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EXPERIMENTAL STUDY AND SIMULATION OF A
CONCURRENT-FLOW GRAIN DRYER

DISSERTATION

Presented in Partial Fulfillment of the Requirements for the Degree Doctor of Philosophy in the Graduate School of The Ohio State University

By


* * * * * *

The Ohio State University
1971

Approved by

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Studies in Heat Transfer. Professor Eric K. Johnson
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LIST OF SYMBOLS

\( k \) - drying constant, hr\(^{-1}\)

\( m \) - \( m(t, r) \) local kernel moisture concentration, dry basis

\( m_a \) - \( m_a(t) \) moisture concentration of the drying air, dry basis

\( \bar{m} \) - \( \bar{m}(t) \) average kernel moisture concentration, dry basis

\( \bar{m}_s \) - surface moisture concentration of the kernel, dry basis

\( r \) - radial position inside the kernel, ft

\( t \) - drying time, hr

\( y \) - kernel location from the drying air inlet, ft

\( \rho \) - bulk density of dry grain, lb/ft\(^3\)

\( \tau \) - time constant of heat transfer, min

\( D \) - grain moisture diffusivity, ft\(^2\)/hr

\( \text{EFF} \) - drying efficiency

\( G \) - mass flow rate of dry air per unit area, lb/ft\(^2\) hr

\( H \) - mass flow rate of dry grain per unit area, lb/ft\(^2\) hr

\( L \) - latent heat of evaporation of grain moisture, BTU/lb

\( P_s \) - saturated vapor pressure of the drying air, lb/in\(^2\)

\( R \) - kernel radius, ft

\( \text{RH} \) - relative humidity of the drying air

\( S_a \) - specific heat of dry air, BTU/lb °F

\( S_g \) - specific heat of dry grain, BTU/lb °F

\( S_v \) - specific heat of water vapor, BTU/lb °F

\( S_w \) - specific heat of water, BTU/lb °F

\( T_a \) - \( T_a(t) \) drying air temperature, °F

\( T_e \) - equilibrium air temperature, °F

\( T_g \) - \( T_g(t) \) grain temperature, °F

\( T_{wb} \) - \( T_{wb}(t) \) drying air wet bulb temperature, °F
Heated air grain dryers are used on many farms and at most grain elevators in the Midwest. Modern corn harvesting equipment (shellers and combines) have a high harvest rate and place a heavy seasonal load upon farm and elevator drying systems. Attempts to speed up drying usually result in decreased grain quality due to uneven drying.

Continuous flow dryers were developed to maintain a more uniform final moisture while drying at a high rate. These dryers introduce one more variable, the grain flow rate, which makes them more difficult to operate efficiently than fixed bed dryers. Yet, the control of dryers, especially continuous flow dryers, is still an art and requires an experienced operator who functions as an intermittent feedback and, maybe, feedforward, loop in what would otherwise be an open loop system. An automatic control system for continuous flow dryers is very desirable.

The design of an automatic control system for a dryer poses some important questions. What is the variable, or variables, that could best be controlled (air temperature, air flow rate, grain flow rate, etc.)? What is the variable, or variables, that could best be sensed (grain moisture, air temperature, air relative humidity, etc.), and where in the drying bed, to give the best control of the final grain "quality"? And what is grain quality?
An experimental answer to these questions would be extremely expensive and time consuming to obtain. It would require investigating the effect of the potential control variables (air temperature, air flow rate, and grain flow rate) on the temperature and moisture profiles of both grain and air. It would also require identifying a variable, or variables, that could be easily sensed and is as close as possible to the air inlet to minimize lag time, and defining its relationship to final grain "quality". These tests would have to cover a wide range of input variables (initial grain temperature and moisture, air relative humidity, etc.). An accurate mathematical model of such a dryer could provide the answer much easier and faster if it could be programmed on a computer such that each drying test might be simulated in less than 0.0001 of the actual preparation and drying time and at comparable savings in cost.

The purpose of this study is to derive an accurate and reliable model of concurrent-flow dryers, program the model on the computer, and utilize the computer results in a preliminary study of possible automatic control systems for the dryer.
REVIEW OF LITERATURE

Many studies have been made to describe the drying process in such products as grain. Some studies were concerned with thin-layer drying models, while others involved the extension of thin-layer models to deep-bed drying.

Early thin-layer studies by Hukill (1954) and Allen (1960) utilized the exponential drying equation which assumes the resistance to moisture movement is at the grain surface. Thus the drying rate is proportional to the amount of moisture to be removed, or

\[ \frac{\text{dm}}{\text{dt}} = -k(m - \eta) . \]

The solution to this differential equation is

\[ \frac{m - m_\text{e}}{m_\text{o} - m_\text{e}} = \exp(-kt) . \]

Babbitt (1949) was among the first to apply the diffusion equation to describe moisture movement in an agricultural crop. His work centered around describing the rate of moisture desorption in wheat. Hustrulid and Flikke (1959) treated a corn kernel as a homogeneous, isotropic sphere and assumed that its moisture distribution was symmetrical with respect to its center and moisture movement was described by diffusion. Assuming a constant diffusivity, moisture movement was described by the following...
equation:
\[
\frac{\partial m}{\partial t} = D \frac{\partial}{\partial r} \left( r^2 \frac{\partial m}{\partial r} \right).
\]

Crank (1964) derived the solution to this equation for a uniform initial moisture distribution, \(m_o\), and a constant surface moisture, \(m_s\), in equilibrium with a fixed environment

\[
\frac{m(r,t) - m_o}{m_s - m_o} = 1 + \sum_{n=1}^{\infty} \frac{(-1)^n}{n} \sin \frac{n\pi r}{R} e^{-Dn^2 \pi^2 t / a^2}.
\]

Integrating this expression over the volume of the sphere and rearranging, the average moisture content of the sphere is given by

\[
\frac{m_s - m_s}{m_o - m_s} = 6 \sum_{n=1}^{\infty} \frac{1}{n^2} e^{-n^2 k t}
\]

where \(k = \frac{D \pi^2}{k^2}\).

Pabis and Henderson (1962) treated the corn kernel as a homogeneous, isotropic brick 2S thick, 2W wide, by 2L long. They used the diffusion equation in rectangular coordinates with a constant diffusivity to describe moisture movement

\[
\frac{\partial m}{\partial t} = \left( \frac{\partial^2 m}{\partial x^2} + \frac{\partial^2 m}{\partial y^2} + \frac{\partial^2 m}{\partial z^2} \right)
\]

where \(m = f(t,x,y,z)\). The solution to this equation with a uniform initial moisture and a constant surface moisture was derived, integrated, and rearranged to obtain the average moisture content

\[
\frac{m_s - m_s}{m_o - m_s} = \frac{512}{6} \sum_{l=0}^{\infty} \sum_{m=0}^{\infty} \sum_{n=0}^{\infty} \beta e^{-\sigma t}
\]

where
They compared this solution with a solution based on a temperature dependent diffusivity and found latter solution to give better moisture prediction. Although the three-dimensional diffusion equation as applied was quite satisfactory, they recommended the solution for the kernel shaped as a sphere because of its simplicity.

Thompson et al. (1968) developed an empirical model based on a second order exponential function to describe thin-layer drying with satisfactory results.

Most of the early work in thin-layer studies were concerned with only the mass transfer in the drying process. Little attention was given to the heat transfer involved in thin-layer studies. However, some studies were made. Wang and Hall (1961) solved the simultaneous partial differential equations describing heat and vapor transfer in a homogeneous, isotropic, symmetrical sphere with a uniform initial moisture and drying constant under constant boundary conditions. Both thermal diffusivity and mass diffusivity were assumed constant. Young (1969) solved similar equations under the same boundary conditions and initial conditions, but assumed the mass diffusivity to be linearly dependent upon the moisture content and the temperature, and the thermal conductivity to be linearly dependent upon moisture. He examined the effect of mass and thermal diffusivities and combined them in a modified Lewis
number which could be used to determine the significance of the heat transfer equation in drying problems. He proposed that the temperature gradient becomes insignificant for a modified Lewis number greater than 60, thereby allowing the heat transfer equation to be dropped (kernel temperature = air temperature) without any serious errors.

Henderson and Pabis (1961) examined the temperature effect on the drying constant and proposed the following relation

\[ -\frac{f}{T} \]

\[ k = \frac{d}{e} \]

where \( d \) and \( f \) are constants related to the grain and \( T \) is its absolute temperature.

Chittenden and Hustrulid (1966) studied the drying constant for shelled corn using the model of Hustrulid and Flikke (1959) and concluded that the constant is a function of drying air temperature, and kernel moisture concentration. They also noted that the surface moisture content giving the best fit was not the moisture content in equilibrium with the environment and that its value depended upon the initial moisture.

Chu and Hustrulid (1968) examined the diffusivity of shelled corn and developed the following expression

\[ D(c,T) = 1.5134 \exp \left\{ \left[ 0.00045 T - 0.05485 \right] \right\} \]

where \( T = \) absolute temperature °K and \( c \) is the local moisture content in percent dry basis. They also found that a surface moisture content higher than the equilibrium moisture content gave a better prediction.
Hamdy and Barre (1969) suggested the presence of a stagnant film layer of air around the kernel, which would account for a surface moisture higher than the moisture in equilibrium with the drying air. They developed a method for evaluating the film coefficient and the mass diffusivity.

Early application of thin-layer drying models to deep-bed analyses was limited to Hukill's equation (1954) for a stationary bed.

\[
\frac{\partial m}{\partial t} = \frac{G S}{\rho L} \frac{\partial T}{\partial y}
\]

Hukill also gave the following solutions to this equation.

\[
\begin{align*}
\frac{m - m_e}{m_e} &= \frac{c y}{e^{c y + e^{k t} - 1}} \\
\frac{T - T_e}{T_0 - T_e} &= \frac{e^{k t}}{e^{c y + e^{k t} - 1}}
\end{align*}
\]

where

\[
c = \frac{k_p L (m - m_e)}{G S a (T - T_e)}
\]

Implicit in the solutions was a unique relationship between \(T\) and \(m\), independent of the stated partial differential equation. Baughman et al. (1970), proposed an independent relation between \(m\) and \(T\) based upon experimental observations.
\[
\frac{\partial m_y}{\partial t} = -\frac{G S_a(T_o - T_e)}{\rho L(m_o - m_y)} \frac{\partial m_y}{\partial y}
\]

where \( m_y \) is the grain moisture at the air inlet. Barre et al. (1970) used both Hukill's and Baughman's equations to derive analytically the solutions offered by Hukill. They also developed expressions for drying rates, efficiencies, and the average moisture ratio.

Extension of other thin-layer models were limited severely by the lack of analytical solutions to the differential equations under variable boundary conditions. With the aid of computers, numerical solutions to these equations became available and deep-bed drying models were developed further.

Boyce (1965) and Henderson and Henderson (1968) extended the exponential thin-layer model to deep-bed drying.

Thompson et al. (1968) applied the second order exponential thin-layer model to describe counter-flow, concurrent-flow, and cross-flow dryers and also compared their performance. They concluded that the counter-flow dryer was the most efficient dryer with the highest capacity, but at the expense of decreased grain quality due to the exposure of the drier grain to the higher air temperature. The concurrent flow dryer was less efficient but produced higher quality grain for a given input air temperature.

Hamdy and Barre (1970) simulated deep-bed drying on the hybrid computer. They employed a comprehensive, highly non-linear mathematical model plus the diffusion thin-layer model to obtain close agreement of the observed moistures and temperatures with the model.
Bakker-Arkema et al. (1969) also simulated a similar model on the digital computer to describe drying in a multiple zone dryer, also with good results.

In 1966, Aguilar and Boyce studied different temperature ratios for measuring the drying efficiency and for dryer control. They concluded that a ratio of the sensible heat used in drying to the sensible heat in the drying air gave more reliable results than using exhaust air temperature.

In 1963, Matthews used a capacitance-type moisture meter to provide a signal to adjust the grain flow rate in a cross-flow dryer. The moisture meter required extensive calibration, and was very sensitive to temperature fluctuations.

Zachariah and Isaacs (1966) simulated moisture control systems for a continuous cross-flow dryer on the digital computer. They compared the control systems by using a performance index defined as

\[ PI = \int T_s e^2 r \, dt \]

where \( T_s \) = the settling time for the response to a step change, 
\( e \) = error in final moisture content, and
\( r \) = rate of grain discharge bu/hr.

Agness and Isaacs (1966) examined exhaust air temperature sensing for controlling grain drying and concluded that the relation between the exhaust air temperature and the average moisture content was dependent upon the following: initial drying air temperature,
the air flow rate, and the initial grain moisture. Changes in these variables required manual correction of the system.

Thin-layer drying studies have been made and the results indicate that diffusion models give better predictions than the logarithmic model. Much of the diffusion work assumes the kernel to be a homogeneous, isotropic sphere, while very little attention has been given to other diffusion models or to any comparison between them. The spherical diffusion model has also been extended to deep-bed drying with the aid of computers and has given quite good predictions.

Different attempts have been made to use deep-bed models in developing dryer control systems, but application has been limited without an accurate, reliable moisture sensor. Dryer controls, time devices, and exhaust air or drying air temperatures related to grain moisture are used for automatic control. The latter is heavily influenced by other drying variables and must be supplemented by manual control. A definite need exists to define this relation in a more sound manner and to examine the effect of the drying variables on this relation. In so doing one should be able to examine the possibility of sensing temperatures elsewhere in the drying bed.
OBJECTIVES

The objectives of this study were to:

1. Develop a mathematical model to describe concurrent-flow drying.
2. Develop a computer program to solve the model.
3. Examine the effects of different initial grain moistures, initial drying air temperatures, grain flow rates, and air flow rates to determine those with the most influence on drying.
THEORETICAL ANALYSIS

Grain drying consists of two processes occurring simultaneously, the transfer of sensible heat energy from the drying air to the grain and the movement of moisture from the grain to the air. In a concurrent-flow dryer these processes occur while the air and the grain are moving in the same direction. This study applies the drying model developed by Iamdy and Barre (1970) to a concurrent-flow dryer.

Kernel Model

The heat energy transferred into the kernel heats the kernel and evaporates some of its moisture. Most of the thin layer studies have focused on the transfer of moisture rather than heat. This may be justified by looking at the Lewis number, or a modification of it as did Young (1969). The Lewis number is defined as the ratio of the thermal diffusivity to the mass diffusivity. Young concluded that for a modified Lewis number greater than 60, the temperature gradient within a body was insignificant compared to the mass concentration gradient. The ratio of the drying time constant, which is inversely proportional to mass diffusivity, to the heating time constant provides the same information. Typical values for shelled corn are 252 min and 1.74 min (Pabis and Henderson, 1962), respectively, and yield a Lewis number of 145. Thus a simpler heat transfer model assuming no temperature gradient
in the kernel, such as derived by Pabis and Henderson (1962) may be used:

\[
(S_g + S_w) \frac{dT}{dt} = -\frac{1}{T_g} S_g (T - T_a) + L \frac{dm}{dt}.
\]

Radial diffusion in a sphere was assumed to describe the moisture movement within the kernel. The following equations were presented by Crank (1964).

\[
\frac{\partial m}{\partial t} = \frac{1}{r^2} \frac{\partial}{\partial r} \left[ r^2 \frac{\partial m_r}{\partial r} \right]
\]

\[
m_g = \frac{3}{R^3} \int_0^R m_r^2 dr.
\]

Equations (1) and (2) are based on the following assumptions:

1. The kernel is assumed to be a homogeneous, isotropic, symmetrical, solid sphere.

2. Resistance to heat transfer lies at the kernel surface (no temperature gradient within the kernel).

3. The rate of heat transfer is proportional to the difference between the air and kernel temperatures; the constant of proportionality being the reciprocal of the time constant of heat transfer.

Deep-Bed Model

**Transient Operation**

Heat and moisture exchange between the air and the grain satisfies energy and mass conservation laws. The rates of heat
lost and moisture gained by the air as it passes through an infinitesimally thin layer of grain in fixed-bed or cross-flow dryers were described by Hamdy and Barre (1970) as follows:

\[ G(S_{a} + m_{a} S_{v}) \frac{\partial E}{\partial y} = - \rho (S_{g} + m_{g} S_{w}) \frac{\partial E}{\partial t} + \rho [L + S_{v}(T_{a} - T_{g})] \frac{\partial E}{\partial t} \]

\[ \frac{\partial m}{\partial y} = - \rho \frac{\partial E}{\partial t} . \]

The equations of heat and moisture exchange between air and grain in a concurrent-flow dryer or counter-flow dryer could be similarly derived as follows:

\[ G(S_{a} + m_{a} S_{v}) \frac{\partial E}{\partial y} + H[(S_{g} + m_{g} S_{w}) \frac{\partial E}{\partial y} - S_{v} \frac{\partial E}{\partial y} (T_{a} - T_{g})] = \rho L \frac{\partial E}{\partial t} \]

\[ \frac{\partial m}{\partial y} + H \frac{\partial E}{\partial y} = 0 . \]

The grain flow rate is positive for concurrent-flow dryers and negative for counter-flow dryers.

**Steady-State Operation**

If a concurrent-flow dryer or a counter-flow dryer is operating in steady-state, and \( t \) is limited to the time a layer is exposed to drying air through its displacement \( y \) from the air inlet, one has

\[ H = \rho \frac{\partial y}{\partial t} . \quad (3) \]

Substituting into the above equations, the following model for steady-state operation can be obtained
Equation (3) implies that y is no longer a coordinate but has become a variable dependent upon t. The number of independent variables would be reduced to two: t and r. The displacement y may be calculated by integrating equation (3) as follows:

\[ y = \int_{0}^{t} \frac{H}{\rho} \, dt . \]  

**Boundary Conditions**

Steady-state operation implies that the temperature and moisture of both air and grain are constant at the dryer inlet, and that the air and grain flow rates are also constant. The initial moisture concentration in the kernel (at the dryer inlet) was assumed uniform. The kernel moisture profile was assumed to have zero gradient at its center (due to symmetry) and to be in equilibrium with the drying air at its surface. The boundary conditions can be written as
The equilibrium moisture $m_e$ was calculated for the air temperature and relative humidity at the kernel surface. The equations of Smith (1947), Henderson (1952), and Strohman and Yoerger (1967) were considered. The equation of Strohman and Yoerger (1967), which is based on the data of Rodriguez-Arias et al. (1963) over a wide range (temperatures from 40 deg F to 140 deg F, and relative humidities from 15 to 80 percent), was selected.

$$\text{RH} = \frac{P}{P_s} = \exp \left( ae^{e} \ln P_s + ce^{e} \right)$$

(8)

where, for shelled corn, $a = 0.8953$, $b = -0.1232$, $c = -5.482$, and $d = -0.1917$.

The constants $b$ and $d$ were not equal and equation (8) had to be solved by iterative methods during the simulation.

The expression for saturated vapor pressure by Brooker (1967) was tried and found inaccurate at higher temperature. Saturated vapor pressure data for air at various temperatures from Keenan
and Keys (1936) were stored in a function generator. The relative humidity was then calculated by

\[ \text{RH} = \frac{14.696}{(0.6219 + \frac{m_a}{P_s})} \]

(9)

Parameters

The time constant of heat transfer into a kernel may be defined after Kreith (1958)

\[ \tau = \frac{\rho_k c V}{h A_s} \]

where \( A_s \) = kernel surface area,
\( c \) = kernel specific heat,
\( \rho_k \) = kernel density,
\( h \) = convective heat transfer coefficient, and
\( V \) = kernel volume.

Henderson and Pabis (1962) examined the inverse of \( \tau \) and described \( c \) and \( \rho_k \) as functions of the kernel moisture. The expression could then be written as

\[ \tau = \frac{\rho_p (1 + m_a)(S + m_a S_w)}{A_s h} V \]

where \( \rho_p \) = dry kernel density. They also gave a procedure for calculating \( h \) for a single, isolated kernel heated under free convection. Noting that \( \rho_p V \) is constant (kernel mass), assuming that \( h \) does not change with drying, and estimating \( A_s \) as inversely proportional to \( \rho^{0.667} \), it could be shown that \( \tau \) decreases by 27 percent when the
grain moisture content decreases from 35 to 20 percent, dry basis. This is based on an increase of 21 percent in ρ (Miles, 1937). In this study τ was assumed constant and taken as 1.74 min. The effect of the 27 percent decrease in τ is insignificant as will be shown later.

The mass diffusivity was found to be dependent upon the kernel temperature and moisture by Chittenden and Hustrulid (1966). Chu and Hustrulid (1968) developed a model for the diffusivity dependent upon both kernel temperature and local kernel moisture. Because of the simulation complexities introduced by the model dependency upon local kernel moisture, the model developed by Henderson and Pabis (1961) which assumed the mass diffusivity to depend upon temperature only was used

$$\frac{D}{R^2} = A \exp \left(\frac{-5023}{T^8 + 459.69}\right)$$

(10)

where A is obtained from the drying tests. Since the temperature within the kernel was assumed uniform, it follows then that the mass diffusivity would also be uniform within the kernel.

The latent heat of vaporization of grain moisture has been found to be dependent upon both the kernel moisture and temperature by Rodriguez-Arias et al. (1963). Chung and Pfost (1969) presented a model to describe the energy of desorption, but limited application to just a few grain temperatures. For this study, the data of Rodriguez-Arias et al. (1963) was stored in a function generator. To correct for other temperatures, the average decrease in latent
heat per deg F was calculated from their data and found equal to 0.692 BTU/lb H2O deg F. The corrected latent heat of vaporization of grain moisture can then be expressed as

$$L = L_{\text{generated}} - (T_g - 80) \times 0.692$$

(11)

The bulk density of shelled corn has been shown by Miles (1937), Patterson et al. (1969), and Gustafson and Hall (1970) to be dependent upon the grain moisture. In the absence of an analytical model, the data presented by Miles for a grain moisture range of 0.1111 d.b. to 0.6667 d.b. was placed in a function generator for use during simulation.

The specific heat of the grain was assumed constant and taken from Pabis and Henderson (1962) equal to 0.2 BTU/lb d.m. deg F.

The values for the specific heat of dry air, water, and water vapor were taken from the Thermodynamics by Obert and Gaggioli (1963), and are 0.24, 1.0, and 0.45 BTU/lb deg F, respectively.

It was desirable to calculate the wet-bulb temperature and the drying efficiency even though not necessary for the simulation. The wet-bulb temperature was calculated according to Brooker (1967). The drying efficiency at any point in the drying bed was calculated by

$$\text{EFF} = \frac{T_a(o) - T_a(t)}{T_a(o) - T_{wb}(o)}$$

(12)
Computer Simulation

The mathematical model is not readily solvable and requires a large analog computer or a digital computer for simulation. The digital computer was chosen because a large analog was not available, and the availability of the CSMP-360 (Continuous System Modeling Program) at The Ohio State University Instructional and Research Computer Center allows one to concentrate upon the model being simulated rather than the numerical techniques associated with programming.

The model has two independent variables, time and position within the kernel and could not be simulated on the computer (using CSMP), which has only one independent variable. The method developed by Hamdy and Barre (1969) was used to eliminate $r$, the position within the kernel. This method eliminates $r$ by dividing the sphere into 10 spherical shells of equal thickness with a node at the mean radius of each shell, and an unknown moisture concentration associated with the shell node. The moisture profile within each shell was assumed to be a parabola. The three coefficients of each parabola were calculated such that the parabola was consistent with the moisture concentrations associated with the shell node and the adjacent nodes on each side. Since the innermost and the outermost parabolas lacked one adjacent node each, the boundary conditions (equations (7)) were used to provide one equation for each. The innermost parabola was calculated such that its gradient vanished at the sphere center, and the outermost parabola was calculated such that it gave a moisture concentration equal to $m_e$ at the
surface. The method transformed the boundary value problem of grain
drying into an initial value problem. The kernel mass transfer
(equation (2)) transformed into the following set of ordinary differ­
etial equations.

\[
\frac{dm_0}{dt} = \frac{100 D}{R^2} (3m_1 - 3m_0)
\]

\[
\text{shells } i=1,2, \ldots, 8
\]

\[
\frac{d m_i}{dt} = \frac{100 D}{R^2} \left( \frac{2i+3}{2i+1} m_{i+1} - 2m_i + \frac{2i-1}{2i+1} m_{i-1} \right)
\]

\[
\text{outermost shell}
\]

\[
\frac{d m_9}{dt} = \frac{100 D}{R^2} \left( \frac{56}{19} m_e - \frac{80}{19} m_g + \frac{24}{19} m_8 \right)
\]

Equations (13) consists of 10 equations describing mass transfer
within the kernel. Since the position variable \( r \) has been elimi­
nated, equations (1), (4), and (5) can be written as ordinary dif­
fferential equations. Rearranging,

\[
\frac{dT_g}{dt} = \frac{-\frac{1}{T_g} S_g (T - T_g) + \frac{1}{T_a} S_a \frac{d m_g}{dt}}{S_g + m_g S_w}
\]

(14)

\[
\frac{dT_a}{dt} = \frac{(T - T_a) \left( \frac{1}{T_g} S_g - \frac{1}{T_v} S_v \frac{d m_g}{dt} \right) H_a}{S_a + m_a S_v}
\]

(15)
\[
\frac{dm_a}{dt} = - \left( \frac{H}{C} \right) \frac{dR}{dt}.
\] (16)

Equations (6) through (16) can now be simulated on the digital computer. Appendix A contains a list of the CSMP statements used to solve them.

The computer program took less than 2 seconds to simulate a 2 hour test; thus it was possible to obtain many simulations in just a few minutes.
EXPERIMENTAL STUDIES

The experimental part of this study involved several concurrent-flow drying tests of shelled corn from the 1970 crop. The equipment for conditioning the air is similar to that used by Whitaker et al. (1969), Singh (1970), and Baughman et al. (1970).

Experimental Equipment

A concurrent-flow dryer with a cross section of 1 square foot and a bed depth of 2 feet was built and connected to the air conditioner. The drying air entered the grain column through two inverted V-shaped air ducts near the top and was exhausted at the grain chute as shown in Figures 1 and 2. A pneumatic vibrator at the outlet controlled the grain flow rate and prevented "bridging" of the moist grain in the dryer while allowing a continuous flow.

The dry and wet bulb temperatures of the drying air and temperatures every 4 inches in the bed were monitored and recorded to the nearest deg F by a 20-point Leeds and Northrup potentiometer with copper-constantan thermocouples as sensors. In addition to the initial and final grain moisture samples, a special screen wire cage 2 x 2 x 24 inches (see Figure 3) filled with undried grain was inserted at the top of the drying bed and allowed to progress through the bed until it spanned the depth. It was removed and samples for moisture were taken and oven dried at 212 deg F until no more moisture loss could be detected by weighing. This usually
required about 24 hours. The air flow through the drying bed was
determined by measuring the pressure drop across a calibrated
multiple orifice steel sheet with 101-5/16 inch holes. The grain
flow was determined by periodic weighings of the discharge of
dried grain from the dryer.

Experimental Procedure

Before a drying test a batch of shelled corn was rewetted in
a mixing drum and stored in sealed plastic containers for 48 hours.
Rewetted shelled corn was found to dry similar to, but slightly
faster than naturally moist shelled corn by Hustrulid (1962).
Before starting a test the shelled corn was mixed further to obtain
as uniform a mix as possible. Samples for moisture were taken as
the hopper above the drying bed was being filled. It was filled
a second time when the wire cage was inserted.

To establish steady operating conditions the dryer was
operated 4 to 6 hours, after which the wet shelled corn was placed
in the storage hopper. The drying test was begun by starting the
pneumatic vibrator. The grain flow rate was steadied and moni-
tored frequently during the tests.

When the air temperatures at the inlet and in the drying bed
became steady, the wire cage was inserted at the top of the bed and
allowed to progress with the other grain until it was removed and
divided into 12 equal length sections for moisture determination.

After several drying tests it became apparent that a uniform
temperature did not exist across the bed because of the effects of
a two duct system. Thermocouples were mounted in the bed directly
Figure 1. Experimental Concurrent-Flow Dryer
Figure 2. Schematic Diagram of Concurrent-Flow Grain Dryer
Figure 3. Sampling Cage and Sectioning Fork
beneath the ducts and also half way between them at 0.5 foot and 2.0 feet below them. The observed temperatures are shown in Figure 4. According to studies by Ives et al. (1959) on air flow from ducts into grain, parallel and uniform air flow did not occur until the air had traveled a distance from the ducts equal to one-half the center to center distance between the ducts. Therefore, the temperature at the same level but midway between the ducts will be less than those directly under the ducts. To relate temperature measurements in the bed to the moisture samples, the air temperature measurements were made in the same location from which the moisture samples were taken. The results of the experimental tests are given in Appendix B.
Figure 4. Observed Temperatures at 0.5 and 2.0 Feet Below the Air Ducts.
RESULTS AND DISCUSSION

The mathematical model previously developed was simulated on the computer and compared with the experimental results of drying tests 10 and 11 (Figures 5 and 6). The agreement between measured and computed moisture profiles is quite good, except near the air inlet. The slight initial deviation was probably due to heat transfer through the uninsulated air ducts and the wire cage used for collecting grain samples to determine the moisture profile. This could have raised the grain temperature before drying started (at y=0) such that its initial temperature was higher than that used in the computer simulation, thereby resulting in a higher initial mass diffusivity and producing a higher initial drying rate than computer predictions.

The agreement between measured and computed air temperature profiles (Figures 5 and 6) is also good except near the air inlet. Experimental errors seem to have been responsible for this initial deviation. First, the non-uniform air flow rate produced lower temperatures at the dryer center (where the wire cage was moving) than directly below the air ducts. Second, the thermocouples used for temperature measurements were unshielded and may have sensed a temperature between those of the air and grain instead of the air temperature. Had this happened, the error in air temperature measurement would have been most serious where air and grain
Figure 5. Observed and Computed Air Temperature and Grain Moisture Profiles for Drying Test No. 10.
Figure 6. Observed and Computed Air Temperature and Grain Moisture Profiles for Drying Test No. 11.
temperatures differ the most, that is, near the air inlet. The indicated temperature would be less than the air temperature due to the lower grain temperature.

In discussing the model parameters the time constant of heat transfer, \( t \), was assumed equal to 1.74 min as calculated by Pabis and Henderson (1962) for free convection. To evaluate the significance of higher or lower values of \( t \), the model was simulated for values of \( t \) equal to 0.6, 1.74, and 4 min as shown in Figure 7. The value of \( t \) less than 1.74 had very little effect on the moisture and temperature profiles, while the larger value of \( t \) had a more pronounced effect. For the experimental drying conditions used, \( t \) equal to 1.74 min is probably too large since air flow rates of 410 and 444 lb/hr ft\(^2\) were used in tests 10 and 11, respectively, thereby creating forced, rather than free, convection.

A typical computer prediction of the air and grain temperature profiles and air and grain moisture profiles are shown in Figures 8 and 9, respectively. It should be noted that the model predicts kernel temperature profiles (Figure 8), thus it is possible to determine the maximum temperature the kernel attains and the length of time the kernel is above a certain temperature level during a drying simulation. This information would be quite valuable in helping to determine final grain quality.

To gain some insight into the steady-state performance of a concurrent-flow dryer, the model was simulated over a wide range of operating conditions, namely: initial grain moisture,
Figure 7. Grain Moisture and Air Temperature Profiles for Different Values of $\tau$. 

GRAIN MOISTURE

AIR TEMPERATURE

$\tau = 0.6 \text{ min}$

$1.74$

$4.0$

DEPTH (ft)

AIR TEMPERATURE (°F)

GRAIN MOISTURE (d.b.)

200

175

150

125

100

75

0

.50

.40

.30

.20

.10

0

.25

.50

.75

1.00

1.25

1.50

1.75

2.00

.50

.40

.30

.20

.10

0

.25

.50

.75

1.00

1.25

1.50

1.75

2.00
Figure 8. Computer Simulation-Air and Grain Temperature Profiles
Figure 9. Computer Simulation—Air and Grain Moisture Profiles
initial grain temperature, initial air moisture, initial air temperature, and air flow rate. Each of these parameters was varied individually for different grain flow rates from a set of reference operating conditions selected to dry a 2 foot deep bed of shelled corn at 52 deg F from 33.33 percent moisture, dry basis, using 322.2 lb/hr ft$^2$ of air at 250 deg F and 0.58 percent moisture, dry basis. Performance curves (Figures 10 through 16) show the effect of varying these parameters on the steady-state operation of the experimental concurrent-flow dryer.

The effects of variation of each of these parameters on the final grain moisture is shown in Figures 10 through 14. As was expected variation in initial grain temperature and initial air moisture (shown in Figures 10 and 11, respectively) had insignificant effects on the final grain moisture and they may be ignored.

The effect of variation of initial air temperature on the final grain moisture (Figure 12) is significant and linear thus indicating that it may be used to control final grain moisture.

Variation of the air flow rate (Figure 13) does produce changes in final grain moisture, however, rather large changes in air flow rate are required to produce changes in final grain moisture.

The effects of variation of initial grain moisture on the final grain moisture is shown in Figure 14, and as expected produced significant changes in the final grain moisture.
Figure 10. Variation of Final Grain Moisture with Initial Grain Temperature for Different Grain Flow Rates
Figure 11. Variation of Final Grain Moisture with Initial Air Moisture for Different Grain Flow Rates
Figure 12. Variation of Final Grain Moisture with Initial Air Temperature for Different Grain Flow Rates

- \( m_{g_0} = 33.33\% \text{ d.b.} \)
- \( G = 322.2 \text{ lb/hr ft}^2 \)
- \( H = 100 \text{ lb/hr ft}^2 \)
Figure 13. Variation of Final Grain Moisture with Air Flow Rate for Different Grain Flow Rates
Figure 14. Variation of Final Grain Moisture with Initial Grain Moisture for Different Grain Flow Rates
Variation of the inlet parameters (initial air temperature, air flow rate, and initial grain moisture) was shown to have significant effects on the final grain moisture for different grain flow rates. Fortunately, the initial air temperature, air flow rate, and grain flow rate can be controlled, while only the initial grain moisture cannot be controlled.

Variations of the initial grain moisture produce significant changes in the final air temperature as shown in Figure 15, but use of final air temperature to sense such changes introduces a large time lag. This time lag can be greatly reduced if the air temperature is sensed at a shallower depth. A depth of 0.5 foot was arbitrarily chosen since this is the shallowest depth at which the computer model gave good agreement with the experimental data. In Figure 16 the effects of variation of initial grain moisture on air temperature at 0.5 foot for different grain flow rates are shown to be significant. Thus, it is possible to use this temperature to sense changes in initial grain moisture.

It is possible (at constant initial air temperature and air flow rate) to develop dryer function curves which relate the final grain moisture to the initial grain moisture and to the air temperature at 0.5 foot and at 2 feet as shown in Figures 17 and 18, respectively.

These dryer function curves may be used in the feedback control system as the function which relates sensed air temperatures to the final grain moisture. This final grain moisture
Final Air Temperature (°F)

Figure 15. Variation of Final Air Temperature with Initial Grain Moisture for Different Grain Flow Rates

\[ T_{oo} = 250°F \]
\[ G = 322.2 \text{ lb/hr ft}^2 \]
\[ H = 20 \text{ lb/hr ft}^2 \]
Figure 16. Variation of Air Temperature at 0.5 Foot Depth with Initial Grain Moisture for Different Grain Flow Rates.

- $T_{ao} = 250^\circ F$
- $G = 322.2 \text{ lb/hr ft}^2$
- $H = 20 \text{ lb/hr ft}^2$

The graph shows the variation of air temperature at 0.5 feet depth with initial grain moisture (d.b.) for different grain flow rates.
Figure 17. Dryer Function Curves Relating Final Grain Moisture to Grain Flow Rate and Air Temperature at 0.5 Foot for Different Initial Grain Moistures
Figure 18. Dryer Function Curves Relating Final Grain Moisture to Grain Flow Rate and Final Air Temperature for Different Initial Grain Moistures
may then be compared with a reference final moisture to generate an error signal to activate the dryer control system (Figure 19). In this manner the dynamic model of the concurrent-flow dryer can be utilized to optimize the control system under transient conditions. When the dynamic model is verified and employed to study the transient operation of the dryer, intelligent, logical application of automatic control theory can be applied.
Figure 19. Automatic Feedback Control Diagram for a Concurrent-Flow Grain Dryer
SUMMARY AND CONCLUSIONS

An analytical model describing heat and mass transfer in a concurrent-flow grain drying was developed and simulated on the digital computer. Experimental tests were conducted to verify the steady-state simulation of the model.

The agreement of the observed with the computed air temperatures and grain moistures was good except for the initial drying near the inlet. This deviation near the inlet was due to the heat transfer through the uninsulated ducts, the high conductivity of the wire cage, the non-uniform air flow of the ducts, and the use of unshielded thermocouples. The effect of each of the inlet conditions upon the performance of the drying system were analyzed and dryer performance curves were developed.

The results of this concurrent-flow drying study warrant the following conclusions:

1. The analytical model developed in this study gives a good description of the heat and mass transfer processes involved concurrent-flow drying.

2. The results of the analytical study show that the relations between the inlet conditions and the temperature are well defined.

3. The temperature and moisture profiles are related to the final moisture content.
4. Initial grain temperature and initial air moisture have an insignificant effect on the final grain moisture.

5. Dryer function curves can be developed which relate final grain moisture to air temperatures sensed in the drying bed for constant air temperature and air flow rate in a steady-state concurrent-flow dryer.
RECOMMENDATIONS FOR FUTURE STUDIES

The results of this study indicate the need for further research in the following areas.

1. Further experimental verification of the transient response of the concurrent-flow drying model.

2. Further study into temperature measurements near the air inlet system and also the effect of duct spacing on the temperature measurements.

3. The development of a rapid, accurate, continuous operation moisture meter capable of monitoring grain moisture at any point in the dryer.

4. The development of models or procedures for calculating the heat transfer time constant for deep-bed drying.

5. Detailed studies into the types of automatic controls best suited for grain dryers.
APPENDIXES
Appendix A. CSMP-360 Concurrent-Flow Drying Program
CONCURRENT-FLOW DRYING MODEL

INITIAL
INCON TAO = 250.0, MAO = 0.006, TGO = 52.0, MGO = 0.3333
PARAMETER T = 1.72, H = 56.0, C = 322.2
CONSTANT SA=0.2405, SV=0.448, SG=0.20, SV=1.0, IC=-3.0, ERR=0.1,
TWB = 108.0
HG = H/G

FUNCTION CURVE7=0.0, 1451.0, 0.10, 1372.0, 0.12, 1290.0, 0.14, 1221.0,
0.16, 1160.0, 0.18, 1147.0, 0.22, 1129.0, 0.26, 1123.0, 0.3, 1120.0,
0.34, 1120.0, 0.38, 1120.0, 0.4, 1120.0, 0.45, 1120.0

FUNCTION CURVE8=40.0, 0.1217, 42.0, 0.1315, 44.0, 0.1419, 46.0, 0.15323,
48.0, 0.16525, 50.0, 0.17811, 52.0, 0.19182, 54.0, 0.20642, 56.0, 0.222,
58.0, 0.2336, 60.0, 0.2563, 62.0, 0.2751, 64.0, 0.2951, 66.0, 0.3164,
68.0, 0.330, 70.0, 0.3631, 72.0, 0.3886, 74.0, 0.4156, 76.0, 0.4443,
78.0, 0.4747, 80.0, 0.5049, 82.0, 0.5410, 84.0, 0.5771, 86.0, 0.6157,
88.0, 0.6556, 90.0, 0.6982, 92.0, 0.7432, 94.0, 0.7908, 96.0, 0.8407,
98.0, 0.8935, 100.0, 0.0492, 102.0, 0.1078, 104.0, 0.1695, 106.0, 0.1345,
108.0, 1.2029, 110.0, 1.2748, 112.0, 1.3504, 114.0, 1.4298, 116.0, 1.5130,
118.0, 1.6006, 120.0, 1.6804, 122.0, 1.8808, 124.0, 1.8897, 126.0, 1.9995,
128.0, 2.1064, 130.0, 2.2272, 132.0, 2.3440, 134.0, 2.4712, 136.0, 2.6042,
138.0, 2.7432, 140.0, 2.888, 142.0, 3.0404, 144.0, 3.199, 146.0, 3.365,
148.0, 3.537, 150.0, 3.711, 152.0, 3.906, 154.0, 4.102, 156.0, 4.306,
160.0, 4.471, 162.0, 5.212, 164.0, 5.721, 166.0, 6.273, 168.0, 6.868,
170.0, 7.510, 172.0, 8.202, 174.0, 8.946, 176.0, 10.605,
178.0, 11.526, 180.0, 12.512, 182.0, 13.568, 184.0, 14.606, 186.0, 15.901,
188.0, 17.186, 190.0, 18.557, 192.0, 20.016, 194.0, 21.567, 196.0, 23.217,
FUNCTION CURVE = \[ 0.111, 41.32, 123.5, 40.797, 136.3, 40.12, 14.94, 2.5, 35.36, 16.27, 3.1, 7.44, 17.64, 37.94, 19.04, 37.22, 8.7, 26.84, 24.0, 45.36, 71.95, 35.68, 64.2, 23.45, 34.27, 25.34, 176, 26.58, 33.43, 28.2, 32.69, 76, 29.87, 3.7, 0.2, 31.57, 31.43, 36.33, 30.84, 35.13, 30.31, 04, 36.98, 29.78, 41.3, 38.88, 29.31, 4.5, 28.40, 28.85, 44.82, 25, 79, 964, 47.05, 29.63, 52, 4.52, 27.28, 28.51, 51.51, 26.98, 53.4, 26.67, 56.25, 76, 4.19, 2.5, 58.73, 26.15, 76, 61.29, 25.8, 12, 63.0, 25.62, 66.66, 25.392 \]

DYNAMIC

\[
\begin{align*}
DR &= 33000 \times F(Y) - 5023 \times ( \text{TG} - 45.69) \\
M1DOT &= DR \times (3.0 \times M1 - 3.0 \times M0) \\
M1DOT &= DR \times (1.6667 \times M2 - 2.0 \times M1 + 0.3333 \times M0) \\
M2DOT &= DR \times (1.40 \times M3 - 2.0 \times M2 + 0.60 \times M1) \\
M3DOT &= DR \times (1.27 \times M4 - 2.0 \times M3 + 0.71 \times M2) \\
M4DOT &= DR \times (1.22 \times M5 - 2.0 \times M4 + 0.7777 \times M3) \\
M5DOT &= DR \times (1.18 \times M6 - 2.0 \times M5 + 0.81 \times M4) \\
M6DOT &= DR \times (1.15 \times M7 - 2.0 \times M6 + 0.84 \times M5) \\
M7DOT &= DR \times (1.13 \times M8 - 2.0 \times M7 + 0.86 \times M6) \\
M8DOT &= DR \times (1.11 \times M9 - 2.0 \times M8 + 0.88 \times M7) \\
M9DOT &= DR \times (1.07 \times M10 - 2.0 \times M9 + 0.90 \times M8) \\
MGDOT &= -0.0078 \times M7 + 0.1320 \times M8 - 0.95 \times M9 + 0.8316 \times M8 \\
M1 = \text{INTGRL}(M0, M1DOT) \\
M2 = \text{INTGRL}(M0, M2DOT) \\
M3 = \text{INTGRL}(M0, M3DOT)
\end{align*}
\]
M4=INTGRL(MCO,M4DOT)
M5=INTGRL(MCO,M5DOT)
M6=INTGRL(MCO,M6DOT)
M7=INTGRL(MCO,M7DOT)
M8=INTGRL(MCO,M8DOT)
M9=INTGRL(MCO,M9DOT)
MG=0.00075*M0+0.00675*M1+0.01875*M2+0.03675*M3+0.06075*M4
+0.09075*M5+0.12675*M6+0.16875*M7+0.207825*M8+0.282175*M9

L1=NLFGEN(CURVE7,MG)
L=L-(TG-EO.0)*0.602
MA = HG*(MGO-MC)+MA0
TGDOT = (L*MCDOT-SG*(TG-TA)*60./T)/(SG+MG*SW)
TADOT = HG*(TG-TA)*60./T-SV*MGDOT)/(SA+MA*SV)
TG=INTGRL(TCO,TGDOT)
TA=INTGRL(TAO,TADOT)
PS=NLFGEN(CURVER,TA)
RH=14.696*MA/(0.6219+MA)/PS
LP = ALOG(PS)
LRH = ALOG(RH)

STE = IMPL(JC, ERR, FSTE)
EM1 = EXP(-0.1232*STE)
EM2 = EXP(-0.1017*STE)
FM = 0.8893*FM1*LP-5.482*EM2-LRH
FMDOT = -0.1103*FM1*LP+1.0509*EM2
FSTF= STE-FM/FMDOT
DSTE=STE/100.0
PROCEDURE MS = EMMY(DSTE)
    IF (DSTE) 1,1,2
        1 MS = 0.0
        GO TO 2
    2 MS = DSTE
    3 CONTINUE
FNDPRO

R = NLFGEN(CURVE, MG)
DEPTH = INTEGRAL(0.0, H/R)
PV = RH*PS
TDB = TA+459.6*

PROCEDURE TWB = WTBLB(PV, TDB)
    PSWB = NLFGEN(CURVE, TWB)
    PSF = (PSWB-PV)/(PSWB-14.696)
    FTWB = TWB-(TDR+3506.8525*PSF)/(1.0+1.4736*PSF)+459.69
    IF (FTWB) 71, 72, 73

71 TWB = TWB+0.1
    PPSWB = NLFGEN(CURVE, TWB)
    PPSF = (PPSWB-PV)/(PPSWB-14.696)
    FPTWB = TWB-(TDR+3506.8525*PPSF)/(1.0+1.4736*PPSF)+459.69
    IF (FPTWB .GE. 0.0) GO TO 72
    GO TO 71

73 TWB = TWB-0.1
    PMSTW = NLFGEN(CURVE, TWB)
    PMSF = (PMSTW-PV)/(PMSTW-14.696)
    FMTWB = TWB-(TDR+3506.8525*PMSF)/(1.0+1.4736*PMSF)+459.69
    IF (FMTWB .LE. 0.0) GO TO 72
    GO TO 73
72 CONTINUE
FNDPRO

EFF = (TAC-7A)/(TAC-TW0)
PROCEDURE TW0 = ITW0LB(TW)
   IF (TIME .GT. 0.0) GO TO 74
   TW0 = TW
FNDPRO

TERMINAL
TIMER DELT=1.0E-11, PRDFL=0.05, FINTIM=2.0, DELMIN=1.0E-12
FINISH DEPTH=2.0
PRINT DEPTH, TA, MA, RH, TWB, TG, MG, EFF, HG
END
STOP
ENDJOB
Appendix B. Experimental Test Data
DRYING TEST NO. 10

\[ T_a(o) = 182.0 \degree F \]
\[ T_g(o) = 77 \degree F \]
\[ m_a(o) = 0.0068 \ d.b. \]
\[ m_g(o) = 0.3367 \ d.b. \]
\[ G = 409 \ lb/hr \ ft^2 \]
\[ H = 52 \ lb/hr \ ft^2 \]
\[ \text{Static Pressure} = 1.00 \ \text{in. H}_2\text{O} \]
\[ \rho = 35.5 \ \text{lb/ft}^3 \]

### Table 1

<table>
<thead>
<tr>
<th>Temperature Profile</th>
<th>Moisture Profile</th>
</tr>
</thead>
<tbody>
<tr>
<td>Depth (in.)</td>
<td>Temperature (( ^\circ F ))</td>
</tr>
<tr>
<td>0 (inlet)</td>
<td>182.0</td>
</tr>
<tr>
<td>2&quot;</td>
<td>121.5</td>
</tr>
<tr>
<td>6&quot;</td>
<td>115.5</td>
</tr>
<tr>
<td>10&quot;</td>
<td>110.1</td>
</tr>
<tr>
<td>14&quot;</td>
<td>105.9</td>
</tr>
<tr>
<td>18&quot;</td>
<td>103.0</td>
</tr>
<tr>
<td>22&quot;</td>
<td>101.4</td>
</tr>
<tr>
<td>24&quot; (exhaust)</td>
<td>101.3</td>
</tr>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
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<td></td>
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<tr>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Temperature directly under the inlet ducts
\[ T (6") = 139.1 \degree F \]
\[ T (24" = \text{exhaust}) = 108.0 \degree F \]

Ambient Conditions
\[ T_{d.b.} = 81 \degree F, \]
\[ T_{wb} = 61 \degree F \]
DRYING TEST NO. 11

\[ T_a(o) = 180^\circ F \]
\[ T_g(o) = 70.0^\circ F \]
\[ m_a(o) = 0.0046 \text{ d.b.} \]
\[ m_g(o) = 0.3287 \text{ d.b.} \]
\[ G = 445 \text{ lb/hr ft}^2 \]
\[ H = 46.0 \text{ lb/hr ft}^2 \]
\[ \text{Static Pressure} = 1.153 \text{ in. H}_2\text{O} \]
\[ \rho = 38.7 \text{ lb/ft}^3 \]

Table 2

<table>
<thead>
<tr>
<th>Depth (in.)</th>
<th>Temperature (°F)</th>
<th>Moisture (d.b.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 (inlet)</td>
<td>180.0</td>
<td>0.2444</td>
</tr>
<tr>
<td>2&quot;</td>
<td>122.8</td>
<td>0.2287</td>
</tr>
<tr>
<td>6&quot;</td>
<td>118.4</td>
<td>0.2099</td>
</tr>
<tr>
<td>10&quot;</td>
<td>114.0</td>
<td>0.2009</td>
</tr>
<tr>
<td>14&quot;</td>
<td>108.8</td>
<td>0.1951</td>
</tr>
<tr>
<td>18&quot;</td>
<td>104.6</td>
<td>0.1877</td>
</tr>
<tr>
<td>22&quot;</td>
<td>102.6</td>
<td>0.1870</td>
</tr>
<tr>
<td>24&quot; (exhaust)</td>
<td>101.7</td>
<td>0.1836</td>
</tr>
<tr>
<td>16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>18</td>
<td></td>
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<tr>
<td>20</td>
<td></td>
<td></td>
</tr>
<tr>
<td>22</td>
<td></td>
<td></td>
</tr>
<tr>
<td>24 (exhaust)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Temperature directly under the inlet ducts

\[ T (6") = 145.8 \]
\[ T (24" = exhaust) = 105.3 \]

Ambient Conditions

\[ T_{d.b.} = 81 \]
\[ T_{wb} = 57.5 \]

\*During drying test No. 11 thermocouples measuring inlet air temperature failed. An upstream thermocouple measuring air temperature before the ducts was used. This thermocouple averaged 183°F during the test. Previous tests under similar conditions showed a 2 to 4°F temperature drop was experienced until the air reached the ducts.*
BIBLIOGRAPHY


