 \text{ US gallon}/\text{ft}^2\text{day} = 1.7 \text{ L}/\text{m}^2\text{h} = 0.47 \mu\text{m}/\text{s}$, and $1 \text{ L}/\text{m}^2\text{h} = 0.28 \mu\text{m}/\text{s}$.

Equation (10-1) is difficult to use in practice, since the membrane thickness z , as well as the solubility S and diffusion D constants, are not known accurately. For this reason, the following simplified equations for the flux are used in membrane separations (Cheryan, 1998; Saravacos and Kostaropoulos, 2002):

$$J = A (\Delta P) = (\Delta P)/R_m \quad (10-2)$$

where $A = SD/z$, the overall permeability of the specific membrane, and $R_m = 1/A$, the overall membrane resistance to mass transfer. In membrane separations, the flux J is usually expressed in volumetric, instead of mass units, i.e., $\text{m}^3/\text{m}^2\text{s}$ or m/s (for low flow rates, $\mu\text{m/s}$). Thus, the units of the overall permeability A become $\text{m}/(\text{s Pa})$, and of the units of the overall resistance R_m will be $(\text{s Pa})/\text{m}$.

In reverse osmosis, the flux equation becomes:

$$J = (\Delta P - \Delta \Pi) / R_m \quad (10-3)$$

where $\Delta \Pi = (\Pi_f - \Pi_p)$, and Π_f , Π_p are the osmotic pressures of the feed and product stream, respectively. In addition to membrane, the resistances of fouling and concentration polarization should be considered (Perry and Green, 1997).

In ultrafiltration and microfiltration, the flux of permeate through the porous membrane is given by the Darcy equation:

$$J = K (\Delta P / \eta z) = \Delta P / (\eta r_m) \quad (10-4)$$

where K (kg/m) is the flow (Darcy) permeability, η (Pa s) is the viscosity, and r_m is the flow (Darcy) resistance. It should be noted that $R_m = \eta r_m$. The fouling and polarization resistances should be added to R_m .

The rejection R of a solute in a membrane separation is calculated from the equation:

$$R = 1 - (C_p / C_f) \quad (10-5)$$

where C_p and C_f are the solute concentrations (kg/m^3) in the product and feed, respectively.

2. Membrane Modules

The composition and preparation of separation membranes is discussed by Cheryan (1998). The method of preparation determines the physical structure and the selectivity of the membrane. Polymeric membranes should have a precise pore size distribution and a thin skin on the surface about 0.5-1 μm thick. Two general types of membranes are used commercially, i.e., flat sheets and hollow fibers (capillary tubes).

Commercial polymeric membranes include cellulose acetate, polyamides, polysulfones, polyacrylonitrile, polyethersulfones, and polypropylene. Experimental trials are needed to determine the suitability of a membrane for a particular separation.

Polymeric materials are sensitive to temperature T and pH , and the following temperature limits should be used: Cellulose acetate $T < 30$ $^\circ\text{C}$, polyamides, polysulfones $T < 80$ $^\circ\text{C}$, and ceramic membranes $T < 130$ $^\circ\text{C}$. A narrow pH range is recommended for cellulose acetate (3 - 8), while polysulfones can be operated over a wider range (3 - 11).

The commercial membrane modules must maintain membrane integrity against damage and leaks, provide sufficient mass transfer, control polarization, and permit cleaning, with minimum waste of energy and water (Perry and Green, 1997). Efficient membrane modules include (Cheryan, 1992, 1998):

1. Hollow fiber-capillary systems with outer diameter of the capillaries about 93 μm , used mainly in reverse osmosis. The liquid flows from the shell into the capillary. In ultrafiltration the liquid flows from the interior of tubes 0.25 - 6 mm diameter.
2. Tubular modules of 12 to 25 mm diameter, supported on the perforated or porous walls of stainless steel tubes. The liquid flows from the interior of the tube outwards.
3. Spiral wound modules are inexpensive arrangements, providing large separation surface. Many membrane layers are wound to produce module diameters up to 400 mm. The liquid feed flows from the outside and it is separated into the permeate, which flows in an internal tube, and the retentate, which is removed as side product.
4. Plate and frame modules, resembling the mechanical pressure (plate and frame) filters (Saravacos and Kostaropoulos).
5. Ceramic modules are usually monoliths of tubular capillaries with typical channel size of 3 mm. Many monoliths are incorporated into a modular housing. A typical commercial module of ceramic membranes has a length of 90 cm, diameter 15 cm and channel diameter 2.5 mm.

3. Membrane Separation Systems

a. Pressure-driven Separations

Most applied membrane separations are pressure-driven, such as ultrafiltration and reverse osmosis. They are single units, with limited application of the multi-stage systems. Figure 10.2 shows the operating principle of a separation unit, consisting of the membrane module, the feed pump, and the accompanying piping for separating the feed into a permeate and a concentrate. The feed pump can be centrifugal for low pressures and positive displacement (piston type) for high pressures. The main product may be the permeate (e.g., desalted water in reverse osmosis) or the concentrate or retentate (e.g., recovered protein in ultrafiltration).

The operating pressure depends on the separation process and it varies in the ranges shown in Figure 10.3.

Recirculation (“feed and bleed system”) of the liquid concentrate is necessary in most applications to maintain high flow rate and increase the efficiency of separation.

Flux reduction of the permeate may be caused by membrane compaction, concentration polarization, or membrane fouling. Fouling is caused by plugging the membrane pores or by cake (gel) formation on the membrane surface.

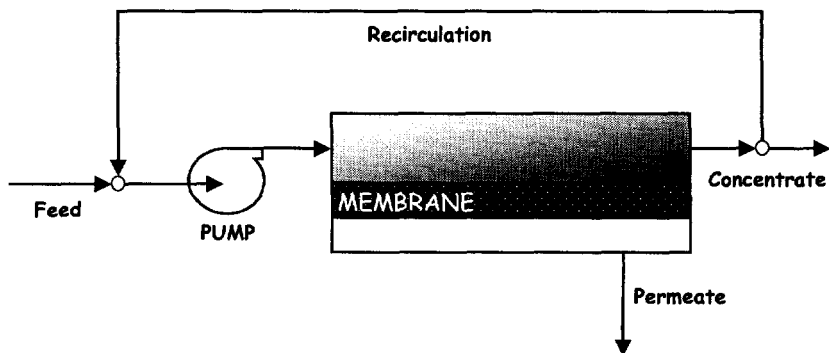


Figure 10.2 Schematic diagram of a membrane separation system.

Fouling can be prevented by selection of the proper membrane, e.g., cellulose acetate is a low-fouling polymer. Process configuration and flow rate can also control fouling. High shear rates, caused by high flow rates, at the membrane surface reduce fouling by removing any deposits formed. An economic balance should be made to determine the optimum flow rate for minimum fouling and cleaning cost for the membranes. The operating pressure should also be optimized in relation to fouling and energy (pumping) cost.

Proteins are the major foulant in membrane separations, especially in ultrafiltration and microfiltration, because of their complex and sensitive physico-chemical structure.

Membrane fouling can be reduced by pulsating flow (periodic increase of pressure), backflushing, and washing with water.

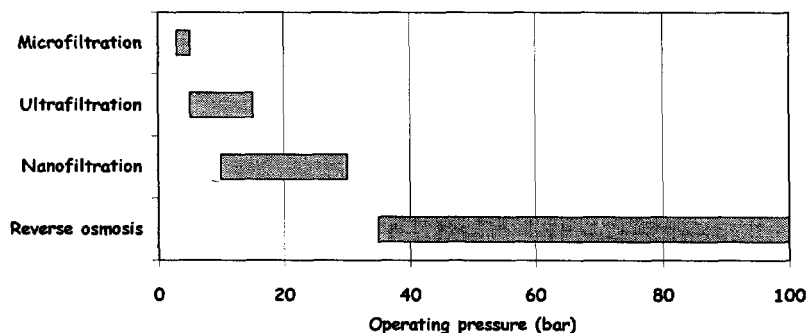


Figure 10.3 Range of operating pressure for membrane separation processes.

Cleaning and sanitizing of fouled membranes involves physical removal of fouling substances, chemical removal of foulants, and hygienic (sanitary) treatment to remove all viable microorganisms (Cheryan, 1998). Detergents, heating and mechanical energy (high velocity) are utilized in a systematic cleaning procedure, similar to the cleaning of food process equipment (Saravacos and Kostaropoulos, 2002). Special care should be taken in cleaning the polymeric membranes, which are sensitive to the cleaning chemicals, the pH, and temperature, compared to the cleaning of stainless steel processing equipment. Cellulose acetate is very sensitive to both temperature and pH, and polyamides are sensitive to chlorine. Ceramic membranes are resistant to chemicals and temperature, but care should be taken in cleaning the gaskets, epoxy resins, etc., which are part of the membrane module.

Turbulent flow is essential during membrane cleaning, employing fluid velocities in the range of 1.5-2.0 m/s in tubular systems. Hollow fibers, used in ultrafiltration, operate in the laminar region, but the high shear rates developed facilitate cleaning. Cleaning time should be 30-60 min and the temperature should not exceed 60 °C.

The cleaning water should be free of iron, which may deposit a yellow fouling film on the surface of the cleaned membrane. Silicates should be removed from the cleaning water, preventing severe fouling of the membranes. Permeate from reverse osmotic desalting units is suitable for cleaning various membranes (Cheryan, 1998).

b. Special Membrane Separations

Several special membrane separation processes are applied in Chemical and Process Engineering. They are based on the selectivity of polymeric membranes to the diffusion and flow of various molecules from mixtures and solutions. The membrane processes are simpler and more economical than the conventional mass transfer processes of distillation, extraction, adsorption, etc (Chapter 9). Thus, gas separations (oxygen/nitrogen) by membranes are replacing distillation, absorption and adsorption.

Two special membrane separations are of interest to food processing, i.e., pervaporation and electrodialysis (Perry and Green, 1997; Saravacos and Kostaropoulos, 2002).

i. Pervaporation

Pervaporation is a separation process, in which one component of a liquid mixture is separated through a permselective membrane. The component is transported by diffusion through the membrane, and desorbed into the permeate space.

In food systems, pervaporation is used either to remove water from liquids through hydrophilic membranes, or organic components through hydrophobic membranes.

The pervaporation membrane system includes heating and recirculation of the feed, condensation of the volatile component, and a vacuum pump (Figure 10.4).

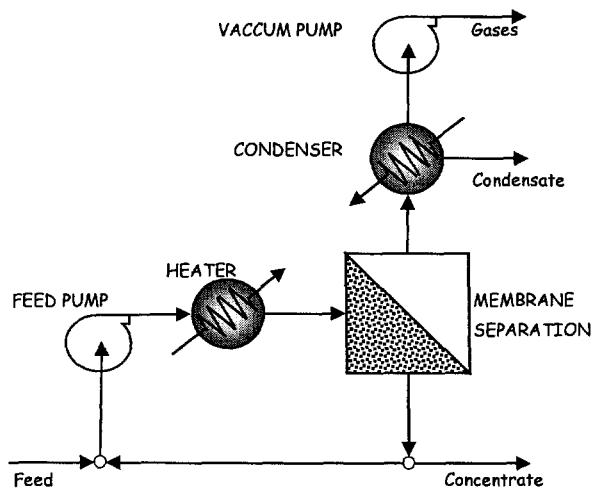


Figure 10.4 Diagram of a pervaporation system.

A typical application of pervaporation is the separation of water (5 % by volume) from the ethanol azeotrope mixture, using a hydrophilic membrane. It can be used for the concentration of fruit juices, by removing only water through an appropriate membrane, while the volatile components are retained in the juice concentrate. The aqueous essence (aroma solution) of fruit juices can be concentrated, removing the water through a hydrophilic membrane. The volatility of food aroma components during pervaporation (Karlsson and Tragardh, 1997) can be analyzed by methods discussed in Chapter 9.

Pervaporation is used for the de-alcoholization of wine, instead of the thermal distillation (spinning cone column) process, described in Chapter 9. Hydrophobic (special rubber) membranes are used to remove ethanol to a level below 0.5 % (Mermelstein, 2000). The process is carried out near room temperature, applying a small pressure difference.

ii. Electrodialysis

Electrodialysis (ED) is a membrane-separation process of electrolytes from non-electrolytes, based on the migration of ions through permselective membranes.

The ED membranes consist of swollen gels of polymers with fixed ionic charges. Cation exchange membranes consist of sulfonated polystyrene copolymerized with divinylbenzene. Anion exchange membranes consist of quaternary anions attached on polystyrene-divinylbenzene polymers (Perry and Green, 1997).

The positive and negative ion exchange membranes are placed alternately in a stack, containing several membranes, similar to the plate and frame filtration system. The ions are induced to migrate by an electric potential, the anions passing the anion exchange membrane, and the cations the cation exchange membrane.

Electrodialysis competes with RO in desalting brackish water (salt concentrations 500 to 5000 ppm TDS). However, it is not economical for desalination of sea water (concentrations above 25 000 ppm TDS).

ED is used in food processing mainly for the demineralization (de-ashing) of whey before evaporation and spray drying. It can be used for the de-acidification of citrus juices (Grandison and Lewis, 1996). ED membranes may have an operating life of about 7 years. Their replacement represents about 50% of the total annual cost, followed by the energy cost of about 20%.

III. MICROFILTRATION AND ULTRAFILTRATION

1. Microfiltration

Microfiltration (MF) is used to remove particles larger than 0.2 μm from solutions and suspensions, e.g., large molecules, particles, and microorganisms (yeasts and bacteria). MF membranes are usually made of porous polymeric materials of tortuous (sponge) structure, or inorganic agglomerates and sinters of micro-particles (metal, metal oxide, graphite, or ceramic). Ceramic membranes are made of monoliths with channels of 2-6 mm diameter.

MF membranes are rated for pore size and liquid flux. The pore size is characterized by special tests, such as the bubble point method and the retention of particles or bacteria of certain sizes (Cheryan, 1998; Saravacos and Kosatropoulos, 2002).

Microfiltration of dilute suspensions (e.g., below 0.5%) is normally carried out by “dead-end” filtration. i.e., flow perpendicular to the membrane surface. Suspensions of higher concentration are treated by cross-filtration (flow parallel to the membranes), which reduces membrane fouling.

MF and other separation membranes are asymmetric, i.e., they have a tight layer (skin) on the top of a more porous structure. The membranes are usually operated with the skin in contact with the feed liquid. However, liquids with high solids loads are better filtered with inverted membranes, which retain the large particles in their pores. Periodic flushing of membranes with water is necessary to maintain an acceptable level of liquid flux.

Spiral wound modules are preferred in MF, because they are cheaper, but they are difficult to operate (fouling) with particles larger than 200 μm . Mechanical filters may be used for removing (pre-filtering) the large particles before MF is applied. Tubular MF modules foul less and they can be cleaned more easily than the spiral wound or hollow fiber systems.

Ceramic (inorganic) membranes are commonly used in MF. They consist of two layers, i.e., a porous support with pores 10 μm or larger, and an active coat of smaller pores. They are used as tubular monoliths, and they are particularly suited for high temperature and high pH applications. Operating conditions of the MF units are, pressure 1-15 bar, and temperature 50-90°C. Temperature has a positive effect on the flux of juices and sugar solutions, since the viscosity decreases sharply at higher temperatures. However, temperature may have a negative effect on the flux of MF membranes treating protein suspensions, due to the precipitation (coagulation) of proteins at high temperatures.

Microfiltration is applied to several separations in food processing, similar to those of ultrafiltration. Clarification of fruit juices, wine and beer by MF can replace conventional mechanical filtration, eliminating the use of filter aids and reducing the process time and, finally, the process cost. Fouling of MF membranes (plugging with large particles) is a problem, which limits its widespread application

MF can be used to remove bacteria and other spoilage microorganisms from liquid foods, such as milk, wine and beer. The milk concentrate can be sterilized separately and re-combined with the sterile milk permeate. The use of MF membranes for sterilization is governed by strict public health regulations (FDA). Fouling of MF membranes is caused mainly by protein components of the foods. The bovine serum albumin (BSA) is normally used as a model protein in fouling studies. Membrane fouling is reduced by backflushing, pulsing, and washing.

Scale-up of membrane plants is almost linear: After a certain size is reached, capacity is increased by adding more modules to the system. Thus, pilot plant tests and data can be used directly in the design of large commercial installations.

2. Ultrafiltration

Ultrafiltration (UF) is used to separate macromolecules and small particles (concentrate) from solvents (water), ions and small molecules (permeate). It can replace evaporation, which may damage heat-sensitive food products. The most important application of UF is in the dairy industry where milk and whey proteins are separated from lactose and other solutes. UF is also finding applications in biotechnological processing and in the treatment of wastewater.

a. UF Systems

UF membranes include cellulose acetate, polyamides, polysulphones, polyvinylidene fluoride, and polyvinyl alcohol-polyethylene copolymers. Ceramic membranes, based on porous alumina and carbon substrates, are also used (Perry and Green, 1997).

UF membranes are characterized by their permeability and retention properties. The membrane permeability is determined by the pore size distribution and the thickness of the active layer. Permeability data are reported at standard conditions, e.g., at $\Delta P = 3.43$ bar and $T = 25$ °C. New membranes may have permeabilities of 0.1 - 1 mm/s, which are reduced considerably during processing.

The separating capacity of the UF membranes is characterized by the molecular weight cut off (MWCO), the maximum size of the molecules that will pass through. The size of polymer molecules is expressed by their molecular weight, e.g., Dextran 250 is a dextrin of molecular weight 250 000 or 250 kD, where 1 kD = 1000 daltons. The separation of polymer molecules is not sharp, since the pore size of UF membranes is not uniform but dispersed, following the normal or log normal distribution models. The MWCO of UF membranes is determined experimentally, using dextrans of known molecular weight, and it is in the range of 2 to 300 kD.

UF membranes are produced in various modular forms: Tubular (diameters 5-25 mm), hollow fiber / capillary (diameters 0.25 - 1 mm), spiral wound, and plate and frame. The hollow fibers require clean feeds, such as protein solutions. The tubular systems are used when the feed contains significant amounts of suspended particles, such as cloudy fruit juices.

Concentration polarization is a serious problem of UF and RO membranes, reducing significantly the flux of permeate at higher operating pressures. At low pressures, the flux increases linearly with the applied pressure. However, above a certain pressure the flux increases slowly or becomes independent of the applied pressure. Equilibrium is established between concentration of the solute on the membrane surface and dispersion into the flowing liquid. Polarization is reduced by cross flow of the liquid, parallel to the membrane surface, which removes the accumulated solute. Flow geometry and flow rate control concentration polarization. Most of the mechanical energy in UF is consumed to reduce concentration polarization.

Fouling of UF membranes is caused by plugging of small pores, selective plugging of larger pores, and cake formation of adsorbed macromolecules and particles. Fouling is evidenced by sharp decline of the membrane flux, after the system has operated for some time. When the flux drops below a pre-determined low level, operation is interrupted and the membrane is cleaned.

Fouling can be reduced by choosing non-fouling membranes, such as cellulose acetate and by proper operating procedures. Cleaning procedures and addition of anti-fouling agents are commonly used.

UF separation systems operate at temperatures in the range 5-45 °C. In general, the flux of permeate increases with increasing temperature, mainly because of a reduction of the liquid viscosity. The viscosity of concentrated sugar solutions drops sharply as the temperature is increased (high activation energy). However, temperature has a smaller effect (low activation energy) on the (apparent) viscosity of non-Newtonian (pseudoplastic) solutions/suspensions (Saravacos and Maroulis, 2001).

Rejection (R) of macromolecules and suspended particles by UF membranes is between 0.9 and 1.0, while for small molecules and ions $R = 0$ to 0.1. Flux rates in UF vary over the range of 5-500 L/m²h. Pumping energy consumption is in the range of 0.5-5 kWh / m³ permeate. The flux rate is related to the membrane, fouling and concentration polarization resistances (Equation 10-4).

Diafiltration is a special case of UF, in which water is added to the concentrate side of the membrane system, facilitating the removal (permeation) of unwanted solutes or ions from the concentrated product. Water is added either batchwise or continuously. Typical applications of diafiltration are the removal of lactose from the protein concentrate of UF-treated whey, and the fractionation of proteins with removal of low-molecular weight components, e.g., ions, sugars, and ethanol (Grandison and Lewis, 1966).

b. Food Applications of UF

UF is used commercially in dairy technology, juice technology, water technology, and wastewater treatment (Cheryan, 1998; Grandison and Lewis, 1996).

Recovery of proteins from cheese whey is an important dairy application. The whey is separated into a protein concentrate and an aqueous permeate containing lactose and minerals, which may be further concentrated by RO or evaporation (Example V.1).

UF concentration of whole milk (2 to 5 times) is used in the preparation of special types of cheese (e.g., feta) and yogurt.

UF is used in combination with RO and ion exchange in the cleaning and desalting of drinking and process water. Hollow-fiber UF membranes of 20 kD (MWCO) are used at pressures about 3 bar with water fluxes 15-75 L/m²h. Energy use is about 0.5 kWh/m³ of water.

UF is used to clarify fruit juices, replacing conventional juice clarification technology (enzyme treatment, filter aids, cake filtration), with significant operating and economic advantages (Cheryan, 1998). The time-consuming conventional operation is replaced by the one-step UF process, obtaining higher juice yields (96-68%). Polymeric membranes of 30 kD (MWCO) are used with fluxes 35-300 L/m²h.

UF ceramic membranes are suitable for juice clarification, because they are more stable and more easily cleaned than polymeric membranes. Typical UF ceramic membranes, used for the clarification of apple juice, have 0.2 μm pores, tube diameter 4 mm and length 85 cm. High flux rates of 400-500 L/m²h are obtained with pressures 1-7 bar (McLellan, 1993; McLellan and Padilla, 1992).

Combination of UF clarification and RO concentration can yield clarified juice concentrates up to about 35 °Brix. Higher concentrations require thermal evaporation.

UF is used in combination with ion exchange to de-bitter citrus juices (removal of bitter components limonin and naringin) The cloudy juice is passed through a UF membrane system to remove suspended particles and colloids, which would plug the ion exchange resins (Cheryan, 1998). The bitter components from the clarified juice are adsorbed on resins, e.g., divinylbenzene copolymer, and the debittered juice is mixed with the separated pulp.

Wine and beer can be clarified by UF membranes, replacing chemical treatment and cake filtration. UF membranes of 0.45 μm pores will retain yeasts and other microorganisms, eliminating the need for thermal pasteurization.

UF and RO separations can be used in the treatment of wastewater of food-processing plants, reducing pollution and recovering water for reuse. A pilot plant system of dairy wastewater treatment is described by Voros et al. (1999).

IV. REVERSE OSMOSIS AND NANOFILTRATION

The design of membrane separation systems can be carried out using simplified software, developed by membrane suppliers. Software used for reverse osmosis is described by Brauns (2001).

Reverse osmosis (RO) is used to remove water from aqueous solutions, employing tight membranes, which retain the dissolved molecules and ions. The applied pressure must be higher than the osmotic pressure of the solution in order to overcome the membrane resistance and the resistances of concentration polarization and membrane fouling, according to the flux Equation (10-3). Nanofiltration (NF) is more selective than RO, retaining the dissolved molecules and the polyvalent ions, but allowing the permeation of the monovalent ions, along with the water. RO operates at pressures 35-100 bar, while lower pressures (10-30 bar) are applied in NF.

The osmotic pressure of aqueous solutions and food liquids depends on the size and concentration of the dissolved molecules (% total solids in water), as shown in Table 10.2 and Figure 10.5 (Cheryan, 1998).

Very high osmotic pressures are obtained in concentrated solutions of small molecules and (monovalent) ions, e.g., salt (sodium chloride) and sucrose solutions. The elevated osmotic pressures limit the application of RO in the production of concentrated salt and sugar solutions (clear juices).

Table 10.2 Osmotic Pressure of Some Aqueous Solutions

| | Concentration (%) | Osmotic pressure (bar) |
|-----------------|----------------------|---------------------------|
| Sodium chloride | 1 | 8 |
| | 3 | 35 |
| Sucrose | 25 | 27 |
| | 53 | 107 |
| | 65 | 200 |
| Milk, nonfat | 9 | 7 |
| Whey | 6 | 7 |
| Orange juice | 11 | 16 |
| Apple juice | 15 | 20 |
| Coffee extract | 28.5 | 35 |

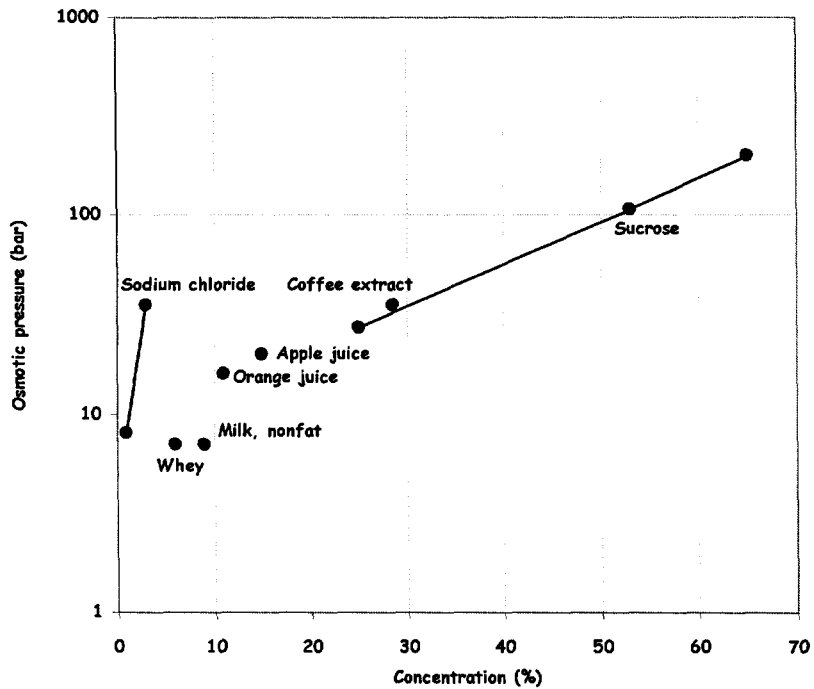


Figure 10.5 Osmotic pressure of some aqueous solutions.

1. RO Systems

RO membranes are rated for (water) flux and % rejection of the solutes. RO membranes, used in water desalting, may have rejections up to 99.7 % (estimated on the basis of rejection of sodium chloride). The rejection of nanofiltration membranes is based on magnesium sulphate (Perry and Green, 1997).

Concentration polarization of RO membranes is caused by increased flux of water, due to high-pressure differential, resulting in fouling and scaling of the membrane surface by precipitated salts. It can be reduced by increasing the cross-flow velocity of the solution, by adding anti-scalants in the feed solution, or by operating at lower pressure gradients. Hollow-fiber and spiral-wound membrane modules operate at lower fluxes with less polarization problems. Membrane fouling with various colloids and soil particles can plug RO membranes, causing a sharp reduction of the permeation rate. Concentration polarization can be prevented by pre-filtration, pH adjustment, and chlorination.

2. Food Applications of RO

Desalting of brackish water and desalination of seawater to produce potable and process water are the most important applications of RO (Perry and Green, 1997). The RO process competes with evaporation in small and medium installations, while evaporation is favored in large-scale applications, using multi-effect systems. Potable water should contain less than 500 ppm of total dissolved solids (TDS), while the concentrations of the brackish and seawater are about 1500 and 30000 ppm TDS, respectively. Water recovery in the desalting of brackish water is high (70-90 %), but low in seawater (25-30 %). Energy requirements for brackish water desalting are low (about 1 kWh/m³), compared to the high-energy requirements for seawater (about 6 kWh/m³).

The economics of water desalination can be analyzed and predicted, using computer spreadsheets, as discussed by Ettouney et al. (2002).

Demineralized water, suitable for steam boilers and other specialized applications, is produced by RO in combination with ion exchange (Chapter 9). The water is first filtered from the suspended substances, then it is treated in an RO unit to remove most of the dissolved solids, and finally it is treated in an ion exchange system to remove any residual salt ions.

RO is used as a pre-evaporation step in the concentration of milk and fruit juices, removing most of the water, before the final concentration in the evaporator. Milk is concentrated by a factor of 2 to 3, using RO. Concentration of the milk by 1.5 times is used for the preparation of yogurt. Whey is concentrated from 6 to 24 % TS. Due to the high osmotic pressure of sugar solutions, most applications of RO are in preparing juice concentrates up to about 25-30 °Brix.

Nanofiltration is used to separate by preferential permeation monovalent ions (sodium chloride) from polyvalent ions and other solids of the whey. It is also used in the deacidification of citrus juices, by removing (permeating) hydrogen ions.

Concentration of cloudy orange juice by RO is limited by membrane fouling with pectins and precipitated hesperidin (a citrus glycoside). Combination of RO and falling film evaporation can reduce significantly the cost of concentration of apple juice (Moresi, 1988).

RO can be used to concentrate wine by removing ethanol and water through a selective membrane. The concentrate is diluted with water to prepare de-alcoholized wine (Mermelstein, 2000), while the permeate can be distilled to recover the ethanol. Pressures of about 30 bar and temperatures 7-13 °C are used. De-alcoholization of wine can also be accomplished by another membrane separation process, pervaporation (Section II.3.b.i of this chapter).

V. APPLICATION TO CHEESE WHEY PROCESSING

A good example of successful application of membrane technology is the processing of cheese whey, a byproduct of the cheese industry. It is the liquid fraction that is drained from the curd during the manufacture of the cheese. Its disposal is a major problem for the dairy industry, as reflected by its composition (Table 10.3). Figure 10.6 presents the milk and cheese whey composition along with the component particle size, while Figure 10.7 shows a general schematic of potential application of membranes in whey processing.

Table 10.3 Typical Composition of Milk and Cheese Whey

| Component | Milk | Cheese Whey |
|--------------|------------|-------------|
| Fat | 3.5 | 0.1 |
| Protein | 3.3 | 0.6 |
| Lactose | 4.9 | 5.0 |
| Salts | <u>0.7</u> | <u>0.6</u> |
| Total solids | 12.4 | 6.3 |

Data from USDA (1978)

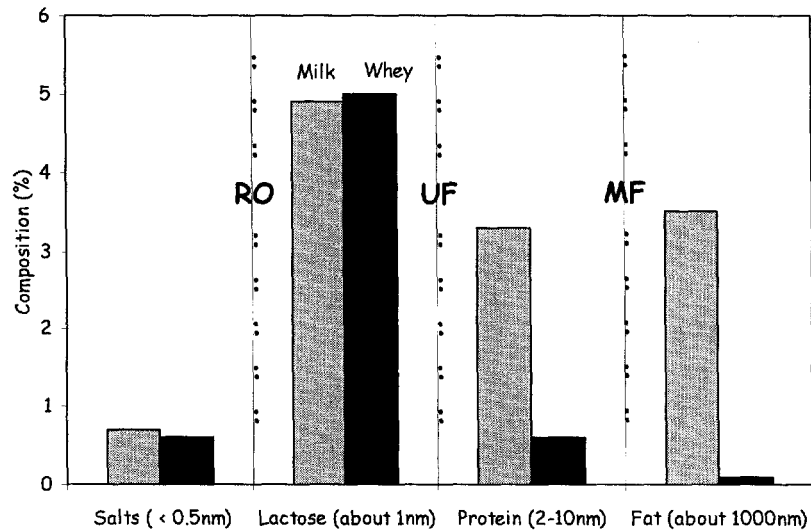


Figure 10.6 Typical composition of milk and cheese whey versus particle size.

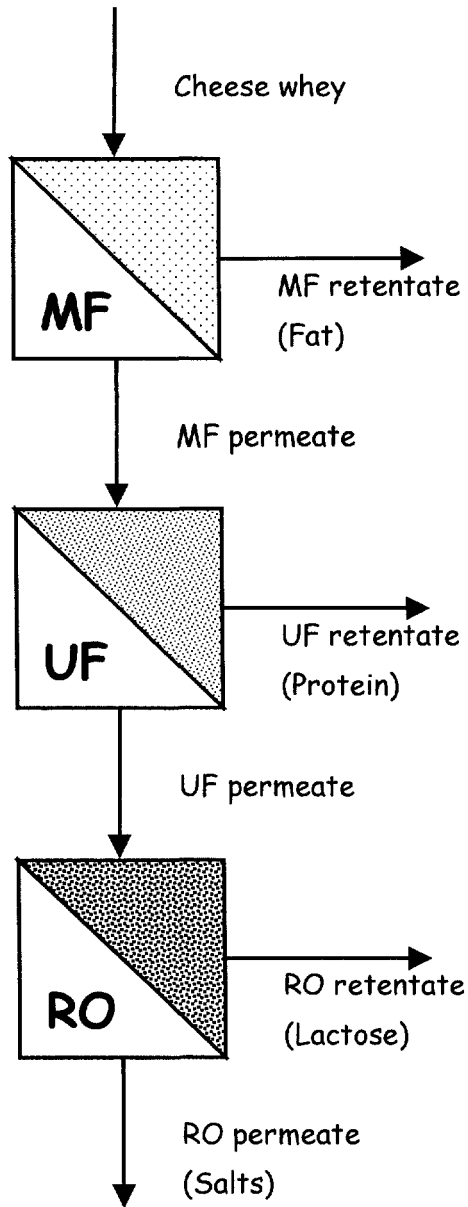


Figure 10.7 Application of membrane separations to cheese whey. MF = micro-filtration, UF = ultrafiltration, RO = reverse osmosis.

1. Design of an Ultrafiltration Process

In this section, a design approach is described for an ultrafiltration unit.

a. Process Description

A typical flowsheet of an ultrafiltration system is presented in Figure 10.8. The process stream at a flow rate V_F (kg/s) and concentration C_{Fj} (kg/kg wb) in each component j enters the feed pump which provides the operating (transmembrane) pressure P_F . The feed stream is mixed with the recycle stream (provided by the recirculation pump) and enters the ultrafiltration unit. The membrane permeate stream contains the rejected components, while the retentate stream is enriched in the rejected components.

b. Process Model

The mathematical model of the process presented in Figure 10.8 is summarized in Table 10.4.

Equation (E01) is the material balance for the water, while Equation (E02) is the material balance for the j component. There exist C such equations, one for each component. Equation (E03) represents the ultrafiltration separation model suggested by Cheryan (1998). Each component j for a specific membrane is characterized by a rejection parameter R_j . There exist C equations of this kind. Equation (E04) calculates the water flux through the membrane. A is the water permeability of the membrane, which depends on pressure, temperature and flow conditions.

Equation (E05) calculates the required total number of tubes, or hollow fibers versus the tube or hollow fiber dimensions. Equation (E06) estimates the required recirculation flow rate to obtain the appropriate velocity in tubes or hollow fibers. Equation (E07) calculates the friction pressure loss inside the tubes or hollow fibers when the Fanning type friction parameter is known. It must be noted that Equations (E05) through (E07) are valid for flow into cylindrical canals, that is tubular on hollow fiber membranes. These equations need some modification for other geometries, such as plate and frame or spiral wound membranes.

Finally, Equations (E08) and (E09) size the required pumps.

$(14+3C)$ variables presented in Table 10.5 are involved in the model of $(7+2C)$ equations presented in Table 10.4, which means that $(7+C)$ degrees of freedom are available, and if we consider a typical problem with $(3+C)$ specifications presented in Table 10.6 four design variables are available for process optimization. Table 10.6, suggests a selection of design variables and the corresponding solution algorithm. The total annualized cost TAC presented in Table 10.7 is used as objective function in process optimization.

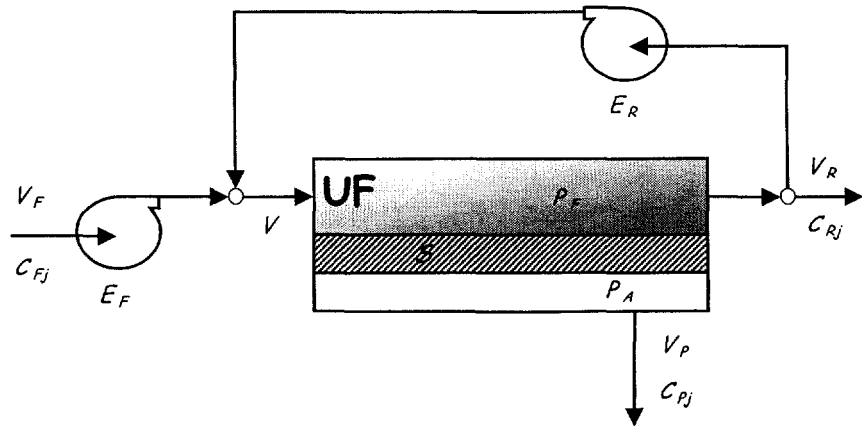


Figure 10.8 Diagram of a typical ultrafiltration unit.

Table 10.4 Process Model of an Ultrafiltration Unit

$$V_F = V_R + V_P \quad (\text{E01})$$

$$V_F C_{Fj} = V_R C_{Rj} + V_P C_{Pj}, \text{ for } j = 1, 2, \dots, C \quad (\text{E02})$$

$$C_{Rj} = C_{Fj} \left(\frac{V_F}{V_R} \right)^{R_j}, \text{ for } j = 1, 2, \dots, C \quad (\text{E03})$$

$$V_P = A S (P_F - P_A) \quad (\text{E04})$$

$$N = \frac{S}{\pi d L} \quad (\text{E05})$$

$$V = N \frac{\pi d^2}{4} u \rho \quad (\text{E06})$$

$$\Delta P = 4 f \left(\frac{L}{d} \right) \frac{\rho u^2}{2} \quad (\text{E07})$$

$$E_R = V \Delta P / \rho \quad (\text{E08})$$

$$E_F = V_F (P_F - P_A) \quad (\text{E09})$$

Table 10.5 Process Variables

| | | |
|------------|----------------|---|
| C_{Fj} | % | Feed concentration of j component |
| C_{Pj} | % | Permeate concentration of j component |
| C_{Rj} | % | Retentate concentration j component |
| d | m | Hollow fiber diameter |
| E_F | kW | Feed pump power |
| E_R | kW | Recycle pump power |
| L | m | Hollow fiber length |
| N | – | Number of fibers |
| P_A | bar | Atmospheric pressure |
| P_F | bar | Operating pressure |
| S | m ² | Membrane area |
| u | m/s | Fluid velocity in fiber |
| V | kg/s | Recycle flow rate |
| V_F | kg/s | Feed flow rate |
| V_P | kg/s | Permeate flow rate |
| V_R | kg/s | Retentate flow rate |
| ΔP | bar | Pressure loss |

Table 10.6 Solution of a Typical Design Problem**Process Specifications**

| | | |
|----------|------|-------------------------------------|
| V_F | kg/s | Feed flow rate |
| C_{Fj} | °C | Feed concentration of j component |
| P_A | bar | Atmospheric pressure |
| V_R | kg/s | Retentate flow rate |

Degrees-of-Freedom Analysis

Process variables $14+3C$ (C is the number of solutes)

Process equations $7+2C$

Degrees of freedom $7+C$

Specifications $3+C$

Design variables 4

Design Variables

| | | |
|-------|-----|-------------------------|
| d | m | Hollow fiber diameter |
| L | m | Hollow fiber length |
| u | m/s | Fluid velocity in fiber |
| P_F | bar | Operating pressure |

Solution algorithm

- (E01) $\rightarrow V_P$
(E03) $\rightarrow C_{Rj}, j = 1, 2, \dots, C$
(E02) $\rightarrow C_{Pj}, j = 1, 2, \dots, C$
(E04) $\rightarrow S$
(E05) $\rightarrow N$
(E06) $\rightarrow V$
(E07) $\rightarrow \Delta P$
(E08) $\rightarrow E_R$
(E09) $\rightarrow E_F$
-

Table 10.7 Cost Analysis**Equipment cost**

$$C_{eq} = C_m S + C_p E_R^{n_p} + C_p E_f^{n_p}$$

Annual operating cost

$$C_{op} = C_e (E_R + E_f) t_y + C_r S$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op}$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r(1+i_r)^{l_f}}{(1+i_r)^{l_f} - 1}$$

Cost Data

| | | |
|-------|----------------------|--|
| C_m | \$/m ² | Membrane unit cost |
| n_m | – | Membrane scaling factor |
| C_p | \$/kW | Pump unit cost |
| n_p | – | Pump scaling factor |
| C_e | \$/kWh | Cost of electricity |
| C_r | \$/m ² yr | Membrane replacement and cleaning cost |
| t_y | h/yr | Annual operating time |
| i_r | – | Interest rate |
| l_f | yr | Lifetime |

c. Application to Cheese Whey

The previous design procedure is applied to the cheese whey ultrafiltration to recover the whey protein as shown in Figure 10.7. Typical process specifications for the design problem are presented in Table 10.8. The necessary technical and cost data are also summarized in Table 10.8 and in Figures 10.9 and 10.10. The data of Figure 10.9 correspond to the performance of a membrane with pores of average size about 5 nm.

The process design results are summarized in Table 10.9. A membrane area of 136 m² is needed for a concentration ratio $V_f/V_R=50$. The corresponding component flow rates are presented in Figure 10.11. Practically all the protein is retained.

The retentate flow rate, and the component concentrations in retentate and permeate versus the membrane area are presented in Figures 10.12 through 10.14, respectively. The protein concentrate can be spray-dried to a dehydrated protein product.

Table 10.8 Process Specifications and Process Technical and Cost Data**Process Specifications**

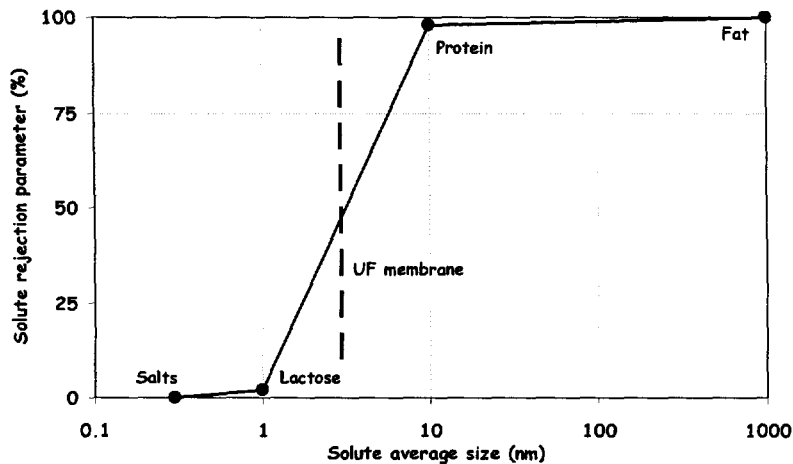
| | | |
|----------|--------------|-------------------------------|
| V_F | = 5.00 ton/h | Feed flow rate |
| V_R | = 0.10 ton/h | Retentate flow rate |
| C_{FF} | = 0.10 % | Feed concentration of fat |
| C_{FP} | = 0.60 % | Feed concentration of protein |
| C_{FL} | = 5.00 % | Feed concentration of lactose |
| C_{FS} | = 0.60 % | Feed concentration of salts |
| P_A | = 1 bar | Atmospheric pressure |

Technical Data

| | | |
|--------|-----------------------------|------------------------------------|
| R_f | = 1.00 | Rejection parameter of fat |
| R_p | = 0.99 | Rejection parameter of protein |
| R_l | = 0.01 | Rejection parameter of lactose |
| R_s | = 0.00 | Rejection parameter of salts |
| A | = 5×10^{-6} m/sbar | Water permeability of the membrane |
| f | = 0.1 | Friction loss coefficient |
| ρ | = 1000 kg/m ³ | Water density |

Cost Data

| | | |
|-------|------------------------------|--|
| C_m | = 1.00 k\$/m ² | Membrane unit cost |
| n_m | = 1.00 | Membrane scaling factor |
| C_p | = 0.50 k\$/kW | Pump unit cost |
| n_p | = 0.65 | Pump scaling factor |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_r | = 0.25 k\$/m ² yr | Membrane replacement and cleaning cost |
| t_y | = 5000 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

**Figure 10.9** Solute rejection parameters in ultrafiltration.

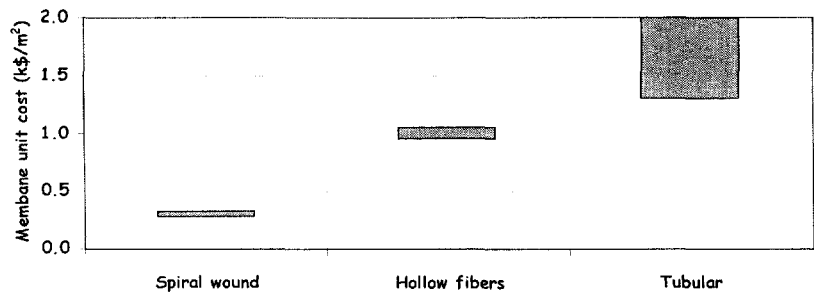


Figure 10.10 Ultrafiltration membrane unit cost of various types. Data from Cheryan (1998).

Table 10.9 Process Design Results

Feed

| | | |
|----------|--------------|--------------------------|
| V_F | = 5.00 ton/h | Flow rate |
| C_{Ff} | = 0.10 % | Concentration of fat |
| C_{Fp} | = 0.60 % | Concentration of protein |
| C_{Fl} | = 5.00 % | Concentration of lactose |
| C_{Fs} | = 0.60 % | Concentration of salts |

Retentate

| | | |
|----------|--------------|--------------------------|
| V_R | = 0.10 ton/h | Flow rate |
| C_{Rf} | = 5.00 % | Concentration of fat |
| C_{Rp} | = 28.9 % | Concentration of protein |
| C_{Rl} | = 5.20 % | Concentration of lactose |
| C_{Rs} | = 0.60 % | Concentration of salts |

Permeate

| | | |
|----------|--------------|--------------------------|
| V_P | = 4.90 ton/h | Flow rate |
| C_{Pf} | = 0.00 % | Concentration of fat |
| C_{Pp} | = 0.02 % | Concentration of protein |
| C_{Pl} | = 5.00 % | Concentration of lactose |
| C_{Ps} | = 0.60 % | Concentration of salts |

Membrane

| | | |
|------------|----------------------|-------------------------|
| S | = 136 m ² | Membrane area |
| d | = 1 mm | Hollow fiber diameter |
| L | = 2 m | Hollow fiber length |
| N | = 21700 | Number of fibers |
| u | = 0.5 m/s | Fluid velocity in fiber |
| P_F | = 3 bar | Operating pressure |
| ΔP | = 1 bar | Pressure loss |

Pumps

| | | |
|-------|----------|--------------------|
| E_F | = 0.3 kW | Feed pump power |
| E_R | = 0.9 kW | Recycle pump power |

Cost

| | | |
|----------|-------------|------------------|
| C_{eq} | = 137 k\$ | Equipment |
| C_{op} | = 35 k\$/yr | Operating |
| TAC | = 55 k\$/yr | Total annualized |

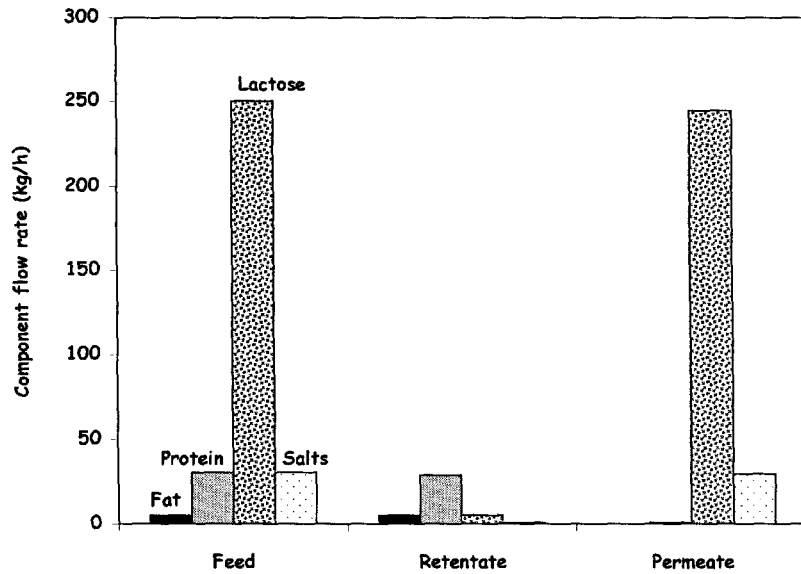


Figure 10.11 Ultrafiltration membrane separation results.

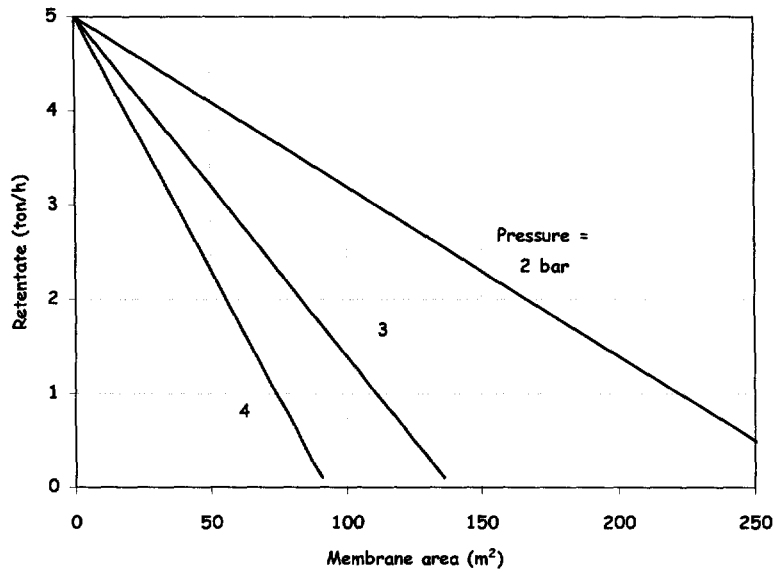


Figure 10.12 Retentate versus ultrafiltration membrane area for various operating pressures.

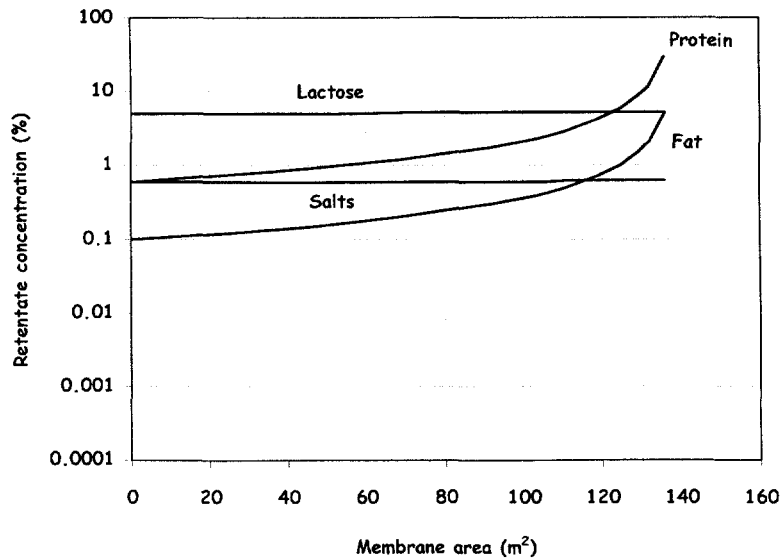


Figure 10.13 Ultrafiltration retentate concentrations versus membrane area.

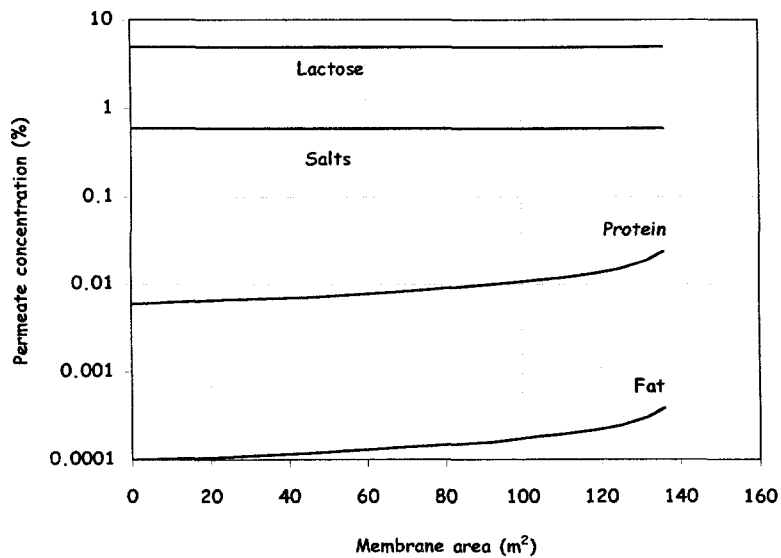


Figure 10.14 Ultrafiltration permeate concentrations versus membrane area.

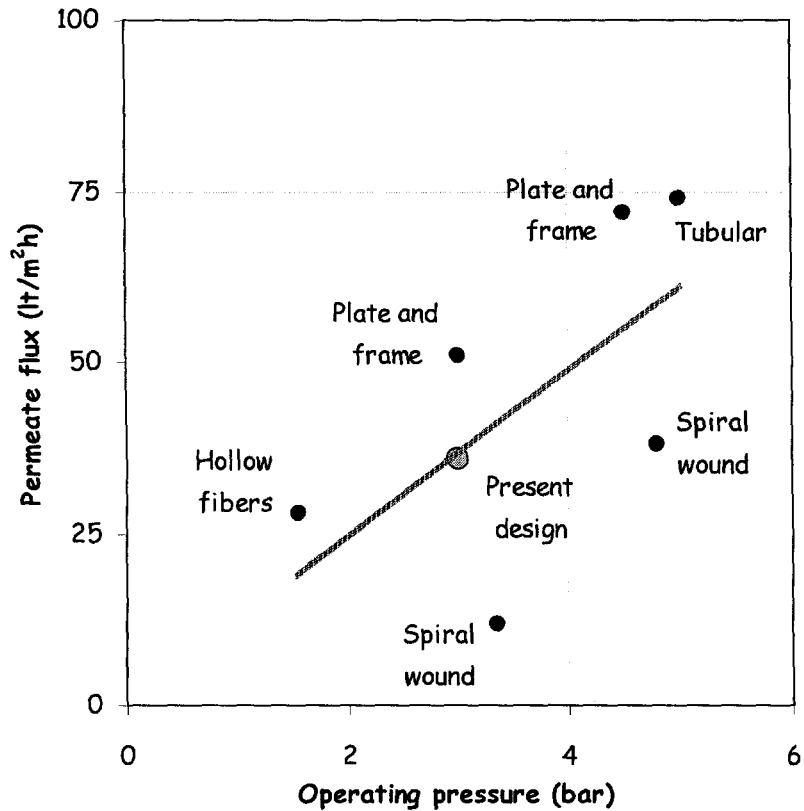


Figure 10.15 Comparison of various UF modules processing cheese whey. Data from Cheryan (1998).

2. Design of a Reverse Osmosis Process

In this section, a design approach is described for a reverse osmosis unit.

a. Process Description

A typical flowsheet of a reverse osmosis system is presented in Figure 10.16. The process stream at a flow rate V_F (kg/s) and concentration C_{Fj} (kg/kg wb) in each component j enters the feed pump which provides the operating (transmembrane) pressure P_F . The feed stream is mixed with the recycle stream (provided by the recirculation pump) and enters the reverse osmosis unit. The membrane permeate stream is poor in rejected components, while the retentate stream is enriched in rejected components.

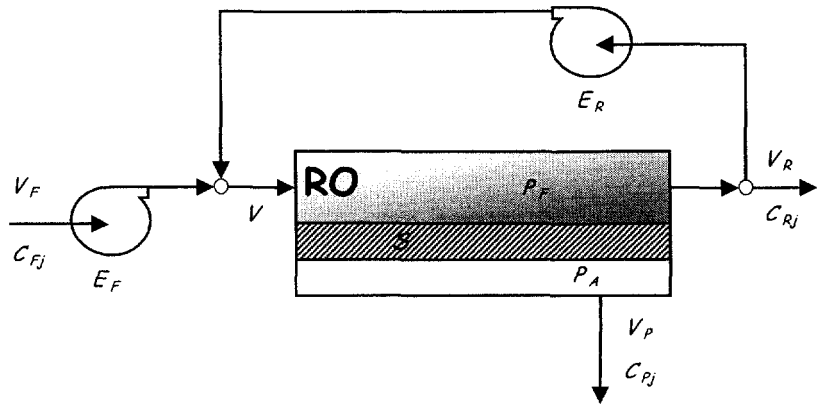


Figure 10.16 Diagram of a typical reverse osmosis unit.

b. Process Model

The mathematical model of the process presented in Figure 10.16 is summarized in Table 10.10.

Equation (E01) estimates the osmotic pressure difference between the two sides of the membrane. Equations (E02) through (E04) describe the reverse osmosis membrane performance. It must be noted that the mechanism, and consequently the corresponding model, is different than the ultrafiltration. Ultrafiltration is a size-separation process as denoted by the model in Table 10.4. Instead, the well known reverse osmosis equations are based on a selective sorption/capillary flow model summarized by Sourirajan, 1977.

Equation (E02) calculates the water flux through the membrane. A is the water permeability of the membrane, which depends on pressure, temperature and flow conditions. The driving force for the water transfer is the difference in the pressure between the two sides of the membrane. Equations (E03) calculate the component permeation through the membrane. The driving force is the difference in concentrations between the two sides of the membrane, while B is the component permeability. There exist C such equations, one for each component. Equations (E04) describe the component diffusion in the boundary layer at the high pressure side of the membrane, where k is the component mass transfer coefficient. There exist C such equations.

Equation (E05) is the material balance for the water, while Equation (E06) is the material balance for the j component. There exist C such equations, one for each component. Equation (E07) calculates the membrane area to obtain the appropriate permeate flow rate.

Equation (E08) calculates the required total number of tubes, or hollow fibers versus the tube or hollow fiber dimensions. Equation (E09) estimates the required recirculation flow rate to obtain the appropriate velocity in tubes or hollow fibers. Equation (E10) calculates the friction pressure loss inside the tubes of hollow fibers when the Fanning type friction parameter is known. It must be noted that Equations (E08) through (E10) are valid for flow in cylindrical canals, that is tubular or hollow fiber membranes. These equations need some modification for other geometries such as plate and frame or spiral wood membranes.

Finally, Equations (E11) and (E12) size the required pumps.

(16+4C) variables presented in Table 10.11 are involved in the model of (9+3C) equations presented in Table 10.10, which means that (7+C) degrees of freedom are available, and if we consider a typical problem with (3+C) specifications presented in Table 10.12 four design variables are available for process optimization. Table 10.12 suggests a selection of design variables and the corresponding solution algorithm. The total annualized cost TAC presented in Table 10.13 is used as objective function in process optimization.

Table 10.10 Process Model of a Reverse Osmosis Unit

$$P_o = p(C_W - C_P) \text{ where } C_W = \sum_{j=1}^C C_{Wj} \text{ and } C_P = \sum_{j=1}^C C_{Pj} \quad (\text{E01})$$

$$J_p = A (P - P_o) \quad (\text{E02})$$

$$J_p C_{Pj} = B_j (C_{Wj} - C_{Pj}) \text{ for } j = 1, 2, \dots, C \quad (\text{E03})$$

$$\frac{C_{Wj} - C_{Pj}}{C_{Rj} - C_{Pj}} = \exp\left(\frac{J_p}{k_j}\right) \text{ for } j = 1, 2, \dots, C \quad (\text{E04})$$

$$V_F = V_R + V_P \quad (\text{E05})$$

$$V_F C_{Fj} = V_R C_{Rj} + V_P C_{Pj} \text{ for } j = 1, 2, \dots, C \quad (\text{E06})$$

$$V_P = S J_p \rho \quad (\text{E07})$$

$$N = \frac{S}{\pi d L} \quad (\text{E08})$$

$$V = N \frac{\pi d^2}{4} u \quad (\text{E09})$$

$$\Delta P = 4 f \left(\frac{L}{d}\right) \frac{\rho u^2}{2} \quad (\text{E10})$$

$$E_R = V \Delta P \quad (\text{E11})$$

$$E_F = V_F (P_F - P_A) \quad (\text{E12})$$

Table 10.11 Process Variables

| | | |
|------------|----------------------------------|---|
| C_{Fj} | % | Feed concentration of j component |
| C_{Pj} | % | Permeate concentration of j component |
| C_{Rj} | % | Retentate concentration j component |
| C_{wj} | % | Wall concentration j component |
| d | m | Hollow fiber diameter |
| E_F | kW | Feed pump power |
| E_R | kW | Recycle pump power |
| L | m | Hollow fiber length |
| N | – | Number of fibers |
| P_A | bar | Atmospheric pressure |
| P_F | bar | Operating pressure |
| P_O | bar | Osmotic pressure |
| S | m ² | Membrane area |
| u | m/s | Fluid velocity in fiber |
| V | kg/s | Recycle flow rate |
| V_F | kg/s | Feed flow rate |
| V_P | kg/s | Permeate flow rate |
| V_R | kg/s | Retentate flow rate |
| ΔP | bar | Pressure loss |
| J_p | m ³ /s/m ² | Water flux through membrane |

Table 10.12 Solution of a Typical Design Problem**Process Specifications**

| | | |
|----------|------|--|
| V_F | kg/s | Feed flow rate |
| C_{Fj} | % | Feed concentration of each j component (C data) |
| C_{Ri} | % | Retentate concentration of the i^{th} component |
| P_A | bar | Atmospheric pressure |

Degrees-of-Freedom Analysis

| | |
|--------------------|---|
| Process variables | $16+4C$ (C is the number of solutes) |
| Process equations | $9+3C$ |
| Degrees of freedom | $7+C$ |
| Specifications | $3+C$ |
| Design variables | 4 |

Design Variables

| | | |
|-------|-----|-------------------------|
| d | m | Hollow fiber diameter |
| L | m | Hollow fiber length |
| u | m/s | Fluid velocity in fiber |
| P_F | bar | Operating pressure |

Table 10.13 Cost Analysis**Equipment cost**

$$C_{eq} = C_m S + C_p E_R^{n_p} + C_p E_F^{n_p} \quad (\text{F01})$$

Annual operating cost

$$C_{op} = C_e (E_R + E_F) t_y + C_r S \quad (\text{F02})$$

Total annual cost (objective function)

$$TAC = e C_{eq} + C_{op} \quad (\text{F03})$$

where the Capital Recovery Factor is calculated from the equation

$$e = \frac{i_r (1 + i_r)^{l_f}}{(1 + i_r)^{l_f} - 1} \quad (\text{F04})$$

Cost Data

| | | |
|-------|----------------------|--|
| C_m | \$/m ² | Membrane unit cost |
| n_m | – | Membrane scaling factor |
| C_p | \$/kW | Pump unit cost |
| n_p | – | Pump scaling factor |
| C_e | \$/kWh | Cost of electricity |
| C_r | \$/m ² yr | Membrane replacement and cleaning cost |
| t_y | h/yr | Annual operating time |
| i_r | – | Interest rate |
| l_f | yr | Lifetime |

c. Application to Cheese Whey

The previous design procedure is applied to the cheese whey processing to recover the lactose after the ultrafiltration processing, as shown in Figure 10.7. Typical process specifications for the design problem are presented in Table 10.14. The necessary technical and cost data are also summarized in Table 10.14 and in Figure 10.17.

The process design results are summarized in Table 10.15. A membrane area of 77 m² at 60 bar can concentrate the lactose in the whey from 5% to 20%. Lactose will crystallize above 20% concentration, fouling the RO membrane. The results are further analyzed in Figures 10.18 and 10.19. Figure 10.18 presents the retentate flow rate versus the membrane area for different operating pressures, while Figure 10.19 presents the lactose concentrations in retentate, permeate and boundary layer versus the membrane area, operating at 60 bar.

The concentrated lactose solution can be crystallized into a crystalline product, or it can be used in industrial fermentations.

Table 10.14 Process Specifications and Process Technical and Cost Data**Process Specifications**

| | | |
|----------|-----------|------------------------------------|
| V_F | = 5 ton/h | Feed flow rate |
| C_{F1} | = 5 % | Feed concentration of lactose |
| C_R | = 20 % | Retentate concentration of lactose |
| P_A | = 1 bar | Atmospheric pressure |

Technical Data

| | | |
|--------|------------------------------|---|
| A | = 5×10^{-7} m/s bar | Water permeability |
| B_l | = 5×10^{-7} m/s | Lactose permeability |
| k_l | = 5×10^{-5} m/s | Boundary layer mass transfer coefficient of lactose |
| f | = 0.1 | Friction loss coefficient |
| ρ | = 1000 kg/m ³ | Water density |
| p | = 1.23 bar/% | Osmotic pressure coefficient |

Cost Data

| | | |
|-------|------------------------------|--|
| C_m | = 1.00 k\$/m ² | Membrane unit cost |
| n_m | = 1.00 | Membrane scaling factor |
| C_p | = 0.50 k\$/kW | Pump unit cost |
| n_p | = 0.65 | Pump scaling factor |
| C_e | = 0.10 \$/kWh | Cost of electricity |
| C_r | = 0.35 k\$/m ² yr | Membrane replacement and cleaning cost |
| t_y | = 5000 h/yr | Annual operating time |
| i_r | = 0.08 | Interest rate |
| l_f | = 10 yr | Lifetime |

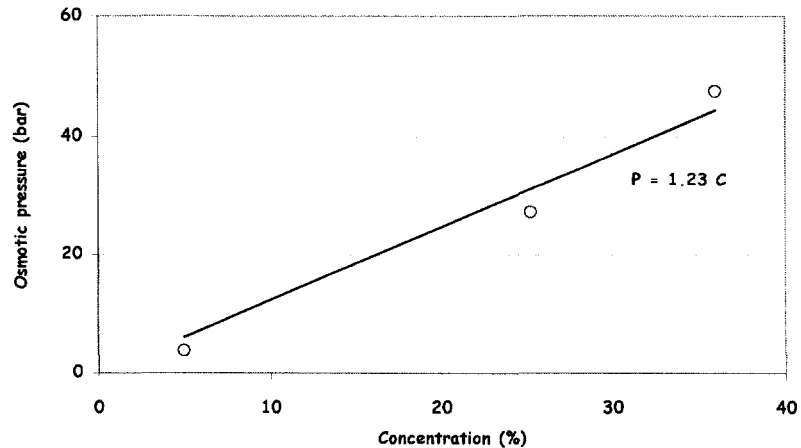
**Figure 10.17** Osmotic pressure of lactose.

Table 10.15 Process Design Results

Feed

| | | |
|-------|--------------|--------------------------|
| V_F | = 5.00 ton/h | Flow rate |
| C_F | = 5.00 % | Concentration of lactose |

Retentate

| | | |
|-------|--------------|---|
| V_R | = 1.07 ton/h | Flow rate |
| C_R | = 20.0 % | Concentration of lactose |
| C_W | = 25.7 % | Boundary layer concentration of lactose |

Permeate

| | | |
|-------|--------------|--------------------------|
| V_P | = 3.93 ton/h | Flow rate |
| C_P | = 0.90 % | Concentration of lactose |

Membrane

| | | |
|------------|-----------------------|-------------------------|
| S | = 76.8 m ² | Membrane area |
| d | = 1 mm | Hollow fiber diameter |
| L | = 2 m | Hollow fiber length |
| N | = 12200 | Number of fibers |
| u | = 0.5 m/s | Fluid velocity in fiber |
| P_F | = 60.0 bar | Operating pressure |
| P_O | = 31.6 bar | Osmotic pressure |
| ΔP | = 1 bar | Pressure loss |

Pumps

| | | |
|-------|----------|--------------------|
| E_F | = 8.2 kW | Feed pump power |
| E_R | = 0.5 kW | Recycle pump power |

Cost

| | | |
|----------|---------------|------------------|
| C_{eq} | = 81.1 k\$ | Equipment |
| C_{op} | = 31.2 k\$/yr | Operating |
| TAC | = 43.3 k\$/yr | Total annualized |

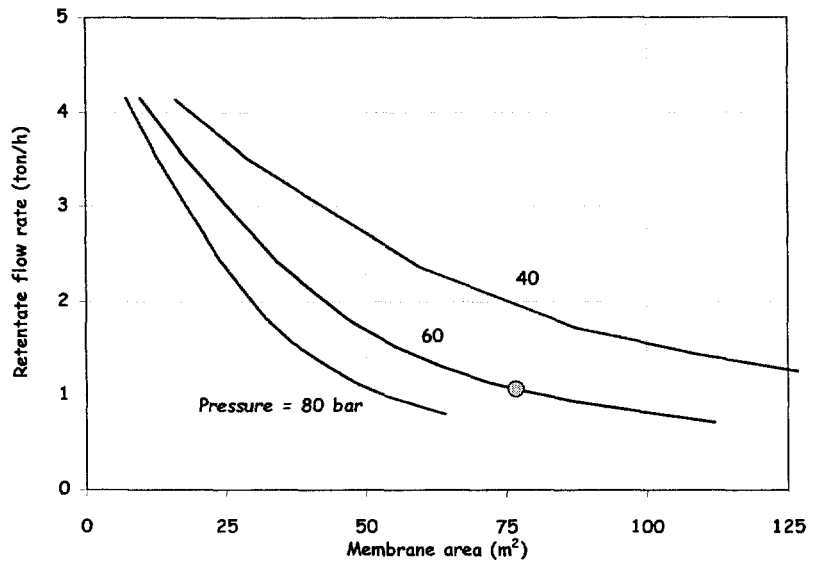


Figure 10.18 Retentate flow rate versus RO membrane area at different pressures.

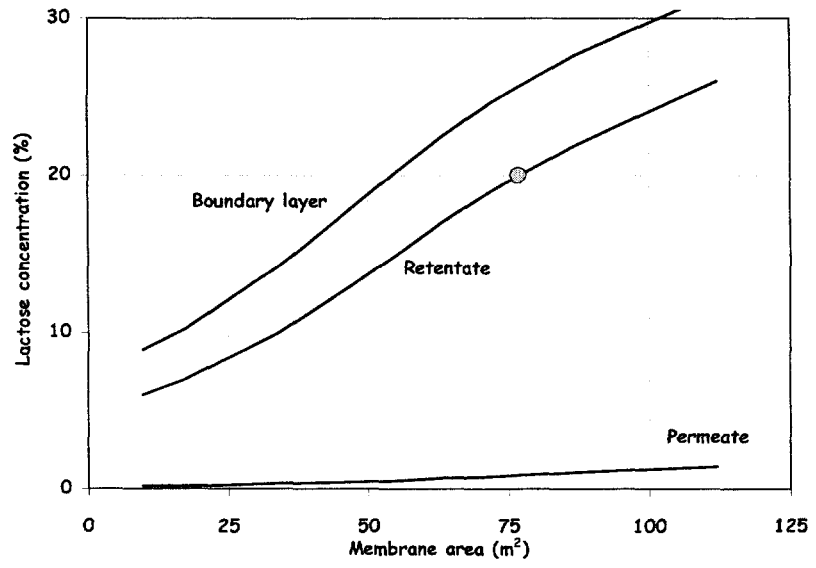


Figure 10.19 Lactose concentrations versus RO membrane area, operating at 60 bar.

NOMENCLATURE

| | | |
|------------|----------------------------------|---|
| A | m/s bar | Water permeability of the membrane |
| B_l | m/s | Lactose permeability |
| C_e | \$/kWh | Cost of electricity |
| C_{Fj} | % | Feed concentration of j component |
| C_{Rj} | % | Retentate concentration of j component |
| C_m | \$/m ² | Membrane unit cost |
| C_p | \$/kW | Pump unit cost |
| C_{Pj} | % | Permeate concentration of j component |
| C_r | \$/m ² yr | Membrane replacement and cleaning cost |
| C_{Rj} | % | Retentate concentration j component |
| C_{Wj} | % | Wall concentration j component |
| d | m | Hollow fiber diameter |
| E_F | kW | Feed pump power |
| E_R | kW | Recycle pump power |
| f | – | Friction loss coefficient |
| i_r | – | Interest rate |
| J_p | m ³ /s/m ² | Water flux through membrane |
| k_l | m/s | Boundary layer mass transfer coefficient of lactose |
| L | m | Hollow fiber length |
| l_f | yr | Lifetime |
| N | – | Number of fibers |
| n_m | – | Membrane scaling factor |
| n_p | – | Pump scaling factor |
| p | bar/% | Osmotic pressure coefficient |
| P_A | bar | Atmospheric pressure |
| P_F | bar | Operating pressure |
| P_O | bar | Osmotic pressure |
| R_f | – | Rejection parameter of fat |
| R_l | – | Rejection parameter of lactose |
| R_p | – | Rejection parameter of protein |
| R_s | – | Rejection parameter of salts |
| S | m ² | Membrane area |
| t_y | h/yr | Annual operating time |
| u | m/s | Fluid velocity in fiber |
| V | kg/s | Recycle flow rate |
| V_F | kg/s | Feed flow rate |
| V_P | kg/s | Permeate flow rate |
| V_R | kg/s | Retentate flow rate |
| wb | – | Wet basis |
| ΔP | bar | Pressure loss |
| ρ | kg/m ³ | Water density |

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Appendix

The techniques used throughout this book require physical properties and technical data in the form of mathematical equations instead of tabulated or graphical format. Thus, the purpose of this appendix is to provide the appropriate equations for the most common thermophysical properties and technical data required in food process design. The proposed equations have been fitted to the available data retrieved from the related literature. In all figures throughout this Appendix, the real data from the literature are presented with points, while the fitted equations are presented with continuous curves.

The data for the properties of water, steam and air (Section II) are retrieved mainly from Pakowski et al. (1991), Perry and Green (1997) and Saravacos and Maroulis (2001). The thermal properties of major food components (Section III) are based on the work of Choi and Okos (1983). The transport properties of selected foods (Section IV) are from Saravacos and Maroulis (2001). The fluid flow and convection heat transfer analysis (Sections V and VI, respectively) can be found in textbooks on fluid flow and heat transfer, such as McCabe et al. (2001) and Levenspiel (1986).

I. CONVERSION FACTORS TO SI UNITS

| From | To (SI) Units | Multiply by |
|----------------------------|--------------------|------------------------|
| atm (760 Torr) | bar | 1.013 |
| bar | Pa | 1×10^5 |
| Btu | kJ | 1.055 |
| Btu/h ft °F | W/m K | 1.729 |
| Btu/h ft ² | W/m ² | 3.154 |
| Btu/h ft ² °F | W/m ² | 5.678 |
| Btu/h | W | 0.293 |
| Btu/lb | kJ/kg | 2.326 |
| cP | Pa s | 0.001 |
| cuft | m ³ | 0.0284 |
| cuft/lb | m ³ /kg | 0.0624 |
| cuft/min (CFM) | m ³ /s | 0.5×10^{-3} |
| ft lb | J | 1.355 |
| ft of water | Pa | 2990 |
| ft | m | 0.305 |
| ft/min (FPM) | m/s | 0.0051 |
| gallons (Imperial) | m ³ | 4.543×10^{-3} |
| gallons (US) | m ³ | 3.785×10^{-3} |
| gallons/min (GPM) | m ³ /s | 0.063×10^{-3} |
| HP (boiler) | kW | 9.80 |
| HP | kW | 0.745 |
| in (inches) | m | 0.0254 |
| in Hg | Pa | 3386 |
| kcal | kJ | 4.18 |
| kg force (kp) | N | 9.81 |
| L (lit, lt, l) | m ³ | 0.001 |
| lb force | N | 4.45 |
| lb mass | kg | 0.454 |
| lb/cuft | kg/m ³ | 16.02 |
| lb/ft s | Pa s | 1.488 |
| lb/ft ² | Pa | 47.9 |
| lb/inch ² (psi) | Pa | 6894 |
| miles | km | 1.609 |
| mm water | Pa | 9.81 |
| P (poise) | Pa s | 0.10 |
| RPM (rpm) | 1/s | 1/60 |
| sq ft (ft ²) | m ² | 0.093 |
| sq in (inch ²) | m ² | 0.645×10^{-3} |
| ton (metric) | kg | 1 000 |
| ton (US, short) | kg | 907.2 |
| ton-refrigeration | kW | 3.51 |
| Torr (mm Hg) | Pa | 133.3 |

K = °C + 273.15

°C = (°F - 32)/1.8

Prefixes: k (kilo) = 10^3 , M (mega) = 10^6 , G (giga) = 10^9
m (milli) = 10^{-3} , μ (micro) = 10^{-6} , n (nano) = 10^{-9}

II. PHYSICAL PROPERTIES OF WATER, STEAM, AND AIR

1. Saturated Pressure of Water

Table A.1 Saturated Pressure of Water

| | |
|-----------------------------------|-----------------------------|
| $P_s = a_1 - \frac{a_2}{a_3 + T}$ | Antoine equation |
| P_s bar | Saturated pressure of water |
| T °C | Temperature |
| $T_t < T < T_c$ | Range of application |
| $T_t = 0.01$ °C | Triple point temperature |
| $T_c = 374.14$ °C | Critical temperature |
| $a_1 = 1.19 \cdot 10^1$ | Antoine constants |
| $a_2 = 3.95 \cdot 10^3$ | |
| $a_3 = 2.32 \cdot 10^2$ | |

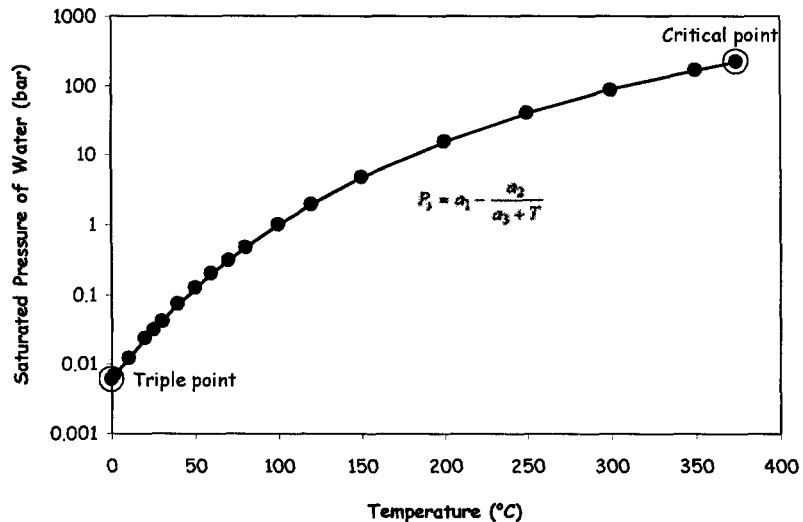


Figure A.1 Saturated pressure of water.

2. Latent Heat of Vaporization of Water

Table A.2 Latent Heat of Vaporization of Water

Equation 1

$$\Delta H = \Delta H_o - (Cp_w - Cp_v)T$$

0–150°C

Range of application

ΔH MJ/kg

Latent heat of vaporization

T °C

Temperature

$\Delta H_o = 2.50$ MJ/kg

Latent heat of vaporization at 0°C

$Cp_w = 4.20$ kJ/kgK

Average specific heat of water

$Cp_v = 1.87$ kJ/kgK

Average specific heat of water vapor

Equation 2

$$\Delta H = \Delta H_o \left(\frac{T_c - T}{T_c} \right)^{1/3}$$

$T_t < T < T_c$

Range of application

ΔH MJ/kg

Latent heat of vaporization

T °C

Temperature

$\Delta H_o = 2.50$ MJ/kg

Latent heat of vaporization at 0°C

$T_t = 0.01$ °C

Triple point temperature

$T_c = 374.14$ °C

Critical temperature

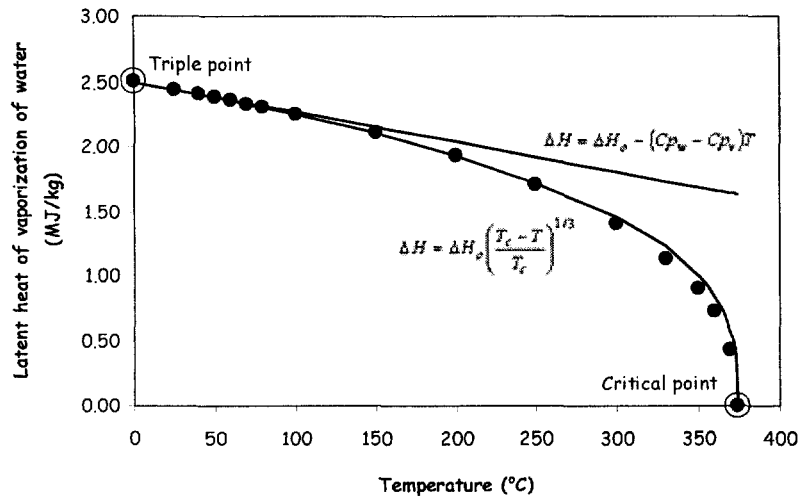


Figure A.2 Latent heat of vaporization of water.

3. Density of Water

Table A.3 Density of Water

$$\rho = a_0 + a_1T + a_2T^2$$

| | | |
|--------|-------------------|-------------|
| ρ | kg/m ³ | Density |
| T | °C | Temperature |

| | | |
|-------|--------------------------|--------------------------------|
| a_0 | = 9.97 10 ² | Constants in the range 0–150°C |
| a_1 | = 3.14 10 ⁻³ | |
| a_2 | = -3.76 10 ⁻³ | |

| | | |
|-------|--------------------------|--------------------------------|
| a_0 | = 9.95 10 ² | Constants in the range 0–350°C |
| a_1 | = 2.91 10 ⁻² | |
| a_2 | = -3.40 10 ⁻³ | |

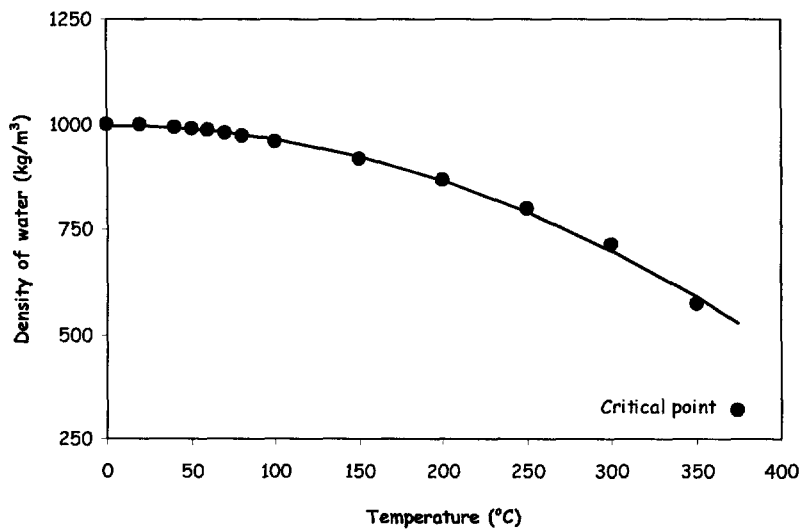


Figure A.3 Density of water.

4. Density of Saturated Steam

Table A.4 Density of Saturated Steam

| | |
|--|-----------------------------|
| $\rho = \frac{M P_s}{R T_a}$ | Ideal gas equation |
| where $P_s = a_1 - \frac{a_2}{a_3 + T}$ | Antoine equation |
| and $T_a = T + 273.15$ | Absolute temperature |
| 0–250°C | Range of application |
| ρ kg/m ³ | Density |
| P_s bar | Saturated pressure of water |
| T °C | Temperature |
| T_a K | Absolute temperature |
| $R = 0.083143$ m ³ bar/kmol K | Ideal gas constant |
| $M = 18.015$ kg/kmol | Molecular weight |
| $a_1 = 1.19 \cdot 10^1$ | Antoine constants |
| $a_2 = 3.95 \cdot 10^3$ | |
| $a_3 = 2.32 \cdot 10^2$ | |

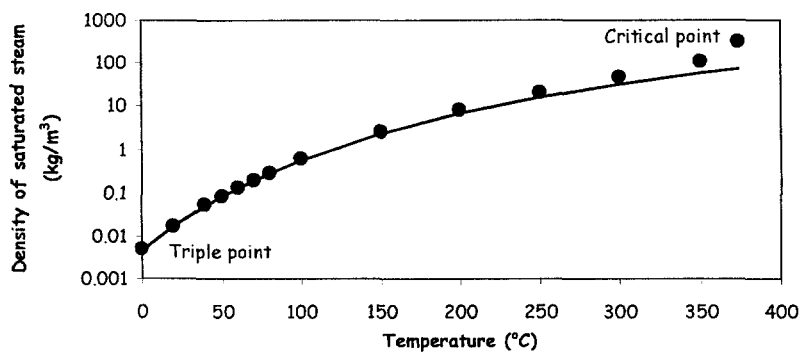


Figure A.4 Density of saturated steam versus temperature.

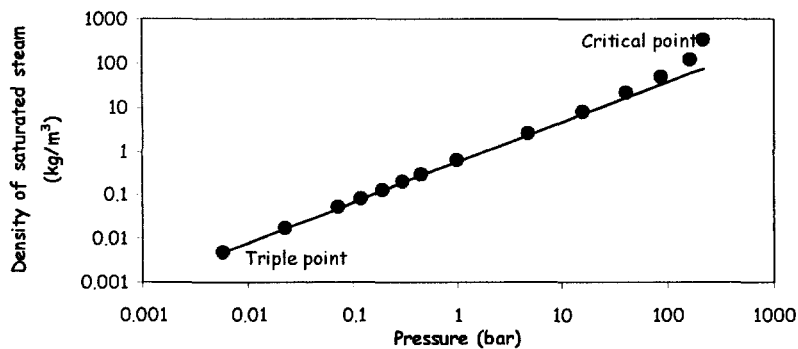


Figure A.5 Density of saturated steam versus pressure.

5. Phase Diagram of Water

Table A.5 Clapeyron Equation for Ice Fusion

$$P = P_o + \frac{\Delta H_f}{\Delta V_f} \ln \left(\frac{273.15 + T}{273.15 + T_o} \right)$$

where

| | | |
|-------|---------|----------------------------|
| P | bar | Pressure |
| T | °C | Temperature |
| P_o | = 1 bar | Normal pressure |
| T_o | = 0 °C | Normal melting temperature |

and

$$\frac{\Delta H_f}{\Delta V_f} = -2.71516 \times 10^6 \text{ bar}$$

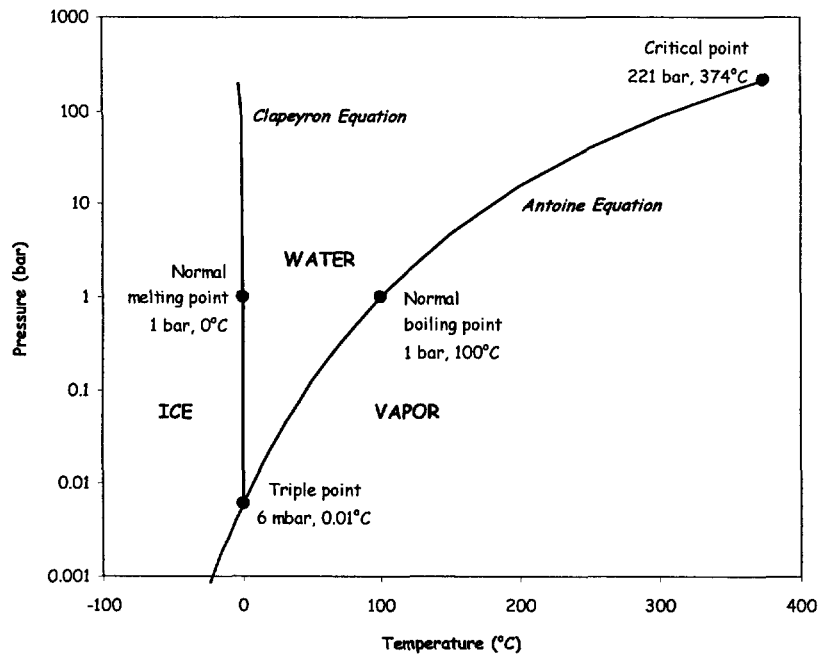


Figure A.6 Phase diagram of water.

6. Density of Air

Table A.6 Density of Air

$$\rho = \frac{MP}{R T_a} \quad \text{Ideal gas equation}$$

where $T_a = T + 273.15$ Absolute temperature

ρ kg/m³ Density
 P bar Pressure
 T °C Temperature
 T_a K Absolute temperature

$R = 0.083143 \text{ m}^3\text{bar/kmol K}$ Ideal gas constant

$M = 18.015 \text{ kg/kmol}$ Molecular weight

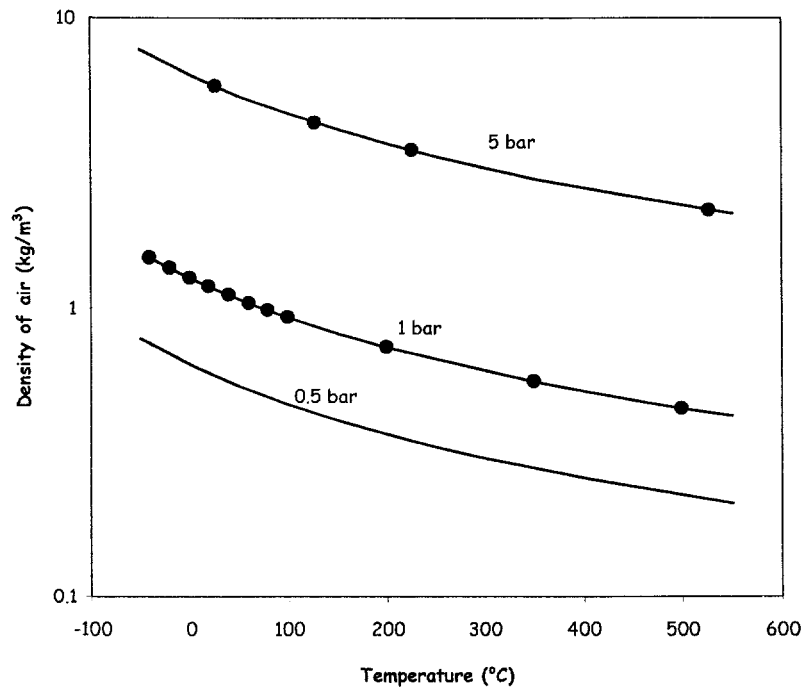


Figure A.7 Density of air.

7. Specific Heat of Water, Steam and Air

Table A.7 Specific Heat of Water, Steam and Air

$$C_p = a_0 + a_1T + a_2T^2$$

| C_p | kJ/kgK | Specific heat |
|-------|-------------------------|---------------------|
| T | °C | Temperature |
| a_0 | = 4.21 | Constants for water |
| a_1 | = $-1.35 \cdot 10^{-3}$ | |
| a_2 | = $1.38 \cdot 10^{-5}$ | |
| a_0 | = 1.87 | Constants for steam |
| a_1 | = $3.07 \cdot 10^{-4}$ | |
| a_2 | = $5.66 \cdot 10^{-7}$ | |
| a_0 | = 1.01 | Constants for vapor |
| a_1 | = $3.16 \cdot 10^{-5}$ | |
| a_2 | = $3.28 \cdot 10^{-7}$ | |

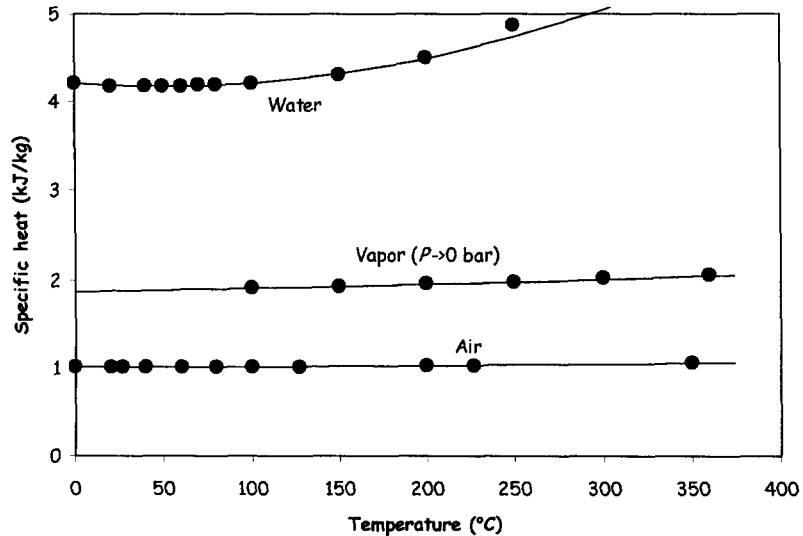


Figure A.8 Specific heat of water, steam and air.

8. Viscosity of Water, Steam and Air

Table A.8 Viscosity of Water, Steam and Air

$$\eta = a_0 + a_1T + a_2T^2 + a_3T^3$$

| | | |
|--------|-------|---------------|
| η | mPa s | Specific heat |
| T | °C | Temperature |

| | | |
|-------|------------------------|--|
| a_0 | $= 7.59 \cdot 10^{-3}$ | Constants for saturated vapor in the range 0–300°C |
|-------|------------------------|--|

| | |
|-------|------------------------|
| a_1 | $= 4.49 \cdot 10^{-5}$ |
|-------|------------------------|

| | |
|-------|-------------------------|
| a_2 | $= -6.13 \cdot 10^{-8}$ |
|-------|-------------------------|

| | |
|-------|-------------------------|
| a_3 | $= 1.44 \cdot 10^{-10}$ |
|-------|-------------------------|

| | | |
|-------|------------------------|--|
| a_0 | $= 8.07 \cdot 10^{-3}$ | Constants for superheated vapor in the range 100–700°C |
|-------|------------------------|--|

| | |
|-------|------------------------|
| a_1 | $= 4.04 \cdot 10^{-5}$ |
|-------|------------------------|

| | |
|-------|------------------------|
| a_2 | $= 1.24 \cdot 10^{-9}$ |
|-------|------------------------|

| | |
|-------|--------------------------|
| a_3 | $= -1.21 \cdot 10^{-12}$ |
|-------|--------------------------|

| | | |
|-------|------------------------|---|
| a_0 | $= 1.69 \cdot 10^{-2}$ | Constants for air in the range 0–1000°C |
|-------|------------------------|---|

| | |
|-------|------------------------|
| a_1 | $= 4.98 \cdot 10^{-5}$ |
|-------|------------------------|

| | |
|-------|-------------------------|
| a_2 | $= -3.19 \cdot 10^{-8}$ |
|-------|-------------------------|

| | |
|-------|-------------------------|
| a_3 | $= 1.32 \cdot 10^{-11}$ |
|-------|-------------------------|

$$\eta = a_0 + a_1T + a_2T^2 + a_3T^{-1}$$

| | | |
|-------|----------------------|--|
| a_0 | $= -1.07 \cdot 10^1$ | Constants for saturated water in the range 0–350°C |
|-------|----------------------|--|

| | |
|-------|------------------------|
| a_1 | $= 1.97 \cdot 10^{-2}$ |
|-------|------------------------|

| | |
|-------|-------------------------|
| a_2 | $= -1.47 \cdot 10^{-5}$ |
|-------|-------------------------|

| | |
|-------|---------------------|
| a_3 | $= 1.82 \cdot 10^3$ |
|-------|---------------------|

9. Thermal Conductivity of Water, Steam and Air

Table A.9 Thermal Conductivity of Water, Steam and Air

$$\lambda = a_0 + a_1T + a_2T^2 + a_3T^3$$

| λ | W/m K | Specific heat |
|-----------|--------------------------|--|
| T | °C | Temperature |
| a_0 | $= 5.70 \cdot 10^{-1}$ | Constants for saturated water in the range 0–350°C |
| a_1 | $= 1.78 \cdot 10^{-3}$ | |
| a_2 | $= -6.94 \cdot 10^{-6}$ | |
| a_3 | $= 2.20 \cdot 10^{-9}$ | |
| a_0 | $= 1.76 \cdot 10^{-2}$ | Constants for saturated vapor in the range 0–300°C |
| a_1 | $= 1.05 \cdot 10^{-4}$ | |
| a_2 | $= -6.71 \cdot 10^{-7}$ | |
| a_3 | $= 3.07 \cdot 10^{-9}$ | |
| a_0 | $= 1.77 \cdot 10^{-2}$ | Constants for superheated vapor in the range 100–700°C |
| a_1 | $= 6.01 \cdot 10^{-5}$ | |
| a_2 | $= 9.51 \cdot 10^{-8}$ | |
| a_3 | $= -3.99 \cdot 10^{-11}$ | |
| a_0 | $= 2.43 \cdot 10^{-2}$ | Constants for air in the range 0–1000°C |
| a_1 | $= 7.89 \cdot 10^{-5}$ | |
| a_2 | $= -1.79 \cdot 10^{-8}$ | |
| a_3 | $= -8.57 \cdot 10^{-12}$ | |

10. Mass Diffusivity of Water Vapor in Air

Table A.10 Mass Diffusivity of Water Vapor in Air

$$D = a_0 \left(\frac{T + 273}{273} \right)^{a_1} P^{a_2}$$

| D | m ² /s | Mass diffusivity |
|-------|------------------------|---|
| T | °C | Temperature |
| P | bar | Pressure |
| a_0 | $= 2.16 \cdot 10^{-5}$ | Constants for saturated vapor in the range 0–1200°C |
| a_1 | $= 1.80$ | |
| a_2 | $= -1.00$ | |

III. THERMAL PROPERTIES OF MAJOR FOOD COMPONENTS

1. Density of Major Food Components

Table A.11 Density of Major Food Components

$$\rho = \frac{1}{\sum \frac{X_i}{\rho_i}}$$

$$\rho_i = b_0 + b_1T + b_2T^2$$

| | | |
|----------|-------------------|---------------------------------------|
| ρ | kg/m ³ | Density of food |
| ρ_i | kg/m ³ | Density of component <i>i</i> |
| X_i | kg/kg | Weight fraction of component <i>i</i> |
| T | °C | Temperature |

| Component | b_0 | b_1 | b_2 |
|--------------|----------|-----------|-----------|
| Protein | 1.33E+03 | -5.18E-01 | |
| Fat | 9.26E+02 | -4.18E-01 | |
| Carbohydrate | 1.60E+03 | -3.10E-02 | |
| Fiber | 1.31E+03 | -3.66E-01 | |
| Ash | 2.42E+03 | -2.81E-01 | |
| Water | 9.97E+02 | 3.14E-03 | -3.76E-03 |
| Ice | 9.17E+02 | -1.31E-01 | |

2. Specific Heat of Major Food Components

Table A.12 Specific Heat of Major Food Components

$$C_p = \sum X_i C_{pi}$$

$$C_{pi} = b_0 + b_1 T + b_2 T^2$$

| | | |
|----------|---------|----------------------------------|
| C_p | kJ/kg K | Specific heat of food |
| C_{pi} | kJ/kg K | Specific heat of component i |
| X_i | kg/kg | Weight fraction of component i |
| T | °C | Temperature |

| Component | b_0 | b_1 | b_2 |
|----------------------|----------|-----------|-----------|
| Protein | 2.01E+00 | 1.21E-03 | -1.31E-06 |
| Fat | 1.98E+00 | 1.47E-03 | -4.80E-06 |
| Carbohydrate | 1.55E+00 | 1.96E-03 | -5.94E-06 |
| Fiber | 1.85E+00 | 1.83E-03 | -4.65E-06 |
| Ash | 1.09E+00 | 1.89E-03 | -3.68E-06 |
| Water | 4.18E+00 | -9.09E-05 | 5.47E-06 |
| Ice | 2.06E+00 | 6.08E-03 | |
| Water (below 0°C) | 4.08E+00 | -5.31E-03 | 9.95E-04 |

3. Thermal Conductivity of Major Food Components

Table A.13 Thermal Conductivity of Major Food Components

$$\lambda = \sum Y_i \lambda_i$$

$$\lambda_i = b_0 + b_1 T + b_2 T^2$$

| | | |
|-------------|--------------------------------|---------------------------------------|
| λ | W/m K | Thermal conductivity of food |
| λ_i | W/m K | Thermal conductivity of component i |
| Y_i | m ³ /m ³ | Volume fraction of component i |
| T | °C | Temperature |

| Component | b_0 | b_1 | b_2 |
|--------------|----------|-----------|-----------|
| Protein | 1.79E-01 | 1.20E-03 | -2.72E-06 |
| Fat | 1.81E-01 | -2.76E-03 | -1.77E-07 |
| Carbohydrate | 2.01E-01 | 1.39E-03 | -4.33E-06 |
| Fiber | 1.83E-01 | 1.25E-03 | -3.17E-06 |
| Ash | 3.30E-01 | 1.40E-03 | -2.91E-06 |
| Water | 5.71E-01 | 1.76E-03 | -6.70E-06 |
| Ice | 2.22E+00 | -6.25E-03 | 1.02E-04 |

IV. TRANSPORT PROPERTIES OF SELECTED FOODS

1. Viscosity of Selected Foods

Table A.14 Apparent Viscosity of non-Newtonian Fluid Foods

Mathematical Model

$$\eta = K \left(\frac{8u}{d} \right)^{n-1}$$

$$K = K_o \exp(BX) \exp \left[\frac{E}{R} \left(\frac{1}{T} - \frac{1}{T_o} \right) \right]$$

$$n = n_o - bX$$

| | | |
|--------|-------------------|-------------------------|
| η | Pa s | Apparent viscosity |
| K | Pa s ⁿ | Consistency coefficient |
| n | – | Flow behavior index |
| u | m/s | Fluid velocity |
| d | m | Equivalent diameter |
| X | kg/kg | Concentration |
| T | K | Temperature |
| T_o | K | Reference temperature |
| R | | Ideal gas constant |

Adjustable Model Parameters

K_o
 B
 E
 n_o
 b

Table A.15 Data for Estimating Apparent Viscosity of Selected Fluid Foods Using the Model in Table A.14

| Material | K_o , Pa s ⁿ | B | E , kJ/mol | n_o | b |
|-------------|---------------------------|-------|--------------|-------|--------|
| Tomato | 1.27 | 0.149 | 15.8 | 0.403 | 0.0028 |
| Orange | 9.28 | 0.077 | 35.0 | 0.950 | 0.0034 |
| Pear | 2.15 | 0.087 | 16.1 | 0.348 | 0.0000 |
| Apple sauce | 26.4 | | | 0.286 | |

2. Thermal Conductivity of Selected Foods

Table A.16 Mathematical Model for Calculating Thermal Conductivity in Foods as a Function of Moisture Content and Temperature

Mathematical Model

$$\lambda = \frac{1}{1+X} \lambda_o \exp\left[-\frac{E_o}{R}\left(\frac{1}{T} - \frac{1}{T_r}\right)\right] + \frac{X}{1+X} \lambda_i \exp\left[-\frac{E_i}{R}\left(\frac{1}{T} - \frac{1}{T_r}\right)\right]$$

λ (W/mK) the thermal conductivity,

X (kg/kg db) the material moisture content,

T (K) the material temperature,

$T_r = 60^\circ\text{C}$ a reference temperature, and

$R = 0.0083143$ kJ/mol K the ideal gas constant.

Adjustable Model Parameters

- λ_o (W / mK) thermal conductivity at moisture $X = 0$ and temperature $T = T_r$
 - λ_i (W / mK) thermal conductivity at moisture $X = \infty$ and temperature $T = T_r$
 - E_o (kJ / mol) activation energy for heat conduction in dry material at $X = 0$
 - E_i (kJ / mol) activation energy for heat conduction in wet material at $X = \infty$
-

Table A.17 Data for Estimating Thermal Conductivity of Selected Foods Using the Model in Table A.16

| Material | λ_i (W/m K) | λ_o (W/m K) | E_i (kJ/mol) | E_o (kJ/mol) |
|-------------------------------|------------------------|------------------------|-------------------|-------------------|
| <u>Cereal products</u> | | | | |
| Corn | 1.580 | 0.070 | 7.2 | 5.0 |
| <u>Fruits</u> | | | | |
| Apple | 0.589 | 0.287 | 2.4 | 11.7 |
| Orange | 0.642 | 0.106 | 1.3 | 0.0 |
| Pear | 0.658 | 0.270 | 2.4 | 1.9 |
| <u>Model foods</u> | | | | |
| Amioca | 0.718 | 0.120 | 3.2 | 14.4 |
| Starch | 0.623 | 0.243 | 0.3 | 0.4 |
| Hylon | 0.800 | 0.180 | 9.9 | |
| <u>Vegetables</u> | | | | |
| Potato | 0.611 | 0.049 | 0.0 | 47.0 |
| Tomato | 0.680 | 0.220 | 0.2 | 5.0 |
| <u>Dairy</u> | | | | |
| Milk | 0.665 | 0.212 | 1.7 | 1.9 |
| <u>Meat</u> | | | | |
| Beef | 0.568 | 0.280 | 2.2 | 3.2 |
| <u>Other</u> | | | | |
| Rapeseed | 0.239 | 0.088 | 3.6 | 0.6 |
| <u>Baked products</u> | | | | |
| Dough | 0.800 | 0.273 | 2.7 | 0.0 |

3. Moisture Diffusivity of Selected Foods

Table A.18 Mathematical Model for Calculating Moisture Diffusivity in Foods as a Function of Moisture Content and Temperature

Mathematical Model

$$D = \frac{1}{1+X} D_o \exp\left[-\frac{E_o}{R}\left(\frac{1}{T} - \frac{1}{T_r}\right)\right] + \frac{X}{1+X} D_i \exp\left[-\frac{E_i}{R}\left(\frac{1}{T} - \frac{1}{T_r}\right)\right]$$

| | |
|--------------------------|--------------------------------|
| D (m ² /s) | the moisture diffusivity, |
| X (kg/kg db) | the material moisture content, |
| T (K) | the material temperature, |
| $T_r = 60^\circ\text{C}$ | a reference temperature, and |
| $R = 0.0083143$ kJ/mol K | the ideal gas constant. |

Adjustable Model Parameters

- D_o (m²/s) diffusivity at moisture $X = 0$ and temperature $T = T_r$,
 - D_i (m²/s) diffusivity at moisture $X = \infty$ and temperature $T = T_r$,
 - E_o (kJ/mol) activation energy for diffusion in dry material at $X = 0$
 - E_i (kJ/mol) activation energy for diffusion in wet material at $X = \infty$
-

Table A.19 Date for Estimating Moisture Diffusivity of Selected Foods Using the Model in Table A.18

| Material | D_i (m^2/s) | D_o (m^2/s) | E_i (kJ/mol) | E_o (kJ/mol) |
|--------------------------|--|--|--------------------------------------|--------------------------------------|
| Cereal products | | | | |
| <u>Corn</u> | | | | |
| - | 4.40E-09 | 0.00E+00 | 0.0 | 10.4 |
| <i>deni</i> | 1.19E-08 | 0.00E+00 | 49.4 | 73.1 |
| <i>grains</i> | 1.15E-09 | 6.66E-11 | 10.2 | 57.8 |
| <i>kernel</i> | 5.87E-10 | 5.32E-10 | 0.0 | 33.8 |
| <i>pericarp</i> | 1.13E-09 | 0.00E+00 | 10.0 | 5.0 |
| <u>Pasta</u> | | | | |
| - | 1.39E-09 | 0.00E+00 | 16.2 | 2.0 |
| <u>Rice</u> | | | | |
| <i>kernel</i> | 9.75E-09 | 0.00E+00 | 12.5 | 2.0 |
| <u>Rough rice</u> | | | | |
| - | 2.27E-09 | 0.00E+00 | 12.7 | 0.7 |
| <u>Wheat</u> | | | | |
| - | 1.94E-09 | 1.30E-09 | 0.0 | 46.3 |
| Fruits | | | | |
| <u>Apple</u> | | | | |
| - | 7.97E-10 | 1.16E-10 | 16.7 | 56.6 |
| <u>Banana</u> | | | | |
| - | 2.03E-09 | 4.66E-10 | 9.9 | 4.6 |
| <u>Grapes</u> | | | | |
| <i>seedless</i> | 5.35E-09 | 0.00E+00 | 34.0 | 10.4 |
| <u>Raisins</u> | | | | |
| - | 8.11E-10 | 1.05E-10 | 21.4 | 50.1 |
| Model foods | | | | |
| <u>Amioca</u> | | | | |
| - | 1.52E-08 | 1.52E-08 | 0.0 | 33.3 |
| <u>Hylon-7</u> | | | | |
| - | 1.96E-08 | 1.96E-08 | 0.0 | 24.2 |
| Vegetables | | | | |
| <u>Carrot</u> | | | | |
| - | 2.47E-09 | 1.54E-09 | 13.9 | 11.3 |
| <u>Garlic</u> | | | | |
| - | 5.33E-10 | 1.68E-11 | 15.4 | 7.1 |
| <u>Onion</u> | | | | |
| - | 1.45E-08 | 0.00E+00 | 70.2 | 10.4 |
| <u>Potato</u> | | | | |
| - | 1.57E-09 | 4.31E-10 | 44.7 | 76.9 |

V. FLUID FLOW IN CIRCULAR TUBES

Table A.20 Fluid Flow in Circular Tubes

$$\Delta p = 4f \left(\frac{L}{d} \right) \left(\frac{\rho u^2}{2} \right)$$

$$Re = \frac{\rho d u}{\eta}$$

$$f = \begin{cases} 16 Re^{-1} & \text{for laminar flow (Re < 2100)} \\ 0.08 Re^{-1/4} & \text{for turbulent flow (Re > 2100) in smooth tubes} \\ 0.017 & \text{for turbulent flow (Re > 2100) in very rough tubes (pipes)} \end{cases}$$

$$E = n F \Delta p$$

| | | |
|------------|-------------------|------------------------------|
| Δp | Pa | Friction pressure loss |
| L | m | Length of tube |
| d | m | Diameter of the tube |
| u | m/s | Superficial velocity |
| ρ | kg/m ³ | Density of fluid |
| η | Pa s | Apparent viscosity of fluid |
| Re | – | Reynolds number |
| f | – | Fanning friction coefficient |
| F | m ³ /s | Fluid flow rate |
| n | – | Pump efficiency |
| E | W | Pumping power |

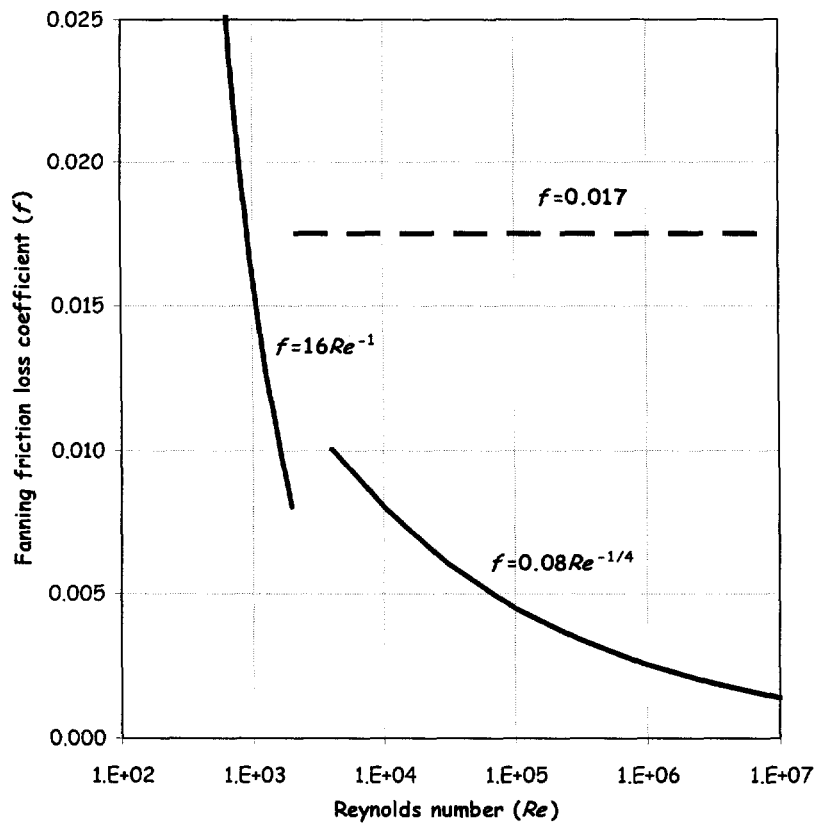


Figure A.9 Friction coefficient versus Reynolds number.

VI. CONVECTION HEAT TRANSFER COEFFICIENTS

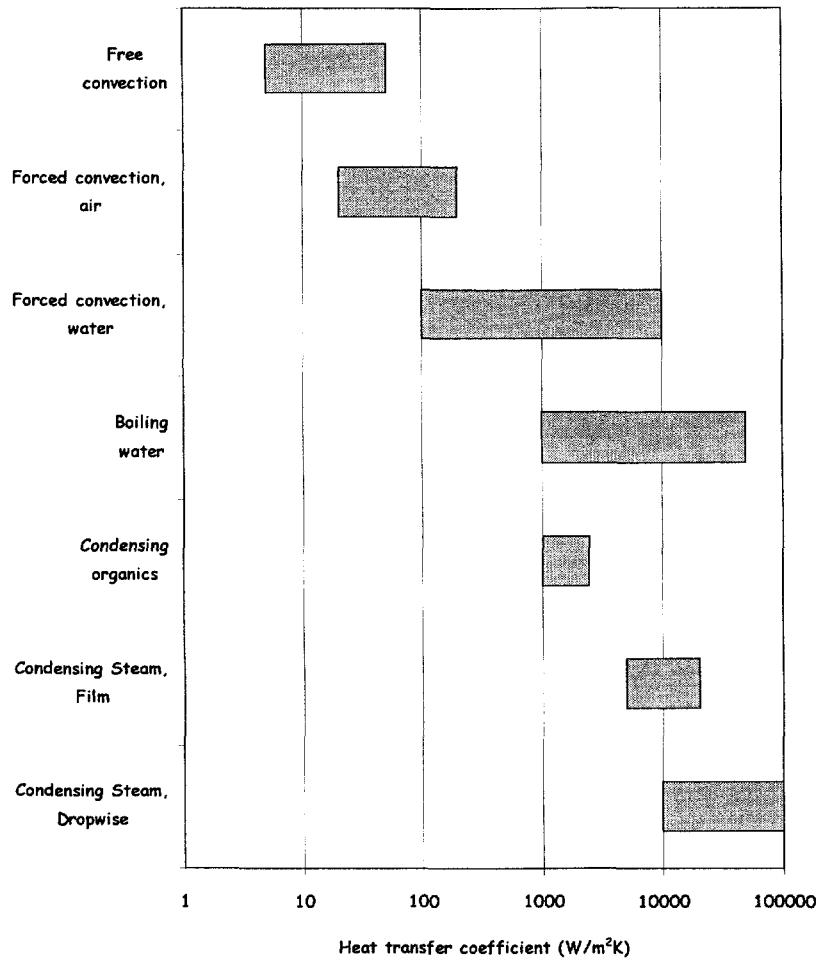


Figure A.10 Heat transfer coefficients.

Table A.21 Convection Heat Transfer in Circular Tubes

Laminar flow

$$Nu = 1.86 \left(\frac{d}{L} \right)^{1/3} (RePr)^{1/3} \phi^{0.14}$$

Turbulent flow

$$Nu = 0.023 \left[1 + \left(\frac{d}{L} \right)^{0.7} \right] Re^{0.8} Pr^{1/3} \phi^{0.14}$$

where

$$Nu = \frac{hd}{\lambda} \quad \text{Nusselt number}$$

$$Re = \frac{\rho du}{\eta} \quad \text{Reynolds number}$$

$$Pr = \frac{C_p \eta}{\lambda} \quad \text{Prandtl number}$$

$$\phi = \frac{\eta}{\eta_w} \quad \text{Correction factor for heating or cooling}$$

$$\frac{d}{L} \quad \text{Entrance effect}$$

and

| | | |
|-----------|--------------------|-------------------------------|
| h | W/m ² K | Heat transfer coefficient |
| d | m | Tube diameter |
| L | m | Tube length |
| u | m/s | Superficial velocity |
| ρ | kg/m ³ | Density |
| λ | W/m K | Thermal conductivity |
| η | Pa s | Viscosity at bulk temperature |
| η_w | Pa s | Viscosity at wall temperature |

NOMENCLATURE

| | | |
|------------|--------------------------------|---|
| C_p | kJ/kg K | Specific heat |
| d | m | Diameter of the tube |
| D | m ² /s | Mass diffusivity |
| E | kW | Pumping power |
| f | – | Fanning friction coefficient |
| F | m ³ /s | Fluid flow rate |
| h | W/m ² K | Heat transfer coefficient |
| K | Pa s ⁿ | Consistency coefficient |
| L | m | Length of tube |
| M | kg/kmol | Molecular weight |
| n | – | Flow behavior index |
| η | – | Pump efficiency |
| P | bar | Pressure |
| P_s | bar | Saturated pressure of water |
| R | m ³ bar/kmol K | Ideal gas constant |
| r | m | Hydraulic radius |
| Re | – | Reynolds number |
| T | °C, K | Temperature |
| T_c | °C, K | Critical temperature |
| T_o | °C, K | Reference temperature |
| T_i | °C, K | Triple point temperature |
| u | m/s | Superficial velocity |
| X_i | kg/kg | Weight fraction of component i |
| Y_i | m ³ /m ³ | Volume fraction of component i |
| Δp | Pa | Friction pressure loss |
| η | Pa s | Apparent viscosity of fluid |
| η_w | Pa s | Apparent viscosity of fluid at wall temperature |
| λ | W/m K | Thermal conductivity |
| ρ | kg/m ³ | Density |

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