Improvement in Grain-Dryer Fuel Efficiency through Heat Recovery

Exhibit 2

#

F. S. Lai, G. H. Foster
ASSOC. MEMBER ASAE — FELLOW ASAE

ABSTRACT

GRAIN drying is an energy intensive operation from which useful heat is discarded. Recovery and reuse of part of this heat appears feasible. A heat-pipe heat exchanger, singly and in combination with a heat pump, was tested experimentally and by mathematical simulation as an approach to heat recovery and reducing energy requirements for grain drying.

INTRODUCTION

The rising costs and decreasing availability of fuels require that process designs incorporate energy economy. In many processes, useful energy is discarded as waste heat. One example is high-temperature grain drying. In the drying of grain, heat is neither gained nor lost, only a change of state is involved. Since the energy input for drying is relatively large, the recovery and use of at least part of the energy should reduce the total energy requirements of existing drying systems.

Efficiency of heat-energy usage has not been emphasized in the design of grain dryers. The performance of grain dryers has been measured largely in terms of drying capacity and operational reliability.

The present energy utilization for drying is relatively inefficient. As much as 25 percent of the energy consumed may be lost through ineffective practices (Rimberg, 1974). We should, therefore, consider the possibility that demands for energy may be moderated by improving the effectiveness of energy utilization without sacrificing the expected output from existing systems.

In recent studies, modified or optimized designs have been considered for reduction of energy requirements for high-temperature grain drying (Bakker-Arkema et al. 1974a; Lerew et al. 1972; Anderson, 1972; Converse, 1972; Morey and Cloud 1973; Roberts and Brooker, 1973; and Morey et al. 1974). In most of these studies, emphasis was on improving the efficiency of conventional grain dryers, rather than on developing new designs and concepts. Exceptions are the studies by Davis (1949), Shove (1953; 1968) and Flikke et al. (1957). They utilized the heat pump principle to recover and recycle the heat.

Davis (1949) studied theoretically the adaptability of the heat pump for continuous drying of shelled corn. Shove (1953) made a similar investigation on a laboratory batch grain dryer. Both indicated that the cost of operating the heat pump to condition air for grain drying would compete favorably with the cost of other methods of supplying energy for drying. However, the relatively high initial capital cost of the equipment may limit its applications. Flikke et al. (1957) extended the above studies to determine some of the economic factors involved in using this method of conditioning air for grain drying.

Reclaiming heat from the outlet air by recycling the air may also improve the energy efficiency of a dryer (Stephens and Thompson, 1975; Bakker-Arkema et al. 1974a). In the basic technique, which has been very successful, the drying system is enclosed so that all the air entering and leaving the dryer can be regulated. In comparing a conventional crossflow dryer with a modified recirculating-type crossflow dryer, Bakker-Arkema et al. (1974a) found that the dryer energy utilization index (total energy required by a dryer to remove 1 lb of moisture from the grain under a set of specified conditions) of a modified recirculating-type crossflow dryer was as much as 16 percent lower than that of a conventional crossflow dryer. Moreover, recirculation of air may improve the quality of the grain, since recirculation maintains higher humidities in the drying air than exist without recirculation, and the drying rate is reduced. The drying can also proceed at lower temperatures. High drying rate and excessive drying temperatures cause both physical and chemical damage to the grain (Brooker et al. 1974).

In this work, we used two heat recovery methods to reclaim the energy from the exhaust air of laboratory scale batch grain dryer. One had a heat-pipe heat exchanger and a heat pump, and one had a heat-pipe heat exchanger only. Tests with no heat recovery were used as a check or control. Two air-circulation modes of operation were used, open-loop and closed-loop. In the open-loop operating mode, the air used in the drying system was not recirculated and was released into the atmosphere. In the closed-loop operating mode, the air was recirculated inside the system. We also developed a simulation model to describe the heat recovery process. Although the present study was restricted to a batch grain dryer, the approach is equally applicable to continuous flow dryers. More specifically, the objectives of this study were: (a) To determine the amount of dryer heat that can be recovered and recycled and the related reduction in energy required for drying grain and (b) to determine the relative effectiveness of the two heat recovery methods tested.

EQUIPMENT COMPONENTS OF HEAT RECOVERY SYSTEM

The heat recovery system for the batch grain dryer considered in this study consisted of the following five sub-

Article was submitted for publication in May 1976; reviewed and approved for publication by the Electric Power and Processing Division of ASAE in January 1977. Presented as ASAE Paper No. 75-3517.

The authors are: F. S. LAI, Chemical Engineer, and G. H. FOSTER, Agricultural Engineer, U.S. Grain Marketing Research Center, ARS, USDA, Manhattan, KS.

Reference to a company or product does not imply approval or recommendation of the product by the U.S. Department of Agriculture to the exclusion of others that may be suitable.

This article is reprinted from the TRANSACTIONS of the ASAE (Vol. 20, No. 3, pp. 579, 580, 581, 582, 583, 584, 1977)
Published by the American Society of Agricultural Engineers, St. Joseph, Michigan

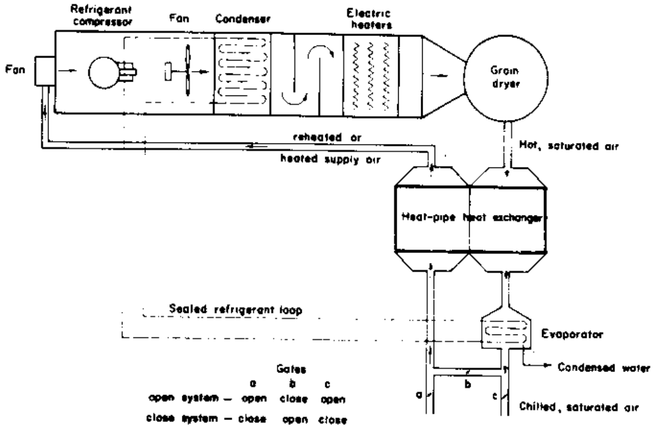

FIG. 1 Plan view of a grain dryer with heat recovery systems.

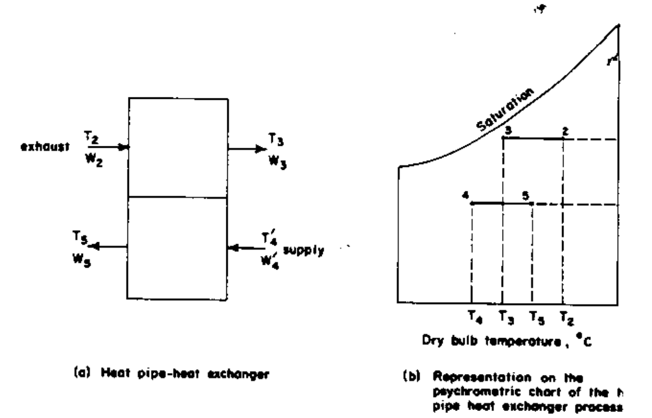

FIG. 2 Heat-pipe heat exchanger subsystem.

systems (Fig. 1):

1 Grain dryer
2 Heat-pipe heat exchanger
3 Evaporator
4 Condenser
5 Electric heater

The operating principle of the heat-pipe heat exchanger is described below. Other subsystems have been previously described in principle (Brooker et al. 1974; Shove, 1953), and additional details are given in conjunction with the description of the experimental equipment.

A heat-pipe heat exchanger is a closed evaporation-condensation system consisting of several rows of heat pipes capable of transferring thermal energy at a high rate. A working fluid and a capillary wick are permanently sealed inside the heat pipe, thereby setting up a liquid-to-vapor-to-liquid circulation cycle. Thermal energy applied to one end of the pipe causes the fluid to vaporize. The vapor then travels to the other end of the pipe where thermal energy is removed. This removal causes the vapor to condense. The condensed liquid then returns through the capillary wick. The evaporation-condensation cycle is continuous as long as one end of the pipe has a heat source and the other end has a medium into which the heat can be dissipated.

Fig. 2 (a) shows a schematic of a heat-pipe heat exchanger for cooling the exhaust air and heating the supply air. Fig. 2 (b) shows the process on the psychrometric chart. The amount of energy recovered can be shown to be

$$kW = 1.23 \times SCMS_s \times (T_5 - T_4) \quad \ldots\ldots\ldots\ldots \quad [1]$$

where

kW	=	heat recovered, kW
$SCMS_s$	=	standard air volumetric flow rate, m^3/s
T_5	=	temperature of supply air leaving, °C
T_4	=	temperature of supply air entering, °C

THEORETICAL AND SIMULATED EVALUATION OF HEAT RECOVERY POTENTIAL

Fig. 3 shows schematically the energy flow in a grain dryer with heat recovery provisions. Suppose that the compressor of the heat pump and the fan used for recirculation of air are driven by electric motors. The motors are supplied with electric energy at rates of w_1 and w_2 for the compressor and fan motors, respectively. The motors deliver work to the compressor and the fan at rate w_c and w_f, and reject heat at rates of q_c and respectively. Since the compressor is completely encl inside the air duct, the rejected heat, q_c, is compl recovered. A portion of the rejected heat from the motor, $kq_f (0<k<1)$, is recovered in the system. electric heater is supplied with energy, w_3. The sy has a rate of overall heat loss, q_{loss}, to the surrou ings. Condensed water of r kg per hour is discha from the system.

Any fixed-bed grain drying model such as developed by Bakker-Arkema et al. (1974b) can applied to drying systems with heat-recovery p sions. Alternately, a simpler model can be used. grain dryer is assumed to be operated with a relati constant input temperature with a given sensible utilization efficiency (SHUE), which is defined as the centage of the available sensible heat utilized evaporation of water. In other words, SHUE is expression of the degree of saturation of the outg drying air and is the moisture removed as the fractio the maximum removal moisture. The energy requirec evaporation of 1 kg of water with a given sensible utilization efficiency is called Sensible Heat Utiliza Index (SHUI). According to the definition, we have

$$SHUI = \frac{Q}{r} \quad \ldots\ldots\ldots\ldots$$

where Q is the sensible heat utilized. The total sens heat available to the system is given by the follo expression:

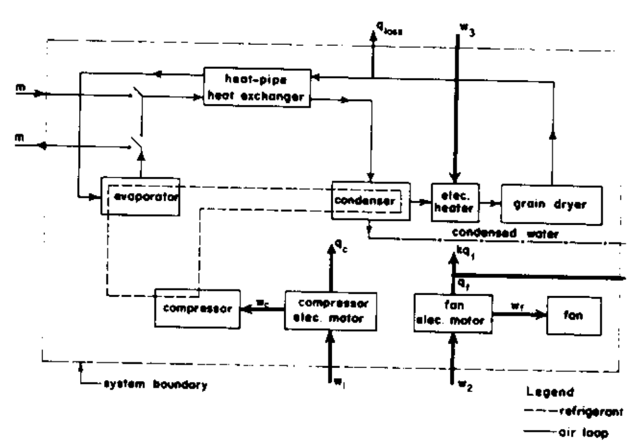

FIG. 3 Schematic of a grain dryer with heat recovery system.

TABLE 1. CONDITIONS FOR GRAIN DRYER HEAT RECOVERY SIMULATION.

Simulation no.	Air-circulation mode	Ambient air temperature, °C	Ambient air relative humidity, percent	Airflow rate ($m^3/sec/m^3$)	Drying air temperature, °C	Grain moisture (w.b.) Initial, percent	Final, percent
I	Open loop	21.1	70	0.67	51.7	25	15
II	Open loop	4.44	70	0.67	51.7	25	15
III	Closed loop	—	—	0.67	51.7	25	15

TABLE 2. RESULTS OF DRYER HEAT RECOVERY SIMULATION.

		Sensible heat util. index (SHUI)			Energy transferred		
Simulation no.	Air-circulation mode	No heat recovery, kWh/kg	Heat pipe only, kWh/kg	Heat pipe and heat pump, kWh/kg	Heat absorbed at evaporator, kW	Heat rejected at condenser, kW	Heat recovered at heat-pipe heat exchanger, kW
I	Open loop	1.18	0.861	0.242	0.674	0.903	0.226
II	Open loop	1.25	0.910	0.252	0.733	0.988	0.378
III	Closed loop	—	—	0.261	0.457	0.988	0.277

$$Q^* = w_1+[1-(1-k)(1-n_f)]\,w_2+w_3-q_{loss}+m(h_1-h_2)-rh_o \quad \ldots\ldots \quad [3]$$

where

n_f = efficiency of the fan electric motor, dimensionless
m = air mass flow rate, kg/hr
h_1 = enthalpy of the intake air, J/kg
h_2 = enthalpy of the outlet air, J/kg
h_0 = enthalpy of condensed water, J/kg
r = condensed water flow rate, kg/hr

Equation [3] is a total energy balance but does not necessarily reflect the actual cost of energy. The actual cost of energy is the sum of w_1, w_2, and w_3. For a fixed requirement of energy Q, we can reduce w_1, w_2, and w_3 by minimizing energy losses.

To determine the amount of heat that can be recovered and recycled and the related reduction in energy requirement for drying grain, we ran several simulations. At first, the Michigan State University fixed-bed simulation model (Bakker-Arkema et al. 1974b) was used. However, the model consumed more computer time than was feasible for this simulation; so a simplified model was used with which a sensible heat utilization efficiency of 50 percent was assumed.

For a system with an electric-motor-driven compressor, it can be shown that the coefficient of performance (C.O.P.) is given by

$$\text{C.O.P.} = \frac{4.19\, n_e R}{H} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [4]$$

where

n_e = efficiency of the electric motor, dimensionless
R = ratio of the heat rejected in the condenser to the heat absorbed in the evaporator, dimensionless
H = kilowatts input to the compressor required per kg-cal/s of refrigeration in the evaporator, dimensionless

In simulating the heat pump, we assumed that the efficiency of both the compressor and the electric motor was at 80 percent. Additional assumptions were made about the single-stage refrigeration cycle:

R = 1.35
H = 1.24

Hence, by equation [4]

C.O.P. = 3.64

The conditions simulated are listed in Table 1. Ambi air temperatures of 21.1 °C (70 °F) and 4.44 °C (40 were considered in the simulation of the open-lo operating mode, representing respectively, the tempe tures in early September and in November in Kans The results of simulations are shown in Table 2 Figs. 4 and 5. Fig. 4 shows the air state with the open-lo operating mode on the psychrometric chart, and Fig shows the closed-loop operating mode. In Fig. 4, ambient air is at state 4′. It reaches state 5 after pass through the heat-pipe heat exchanger. From the h exchanger through the condenser or heater, or both, reaches state 1. State 2 represents air from the dr exhaust. State 3 is at the exiting side of the h exchanger, and state 4 is at the outlet of the evaporat In Fig. 5, state 4′ and 4 coincide, since the air-circulati loop is closed.

We effected as much as 30 percent reduction in SH by utilizing the heat-pipe heat exchanger alone (Table The SHUI reduction is similar for ambient temperatu of both 21.1 and 4.44 °C, even though 67 percent mo

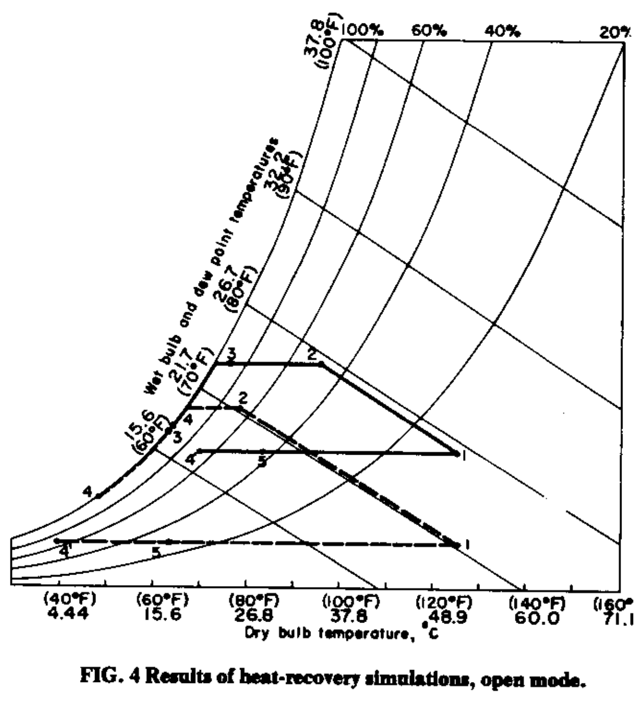

FIG. 4 Results of heat-recovery simulations, open mode.

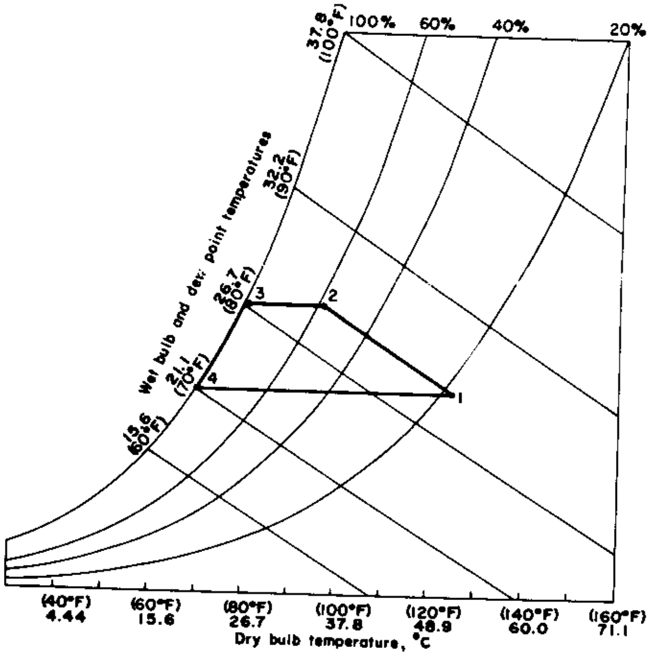

FIG. 5 Results of heat-recovery simulations, closed mode.

energy was recovered by the heat-pipe heat exchanger operation at low ambient. This apparent inconsistency results from the assumption of a SHUE of 50 percent in both cases. Normally, the utilization efficiency is increased at higher ambient temperatures. When the heat-pipe heat exchanger and the heat pump were used in combination, the reduction in SHUI was 80 percent for the open-loop operating mode at both ambient temperatures. The only energy requirement in the closed-loop operating mode was the energy needed to drive the heat pump and to circulate the air, because the system was simulated under the assumption of no heat losses. These results will be compared with the experimental results in the next section.

STUDY OF EXPERIMENTAL HEAT RECOVERY SYSTEM ON A BATCH GRAIN DRYER

Description of the Experimental System

The arrangement of the experimental grain dryer with heat recovery is identical to that of the theoretical heat recovery system shown in Fig. 1. The heat-pipe heat exchanger transfers a substantial amount of sensible heat from the dryer exhaust air to the low-temperature side of the heat-pipe heat exchanger, thereby reducing the load on the heat pump. The air then passes through the evaporator of the heat pump, where it is cooled below the dew point. Some of the moisture is condensed from the air and discharged from the system. The air then leaves the evaporator and either is released into the atmosphere or reintroduced into the supply side of the heat-pipe heat enchanger and recycled. The air picks up the heat transferred to the low-temperature side of the heat-pipe heat exchanger on its way to the fan on the dryer. Air from the fan is forced over the condensing unit—picking up heat taken from the exhaust air by the evaporator, then over the electric-resistance heaters, and finally through the corn in the dryer. We insulated the metal air duct, grain dryer, heat-pipe heat exchanger, evaporator, and compressor enclosure with 2.54 cm of blanket-type insulation to reduce heat loss from the system. The components of the system are described mo fully below.

The batch grain dryer was a cylindrical bin that wa 76.2 cm (30 in.) in diameter and 1.22 m (4 ft) high. false perforated floor above a solid floor formed a plenu where the air entered. The drying container was fille to a depth of about 38.1 cm (15 in.) with 127 kg (5 bu of corn. Thermocouples were installed in the plenu at 15-cm (6-in.) intervals above the perforated flo along the center axis of the bin.

The heat pump was a modified 0.559 kW (3/4-hp 3/4-ton) refrigerator with its evaporator and condens separated (Fig. 1).

The heat-pipe heat exchanger was a custom bui Iso-Fin* coil unit. The face area was about 30.5 x 76.2 c (12 x 30 in.), half on the high-temperature side an half on the low-temperature side. The coil was of si row aluminum construction. The unit was an air-to-a type that transferred heat between outgoing and incomin air streams in a counterflow arrangement. The operabl temperature range of the unit was from -40 °C to 93.3 °C The tube core was 1.59 cm (5/8 in.) O.D. seamles copper tubing expanded into fin collars so that permanent, tight metal-to-metal bond was formed. Th secondary surface was made of plate-type aluminu fins (0.0254 cm). The fins had die-formed collars tha completely covered the tubes, and they had a corrugate surface for efficiency of heat-transfer. We provided partition at the center of the unit to separate the out going and incoming air streams.

A fan pulled air through a 25.4-cm (10-in.) diamete metal duct that connected the dryer exhaust to th heat-pipe heat exchanger, the evaporator, and the fan The fan was a 30.5-cm (12-in.), backward-curved centrifugal blower that was powered by a 1/2-hp motor. We used a variable-speed drive arrangement to provide a range o air volumes.

We used thermocouples to measure air temperatures throughout the system, and recorded temperatures continuously during tests. We used a Pitot tube with hook gage and a velometer to determine the air velocity through a long-radius nozzle of known diameter. The electrica current to the fan and compressor motors was measured. and the total energy input was calculated from the measurements. The energy usage, weight of grain, water condensed, and operating time were recorded. The grain was sampled and the moisture content was measured on an electronic moisture tester.

Experimental Procedure

For closed-loop operating mode, gate valves a and c (Fig. 1) were closed and b was open. The valves were in the opposite position for the open-loop operating mode. The heat-pump system was turned off when only the heat-pipe heat exchanger was used. The corn used was harvested in 1975 at moisture contents of about 20 to 25 percent (w.b.) and stored in a cooler at about 4.44 °C (40 °F). All the tests were conducted on naturally wet grain. During the test the water condensed from the evaporator was collected and measured. At the end of the experiment, the moisture content and the final weight of the corn were measured and recorded.

Tests were conducted at two airflow rates. The maximum drying air temperature entering the grain dryer averaged 51.7 °C (125 °F). The corn was dried to about 15 percent (w.b.).

TABLE 3. CONDITIONS FOR DRYER HEAT RECOVERY TESTS.

Test no.	Corn weight		Moisture — w.b.		Outdoor air		Drying air temp., °C	Air-flow rate* m^3/s	Drying time, hr
	Initial, kg	Final, kg	Initial, percent	Final, percent	Temp., °C	R.H., percent			
1	127	116.1	20.7	13.9	—	—	37.7	0.142	7
2	127	115.2	20.7	12.6	—	—	51.7	0.142	7
3	127	116.1	20.7	13.3	—	—	51.7	0.142	7
4	127	112.5	25.5	15.9	21.1	60	51.7	0.142	5
5	127	112.5	25.6	15.8	23.9	55	51.7	0.142	5
6	127	112.9	24.0	13.9	23.9	52	51.7	0.142	5
7	127	112.9	24.5	15.0	21.1	40	51.7	0.0944	5
8	127	112.5	24.3	14.5	21.1	40	51.7	0.0944	5
9	127	113.4	24.5	15.4	21.1	35	51.7	0.0944	5

*Airflow is approximate - system was not totally airtight.

Nine tests were conducted. The conditions for dryer heat recovery tests are given in Table 3. We conducted three tests (Nos. 1-3) in the closed operating mode, using both the heat-pipe heat exchanger and the heat-pump. Two tests (Nos. 4 and 7) were conducted in the open operating mode but with the same heat recovery components in operation. We conducted two tests (Nos. 5 and 8) in the open-loop operating mode, using the heat-pipe heat exchanger. Two tests (Nos. 6 and 9) were conducted without heat recovery and were used as checks or controls.

RESULTS AND DISCUSSION

Table 4 summarizes results of the heat recovery tests. The energy requirements for drying were expressed by the previously defined Sensible Heat Utilization Index (SHUI).

The amount of water removed (next to the last column in Table 4) was calculated from the difference between the initial and final weights of the grain. In the open-loop operating mode, the water collected from the evaporator was not equal to the total water removed, because a substantial amount of moisture was released into the atmosphere. In the closed-loop operating mode, about 91 percent of the water removed from the grain was recovered at the condensate drain. The other 9 percent was lost, probably in air that leaked from the system.

Part of the energy was not recovered in the experimental system. The amount of sensible energy required to heat the grain from the room temperature (21.1 °C) to the drying temperature (51.7 °C) in test No. 1 accounted for about 25 percent of the total energy required for drying. About 20 percent of the heat went into warming the grain with the same heat-recovery method (heat pump and heat-pipe heat exchanger in combination) in the open-loop operating mode. In tests with the heat-pipe heat exchanger heat recovery only, and in those with no heat recovery, only about 10 percent of the input energy was required to heat the grain.

Now we will compare the experimental results with those from the simulation. For the open-loop operating mode, let us consider tests Nos. 4-6. The simulated conditions of simulation No. 1 (Table 1) and the actual operating condition of these three tests (Table 3) were nearly the same. With no heat recovery (test No. 6, Table 4), the energy required per kg of water removed was 1.48 kWh/kg in the experiment and 1.18 kWh/kg (Table 2) in the simulation. For heat-pipe heat-exchanger heat recovery only in the same operating mode, the SHUI was 1.33 and 0.861 kWh/kg for the experimental (Table 4) and simulated (Table 2) results, respectively. For the heat pump and heat-pipe heat exchanger combinations in open-loop operating mode, the SHUI was 0.853 and 0.242 kWh/kg for the experimental (Table 4) and simulated (Table 2) results, respectively. The SHUI from the simulation results was less than that from the experimental results for all tests in which heat recovery was used. This result was due in part to heat or other losses in the experiments. In general, the simulation model predicted the SHUI reasonably well.

Heat recovered by the heat-pipe heat exchanger was greater when the ambient air temperature was low during operation in open mode. A comparison of the results of simulations Nos. I and II (see Table 1) shows that heat recovery by the heat-pipe heat exchanger under simulation No. I (21.1 °C, 70 percent RH) was lower than that under simulation No. II (4.44 °C, 70 percent). At 4.44 °C (40 °F) ambient temperature, the heat recovered by the heat-pipe heat exchanger was 40 percent greater than that at 21.1 °C (70 °F) ambient; however, this advantage was partially offset by the greater amount of heat required to warm the grain from the

TABLE 4. RESULTS OF DRYER HEAT RECOVERY TESTS.

Test No.	Air-circulation mode	Heat recovery method(s) used	Sensible heat (SH) inputs				SH gain in grain, kWh	SH for drying, kWh	Water removed, kg	SHUI* kWh/kg
			Ambient air, kWh	Fan, kWh	Heat pump, kWh	Heater, kWh				
1	Closed loop	Heat pipe + heat pump	—	0.527	6.15	3.66	2.05	8.29	9.98	0.830
2	Closed loop	Heat pipe + heat pump	—	0.527	6.15	7.41	4.10	9.98	11.8	0.846
3	Closed loop	Heat pipe + heat pump	—	0.527	4.74	7.41	4.10	8.56	10.9	0.788
4	Open loop	Heat pipe + heat pump	1.26	0.381	4.22	8.56	2.05	12.4	14.5	0.853
5	Open loop	Heat pipe only	1.44	0.381	—	19.9	2.05	19.6	14.7	1.33
6	Open loop	None	1.93	0.381	—	20.6	2.05	20.9	14.1	1.48
7	Open loop	Heat pipe + heat pump	1.96	0.176	2.17	7.35	2.05	9.60	14.1	0.683
8	Open loop	Heat pipe only	1.96	0.176	—	19.6	2.05	21.3	14.5	1.46
9	Open loop	None	2.28	0.176	—	20.9	2.05	21.3	13.6	1.56

*Sensible heat utilization index - sensible heat in drying air - kWh/kg of water removed.

lower temperature.

In the closed-loop operating mode, the only energy losses were by conduction through walls and air exchange due to leaks. For the ideal case, the energy requirement of heaters for drying was zero. The SHUI from the simulation result in the closed-loop operating mode was 0.261 kWh/kg. The conditions of the air inside the system were independent of the ambient conditions. Hence, the dryer can be operated at almost any selected air temperature and humidity levels. Under such conditions, the insulation of the drying system becomes important. In the experiments, SHUI was about 0.788 kWh/kg, which indicated a substantial energy loss.

In the open-loop operating mode, the effectiveness of each of the two heat-recovery methods can be compared with normal dryer operation without heat recovery. From the experiments, the SHUI when operating with both the heat-pipe heat exchanger and heat pump was about 42 to 56 percent less than that when operating without heat recovery. In the simulated results, the reduction in SHUI was 80 percent. The use of the heat-pipe heat exchanger alone in the experiment reduced SHUI about 6 to 10 percent. In the simulated results, an advantage of about 30 percent was gained in energy utilization from use of the heat-pipe heat exchanger.

A decision concerning the recirculation of the air should be made based on ambient atmosphere conditions as well as on grain quality considerations.

The drying of 1 kg of grain from 20 to 15 percent moisture content requires the removal of 0.0625 kg of water. If we assume that 1.29 kWh of sensible heat must be supplied for each kg of water removed, then for each kg of grain dried, 0.0806 kWh must be transferred from the dryer exhaust air to the intake air. Assuming also that 1.127 kg-cal/s of refrigeration can be supplied per kW of energy input, 17.1 kW would be required per ton of hourly dryer capacity. The first cost of the heat pump would probably be about twice the cost of the dryer. If one-third of the heat can be recovered by the heat exchanger, the heat pump requirement would be reduced to about 11.4 kW for a 1 ton-per-hr dryer. If fuel prices increase to the extent that dryer operating costs significantly exceed ownership costs, then such investment in heat recovery systems may be justified.

CONCLUDING REMARKS

The analysis and synthesis of a grain-dryer heat recovery system presented here were made for determination of some of the factors related to the recovery of waste heat for saving energy in drying. These include: (a) The amount of dryer heat energy that can be recovered and recycled and the related reduction in energy required for drying grain and (b) the relative effectiveness of the two proposed heat recovery methods.

The analysis and test results reported indicate that it is possible to recover and recycle heat from a convection dryer under the conditions assumed or used in these experimental tests. In fact, under ideal conditions—no heat transfer into or out of system—it is theoretically possible to reduce the energy input to the energy required to drive the heat pump. However, the heat recovery equipment cost will exceed the cost of the dryer.

The heat pipe used in this study is a recently developed heat transfer device. Although more expensive than other types of heat exchangers, it will transfer heat efficiently over a sufficient distance that intake and exhaust ducts can be arranged side by side. It is essentially maintenance free, requires no power input, and should have a long life.

The performance of the heat-pipe heat exchanger depends directly on the temperature difference between the hot and cold sections. During warm weather, the temperature of the dryer exhaust air and that of the intake air may differ only slightly. Thus, in the open-loop operating mode the heat pipe works most effectively either during cold weather or when drying is almost complete and the temperature of the dryer exhaust air is near that of the intake air. Of course, the heat-pipe heat exchanger will not work by itself in the closed-loop operating mode since there is no temperature difference to effect heat transfer.

A combination of a heat-pipe heat exchanger and a heat pump offers considerable advantage in recovering dryer heat and makes a heat-pipe heat exchanger effective in a closed system. The evaporator of the heat pump, located between the hot and cold sides of the heat-pipe heat exchanger, lowers the temperature of the air to provide the gradient required to make the heat pipe device function. The heat-pipe heat exchanger extracts the sensible heat from the exhaust air and leaves the latent heat to be condensed and removed by the evaporator.

References

1 Anderson, R. J. 1972. Commercial concurrent flow heating-counterflow cooling grain dryer: The Anderson model. ASAE Paper No. 72-846. ASAE, St. Joseph, MI 49085.

2 Bakker-Arkema, F. W., S. Sokhansay, and M. G. Roth. 1974a. Energy requirements for a two-stage recirculating counterflow grain dryer. ASAE Paper No. 74-3005, ASAE, St. Joseph, MI 49085.

3 Bakker-Arkema, F. W., L. E. Lerew, S. F. De Boer and M. G. Roth. 1974b. Grain dryer simulation. Research Report No. 224, Agricultural Experimental Station, Michigan State University, East Lansing, MI.

4 Brooker, D. B., F. W. Bakker-Arkema, and C. W. Hall. 1974. Drying cereal grains. The AVI Publishing Co., Westport, CT.

5 Converse, J. W. 1972. A commercial crossflow-counterflow grain dryer: The HC. ASAE Paper No. 72-828, ASAE, St. Joseph, MI 49085.

6 Davis, C. P., Jr. 1949. A study of the adaptability of the heat pump to dry shelled corn. Unpublished master's thesis, Purdue Univ., Lafayette, IN.

7 Flikke, A. M., H. A. Cloud, and A. Hustrulid. 1957. Grain drying by heat pump. AGRICULTURAL ENGINEERING 38(8):592-597.

8 Lerew, L. E., F. W. Bakker-Arkema, and R. C. Brook. 1972. Simulation of a commercial crossflow dryer: The Hart-Carter model. ASAE Paper No. 72-829, ASAE, St. Joseph, MI 49085.

9 Morey, R. V. and H. A. Cloud. 1973. Simulation and evaluation of a multiple column crossflow grain dryer. TRANSACTIONS of the ASAE 16(5):114.

10 Morey, R. V., H. A. Cloud, and W. E. Lueschen. 1974. Practices for efficient utilization of energy for drying corn. ASAE Paper No. 74-3451, ASAE, St. Joseph, MI 49085.

11 Rimberg, D. 1974. Utilization of waste heat from power plants. Noyes Data Corp., Park Ridge, NJ.

12 Roberts, D. E. and D. B. Brooker. 1973. Grain drying with a circulator. ASAE Paper No. 73-323, ASAE, St. Joseph, MI 49085.

13 Shove, G. C. 1953. A laboratory investigation of the adaptability of the heat pump to batch drying of shelled corn. Unpublished master's thesis, Kansas State University, Manhattan, KS.

14 Shove, G. C. 1968. Application of dehydrofrigidation to shelled-corn conditioning. TRANSACTIONS of the ASAE 11(3):312-317.

15 Stephens, G. R. and T. L. Thompson. March 21-22, 1975. Improved crossflow grain dryer design using simulation. Paper No. NC 75-301 presented at the Mid-Central Region ASAE meeting, St. Joseph, MO.