MODEL DEVELOPMENT AND ECONOMIC EVALUATION OF A SENSIBLE HEAT STORAGE UNIT UTILIZED IN A SOLAR-DEHUMIDIFICATION LUMBER DRYING SYSTEM

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ABSTRACT

A mathematical model for a rock-bed energy storage system has been developed for a solar-dehumidification lumber dryer and shown to give good correlation with experimental data. The drying rate of yellow poplar lumber was mathematically related to the collector area and storage size. A simple economic model was used to compare solar, solar-dehumidification, dehumidification, and wood-fired boiler drying systems.

Keywords: Solar energy, wood drying, heat pump, drying model, simulation.

NOMENCLATURE

A = bed cross-sectional flow area
A_c = collector surface area
$\overset{*}{A}_c$ = collector surface area per unit wood volume
A_s = heat transfer surface area
Bi = Biot number = $hR^2/(3k(1 - e))$
c_p = specific heat
D = rock diameter
e = void fraction
G = mass flow rate per unit cross sectional area
h = fluid-to-rock volumetric heat transfer coefficient
h_{amb} = heat transfer coefficient for bed-to-ambient air
k = rock thermal conductivity
L = bed length
$\dot{m}$ = mass flow rate
MC = moisture content (oven-dry basis)
N = number of rock-bed layers = $L/\Delta x$
NTU = number of transfer units = $hAL/(\dot{m}c_p)_f$
NTU* = $(U\Delta A_s)i/(\dot{m}C_p)_f$
NTUC = corrected number of transfer units (equation 12)

ΔP = pressure drop across the rock-bed
R = rock radius
Re = Reynolds number = $\dot{m}D/(A\mu)$
R_w = wood drying resistance
t = time
t^* = $t(\dot{m}c_p)_f/[(\rho c_p)_b(1 - e)AL]$
T = temperature
T^+ = temperature at the new time
U = overall heat transfer coefficient
V = rock-bed volume
$\overset{*}{V}$ = rock-bed volume per unit wood volume
W = humidity ratio of the kiln air (kg-water vapor/kg-dry air)
W_s = saturated humidity ratio at the wet-bulb temperature (kg-water vapor/kg-dry air)
x = distance in the rock-bed from the inlet

Greek symbols

ρ = density
μ = viscosity
Δ = finite change
η = factor, equation 5 and 6

Subscripts

amb = ambient conditions
b = rock conditions
f = fluid conditions
i = layer "i" of the rock bed
in = inlet conditions
w = water

INTRODUCTION

Lumber to be used for most wood products requires drying before it can be manufactured into a stable finished product that will perform satisfactorily. Drying is the most energy-consuming step in lumber processing. If solar energy could be used to dry lumber, a clean, inexpensive, and inexhaustible energy source would be harnessed. However, solar kilns require longer drying times than conventional kilns because of limited kiln temperatures, which are generally less than 60 C. Other factors such as lack of heat supply at night or during cloudy weather and unpredictable drying conditions as a result of changes in the weather also affect the operation of solar kilns.

Another technique currently being used by the lumber industry to reduce the energy cost of drying wood is dehumidification with heat pumps. Moisture in the kiln air is removed in the evaporator instead of during venting as in a conventional kiln. This reduces energy loss from the kiln. Although they consume less energy, dehumidification kilns require about the same amount of energy costs as conventional kilns, but their initial capital cost can be lower.

While there have been several theoretical models presented in the literature for the individual components (e.g., collector, storage, kiln, etc.), none has included the subsystems of a dehumidification unit and energy storage in the complete system model. The purpose of this paper is to present a computer simulation model for a combined solar-dehumidification wood drying system with a heat storage unit to achieve a better initial heat-up of wood and to assist and control the drying schedule. This simulation model can be used in the design and optimization of specific solar-dehumidification kilns. A simple economic model is also used to compare solar-dehumidification drying to other drying methods.

LITERATURE REVIEW

Solar wood drying systems

Even though there is a current energy glut, long-term problems of energy availability and costs will not go away. There is still significant interest in the use of solar energy for drying lumber. Although much research has been done, commercial application of solar drying has been limited. The relatively small capacities of solar kilns developed to date make them impractical for most commercial use in the United States; however, solar kilns can be practical for developing countries.

Successful operation of a number of such units has been reported in the literature. However, only the articles dealing with solar wood drying systems that have included energy storage in their research are discussed here. A good summary paper that outlines the improvements in solar wood drying is presented by Hall et al. (1981) and includes an annotated bibliography.

Read et al. (1974) have studied a solar kiln with the solar collector separate from the kiln and a rockpile thermal storage unit below the kiln. This made it possible to store additional thermal energy during the day and use it at night, thus reducing the effect of daily temperature differences and providing better drying conditions. Drying 2.5 cm alpine ash (*Eucalyptus delegatensis*) required an 80% increase in drying time compared to a conventional steam-heated kiln. They concluded that the rock bed should be isolated from the wood charge to be more effective. Their economics showed that at current back-up fuel prices, their solar system was only marginally effective.

Chen and Helton (1985) used black painted concrete blocks in a passive type solar kiln to augment the drying of yellow poplar lumber. Little (1984) has reported experimental work done on a commercial size lumber dryer 240 m^3 (100,000 board feet) that uses solar energy to augment the heat supplied from natural gas. Water-filled solar collectors (232 m^2) heat an 18,170-liter (water) energy storage tank that heats the kiln through an air-to-water heat exchanger. After some initial operational problems were solved, the system seemed to perform adequately. No detailed economic comparison was performed in their research. Other work describing solar-dehumidification drying enhanced by an energy storage has not been found in the literature.

Dehumidification wood drying systems

Dehumidification wood drying systems, though used in Europe and Asia for a number of years, are now used in the United States. Current dehumidification systems are able to heat up to 100 C and thus dry at a rate comparable to conventional steam kilns. Wood-drying quality is at least as good as in conventional kilns, and it appears that capital costs are less than steam kilns with operating costs about the same (since expensive electrical energy is used to drive the system). Several articles in the literature compared dehumidification with other drying systems (Wengert 1980; Milota and Wilson 1984; Lee and Harris 1984).

One source (Hogan et al. 1976) has developed a detailed computer code that calculates the heat transfer and thermodynamic states of the refrigerant at various points in the cycle of the dehumidifier. While this approach is very exact, it can be very expensive in terms of computer time. It is usually more convenient to curve-fit the output (moisture removal rate, compressor power, etc.) as a function of kiln operating conditions (usually kiln dry-bulb and wet-bulb temperature) (Helmer et al. 1982). This is the technique used in the present model.

Mathematical models of wood dryers

Several detailed mathematical models for kilns have been found in the literature. D. J. Close (1975) has presented the simulation of a solar timber dryer with a rock-bed storage system. Experimental data were used to establish reasonable values for such kiln and timber parameters as the collector area, collector inclination to the horizontal, kiln size and insulation, storage size, storage material properties, and power ratings of the circulating fan and auxiliary electric heaters. The system operated in the following manner: The dryer was made to follow a set schedule of kiln airdry- and wet-bulb temperatures. If the kiln temperature dropped 5 C below this set schedule, then heat was added to the kiln from the collector through the flow of hot air. If the kiln temperature rose above the scheduled value, air from the collector was passed through the storage unit. If the kiln temperature dropped by more than 2 C below the scheduled value, then heat would be added through the electrical heaters and not from the storage unit. The storage unit would add heat only if the kiln temperature was maintained at a satisfactory level. The reasons why such control strategy was used are unclear.

The solar radiation input was modelled by assuming a sine function with half period the same as the day length and with the integrated input over the period equal to the measured input. An hourly record of total insolation was made from a pyranometer for a period of twenty days. Ambient humidity was assumed constant, but recorded ambient dry-bulb temperature variations were input to the program.

The comparison of simulation results with the experimental data showed that the program could not predict the level, as well as the diurnal variations in the kiln dry-bulb temperature. Unfortunately, discrepancies of the order of 7 C to 10 C beween the simulation and the experimental results for the kiln temperatures were reported, indicating that the general results of their simulation study are of questionable value.

Duffie and Close (1978) have presented a simulation model that was used for determining the optimized design of a solar timber dryer equipped with a gravel bed and an adsorbent bed. The annual costs of systems using oil, solar energy, and electricity as the energy source were estimated at $919, $1,088, and $1,459, respectively. The adsorbent storage system proved to be superior to the gravel bed with all the auxiliary energy sources and collector designs.

PaLancz (1984) developed a similar simulation program for a solar-dehumidification dryer system but had no data to compare with his model. Wengert et al. (1984) have recently developed a solar kiln model for use on a microcomputer.

MODEL DEVELOPMENT

Helmer et al. (1982) have developed a solar-dehumidification kiln model. Knowing the ambient weather conditions, the dehumidifier parameters, the kiln geometry, the type of solar collector, and the initial conditions of the wood and air, the change in air humidity and air dry-bulb temperature can be calculated at any time period. The model also calculates wood moisture content, air temperature and humidity, and electrical energy consumptions of compressor and blowers for any time period. Their experimental results agreed well with their computer simulation. This model forms the base for the simulation model presented here. Only the changes in the computer model will be mentioned.

The drying rate equation was slightly modified as indicated in the equation to use a driving potential based on the difference between the humidity ratio saturated at the kiln wet-bulb temperature and the humidity ratio evaluated at the kiln conditions:

$$\dot{m}_w = (W_s - W)/R_w \tag{1}$$

where

$$R_w = -0.00360 + 0.342/MC \tag{2}$$

Figure 1 illustrates the drying rate correlation used in the program. This equation is in a simpler form than previously used and yet it shows good comparison to the previous drying data for yellow poplar extracted from the work presented by Chen and Helmer (1984).

Sensible thermal storage systems

Thermal storage systems, in general, involve storing of energy either as a sensible heat in liquid or solid medium, or as heat of fusion in selected phase change materials. Air-heated collectors, used mostly for wood drying, typically have rock-beds as sensible energy storage systems.

The optimum diameter of the pebbles may vary from 1.2 to 7.6 cm. Uniform diameters give more uniform air flow passages. Rocks with smaller diameters provide more surface area to the air flow, so they are more efficient in terms of heat transfer between the rocks and the air flow.

Static pressure drop through rock storage depends on the velocity, rock-bed length, void fraction, rock surface area factor, and rock size. The fan energy consumption is, of course, greatly dependent upon the air flow pressure drop. Larger rock sizes minimize pressure drops; but their centers are not used to store heat. So a compromise is sought for the rock size since rocks should be small enough to affect good heat transfer but large enough to minimize pressure drop.

Numerous storage models have been reported in the literature. The reader is referred to Schmidt and Willmott (1981) for an excellent reference book on the subject.

Though rock-beds are widely used for thermal storage in conjunction with solar energy systems, necessary data for determining design parameters such as bed dimensions, rock size, and air flow rates are available only in the form of rules of thumb; and even when available, these experimental data are applicable only for the specific situations and experimental conditions. Though a few closed-form analytical solutions exist for heat transfer processes inside a rock-bed, their use-

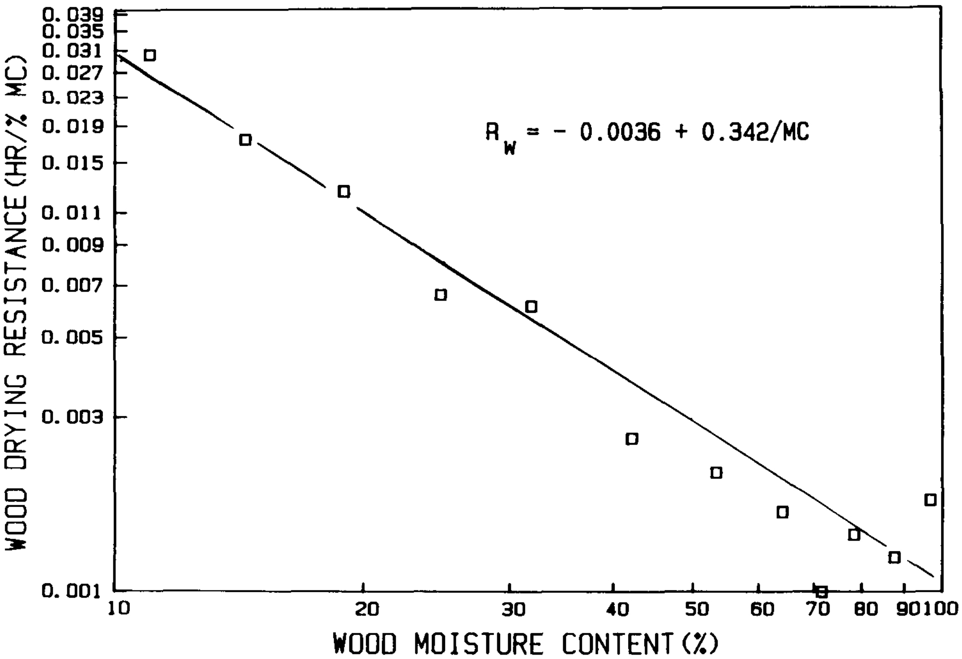

FIG. 1. Yellow poplar wood drying resistance variation with moisture content.

fulness for innovative designs involving variable flow rates, rock sizes, and circulation patterns is limited to simple unrealistic boundary conditions. But a numerical solution obtained by solving the basic equations makes it possible to simulate various design parameters easily.

The model used in the present simulation was derived from work done by Hughes et al. (1976) and Mumma and Marvin (1976). In this method of modelling, the rock-bed is assumed to be divided in N layers, each of thickness Δx, as shown in Fig. 2. The rock-bed temperature is assumed to be uniform in any layer. The air velocity through the bed is also assumed constant. The air temperature inside the bed is assumed to have an exponential temperature profile in the direction of flow.

Heat exchanger theory applied to this bed layer gives the air temperature variation in the direction of flow,

$$\frac{T_{f,i+1} - T_{b,i}}{T_{f,i} - T_{b,i}} = \mathrm{Exp}[-NTU(\Delta x/L)] \tag{3}$$

The energy added to the air then becomes

$$(\dot{m}c_p)_f(T_{f,i} - T_{f,i+1}) = (\dot{m}c_p)_f(T_{f,i} - T_{b,i})(1 - \mathrm{Exp}(-NTU/N)) \tag{4}$$

The bed temperature variation with time can be obtained from an energy balance on the rock-bed layer yielding

$$\frac{dT_{b,i}}{dt^*} = \eta N(T_{f,i} - T_{b,i}) \tag{5}$$

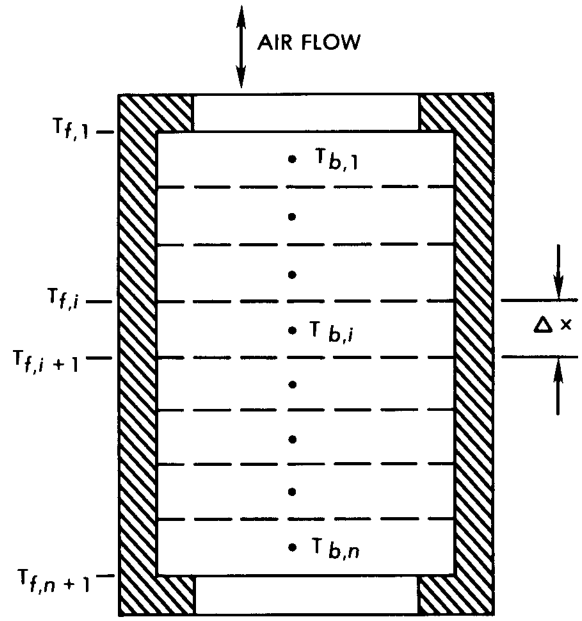

FIG. 2. Numerical node model for a rock-bed energy storage system.

where

$$\eta = 1 - \mathrm{Exp}(-\mathrm{NTU}/\mathrm{N}) \tag{6}$$

and

$$t^* = t(\dot{m}c_p)_f/[(\rho c_p)_b(1 - e)AL] \tag{7}$$

Including the heat loss to the environment results in, for layer "i",

$$\frac{dT_{b,i}}{dt^*} = N(T_{f,i} - T_{b,i}) + \{(UA_s)_i/(\dot{m}c_p)_f\}(T_{amb} - T_{b,i}) \tag{8}$$

where (UA) is the product of overall heat loss coefficient between the fluid and ambient multiplied by the heat loss surface area for node "i".

This equation can be solved by the Crank-Nicolson method as suggested by Hughes et al. (1976). Here, $(dT_{b,i})/(dt^*)$ is replaced by $(T^+_{b,i} - T_{b,i})/\Delta t^*$ and T_b on the right side of the equation is replaced by $(T^+_{b,i} + T_{b,i})/2$, thus giving the new bed temperature after a time period Δt^*;

$$T^+_{b,i} = \frac{\eta NT_{f,i} - NT_{b,i}/2 + NTU^*T_{amb} - NTU^*T_b/2}{1/\Delta t^* + \eta N/2 + NTU^*/2} \tag{9}$$

where

$$NTU^* = U\Delta A_s/(\dot{m}c_p)_f \tag{10}$$

Air temperatures can be calculated from rearranging Eq. (3) to yield:

$$T_{f,i+1} = T_{b,i} + T_{f,i}\mathrm{Exp}(-\mathrm{NTU}/\mathrm{N}) \tag{11}$$

At a given time t* knowing the bed layer temperature and fluid inlet temperature for that layer, the new bed layer temperature and fluid outlet temperature can be calculated. So starting at the first layer, all the new bed layer temperatures and fluid outlet temperatures can be calculated. Then the new bed temperatures can be calculated from Eq. (9) and the whole process repeated for new air temperatures in the rock-bed.

Though this model does not directly account for the temperature gradients within the pebbles, a slight correction in the value of NTU can relax this constraint on the model using

$$\mathrm{NTUC} = \mathrm{NTU}/(1 + \mathrm{Bi}/5) \tag{12}$$

The model was also modified to include the effects of internal conductive heat flow that takes place between rock layers.

The pressure drop across the bed determines the size of the blower motor required to move the air through the bed. The following equation for pressure drop is due to McCorquodale et al. (1978):

$$\Delta P = \frac{2LG^2(1 - e)}{\rho_{air} De^{3/2}} \left[4.74 + 332 \frac{(1 - e)}{e^{3/2}} \frac{\mu}{GD} \right] \tag{13}$$

Eshleman et al. (1977) have reported a reliable equation for the calculation of packed bed convective coefficients. The convective coefficient, h, can be predicted using

$$h = 2.88(1 - e)(k/D^2)\mathrm{Re}^{(0.00534D + 0.627)} \tag{14}$$

This equation was used for predicting the variable transfer coefficients for rock-bed storage model.

EXPERIMENTAL APPARATUS

A small scale solar-dehumidification lumber dryer system was built in Carterville, Illinois, to provide drying data to be compared with the results of the simulation model (Fig. 3). This kiln (1.2 m^3) was built to handle a charge smaller than industrial wood-drying kilns which are in the range of 50 m^3 to 200 m^3. The complete description of the kiln and supporting equipment can be found elsewhere (Helmer et al. 1982), and therefore the solar kiln will be only briefly described here.

Solar dehumidification kiln construction

The kiln itself measures 2.4 m by 2.4 m by 2.0 m high and is constructed of wood. The floor, walls, and ceiling are insulated with 5 cm of styrene foam and 15 cm of fiberglass. The 22.6 m^2 of double-glazed air solar collectors are site-built from aluminum beverage containers. Air flow rates through the collectors are set at about 20 liters/sec-m^2 and are circulated by two centrifugal blowers. A 0.75-kW (1-hp) motor drives the kiln fan that continuously circulates the air in the kiln. The high-temperature dehumidification system uses a 1.5-kW (2-hp) motor. A heat pipe heat exchanger is used in the dehumidification unit to improve the water extraction rate. This heat exchanger is placed in the inlet and outlet air flow paths of the dehumidifier to precool the kiln air going in to the evaporator as well as heat up the air leaving the condenser.

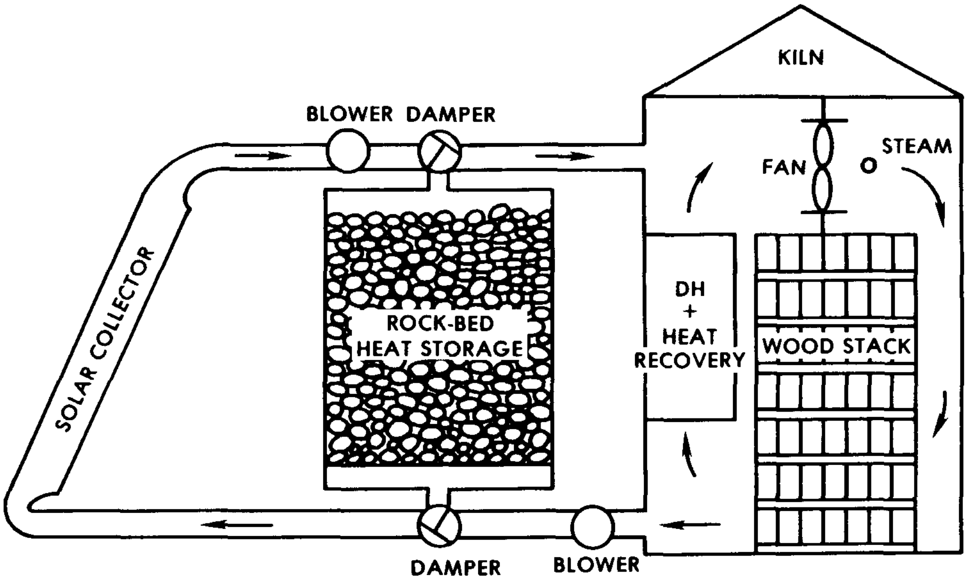

FIG. 3. Solar-dehumidification kiln with rock-bed energy storage system.

Heat storage system

The solar heat storage unit for the kiln used a rock-bed sensible heat storage system. Enough rock volume (3.6 m^3) was used to supply ènough heat to the kiln for one day's operation if no solar radiation was available (Fig. 4). A volumetric thermal capacity of 1.6 MJ/m^3-C was used in the design (assuming a 30% void fraction). The size would supply the kiln with 131,000 kJ of energy, if the heat storage was discharged from an initial temperature of 60 C to a final temperature of 38 C. The rock-bed was 1.2 m in the direction of flow, and the flow cross-section was 1.2 m by 2.4 m. The cross-sectional area was designed to give a pressure drop of 0.41 cm of water at 47.2 liters/sec of air flow. The walls of the storage chamber were made of 3.8 cm × 20.3 cm studs on 41-cm spacing with 1.9-cm plywood used as the inside wall with steel tie bolts used to reduce the side wall deflection. Twenty-three cm of fiberglass insulation and 1.2 cm of rigid polyurethane insulation board were used on the walls and bin top. Aluminum siding covered the outside of the storage chamber.

The weight of the rocks was supported by a layer of expanded metal lath on top of a series of bond beam blocks. The floor of the bin was covered with 7.5 cm of rigid insulation and a plastic vapor barrier under 1.9-cm-thick plywood. A 15-cm concrete slab supported the complete storage unit. The energy storage material, rocks, were obtained from washed river gravel, and ranged from 2.5 cm to 7.5 cm in diameter.

All air ducts were constructed from 20-cm-diameter sheet metal and were insulated with 3.8 cm of polyurethane foam. Two rectangular inlet ducts (20 cm by 122 cm) were located at the bottom side of each end of the storage bin. Two 46-cm by 46-cm ducts on the top of the storage unit were attached to two variable speed centrifugal blowers. These blowers were placed downstream from the solar collectors to reduce infiltration energy losses from the collectors. The air flow

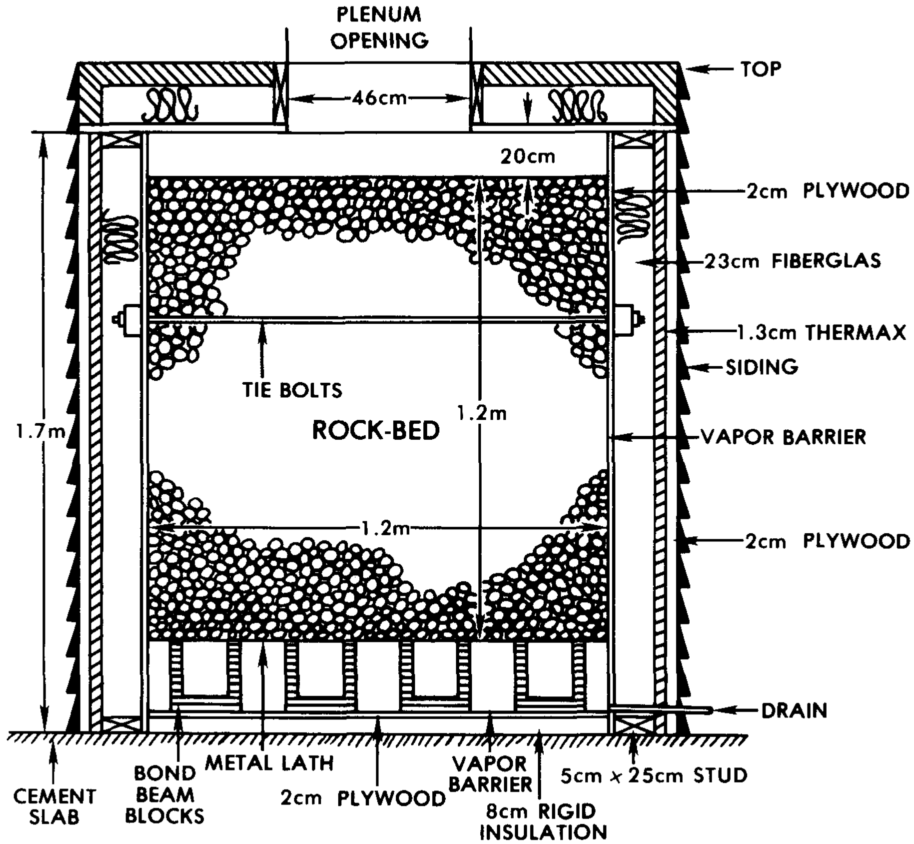

FIG. 4. Rock-bed energy storage system–side view.

from the kiln, collectors, and the heat storage unit was controlled by motorized dampers at the inlet and outlet of the storage bin. Copper-constantan thermocouples were installed at the bed heights of 33 cm, 66 cm, and 100 cm to measure and control the charging/discharging of the heat storage unit.

System control

The heating and cooling components in the wood drying system were controlled by two separate systems: one for the kiln/collector/storage system and one for the dehumidifier. The dehumidifier control was activated by a humidistat located in the kiln. When the relative humidity fell below the set value, the dehumidifier was turned off. The humidistat was gradually reduced from 80 to 25% relative humidity as the moisture content of the wood decreased.

A differential temperature controller activated the energy flow in the kiln/collector/storage system. The controller continuously monitored and compared the set temperature, the kiln dry-bulb temperature, the solar collector outlet temperature, and the top rock-bed storage temperature. It then activated the blowers and dampers on the basis of the differences in those temperatures. The control was as follows:

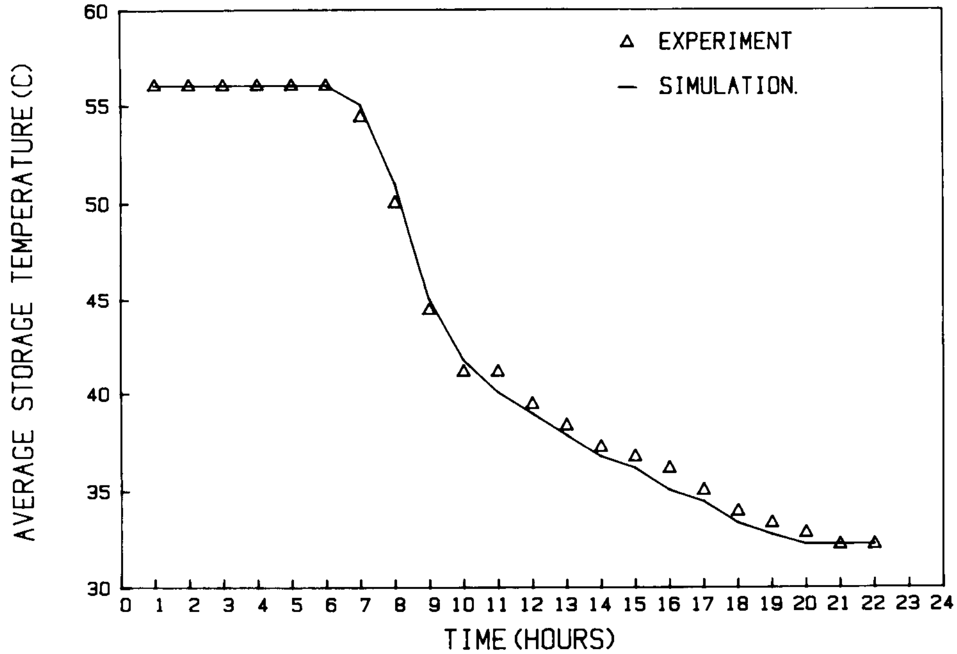

FIG. 5. Comparison of simulated and measured average rock-bed temperatures.

1. Heat was supplied to the kiln whenever the kiln dry-bulb temperature was below the set value.
   a. Heat to the kiln was first supplied by the collector.
   b. If the solar collector was not hot enough, then heat was supplied by the storage unit.
2. If the kiln dry-bulb temperature was above the set temperature and the solar collector was hotter than the storage temperature, then the solar collector was activated to heat the storage unit.

EXPERIMENTAL METHODS

The ambient and kiln temperatures (wet- and dry-bulb), inlet and outlet temperatures of the solar collector, and the top, center, and bottom temperatures of the rock-bed heat storage unit were recorded at 10-minute intervals. Solar radiation arriving perpendicular to the tilted collector surface was measured by a silicon pyranometer and recorded as instantaneous and integrated values.

Electric energy consumed by the compressor motor, the condenser, and evaporator blower was recorded by watt-hour meters. The electric current for the collector blowers, storage blowers, and the main fan were measured by an ammeter. The hours of operation of the fan and blowers were automatically recorded, and energy consumption was then calculated from these data.

Six boards were chosen for moisture samples and the weights of these boards were measured twice a day. Test runs were terminated when the average moisture content of the high three sample boards reached 8% moisture content. The lumber was conditioned approximately 6 to 8 hours with steam from a 7.5 kW, 11 kg-of-steam/hr electric steam boiler to relieve the drying stresses in the wood before

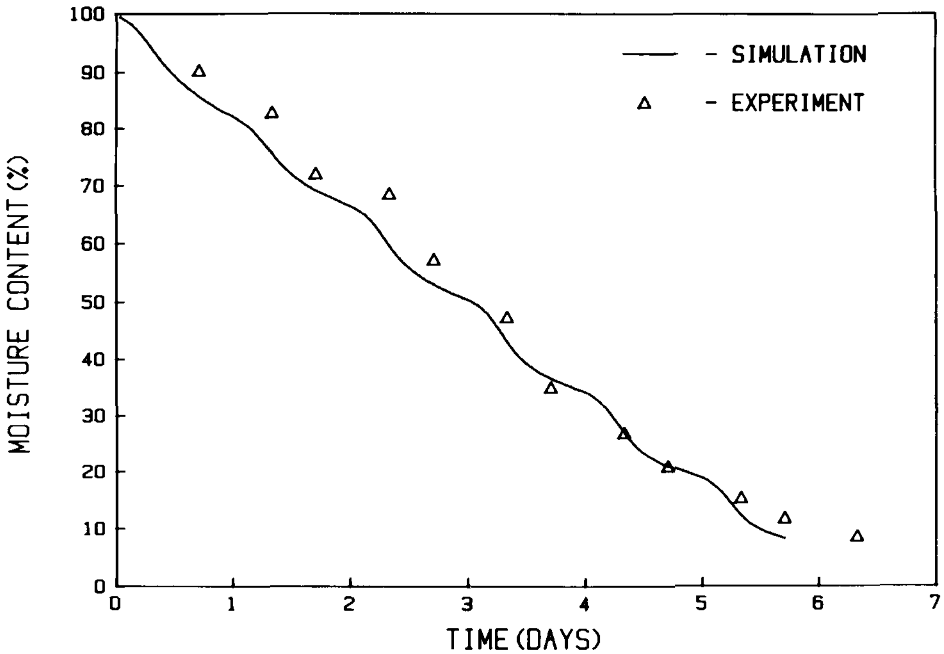

FIG. 6. Yellow poplar drying curve–experiment vs. computer simulation.

discharge from the kiln. At the end of the run the boards were measured for twist, bow, and crook as well as inspected for surface checking, end checking, and collapse.

Model verification

Experimental data from drying a charge of yellow poplar lumber were used to verify the performance of the system simulation model. Data were taken during July of 1983 to dry 1.2 m^2 (500 board feet) of 2.5-cm lumber (Chen and Helmer 1984). The green moisture content was 99.4%. The energy storage system was charged to 60 C prior to the run through the operation of the solar collector. All of these independent data (ambient temperatures, ambient solar radiation, initial kiln and storage conditions, as well as the kiln, storage, and collector characteristics) were used as input to the computer simulation model.

Figure 5 shows the comparison between the measured and simulated average rock-bed temperatures during the first day of drying operation. This figure illustrates the good correlation obtained by the model. Other days' results are similar, but unimportant, since the storage was not high enough in temperature to dry the wood. The experimental and computer results indicated that after about the third day the rock-bed remains idle because its average temperature is lower than the kiln temperature.

Table 1 is a summary of the comparison between the experimental drying data and the results of the computer simulation. Here it is seen that there is good correlation betweeen the kiln dry- and wet-bulb temperatures as well as the drying time. Figure 6 is a graphical comparison of the hourly computer simulation and the experimental drying data taken twice daily. The simulation predicts the same

TABLE 1. *Comparison of experimental solar dehumidification drying data for 2.5-cm yellow poplar lumber with the results of the computer simulation model.*

Parameter	Experimental	Simulation
Average kiln dry-bulb temperature (°C)	47.4	46.3
Average kiln wet-bulb temperature (°C)	41.6	43.4
Drying time (days)	6.3	5.7
Compressor energy consumption (kWh)	108	113
Rock bed blower energy consumption (kWh)	9.0	8.4

trend in diurnal drying rate variation as was measured experimentally. There is also good agreement in the drying curve and the overall drying time to 8% moisture content. The above comparison indicates that the computer model is adequate to predict the drying performance and energy consumption for this type of drying system.

SIMULATION RESULTS FOR DIFFERENT SIZE COMPONENTS

One of the questions that confronts the drying system contractor or builder is to determine the correct size of the components for the given amount and type of wood to be dried. The significant subsystems identified in the present system are the collector size and the energy storage size. Since the heat pump is not the main focus of the present work, its size was kept constant for the variation in parameters indicated below, the only parameters that were varied being the collector size and rock-bed volume.

Figure 7 shows the results of the simulation where the collector area was varied from 11.3 m^2 to 45.3 m^2 and the storage volume varied from 1.8 m^3 to 10.8 m^3. Figure 7 indicates that the storage size really has a small influence on the drying time. This is consistent with the fact that the storage energy supply was depleted after about the first day's operation. After that time the storage was not used. The average storage temperature was below the average kiln temperature, and therefore no energy from the storage could be added to the kiln.

Figure 7 also shows that there is a significant change in drying time with collector area; however beyond about 40 m^2 (or 80 m^2/1,000 board feet), there is little advantage to increasing the size of the collector. The effect of storage size and collector area was correlated according to the equation:

$$\text{Drying time (hours)} = 232.5 - 4.097A - 3.045V + 0.042A_c^2 + 0.0676V^2 + 0.0342(A_cV) \quad (15)$$

A first approximation for a general equation for the drying time as a function of collector area per unit of wood volume and storage volume per volume of wood yields

$$\text{Drying time (hours)} = 232.5 - 8.19\overset{*}{A} - 6.09\overset{*}{V} + 0.168\overset{*}{A}_c^2 + 0.2704\overset{*}{V}^2 + 0.1368(\overset{*}{A}_c\overset{*}{V}) \quad (16)$$

where

$\overset{*}{A}_c$ = collector area/wood volume (m^2/m^3)
$\overset{*}{V}$ = storage volume/wood volume (m^3/m^3)

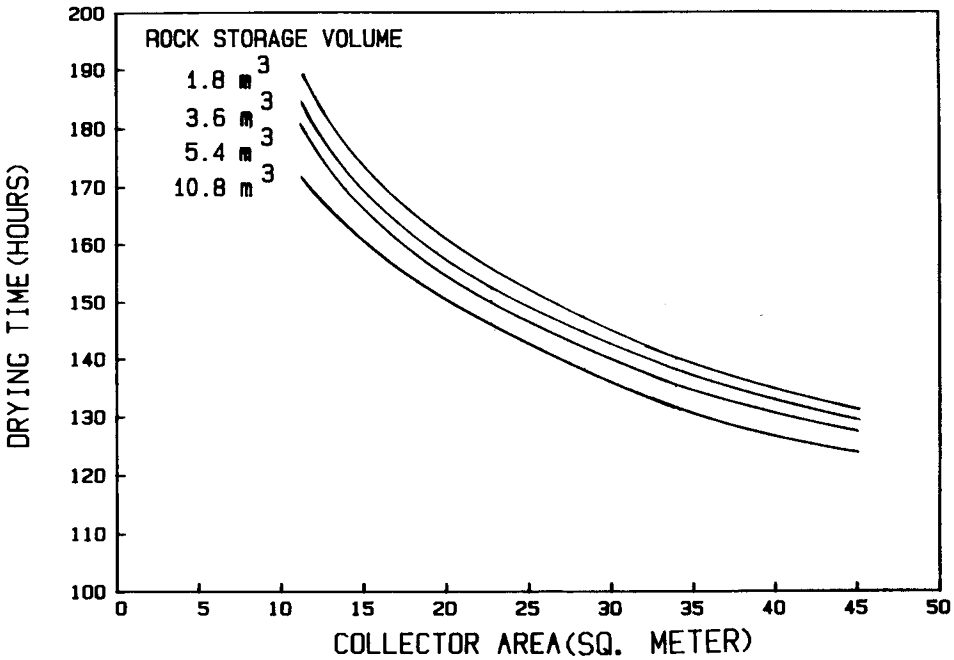

FIG. 7. Variation in drying time with collector area and storage volume.

ECONOMICS

Residential solar heating systems have been extensively analyzed according to various economic methods (e.g., rate-of-return, life-cycle costing, etc.). Unfortunately economic analyses for solar lumber drying systems are almost nonexistent in the literature, and the few that do exist usually include only a simple pay-back analysis that does not incorporate the time value of money.

In any solar system the increased initial cost of the equipment is balanced by the decreased fuel costs. The solar expenditure may or may not be economic, depending on the cost of the back-up fuel displaced, the escalation rate of the back-up fuel cost, the current interest rate for the money borrowed to finance the solar expenditure, the rate of increase (or decrease) in the interest rate, the expected life of the system, and other factors.

In the present economic analysis, a 120 m^3 (50,000 board feet) kiln drying 17 charges of yellow poplar lumber per year with fuel oil was considered. The kiln efficiency was assumed to be 50%. The capital costs were calculated for a system replacing a conventional steam boiler unit fired with fuel oil. Four types of replacement systems were considered: wood-fired boiler, dehumidification, dehumidification with solar, and solar heating only. The wood and fuel oil boilers had efficiencies that were assumed to be 65% and 82.5%, respectively. Incremental fuel costs were calculated on the difference between the fuel cost for the stated case and the fuel oil boiler system. Likewise, electrical operating costs were determined from the difference between the electrical operating costs of the given case and the fuel oil boiler system.

The wood-fired boiler had an initial cost of \$48,750 based on estimates of \$143/kg-steam-hour systems obtained from quotes of several manufacturers. The de-

TABLE 2. *Economic comparison of solar, solar-dehumidification, dehumidification, and wood-fired boiler drying systems assuming a 120-m³ (50,000 board feet) kiln.*

	System			
	Solar	Solar-D.H.	D.H.	Wood-fired boiler
Total net savings	\$74,217	\$491,857	\$909,496	\$1,772,157
Years until fuel savings equals investment	11	7	1	2

humidification capital costs were based on \$1/board foot capacity or a total of \$25,000 for dehumidification with solar, and \$50,000 for dehumidification alone. The solar costs were calculated on a solar collector area of 67.3 m^2 calculated from the sizing method by Helmer (1985) and using cost data from Mueller (1981) and a cost rate of \$393/square meter of collector area obtained from existing costs from a similar solar system in the field (Little 1984). This cost includes the cost of the collectors, ducting, controls, and general site construction. In the combined solar-dehumidification system, it was assumed that half of the required drying energy was supplied by each system.

The fuel escalation rate was assumed to be 10%/year. The interest rate was assumed to be 12%. The economic period over which the analysis was performed was 20 years. Fuel oil was assumed to cost \$0.40 per liter. Wood fuel was assumed to be available at \$8.00/ton. Electricity to power the dehumidifier and the fans and blowers for the solar kiln was estimated at 5 cents/kilowatt-hour.

The net total savings over the 20-year life for each system was calculated from the sum of the yearly results of:

$$\text{Net savings} = \underset{\text{savings}}{\text{total fuel}} - \underset{\text{costs}}{\text{electricity}} - \underset{\text{and interest}}{\text{loan principal}} \tag{17}$$

Table 2 presents the results. Here it is seen that the wood-fired system fared the best (in part because of low fuel cost compared to fuel oil). The solar system performed the poorest economically mainly because of the high capital cost of the collector. Adding the dehumidification system improved the economics of the solar system significantly. The second item in this table indicates the number of years it takes the total fuel savings (including escalation) to equal the capital expenditure (neglecting interest); this is like a "simple pay-back time." The wood-fired and dehumidification systems had a pay-back time of about 1 to 2 years, while the solar systems had pay-back times of 7 and 11 years.

CONCLUSIONS

A computer simulation model was developed that adequately simulates the performance of a solar-dehumidification lumber-drying system equipped with a rock-bed energy storage system. The rock-bed helped heat up the initial charge of lumber during the first day or two and was unutilized thereafter. The cost of the storage system is probably unjustified in comparing it to the decrease in drying time. Solar collector sizes beyond 80 m^2/1,000 board feet are probably not justified by the increase in drying performance. Without governmental tax credits, solar systems are not economically competitive with wood-fired boilers or dehumidification drying systems.

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