

ANALYSIS OF SOLID DESICCANT

ROTARY DEHUMIDIFIERS

by

ERIC VAN DEN BULCK

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ABSTRACT

This work presents an analysis of solid desiccant rotary dehumidifiers. The conservation equations for mass and energy that model the dehumidifier with infinite transfer coefficients are analyzed by means of a wave theory. Two techniques are used: the method of characteristics and the shock wave method. The former is applicable for modeling the processing period of the dehumidifier, the latter for modeling the regenerating period. The techniques are complementary. When combined, they provide a set of analytical equations which allow for fast and accurate predictions of the performance of an ideal dehumidifier.

Empirical correlations for the effectiveness of a rotary silica gel dehumidifier with finite transfer coefficients are presented. The effectiveness is correlated as a function of the inlet conditions and a set of dimensionless transfer coefficients. Combined with the ideal dehumidifier model, an ϵ -NTU model is obtained that allows rapid calculation of the dehumidifier performance. The method is accurate for a wide range of inlet conditions, operating parameters and transfer numbers.

The use of dehumidifiers in desiccant cooling systems is investigated. Wheel revolution speed and regenerating mass flow rate are optimized with respect to minimal energy use for a given cooling

load. The sensitivity of the energy supply with respect to those parameters is studied and the optimal control strategy for a dehumidifier cooling system is discussed.

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NOMENCLATURE

a_i	Eq. (2.13), state property functions
A_j	total heat transfer area of the dehumidifier matrix in period j [area]
c_p	moist air specific heat [energy/(dry mass-temp)]
C^*	ratio of minimum to maximum fluid capacity rates for a thermal regenerator; or, Table 3.3, ratio of operating parameters [dimensionless]
C_ϵ	Table 3.3, effectiveness correction factor [dimensionless]
COP	coefficient of performance, ratio of cooling output to energy input [dimensionless]
F_i	ith combined heat and mass potential [dimensions arbitrary]
G_i	Eq. (2.22), integrating function for the F_i potential [dimensions arbitrary]
h	heat transfer coefficient [energy/(area-time-temp)]
h^*	Table 3.2, ratio of the isothermal differential heat of adsorption to the heat of vaporization [dimensionless]
h_{fg}	heat of vaporization [energy/mass]
h_w	mass transfer coefficient [mass/(area-time)]
i	moist air enthalpy [energy/dry mass]
i_s	Table 3.2, isothermal differential heat of adsorption [energy/dry mass]
i_v	heat of vaporization [energy/mass]
i_{wv}	water vapor enthalpy [energy/mass]
I	Table 3.2, matrix enthalpy [energy/dry mass]
L	axial flow length through the matrix [length]

Nomenclature (continued)

Le	Eq. (2.66), overall Lewis number [dimensionless]
$\dot{m}$	mass flow rate [mass/time]
M_d	mass of desiccant in the dehumidifier matrix [mass]
M_f	mass of air in the dehumidifier matrix [mass]
NTU	number of transfer units [dimensionless]
NTU_t	Eq. (2.66), number of transfer units for heat transfer [dimensionless]
NTU_o	overall number of transfer units [dimensionless]
NTU_w	Eq. (2.66), number of transfer units for mass transfer [dimensionless]
P_t	Table 3.1, total air pressure [force/area]
P_{ws}	Table 3.1, saturation pressure of water vapor [force/area]
P_{wv}	Table 3.1, partial pressure of water vapor [force/area]
Q	thermal energy supply [energy]
r	relative humidity of moist air [dimensionless]
RPM	dehumidifier wheel revolution speed [time^{-1}]
S_i	Eq. (2.36), i th shock wave speed [dimensionless]
t	temperature
T	time required for a complete rotation of the matrix [time]
w	moist air humidity ratio [dimensionless]
W	matrix water content [dimensionless]
x	axial coordinate measured from period entrance [dimensionless]

Nomenclature (continued)

z	axial displacement through matrix measured from period entrance [length]
Γ_i	Eq. (2.4), i th operating parameter of the rotary dehumidifier [dimensionless]
δ	perturbation of variable
Δ	difference
ϵ	effectiveness [dimensionless]
ϵ_{cf}	counterflow heat exchanger effectiveness [dimensionless]
ϵ_h	Eq. (3.5), enthalpy effectiveness of the rotary dehumidifier [dimensionless]
ϵ_R	rotary heat exchanger effectiveness [dimensionless]
ϵ_t	temperature effectiveness of the rotary dehumidifier [dimensionless]
ϵ_w	Eq. (3.5), humidity ratio effectiveness [dimensionless]
θ	time
θ_j	duration of period j [time]
λ_i	Eq. (2.18), i th characteristic wave speed [dimensionless]
τ	time coordinate [dimensionless]
τ_{dj}	dwelt time of a fluid particle in period j [time]

Subscripts

a, b	alternate moist air-desiccant states
$A, B, \dots$	Fig. 2.16, various operating points of the ideal dehumidifier
d	desiccant
f	evaluated at fluid state

Nomenclature (continued)

i	combined potential index
int	evaluated at intersection point of F_1 characteristic through the period j inlet state and the F_2 characteristic through the period 3-j inlet state
id	ideal outlet state
j	period index
l	face index, l = 1-inlet, l = 2-exit
load	load
m	evaluated at, or in equilibrium with, the matrix state
min	minimum
opt	optimal
p	processing
P	processing
R	regenerating
reg	regenerating
room	evaluated at room air state
s	evaluated at shock state
sys	system
t	heat transfer or temperature
v	vapor, or vaporization
w	mass transfer or moisture
wv	water vapor

Nomenclature (continued)

Superscripts

- . rate
- average value

CHAPTER 1

INTRODUCTION

Air conditioning, drying and cooling systems need mechanical work or a heat source at high or low temperature to operate in a continuous cycle [1]. The impending depletion and rising cost of nonrenewable energy sources have caused a boost in the research on the utilization of renewable energy sources. One promising area is desiccant air conditioning [2]. In these systems, a process air stream is first dehumidified by passing it through a solid adsorbent matrix and subsequently cooled to meet the air conditioning load.

1.1 Solid Desiccant Air Conditioning Systems

Various configurations for air conditioning systems utilizing a solid desiccant dehumidifier have been presented and studied [2,5,7,9,10,16,22]. An excellent overview is given in [3].

A ventilation cycle configuration is shown schematically in Figure 1.1. Process air is drawn from the outside and is passed through the dehumidifier where it is dehumidified and simultaneously heated. This dry air stream then enters a regenerator where it is cooled to approximately room temperature. It is further processed in an evaporative cooler and enters the dwelling cool and dry enough to maintain room comfort. Regenerating air is drawn from the room, humidified and concurrently cooled in an evaporative cooler,

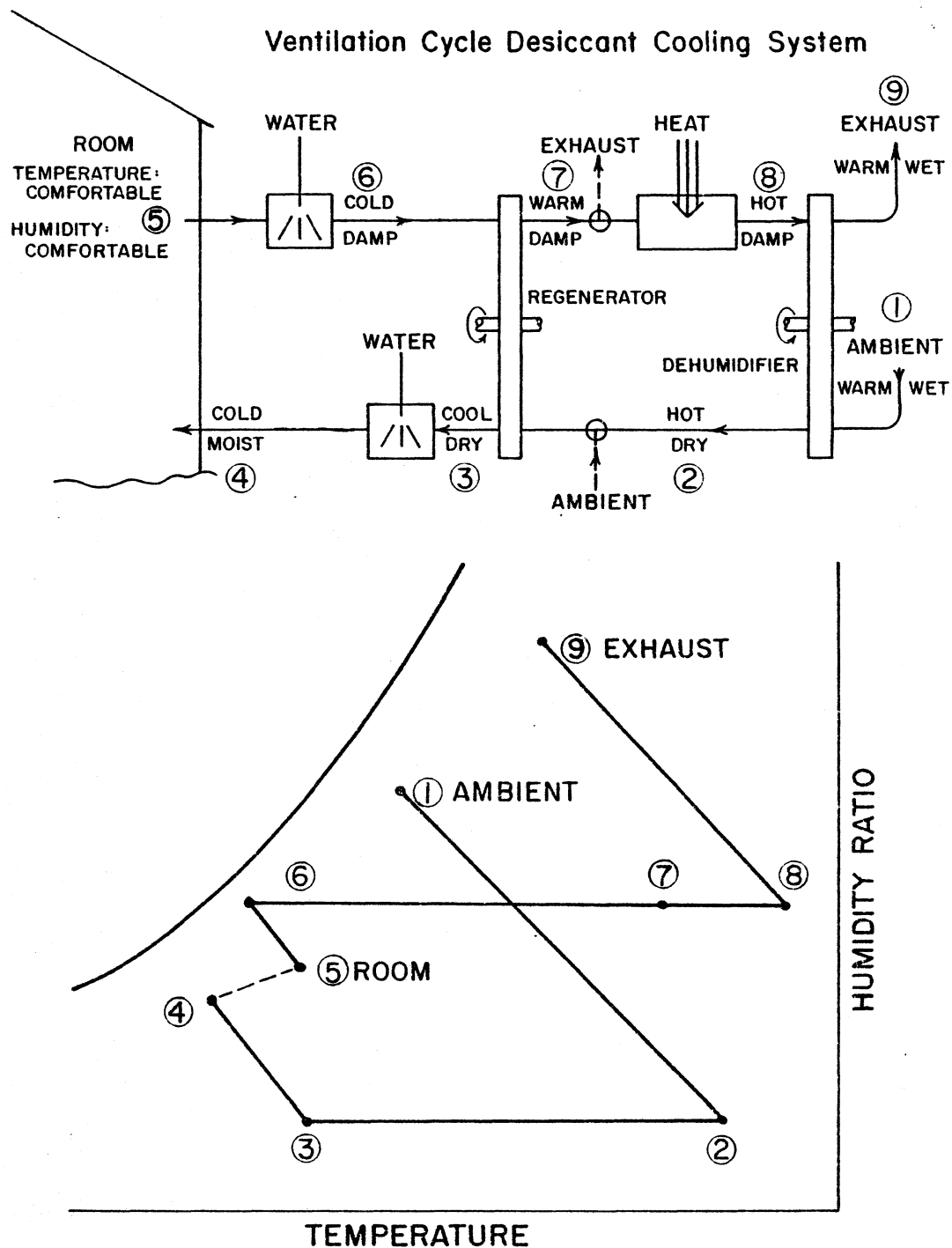


Figure 1.1 Schematic and psychrometric diagram of a ventilation cycle desiccant cooling system (from ref [3])

and subsequently heated by heat exchange with the process air stream in the regenerator. The air stream is further heated by an auxiliary energy source, such as gas or solar, and is then passed through the adsorbent matrix to regenerate the moist desiccant.

In contrast to the ventilation cycle which supplies processed ambient air to the conditioned space, the recirculation cycle, illustrated in Figure 1.2, uses room air to process in the dehumidifier while ambient air is utilized to regenerate the matrix. The choice of cycle depends on ambient and operating correlations.

The prototype systems currently under development [7,9,10] are based on these two cycles, but alternatives such as the Dunkle cycle [2], have been suggested. Desiccant air conditioners in combination with a conventional vapor compression unit have been studied and show good prospects for future utilization in commercial applications [5,6]. Cooled dehumidifiers have been proposed as an alternative to adiabatic operated dehumidifiers, however, performance improvement is questionable [7,8].

Three complete prototype desiccant cooling systems have been developed and tested under DOE's National Solar Desiccant Cooling Program [4,12]. The AiResearch cooling system [9] employs a silica gel dehumidifier, arranged as a packed bed rotating drum, in a ventilation type air conditioning unit. In the IGT SOLAR-MEC systems [10], the dehumidifier is arranged as a rotating wheel with axial counterflow of both air streams. Molecular sieve is used as the desiccant. The IIT design [7] uses two fixed cross-cooled

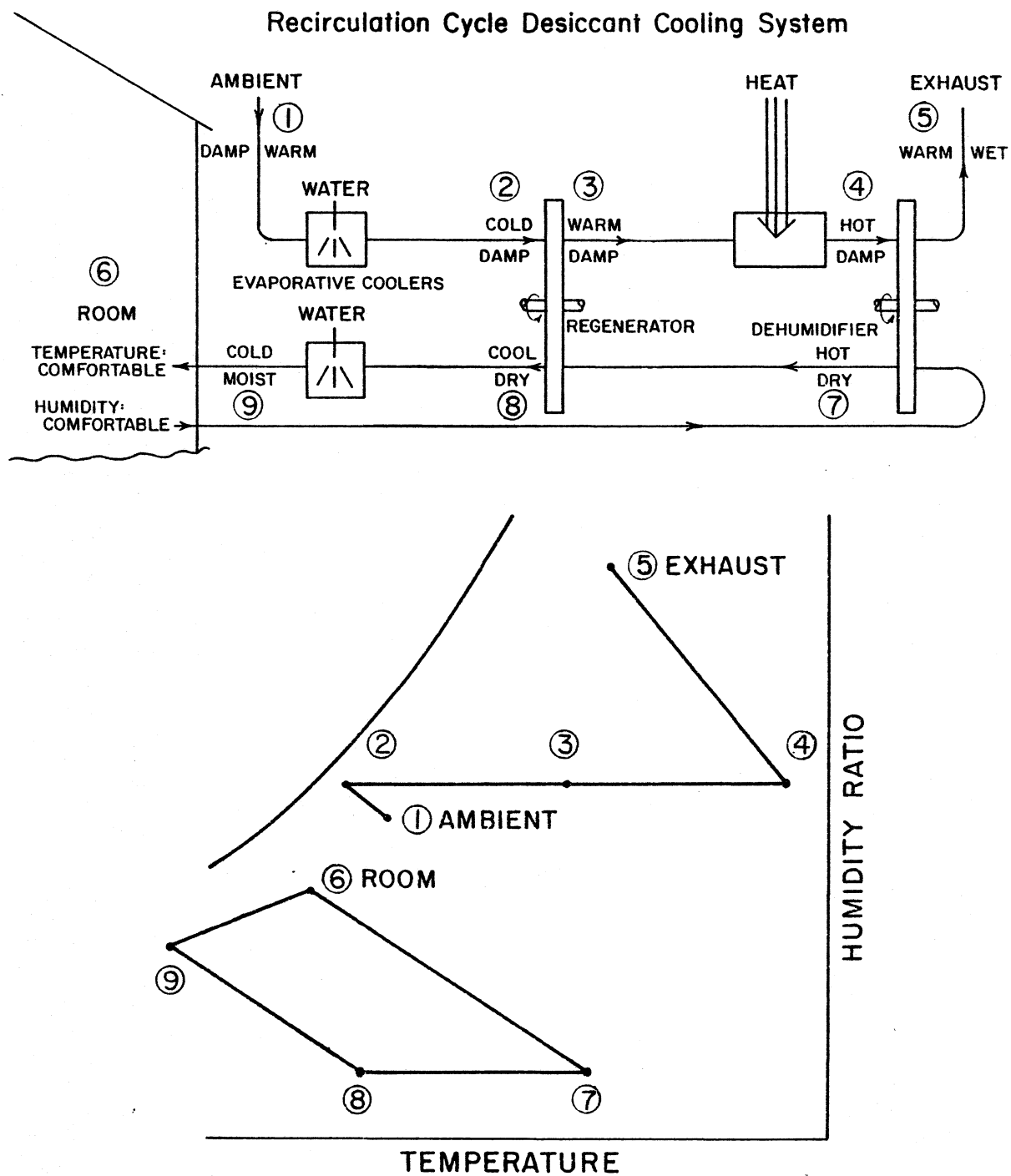


Figure 1.2 Schematic and psychrometric diagram of a recirculation cycle desiccant cooling system (from ref [3])

silica gel matrices that are operated intermittently to dehumidify the process air stream.

While the dehumidifier itself can have several configurations, the design showing the best prospects is the rotary counterflow conception. This design combines high performance with compactness. Advanced dehumidifiers are presently designed combining low pressure drop characteristics with high heat and mass transfer coefficients. The University of California-Los Angeles [11] has tested a dehumidifier in which the air flows through passages coated with a single layer of fine silica gel particles, while the Solar Energy Research Institute [12] is investigating the performance of parallel-passage dehumidifiers.

1.2 Dehumidifier Simulation Models

Numerous publications on computer simulations of desiccant dehumidifier cooling systems have appeared in the literature [8,16,21,22,23,24]. To simulate the dehumidifier performance is, however, a real challenge since it involves a complex combination of both heat and mass transfer. Two kinds of computer simulation models presently exist: those based on finite difference methods and those based on an approximate theoretical analysis.

Several codes [13,14,15] exist that model the dehumidifier by solving the conservation and transfer equations for heat and mass by means of a finite difference method. The matrix is divided into

a two dimensional time-space grid, and the changes in enthalpy and humidity ratio of the matrix and the air stream when passing through a grid element are calculated using various approximations. These codes seem to agree well with experimental data of dehumidifier performance [16,17].

Approximate analytical theories are also available for estimating the dehumidifier outlet states for given inlet conditions, operating parameters and transfer coefficients [13,18,19]. They are all based on a mathematical theory in which the temperature and humidity ratio of the air stream are combined into two potentials. When the conservation equations for mass and energy are expressed in terms of these potentials, they become analogous to the energy equation for a thermal regenerator. Based on this analogy the dehumidifier performance may then be calculated by using existing regenerator performance relations [20].

However, both methods have their flaws. Finite difference programs are very flexible but involve considerable computation, and can therefore not be used in long term system simulations. The analogy theories require far less computation but are only accurate for a limited range of operating parameters and transfer numbers [3].

1.3 Thesis objectives

In a dehumidifier, heat and mass are transferred between the air and the desiccant by convective transport through the boundary

layer in the air, and by diffusional transport inside the desiccant particles. The ratio of overall resistance for mass transfer to overall resistance for heat transfer is defined as the Lewis number.

The initial objective of this thesis was to investigate the effect of the Lewis number on the performance of a rotary dehumidifier. This effect can be described in terms of an effectiveness, in which the dehumidifier is compared to an ideal dehumidifier which assumes infinite transfer coefficients for both heat and mass. This approach leads to the formulation of an ϵ -NTU method for rotary heat and mass exchangers, which is introduced in this work.

In Chapter 2 the ideal dehumidifier is modeled. The governing conservation equations for mass and energy are analyzed by means of a wave analysis. Equations are provided to calculate the dehumidifier outlet states as a function of the inlet states and two dimensionless operating parameters.

Chapter 3 presents the effectiveness of a rotary dehumidifier as a function of two dimensionless transfer coefficients and the Lewis number. Correlations for effectiveness are presented for a silica gel dehumidifier. When combined with the ideal dehumidifier performance equations, a method is obtained which allows rapid calculation of the dehumidifier performance. The method is accurate for a wide range of inlet conditions, operating parameters and transfer numbers.

In system operation the various operating parameters of the dehumidifier, such as wheel revolution speed or regenerating mass

flow rate, must be specified. However, little has been done so far concerning optimizing these parameters with respect to minimal energy use for a given cooling load. In Chapter 4 the use of dehumidifiers in desiccant cooling systems is analyzed, and optimum values for the various operating parameters of the dehumidifier are given. The sensitivity of the energy supply with respect to these parameters is investigated, and the optimal control strategy for a dehumidifier system is discussed.

Chapter 5 provides a summary and recommendations for future research.

CHAPTER 2

THE IDEAL ROTARY DEHUMIDIFIER

In this chapter, a theory is presented for predicting the performance of a rotary heat and mass exchanger with infinite transfer coefficients for heat and mass. To calculate the outlet states for given inlet conditions, only the conservation equations for mass and energy thus need to be considered.

These equations, when expressed for a differential matrix element, take the form of a set of non-linear hyperbolic partial differential equations with respect to time and space. Representative equations for this type of P.D.E.s are the compressible gas dynamic equations and, because of their importance, appropriate analytical techniques have been developed to analyze this type of equations [30,31].

One technique, known as the method of characteristics, has been applied to the conservation equations of an ideal dehumidifier [18]. Based on this analysis, the analogy theories were introduced to predict real dehumidifier performance [13,18,19]. However, it is known that variables such as fluid state properties or velocities, described by nonlinear hyperbolic P.D.E.s, can show a discontinuous behavior in time and space [31]. In these cases, depending on the initial value problem associated with the equations, the method of characteristics is not applicable.

In this chapter it is shown that, for a certain range of the operating parameters of a rotary dehumidifier, the performance of the dehumidifier is governed by such discontinuous behavior. For this operating region, where the analogy theory fails, a theory is introduced in which the conservation equations are analyzed by means of the shock wave technique. The techniques are complementary. When combined they provide a set of analytical equations which allows prediction of the performance of an ideal dehumidifier for the entire range of operating parameters, and for any inlet condition. These equations are presented in the final part of this chapter.

In section 2.1, the dehumidifier nomenclature is defined. The model with the assumptions is formulated in section 2.2 and subsequent sections present the method of characteristics and the shock wave method, as applied to the dehumidifier conservation equations. Finally, the presented theory is compared with a general computer code which solves the governing equations numerically.

2.1 System Description

The nomenclature and coordinate system for the rotary dehumidifier are illustrated in Figure 2.1. The adsorbent matrix is arranged as a rotating cylindrical wheel of length L and has a total mass of dry desiccant M_d .

Two air streams are blown in counterflow through the regenerator. The process air stream, indicated by subscript $j=1$, has a

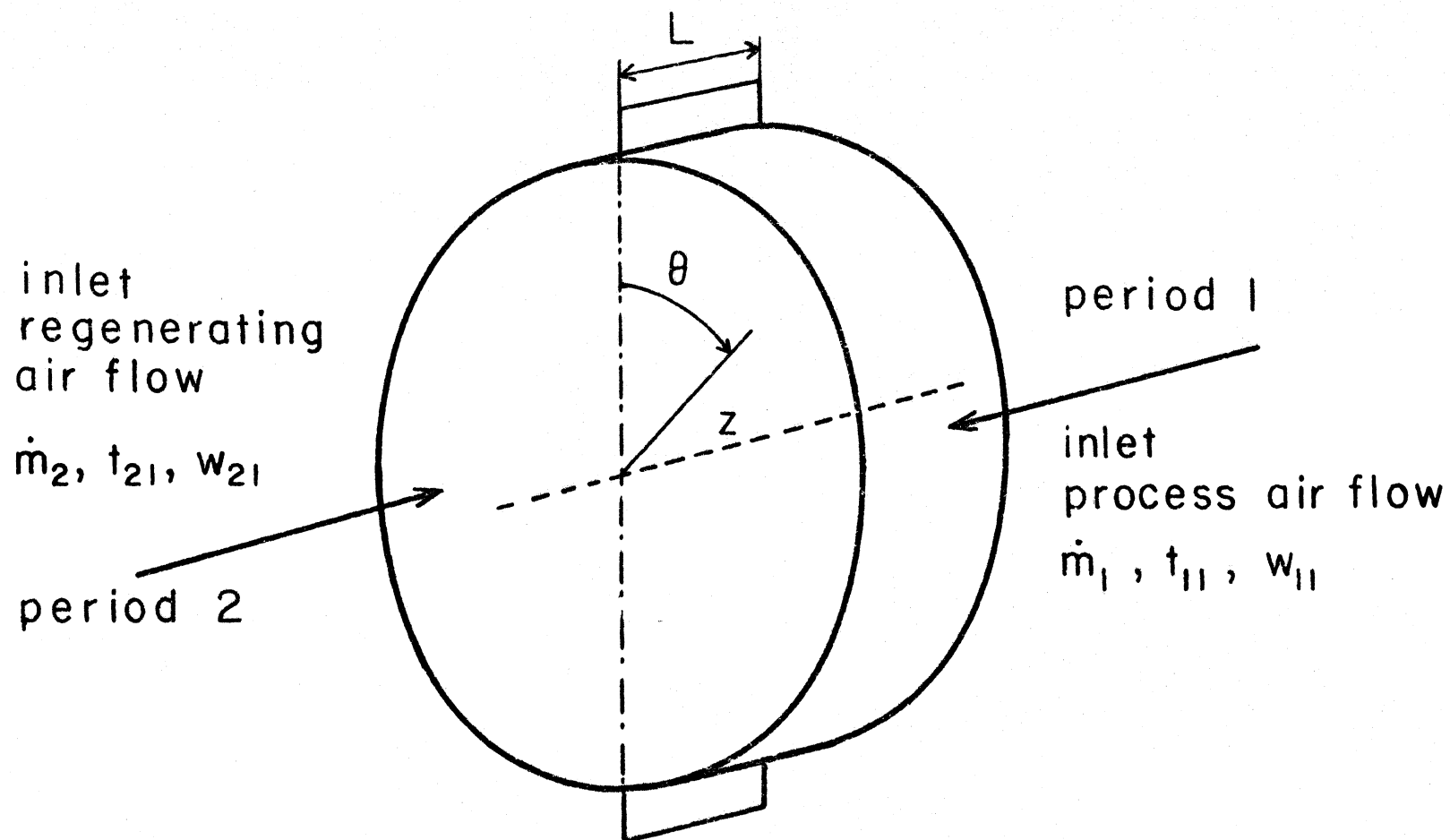


Figure 2.1 Nomenclature and coordinate system for the rotary heat and mass exchanger

low temperature and high relative humidity while the regenerating air stream, indicated by subscript $j=2$, has a high temperature and low relative humidity. The inlet face of the wheel is indicated by subscript $l=1$, while the corresponding exit face is indicated by subscript $l=2$. Both indices are used to describe the various state properties of the alternate air streams: the temperature, t_{jl} , the relative humidity ratio, w_{jl} , and the specific enthalpy i_{jl} . Dry air mass flow rates are symbolized by $\dot{m}_j$.

The additional subscripts f and m may be used to distinguish between fluid and matrix properties. Upper case characters W_m and I_m are used to symbolize the matrix water content and specific enthalpy.

For each period, the axial coordinate, z , is defined as positive in the fluid flow direction, while the rotary position is indicated by the time coordinate θ . The duration of period j is θ_j , and the time of a complete wheel rotation is $T = \theta_1 + \theta_2$.

2.2 Model Formulation and Assumptions

2.2.1 Assumptions

The model which describes the exchange of heat and mass between the moist air and the desiccant matrix, and which is presented in the next section, is based on the following standard assumptions [3]:

1. The matrix is modeled as being of parallel passage form, consisting of a homogeneous solid with constant matrix

characteristics and porosity, through which an air-water vapor mixture flows with constant velocity. Pressure drop effects are neglected.

2. The state properties of the air streams are spatially uniform at the inlet of each period. There is no mixing or carry-over of both streams.
3. A one dimensional approach is applied: there is no radial variation of fluid or matrix states, and diffusion fluxes of heat and mass due to tangential gradients of matrix and air state properties are neglected.
4. The axial heat conduction and water vapor diffusion flux are negligible in both the matrix and the air streams.
5. The thermal and moisture capacities of the air entrained in the matrix is negligible compared to the matrix capacities.
6. Transport of water vapor occurs only through ordinary diffusion and transport of heat occurs only through ordinary heat conduction. Flux coupling is neglected.
7. The heat and mass transfer processes between the desiccant matrix and the air stream can be described by lumped transfer coefficients.

In order to model the ideal dehumidifier, the following additional assumption is made:

8. At all times, each differential desiccant-moist air subsystem is in complete thermodynamic equilibrium (i.e., thermal and vapor pressure equilibrium). This assumption requires in-

finite overall transfer coefficients for both heat and mass.

In addition to these assumptions, the counterflow geometry will be used, and only the steady state performance of the dehumidifier is considered.

2.2.2 Model Formulation

Under the assumptions 1 to 7, the conservation and transfer rate equations for period j of the heat and mass regenerator have been written as [13]:

water vapor conservation:

$$\frac{L}{\tau_{df}} \frac{\partial w_f}{\partial z} + \frac{M_{dj}}{M_{fj}} \frac{\partial w_m}{\partial \theta} = 0$$

water vapor transfer rate:

$$\frac{L}{\tau_{dj}} \frac{\partial w_f}{\partial z} = \frac{h_{wj} A_j}{M_{fj}} (w_m - w_f) \quad j = 1, 2 \quad (2.1)$$

energy conservation:

$$\frac{L}{\tau_{dj}} \frac{\partial i_f}{\partial z} + \frac{M_{dj}}{M_{fj}} \frac{\partial i_m}{\partial \theta} = 0$$

energy transfer rate:

$$\frac{L}{\tau_{dj}} \frac{\partial i_f}{\partial z} = \frac{h_j A_j}{M_{fj}} (t_m - t_f) + i_{wv} \frac{h_{wj} A_j}{M_{fj}} (w_m - w_f)$$

In these expressions, written with respect to a coordinate system fixed to the rotating wheel, τ_{dj} is the dwell time of the air in period j ; M_{fj} and M_{dj} are the entrained dry air mass, respectively, dry desiccant mass for period j ; A_j is the total internal transfer area between the air and the desiccant; h_j is the lumped heat transfer coefficient; h_{wj} is the lumped mass transfer coefficient; i_{wv} is the water vapor specific enthalpy; and w_m is the humidity ratio of the air if the air would be in thermodynamic equilibrium with the desiccant at the temperature of the desiccant, t_m , and the water content of the desiccant, W_m .

If further assumption 8 is considered,

$$\begin{aligned} t_{fj} &= t_m \\ w_{fj} &= w_m \end{aligned} \quad j = 1, 2 \quad (2.2)$$

the conservation equations may be written as

$$\frac{L}{\tau_{dj}} \frac{\partial w_m}{\partial z} + \frac{M_{dj}}{M_{fj}} \frac{\partial W_m}{\partial \theta} = 0 \quad (2.3)$$

$$\frac{L}{\tau_{dj}} \frac{\partial i_m}{\partial z} + \frac{M_{dj}}{M_{fj}} \frac{\partial i_m}{\partial \theta} = 0$$

The operating parameters Γ_1 and Γ_2 are defined as

$$\Gamma_j = \frac{M_{dj} \tau_{dj}}{\theta_j M_{fj}} = \frac{M_d}{T \dot{m}_j} \quad j = 1, 2 \quad (2.4)$$

and the following dimensionless coordinates are introduced

$$x = \frac{z}{L}$$

$$\tau = \frac{\theta}{\tau_{dj}} \frac{M_{fj}}{M_{dj}} = \frac{\theta}{\theta_j} \frac{1}{\Gamma_j} \quad (2.5)$$

The conservation equations may then be expressed as

$$\frac{\partial w_m}{\partial x} + \frac{\partial W_m}{\partial \tau} = 0$$

$$\frac{\partial i_m}{\partial x} + \frac{\partial I_m}{\partial \tau} = 0 \quad (2.6)$$

where

$$0 \leq x \leq 1$$

$$0 \leq \tau \leq \frac{1}{\Gamma_j} \quad (2.7)$$

Equations (2.6) are coupled through the thermodynamic property relationships for the desiccant-air-water vapor mixture:

$$w_m = w_m (W_m, t_m) \quad (2.8.a)$$

which gives the humidity ratio of the air, w_m , in equilibrium with the desiccant at the temperature t_m and the desiccant water content W_m for a given pressure, which is assumed constant. Relationship (2.8.a) is related to the adsorption isotherm of the desiccant;

$$i_m = i_m (w_m, t_m) \quad (2.8.b)$$

which gives the specific enthalpy of the moist air for given humidity ratio and temperature of the air;

$$I_m = I_m(W_m, t_m) \quad (2.8.c)$$

which gives the specific enthalpy of the desiccant for given moisture content and temperature of the adsorbent.

The initial conditions for this system of equations are:

$$\begin{aligned} w_f(x=0, \tau) &= w_{j1} \\ i_f(x=0, \tau) &= i_{j1} \end{aligned} \quad 0 \leq \tau \leq \frac{1}{\Gamma_j}; j = 1, 2 \quad (2.9)$$

The periodic equilibrium boundary conditions for the matrix state properties are:

$$\begin{aligned} \lim_{\tau_1 \rightarrow (1/\Gamma_1)^-} W_m(x, \tau_1) &= \lim_{\tau_2 \rightarrow 0^+} W_m(1-x, \tau_2) \\ \lim_{\tau_1 \rightarrow (1/\Gamma_1)^-} I_m(x, \tau_1) &= \lim_{\tau_2 \rightarrow 0^+} I_m(1-x, \tau_2) \end{aligned} \quad 0 \leq x \leq 1; j = 1, 2 \quad (2.10)$$

$$\begin{aligned} \lim_{\tau_1 \rightarrow 0^+} W_m(x, \tau_1) &= \lim_{\tau_2 \rightarrow (1/\Gamma_2)^-} W_m(1-x, \tau_2) \\ \lim_{\tau_1 \rightarrow 0^+} I_m(x, \tau_1) &= \lim_{\tau_2 \rightarrow (1/\Gamma_2)^-} I_m(1-x, \tau_2) \end{aligned}$$

Equations (2.6) combined with the property relationships (2.8) and the initial and boundary conditions, (2.9), (2.10), form a sys-

tem of two coupled conservation laws. Each is a hyperbolic partial differential equation, and is non-linear because of the non-linear property relationships. The behavior of the solutions to this type of equations is well known, and it has been proved that two kinds of solutions exist [30].

Equations (2.6) were derived based on the continuum assumption, in which it is presumed that the mean free path length of the molecules involved is small compared to the dimensions of interest. However, phenomena such as detonation waves, shock waves or Knudsen flow do not satisfy this assumption. The conservation laws for these phenomena of course still exist, but they have to be expressed as integral equations rather than as differential equations. Solutions to the differential equations that satisfy the continuum assumption and obey the conservation laws are called genuine or differentiable solutions, in contrast to the solutions to the integral equations that do not satisfy the continuum assumption but still obey the same conservation laws. These solutions are called generalized solutions [30].

Differentiable solutions are obtained by analyzing the differential conservation equations (2.6) by means of the method of characteristics. This technique is outlined in section 2.3. It is shown that these solutions are applicable when a desiccant at high temperature and low water content is being processed by a process air stream at low temperature and high relative humidity.

Generalized solutions are obtained by analyzing the integral conservation equations by means of the shock wave method. These equations are presented and analyzed in section 2.4. It is shown that these solutions are applicable when a desiccant at low temperature and high water content is being regenerated by a regenerating air stream at high temperature and low relative humidity.

2.3 The Method of Characteristics

2.3.1 Theoretical Analysis

The method of characteristics that is outlined in [28-30] will focus on the conservation equations (2.6)

$$\frac{\partial w_m}{\partial x} + \frac{\partial W_m}{\partial \tau} = 0 \quad (2.11)$$

$$\frac{\partial i_m}{\partial x} + \frac{\partial I_m}{\partial \tau} = 0$$

The state properties i_m , I_m and w_m may be eliminated using the property relationships (2.8). Introducing the functions a_i , equations (2.11) may be expressed as

$$a_1 \frac{\partial t_m}{\partial x} + a_2 \frac{\partial W_m}{\partial x} + \frac{\partial W_m}{\partial \tau} = 0 \quad (2.12)$$

$$a_4 \frac{\partial t_m}{\partial x} + a_3 \frac{\partial t_m}{\partial \tau} + a_5 \frac{\partial W_m}{\partial \tau} = 0$$

where the functions $a_i(t_m, W_m)$ are defined as

$$\begin{aligned}
a_1(t_m, W_m) &= \frac{\partial w_m}{\partial t_m} \bigg) W_m \\
a_2(t_m, W_m) &= \frac{\partial w_m}{\partial W_m} \bigg) t_m \\
a_3(t_m, W_m) &= \frac{\partial I_m}{\partial t_m} \bigg) W_m \\
a_4(t_m, W_m) &= \frac{\partial i_m}{\partial t_m} \bigg) W_m \\
a_5(t_m, W_m) &= \frac{\partial I_m}{\partial W_m} \bigg) t_m - \frac{\partial i_m}{\partial w_m} \bigg) t_m
\end{aligned} \tag{2.13}$$

These functions are derivatives of state property functions and therefore are state properties too, only dependent on the desiccant-air-water vapor mixture considered.

To obtain the solution to equations (2.12), $t_m(x, \tau)$ and $W_m(x, \tau)$, these equations are multiplied by $d\tau$, $d\tau \neq 0$, and after the following substitutions are made

$$\frac{\partial t_m}{\partial \tau} d\tau = dt_m - \frac{\partial t_m}{\partial x} dx$$

(2.14)

$$\frac{\partial W_m}{\partial \tau} d\tau = dW_m - \frac{\partial W_m}{\partial x} dx$$

the conservation equations may be written as

$$\begin{aligned}
 (a_1 d\tau) \frac{\partial t_m}{\partial x} + (a_2 d\tau - dx) \frac{\partial W_m}{\partial x} &= -dW_m \\
 (a_3 dx - a_4 d\tau) \frac{\partial t_m}{\partial x} + (a_5 dx) \frac{\partial W_m}{\partial x} &= a_3 dt_m + a_5 dW_m
 \end{aligned} \tag{2.15}$$

This set of partial differential equations is a linear set of algebraic equations in the unknown variables $\frac{\partial t_m}{\partial x}$ and $\frac{\partial W_m}{\partial x}$ with non-constant coefficients. The determinant of this system of equations may be set to zero by a proper choice of dx and $d\tau$

$$\begin{vmatrix} a_1 d\tau & a_2 d\tau - dx \\ a_3 dx - a_4 d\tau & a_5 dx \end{vmatrix} = 0 \tag{2.16}$$

The variable λ is introduced, where λ is defined by

$$dx = \lambda d\tau \tag{2.17}$$

Using this new variable, equation (2.16) may be written as

$$a_1 a_5 \lambda - (a_2 - \lambda)(a_3 \lambda - a_4) = 0 \tag{2.18}$$

This is a quadratic equation in λ with real coefficients and has two roots λ_1 and λ_2 . Because the functions a_i are state property functions, λ_1 and λ_2 are state property functions too.

$$\lambda_i = \lambda_i(t_m, W_m) \quad i = 1, 2 \quad (2.19)$$

Because equations (2.11) are the expressions for physical conservation laws, λ_1 and λ_2 have to be real functions of t_m, W_m [30].

It has been proven that a hyperbolic set of P.D.E.s has a unique solution determined over a certain domain in the $x - \tau$ plane [30]. Hence, if equation (2.16) is satisfied, system (2.15) has a non-trivial solution (i.e., different from the zero-solution) if and only if

$$\begin{vmatrix} a_1 d\tau & -dW_m \\ a_3 dx - a_4 d\tau & a_3 dt_m + a_5 dW_m \end{vmatrix} = 0 \quad (2.20)$$

This equation may be written as

$$a_1 a_3 dt_m + (a_1 a_5 + a_3 \lambda_i - a_4) dW_m = 0 \quad (2.21.a)$$

where

$$dx = \lambda_i d\tau \quad i = 1, 2 \quad (2.21.b)$$

Equation (2.21.b) determines two curves in the $x - \tau$ plane, which are called the characteristic lines of the P.D.E.s. Equations (2.21) are called the differential expressions of the characteristics [28]. This set of equations is identical to equations (2.11) and may be physically interpreted as expressions of state property waves.

Along the characteristic line (2.21.b), equation (2.21.a) holds. If $G_i(t_m, W_m)$ is the integrating factor for this differential expression, $F_i(t_m, W_m)$ may be defined as

$$dF_i = G_i a_1 a_3 dt_m + G_i (a_1 a_5 + a_3 \lambda_i - a_4) dW_m \quad i = 1, 2 \quad (2.22)$$

Along a characteristic line, the state property function F_i satisfies the equation

$$dF_i = 0 \quad i = 1, 2 \quad (2.23)$$

and therefore are constant on a characteristic line. Thus values of constant F_i propagate through the matrix at speed λ_i and this process may be interpreted as a state property wave moving with speed λ_i . This wave speed is a local speed and depends only on the local state properties at the point of interest.

The functions F_i are the Riemann invariants of the P.D.E.s [30]. They have been introduced for desiccant systems by Banks et al. [18], and are called the F_i potentials because they are state property functions of the desiccant-air-water vapor system.

The Method of Characteristics for Nominal Silica Gel

The state property relationships for nominal silica gel are given in Tables 3.1 and 3.2. Using these relationships, Jurinak [3] has developed three-parameter curve fits to the F_1 and F_2 potentials as a function of t_m and w_m . They are given by

$$\begin{aligned}
 F_1 &= 10. + 12. \left[- \frac{2865.}{(t_m + 273.)^{1.490}} + 4.244 w_m^{0.8624} \right] \\
 F_2 &= 3. + 12. \left[\frac{(t_m + 273.)^{1.490}}{6360.} - 1.127 w_m^{0.07969} \right]
 \end{aligned}
 \tag{2.24}$$

It should be reminded that if F_i is an invariant so is F_i^* where

$$F_i^* = f(F_i) \tag{2.25}$$

and where f is any arbitrary differentiable function. Numerical evaluation of equations (2.24) show that lines of constant F_1 resemble lines of constant enthalpy in a psychrometric chart, whereas lines of constant F_2 resemble lines of constant relative humidity ratio. Figure 2.2 illustrates an $F_1 - F_2$ chart with lines of constant temperature and humidity ratio. If F_1 is constant, the temperature increases as the humidity ratio decreases. If F_2 is constant, the temperature increases as the humidity ratio increases.

Lines of constant λ_1 are shown in an $F_1 - F_2$ chart in Figure 2.3, and lines of constant λ_2 are shown in Figure 2.4. λ_1 is the speed at which the F_1 potential moves through the desiccant bed, while λ_2 is the corresponding speed for the F_2 potential. Because λ_1 is about 10 to 100 times larger than λ_2 , the F_1 wave will propagate much faster through the desiccant matrix than the wave carrying the F_2 invariant. From Figures 2.3 and 2.4 it follows that λ_1 and λ_2 are both positive dimensionless numbers.

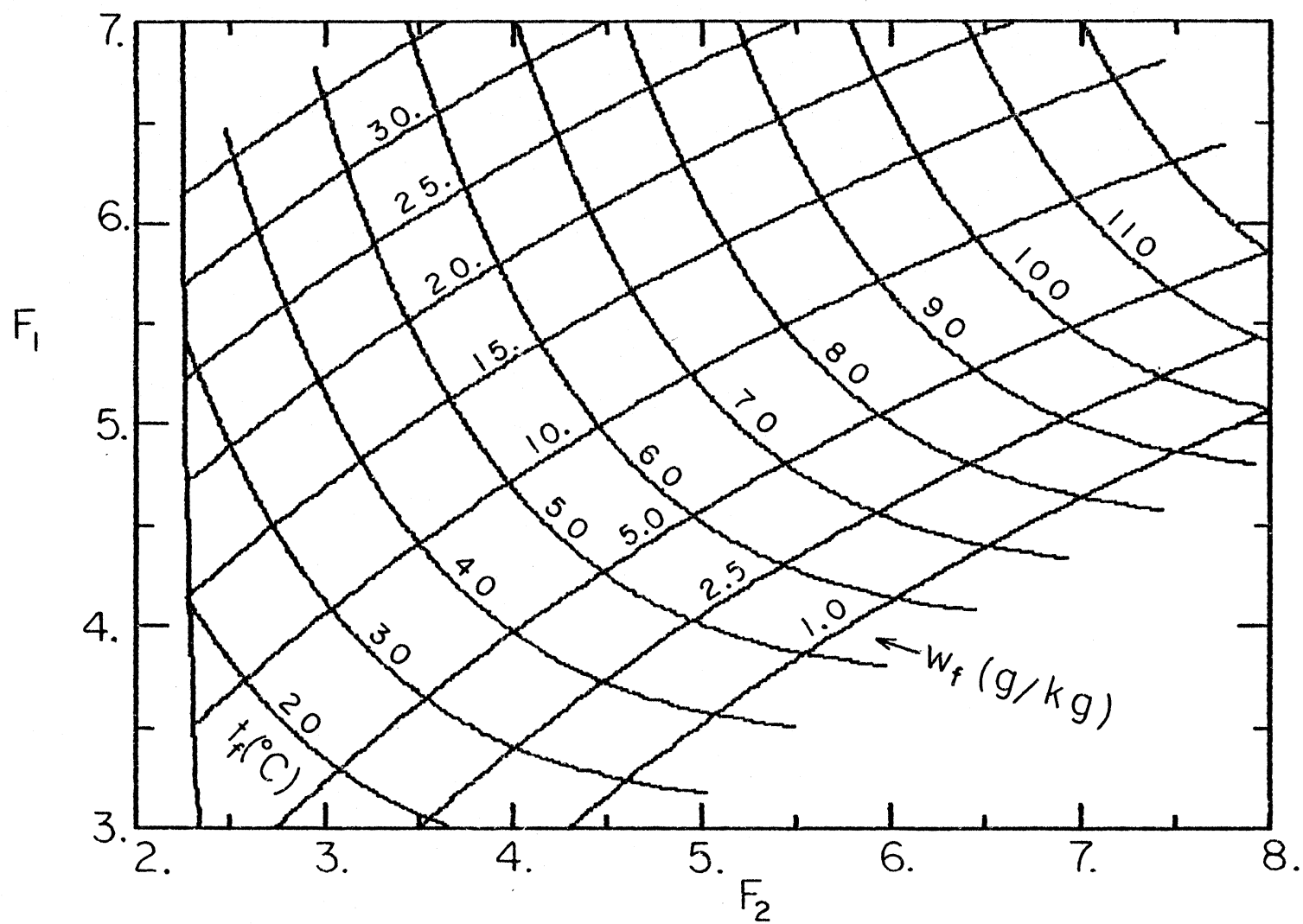


Figure 2.2 Air temperature and humidity ratio in an F_1 - F_2 chart for moist air-silica gel

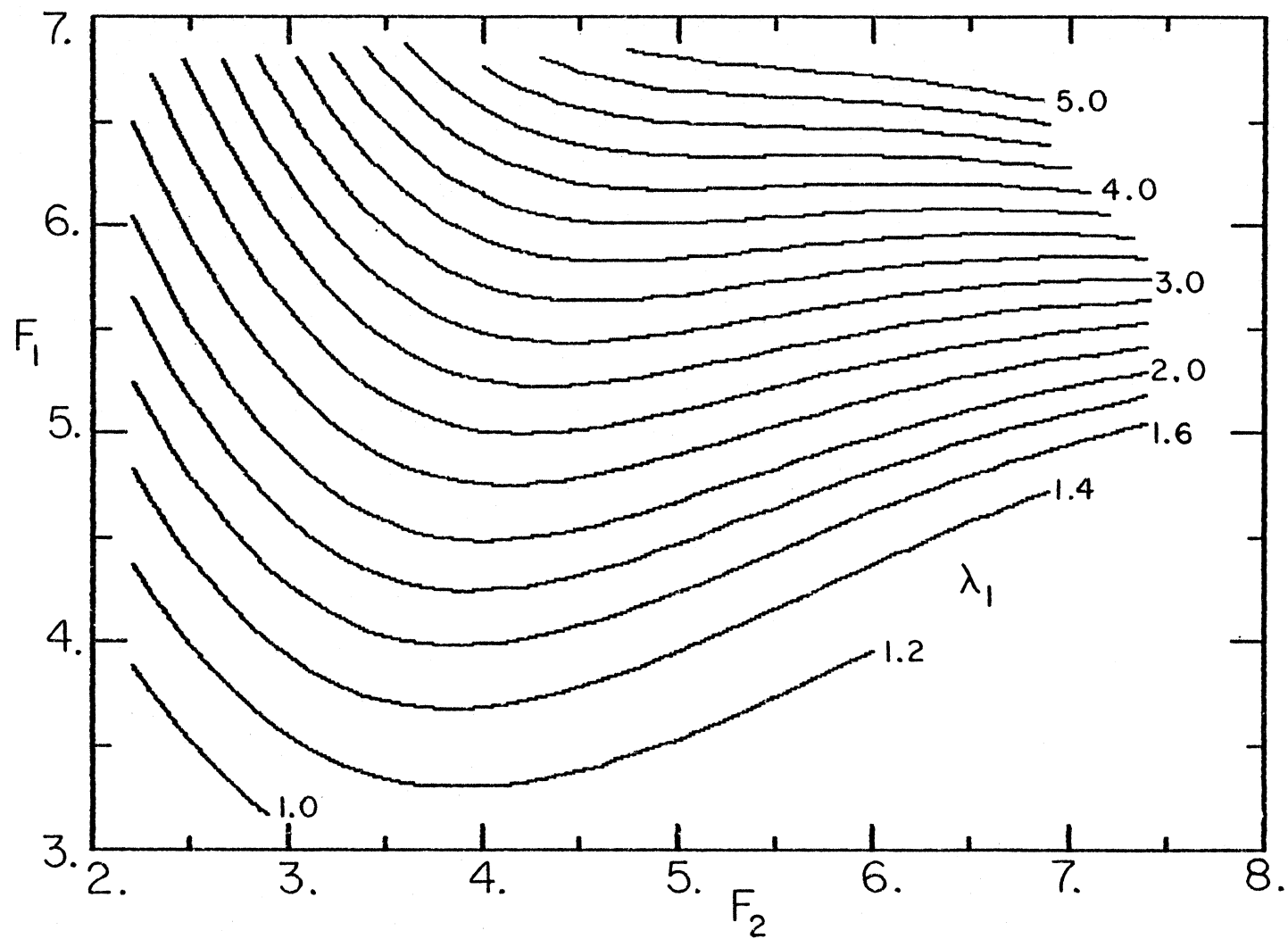


Figure 2.3 Wave speed λ_1 in an F_1 - F_2 chart for moist air-silica gel

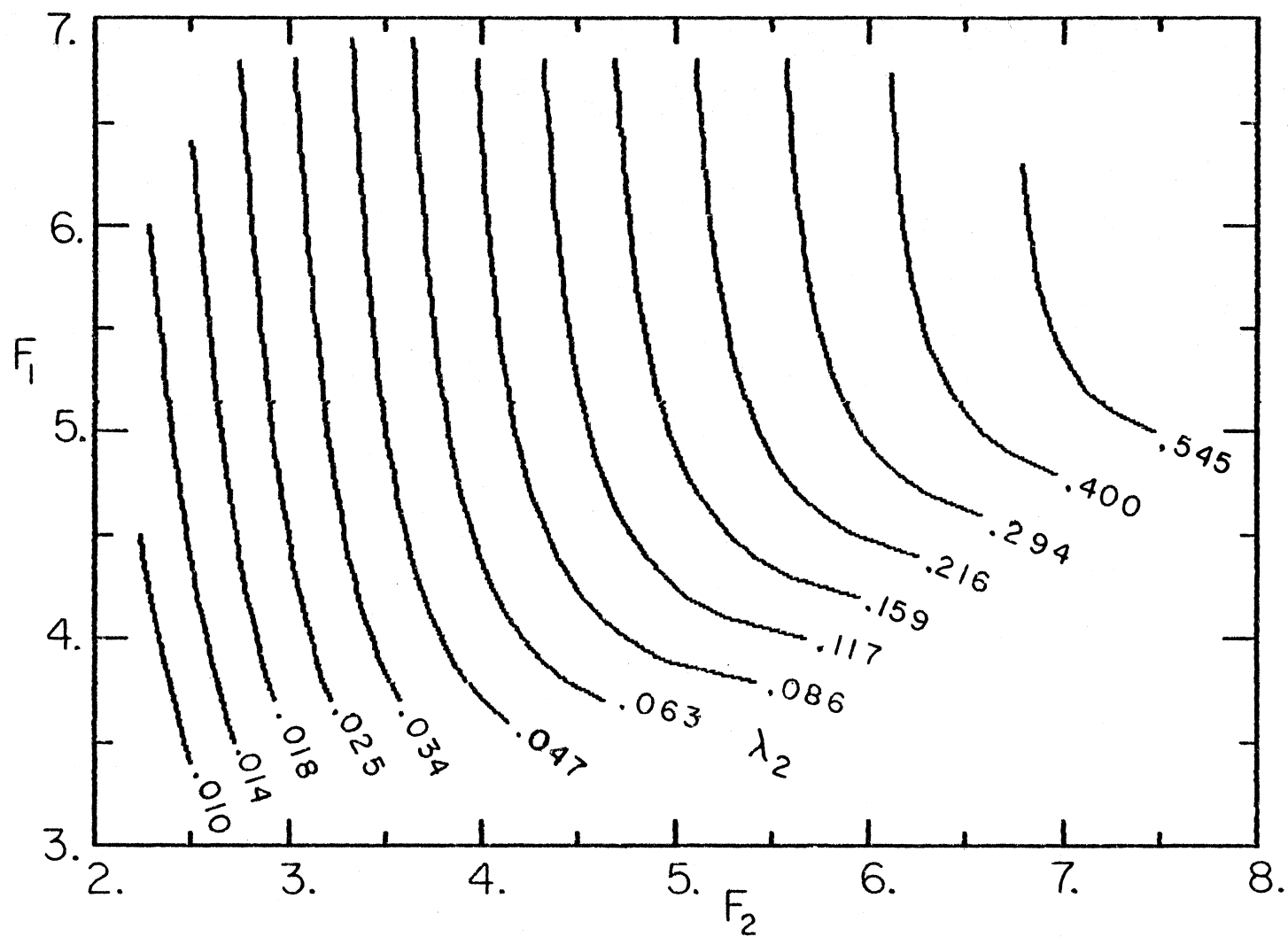


Figure 2.4 Wave speed λ_2 in an F_1 - F_2 chart for moist air-silica gel

Expansion Waves and Shock Waves

Figure 2.3 shows that λ_1 increases with increasing F_1 for constant F_2 , and Figure 2.4 shows that λ_2 increases with increasing F_2 for constant F_1 . Such behavior may cause expansion waves or shock waves to occur, depending on the initial value problem associated with the conservation equations (2.11). Figure 2.5 illustrates various possible F_i distributions associated with the flow of moist air through a desiccant matrix. In Figure 2.5.a, the leading edge of the F_1 wave will move faster than the trailing edge, hence the F_1 wave will further expand as it moves through the desiccant. In Figure 2.5.b, the leading edge of the F_1 wave will move slower than the trailing edge, thus an F_1 shock wave will originate. Figures 2.5.c and 2.5.d show a similar behavior for an F_2 wave.

Figure 2.2 shows that, if the following ranges hold for the process and regenerating air stream inlet state properties

$$\begin{aligned} t_{11} &\leq 35^\circ\text{C} \\ w_{11} &\leq 0.020 \text{ kg/kg} \\ t_{21} &\geq 60^\circ\text{C} \\ w_{21} &\geq 0.010 \text{ kg/kg} \end{aligned} \tag{2.26}$$

then the following relationships hold for the F_i potentials

$$\begin{aligned} F_{1,11} &< F_{1,21} \\ F_{2,11} &> F_{2,21} \end{aligned} \tag{2.27}$$

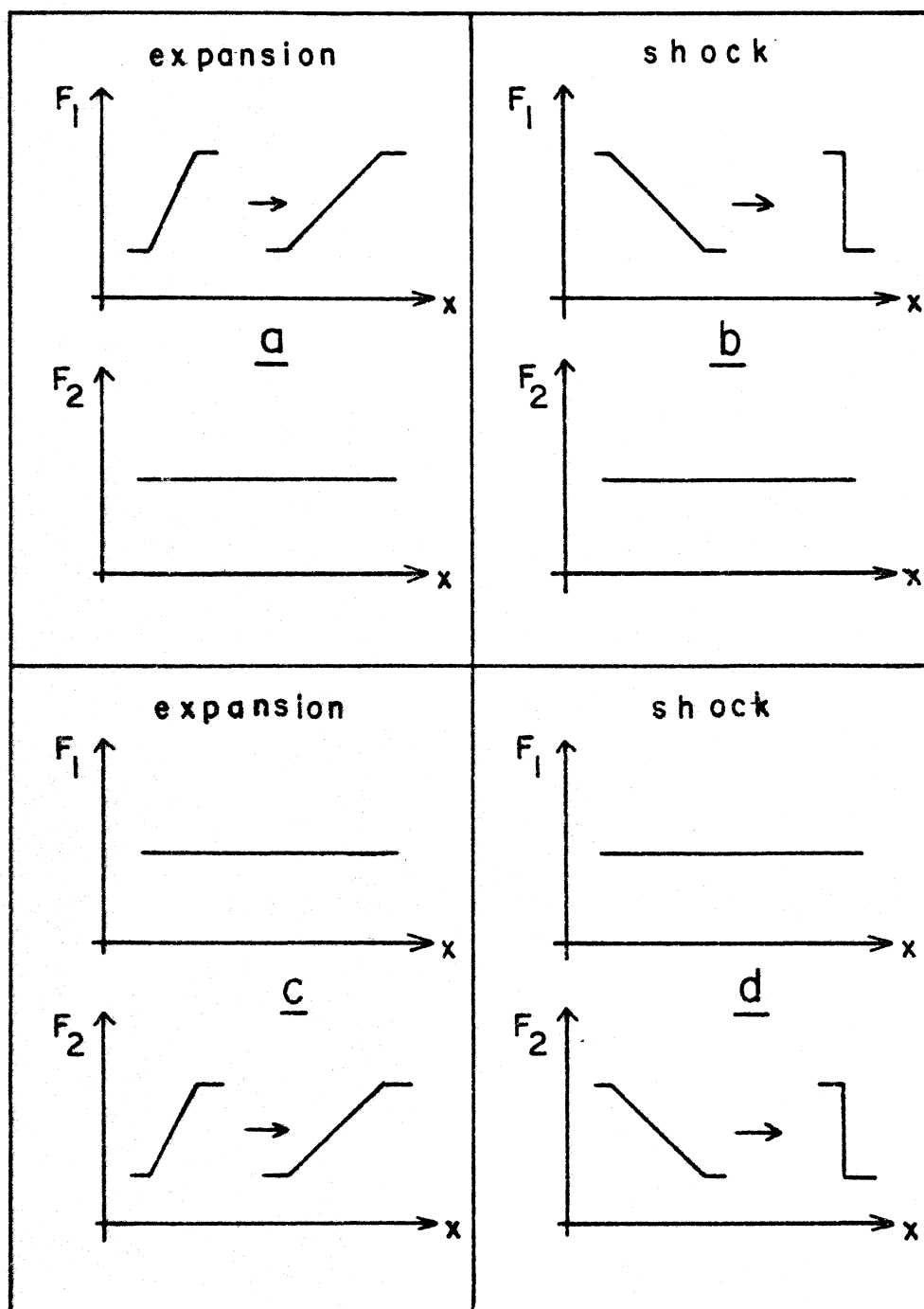


Figure 2.5 Typical initial conditions for moist air flow through a desiccant matrix

Relationships (2.26) cover the entire range of inlet values of interest for the process and regenerating air streams.

Figure 2.6.a illustrates the initial value problem associated with the flow of a process air stream through a desiccant matrix, which is initially in equilibrium with the regenerating air inlet stream. Both waves start out as square waves but will develop in expansion waves, according to Figure 2.5. Figure 2.6.b illustrates the corresponding problem for the sudden flow of a regenerating air stream through a desiccant matrix, which is initially in equilibrium with the process air inlet stream. Again the waves start out as square waves but now will propagate through the desiccant matrix as shock waves.

It thus has been shown that, if the desiccant matrix is being processed by the process air stream, expansion waves will occur. In this case, the method of characteristics is a suitable technique to provide the solution to the initial value problem (2.9) associated with the conservation equation (2.11). It also has been shown that, if the desiccant is being regenerated by the regenerating air stream, shock waves will originate. In this case, the state properties of the air stream and the desiccant vary discontinuously in time and space and the method of characteristics is not applicable. Another method has to be used to provide the solutions to equations (2.11), and this method is outlined in section 2.4.

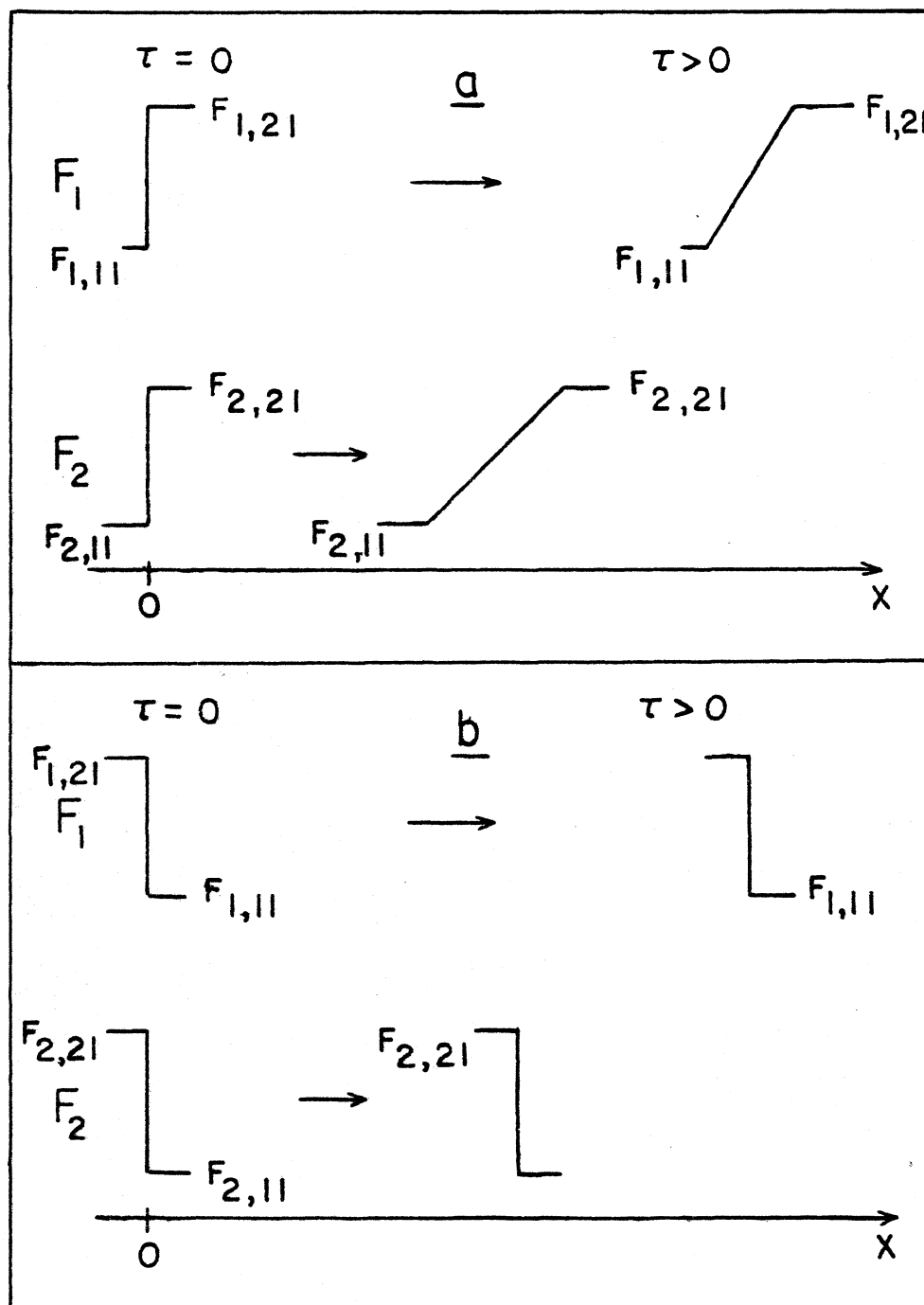


Figure 2.6 Initial value problems associated with processing (a) and regenerating (b) a desiccant matrix

2.3.2 Application: Processing a Desiccant

1. Processing a desiccant which is initially in equilibrium with the regenerating air inlet state.

The characteristic curves defined by (2.21.b) may be drawn in an $x - \tau$ plane. Two and only two characteristic lines (i.e., the F_1 and F_2 characteristics) are passing through each point. Lines carrying two different values for the F_1 potentials can only intersect in a point of discontinuity. Such discontinuity is a shock and is discussed in section 2.4. Figure 2.7 illustrates a typical characteristic diagram showing the solution to the initial value problem, corresponding to the sudden flow of a process air stream through a desiccant matrix which is initially in equilibrium with the regenerating air inlet state. The initial and boundary conditions are

$$\begin{aligned}
 &\text{at } \tau = 0, \quad t(x, \tau) = t_{21} \quad \text{for all } x \geq 0 \\
 &\quad \quad \quad w(x, \tau) = w_{21} \\
 &\text{at } x = 0, \quad t(x, \tau) = t_{11} \quad \text{for all } \tau > 0 \\
 &\quad \quad \quad w(x, \tau) = w_{11}
 \end{aligned} \tag{2.28}$$

Initial value problems like this are called Riemann problems [30].

The F_1 and F_2 waves start out as square waves but propagate through the desiccant as two expanding rarefaction waves, as illustrated in Figure 2.8. This figure shows the distribution of various

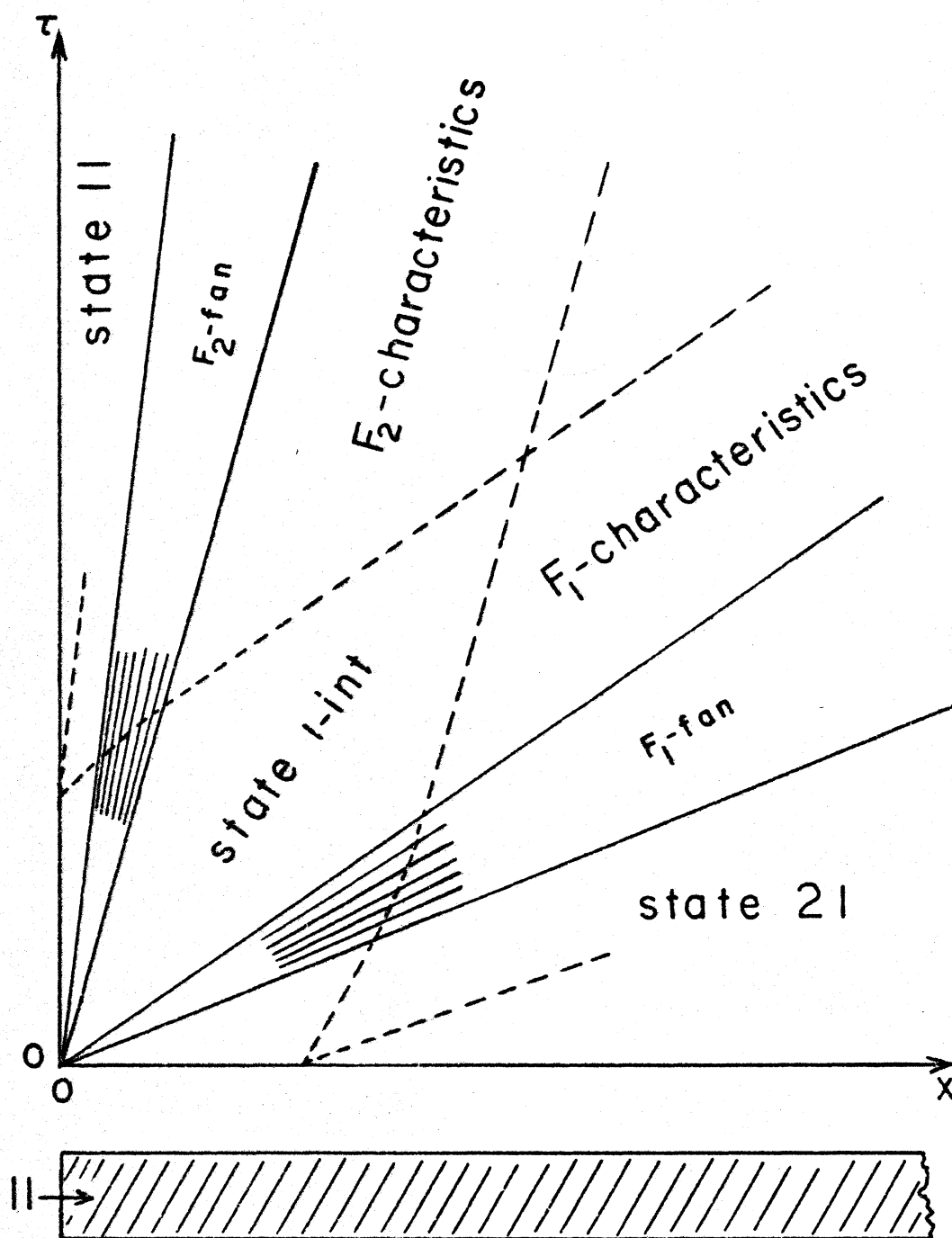


Figure 2.7 Characteristic diagram for processing a fully regenerated desiccant matrix

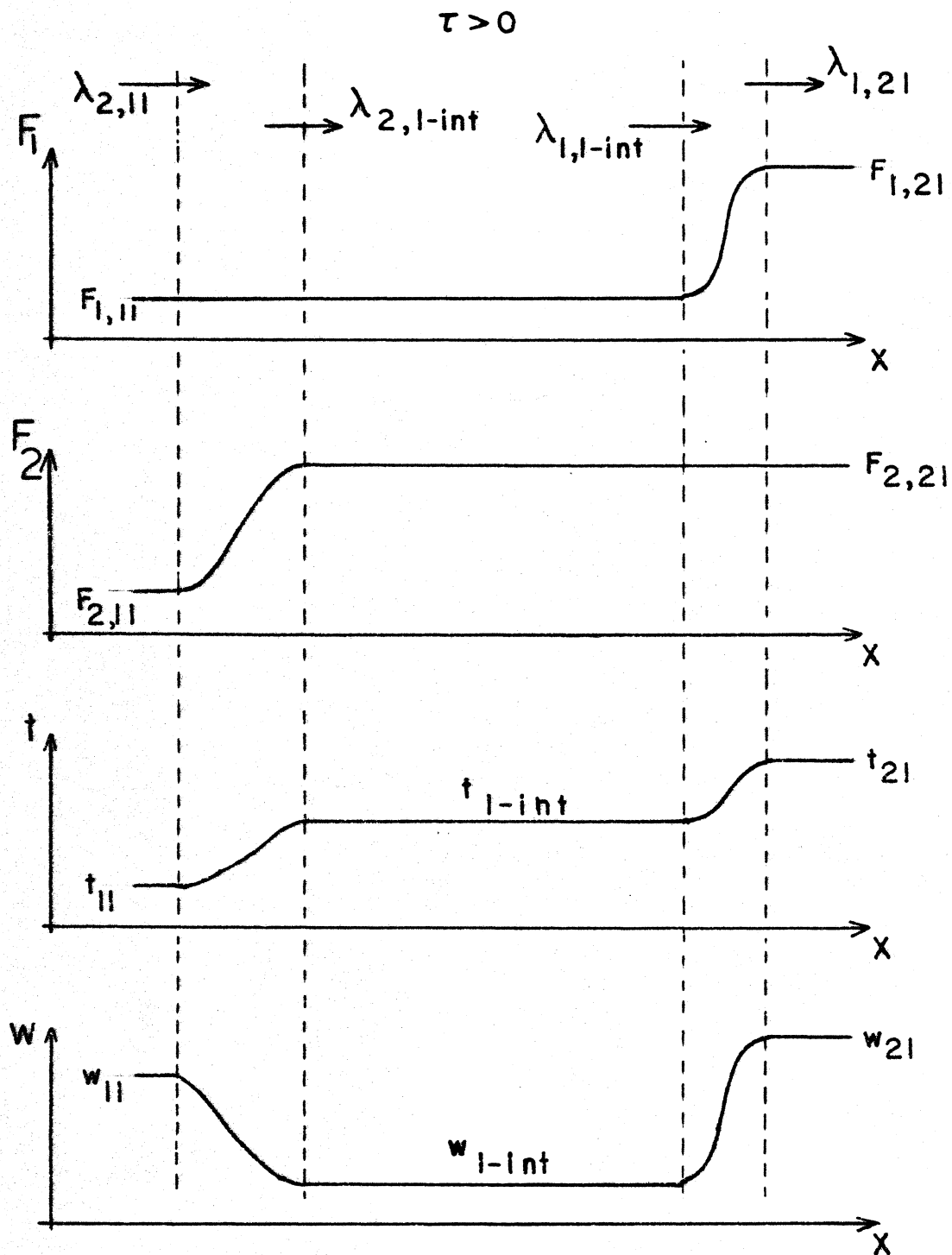


Figure 2.8 Various waves moving through a desiccant matrix that is being processed

state properties of the air stream throughout the desiccant matrix. The rarefaction waves correspond to the two centered diverging fans of lines of constant F_i potential in the $x - \tau$ plane in Figure 2.7. The leading edge of the first wave (i.e., the F_1 wave) carries $F_{1,21}$ and moves at the highest speed, $\lambda_{1,21}$, through desiccant in equilibrium with the regenerating air inlet state. This is the state to the right of the centered F_1 fan. The trailing edge of the second wave (i.e., the F_2 wave) carries $F_{2,11}$ and moves at the lowest speed, $\lambda_{2,11}$, leaving the desiccant in equilibrium with the process air stream. This is the state to the left of the centered F_2 fan in Figure 2.7.

The trailing edge of the F_1 wave carries $F_{1,11}$ and the leading edge of the F_2 wave carries $F_{2,21}$. The state between the two waves is given by

$$\begin{aligned} F_1(t, w) &= F_1(t_{11}, w_{11}) = F_{1,11} \\ F_2(t, w) &= F_2(t_{21}, w_{21}) = F_{2,21} \end{aligned} \tag{2.29}$$

These equations are based on the theorem [30] that, along the F_i characteristic line, the value of the F_i potential does not change if that line passes through a (centered) rarefaction fan of potential F_{3-i} . Figure 2.7 shows an example of an F_2 characteristic line moving through the F_1 fan and a token F_1 characteristic moving through the F_2 fan. Both lines intersect inside the region between the two fans and provide the first-intersection point state,

t_{1-int} , w_{1-int} , which is the solution to (2.29) [18].

Figures 2.2, 2.3 and 2.4 show that if relationships (2.26) are satisfied, the following relationship holds

$$0 < \lambda_{2,11} < \lambda_{2,1-int} < \lambda_{1,1-int} < \lambda_{1,21} \quad (2.30)$$

This explains the existence of two separated centered rarefaction waves in Figure 2.7. Such "a posteriori" existence prove is typical for the solution of Riemann problems. In general, many fan and/or shock configurations are conceivable [30].

Inside the F_1 fan, the solution to (2.11) is given by [30]

$$\lambda_1(t, w) = \frac{x}{\tau} \quad (2.31.a)$$

$$F_2(t, w) = F_{2,21}$$

and inside the F_2 fan, the solution is given by

$$\lambda_2(t, w) = \frac{x}{\tau} \quad (2.31.b)$$

$$F_1(t, w) = F_{1,11}$$

Equations (2.29) and (2.31) are non-linear and may be solved by an iterative method.

Finally the solution to (2.11) and (2.18) is given by three regions of constant state properties: (t_{21}, w_{21}) ; (t_{1-int}, w_{1-int}) ; (t_{11}, w_{11}) ; and by two regions of expanding waves for which the

is a very thin transition region as the result of a shock wave propagating through the desiccant. The F_1 potentials of the shock state are in good approximation given by (2.40)

$$F_{1, 2-s} \simeq F_{1,21} \quad (2.33)$$

$$F_{2, 2-s} \simeq F_{2,11}$$

In this case, no F_2 wave is generated at location $x = 0$ and no F_1 wave at $x = x_o$.

The solution to the Riemann problem (2.32) is illustrated in Figure 2.9. Four regions of constant state properties (t_{11}, w_{11}) , (t_{21}, w_{21}) , (t_{2-s}, w_{2-s}) and (t_{1-int}, w_{1-int}) are separated by two centered rarefaction waves, each originating from a different location. The two fans are slightly curved when they intersect with each other because the wave speeds λ_1 change. The state properties of the air inside the regions covered by the fans are given by equations analogous to (2.31).

2.4 The Shock Wave Method

2.4.1 Theoretical Analysis

In section 2.3, it is shown that the regeneration of a desiccant matrix in equilibrium with the process air inlet state causes the propagation of two shock waves through the matrix. The speed at which these shocks move and the state properties between the

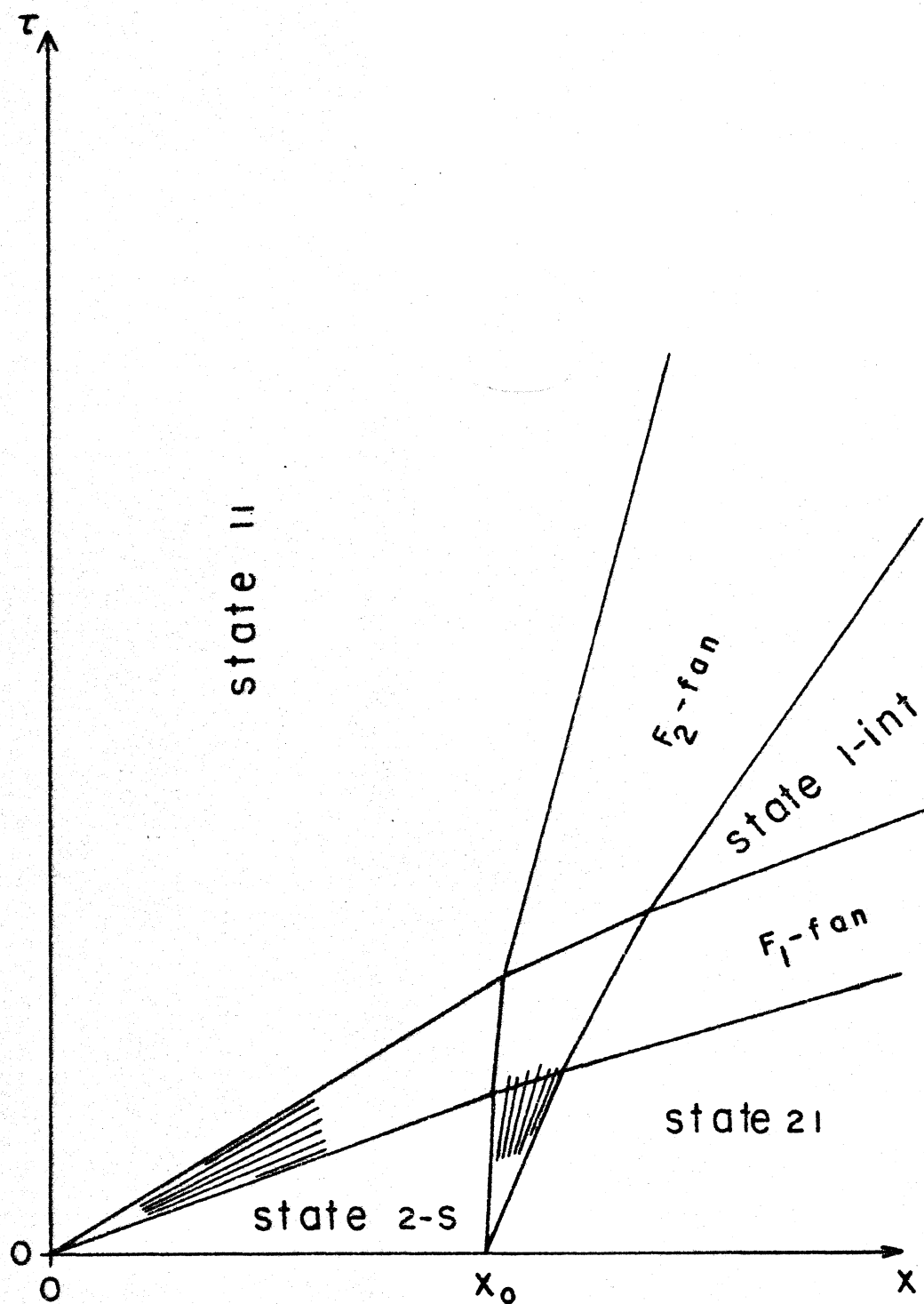


Figure 2.9 Characteristic diagram for processing a partially regenerated desiccant matrix

two shocks cannot be calculated using the method of characteristics. Rather a generalized shock wave method has to be used [30,31]. The analysis outlined here will proceed immediately from the integral conservation laws for water vapor and enthalpy

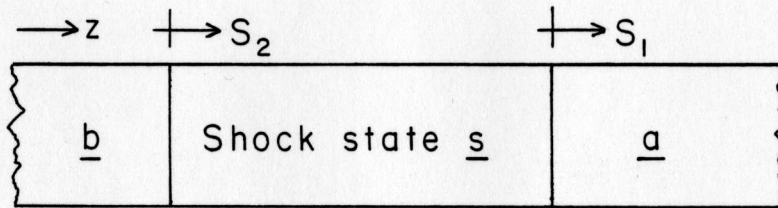


Figure 2.10 Illustration of shockwaves moving through a desiccant matrix

In Figure 2.10 it is assumed that two shock waves propagate through a desiccant matrix at speeds S_1 and S_2 . The state of the air before the first shock is a given state a , while the air state after the second wave is a given state b . The state between the two shocks is called the shock state and is symbolized by state s . It is assumed that the transition of the state properties in the shock waves occurs in a very thin region of negligible thickness.

The conservation equations are applied to the control volume with boundaries at the location of the first shock at time θ , z , and the location of that shock at time $\theta + d\theta$, $z + dz$, as illustrated in Figure 2.11.

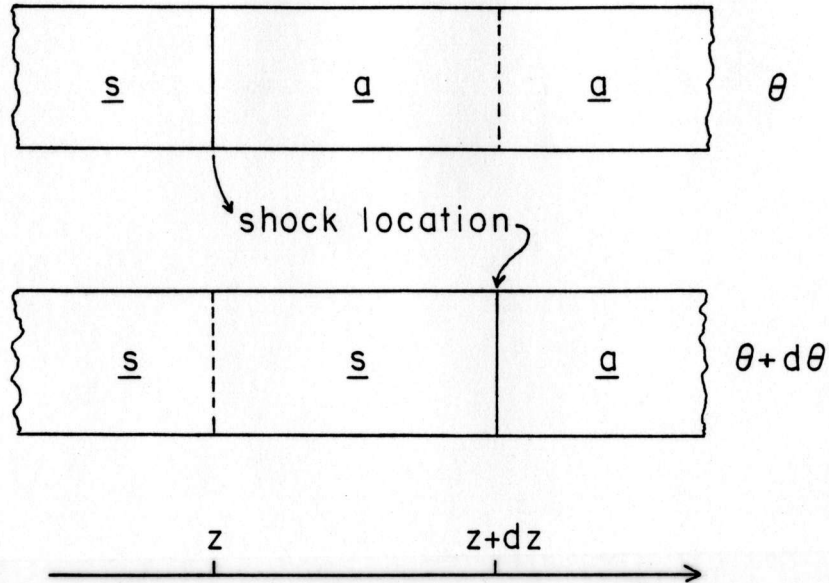


Figure 2.11 Control volume for the shock wave analysis

These equations may be written as

water vapor conservation:

$$\frac{L}{\tau_{dj}} (w_s - w_a) d\theta = \frac{M_{dj}}{M_{fj}} (W_s - W_a) dz$$

$$j = 1 \quad (2.34)$$

energy conservation:

$$\frac{L}{\tau_{dj}} (i_s - i_a) d\theta = \frac{M_{dj}}{M_{fj}} (I_s - I_a) dz$$

The dimensionless variables x and τ are used as defined by (2.5), and the dimensionless shock wave velocity S_1 is introduced

$$S_1 = \frac{dx}{d\tau} \quad (2.35)$$

This shock speed is analogous to the expansion wave speed λ_1 ,
(2.17).

The conservation equations for the first shock may be expressed as

$$(w_s - w_a) = S_1 (W_s - W_a) \quad (2.36)$$

$$(i_s - i_a) = S_1 (I_s - I_a)$$

Equations (2.36) are two equations in the three unknown variables S_1 , t_s and w_s . Two additional equations may be obtained by applying the same control volume analysis to the second shock. The conservation equations are

$$(w_b - w_s) = S_2 (W_b - W_s) \quad (2.37)$$

$$(i_b - i_s) = S_2 (I_b - I_s)$$

S_1 and S_2 may be eliminated from equations (2.36) and (2.37), and the following expressions are obtained

$$(w_s - w_a) (I_s - I_a) = (W_s - W_a) (i_s - i_a) \quad (2.38)$$

$$(w_b - w_s) (I_b - I_s) = (W_b - W_s) (i_b - i_s)$$

Equations (2.38) are two equations in the two unknowns t_s and w_s , and are called the (generalized) Rankine-Hugoniot jump condition [30].

Equations (2.38) need to be solved numerically in combination with the state property relationships (2.8), to provide the shock state properties (t_s, w_s) for given state properties (t_a, w_a) and (t_b, w_b) . For a desiccant-moist air system, these equations have two distinct solutions, of which only one is physically possible. This solution satisfies the jump conditions (2.38) and also the following relationships

$$\begin{aligned}\lambda_{1,a} < S_1 < \lambda_{1,s} \\ \lambda_{2,s} < S_2 < \lambda_{2,b}\end{aligned}\tag{2.39}$$

where S_1 and S_2 are given by (2.36), respectively, (2.37). When applied to the equations of compressible flow, this condition amounts to requiring that material which crosses the discontinuity should suffer an increase of entropy. For this reason condition (2.39) is called the (generalized) entropy condition [30].

The shock state properties and the shock wave velocities are dependent on both states a and b. The shock wave velocities S_i are not state properties of the shock state s.

The Shock Wave Method for Nominal Silica Gel

The jump expressions (2.38) may be solved for nominal silica gel. State a is the process air inlet state and state b is the regenerating air inlet state. Two solutions, 1-s and 2-s are obtained. Numerical calculations show that these solutions may be approximated by the first and second intersection point.

$$F_{1,1-s} \simeq F_{1,1-int} = F_{1,11} \quad (2.40.a)$$

$$F_{2,1-s} \simeq F_{2,1-int} = F_{2,21}$$

and

$$F_{1,2-s} \simeq F_{1,2-int} = F_{1,21} \quad (2.40.b)$$

$$F_{2,2-s} \simeq F_{2,2-int} = F_{2,11}$$

Solution 2-s satisfies the entropy condition (2.39) and is therefore the solution that is physically possible.

2.4.2 Application: Regenerating a Desiccant

1. Regenerating a desiccant which is initially in equilibrium with the process air inlet state.

The initial value problem is given by

$$\begin{aligned}
&\text{at } \tau = 0, \quad t(x, \tau) = t_{11} \quad \text{for all } x \geq 0 \\
&\quad \quad \quad w(x, \tau) = w_{11} \\
&\quad \quad \quad (2.41) \\
&\text{at } x = 0, \quad t(x, \tau) = t_{21} \quad \text{for all } \tau > 0 \\
&\quad \quad \quad w(x, \tau) = w_{21}
\end{aligned}$$

The solution to (2.11) and (2.41) may be illustrated in an $x - \tau$ diagram, showing characteristic lines and shock lines. Shock lines are generated where two F_1 (or F_2) characteristics intersect [30].

The F_1 and F_2 waves start out as square waves and propagate through the matrix at constant speeds, as illustrated in Figure 2.13. This corresponds to the two shock lines in the characteristic diagram illustrated in Figure 2.12. The first shock wave moves at the highest speed, S_1 , through desiccant in equilibrium with the process air inlet state. This is the state to the right of the first shock line. The second shock wave moves at the lowest speed, S_2 , leaving the desiccant in equilibrium with the regenerating air inlet state. The state between the two shocks is the shock state 2-s.

It has been proven [30] that, if an F_1 characteristic line intersects with a shock line formed by the collision of F_{3-i} characteristics, the F_1 potential changes slightly in order to satisfy the jump condition (2.38). Thus the F_1 value on an F_1 characteristic changes slightly when passing through the second shock wave. The F_2 value along a F_2 characteristic changes slightly when that

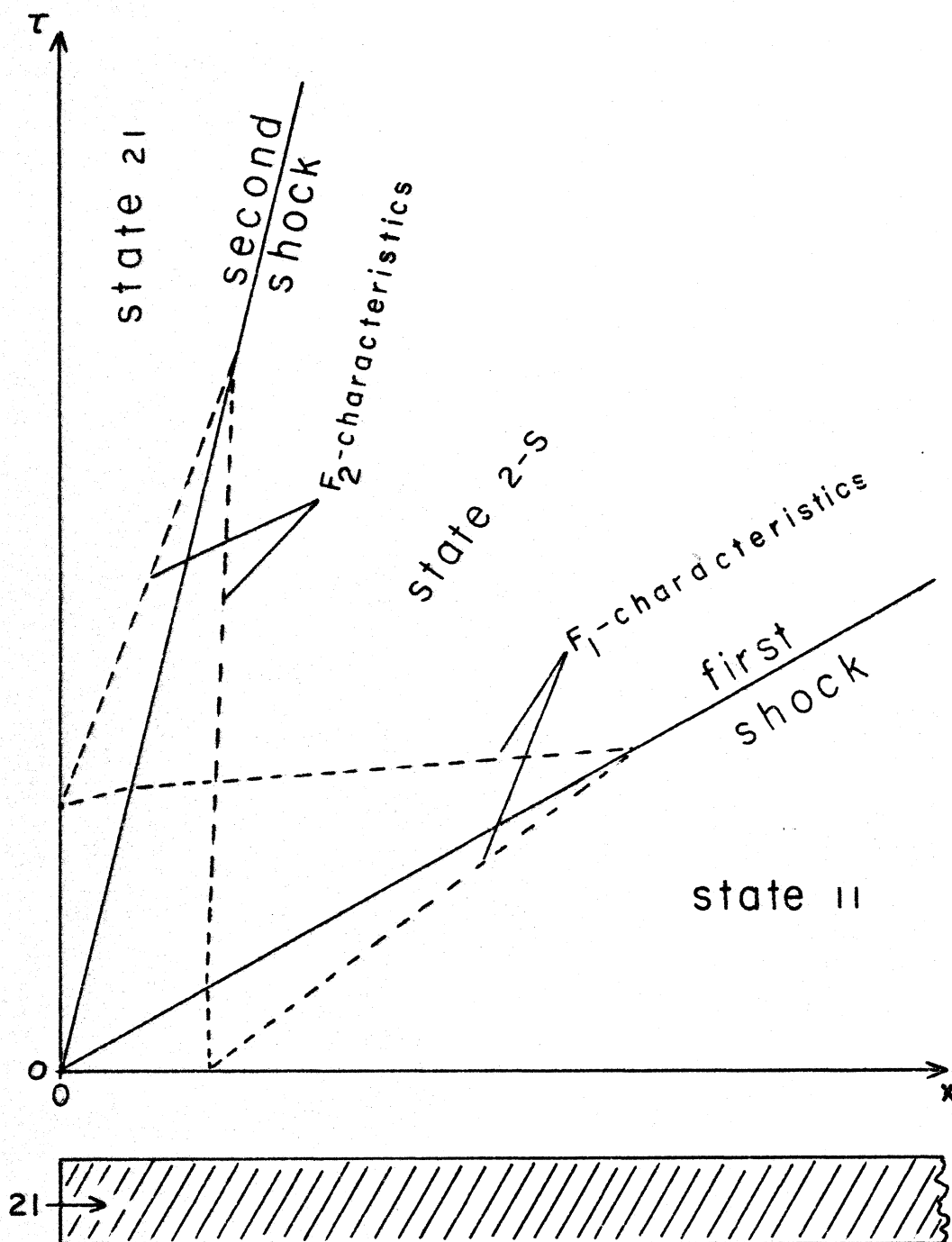


Figure 2.12 Characteristic diagram for regenerating a fully processed desiccant matrix

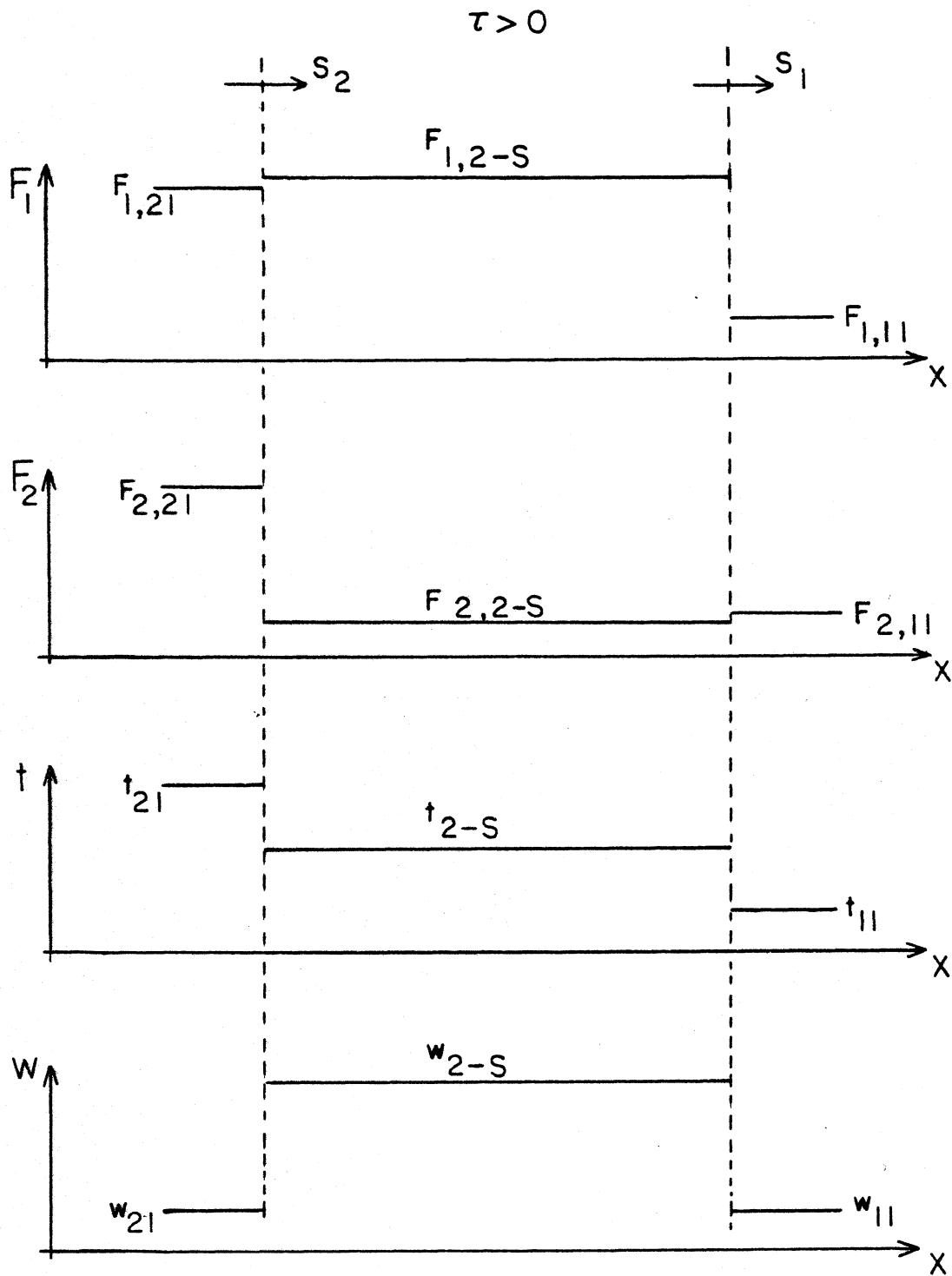


Figure 2.13 Various waves moving through a desiccant matrix that is being regenerated

line passes through the first shock. This explains the relationships (2.40).

Finally, the solution to (2.11) and (2.41) is given by three regions of constant state properties (t_{11}, w_{11}) , (t_{2-s}, w_{2-s}) and (t_{21}, w_{21}) . As an example, Table 2.1 lists some typical state property values for moist air-nominal silica gel.

2. Regenerating a partially processed desiccant.

The following initial value problem is of interest to the discussion of the "regenerating line," presented in section 2.5. This problem is generated when the flow of a process air stream through a desiccant is suddenly stopped, and regenerating air starts flowing in the reverse direction. During processing, the first rarefaction wave has completely moved through the desiccant, and the leading edge of the F_2 rarefaction wave has just reached the end face of the desiccant layer.

$$\begin{aligned}
 &\text{at } \tau = 0, \quad t(x, \tau) = t_{1-\text{int}} \quad \text{for } x = 0 \\
 &\quad \quad \quad w(x, \tau) = w_{1-\text{int}} \\
 &\quad \quad \quad F_1(t, w) = F_{1,11} \quad \text{for all } x, \quad 0 \leq x \leq x_0 \\
 &\quad \quad \quad \lambda_2(t, w) = (1 - x) \Gamma_1 \\
 &\quad \quad \quad t(x, \tau) = t_{11} \quad \text{for all } x, \quad x_0 \leq x \leq 1 \\
 &\quad \quad \quad w(x, \tau) = w_{11} \\
 &\text{at } x = 0, \quad t(x, \tau) = t_{21} \quad \text{for all } \tau > 0 \\
 &\quad \quad \quad w(x, \tau) = w_{21}
 \end{aligned} \tag{2.42}$$

Table 2.1 Typical State Property Data for Nominal Silica Gel

State	1l	2l	1-int*	2-int*	1-s	2-s
t (°C)	25.00	85.00	59.61	37.28	58.52	37.82
w (g/kg)	15.00	15.00	2.75	31.96	2.49	33.02
F ₁ (/)	4.33	6.01	4.32	6.02	4.27	6.11
F ₂ (/)	2.50	5.38	5.35	2.46	5.37	2.46
λ ₁ (/)	1.359	3.728	1.557	2.265	1.510	2.320
λ ₂ (/)	0.013	0.248	0.146	0.017	0.141	0.017
S ₁ = 1.7904			S ₂ = 0.0519			

*See text.

Figure 2.14 illustrates the solution to (2.11) and (2.42) in the $x - \tau$ plane. A strong first shock is generated and propagates at high velocity through the desiccant. This propagation speed gradually decreases as the shock wave moves further through the matrix, due to the continuous change of the state properties in front of that shock. E.g., for the inlet states of Table 2.1, the initial speed is 2.40 and the "steady-state" speed ($x > x_0$) is 1.77.

Initially, the F_2 potential is continuously distributed throughout the matrix. As the F_2 wave moves through the matrix, the slope of that wave gradually increases until a small shock is generated. The strength of that shock grows slowly as the shock proceeds and, eventually, a strong second shock is formed. The velocity of this wave gradually decreases as the shock becomes stronger. E.g., for the inlet states of Table 2.1, this velocity decreases from an initial value of 0.249 to a final value of 0.021.

Numerical solution of (2.11) and the initial value problem (2.42) by a scheme analogous to Glimm's Random Choice Method [33], shows that the shock state in front of the second shock approaches more and more the second intersection point (2.40), when the initial distribution of the F_2 rarefaction wave is more spread out throughout the matrix at $\tau = 0$.

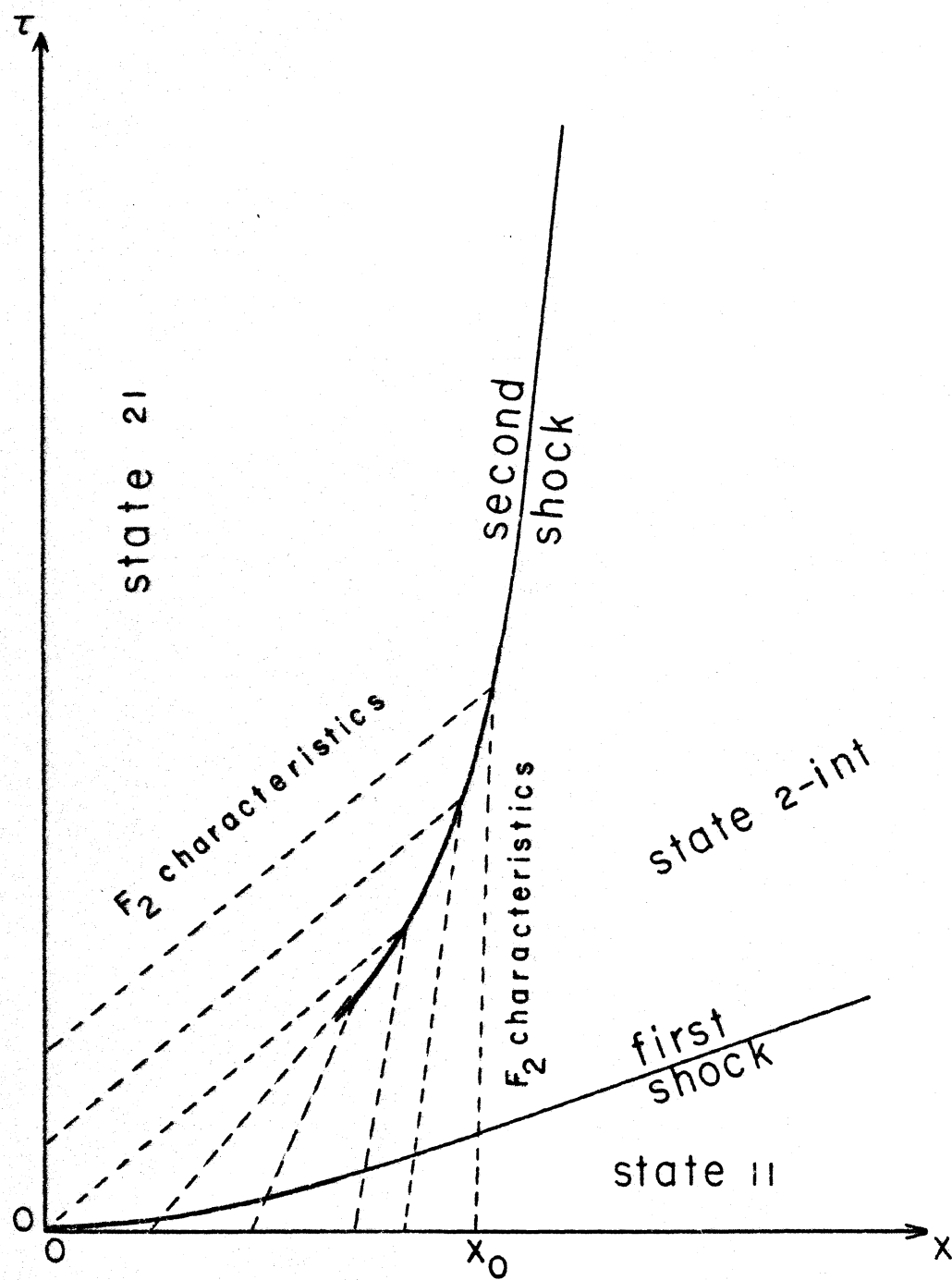


Figure 2.14 Characteristic diagram for regenerating a partially processed desiccant matrix

2.5 The Ideal Rotary Dehumidifier

2.5.1 The Operating Chart

Figure 2.15 illustrates a typical combined characteristic or wave diagram for a counterflow ideal dehumidifier, showing the rarefaction waves during the processing period and the shocks during the regenerating period. The distance that the various waves move through the desiccant depends on the velocity of the wave (2.19), (2.36), which is a function of the inlet states, and on the duration of each period, $1/\Gamma_j$ (2.7). If the trailing edge of the F_2 rarefaction wave reaches the exit face of the desiccant wheel during the processing period (period 1), the desiccant is in complete equilibrium with the process air inlet state before the matrix enters the regenerating period (period 2). Similarly, if the second shock propagates completely through the matrix during the regenerating period, the desiccant is in complete equilibrium with the regenerating air inlet state before it enters the processing period. This leads to the concept of an operating chart, as illustrated in Figure 2.16. Various operating regions are shown in a $\Gamma_1 - \Gamma_2$ diagram for given inlet conditions.

The solid line A-B-D in Figure 2.16 represents the regenerating line.

$$\Gamma_{2,R} = \Gamma_{2,R}(\Gamma_1) \quad (2.43)$$

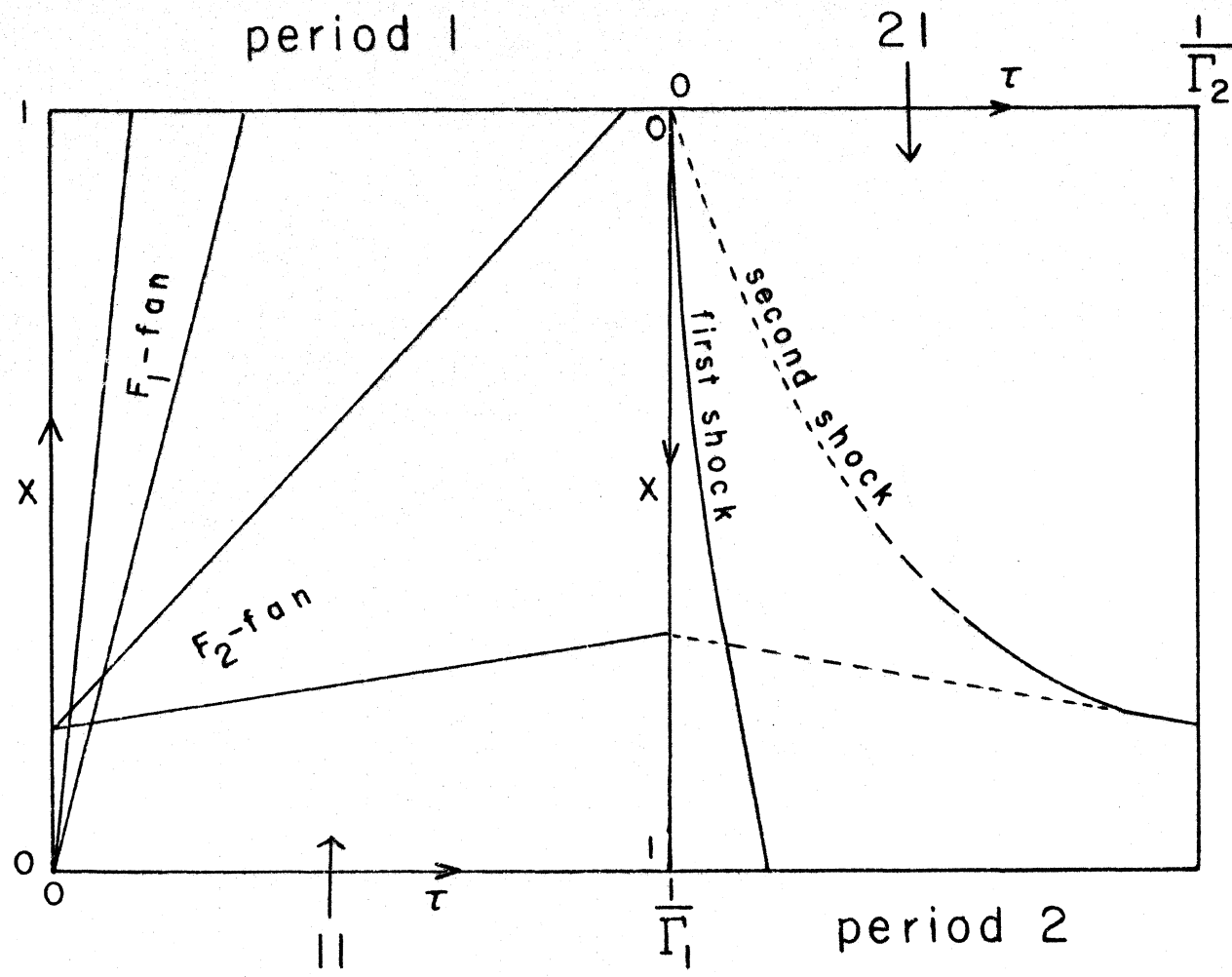


Figure 2.15 Wave diagram for the ideal dehumidifier

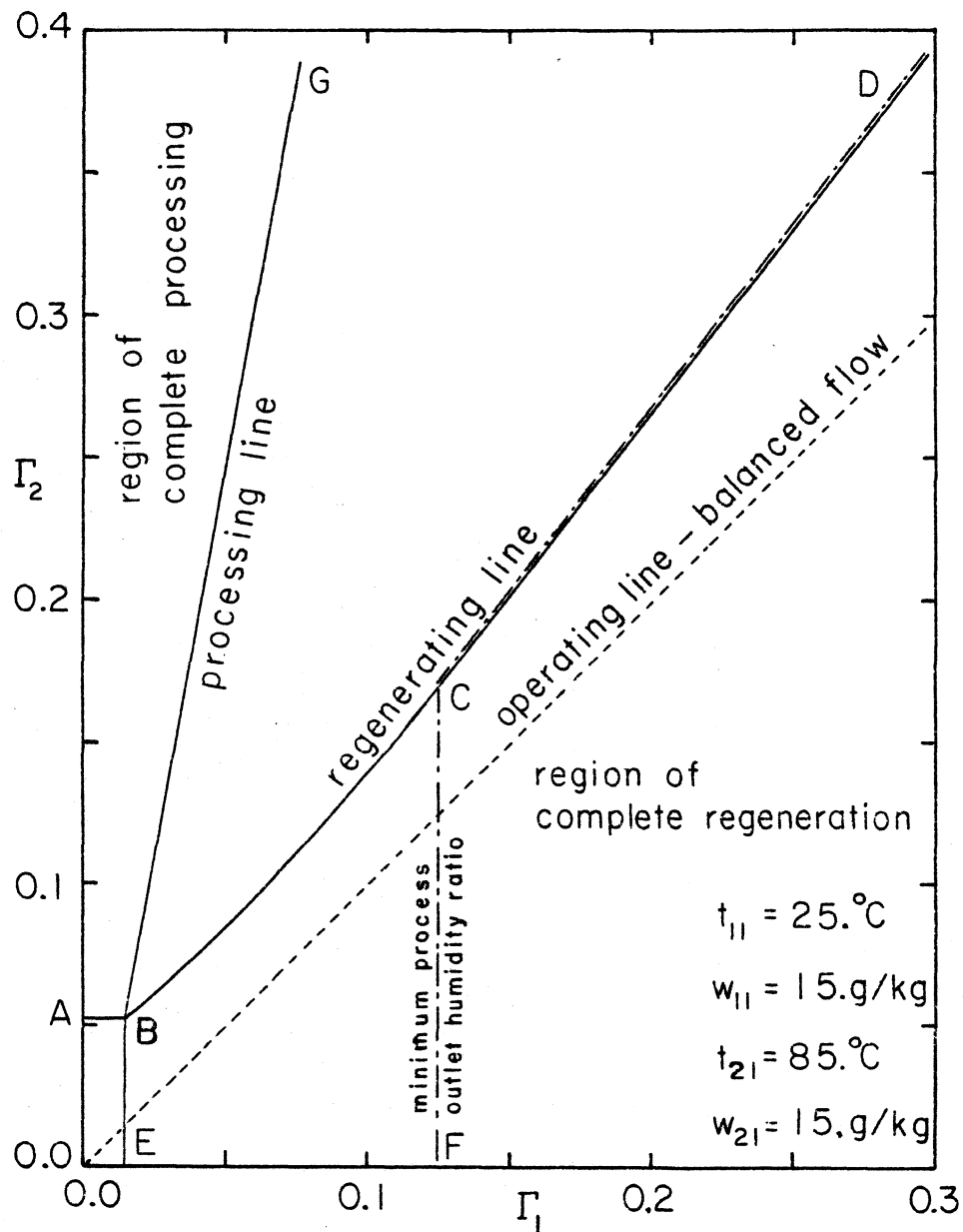


Figure 2.16 Operating chart for the ideal dehumidifier

For any operating point (Γ_1, Γ_2) to the right of the regenerating line,

$$\Gamma_2 \leq \Gamma_{2,R}(\Gamma_1) \quad (2.44)$$

the second shock has completely moved through the matrix during the regenerating period. Thus all the desiccant particles are then in equilibrium with the regenerating air inlet state, before they change period. This line has a positive slope because if more air is being processed for a given rotational speed, then it takes more air to completely regenerate the desiccant matrix.

The solid line E-B-G in Figure 2.16 represents the processing line of the dehumidifier

$$\Gamma_{1,P} = \Gamma_{1,P}(\Gamma_2) \quad (2.45)$$

For any operating point (Γ_1, Γ_2) to the left of the processing line

$$\Gamma_1 \leq \Gamma_{1,P}(\Gamma_2) \quad (2.46)$$

the trailing edge of the F_2 wave has completely moved through the matrix. Thus all the desiccant particles reach equilibrium with the process air inlet state during the processing period, before they change period. Also this line has a positive slope.

The line segment A-B is a straight line with zero slope.

$$\Gamma_{2,A} = \Gamma_{2,B} = S_2 \quad (2.47)$$

where S_2 is the speed of the second shock when regenerating a fully

processed desiccant. The line segment E-B is a straight line with infinite slope. The following equation holds

$$\Gamma_{1,E} = \Gamma_{1,B} = \lambda_{2,11} \quad (2.48)$$

It is shown in section 2.5.2 that the process air mean outlet humidity ratio is minimum for certain values of Γ_1 and Γ_2 . The dashed line F-C-D in Figure 2.16 represents the line of minimum mean outlet humidity ratio of the process air stream. This minimum is constant for the segment F-C but increases slowly along the line C-D as Γ_1 increases. The line F-C is a straight line with infinite slope, because the segment lies in the region of complete regeneration of the desiccant, and hence is not dependent on Γ_2 . It is shown in the next section that the value $\Gamma_{1,F} = \Gamma_{1,C}$ is slightly less than $\lambda_{2,1-int}$. Outside the region of complete regeneration, the line of minimum outlet humidity ratio coincides with the regenerating line of the dehumidifier.

The processing line and the regenerating line are almost straight lines going through the origin. Instead of using Γ_1 and Γ_2 as the describing parameters, Γ_1 and

$$\frac{\Gamma_1}{\Gamma_2} = \frac{\dot{m}_2}{\dot{m}_1} \quad (2.49)$$

may therefore be used.

2.5.2 Performance Correlations

In this section, correlations are presented to calculate the

dehumidifier outlet states as a function of the inlet conditions and the operating parameters Γ_1 and Γ_2 . These performance correlations are valid either for the region of complete regeneration or the region of complete processing. Correlations for the processing line and the regenerating line are provided.

1. The Region of Complete Regeneration

The mean outlet state properties of the process air stream are given by

$$\begin{aligned}\bar{w}_{12} &= \Gamma_1 \int_0^{1/\Gamma_1} w_{12}(\tau) d\tau \\ \bar{i}_{12} &= \Gamma_1 \int_0^{1/\Gamma_1} i_{12}(\tau) d\tau\end{aligned}\tag{2.50}$$

where $w_{12}(\tau)$ and $i_{12}(\tau)$ are given by

$$\begin{aligned}w_{12}(\tau) &= w_m(x=1, \tau) \\ i_{12}(\tau) &= i_m(x=1, \tau)\end{aligned}\tag{2.51}$$

and where $w_m(x, \tau)$ and $i_m(x, \tau)$ are the solutions to (2.11) and the initial value problem (2.28).

Figure 2.17 shows the instantaneous value and the mean value of the humidity ratio of the process air outlet as functions of time. Based on the characteristic diagram in Figure 2.7, the following

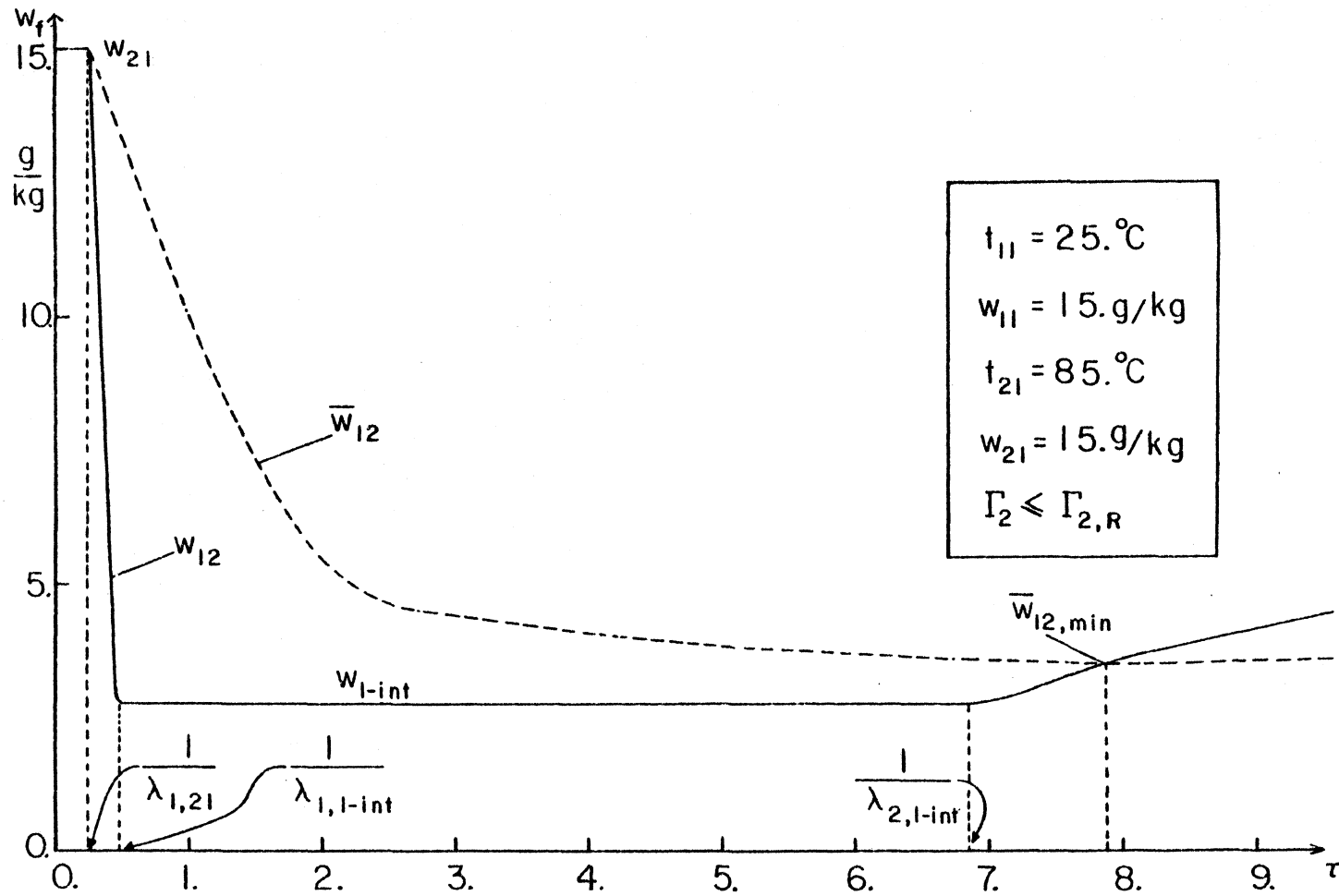


Figure 2.17 Process air outlet humidity ratio as a function of time

expressions may be obtained

$$\text{for } 0 \leq \tau \leq \frac{1}{\lambda_{1,21}}$$

$$w_{12}(\tau) = w_{21}$$

$$i_{12}(\tau) = i_{21}$$

$$\text{for } \frac{1}{\lambda_{1,21}} \leq \tau \leq \frac{1}{\lambda_{1,1-\text{int}}}$$

$$w_{12}(\tau) = w_m(\lambda_1 = 1/\tau, F_2 = F_{2,12})$$

$$i_{12}(\tau) = i_m(\lambda_1 = 1/\tau, F_2 = F_{2,21})$$

$$\text{for } \frac{1}{\lambda_{1,1-\text{int}}} \leq \tau \leq \frac{1}{\lambda_{2,1-\text{int}}} \quad (2.51)$$

$$w_{12}(\tau) = w_{1-\text{int}}$$

$$i_{12}(\tau) = i_{1-\text{int}}$$

$$\text{for } \frac{1}{\lambda_{2,1-\text{int}}} \leq \tau \leq \frac{1}{\lambda_{2,11}}$$

$$w_{12}(\tau) = w_m(\lambda_2 = 1/\tau, F_1 = F_{1,11})$$

$$i_{12}(\tau) = i_m(\lambda_2 = 1/\tau, F_1 = F_{1,11})$$

$$\text{for } \frac{1}{\lambda_{2,11}} \leq \tau$$

$$w_{12}(\tau) = w_{11}$$

$$i_{12}(\tau) = i_{11}$$

The integration (2.50) may be performed using the alternate expressions provided in (2.51). E.g., for the case

$$\lambda_{2,1-\text{int}} \leq \Gamma_1 \leq \lambda_{1,1-\text{int}} \quad (2.52)$$

it follows that

$$\bar{w}_{12} = w_{1-\text{int}} + \Gamma_1 \left[\frac{w_{21}}{\lambda_{1,21}} + \int_{\frac{1}{\lambda_{1,21}}}^{\frac{1}{\lambda_{1,1-\text{int}}}} w_m d\tau - \frac{w_{1-\text{int}}}{\lambda_{1,1-\text{int}}} \right] \quad (2.53)$$

$$\bar{i}_{12} = i_{1-\text{int}} + \Gamma_1 \left[\frac{i_{21}}{\lambda_{1,21}} + \int_{\frac{1}{\lambda_{1,21}}}^{\frac{1}{\lambda_{1,1-\text{int}}}} i_m d\tau - \frac{i_{1-\text{int}}}{\lambda_{1,1-\text{int}}} \right]$$

These relationships show that, for the range (2.52), $\bar{w}_{12}$ and $\bar{i}_{12}$ are linear functions of Γ_1 . The integrals appearing in (2.53) may be calculated by Simpson's 3-point integration scheme.

The value of Γ_1 for which $\bar{w}_{12}$ is minimum is obtained when $\bar{w}_{12}(\tau)$ intersects with $w_{12}(\tau)$ in Figure 2.17. This minimum is not well defined. Therefore, a matrix operating at certain inlet conditions does not have to be tuned at a well defined rotational speed.

Correlations (2.51) and (2.53) are valid only for the region of complete regeneration

$$\Gamma_2 \leq \Gamma_{2,R}(\Gamma_1) \quad (2.54)$$

2. The Regenerating Line

The following assumptions are made to obtain the expression for the regenerating line:

1. The second shock wave is fully developed during the regeneration period. The continuous F_2 wave is completely converted into a strong shock.
2. The leading edge of the F_2 rarefaction wave has not reached the regenerating inlet face of the matrix during the processing period.

Assumption 1 has been verified by solving (2.11) and the initial value problem (2.42) by a numerical scheme, analogous to Glimm's sampling procedure [33] that has been developed to solve shock wave problems. (2.11) and (2.42) were solved for a number of operating parameters and inlet conditions. Assumption 2 may be expressed as

$$\Gamma_1 \geq \lambda_{2,1-int} \quad (2.54)$$

This assumption is closely met for most operating points for a well defined dehumidifier.

Assumption 1 implies that the shock state driven by the second shock may be approximated by the second intersection point (2.4.2-2 and Eq. (2.40)). In this case, the outlet humidity ratio of the regenerating air stream is given by

$$\bar{w}_{22} = \Gamma_{2,R} \left[\frac{w_{11}}{\bar{s}_1} + w_{2-int} \left(\frac{1}{\Gamma_{2,R}} - \frac{1}{\bar{s}_1} \right) \right] \quad (2.55)$$

which is analogous to expression (2.53). $\bar{s}_1$ denotes the average speed of the first shock when regenerating a partially processed matrix (2.4.2. - 2).

Based on the expressions for the operating parameters Γ_j , (2.4), it may be shown that an overall mass balance on the dehumidifier yields

$$\frac{w_{11} - \bar{w}_{12}}{\Gamma_1} = \frac{\bar{w}_{22} - w_{21}}{\Gamma_{2,R}} \quad (2.56)$$

If (2.56) is substituted in (2.55), the following expression may be obtained for $\Gamma_{2,R}$

$$\Gamma_{2,R} = \Gamma_1 \frac{(w_{2-int} - w_{21})}{(w_{11} - \bar{w}_{12}) + \frac{\Gamma_1}{\bar{s}_1} (w_{2-int} - w_{11})} \quad (2.57)$$

The mean humidity ratio of the process air outlet state may be expressed as (2.53)

$$\bar{w}_{12} = w_{1-int} + \frac{\Gamma_1}{\bar{\lambda}_1} (w_{21} - w_{1-int}) \quad (2.58)$$

where $\bar{\lambda}_1$ is the average speed of the F_1 rarefaction wave when processing a fully regenerated matrix (2.4.1 - 1). Substituting (2.58) in (2.57) yields

$$\Gamma_{2,R} = \Gamma_1 \frac{(w_{2-int} - w_{21})}{(w_{11} - w_{1-int}) + \Gamma_1 \left(\frac{w_{2-int} - w_{11}}{\bar{S}_1} - \frac{w_{21} - w_{1-int}}{\bar{\lambda}_1} \right)} \quad (2.59)$$

Accuracy values for $\bar{S}_1$ and $\bar{\lambda}_1$ are not needed. Numerical evaluations of expression (2.59) shows that the second term of the denominator represents a correction term. $\bar{\lambda}_1$ and $\bar{S}_1$ may be calculated by

$$\bar{S}_1 = S_1 \quad (2.60)$$

$$\frac{1}{\bar{\lambda}_1} = \frac{0.5}{\lambda_{1,21}} + \frac{0.5}{\lambda_{1,1-int}}$$

3. The Region of Complete Processing

Equations (2.11) in combination with the initial value problem (2.41) may be solved by the shock wave method. The solution is shown in the characteristic diagram, Figure 2.12. Depending on the value of Γ_2 , two sets of equations may be obtained from Figure 2.12.

$$a. \quad \Gamma_2 > S_2$$

The following equations may be obtained which give the mean state properties for the regenerating air outlet stream

$$\begin{aligned}\bar{w}_{22} &= \Gamma_2 \left[\frac{w_{11}}{S_1} + w_{2-s} \left(\frac{1}{\Gamma_2} - \frac{1}{S_1} \right) \right] \\ \bar{i}_{22} &= \Gamma_2 \left[\frac{i_{11}}{S_1} + i_{2-s} \left(\frac{1}{\Gamma_2} - \frac{1}{S_1} \right) \right]\end{aligned}\quad (2.61)$$

Using the mass balance (2.56) and an analogous energy balance, the following expressions are obtained for the process air outlet state

$$\begin{aligned}\bar{w}_{12} &= w_{11} - \Gamma_1 \left(\frac{w_{2-s} - w_{21}}{\Gamma_2} - \frac{w_{2-s} - w_{11}}{S_1} \right) \\ \bar{i}_{12} &= i_{11} - \Gamma_1 \left(\frac{i_{2-s} - i_{21}}{\Gamma_2} - \frac{i_{2-s} - i_{11}}{S_1} \right)\end{aligned}\quad (2.62)$$

These expressions are analogous to expressions (2.53)

b. limiting case for $\Gamma_2 < S_2$

Figure 2.12 provides the solution to (2.11) and (2.41). The mean outlet state properties of the process air stream are given by

$$\begin{aligned}\bar{w}_{12} &= w_{11} - \Gamma_1 \left(\frac{w_{2-s} - w_{21}}{S_2} - \frac{w_{2-s} - w_{11}}{S_1} \right) \\ \bar{i}_{12} &= i_{11} - \Gamma_1 \left(\frac{i_{2-s} - i_{21}}{S_2} - \frac{i_{2-s} - i_{11}}{S_1} \right)\end{aligned}\quad (2.63)$$

Equations (2.62) and (2.63) are valid only in the region of complete processing, (2.46).

4. The processing line

The correlation for the processing line, (2.45), may be obtained by analyzing the initial value problem (2.11), (2.32). From the characteristic diagram, shown in Figure 2.9, the following correlation may be obtained

$$\Gamma_{1,p} = \frac{\lambda_{2,11}}{1 - \left(1 - \frac{s_2}{\Gamma_2}\right) \left(\frac{\bar{\lambda}_1 - \lambda_{2,11}}{\bar{\lambda}_1 - \lambda_{2,2-s}}\right)} \quad (2.64)$$

where $\bar{\lambda}_1$ is given by

$$\bar{\lambda}_1 = \frac{\lambda_{1,11} + \lambda_{1,2-s}}{2} \quad (2.65)$$

At $\Gamma_1 = \Gamma_{1,R}$, the trailing edge of the F_2 -rarefaction wave just reaches the exit face of the regenerator wheel during the processing period.

5. Summary

A survey of the various correlations that are presented in the previous paragraphs may be shown in graphical form. Figure 2.18 shows the mean humidity ratio of the process air outlet state, $\bar{w}_{12}$, as a function of Γ_1 . Figure 2.19 shows the mean enthalpy of that outlet state, $\bar{i}_{12}$.

Equations (2.50), (2.51) are shown in Figures 2.18 and 2.19. The expressions for the limiting cases, respectively, equation (2.53) to the right and equation (2.63) to the left are also illus-

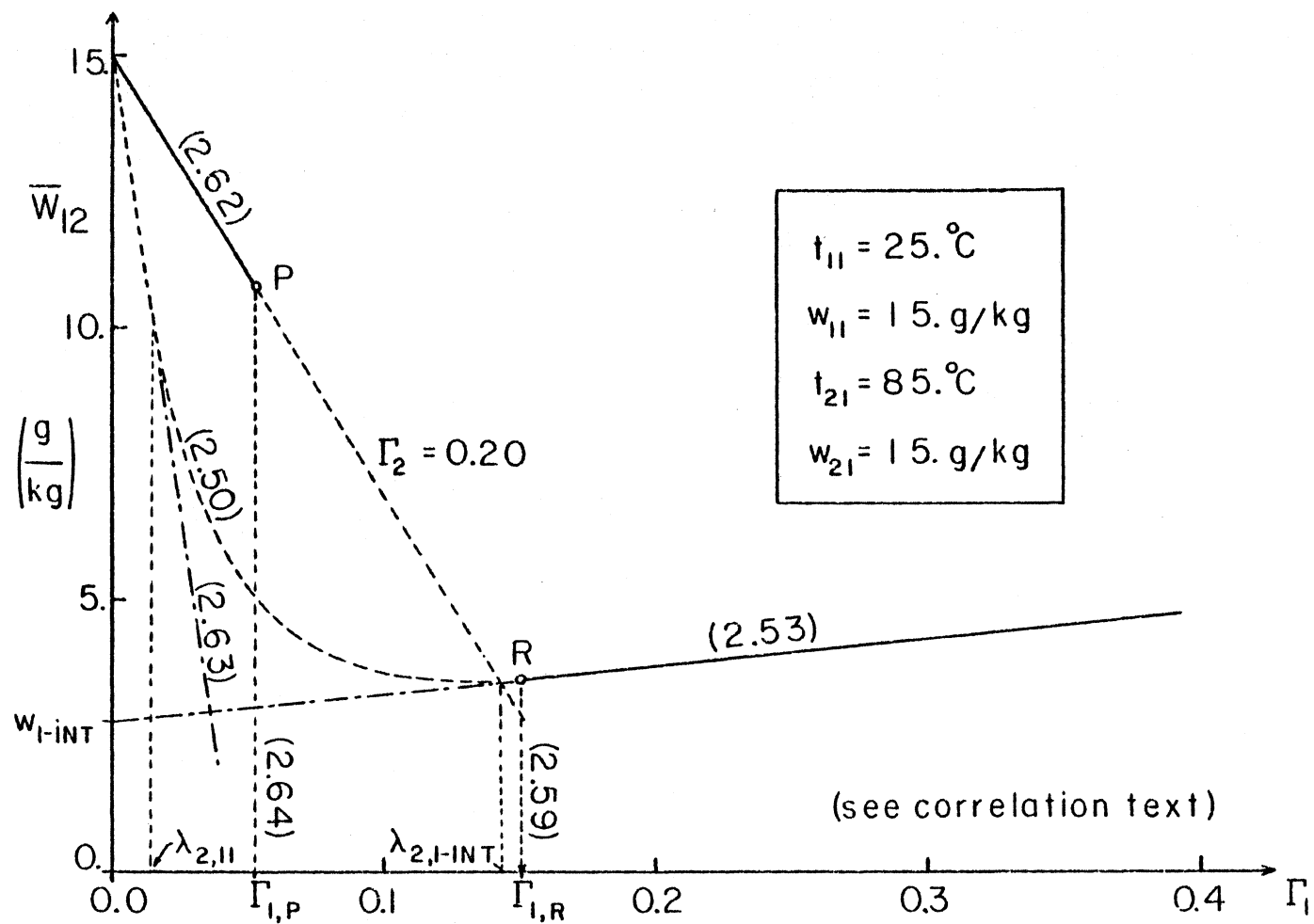


Figure 2.18 Graphical presentation of process air outlet humidity ratio correlation

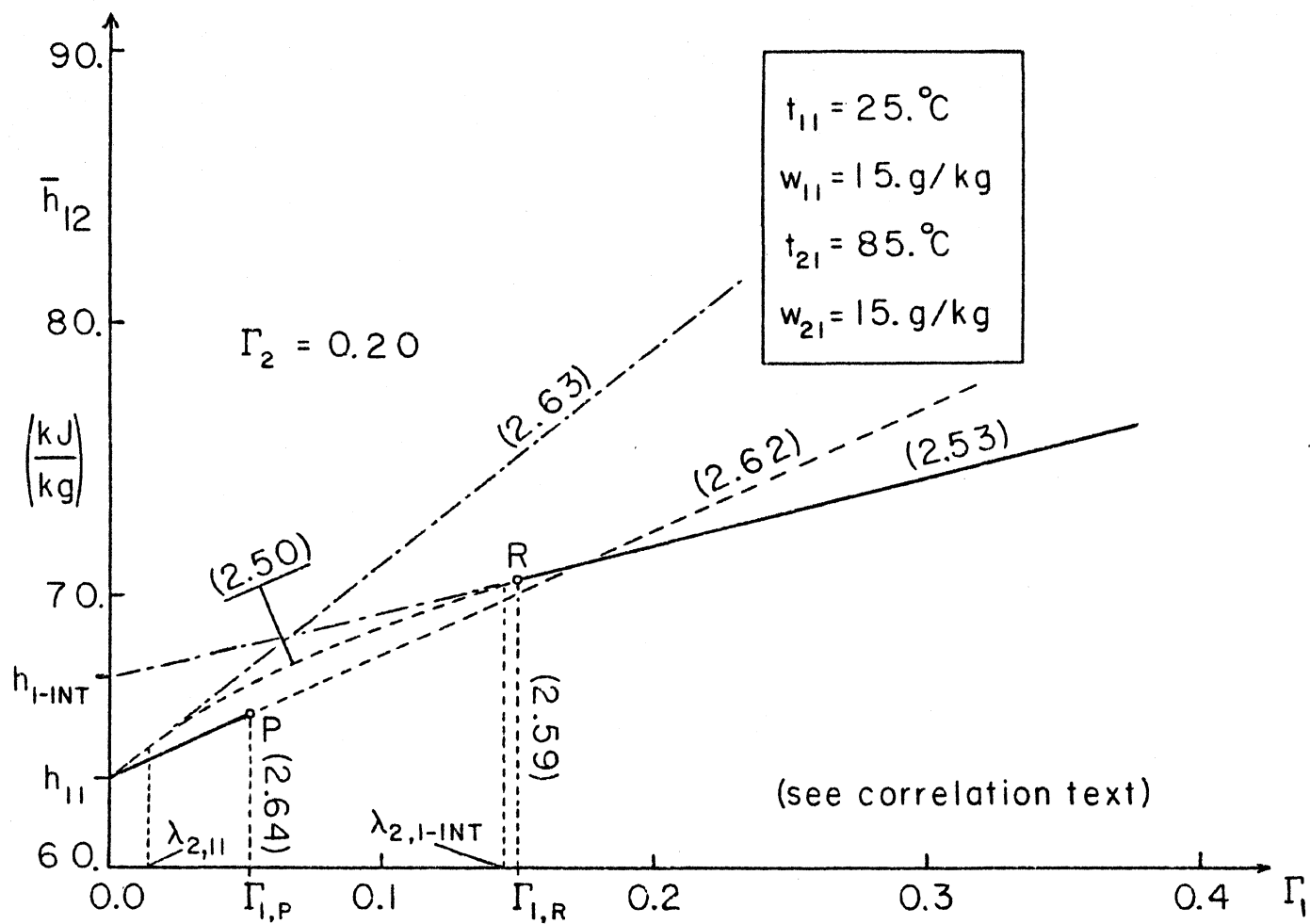


Figure 2.19 Graphical presentation of process air outlet enthalpy correlations

trated. These equations are valid in the region of complete regeneration of the matrix, (2.44), and independent of Γ_2 .

For a given value of Γ_2 , $\Gamma_{1,R}$ may be obtained from equation (2.59) and $\Gamma_{1,P}$ may be obtained from equation (2.64). Equation (2.50) holds for Γ_1 greater than $\Gamma_{1,R}$. Point R in Figure 2.18 and 2.19 symbolizes this limit. Correlation (2.62) is also shown in these Figures. This correlation is valid for Γ_1 less than $\Gamma_{1,P}$. This limit is marked by point P in the Figures.

For intermediate values of Γ_1 , no correlation is available because of the complex interaction between the two wave phenomena in both periods. As follows from Figures 2.18 and 2.19, negligible errors are introduced if points P and R are connected by a parabola. $\bar{w}_{12}(\Gamma_1)$ and $\bar{i}_{12}(\Gamma_1)$ are continuous curves with a continuous first derivative at point P and a discontinuous first derivative at point R, because of the shock waves during the regenerating period.

2.5.3 Comparison with Existing Solution Methods

To predict the performance of a dehumidifier with finite transfer coefficients, a general computer code, MOSHMX [13], has been developed that solves equations (2.1) with the initial conditions (2.9) and the boundary conditions (2.10). The program is based on a second order finite difference method with extrapolation to zero grid size. This computer program was used to compare the proposed correlations (2.50-2.65) with performance values provided by that program.

The program makes use of the following dimensionless transfer coefficients

$$NTU_{t,j} = \frac{h_j A_j}{\dot{m}_j \left(\frac{\partial i_f}{\partial t_f} \right)_{w_f}} \quad j = 1, 2$$

$$NTU_{w,j} = \frac{h_{w,j} A_j}{\dot{m}_j} \quad j = 1, 2 \quad (2.66)$$

$$Le_j = \frac{NTU_{t,j}}{NTU_{w,j}}$$

and assumes for the Lewis numbers Le_j

$$Le_1 = Le_2 = Le \quad (2.67)$$

In the limit

$$NTU_{t,j} \rightarrow \infty$$

$$NTU_{w,j} \rightarrow \infty \quad (2.68)$$

the real dehumidifier performance approaches that of an ideal dehumidifier. In practice, as is shown in Chapter 3, $NTU = 20$ is sufficiently high to obtain approximately ideal dehumidifier performance.

Figure 2.20 shows $\bar{w}_{12}$ as a function of Γ_1 for a fixed value of Γ_2 and various values of $NTU_{t,j}$. The convergence to the proposed

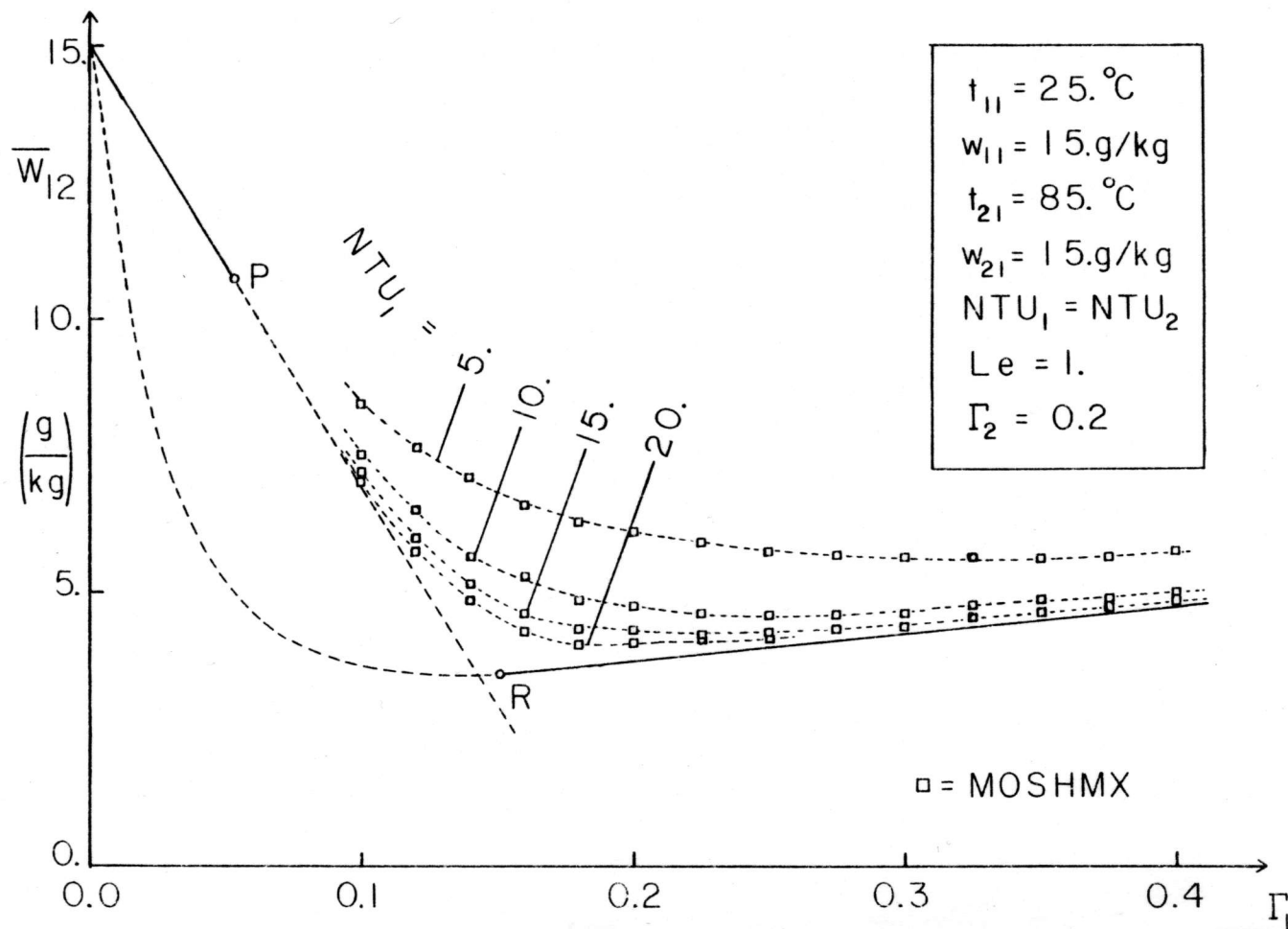


Figure 2.20 Comparison of humidity ratio correlations with MOSHMX and NTU as parameter

correlations, as summarized in Figure 2.18, is clearly shown.

Figure 2.21 shows $\bar{w}_{12}(\Gamma_1)$ for different values of Γ_2 and fixed NTU, and again convergence is shown.

2.5.4 Performance Charts

The correlations that are presented in the previous sections may be implemented in a computer program. Based on such program, several performance charts can be generated showing the effect of the inlet conditions and operating parameters on the outlet state of the process air stream. Appendix A presents such charts in which the mean values of the process air outlet humidity ratio and temperature are shown as functions of Γ_1 .

Figure A.3 shows the effect of Γ_1 and Γ_2 on the outlet humidity ratio and temperature of the process air stream. For a given pair of inlet states, the rotary dehumidifier has a maximal drying capacity independent of Γ_1 and Γ_2 . If Γ_2 is lower than a certain critical value which is dependent on the inlet conditions, then this maximum capacity may be achieved. The corresponding value of Γ_1 , at which the outlet humidity ratio is minimal, is independent of Γ_2 . If Γ_2 is higher than the critical value, the minimal obtainable outlet moisture content increases and becomes a function of Γ_2 . This minimum may be obtained only for one particular value of Γ_1 , which is dependent on Γ_2 .

Figure A.1 shows the effect of the process air inlet state on the outlet state of that air stream. If the process inlet tempera-

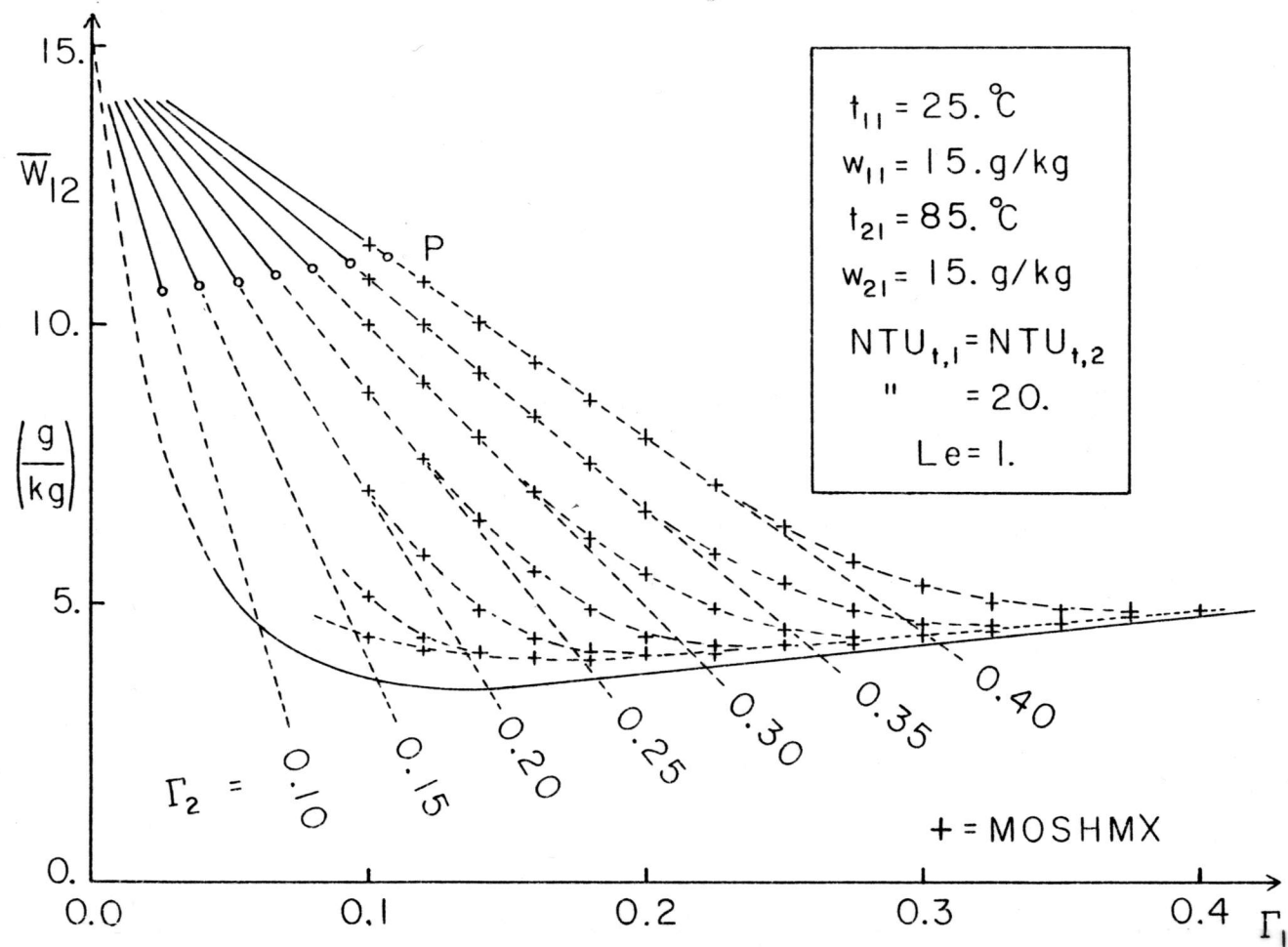


Figure 2.21 Comparison of humidity ratio correlations with MOSHMX and Γ_2 as parameter

ture is increased, the maximal drying capacity decreases and the corresponding optimal value of Γ_1 shifts to a slightly higher value. The optimal ratio Γ_1/Γ_2 increases slightly. It follows from Figure A.1 that it is safe practice to operate the dehumidifier at a value of Γ_1 which is slightly higher than the optimal value. If the process inlet humidity ratio is increased, the minimal obtainable outlet moisture content increases substantially and the optimal ratio of Γ_1/Γ_2 changes to significantly higher values.

Figure A.2 shows the effect of the regenerating air inlet state on the process air outlet state properties. If the regenerating air inlet temperature is increased, the optimal value of Γ_1 and the minimum obtainable outlet humidity ratio decreases. The performance of the ideal rotary dehumidifier is only slightly influenced by the regenerating air inlet humidity ratio, as follows from Figure A.2.

CHAPTER 3

THE ϵ -NTU METHOD FOR ROTARY DEHUMIDIFIERS

In the previous chapter a method has been introduced which allows rapid calculation of the performance of an ideal dehumidifier. In this chapter, the correlations that are provided by that method are used in combination with effectiveness correlations for the humidity ratio and the temperature, to calculate the outlet states of a dehumidifier with finite transfer coefficients.

The idea of using an effectiveness technique to compute the outlet states of a heat and mass exchanger has been introduced by Banks et al. [13,18,19]. Based on this concept, two methods have been proposed: the intersection point method [13,18] and the non-linear analogy method. Both methods are extensively discussed by Jurinak [3]. The analogy theory provides the basis for both methods [13]. In this theory, the characteristic expressions (2.21) for the ideal dehumidifier are made analogous to the characteristic expressions for the ideal thermal regenerator, by uncoupling equations (2.21) through averaging the wave speeds λ_1 by integration. Using these averaged wave speeds, the various Number of Transfer Units (2.66) are then transformed into analogous dimensionless transfer coefficients. The temperature effectiveness of a thermal regenerator with these analogous transfer coefficients will then approximate the effectiveness of the F_1 and F_2 potentials of the corresponding dehumidifier. However, because the correspondence between both re-

generators relies on the characteristic expressions and the wave speeds λ_i are non-linear with respect to the F_i potentials, the transformation is only valid in the region of complete regeneration. Moreover, the analogy theory only holds when the Lewis number (2.66) equals unity [3].

The effectiveness correlations which are introduced in this chapter are valid for silica gel dehumidifiers. They hold for a wide range of inlet conditions, operating parameters, transfer numbers and for $Le \geq 1$.

3.1 Dependent Variables and Dimensionless Groups

Using the dimensionless transfer coefficients defined in (2.66), the conservation and transfer rate equations (2.1) may be expressed as

$$\begin{aligned} \frac{\partial w_f}{\partial x} + \frac{\partial w_m}{\partial \tau} &= 0 \\ \frac{\partial w_f}{\partial x} &= NTU_{w,j} (w_m - w_f) \end{aligned} \tag{3.1}$$

$$\frac{\partial i_f}{\partial x} + \frac{\partial i_m}{\partial \tau} = 0$$

$$\frac{\partial i_f}{\partial x} = NTU_{t,j} \frac{\partial i_f}{\partial t_f} (t_m - t_f) + i_{wv} NTU_{w,j} (w_m - w_f)$$

where x and τ are the dimensionless space and time coordinate, as

defined in (2.5). This set of equations has to be combined with the initial and boundary conditions (2.9), (2.10), and the state property relationships (2.8). These relationships are provided in the next section for the system silica gel-moist air.

The solution to equations (3.1) consists of determining the mean outlet state properties of one of the two fluid streams, e.g.

$$\begin{aligned}\bar{w}_{12} &= \Gamma_1 \int_0^{1/\Gamma_1} w_{f1}(x=1, \tau) d\tau \\ \bar{i}_{12} &= \Gamma_1 \int_0^{1/\Gamma_1} i_{f1}(x=1, \tau) d\tau\end{aligned}\tag{3.2}$$

The outlet state of the other fluid stream may then be obtained by applying an overall mass and energy balance

$$\begin{aligned}\frac{(\bar{w}_{11} - \bar{w}_{12})}{\Gamma_1} &= \frac{(\bar{w}_{22} - w_{21})}{\Gamma_2} \\ \frac{(\bar{i}_{12} - i_{11})}{\Gamma_1} &= \frac{(i_{21} - \bar{i}_{22})}{\Gamma_2}\end{aligned}\tag{3.3}$$

For given desiccant and fluid stream properties (2.8) it follows from equations (3.1) and the initial and boundary condition (2.9), (2.10), that the outlet state of the process air stream is a function of

$$\begin{aligned}\bar{w}_{12} &= \bar{w}_{12} (t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_1, \Gamma_2, NTU_{t,1}, NTU_{t,2}, Le) \\ \bar{i}_{12} &= \bar{i}_{12} (t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_1, \Gamma_2, NTU_{t,1}, NTU_{t,2}, Le)\end{aligned}\tag{3.4}$$

where the Lewis number is defined as the ratio of overall resistance to mass transfer to the overall resistance to heat transfer, as expressed in (2.66). It is assumed that the Lewis number is the same for both periods (2.67). Expressions (3.4) form the basis for the ϵ -NTU method which is introduced in this chapter. First, the thermodynamic property relationships for nominal silica gel-air-water vapor are presented.

3.2 Thermodynamic Property Relationships

The effectiveness correlations that are presented in this text are valid for silica gel as the desiccant and a mixture of water vapor and air as the fluid streams. Table 3.1 lists the various equations used for the water vapor and moist air properties. The total pressure p_t is assumed to be constant at 1 atm. Table 3.2 defines the state properties for nominal silica gel.

3.3 Effectiveness

3.3.1 Definition

Because two state properties of the moist air are required to fully characterize the outlet state, it is necessary to define two

Table 3.1 Thermodynamic Properties of Water Vapor and Moist Air
(from ref [3])

water vapor

enthalpy $i_{wv} = 1800.3 t_f + 2502680.$

heat of vaporization $i_v = 2502680. - 2386.5 t_f$

saturation pressure $\log_{10} \left[\frac{P_{ws}}{218.167} \right] = \frac{-z}{t_f + 273.15} \frac{a + bz + cz^3}{1 + dz}$

where $z = 374.12 - t_f$

$a = 3.2437814$

$b = 5.86826 \times 10^{-3}$

$c = 1.1702379 \times 10^{-8}$

$d = 2.1878462 \times 10^{-3}$

moist air

humidity ratio $w_f = 0.62198 \frac{P_{wv}}{P_t - P_{wv}}$

relative humidity $r = \frac{P_{wv}}{P_{ws}}$

enthalpy $i_f = 1004.8 t_f + w_f (1800.3 t_f + 2502680.)$

where $t_f \equiv [^{\circ}\text{C}]$

$w_f \equiv [\text{kg/kg dry air}]$

$i_f \equiv [\text{J/kg dry air}]$

$p \equiv [\text{atm}]$

Table 3.2 Thermodynamic Properties of Nominal Silica Gel (from ref [3])

equilibrium isotherm

$$r = (2.112 W_m)^{h^*} (29.91 p_{ws})^{h^*-1}$$

heat of adsorption

$$i_s = h^* i_v$$

$$h^* = 1.0 + 0.2843 \exp (-10.28 W_m)$$

enthalpy

$$I_m = (921.096 + 4186.8 W_m) t_m + \int_0^{W_m} (i_v - i_s) dW_m$$

where $W_m \equiv [\text{kg/kg dry desiccant}]$

$I_m \equiv [\text{J/kg dry desiccant}]$

effectiveness expressions, e.g., one for the humidity ratio and one for the enthalpy. Both will be defined for the process air outlet state.

The effectiveness for humidity ratio is defined as

$$\epsilon_w = \frac{w_{11} - \bar{w}_{12}}{w_{11} - (\bar{w}_{12})_{id}} \quad (3.5.a)$$

and for enthalpy

$$\epsilon_h = \frac{\bar{i}_{12} - i_{11}}{(\bar{i}_{12})_{id} - i_{11}} \quad (3.5.b)$$

The subscript "id" means the corresponding outlet state for the ideal dehumidifier

$$(\bar{w}_{12})_{id} = \lim_{\substack{NTU_{w,j} \rightarrow \infty \\ NTU_{t,j} \rightarrow \infty}} \bar{w}_{12} \quad (3.6)$$

$$(\bar{i}_{12})_{id} = \lim_{\substack{NTU_{w,j} \rightarrow \infty \\ NTU_{t,j} \rightarrow \infty}} \bar{i}_{12}$$

In the previous chapter it is shown that the outlet state of an ideal dehumidifier can be obtained analytically without solving equations (3.1) and taking the limit (3.6). Using equations (3.5) and the correlations provided by the ideal dehumidifier model, the

outlet state properties may then be calculated provided that ϵ_w and ϵ_h are readily computable.

From expressions (3.4) it follows that

$$\begin{aligned}\epsilon_w &= \epsilon_w(t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_1, \Gamma_2, NTU_{t,1}, NTU_{t,2}, Le) \\ \epsilon_h &= \epsilon_h(t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_1, \Gamma_2, NTU_{t,1}, NTU_{t,2}, Le)\end{aligned}\quad (3.7)$$

3.3.2 Effectiveness Correlation for Humidity Ratio

The effectiveness ϵ_R of a rotary counterflow heat exchanger may be correlated by [20]

$$\epsilon_R = C_\epsilon \epsilon_{cf} \quad (3.8)$$

where ϵ_{cf} is the expression for the effectiveness of a counterflow heat exchanger

$$\epsilon_{cf} = \frac{1 - \exp(-NTU_0 (1-C^*))}{1 - C^* \exp(-NTU_0 (1-C^*))} \quad (3.9)$$

C^* is the heat capacity rate ratio, which is always less than or equal to 1, and NTU_0 is the overall Number of Transfer Units. C_ϵ is a correction factor less than 1, taking into account the rotary aspect of the regenerator [20].

In a dehumidifier, the humidity ratio of the air streams can only change by mass transfer. Similarly, in a rotary heat exchanger the temperature of the air streams can only change by heat transfer.

Hence, it may be expected by analogy that ε_w may be correlated with an expression analogous to equation (3.8). Humidity ratio effectiveness values obtained from the computer code MOSHMX [13] and the ideal dehumidifier model [chapter 2] are compared to equation (3.9) in Figure 3.1. In this Figure, C^* and NTU_0 are set equal to

$$\Gamma_1 \leq \Gamma_{1R} \quad C^* = \frac{\Gamma_1}{\Gamma_{1R}} \quad (3.10.a)$$

$$\frac{1}{NTU_0} = \frac{C^*}{NTU_{w,1}} + \frac{1}{NTU_{w,2}}$$

$$\Gamma_1 > \Gamma_{1R} \quad C^* = \frac{\Gamma_{1R}}{\Gamma_1} \quad (3.10.b)$$

$$\frac{1}{NTU_0} = \frac{1}{NTU_{w,1}} + \frac{C^*}{NTU_{w,2}}$$

where Γ_{1R} is given by the regenerating line correlation (2.59).

Figure 3.1 shows that there is an analogy between the ratio Γ_1/Γ_{1R} for a heat and mass exchanger and the capacity rate ratio for a counterflow heat exchanger. The humidity ratio effectiveness is minimized at $\Gamma_1 = \Gamma_{1R}$. The performance charts of the ideal dehumidifier (Appendix A) show that the derivative of $(\bar{w}_{12})_{id}$ is discontinuous with respect to Γ_1 for $\Gamma_1 = \Gamma_{1R}$. Finite transfer coefficients tend to smooth abrupt changes of the outlet state properties. Hence, the derivation of $\bar{w}_{12}$ from $(\bar{w}_{12})_{id}$ will be largest for $\Gamma_1 = \Gamma_{1R}$.

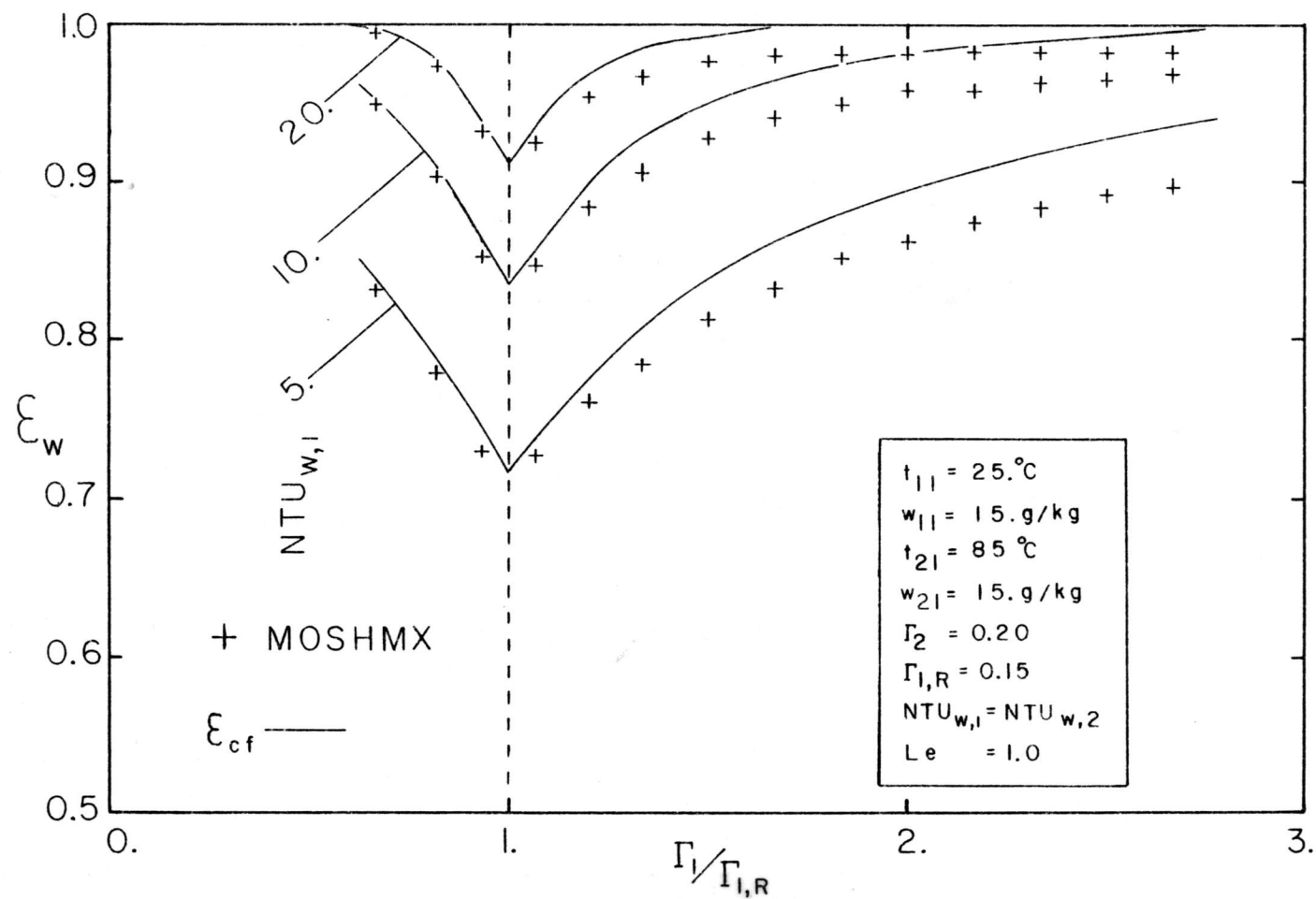


Figure 3.1 Comparison of dehumidifier effectiveness values with the counterflow heat exchanger effectiveness correlation

Figure 3.1 shows further, that for $\Gamma_1 < \Gamma_{1R}$ the agreement between (3.9) and the computed points is within a few percent, but for $\Gamma_1 > \Gamma_{1R}$ correlation (3.9) overpredicts the effectiveness. In a recuperator there is steady state direct heat transfer from one fluid stream to the other by conduction through a wall. In a regenerator, heat and mass are first transferred from the fluid stream to the matrix in a transient type of saturation process, followed by the desorption of the matrix where heat and mass are transferred in the reverse direction. Apparently, this transient switching phenomena causes the effectiveness of that transfer process to drop, compared to the direct transfer type process.

Similar to the counterflow heat regenerator, the deviation of ϵ_w from correlation (3.9) and (3.10) is absorbed in the correction factor C_ϵ . The final moisture effectiveness is given in Table 3.3. For the case in which $\Gamma_1 < \Gamma_{1R}$, a constant value for C_ϵ is an adequate choice as follows from Figure 3.1. On the other hand, if the dehumidifier is operated far enough into the region of complete regeneration ($\Gamma_1 > \Gamma_{1R}$), eventually ϵ_w will become independent of Γ_2 and $NTU_{w,2}$. The adsorption process during the processing period will then resemble the transient response of a porous matrix, initially at uniform state. This is analogous to the insulated duct problem for heat transfer alone [32]. In order to obtain an analogy with the insulated duct problem, a dimensionless time has to be introduced depending on the speed at which the inlet state propagates through the matrix. In the region of complete regeneration, only

Table 3.3 Correlation for process air humidity ratio effectiveness

$$\epsilon_w = C_\epsilon \epsilon_{cf}$$

1. ϵ_{cf} is given by

$$NTU_{w,j} = NTU_{t,j} / Le$$

$$F_{Le} = 1. + 0.498 (\ln(Le))^{\frac{1}{1.546}}$$

$$NTU_{w,j}^* = NTU_{w,j} F_{Le}$$

$$\Gamma_{1R} = \Gamma_{1R} (t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_2) \quad \text{eq (2.59)}$$

for $\Gamma_1 \leq \Gamma_{1R}$:

$$C^* = \Gamma_1 / \Gamma_{1R}$$

$$\frac{1}{NTU_{w,o}} = \frac{C^*}{NTU_{w,1}^*} + \frac{1}{NTU_{w,2}^*}$$

for $\Gamma_1 > \Gamma_{1R}$

$$C^* = \Gamma_{1R} / \Gamma_1$$

$$\frac{1}{NTU_{w,o}} = \frac{1}{NTU_{w,1}^*} + \frac{C^*}{NTU_{w,2}^*}$$

$$\underline{C^* < 1} \quad \epsilon_{cf} = \frac{(1 - \exp(-NTU_{w,o} (1-C^*)))}{(1 - C^* \exp(-NTU_{w,o} (1-C^*)))}$$

$$\underline{C^* = 1} \quad \epsilon_{cf} = \frac{NTU_{w,o}}{1 + NTU_{w,o}}$$

Table 3.3 (continued)

2. C_ε is given by

$$\underline{\Gamma_1 \leq \Gamma_{1R}} \quad C_\varepsilon = 0.99$$

$$\underline{\Gamma_1 > \Gamma_{1R}} \quad C_\varepsilon = C + (0.99 - C) \exp \left[- \frac{\left(\frac{1}{C^*} - 1 \right)}{b} \right]$$

$$C = a \exp \left(- \left(\frac{\bar{\lambda}_1}{\frac{\Gamma_1}{80.}} - 1 \right)^2 \right)$$

$$a = 1. - \exp \left(- \frac{NTU_{w,1}^*}{1.52} \right)$$

$$b = 4.6 \ln \left(\frac{5.88}{NTU_{w,1}^* 0.48} \right)$$

$$\bar{\lambda}_1 = \frac{1}{2} \left(\lambda_{1,21} + \lambda_{1,1-int} \right)$$

the first wave is important for a heat and mass exchanger, hence the corresponding dimensionless time is given by $\bar{\lambda}_1/\Gamma_1$. This ratio is used in the expression for the correction factor C_ϵ for $\Gamma_1 > \Gamma_{1R}$. However, the analogy with the insulated duct problem only holds if Γ_1 is larger than Γ_{1R} to an extent depending on the transfer parameter $NTU_{w,1}$. In the effectiveness correlation it is therefore assumed that the correction factor C_ϵ changes smoothly with Γ_1 , from the constant value valid outside the region of complete regeneration to the limiting value provided by the insulated duct analogy.

To incorporate the influence of the Lewis number into correlation (3.8), an effective Number of Transfer Units $NTU_{w,j}^*$ is defined for each period, based on an empirical correction factor F_{Le} that is a function of the Lewis number. Numerical constants that appear in the expressions in Table 3.3 reflect the empirical character of the correlation.

The accuracy of this correlation has been estimated by comparing effectiveness values obtained from that correlation with effectiveness values provided by MOSHMX and the ideal dehumidifier theory. Based on 62 points covering the range of parameters listed in Table 3.5, the mean value of the error (i.e., the bias) is 0.2% and the standard deviation is 1.3%.

3.3.3 Effectiveness correlation for enthalpy and temperature

In a dehumidifier, enthalpy is exchanged by both heat and mass transfer while moisture is exchanged only by the latter. As a re-

sult, correlating the enthalpy effectiveness, ϵ_h , is far more challenging than correlating the humidity ratio effectiveness, ϵ_w . However, the ratio of the change of enthalpy of the process air stream from inlet to exit to the enthalpy difference of both air streams is typically about 1/8 for well defined dehumidifiers. As is illustrated by the following error analysis, an accurate expression for ϵ_h is therefore not needed.

The temperature of the process air outlet state is given by

$$\bar{t}_{12} = \frac{(\bar{i}_{12} - 2502680 \cdot \bar{w}_{12})}{(1004.8 + 1800.3 \bar{w}_{12})} \quad (3.11)$$

The error in the temperature, $\delta \bar{t}_{12}$, due to an error in the enthalpy, $\delta \bar{i}_{12}$, is given by

$$\delta \bar{t}_{12} = \frac{\delta \bar{i}_{12}}{(1004.8 + 1800.3 \bar{w}_{12})} \quad (3.12)$$

This equation may also be written as

$$\delta \bar{t}_{12} = \frac{((\bar{i}_{12})_{id} - i_{11})}{(1004.8 + 1800.3 \bar{w}_{12})} \cdot \frac{\delta \bar{i}_{12}}{((\bar{i}_{12})_{id} - i_{11})} \quad (3.13)$$

If the following typical numerical values are inserted

$$\begin{aligned} (\bar{i}_{12})_{id} - i_{11} &= 10,000 \text{ J/kg} \\ \bar{w}_{12} &= 0.005 \text{ kg/kg} \end{aligned} \quad (3.14)$$

and the last term of equation (3.13) is replaced by

$$\frac{\delta \bar{i}_{12}}{((\bar{i}_{12})_{id} - i_{11})} = \delta \epsilon_h \quad (3.15)$$

where $\delta \epsilon_h$ is the absolute error in the enthalpy effectiveness, then equation (3.13) may finally be written as

$$\delta \bar{t}_{12} \approx 10 \delta \epsilon_h \quad (3.16)$$

Enthalpy effectiveness values obtained from MOSHMX and the ideal dehumidifier model show that, for $Le=1$ and $NTU_{t,j} \geq 10$, the effectiveness differs from 100% by not more than 4%. Hence if the enthalpy effectiveness is assumed to be constant and equal to unity, then it follows from equation (3.16) that the temperature error is within 0.4°C .

If the Lewis number is higher than unit, the enthalpy effectiveness may be substantially higher than 100%. Indeed, for increasing resistance to mass transfer, the dehumidifier performance approaches that of a thermal regenerator. It is shown [13,20] that, for heat transfer alone, the enthalpy change of the process air stream is typically about 50% higher than for both heat and mass transfer. The parameter of importance to include the Lewis number effect therefore has to be

$$\epsilon_h = \epsilon_h \left(\frac{(\Delta \bar{i}_t)_{id}}{(\Delta \bar{i}_{t,w})_{id}} \right) \quad (3.17)$$

$(\Delta \bar{i}_t)_{id}$ is the enthalpy change of the process air stream from inlet to exit for heat transfer alone while $(\Delta \bar{i}_{t,w})_{id}$ is the enthalpy change for both heat and mass transfer. Infinite transfer coefficients may be assumed in both cases, thus $(\Delta \bar{i}_t)_{id}$ may be calculated using the ideal regenerator theory [13,20], and $(\Delta \bar{i}_{t,w})_{id}$ is calculated using the ideal dehumidifier theory.

The final correlation for the enthalpy effectiveness is given in Table 3.4. The numerical constants that appear in the equations reflect the empirical character of the proposed correlation. The same accuracy analysis as conducted for the humidity ratio shows that the mean error for this effectiveness is 0.6% and the standard deviation is 3.5%. The temperature effectiveness may now be calculated based on the correlations for ϵ_w and ϵ_h . The mean error for this outlet temperature effectiveness, based on 62 points, is 0.3% and the standard deviation is 1.5%. These values are again valid for the range of parameters as given in Table 3.5.

3.4 Effectiveness Charts

Based on the correlations presented in Tables 3.3 and 3.4, the effectiveness may be plotted against the overall Number of Transfer Units with the operating parameters Γ_j and the inlet conditions as parameters. Figures B.1 to B.4 in the Appendix show such effectiveness charts for a rotary silica gel dehumidifier. The + marks are effectiveness values derived from MOSHMX and the ideal dehumidifier model. The solid curves are based on the effectiveness correlations,

Table 3.4 Correlation for process air enthalpy effectiveness

$$\varepsilon_h = 1. + a b \left(\frac{(\Delta i_t)_{id}}{(\Delta i_{t,w})_{id}} - 1. \right)$$

$$\text{where } (\Delta i_{t,w})_{id} = (\bar{i}_{12})_{id} - i_{11}$$

$$(\Delta i_t)_{id} = \Gamma_1 c_m (t_{21} - t_{11})$$

$$c_m = 921. + 4187 \bar{W}_m$$

$$\bar{W}_m = W_m (\bar{t}_f, \bar{w}_f) \text{ where } \bar{t}_f = \frac{t_{11} + t_{21}}{2}$$

$$\bar{w}_f = \frac{w_{11} + w_{21}}{2}$$

a and b are given by

$$\text{for } \underline{\Gamma_1 \leq \Gamma_{1R}} \quad C^* = \Gamma_1 / \Gamma_{1R}$$

$$\frac{1}{NTU_{t,o}} = \frac{C^*}{NTU_{t,1}} + \frac{1}{NTU_{t,2}}$$

$$\text{for } \Gamma_1 > \Gamma_{1R} \quad C^* = \Gamma_{1R} / \Gamma_1$$

$$\frac{1}{NTU_{t,o}} = \frac{1}{NTU_{t,1}} + \frac{C^*}{NTU_{t,2}}$$

$$a = \exp \left[-0.5 \left(\frac{NTU_{t,o}}{10.} - 1. \right) \right]$$

$$b = 1. - \exp \left(- \frac{(Le - 1)}{25.} \right)$$

Table 3.5 Investigated range for effectiveness correlations

t_{11}	:	20.	. . .	35.	°C
w_{11}	:	0.010	. . .	0.020	kg/kg
t_{21}	:	70	. . .	100.	°C
w_{21}	:	0.010	. . .	0.020	kg/kg
Γ_1	:	0.10	. . .	0.40	
Γ_2	:	0.10	. . .	0.40	
$NTU_{t,1}$	:	10.	. . .	40.	
$\frac{NTU_{t,2}}{NTU_{t,1}}$	:	0.5	. . .	2.0	
Le	:	1	. . .	20.	

as presented in this chapter. The NTU_t parameter is defined as

$$\frac{1}{NTU_t} = \frac{1}{2} \left(\frac{1}{NTU_{t,1}} + \frac{1}{NTU_{t,2}} \right) \quad (3.18)$$

Figures B.1-B.2 show the influence of the inlet variables on the effectiveness. For all inlet variables considered, the operating point (Γ_1, Γ_2) is located inside the region of complete regeneration. The closer the operating point moves towards the regenerating line, the lower the effectiveness becomes. The change of effectiveness with inlet variables may therefore be explained by the following expressions

$$\begin{aligned} \Gamma_{1R} &= \Gamma_{1R}(t_{11}, w_{11}, t_{21}, w_{21}, \Gamma_2) \quad (\text{see eq. (2.59)}) \\ \left. \frac{\partial \Gamma_{1R}}{\partial \Gamma_2} \right)_{t_{11}, w_{11}, t_{21}, w_{21}} &> 0 \\ \left. \frac{\partial \Gamma_{1R}}{\partial t_{11}} \right)_{\Gamma_2, w_{11}, t_{21}, w_{21}} &> 0 \\ \left. \frac{\partial \Gamma_{1R}}{\partial w_{11}} \right)_{t_{11}, \Gamma_2, t_{21}, w_{21}} &> 0 \\ \left. \frac{\partial \Gamma_{1R}}{\partial t_{21}} \right)_{t_{11}, w_{11}, \Gamma_2, w_{21}} &< 0 \\ \left. \frac{\partial \Gamma_{1R}}{\partial w_{21}} \right)_{t_{11}, w_{11}, t_{21}, \Gamma_2} &< 0 \end{aligned} \quad (3.19)$$

Figure B.3 shows the change of effectiveness when the operating point slides along the line of balanced operation. Because the regenerating line is almost a straight line going through the origin, the ratio Γ_1/Γ_{1R} remains approximately constant. Thus, the effectiveness change is negligible for this case. Figure B.3 also shows the influence of Γ_2 for fixed Γ_1 . This influence may again be explained by equation (3.19).

The ratio of the NTU parameters in both periods has only a slight effect on the effectiveness as shown in Figure B.4. This has also been reported for the case of a thermal regenerator [20]. Figure B.4 also shows the influence of the Lewis number on the effectiveness. The influence on the humidity ratio effectiveness is more substantial than the influence on the temperature effectiveness. While for $Le < 5$, moisture effectiveness values of 90% may still be achieved, for $Le > 10$, this effectiveness drops substantially.

CHAPTER 4
DESIGN PARAMETER OPTIMIZATION FOR DEHUMIDIFIERS
IN DESICCANT COOLING SYSTEMS

Numerous publications present computer simulation results of desiccant cooling and dehumidification systems [8,16,21,22,23,24]. However, design values for the dehumidifier operating parameters which optimize system performance are still lacking. Regenerating air stream inlet temperatures for comparable experimental prototype systems are reported ranging from as low as 65°C [7] to as high as 103°C [25]. Wheel revolution speed and amount of desiccant used are important design parameters, yet are not always mentioned in system descriptions [12], or are neglected by modeling the dehumidifier as an infinite capacity machine [26]. Although tuning the ratio of regenerating air to process air mass flow rate to an optimal value allows considerable savings in regeneration heat required, system simulations incorporating unbalanced flow through the regenerator have only recently been performed [3].

Jurinak [3] investigated the influence of regenerating air inlet temperature, wheel revolution speed and mass flow rate ratio on the performance of solid desiccant dehumidifier cooling systems. Values for the parameters Γ_1 and Γ_1/Γ_2 of 0.15 and 0.60, respectively, for the recirculation mode, and values of 0.15 and 0.80 for the ventilation cycle are recommended. The COP based on thermal energy input shows a maximum between 65°C and 85°C regenerating tempera-

ture, while the COP based on electrical energy input is maximal at 105°C. These values, however, are the results of a calculation which optimizes only one parameter at a time.

Ingram et al. [10] present performance charts for solid desiccant rotary dryer systems. These charts show the process outlet state of the dehumidifier as a function of the inlet conditions for a set of optimal design parameters, including a dimensionless wheel period and number of transfer units. The optimization was performed with respect to minimal outlet humidity ratio of the process air stream. However, the presented results do not include the effect of unbalanced flow operation.

This chapter presents a three parameter optimization procedure for desiccant dehumidifier systems to minimize the purchased energy supply for a given cooling load. The calculations are based on the ϵ -NTU method, which is outlined in the preceeding chapters of this text. The optimization model with its assumptions is outlined in section 4.1. Section 4.2 presents optimal values for mass flow ratio, wheel revolution speed and regenerating temperature in graphical form. An analysis of the sensitivity of the required regenerating heat supply to the deviation of these parameters from their optimal values is given in section 4.3. A discussion of various control strategy options for dehumidifier cooling systems is provided in section 4.4.

4.1 Model Formulation and Assumptions

The analysis will focus on the dehumidifier-regenerator subsystem that is common to both the ventilation and recirculation cycles and schematically shown in Figure 4.1.

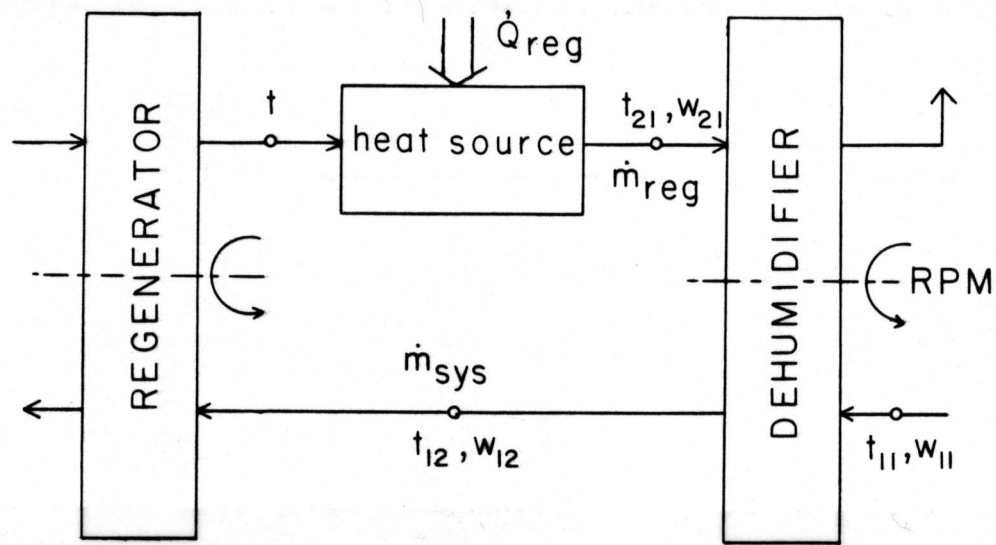


Figure 4.1 Dehumidifier-regenerator subsystem of a desiccant cooling system

The regenerating heat supply, $\dot{Q}_{reg}$, is given by

$$\dot{Q}_{reg} = \dot{m}_{reg} c_p (t_{21} - t) \quad (4.1)$$

where $\dot{m}_{reg}$ is the regenerating air stream mass flow rate, t is the temperature at the exhaust of the heat exchanger, and c_p is the specific heat for constant pressure of the air stream. Introducing

Δt , t may be expressed as

$$t = t_{12} - \Delta t \quad (4.2)$$

where Δt is a measure of the heat exchanger performance. Using the parameters Γ_1 and Γ_2 , $\dot{m}_{reg}$ may be expressed by

$$\dot{m}_{reg} = \frac{\Gamma_1}{\Gamma_2} \dot{m}_{sys} \quad (4.3)$$

where $\dot{m}_{sys}$ is the required process air stream mass flow rate.

Applying an energy balance on the conditioned space yields

$$\dot{m}_{sys} = \frac{\dot{Q}_{load}}{\Delta h} \quad (4.4)$$

where Δh is the difference in enthalpy between the processed air entering the room and the room air (states 4 and 5 in Figure 1.1, 9 and 6 in Figure 1.2), and $\dot{Q}_{load}$ is the total cooling load.

Assumptions

The following assumptions are made in the analysis.

i) Because the variation of Δt with inlet conditions and mass flow rates through the heat exchanger is small compared to the temperature increase provided by the heat source,

$$\delta(\Delta t) \ll t_{21} - t \quad (4.5)$$

it is assumed that Δt is constant equal to 4°C. In Figure 1.1, typical temperatures for states 2,3,6,7 are, respectively, 61°C,

26°C, 20°C and 57°C. These temperatures correspond to a heat exchanger effectiveness of 90%, which is typical of a high performance rotary regenerator [20].

ii) The temperature of the process air at the exhaust of the heat exchanger is assumed to be equal to the room temperature. This is a very good approximation for the ventilation cycle. In Figure 1.1, the difference in temperature between states 3 and 6 is typical 6°C, which is approximately the difference between the dry and wet bulb room air temperature. The assumption is not as good for the recirculation cycle (states 8 and 6 in Figure 1.2), but the results of this analysis for the recirculation cycle based on this assumption agree with previous work [3]. Bearing in mind that evaporative cooling is essentially a constant enthalpy process, equation (4.4) may be written as

$$\dot{m}_{\text{sys}} = \frac{\dot{Q}_{\text{load}}}{h_{\text{fg}} (w_{\text{room}} - w_{12})} \quad (4.6)$$

where w_{room} is the room air humidity ratio and h_{fg} is the latent heat of vaporation of the water vapor.

Substituting (4.2), (4.3) and (4.6) into (4.1), the regenerating heat supply may be expressed as

$$\dot{Q}_{\text{reg}} = \frac{c_p}{h_{\text{fg}}} \frac{\Gamma_1}{\Gamma_2} \frac{(t_{21} - t_{21} + \Delta t)}{(w_{\text{room}} - w_{12})} \dot{Q}_{\text{load}} \quad (4.7)$$

iii) The dehumidifier transfer parameters are

$$NTU_{t,1} = 15.$$

$$NTU_{t,2} = NTU_{t,1} \frac{\Gamma_2}{\Gamma_1} \quad (4.8)$$

$$Le = 1.$$

These transfer coefficients imply that the resistance for heat and mass transfer between the air stream and the surface of the desiccant particles dominates the overall transfer process and that the flow of the air through the matrix is laminar.

The cooling load, $\dot{Q}_{load}$, the process air inlet state (t_{11}, w_{11}), the regenerating air inlet humidity ratio, w_{21} , and the room air state, w_{room} , are specified. With assumptions i) through iii), the regeneration heat required, given by equation (4.7), may then be minimized as a function of t_{21} , Γ_1 and Γ_2 . The results of these calculations are presented in the next section.

4.2 Operating Parameter Design Values for Rotary Dehumidifiers

4.2.1 Regenerating Air Mass Flow Rate and Wheel Revolution Speed

For given values of t_{11} , w_{11} , t_{21} and w_{21} the regenerating heat supply was calculated for values of Γ_1 and Γ_1/Γ_2 , ranging from, respectively, 0.05 to 0.30 with steps of 0.025 and 0.4 to 1.1 with steps of 0.1. The cooling load was held constant and the room air humidity ratio was set to 0.0111 kg/kg. Table 4.1 provides typical results. In this table, the regenerating heat supply is normalized with respect to the minimum value that occurs within the table (at

Table 4.1 Normalized regenerating heat supply as a function of Γ_1 and Γ_1/Γ_2
 Process air inlet state (30°C, 0.015 kg/kg)
 Regenerating air inlet state (85°C, 0.017 kg/kg)

Γ_1	0.100	0.125	0.150	0.175	0.200
Γ_1/Γ_2					
0.6	1.190	1.217	1.262	1.312	1.366
0.7	1.082	1.047	1.060	1.083	1.108
0.8	1.107	1.022	<u>1.000</u>	1.011	1.024
0.9	1.146	1.052	1.024	1.024	1.043
1.0	1.221	1.114	1.080	1.086	1.096

$\Gamma_1 = 0.15$ and $\Gamma_1/\Gamma_2 = 0.8$). The values of Γ_1 and Γ_1/Γ_2 for which Q_{reg} is minimum for various process and regenerating air inlet states are listed in Tables 4.2 and 4.3.

Table 4.1 shows that the regeneration heat required is less sensitive to Γ_1 than to Γ_1/Γ_2 . Tables 4.2 and 4.3 show that the influence of t_{11} and w_{21} on the optimal value of Γ_1/Γ_2 is of second order compared to the influence of w_{11} and t_{21} . The optimal values of Γ_1 and Γ_1/Γ_2 may therefore be averaged with respect to t_{11} and w_{21} , and correlated as a function only of w_{11} and t_{21} . These averaged optimal values are shown in Figure 4.2 for specified process air inlet humidity ratio and regenerating air inlet temperature.

The results in Figure 4.2 demonstrate that an increase in w_{11} for a given regenerating temperature causes both $\Gamma_{1,\text{opt}}$ and $(\Gamma_1/\Gamma_2)_{\text{opt}}$ to increase. To extract more moisture from the process air stream, the matrix should be operated at a higher regenerating mass flow rate, contain more desiccant material, and/or rotate at a higher speed. When increasing the regenerating temperature t_{21} for a given process humidity ratio, $\Gamma_{1,\text{opt}}$ increases but $(\Gamma_1/\Gamma_2)_{\text{opt}}$ decreases. The process outlet humidity ratio will decrease (see Appendix A) and hence also the process mass flow rate. Therefore, a lower regenerating flow rate is required. The optimal amount of adsorbent in the dehumidifier or its revolution speed might increase or decrease depending on the inlet conditions. Points marked with V and R in Figure 4.2 show typical data for the ventilation and recirculation cycle. The values for $(\Gamma_1/\Gamma_2)_{\text{opt}}$ for these points

Table 4.2 Optimal values for Γ_1 and (Γ_1/Γ_2) as a function of the process air inlet state and the regenerating air inlet temperature

w_{11} (g/kg)	$\Gamma_{1,opt}$				$(\Gamma_1/\Gamma_2)_{opt}$			
	10.0	12.5	15.0	17.5	10.0	12.5	15.0	17.5
t_{11} (°C)	$t_{21} = 70^\circ\text{C}$				$w_{21} = 0.017 \text{ kg/kg}$			
25.0	0.050	0.075	0.100	0.100	0.5	0.7	0.8	1.0
30.0	0.050	0.075	0.100	0.100	0.5	0.7	0.8	1.0
35.0	0.050	0.100	0.100	0.100	0.5	0.8	0.9	0.9
	$t_{21} = 85^\circ\text{C}$				$w_{21} = 0.017 \text{ kg/kg}$			
25.0	0.075	0.100	0.125	0.150	0.5	0.6	0.8	0.9
30.0	0.075	0.125	0.150	0.175	0.5	0.7	0.8	0.9
35.0	0.100	0.125	0.150	0.175	0.6	0.7	0.8	1.0
	$t_{21} = 100^\circ\text{C}$				$w_{21} = 0.017 \text{ kg/kg}$			
25.0	0.100	0.125	0.175	0.200	0.5	0.6	0.7	0.8
30.0	0.100	0.150	0.175	0.200	0.5	0.6	0.7	0.8
35.0	0.150	0.200	0.200	0.200	0.6	0.7	0.8	0.8

Table 4.3 Optimal values for Γ_1 and (Γ_1/Γ_2) as a function of the process air inlet state and the regenerating air inlet humidity ratio

w_{11} (g/kg)	$\Gamma_{1,opt}$				$(\Gamma_1/\Gamma_2)_{opt}$			
	10.0	12.5	15.0	17.5	10.0	12.5	15.0	17.5
t_{11} (°C)	$t_{21} = 85.0^\circ\text{C}$				$w_{21} = 0.014$ kg/kg			
25.0	0.075	0.125	0.150	0.175	0.5	0.7	0.8	0.9
30.0	0.100	0.125	0.175	0.175	0.6	0.7	0.8	0.9
35.0	0.125	0.150	0.175	0.200	0.6	0.8	0.9	1.0
	$t_{21} = 85.0^\circ\text{C}$				$w_{21} = 0.017$ kg/kg			
25.0	0.075	0.100	0.125	0.150	0.5	0.6	0.8	0.9
30.0	0.075	0.125	0.150	0.175	0.5	0.7	0.8	0.9
35.0	0.100	0.125	0.150	0.175	0.6	0.7	0.8	1.0
	$t_{21} = 85.0^\circ\text{C}$				$w_{21} = 0.020$ kg/kg			
25.0	0.075	0.100	0.125	0.150	0.5	0.6	0.8	0.9
30.0	0.075	0.125	0.150	0.150	0.5	0.7	0.8	0.9
35.0	0.100	0.125	0.150	0.175	0.6	0.7	0.8	1.0

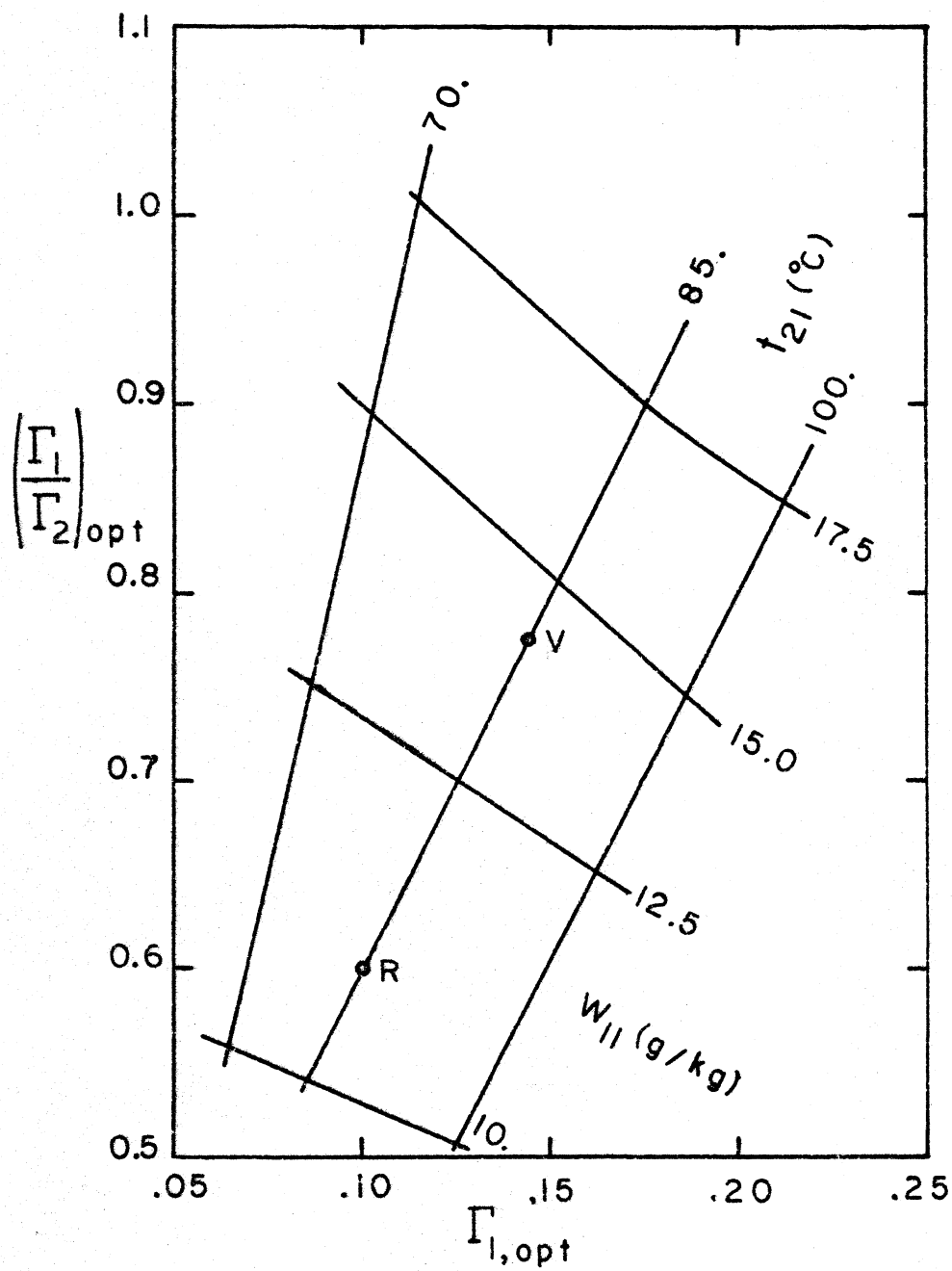


Figure 4.2 Regenerating temperature and process humidity ratio in a $\Gamma_{1,opt} - \left(\frac{\Gamma_1}{\Gamma_2}\right)_{opt}$ chart

correspond to the values reported by Jurinak [3], which were obtained by a complete system analysis.

Tables analogous to Table 4.1 listing w_{12} instead of $\dot{Q}_{reg}$ show that, if the dehumidifier is operated at the optimal values of Γ_1 and Γ_2 for the given inlet conditions, the outlet humidity ratio of the process air stream equals the minimum obtainable value for these inlet conditions. Also, the regenerating mass flow rate is the minimum flow rate that can still regenerate the matrix without decreasing the cooling capacity of the system. Indeed, minimizing the regenerating heat supply for fixed inlet conditions corresponds to minimizing the regenerating mass flow rate. Hence, choosing the right values for Γ_1 and Γ_2 yields another advantage: it minimizes both mass flow rates and therefore the electrical fan power while producing more dehumidification.

4.2.2 Regenerating Air Inlet Temperature

Assuming a "tuned" dehumidifier, i.e., assuming optimal values for Γ_1 and Γ_2 , the regenerating heat supply may be correlated versus t_{11} , w_{11} , t_{21} and w_{21} . Figure 4.3 gives $\dot{Q}_{reg}$ as a function of the regenerating temperature and process humidity ratio for given process temperature and regenerating humidity ratio. $\dot{Q}_{reg}$ is normalized with respect to the minimum regeneration heat required occurring at $w_{11} = 0.010$ kg/kg and $t_{21} = 85^\circ\text{C}$.

If the dehumidifier is used in the recirculation cycle mode ($w_{11} = 0.010 \dots 0.013$ kg/kg), Figure 4.3 shows that the regenerating

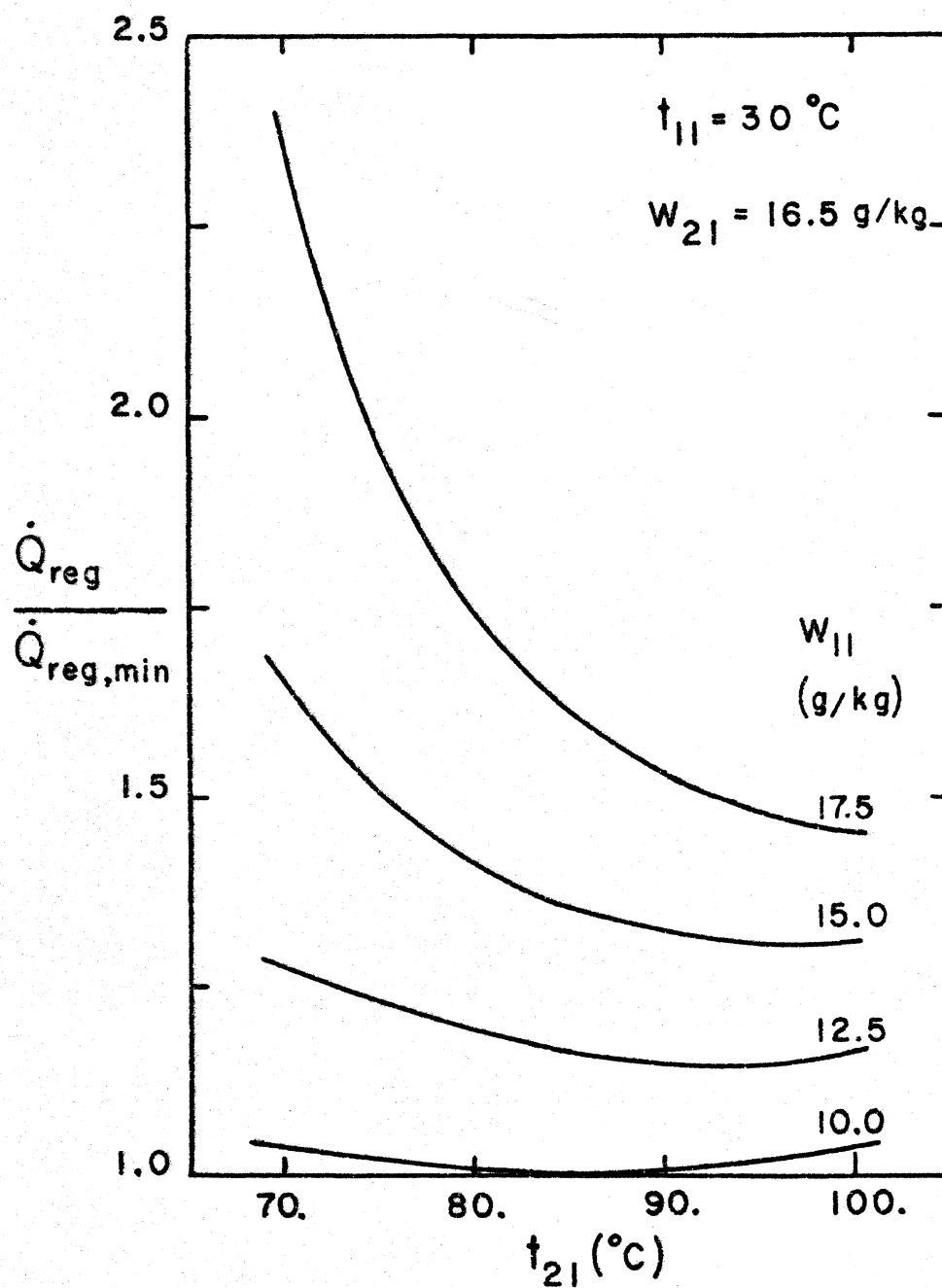


Figure 4.3 Regeneration heat required as a function of regenerating temperature

heat supply is only a weak function of the regenerating temperature. Thus, the COP based on thermal energy input is not sensitive to an optimal choice of the regenerating temperature, for optimal choices for Γ_1 and Γ_2 . However, if the dehumidifier is used in the ventilation cycle mode, higher process humidity ratios (0.015–0.0175) result and a high regenerating temperature is recommended. Previous studies have shown the COP based on thermal energy input to be maximal for t_{21} between 65°C and 85°C, but these studies were performed either assuming balanced flow [16], or fixed values for Γ_1 and Γ_2 [6]. Also, these studies do not presume a fixed cooling load, as is done here, but instead presume a fixed process mass flow rate.

Increasing the regenerating temperature will lower the required process mass flow rate because the minimum obtainable process outlet humidity ratio decreases. Also, this allows the dehumidifier to be operated at a lower value for $(\Gamma_1/\Gamma_2)_{\text{opt}}$, hence, the regenerating mass flow rate will decrease twice. The COP of desiccant cooling systems based on electrical energy input will therefore increase with increasing regenerating temperature [5,6]. Thus, to maximize system performance, the regenerating temperature should be high. All experimental prototype air conditioning systems using a silica gel dehumidifier operate at a regenerating temperature between 80°C and 100°C [3,7,8].

A recirculation cycle operates at a lower process temperature than a ventilation cycle. Because Figure 4.3 shows the results of

an analysis assuming a fixed process inlet temperature, this chart cannot be used for comparing ventilation and recirculation cycle systems.

4.3 Sensitivity Analysis

Optimal values for the regenerating temperature, regenerating mass flow rate and wheel speed have been presented in the previous section. If these parameters deviate from their respective optimal values, the required regenerating heat supply will increase. This increase will be a function of the extent of the deviation, but will also be dependent on the strategy employed to control the dehumidifier system.

In this section, the sensitivity of the required regenerating heat supply to wheel speed, regenerating mass flow rate and inlet temperature is discussed. In the control strategy, the process air mass flow rate is adjusted to provide the desired cooling load. Additional control options are considered.

The analysis is carried out for two typical dehumidifier parameter sets, reflecting the use of the dehumidifier in a ventilation and recirculation cycle system. The parameters listed in Table 4.4 are based on the ARI standard room and ambient states of (26.7°C, 0.0111 kg/kg) and (35.0°C, 0.0142 kg/kg), as specified by the Solar Energy Research Institute for testing desiccant air conditioners [27].

Table 4.4 Parameters for sensitivity analysis

VENTILATION CYCLE		RECIRCULATION CYCLE	
t_{11}	= 35.0 °C	t_{11}	= 26.7 °C
w_{11}	= 0.0142 kg/kg	w_{11}	= 0.0111 kg/kg
t_{21}	= 85.0 °C	t_{21}	= 85.0 °C
w_{21}	= 0.0140 kg/kg	w_{21}	= 0.0190 kg/kg
$\Gamma_{1,opt}$	= 0.170	$\Gamma_{1,opt}$	= 0.093
$\left(\frac{\Gamma_1}{\Gamma_2}\right)_{opt}$	= 0.823	$\left(\frac{\Gamma_1}{\Gamma_2}\right)_{opt}$	= 0.600

4.3.1 Regenerating Air Mass Flow Rate

If the regenerating mass flow rate is increased above the critical minimal limit required to provide the given cooling load, the regenerating heat supply will increase. This increase depends primarily on the extent of the deviation, and also on the control strategy employed. Control options might be to let the wheel speed be constant, vary linearly with the process mass flow rate or vary linearly with the regenerating mass flow rate. Similar control options may be considered for the regenerating temperature.

For a given set of inlet conditions, the regenerating heat supply may be calculated from equation (4.7) if Γ_1 and Γ_2 are known. For a given deviation of m_{reg} and a constant cooling load, equations (4.1-4.8) may be used to generate the following sets of equations,

which give Γ_1 and Γ_2 as a function of $(\dot{m}_{reg})/(\dot{m}_{reg})_{opt}$.

i) Wheel speed stays constant,

$$\begin{aligned}\Gamma_1 &= \Gamma_{1,opt} \frac{(w_{room} - w_{12})}{(w_{room} - w_{12,opt})} \\ \Gamma_2 &= \Gamma_{2,opt} \frac{(\dot{m}_{reg})_{opt}}{(\dot{m}_{reg})}\end{aligned}\tag{4.9}$$

ii) Wheel speed is proportional to the process mass flow rate,

$$\begin{aligned}\Gamma_1 &= \Gamma_{1,opt} \\ \Gamma_2 &= \Gamma_{2,opt} \frac{(\dot{m}_{reg})_{opt}}{(\dot{m}_{reg})} \frac{(w_{room} - w_{12,opt})}{(w_{room} - w_{12})}\end{aligned}\tag{4.10}$$

iii) Wheel speed is proportional to the regenerating mass flow rate,

$$\begin{aligned}\Gamma_1 &= \Gamma_{1,opt} \frac{(\dot{m}_{reg})}{(\dot{m}_{reg})_{opt}} \frac{(w_{room} - w_{12})}{(w_{room} - w_{12,opt})} \\ \Gamma_2 &= \Gamma_{2,opt}\end{aligned}\tag{4.11}$$

where w_{12} and $w_{12,opt}$ are given by

$$\begin{aligned}w_{12} &= \bar{w}_{12}(t_{11}; w_{11}; t_{21}; w_{21}; \Gamma_1; \Gamma_2) \\ w_{12,opt} &= \bar{w}_{12}(t_{11}; w_{11}; t_{21}; w_{21}; \Gamma_{1,opt}; \Gamma_{2,opt})\end{aligned}\tag{4.12}$$

The inlet conditions are listed in Table 4.4.

Equations (4.9)-(4.12), (4.10)-(4.12), (4.11)-(4.12), have to be solved iteratively in combination with (4.7) to obtain $\dot{Q}_{reg}$ as a function of $(\dot{m}_{reg})/(\dot{m}_{reg})_{opt}$. Figure 4.4 shows the relative increase of $\dot{Q}_{reg}$ as a function of the relative increase of the regenerating flow rate for the recirculation and ventilation cycle.

The results in Figure 4.4 show only minor differences between the ventilation and recirculation cycle systems. Also, the influence of the different control strategy options is small. Increasing the mass flow rate to 20% above the minimal value required, causes an increase of about 10% in required regenerating heat supply for the same cooling load. Hence, considerable energy savings may be accomplished by tuning the regenerating mass flow rate to its proper value. This is the minimal value required for the given cooling load and inlet conditions, and that cooling load cannot be met if $\dot{m}_{reg}$ is decreased below this value.

4.3.2 Wheel Revolution Speed

In this section, the sensitivity of the dehumidifier performance to deviations of the wheel rotation speed from its optimal value is investigated. Various control strategies for the regenerating mass flow rates are examined, while the regenerating temperature is assumed constant.

For given inlet conditions and cooling load, equations (4.1-4.8) may be used to calculate Γ_1 and Γ_2 as a function of RPM, the wheel

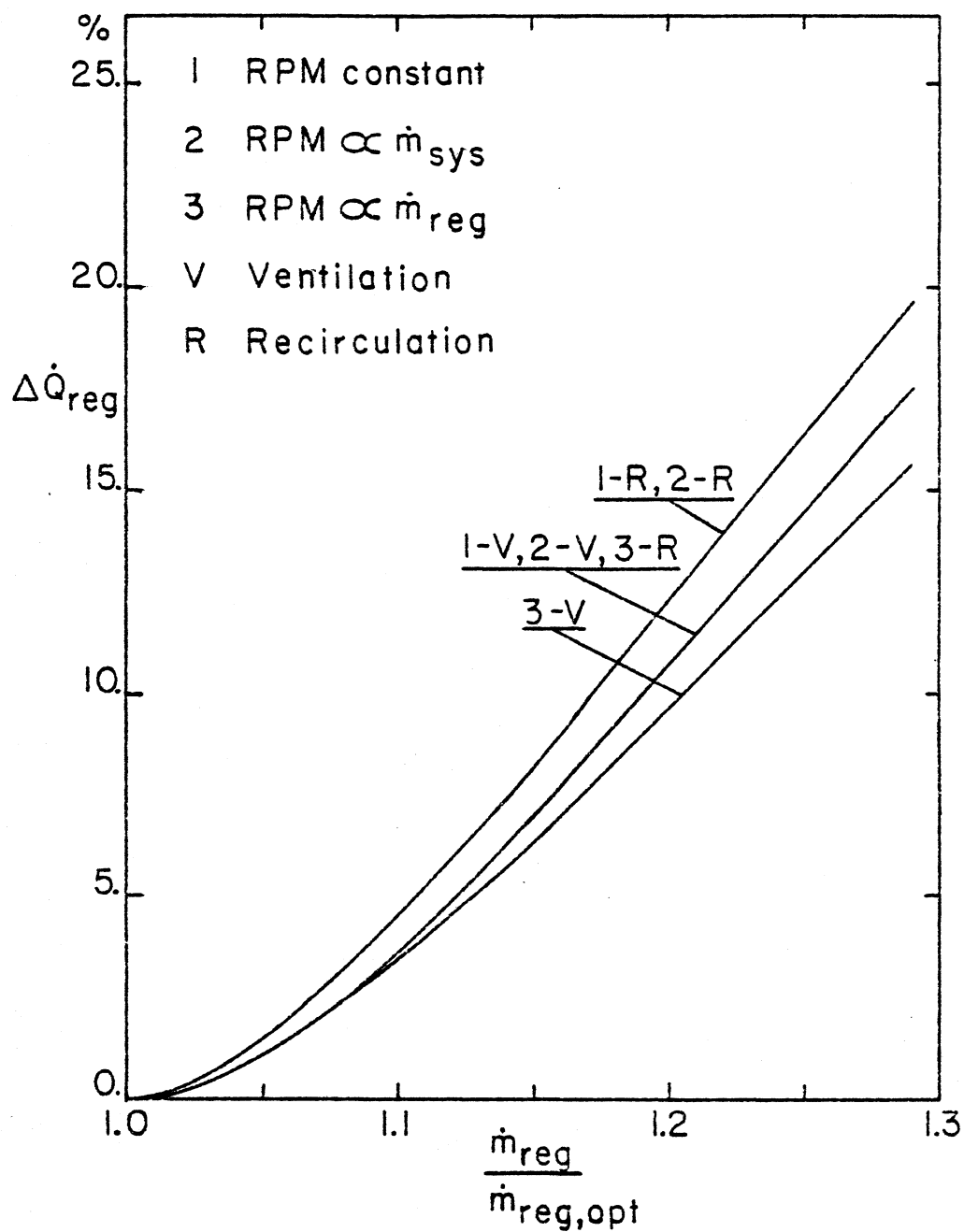


Figure 4.4 Influence of regenerating flow rate variation on regeneration heat required

speed of the dehumidifier. The following sets of equations were used to generate the results of the analysis.

i) Regenerating mass flow rate is constant,

$$\begin{aligned}\Gamma_1 &= \Gamma_{1,opt} \frac{(\text{RPM})}{(\text{RPM})_{opt}} \frac{(w_{room} - w_{12})}{(w_{room} - w_{12,opt})} \\ \Gamma_2 &= \Gamma_{2,opt} \frac{(\text{RPM})}{(\text{RPM})_{opt}}\end{aligned}\tag{4.13}$$

ii) Regenerating mass flow rate is proportional to the process mass flow rate,

$$\begin{aligned}\Gamma_1 &= \Gamma_{1,opt} \frac{(\text{RPM})}{(\text{RPM})_{opt}} \frac{(w_{room} - w_{12})}{(w_{room} - w_{12,opt})} \\ \Gamma_2 &= \Gamma_{2,opt} \frac{(\text{RPM})_{opt}}{(\text{RPM})} \frac{(w_{room} - w_{12,opt})}{(w_{room} - w_{12})}\end{aligned}\tag{4.14}$$

iii) Regenerating mass flow rate is proportional to the wheel revolution speed,

$$\begin{aligned}\Gamma_1 &= \Gamma_{1,opt} \frac{(\text{RPM})}{(\text{RPM})_{opt}} \frac{(w_{room} - w_{12})}{(w_{room} - w_{12,opt})} \\ \Gamma_2 &= \Gamma_{2,opt}\end{aligned}\tag{4.15}$$

Equations (4.13), (4.14) and (4.15) in combination with (4.12) and (4.7) may be solved numerically to calculate the required regenerating heat supply as a function of $(\text{RPM})/(\text{RPM})_{opt}$. The results for the ventilation and recirculation cycle are presented

in Figure 4.5. These results show that, by keeping the regenerating mass flow rate proportional to the process mass flow rate, the regenerating heat supply is least influenced by wheel speed perturbations. In this case, the wheel speed may vary from -20% to +30% without significantly affecting $\dot{Q}_{reg}$. Positive wheel speed deviations are less influential than negative deviations. The dehumidifier performance is, in general, slightly more sensitive to the wheel revolution speed for the ventilation cycle than for the recirculation cycle.

4.3.3 Regenerating Air Inlet Temperature

In section 4.2 it is shown that $\Gamma_{1,opt}$ and $\Gamma_{2,opt}$ are primarily a function of the regenerating temperature for a given process humidity ratio. Moreover it is shown that, if Γ_1 and Γ_2 are controlled with the regenerating temperature, the required regenerating heat supply varies only slightly with the regenerating temperature, provided that this temperature is chosen properly. However, this assumption requires a rather complex control strategy, in which the regenerating mass flow rate and wheel speed are tied to both the regenerating temperature and process mass flow rate. Figure 4.6 shows the relative variation of the required regenerating heat supply with regenerating temperature for four other simple control options. Equations (4.1-4.8) provide the information to generate the equations that are needed to compute the new values for Γ_1 and Γ_2 as a function

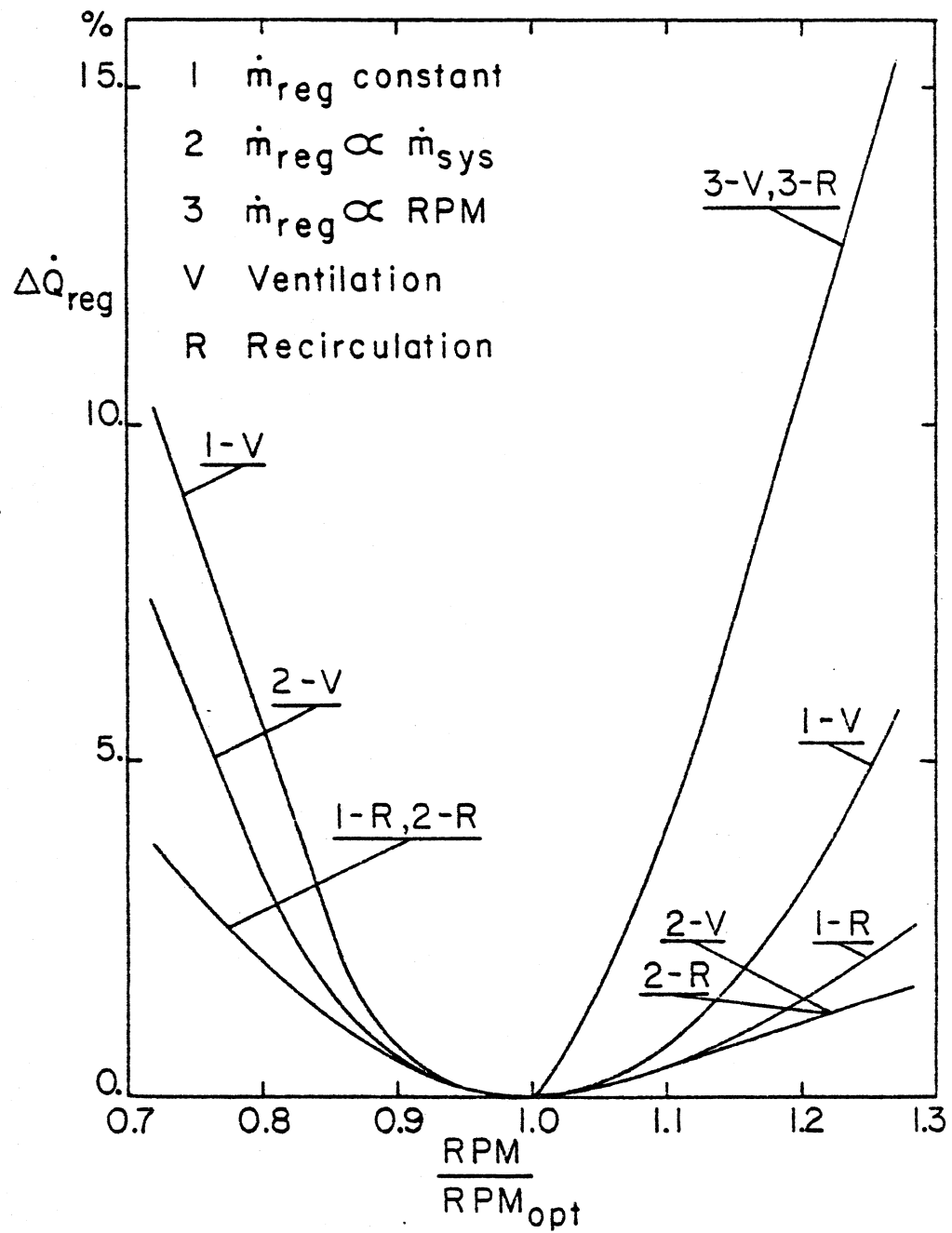


Figure 4.5 Influence of wheel revolution speed variation on regeneration heat required

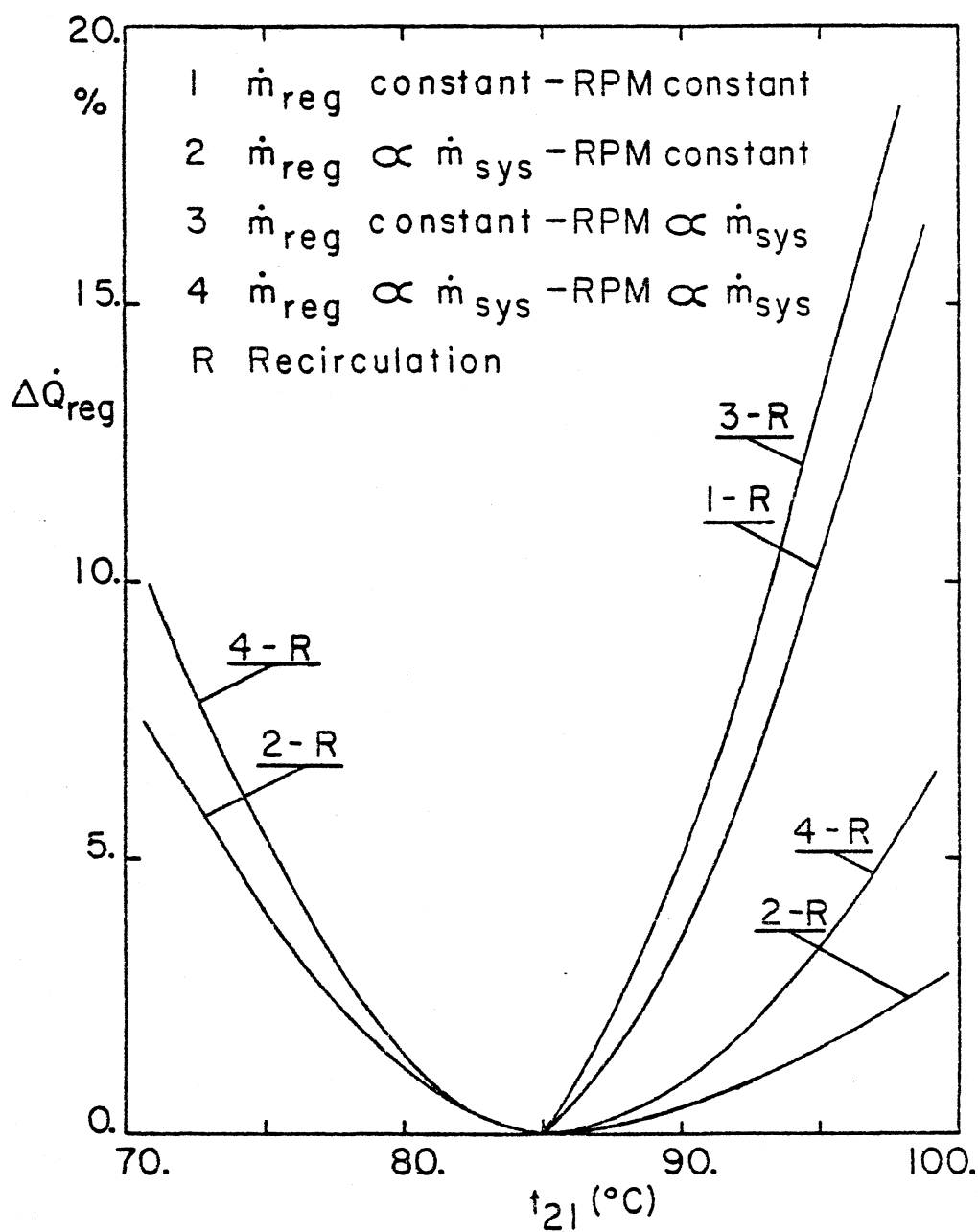


Figure 4.6 Influence of regenerating temperature variation on regeneration heat required

of t_{21} , for given inlet conditions and cooling load. These equations are given below, and again have to be solved in combination with (4.12) and (4.7).

i) $\dot{m}_{\text{reg}}$ and wheel RPM are constant,

$$\Gamma_1 = \Gamma_{1,\text{opt}} \frac{(w_{\text{room}} - w_{12})}{(w_{\text{room}} - w_{12,\text{opt}})} \quad (4.16)$$

$$\Gamma_2 = \Gamma_{2,\text{opt}}$$

ii) $\dot{m}_{\text{reg}}$ is proportional to $\dot{m}_{\text{sys}}$, and wheel RPM is constant,

$$\begin{aligned} \Gamma_1 &= \Gamma_{1,\text{opt}} \frac{(w_{\text{room}} - w_{12})}{(w_{\text{room}} - w_{12,\text{opt}})} \\ \Gamma_2 &= \Gamma_{2,\text{opt}} \frac{(w_{\text{room}} - w_{12,\text{opt}})}{(w_{\text{room}} - w_{12})} \end{aligned} \quad (4.17)$$

iii) $\dot{m}_{\text{reg}}$ is constant and wheel RPM is proportional to $\dot{m}_{\text{sys}}$,

$$\begin{aligned} \Gamma_1 &= \Gamma_{1,\text{opt}} \\ \Gamma_2 &= \Gamma_{2,\text{opt}} \frac{(w_{\text{room}} - w_{12,\text{opt}})}{(w_{\text{room}} - w_{12})} \end{aligned} \quad (4.18)$$

iv) $\dot{m}_{\text{reg}}$ and wheel RPM are proportional to $\dot{m}_{\text{sys}}$,

$$\begin{aligned} \Gamma_1 &= \Gamma_{1,\text{opt}} \\ \Gamma_2 &= \Gamma_{2,\text{opt}} \end{aligned} \quad (4.19)$$

Figure 4.6 shows only the results for the recirculation cycle. Curves for the ventilation cycle show the same trends. The results show that the regenerating heat supply is least affected by the regenerating temperature for control option ii). In this case the regenerating temperature may vary from 75°C to 100°C without significantly increasing $\dot{Q}_{reg}$. Options in which the regenerating mass flow rate is kept constant are to be avoided, while it is preferable to keep the matrix rotating at a constant speed.

4.4 Control Strategies for Desiccant Dehumidifier Cooling Systems

The optimal strategy for controlling a dehumidifier cooling system will provide the given cooling load with the least possible regenerating heat supply. This control strategy will not only maximize the COP based on thermal energy input but also the COP based on electrical energy input. An ideal controller would therefore sense room state and ambient conditions and adjust the process and regenerating mass flow rate, the wheel revolution speed and the regenerating temperature to optimal values. However, such task can only be accomplished by a direct digital control system. Since this might not pay off for small scale applications, simpler control strategies are needed.

The sensitivity analysis, discussed in section 4.3, allows one to examine the influence of different control strategy options. If it is assumed that the process mass flow rate is adjusted to provide the given cooling load, the following control options are

recommended:

- i) The regenerating mass flow rate should vary in proportion to the process mass flow rate.

At the optimal point, the outlet humidity ratio of the process air stream is minimal. Hence, any deviation from this optimal point will increase the process mass flow rate. If, however, the regenerating mass flow is increased too, the regenerating capacity of the matrix increases. This will in turn lower the process outlet humidity ratio, and hence, the process mass flow rate will drop. Thus, this control strategy is inherently stable, and will always provide the least possible flow rates for the following set of given parameters:

- ratio of regenerating mass flow rate to process mass flow rate,
- wheel revolution speed,
- regenerating temperature,
- room state, cooling load and ambient conditions.

It is important to choose and maintain a proper value for the ratio of regenerating mass flow rate to process mass flow rate.

- ii) The wheel speed may be kept constant.

As has been shown, the influence of wheel speed control on the regenerating energy demand is only of secondary order. Since the dehumidifier wheel rotates at a very low speed, it is important

that this speed may be set constant. The choice of the speed is not critical. Deviations of -20% to +30% of the optimal speed do not significantly decrease the COP of the system, provided that condition i) has been met.

iii) The regenerating temperature should be high.

To maximize the COP based on electrical energy input, the regenerating temperature should be set to 85°C ... 100°C. The COP based on thermal energy input is only weakly influenced by the regenerating temperature. Once this temperature has been chosen, the wheel speed and the ratio of regenerating to process mass flow rate can be determined, according to Figure 4.2.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

This work presents an analysis of solid desiccant rotary dehumidifiers. The ideal dehumidifier is modeled by a wave analysis, and correlations for the effectiveness of a silica gel dehumidifier are presented. The use of dehumidifiers in desiccant cooling systems is investigated, and control strategies for these systems are discussed. The following conclusions can be made with regard to the three main chapters in this thesis:

1. The governing mass and energy conservation equations for a dehumidifier with infinite transfer coefficients are analyzed using the method of characteristics and the shock wave method. The method of characteristics is applicable when the desiccant is being processed by the process air stream. The shock wave method is applicable when the desiccant is being regenerated by the regenerating air stream. Depending on the operating parameters Γ_1 and Γ_2 , the performance of the ideal dehumidifier is determined either by the propagation of two expansion waves during the processing period or by the propagation of two shock waves during the regenerating period. A dehumidifier operating chart is introduced in the form of a Γ_1 - Γ_2 diagram which shows these two regions. In each region, either the shock wave method or the method of characteristics provides analytical performance correlations. The process air outlet state of the ideal dehumidifier calculated using these correlations is in good

agreement with that obtained from a finite difference computer program for a dehumidifier with high heat and mass transfer coefficients. No assumptions are made with respect to the desiccant or the fluid state properties. Therefore, the analysis may be applied to other desiccants such as molecular sieves, manganese dioxide or lithium chloride.

2. The finite difference program and the ideal dehumidifier performance correlations were used to establish empirical correlations for the effectiveness of a dehumidifier with finite transfer coefficients. Correlations for the moisture and enthalpy effectiveness of a silica gel dehumidifier are presented. The principles that are used to generate these correlations are general and may be applied to dehumidifiers using other desiccants. An ϵ -NTU model, incorporating the correlations for the effectiveness and the ideal dehumidifier performance, allows rapid and accurate calculation of the dehumidifier performance. The model is valid for a wide range of inlet conditions, operating parameters and transfer coefficients and is suitable to be used in detailed long term performance simulations of desiccant cooling systems.

3. The performance of the rotary heat exchanger-rotary dehumidifier subsystem in desiccant cooling systems is analyzed. For a given cooling load, the required regeneration heat supply can be minimized by choosing appropriate values for the regenerating air mass flow rate and the wheel rotation speed. A map is presented showing

optimal values for rotating speed and regenerating flow rate as a function of the regenerating air inlet temperature and the process air inlet humidity ratio. This regenerating temperature is further optimized as a function of the process humidity ratio. In the analysis, the control strategy adjusts the process air mass flow rate to provide the required cooling load. Additional control options are considered and the sensitivity of the regeneration heat required to the wheel speed, regenerating air mass flow rate and inlet temperature is discussed. Substantial energy savings can be achieved by reducing the regenerating mass flow rate from balanced flow to between 60% and 80% of the process mass flow rate, depending on the inlet conditions of the dehumidifier. The results show that the ratio of the regenerating mass flow rate to the process mass flow rate should be kept fixed at the optimal value. Fine tuning of the regenerator wheel rotation speed however, is less important.

The following suggestions may serve as a guidance for future research in the field of desiccant deployment for air conditioning purposes:

1. A suitable model for rotary dehumidifiers incorporating moisture adsorption hysteresis effects of the desiccant should be developed. In the ideal dehumidifier theory, the adsorption process during the processing period is modeled separately from the desorption process during the regenerating period. The theory may therefore be extended to account for this effect.

2. The analogy theory has been applied to other desiccants than silica gel [13]. Expressions for the F_i potentials used in the method of characteristics should be established for other standard desiccants, such as molecular sieves, manganese dioxide and lithium chloride. Based on these expressions, the ideal dehumidifier model can be extended to calculate the performance of dehumidifiers employing these other desiccants.

3. The dehumidifier is modeled assuming lumped transfer coefficients for heat and mass between the desiccant and the air streams. Allander [35] showed that, for nominal silica gel packed beds, the heat and mass transfer coefficients at the air side may be multiplied by a correction factor less than 1 to account for diffusion resistance effects inside the desiccant particles. That approach should be extended to other desiccants and flow geometries. Pesaran et al. [34] solved the diffusion rate and conservation equations for moist air-silica gel by means of a finite difference scheme. Results obtained with that model are in good agreement with experimental results.

4. Optimal control strategies for rotary dehumidifiers in desiccant cooling and drying systems should be determined. The optimization analysis should be based upon long term system simulations, detailed component models, and should take into account the cost and feasibility of the various control options considered.

APPENDIX A

Performance Charts for the Ideal Rotary Dehumidifier

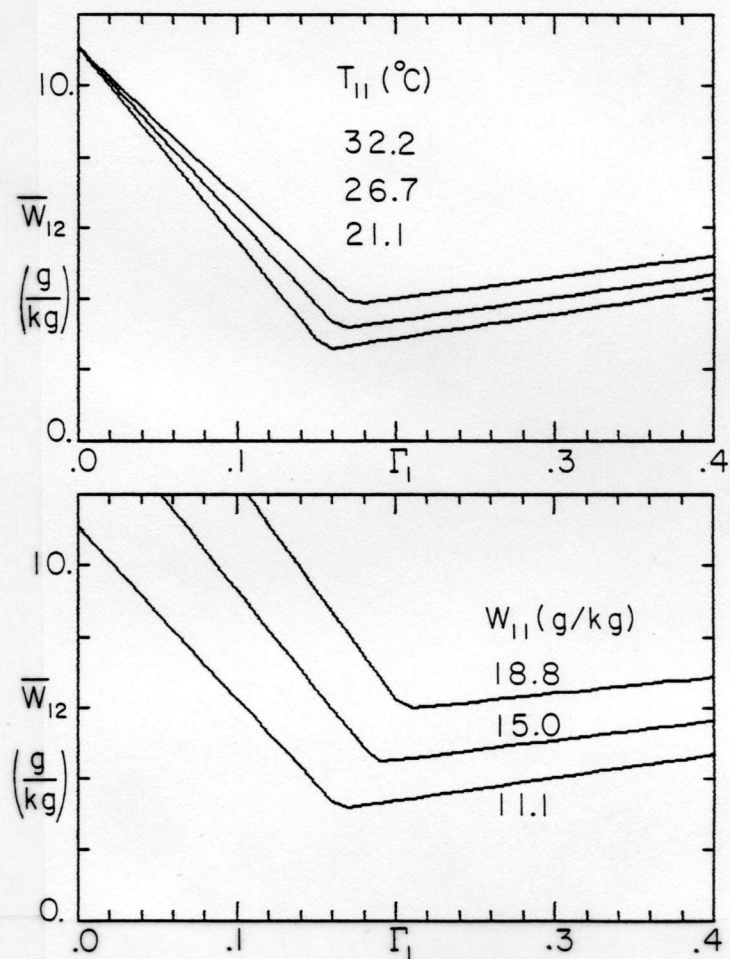
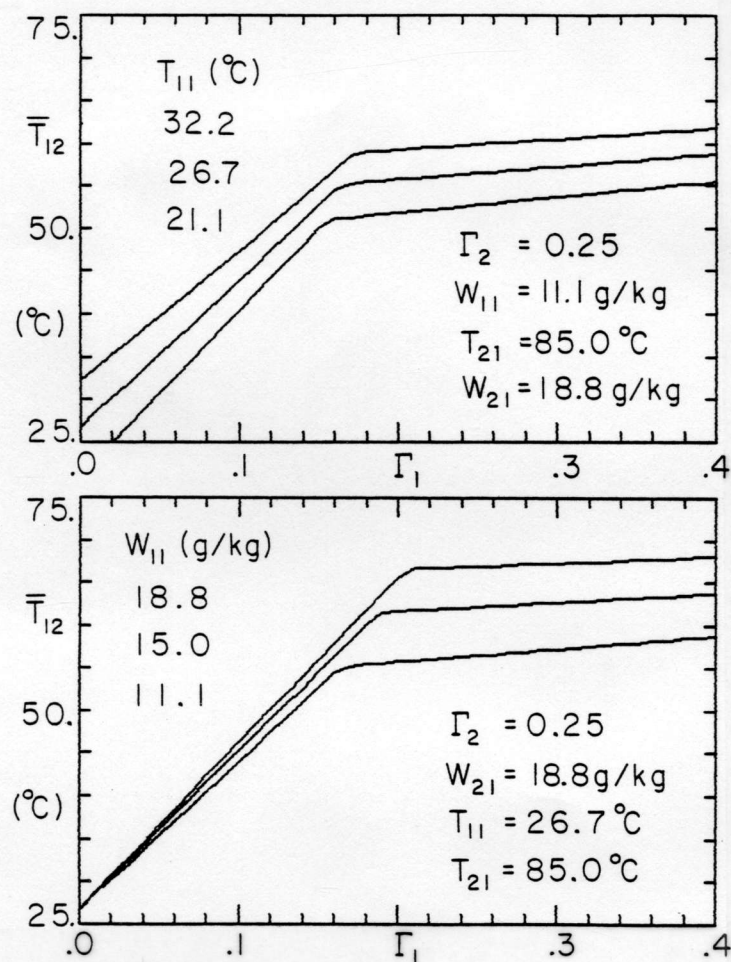


Figure A.1 Influence of process air inlet state properties on process air outlet state properties

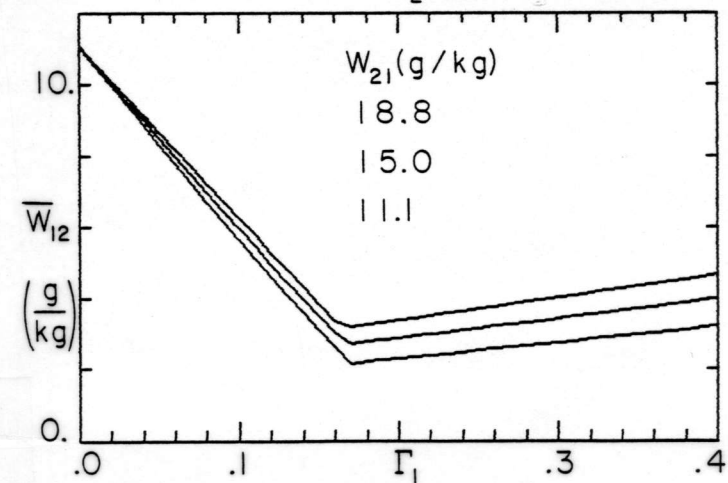
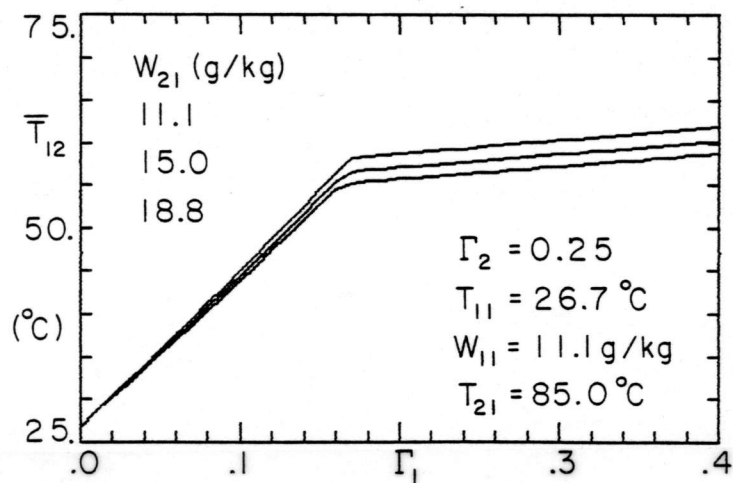
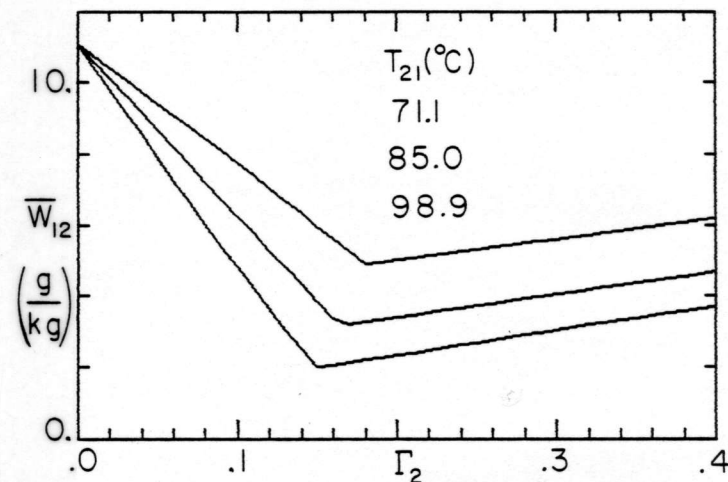
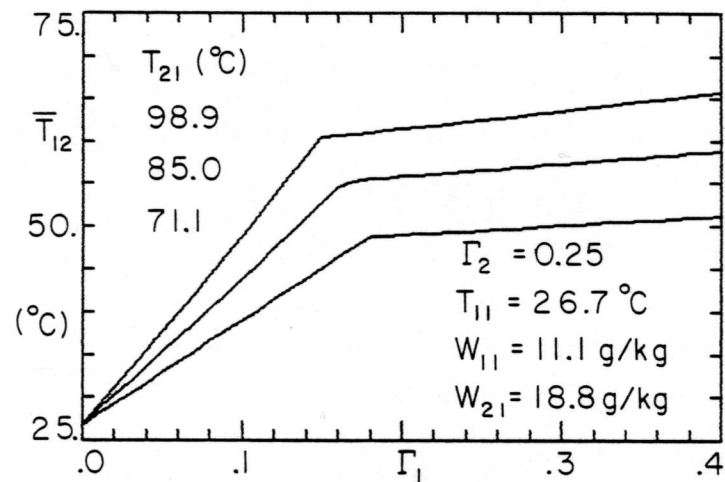


Figure A.2 Influence of regenerating air inlet state properties on process air outlet state properties

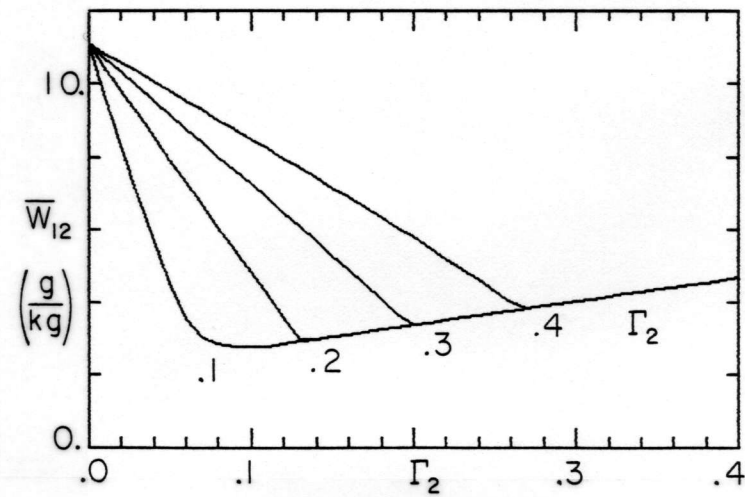
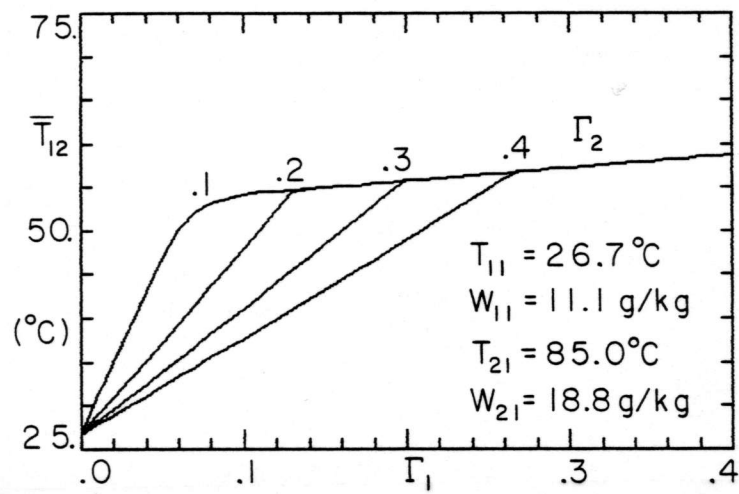


Figure A.3 Influence of operating parameters Γ_1 and Γ_2 on process air outlet state properties

APPENDIX B

Effectiveness Charts for a Rotary

Silica Gel Dehumidifier

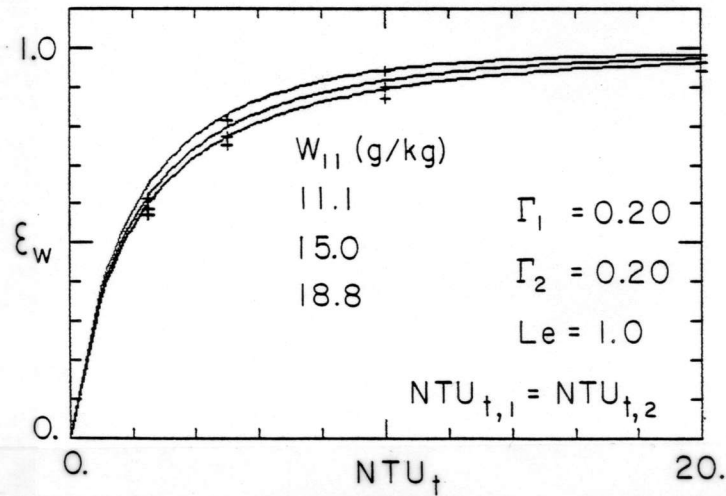
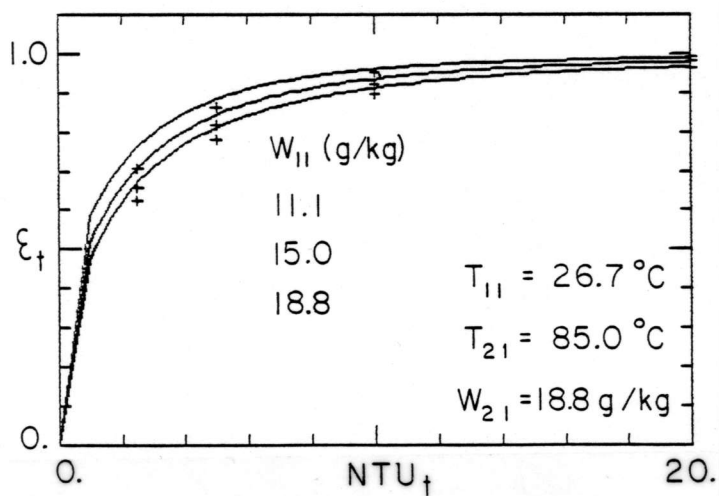
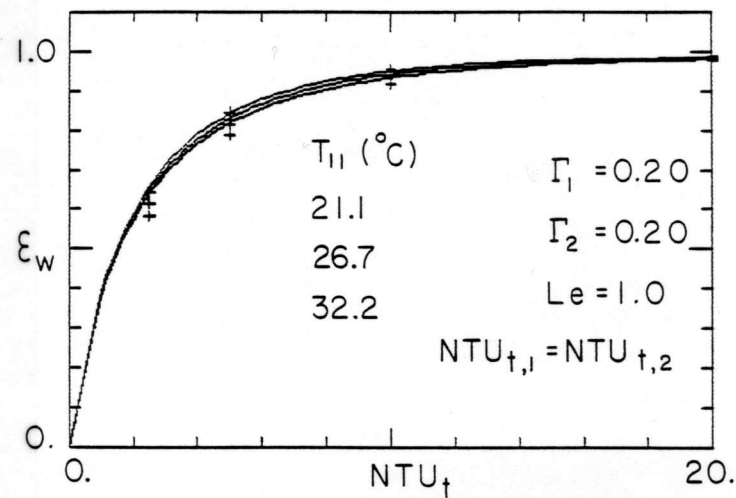
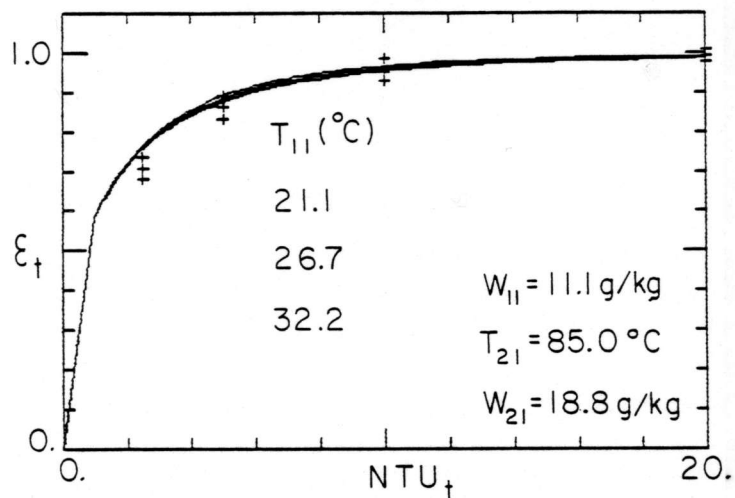


Figure B.1 Influence of process air inlet state properties on process air temperature and humidity ratio effectiveness

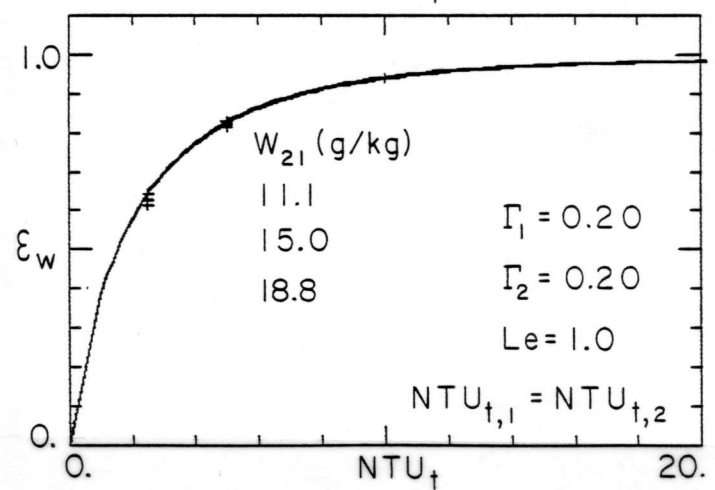
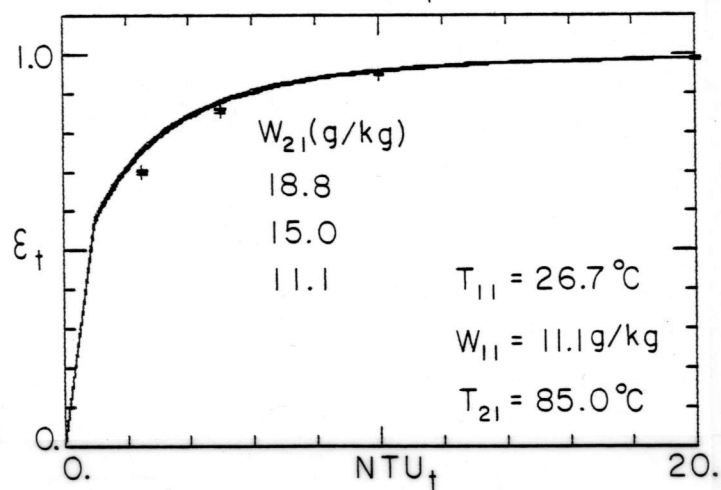
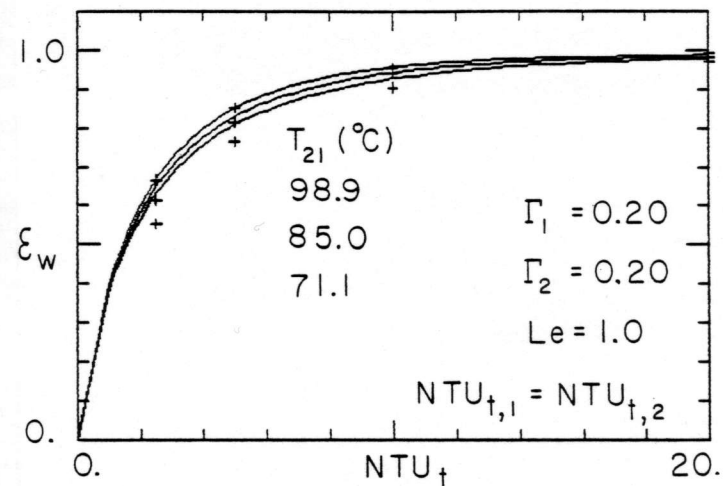
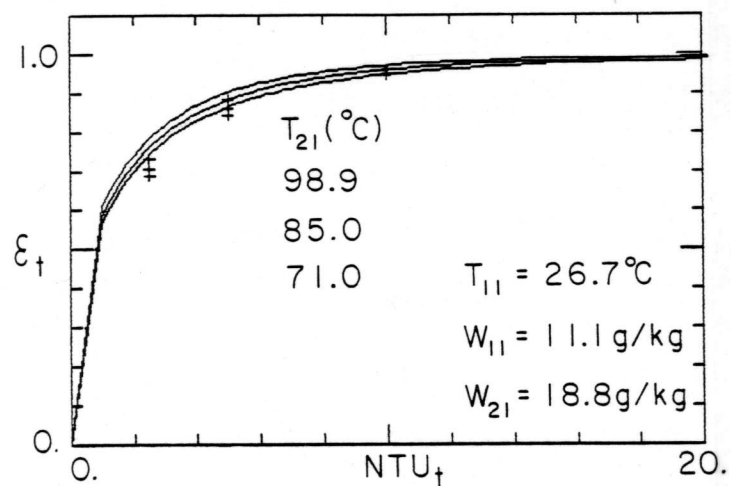


Figure B.2 Influence of regenerating air inlet state properties on process air temperature and humidity ratio effectiveness

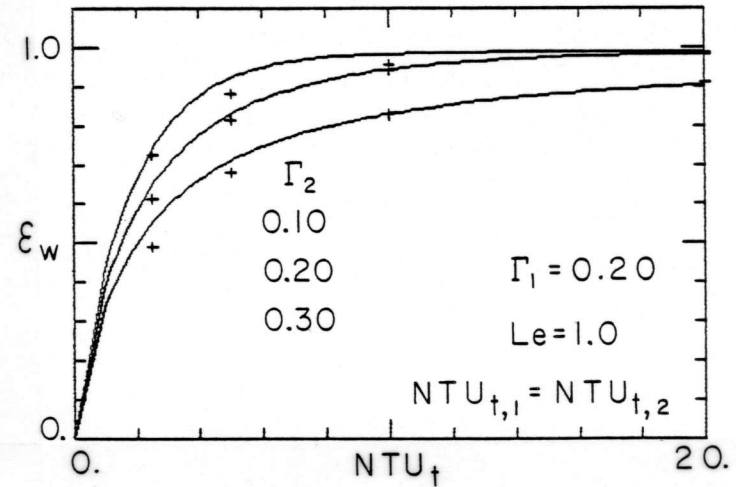
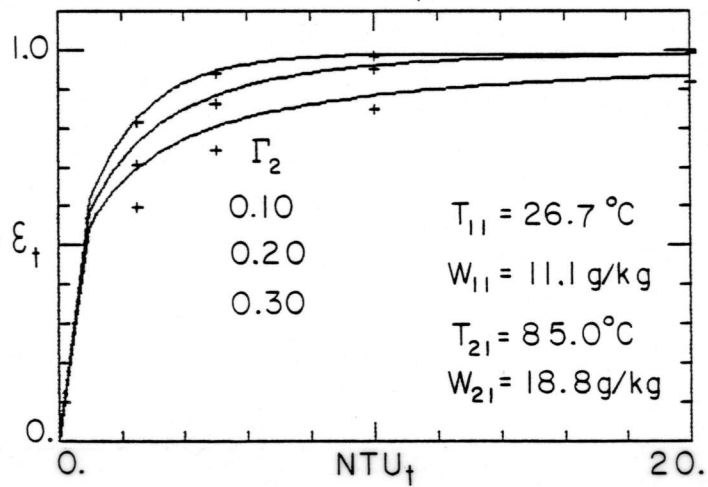
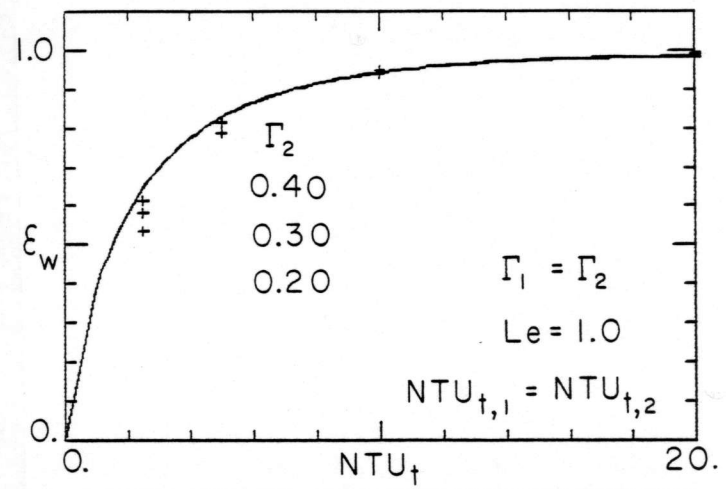
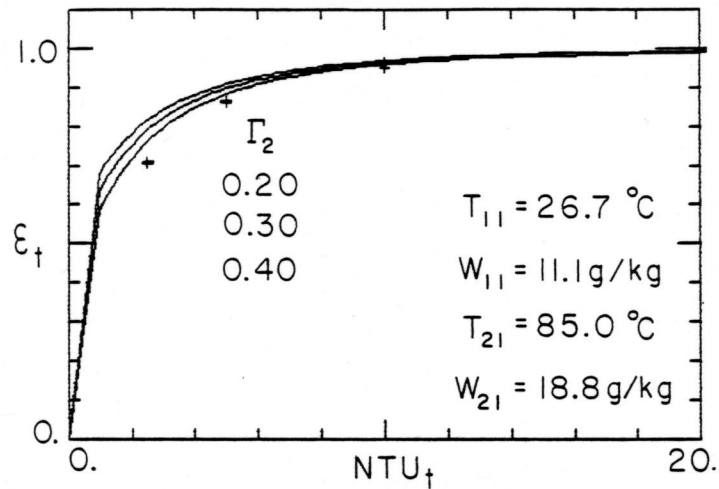


Figure B.3 Influence of operating parameters Γ_1 and Γ_2 on process air temperature and humidity ratio effectiveness

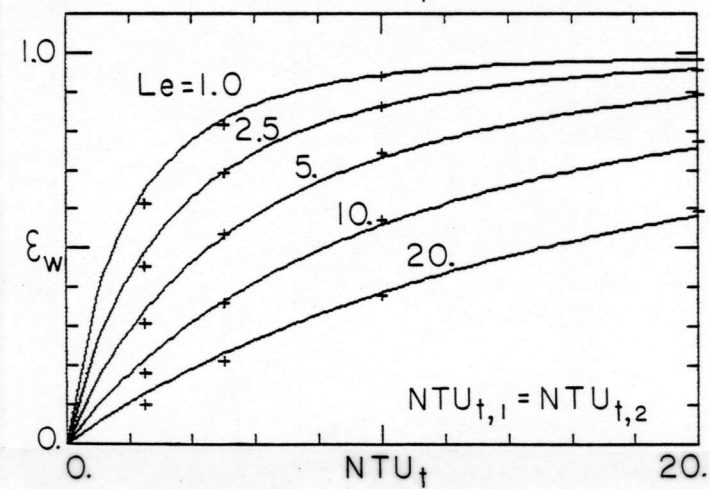
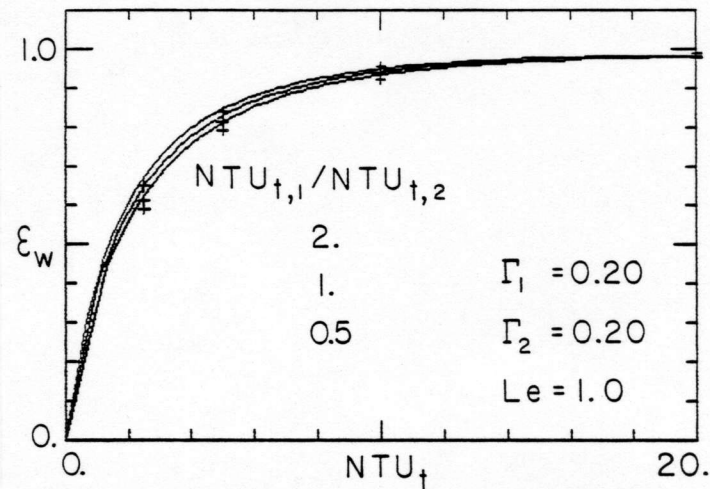
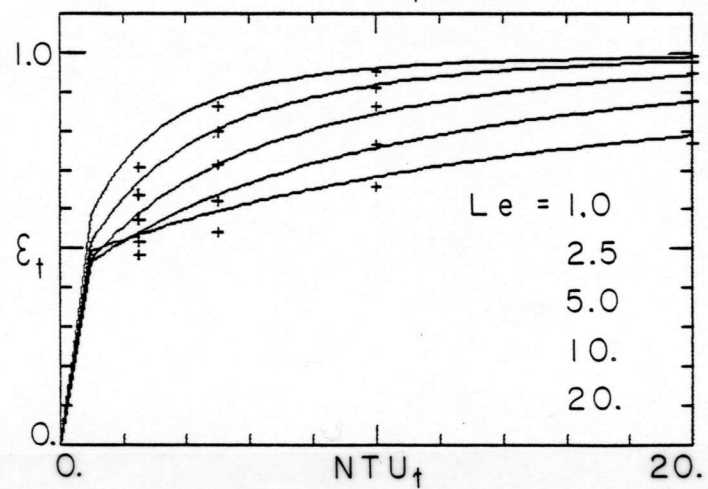
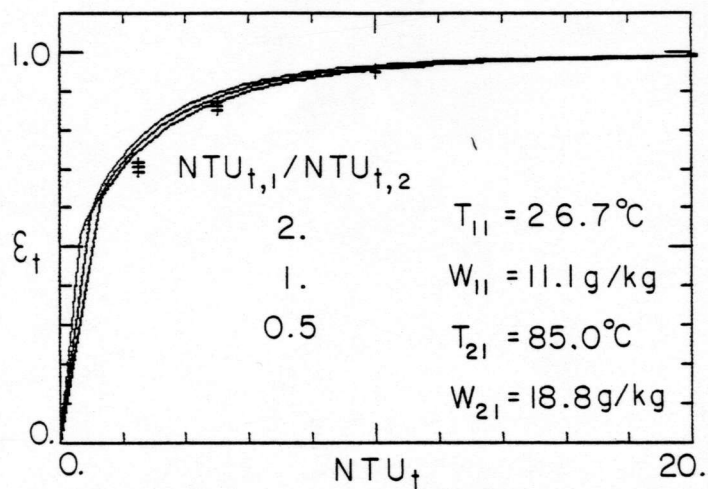


Figure B.4 Influence of transfer parameters and Lewis number on process air temperature and humidity ratio effectiveness

REFERENCES

1. Carnot, N.L.S., Reflections sur la Puissance Motrice du Feu et sur les Machines, Bachelier, Paris, (1824).
2. Dunkle, R.V., "A Method of Solar Air Conditioning," Mech. and Chem. Eng. Trans., Inst. Engns. Australia, MCI, 73, (1965).
3. Jurinak, J., "Open Cycle Solid Desiccant Cooling--Component Models and Systems Simulations," Ph.D. thesis, University of Wisconsin-Madison, (1982).
4. Proceedings of the 1981 Annual DOE Active Heating and Cooling Contractor's Review Meeting, U.S. Department of Energy, (1981).
5. Sheridan, J.C., and Mitchell, J.W., "Hybrid Solar Desiccant Cooling Systems," Proceedings of the 1982 Annual Meeting of AS/ISES, (1982).
6. Howe, R.R., Beckmann, W.A., and Mitchell, J.W., "Commercial Applications for Solar Hybrid Desiccant Systems," Proceedings of the 1983 Annual Meeting of ASSES, (1983).
7. Monnier, J.B., Worek, W.M., and Lavan, Z., "Performance of a Cross-Cooled Solar-Powered Desiccant Cooling System," Illinois Institute of Technology Report No. DES-81-1, (1982).
8. Shultz, K.J., and Mitchell, J.W., "The Performance of Solar-Desiccant Systems Using Cooled Desiccant Beds," Proceedings of the 1982 Annual Meeting of AS/ISES, 585-590, (1982).
9. Rousseau, J., "Development of a Solar Desiccant Dehumidifier: Phase II Technical Progress Report," AiResearch Manufacturing Company Report No. 81-17773, (1981).
10. Wurm, J., et al., "SOLAR-MEC Development Program: Semi-annual Progress Report," Institute of Gas Technology Report No. COO-4495-23, (1979).
11. Biswas, P., "An Experimental Study of the Performance of a Laminar Flow Silica Gel Desiccant Packing Suitable for Solar Air Conditioning Applications," MS thesis, UCLA, (1981).
12. Barlow, R.S., "An Assessment of Dehumidifier Geometries for Desiccant Cooling Systems," SERI/TR-252-1529, (1983).

13. Maclaine-cross, I.L., "A Theory of Combined Heat and Mass Transfer In Regenerators," Ph.D. thesis, Department of Mechanical Engineering, Monash University, Australia, (1974).
14. Barlow, R., "Analysis of the Adsorption Process and of Desiccant Cooling Systems—A Pseudo-Steady-State Model for Coupled Heat and Mass Transfer," SERI/TR-631-1330, (1982).
15. Pla Barby, F., "Solid Desiccant Air Conditioning with Silica Gel Using Solar Energy," Ph.D. thesis, The University of Texas, Austin, (1982).
16. Schlepp, D., "A High Performance Dehumidifier for Solar Desiccant Cooling Systems," SERI/TP-252-1979, (1983).
17. Ingram, J.B., and Vliet, G.C., "Solid Desiccant Rotary Dryer Performance Charts," Progress in Solar Energy, 603-608, (1982).
18. Maclaine-cross, I.L., and Banks, P.J., "Coupled Heat and Mass Transfer in Regenerators—Prediction Using an Analogy with Heat Transfer," Int'l Journal Heat Mass Transfer, 15, 1225-1242, (1972).
19. Banks, P.J., "Prediction of Coupled Heat and Mass Transfer in Fluid Flow through a Porous Matrix by the Nonlinear Analogy Method," Submitted to the ASME Journal of Heat Transfer, (1983).
20. Shah, R.K., "Thermal Design Theory for Regenerators," Heat Exchangers: Thermal-Hydraulic Fundamentals and Design, edited by S. Kakac, A.E. Bergles, and F. Mayinger, Hemisphere Publishing Corp., New York, (1981).
21. Jurinak, J.J., and Mitchell, J.W. "Recirculation of Purged Flow in an Adiabatic Counterflow Rotary Dehumidifier," ASME Paper 83-HT-1, to be published in ASME Journal of Heat Transfer, (1984).
22. Jurinak, J.J., Mitchell, J.W., and Beckmann, W.A., "Open Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications," to be published in ASME Journal of Solar Energy Engineering, (1984).
23. Lunde, P., "Solar Desiccant Air Conditioning with Silica Gel," Proc. 2nd Workshop on the Use of Solar Energy for Cooling of Buildings, SAN/1122-76/2, 280, (1976).

24. Howe, R.R., Beckmann, W.A., and Mitchell, J.W., "Factors Influencing the Performance of Commercial Hybrid Desiccant Air Conditioning Systems," to be published in Proceedings of the 18th Intersociety Energy Conversion Engineering Conference, (1983).
25. Macriss, R.A., and Zawacki, T.S., "High COP Rotating Wheel Solid Desiccant System," Proc. 9th Energy Technology Conference, Washington, D.C., (1982).
26. Howe, R.R., "Model and Performance Characteristics of a Commercially-Sized Hybrid Air Conditioning System...", M.S. thesis, University of Wisconsin-Madison, (1983).
27. Schlepp, D., "SERI Test Standard: Methods of Testing for Rating Solar Assisted Desiccant Cooling Systems," SERI, (1979).
28. Berezin, I.S., and Zhidkov, N.P., Computing Methods, Addison Wesley Pub. Co., Reading, Mass., (1965).
29. Abbot, M.B., An Introduction to the Method of Characteristics, Thames and Hudson, (1966).
30. Lax, P., Hyperbolic Systems of Conservation Laws and the Mathematical Theory of Shockwaves, SIAM, Philadelphia, (1973).
31. Courant, R., and Friedrichs, K.O., Supersonic Flow and Shock Waves, Interscience, New York, (1948).
32. Kays, W.M. and London, A.L., "The Transient Response of Heat Exchangers," Compact Heat Exchangers, Second Edition, McGraw-Hill, New York, (1964).
33. Chorin, A.J., "Random Choice Solution of Hyperbolic Systems," Journal of Computational Physics, 22, 517-533, (1976).
34. Pesaran, A.A., and Mills, A.F., "Modeling of Solid Side Mass Transfer in Desiccant Particles," to be presented at ASME Conference, Las Vegas, (1984).
35. Allander, C.G., "Untersuchung des Adsorptionsvorganges in Adsorbentenschichten mit linearer Adsorptionsisotherme," K. Tekniska Högskolans Handlingar, 70, (1953).