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*Of experience*

## MODELLING AND DESIGN OF **CONVEYOR BELT DRYERS**



In Association with SVCH-Technologii, Moscow (Russia)  
ISO 9001:2015 | ISO 14001:2015 | ISO 45001:2018

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Enhance the value of customer operation through our customer need centric engineering solution.



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## ABSTRACT

A mathematical model suitable for the design of conveyor belt dryers was developed. The objective of design was the evaluation of optimum flowsheet structure, construction characteristics, and operational conditions. The methodology followed was based on constructing a superstructure which involves a large number of minor structures, and optimizing it by means of non-linear mathematical programming techniques. The optimization problem can be stated in an equivalent way so that the computational effort involved in its solution is greatly reduced. This decomposition strategy transforms the original problem into an optimization problem of a major and a minor stage.

## NOTATION

$\alpha_w$	Water activity of air stream leaving the product
$A$	Area of chamber belt ( $m^2$ )
$A_i$	Constants in eqn (26), $i=1,2,3$
$A_{MAX}$	Maximum constructed area of conveyor belt for each chamber ( $m^2$ )
$A_{ST}$	Area of heat exchanger ( $m^2$ )
$C_{CP}$	Capital annual cost (US\$/year)
$C_E$	Cost of electricity (US\$/kWh)
$C_{OP}$	Operational annual cost (US\$/year)
$CP_A$	Specific heat of air (J/kg K)
$CP_S$	Specific heat of dry solid (J/kg)
$CP_V$	Specific heat of vapor (J/kg K)
$CP_W$	Specific heat of water (J/kg)
$C_{ST}$	Cost of steam (US\$/kg)
$C_T$	Total annual cost (US\$/year)

$e$	Percentage of capital cost on an annual rate
$E$	The electrical power consumed for the operation of fans(kWh)
$F_A$	Flow rate of fresh air stream (kg/s dry basis)
$F_{AC}$	Flow rate of drying air stream (kg/s dry basis)
$F_S$	Flow rate of product stream (kg/s dry basis)
$F_{ST}$	Flow rate of steam (kg/s)
$h_A$	Specific enthalpy of an air stream as a function of its temperature and absolute humidity (J/kg)
$h_S$	Specific enthalpy of a product stream as a function of its temperature and moisture content (J/kg)
$k_M$	Drying constant as a function of drying air stream temperature, absolute humidity and velocity, and characteristic dimension of product particles (1/s)
$M$	The number of drying chambers in the superstructure
$n_i$	The number of drying chambers of drying section $i$
$n_j$	Constants in eqn (32), $i= S, D, F, ex$
$N$	The number of drying sections in the superstructure
$P$	Total pressure (Pa)
$P^0$	Water vapour pressure at saturation (Pa)
$Q$	Exchanged heat rate (W )
$r_c$	Characteristic dimension of product particles (m)
$t$	Residence time (s)
$t_{OP}$	Annual operating time (h/year)
$T_A$	Temperature of rejected air stream ( $^{\circ}C$ )

## INTRODUCTION

$T_{A0}$	Temperature of fresh air stream (°C)
$T_{AC}$	Temperature of drying air stream (°C)
$T_{AM}$	Temperature of mixed recirculation and fresh air streams (°C)
$T_s$	Temperature of product stream on leaving the chamber (°C)
$T_{s0}$	Temperature of product stream on entering the chamber (°C)
$T_{SMAX}$	Maximum temperature level, above which no thermal degradation effects are observed (°C)
$T_{ST}$	Temperature of steam (°C)
$U_{ST}$	Overall heat transfer coefficient (W/m <sup>2</sup> K)
$V_A$	Air velocity through product (m/s)
$X_A$	Absolute humidity of rejected air stream (kg/kg dry basis)
$X_{AC}$	Absolute humidity of drying air stream (kg/kg dry basis)
$X_{A0}$	Absolute humidity of fresh air stream (kg/kg dry basis)
$X_s$	Moisture content of product stream on leaving the chamber (kg/kg dry basis)
$X_{s0}$	Moisture content of product stream on entering the chamber (kg/kg dry basis)
$\alpha_j$	Constants in eqn (32), $j = S, D, F, ex$
$\beta_i$	Constants in eqn (25), $i = 0, 1, 2, 3, 4$
$\gamma_i$	Constants in eqns (22-24), $i = 0, 11, 12, 21, 22$
$\Delta H$	Latent heat of vaporization for water (J/kg)
$\Delta H_0$	Latent heat of vaporization of water at reference temperature (J/kg)
$\Delta P$	The mean pressure drop in the chamber (Pa)
$\Delta T$	The temperature difference of drying air stream on passing through the product (°C)
$\Delta T_{MAX}$	Maximum temperature difference of drying air stream through product (°C)
$\lambda_B$	Ratio of molecular weights of water and air (=0.62198)
$\rho_{s0}$	Belt load at the entrance (kg/m <sup>2</sup> wet basis)
$\rho_A$	Density of air (kg/m <sup>3</sup> )
$\rho_{SMAX}$	Maximum belt load (kg/m <sup>2</sup> wet basis)

Process design is principally a combination of structural and parametric optimization efforts, carried out on the flowsheet options available for the various stages of processing. In the case of dryers, design has become an increasingly challenging problem aiming at the evaluation of the proper flowsheet structure and the optimum construction characteristics and operating conditions of each unit in the overall design. However, most design efforts in this field face problems of extreme difficulty related to complex drying conditions that include many interconnected and opposing phenomena, chiefly related to the complicated nature of drying. Although the modelling of drying processes is well developed with adequate understanding of the process itself, most models incorporate a large number of thermophysical properties and transport coefficients, which in most cases are only imprecisely known, producing inaccurate or erroneous results on large-scale industrial applications. Furthermore, the case of appropriate structure determination is usually a complex problem, tackled solely by means of empirical or semiempirical methods.

Ahn et al. (1964) used dynamic programming techniques in order to study optimal air distribution patterns in crossflow grain dryers.

Thompson (1967) developed multivariable search techniques, based on single-dimensional search algorithms, for use in studying the optimal design of convection grain dryers. His study involved both single-stage crossflow and concurrent flow dryers. Farmer (1972) developed a dynamic programming algorithm for the single-staged concurrent flow dryer with a counterflow cooler. The objective function considered energy costs and used grain quality constraints. Thyngson and Grossmann (1970) presented a mathematical model for the modelling and optimization of a through-circulation packed bed dryer. Brook and Bakker-Arkema (1978) determined the optimum operational parameters and size of two-stage and three-stage concurrent flow grain dryers with intermediate tempering stages. The objective function was based on energy and capital costs. The operational parameters were constrained by the desired final moisture content and the maximum allowable value of important grain quality factors. Becker et al. (1984) used a simple process model applicable for microcomputer-based on-line applications in order to optimize the operation of the dryer on minimizing the specific drying cost. Bertin and Blazquez (1986) presented a mathematical model for a tunnel-dehydrator of the California type for plum drying, and optimized the production rate of the dryer.

Kaminski et al. (1989) used two methods of multi-objective optimization in order to analyse the process conditions of L-lysine drying in a fluidized bed dryer. Results obtained were compared to those of one-objective optimization.

Chen (1990) developed a mathematical model based on liquid diffusion theory and basic heat and mass transfer principles in order to simulate and optimize a two-stage drying system, which involved fluidized and fixed bed dryers.

Vagenas and Marinos-Kouris (1991) presented a mathematical model for the design and optimization of an industrial dryer for Sultana grapes. The optimum conditions were given by the minimum thermal load of the dryer per unit mass of dry product.

This paper presents a mathematical model for the case of conveyor belt dryers suitable for design purposes. The total flowsheet is optimized structurally and parametrically in a simultaneous way, by developing a superstructure of all potential structures and determining a solution to the problem through optimization of the total annual cost of the plant, by means of non-linear mathematical programming techniques. The final feasible solution contains valid and non-valid units which form in this way the optimum structure of the flowsheet. Furthermore, the computed optimum values of the decision variables involved determine the best construction characteristics and operating conditions of the proposed flowsheet. In addition to the above, a decomposition strategy is also proposed, so that the computational effort involved in optimizing the superstructure scheme is substantially reduced.

## MODELLING AND DESIGN

### Analysis of structure

An industrial conveyor belt dryer is made up of drying chambers placed in series. A drying chamber is actually the elementary module whose repetition forms the whole plant. For best performance, drying chambers are grouped together into drying sections. All chambers participating in a drying section are provided with a common conveyor belt, on which the product to be dried is uniformly distributed at the entrance. Obviously, redistribution of the product takes place when it leaves a drying section and enters the one that follows. A typical flowsheet comprising the above-mentioned structure is shown in Fig. 1.

Each drying chamber is equipped with an individual heating utility and fans for air circulation through the product. Air is heated by means of heat exchange units that operate with steam. On entering the chamber, fresh air is mixed with the recirculated air at a point below the heat exchangers. It is common practice that within each drying chamber, the temperature and humidity of the drying air stream entering the product, as well as its temperature difference while leaving it, are controlled. In this case, the final control elements are the steam valve and the chamber dampers that regulate the exchanged heat rate and the flow rate of fresh air entering the chamber, respectively. The interior of a typical drying chamber as well as the arrangement of its control facilities are shown in Fig. 2.

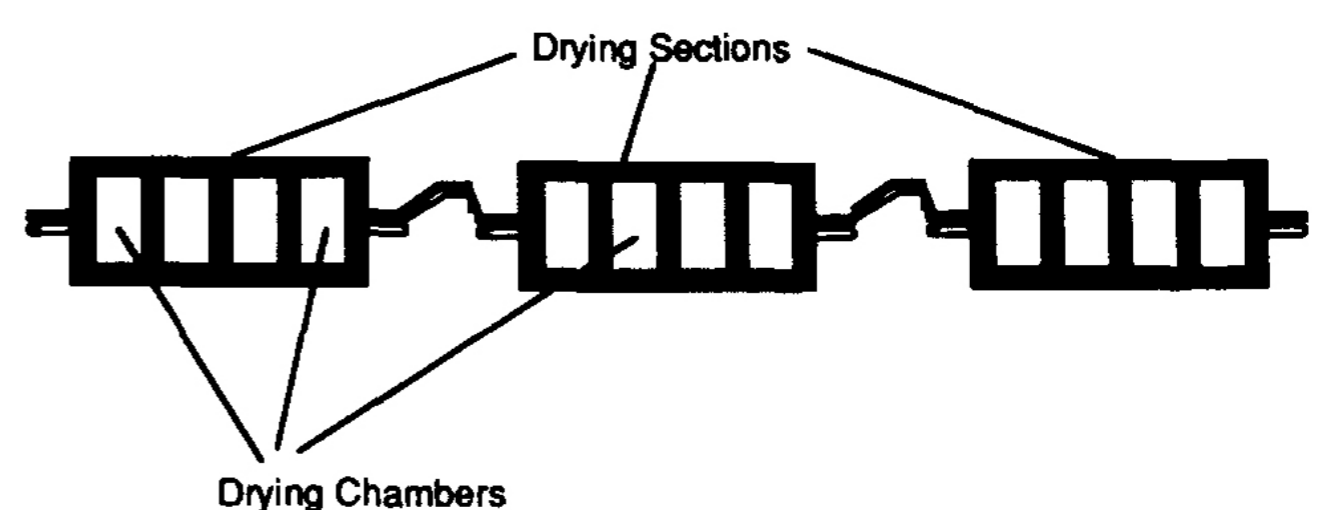


Fig. 1. Typical flowsheet in a dehydration plant.

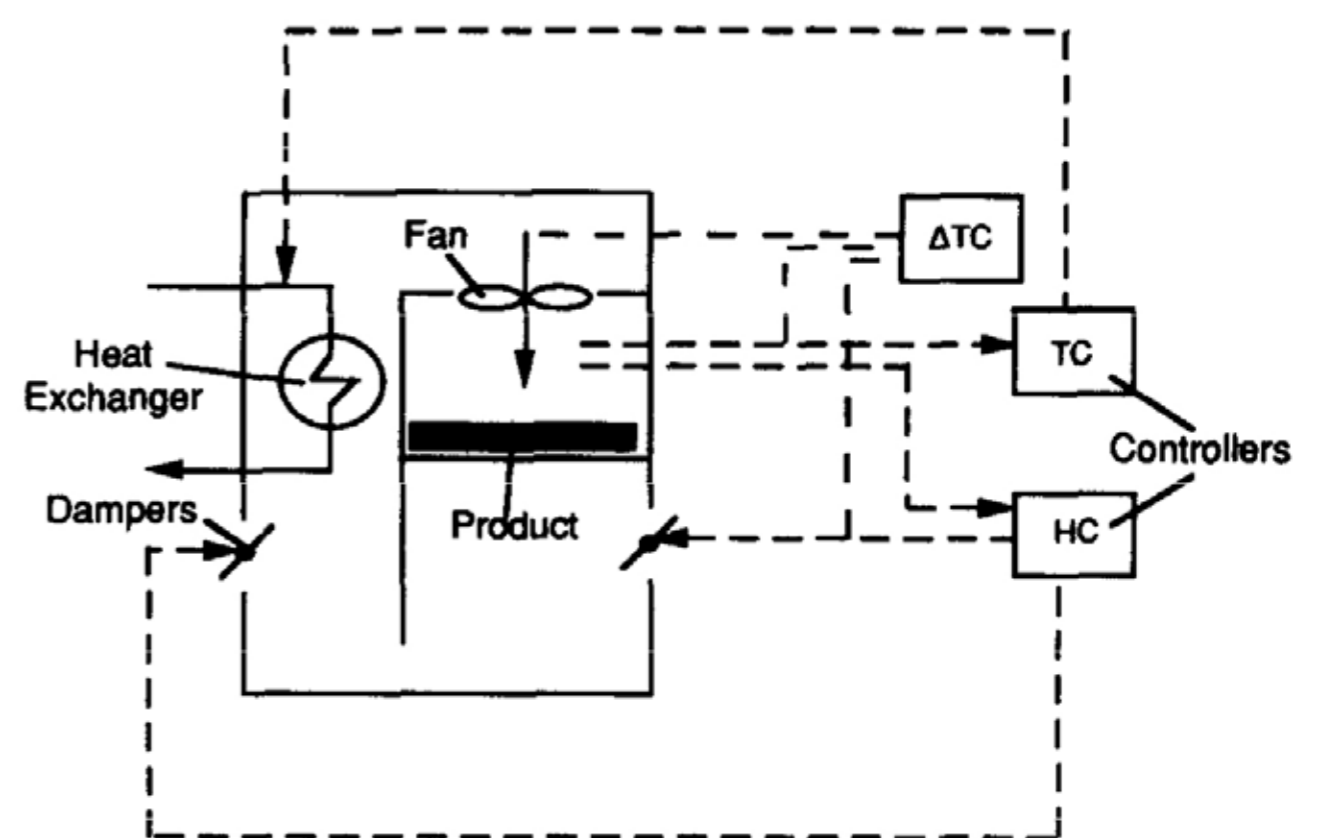


Fig. 2. Interior of a typical drying chamber.

Obviously, the objective of a design strategy can now be clearly stated. Given a specified product with a known flow rate, to be dried from an initial to a desired final moisture content level, the following must be determined:

- (i) the optimum number of drying sections as well as the number of chambers per section (flowsheet structure);
- (ii) the appropriate sizing of chambers (construction characteristics);
- (iii) the best set points of the controllers (operating conditions).

## Mathematical model

Since the dryer examined is made up of similar elementary units, that is to say drying chambers, the overall mathematical model will be formed by repetition of the ones that each individual component contributes. The mathematical model of a drying chamber involves heat and mass balances of air and product streams, as well as heat and mass transfer phenomena that take place during drying. The resulting equations are subject to quality, construction, and thermodynamic constraints that are also taken into consideration. The arrangement of product and air streams within each drying chamber as well as the conditions of each stream are shown in Fig. 3.

The overall mass balance of the drying chamber is given by the equation:

$$F_A(X_A - X_{A0}) = F_S(X_{S0} - X_S) \quad (1)$$

where  $X_{A0}$  is the absolute humidity of fresh air stream (kg/kg dry basis),  $X_A$  is the absolute humidity of rejected air stream (kg/kg dry basis),  $F_A$  is the flow rate of fresh air stream (kg/s dry basis),  $X_{S0}$  is the moisture content of product stream on entering the chamber (kg/kg dry basis),  $X_S$  is the moisture content of product stream on leaving the chamber (kg/kg dry basis) and,  $F_S$  is the flow rate of product stream (kg/s dry basis).

### Modelling and design of conveyor belt dryers

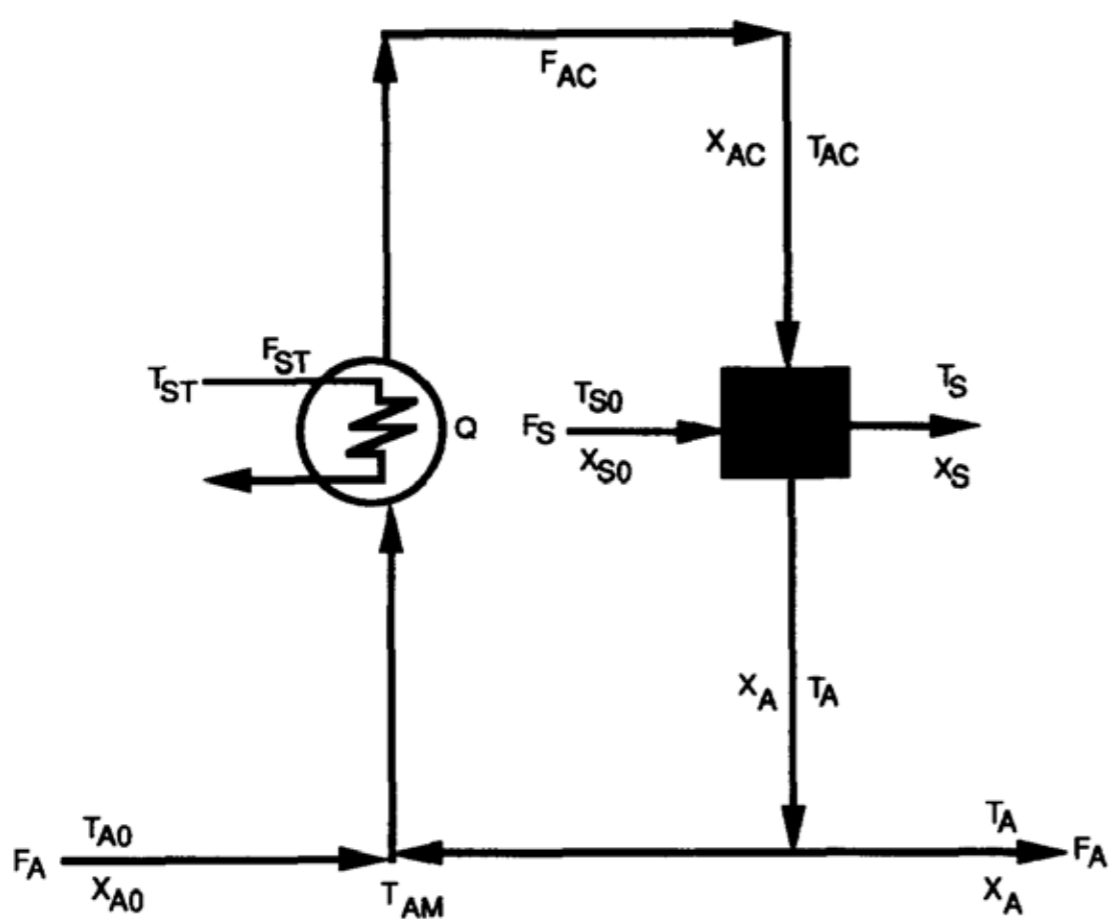


Fig. 3. Arrangement of product and air streams within each drying chamber.

The mass balance in the drying compartment is given by the equation:

$$F_{AC}(X_A - X_{AC}) = F_S(X_{S0} - X_S) \quad (2)$$

where  $X_{AC}$  is the absolute humidity of drying air stream (kg/kg dry basis) and,  $F_{AC}$  is the flow rate of drying air stream (kg/s dry basis).

The overall heat balance in the drying chamber, assuming negligible heat losses, is expressed by the following equation:

$$Q = F_A [h_A(T_A, X_A) - h_A(T_{A0}, X_{A0})] + F_S [h_s(T_s, X_s) - h_s(T_{S0}, X_{S0})] \quad (3)$$

where  $T_{A0}$  is the temperature of fresh air stream ( $^{\circ}\text{C}$ ),  $T_A$  is the temperature of rejected air stream ( $^{\circ}\text{C}$ ),  $Q$  is the exchanged heat rate (W),  $T_{S0}$  is the temperature of product stream on entering the chamber ( $^{\circ}\text{C}$ ) and,  $T_s$  is the temperature of product stream on leaving the chamber ( $^{\circ}\text{C}$ ).

The overall balance in the drying compartment is given by the equation:

$$F_{AC} [H_A(T_{AC}, X_{AC}) - h_A(T_A, X_A)] = F_S [h_s(T_s, X_s) - h_s(T_{S0}, X_{S0})] \quad (4)$$

where  $T_{AC}$  is the temperature of drying air stream ( $^{\circ}\text{C}$ ). Heat and mass transfer phenomena during drying are indeed complicated. They involve coupled transfer mechanisms both within the solid and the gas phases. A mathematical model explicitly accounting for all transfer mechanisms should not be considered to be appropriate for design purposes since it demands considerable computational time. Computational time is of major importance when a mathematical model is to be solved repeatedly in an optimization convergence procedure. In this case, a simplified model is considered to be more suitable. The empirical model used, has an exponential form and contains a mass transfer coefficient of a phenomenological nature, which is usually called the drying constant. The drying constant chiefly accounts for mass diffusion within the solid phase, but it also embodies boundary layer phenomena when it is considered to be a function of all process variables affecting drying. In this way, ample accuracy is combined with sufficient low computation time. On the basis of the above, mass transfer is expressed by the following equation (Bruin & Luyben, 1980):

$$X_S = X_{SE}(T_A, \alpha_w) + [X_{S0} - X_{SE}(T_A, \alpha_w)] \exp[-k_m(T_A, X_A, V_A, r_c)t] \quad (5)$$

where  $V_A$  is the air velocity through the product (m/s),  $r_c$  is the characteristic dimension of the product particles (m),  $\alpha_w$  is the water activity of the air stream leaving the product,  $t$  is the residence time (s), and  $k_m$  is the drying constant as a function of drying air stream temperature, absolute humidity and velocity, and characteristic dimension of product particles (1/s).

The air water activity involved in eqn (5) is calculated as follows:

$$a_w = X_A P / [ (\lambda_B + X_A) P^o(T_A) ] \quad (6)$$

The heat transfer is chiefly controlled by the heat transfer coefficient at the air boundary layer. For the purpose of developing the particular mathematical model it is assumed that the heat transfer coefficient takes a value high enough to allow the product stream leaving the chamber to be in thermal equilibrium with the air stream leaving the product. This assumption removes the need for an unnecessary differential equation which would not improve the model greatly. On the basis of the above, the following equation is used:

$$T_s = T_A \quad (7)$$

The distribution of the product on the conveyor belt is characterized by the belt load variable. This variable is expressed in units of mass of product placed on the belt per unit of area, and varies with position on the belt. Its value at the entrance of each drying chamber can be calculated by the following equation:

$$\rho_{s0} = F_s (1 + X_{s0}) t/A \quad (8)$$

where  $\rho_{s0}$  is the belt load at the entrance (kg/m<sup>2</sup> wet basis), and  $A$  is the area of chamber belt (m<sup>2</sup>).

The air velocity through product involved in eqn (5), is computed as follows:

$$V_A = F_{AC} (1 + X_{AC}) / A \rho_A \quad (9)$$

The heat balance at the heat exchangers of the chamber are given by the following equations:

$$Q = F_{ST} \Delta H (T_{ST}) \quad (10)$$

$$Q = A_{ST} U_{ST} [ (T_{ST} - T_{AM}) - (T_{ST} - T_{AC}) ] / \ln [ (T_{ST} - T_{AM}) / (T_{ST} - T_{AC}) ] \quad (11)$$

where  $T_{ST}$  is the temperature of steam (°C),  $F_{ST}$  is the flow rate of steam (kg/s),  $U_{ST}$  is the overall heat transfer coefficient (W/m<sup>2</sup> K),  $A_{ST}$  is the area of heat exchanger (m<sup>2</sup>) and  $T_{AM}$  is the temperature of mixed recirculation and fresh air streams (°C).

The temperature of mixed recirculation and fresh air streams can be calculated by means of the following heat balance equation:

$$F_{AC} h_A (T_{AM}, X_{AC}) = F_A h_A (T_{A0}, X_{A0}) + (F_{AC} - F_A) h_A (T_A, X_A) \quad (12)$$

Furthermore, we introduce the temperature difference,  $\Delta T$ , of the drying air stream across the product:

$$\Delta T = T_{AC} - T_A \quad (13)$$

The electrical power,  $E$ , consumed by the operation of fans is given by the equation:

$$E = \Delta P F_{AC} \quad (14)$$

Equations (1) - (14) constitute the mathematical model of each individual drying chamber (Kira-noudis, 1992). The variables in this model are constrained by thermodynamic, construction and quality factors. These constraints must also be taken into consideration, for they determine the feasible region of decision variables.

To begin with, the load of the conveyor belt should not exceed a maximum value that would guarantee a sufficiently low thickness of product layer placed on the belt. This is intended to prevent non-uniform drying due to the creation of axial mass and temperature gradients within the product.

Furthermore, this is also dictated by construction reasoning since overloading of the conveyor belt will greatly affect its performance. On the basis of the above, the following constraint is proposed:

$$0 \leq \rho_{s0} \leq \rho_{sMAX} \quad (15)$$

where  $\rho_{sMAX}$  is the maximum belt load (kg/m<sup>2</sup> wet basis).

In the same sense, the temperature change in the drying air stream, on passing through the product, should not exceed a maximum value which guarantees uniform drying, that is to say:

$$0 \leq \Delta T \leq \Delta T_{MAX} \quad (16)$$

where  $\Delta T_{MAX}$  is the maximum temperature change in the drying air stream across the product (°C).

Thermodynamics dictate that the moisture content of the product stream on leaving the product should be greater than the corresponding equilibrium moisture content imposed by the operating conditions of the chamber:

$$X_s \geq X_{SE} (T_A, a_w) \quad (17)$$

To prevent thermal degradation of the product, its temperature should not exceed an upper limit that affects its quality. Thus:

$$T_{A0} \leq T_S \leq T_{SMAX} \quad (18)$$

where  $T_{SMAX}$  is the maximum temperature level, below which no thermal degradation effects are observed ( $^{\circ}C$ ).

Finally, construction reasons dictate that the area of the conveyor belt can not exceed a maximum value. Hence:

$$0 \leq A \leq A_{MAX} \quad (19)$$

where  $A_{MAX}$  is the maximum constructed area of conveyor belt for each chamber ( $m^2$ ).

Inequalities (15)-(19) constitute the constraints for the variables of the mathematical model for each drying chamber. Equations (1)-(14) involve certain thermophysical properties as well as transport coefficients used for model evaluation. These qualities are analysed below.

The specific enthalpy of an air stream, as a function of its temperature and absolute humidity, is given by the following equation, assuming a reference temperature of  $0^{\circ}C$ :

$$h_A = C_{p_A} T_A + X_A (\Delta H_0 + C_{p_V} T_A) \quad (20)$$

The specific enthalpy of a product stream, as a function of its temperature and moisture content, can be calculated by means of the following equation, assuming a reference temperature of  $0^{\circ}C$ :

$$h_S = C_{p_S} T_S + X_S C_{p_W} T_S \quad (21)$$

The equilibrium moisture content of desorption of the product as a function of water activity and temperature of the surrounding air, can be computed by using the well-known Guggenheim-Anderson-de Boer (GAB) equation, which is successfully used in the case of foods (Maroulis et al., 1988; Kiranoudis et al., 1993):

$$X_{SE} = \gamma_0 C K a_w / [(1 - K a_w) (1 - (1 - C) K a_w)] \quad (22)$$

$$C = \gamma_{11} \exp(\gamma_{12} / RT_A) \quad (23)$$

$$K = \gamma_{21} \exp(\gamma_{22} / RT_A) \quad (24)$$

where  $\gamma_0, \gamma_{11}, \gamma_{12}, \gamma_{21}$  and  $\gamma_{22}$  are constants.

The drying constant used in eqn (5) as a function of temperature, absolute humidity and velocity of the drying air stream, as well as the characteristic dimension of the material particles, is given by the following empirical equation (Kiranoudis et al., 1993):

$$K_M = \beta_0 r_C \beta^1 T_A \beta^2 X_A \beta^3 V_A \beta^4 \quad (25)$$

where  $\beta_0, \beta_1, \beta_2, \beta_3$  and  $\beta_4$  are constants.

Water vapour pressure can be computed by means of the Antoine equation (Pakowski et al., 1991):

$$\ln P^0 = A_1 - A_2 / (A_3 + T_A) \quad (26)$$

where  $A_1, A_2$  and  $A_3$  are constants.

Latent heat of water vaporization is well known to be a function of temperature but can be assumed constant in the temperature range studied.

## Degrees of freedom analysis

The mathematical model of each drying chamber is described by eqns (1)-(14), and constraints (15)-(19). It involves 14 equations and 29 variables. The degrees of freedom (d.f.) analysis of the particular model is shown in Table 1. The four decision variables for each drying chamber were chosen to be the absolute humidity and temperature of the air stream leaving the product, the temperature change in the drying air stream passing through the product, and the belt load at the entrance is the chamber.

Let us assume that the plant examined consists of  $N$  drying sections and that  $n_i$  drying chambers make up its  $i$ th section. Let us denote each Degrees of Freedom Analysis for the Mathematical Model of the Drying Chamber

TABLE 1

Equation	Property	Given variables	Calculated variables	Design variables
1	—	$X_{S0}, X_S, X_{A0}, F_S$	$F_A$	$X_A$
2	—	—	$F_{AC}, X_{AC}$	—
3	$h_A, h_S$	$T_{A0}, T_{S0}$	$Q, T_S$	$T_A$
4	—	—	$T_{AC}$	—
5	$k_M, X_{SE}$	$r_C$	$V_A, t$	—
6	$P^0$	$P$	$a_w$	—
7	—	—	—	—
8	—	—	$A$	$\rho_{S0}$
9	$\rho_A$	—	—	—
10	$\Delta H$	$T_{ST}$	$F_{ST}$	—
11	—	$U_{ST}$	$A_{ST}, T_{AM}$	—
12	—	—	—	—
13	—	—	—	$\Delta T$
14	—	$\Delta P$	$E$	—
Total	7	11	14	4
Number of variables:				29
Number of equations:				14
Free variables:				15
Given variables:				11
Decision variables:				4

chamber by the couple  $(i, j)$  where  $j$  stands for the  $i$ th chamber of the  $i$ th section. Clearly, if  $M$  is the total number chambers, then:

$$M = \sum_{i=1}^N n_i \quad (27)$$

The overall mathematical problem will contain eqns (1) - (14) and constraints (15)-(19) written M times. Since the number of decision variables in the case of a drying chamber is 4, the number of decision variables in the case of the whole plant is expected to be 4M. However, we must add to this figure the (M - 1) moisture content variables of the intermediate product streams, and subtract the (M - N) correlations between the belt load variables for all chambers participating in a section. Clearly, in each section only one belt load variable can be chosen as a decision variable, i.e. the one at the entrance. The belt load variables at any point of the conveyor belt can be calculated by the following equation:

$$\rho_s = (1+X_s) \rho_{s0} / (1+X_{s0}) \quad (28)$$

On the basis of the above, the total number of decision variables in this case is:

$$\text{d.f.} = 3M + N (M - 1) \quad (29)$$

## Economic evaluation

The economic evaluation of the plant is based on the determination of its total annual cost. It involves two components, the capital and the operational costs, as shown below:

$$C_T = eC_{CP} + t_{OP}C_{OP} \quad (30)$$

where  $C_T$  is the total annual cost (US\$/year),  $C_{CP}$  is the capital annual cost (US\$/year) and,  $C_{OP}$  is the operational annual cost (US\$/h).

The operating cost concerns thermal and electrical energy consumed at heat exchangers and fans, respectively. Thus:

$$C_{OP} = \sum_{i=1}^N \sum_{j=1}^{n_i} (C_{ST} F_{ST(i,j)} + C_E E_{(i,j)}) \quad (31)$$

where  $C_{ST}$  is the unit cost of steam (US\$/kg) and  $C_E$  is the unit cost of electricity (US\$/kWh).

The capital cost is affected by the belt area of each drying chamber, i.e. concerns construction expenses, the area of heat exchanges involved and the drying air flow rate that determines the costs of fans. All cost components obey economy of scale laws. Furthermore, the capital cost is affected by the cost of the conveyor belt which is determined by the area of belt within a drying section. These components similarly obey an economy of scale law. Hence:

$$C_{CP} = \sum_{i=1}^N \left[ a_s \left( \sum_{j=1}^{n_i} A_{(i,j)} \right)^{n_s} + \sum_{j=1}^{n_i} \left( a_D A_{(i,j)}^{n_D} + a_F F_{AC(i,j)}^{n_F} + a_{ex} A_{ST(i,j)}^{n_{ex}} \right) \right] \quad (32)$$

where  $a_s, a_D, a_F, a_{ex}, n_s, n_D, n_F$  and  $n_{ex}$  are constants. The resulting total annual cost function is expressed in present currency.

## Construction of the superstructure

The total flowsheet can now be formulated as a superstructure that involves a large number of minor structures as well as the mathematical model of these structures. The superstructure created can then be optimized, using the total annual cost as the objective function. In this way the best flowsheet is determined, by carrying out structural and parametric optimization in a simultaneous way.

On the basis of the above, the optimization problem in question takes the form:

$$\min_{xy} f(x, y) \quad (33)$$

under the constraints:

$$g(x, y) \leq 0 \quad (34)$$

where  $f()$  is the objective function described above and  $g()$  is the mathematical model described in details above,  $x$  is the vector of intermediate decision variables (moisture content of intermediate streams) and,  $y$  is the vector of internal decision variables for each chamber (temperature, absolute humidity, and temperature difference of drying air stream passing through product).

The superstructure will normally contain a sufficiently large number of drying sections, each one involving a reasonable number of drying chambers. The optimum solution will contain both valid and non-valid chamber units. For non-valid chamber units, the moisture content of the product stream entering the chamber is equal to the moisture content of the product stream leaving it, resulting in this way in a degenerate flowsheet that will contain the best structure. Clearly, non-valid units will not contribute to the total annual cost of the plant, while for valid units the optimum decision variables, that is to say construction characteristics and operating conditions, are determined.

## DECOMPOSITION STRATEGY

The optimization problem discussed above involves a large number of decision variables. It is therefore considered important if a decomposition strategy is proposed in order to reduce the computation labor. The problem described by eqns (33) and (34) can be transformed into its equivalent optimization problem described by the following equations (Kiranoudis, 1992):

$$\min_x f^*(X) \quad (35)$$

under the constraints:

$$g^*(X) \leq 0 \quad (36)$$

where:

$$f^*(X) = \sum_{i=1}^M \min_{y_i} f(x, y_i) \quad (35)$$

where  $y_i$  is the vector of the decision variables in each drying chamber stage.

In this way, the initial problem changes to one that involves a major and a minor optimization stage. The decision variables of the major optimization problem are only the  $M-1$  moisture content values of the intermediate product streams. The minor optimization problem comprises  $M$  sequential optimizations that involve only three decision variables. The computational effort in this case can be reduced to approximately 20% of the one necessary for solving the original problem.

## DESIGN CASE

The proposed methodology was applied to the design of a dehydration plant for drying of 2400 t/year raw potato. The raw material is cut into 10-mm cubes and enters the processing dryer at a rate of 200 kg (dry solid)/h. Its initial moisture content is 5 kg/kg (dry basis) and its desired level at the exit of the dryer is 0.05 kg/kg (dry basis). The product should not be heated to temperature levels exceeding 75°C, to prevent degradation. The maximum constructed chamber area for the plant is 5 m<sup>2</sup>. Belt load at the entrance of each drying section is kept at a level of 50 kg/m<sup>2</sup> (wet basis). The operation horizon of the plant is 2000 h/year and its capital cost will be paid off after a period of 5 years.

The superstructure constructed comprised of a total number of 10 drying sections, each one consisting of 15 drying chambers, i.e. 150 drying chambers. The total number of design variables was 609. The optimization of the total annual cost was achieved by means of sequential quadratic programming algorithm as implemented in the form of a subroutine (E04UCF/NAG). The code ran on a Sony Workstation under UNIX. The result was 13 valid drying chamber units distributed into three drying sections. The first drying section of the optimum structure involved five drying chambers, while the second and the third ones involved four drying chambers each. The optimum chamber area for each chamber was found to be equal to the constraint of 5 m<sup>2</sup> for all chambers. The temperature of the product at the exit of each chamber was found to be uniform for all chambers at the constraint level of 75°C.

The total annual cost for the plant was found to be 122000 US\$/year distributed into capital expenses (42%) and operational needs (58%). This performance results in a total processing expense of US 5 cents/kg (wet basis). After 5 years of operation, this cost will be reduced to almost 3 US cents (assuming negligible maintenance cost), since all capital cost will be paid off. This value is almost 10% of the raw material price in the market and, therefore, the drying process cost is only a small portion of the final price of the dehydrated product.

The way that material moisture content potential varied along the optimum structure is shown in Fig. 4(a). In this figure, actual and equilibrium material moisture content at the exit of each drying chamber is plotted against air water activity of the rejected air stream. The potential is maintained high even at the last stages of drying. The general impression is that material moisture content decreases smoothly except for the points where a new section is introduced. In Fig. 4(b), material moisture content at the exit of each drying chamber is plotted against the actual number of the chamber involved. Material moisture content decreases considerably in the last stages of drying affected by the optimal section changeovers and operating conditions involved.

The way that air drying temperature and humidity varied along sections is shown in Fig. 5.

The area of each chamber and the area of heat exchangers used is shown in Fig. 6. Chamber area is uniform in all sections while the two other variables exhibit similar peaked curves when a new chamber is introduced.

Figure 7 shows the amount of heat exchanged and the consumption of electricity in each chamber. Figure 8 shows the way that air stream flow rates vary at each chamber. While the drying air stream flow rate decreases at the last stages of drying, the amount of fresh air needed increases. In this way the appropriate drying potential is maintained.

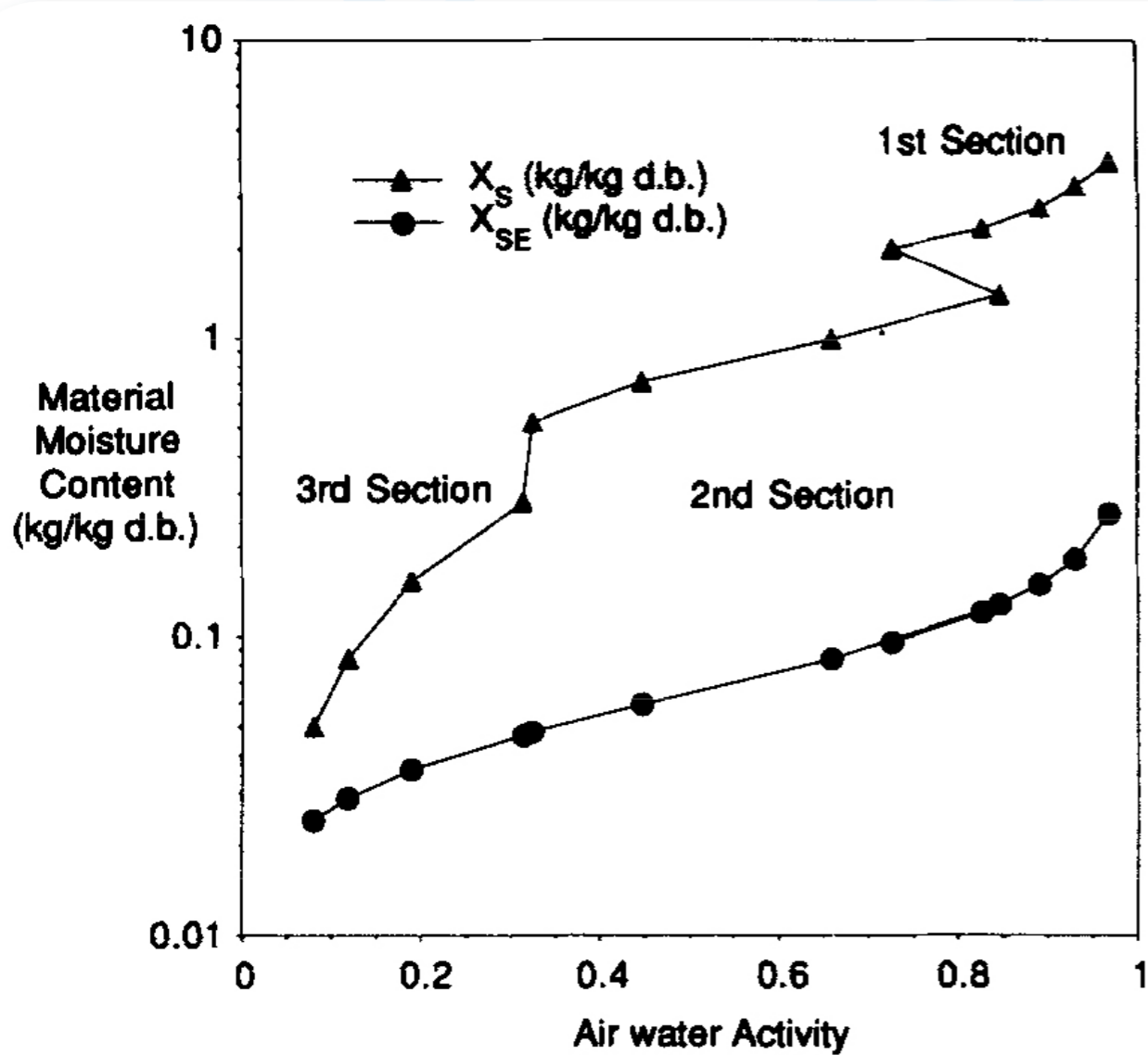
The total annual cost and its components for each chamber involved are presented in Fig. 9. Evidently, the first chamber consumes the largest amount of energy for water evaporation and product heating and, therefore, its contribution to the total operational cost is important.

As observed above (Figs 4-9), discontinuities were found for most of the variables examined at points where a new section was introduced. This can be explained by the fact that at each drying section, the velocity of the conveyor belt, and therefore the residence time, changes (12, 4-8,

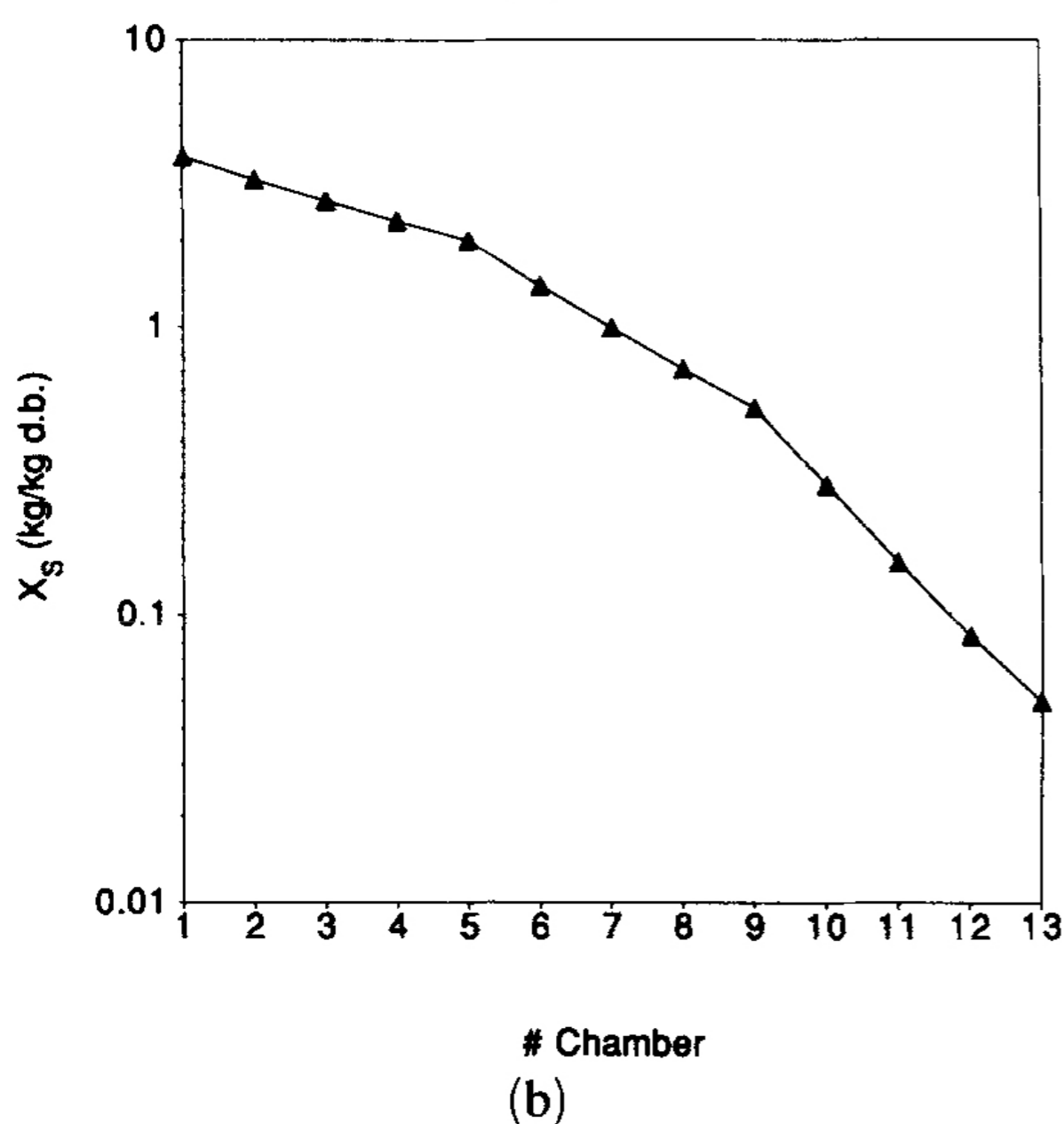
Fig. 4. (a) Actual and equilibrium material moisture content in each chamber of the optimum structure. (b) Material moisture content at the exit of each drying chamber of the optimum structure.

and  $2 \times 5\text{m} / \text{h}$  in the first, second, and third sections, respectively) which results completely new drying characteristics.

The decomposition strategy was applied to the superstructure problem mentioned above. The contours of the transformed cost function  $f^*$  are shown in Fig. 10 for the second chamber of the second section and the last chamber of the third section.



(a)



(b)

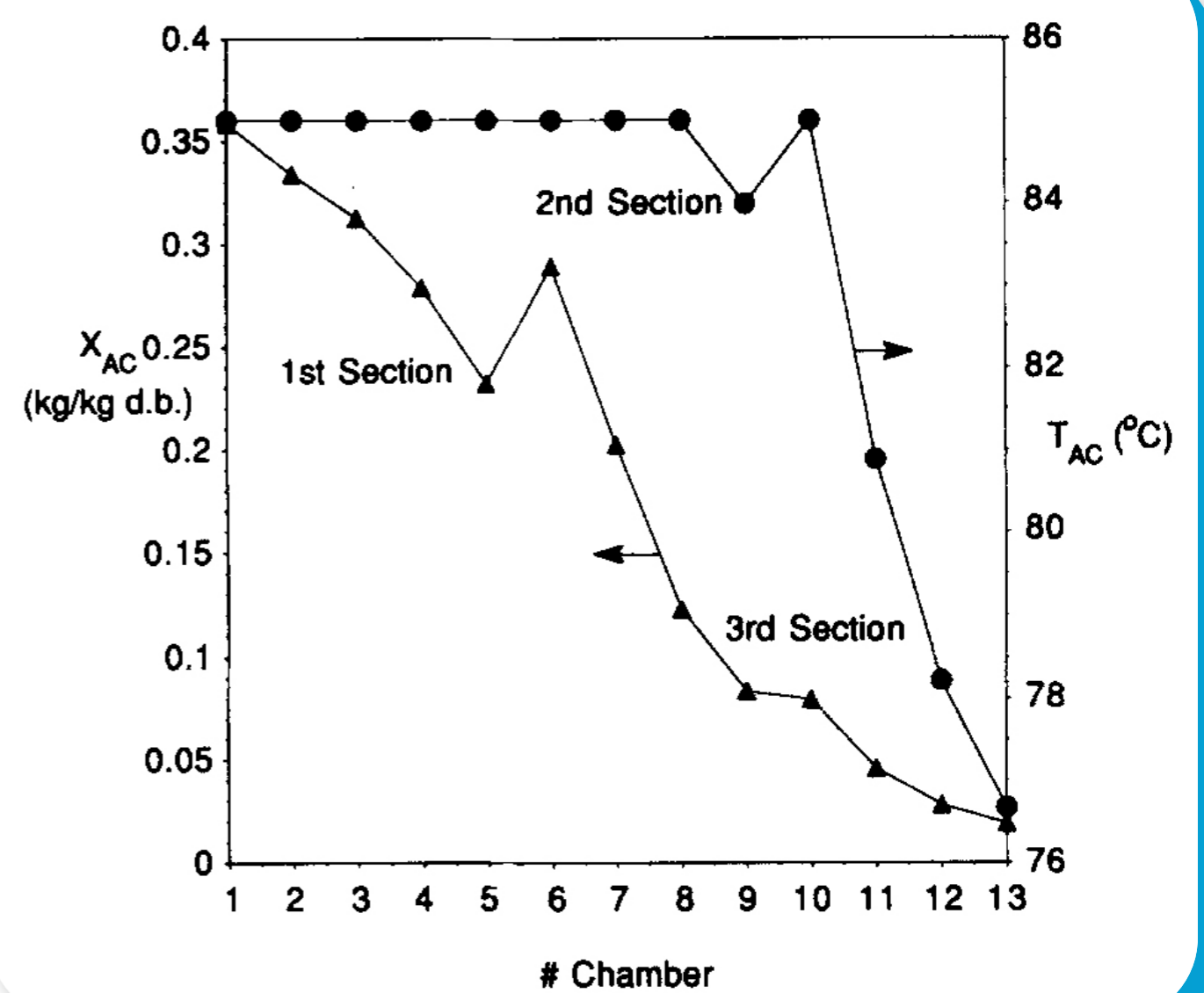


Fig. 5. Absolute humidity and temperature of drying air in each chamber of the optimum structure.

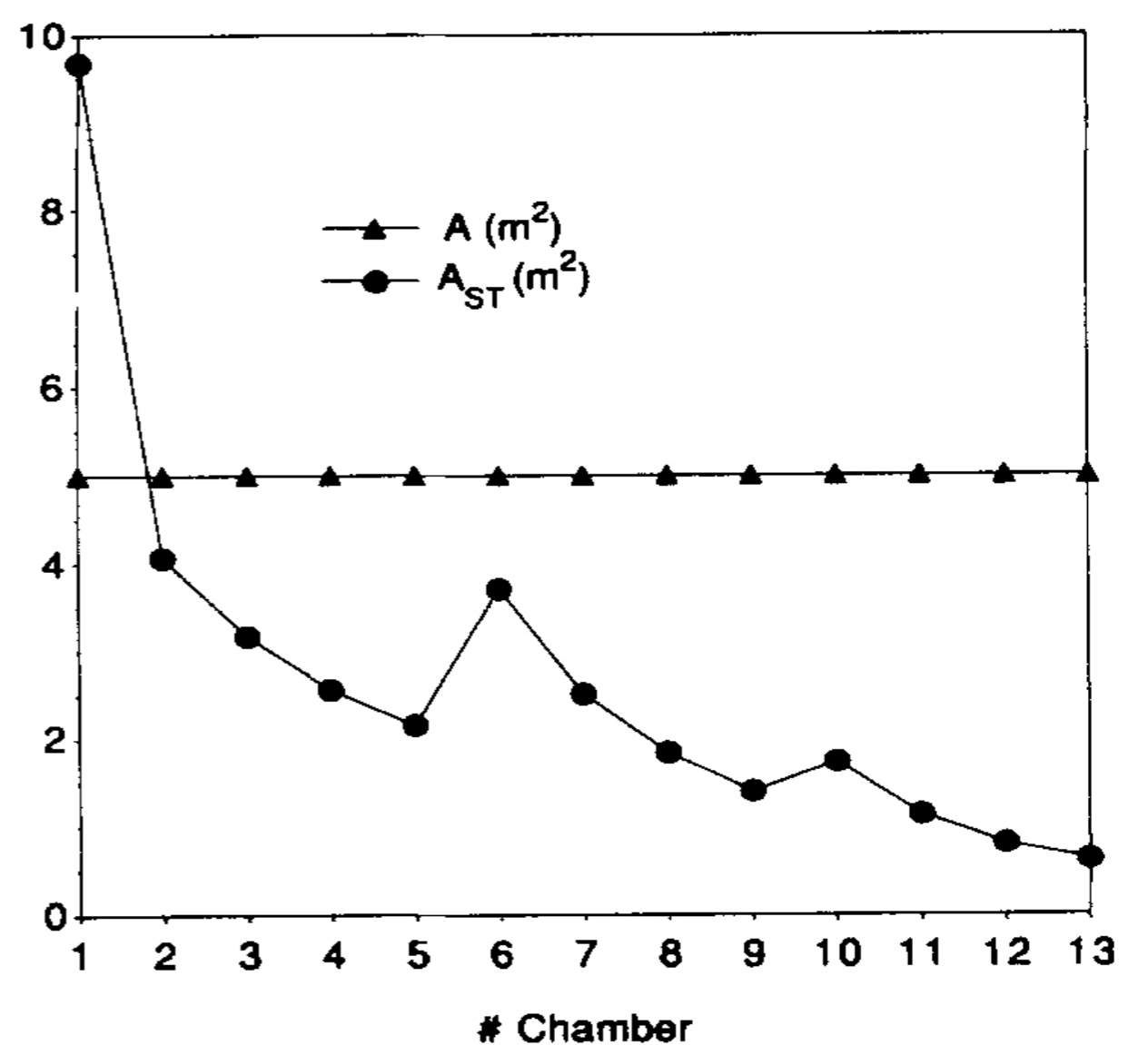


Fig. 6. Chamber area and heat exchanger area for each chamber of the optimum structure.

The contours are presented in a two-variable space, using two of the most significant variables, i.e. air temperature and humidity. The third variable was kept constant at its optimum level. On these diagrams the model constraints are also included. Clearly, there is a global optimum of this chamber cost,

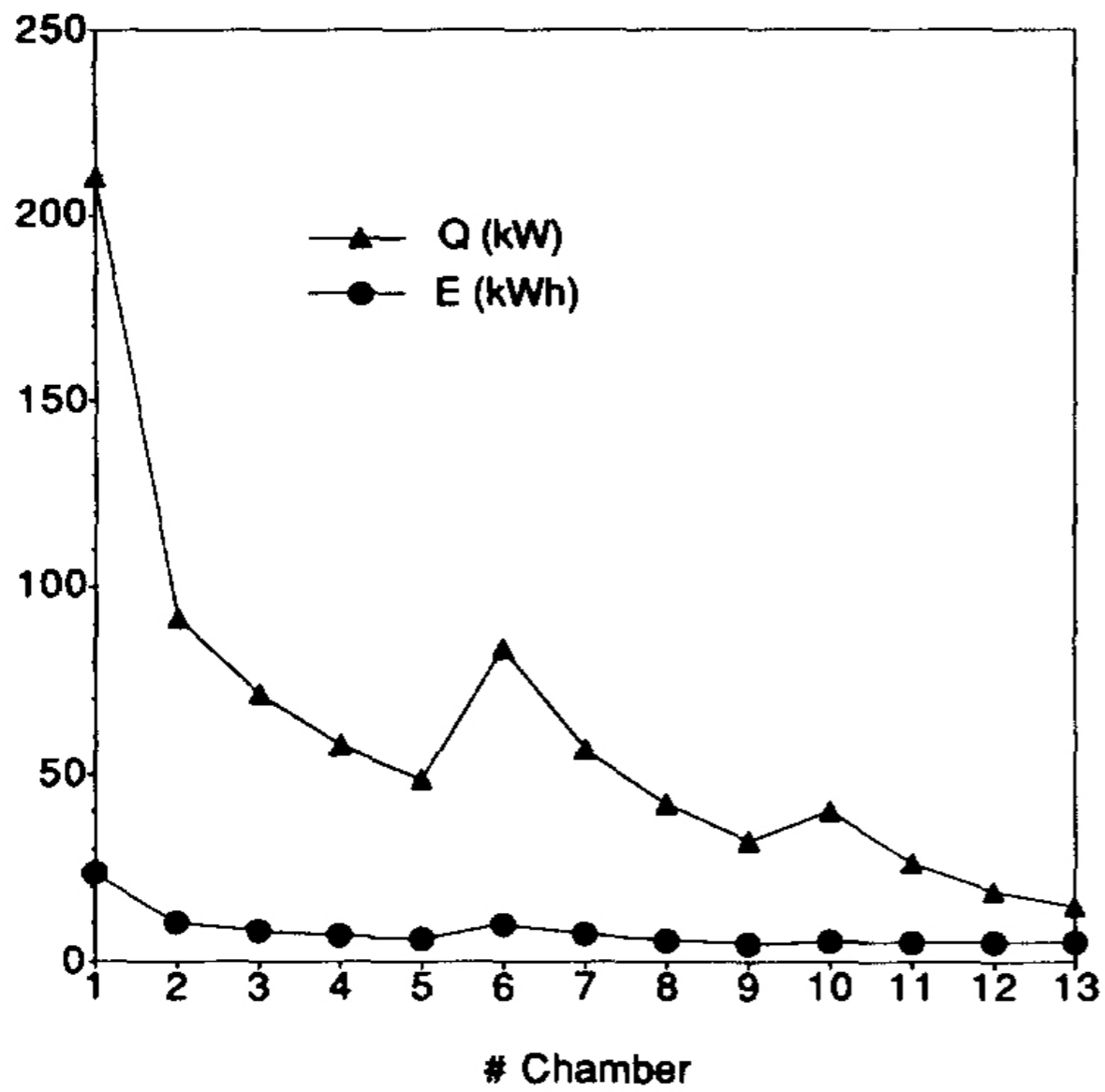


Fig. 7. Thermal and electrical power consumption in each chamber of the optimum structure.

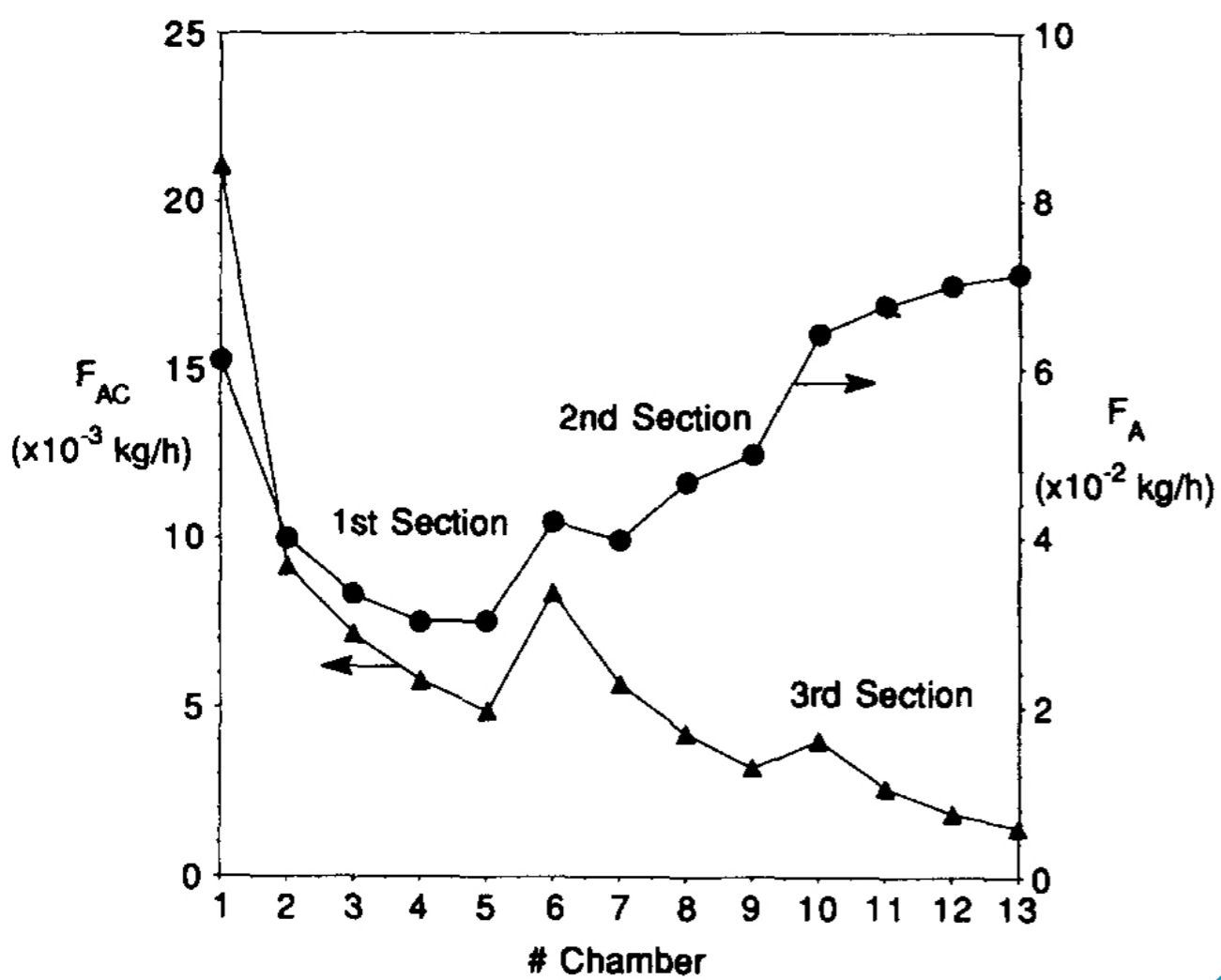


Fig. 8. Air stream flow rates within each chamber of the optimum structure.

which can be achieved only at temperature levels far above the constrained temperature value.

## CONCLUSION

The present analysis shows that a strict theoretical method can be used for the design of conveyor belt dryers. For design purposes a mathematical

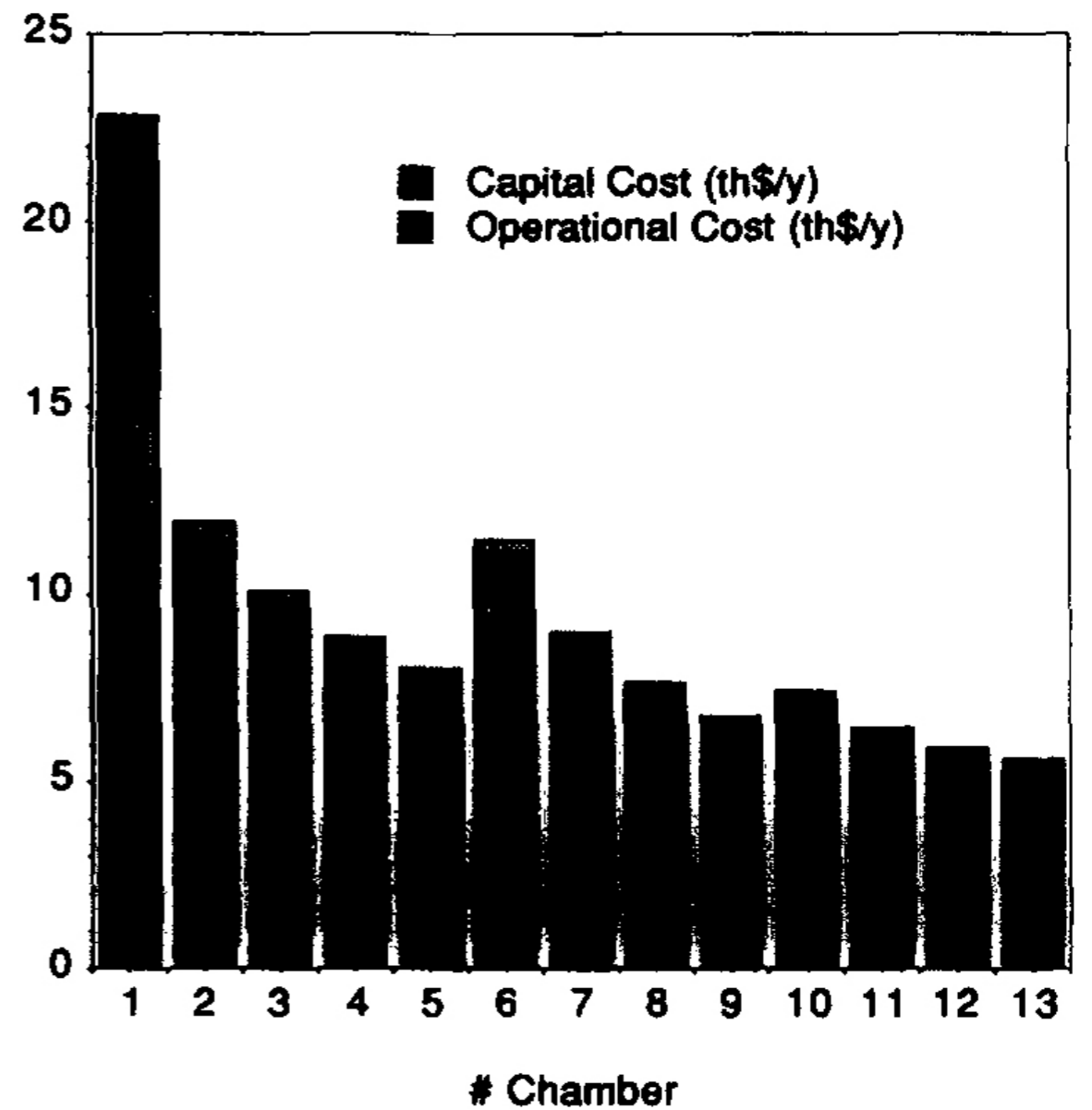


Fig. 9. Total annual cost and its components for each chamber of the optimum structure.

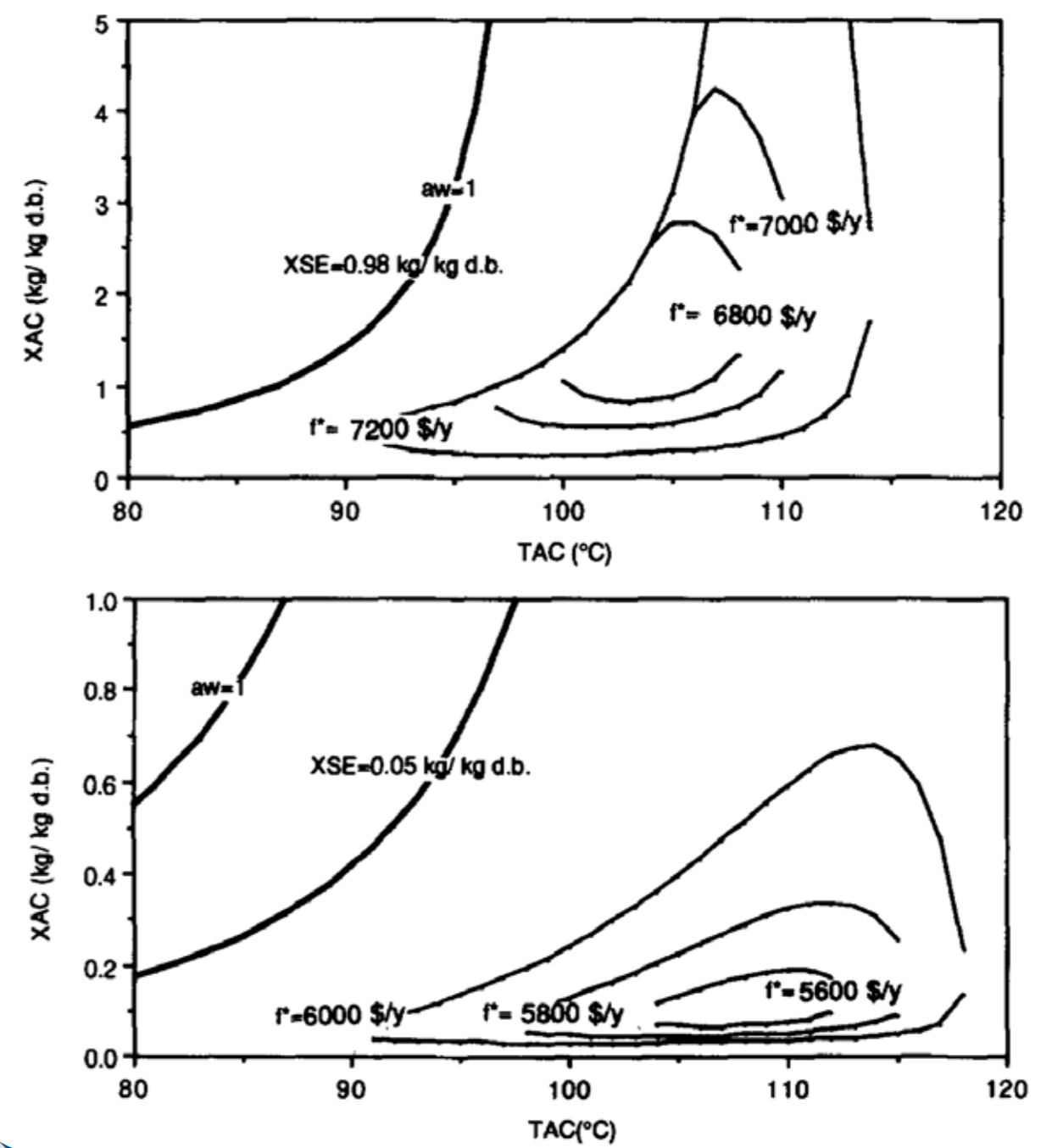


Fig. 10. Contours of the transformed chamber cost for two chambers of the optimum structure.

model was developed and used for the construction of a superstructure that contained a large number of minor structures. The optimum structure found by using non-linear mathematical programming techniques, was determined by minimizing the total annual cost of the plant, and in this way the optimum construction characteristics and operating conditions were determined simultaneously with the best structure. The computational effort involved can be greatly reduced if the initial optimization problem is transformed into an equivalent one that performs optimization in major and minor optimization stages.

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