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ENERGY CONSERVATION IN FRUIT DEHYDRATORS
UTILIZING RECIRCULATION OF EXHAUST AIR
AND HEAT RECOVERY HEAT EXCHANGERS

Final Report

By
John E. Groh

December 1977

Work Performed Under Contract No. EY-76-S-02-2916

University of Arizona
Environmental Research Laboratory
Tucson, Arizona

U. S. DEPARTMENT OF ENERGY

Division of Industrial Energy Conservation

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Final Report

John E. Groh
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December 1977
(Revised February 1978)

Prepared for

THE U.S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
UNDER CONTRACT NO. E(11-1)-2916

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ABSTRACT

Dehydration of fruit in the United States is often done by means of a tunnel dehydrator utilizing large quantities of fossil fuel. Existing dehydrators have been designed to operate with maximum product through-put and with little regard for energy efficiency. Incorporating controlled recirculating air dampers and thermal energy recovery equipment on the exhaust air, calculations and preliminary tests show that the energy required in dehydration may be cut by up to 40%.

During this work, one tunnel was modified and upgraded in a commercial 24-tunnel facility to demonstrate the potential savings. A primary element of this program, the heat recovery heat exchanger, operated below specifications. The manufacturer, Hughes Aircraft Company, has determined that the heat exchanger design was based on faulty data, and is constructing a second exchanger which can be installed and tested during a subsequent program.

I. Introduction

This report describes the work accomplished by the Environmental Research Laboratory of the University of Arizona in cooperation with the Superior Farming Company of Bakersfield, California, under contract E(11-1) 2916 with the Energy Research and Development Administration during the period May 26, 1976 to July 31, 1977.

This work was initiated in the summer of 1976. It consisted of selecting a heat recovery heat exchanger, an exhaust air recirculation damper and doing the support mechanical design for the installation in one dehydrator tunnel of a 24-tunnel facility at the Superior Farming Company's Commodity Center located at Bakersfield, California. The purpose of this work was to demonstrate an energy savings of up to 40% of that now used without exhaust gas recirculation and a heat recovery heat exchanger.

A seminar is to be held for dehydrator operators to disseminate the results of a subsequent program.

II. Background

Since 1966, about three to four percent of the total fruit produced in the United States has been processed into dried fruit. In 1973, there were 436,000 tons of fruit dehydrated from some 1.6 million tons of fresh fruit (1).

Most fruit dehydrators operating in the United States today are tunnel dehydrators which operate in the once-through configuration; i.e., ambient air is drawn into the dehydrator burner section where it is heated by a free-standing burner to the

control temperature of 160 to 185°F, depending on the type of fruit being dried. This air, a mixture of primary and secondary air as well as products of combustion, passes through a blower, then through the product. After picking up moisture from the fruit and cooling to approximately 130 to 155°F, the air is exhausted to the atmosphere. The high temperature and humidity levels of this exhaust air represent a considerable energy loss.

A single dehydrator tunnel will typically dry seven to eight tons of fresh fruit in a 24-hour period, utilizing an airflow rate of approximately 50,000 CFM. Energy input to such a system can average 3 million Btu/hr over the 24-hour period. A 24-tunnel dehydrator facility will therefore consume 150×10^9 Btu of fossil fuel in a normal drying season to produce 1,760 tons of dehydrated fruit.

Most fruit dehydrating processes are of the continuous through-put type. A dehydrator tunnel will contain perhaps 12 steel cars in a row, each holding 25 wooden trays. The fruit is laid out on the trays so that air can pass over each layer and progress horizontally through the cars. The cars are then stepped sequentially through the tunnel so that the fruit is exposed to the air at various conditions of moisture at various times. Tunnels may be operated with either parallel or counterflow; i.e., the cars can be moved either in the direction of airflow or opposite to it (see Figure 1). In the first case, the wet fruit just placed in the tunnel comes in contact with the driest, hottest air, and the driest fruit sees the relatively spent "cooler, moist" air. In the counterflow configuration, the wet

Fig. 1 DEHYDRATOR TUNNEL SECTION VIEW

fruit sees the spent air just prior to exhaust, and the finished fruit sees the air just off the air blower. A typical operating procedure might be for one car to be loaded into and another to be removed from a 12-car tunnel every two hours.

Figure 2 shows a typical multi-tunnel dehydrator facility.

The actual drying mechanism has been studied extensively, and it has been determined that air velocity, air temperature, and air moisture content all affect the drying rate of fruit in somewhat complex ways (2). For example, during the initial drying stages, the rate of drying is limited by the rate of evaporation from the surface of the fruit. Whereas in the latter stages of drying, the rate is limited by the resistance to movement of moisture from the interior of the fruit to the surface. For this reason, there is a tendency to subject the drier fruit to higher temperatures; therefore, counterflow operation is often preferred. However, under these counterflow conditions in which the relatively dry fruit sees the warmest air, it is possible to cause some fruit damage in the form of case hardening, so that generalized rules are difficult to employ. Dehydrating is somewhat of an art, and the art has been developed for minimum tunnel drying time with little regard for energy conservation.

III. Statement of the Problem

The potential for energy savings lies in the fact that tremendous quantities of warm 130°F humid air exhaust the dehydrator continuously. The incentive for recovering a portion of this energy is a balance between the potential fuel savings and the

Figure 2-A Dehydrator Tunnel with Fruit Trays
Air Exhaust End

Figure 2-B Dehydrator Tunnel - Warm End

hardware cost, plus a modest increase in drying time. Energy conservation can be approached in two ways: air recycling through a by-pass damper, and thermal energy recovery through the heat exchanger equipment.

The extent to which air can be recirculated is limited by the moisture content in the air and the effect that moisture has on the rate of drying. This will vary throughout the dehydrating season, depending on ambient air conditions and the fruit being dried. The extent to which thermal energy should be recovered from the exhaust air will depend on an economic analysis of capital cost of the equipment, plus increased fan horsepower associated with the heat recovery equipment. Therefore, the recirculation air should be controlled through a feedback continuously-modulating control system, and the heat recovery equipment should be selected for optimum economic return.

The normal dehydrator season in the Bakersfield, California, area is August through mid-November. Normally prunes are produced in August and raisins from September into November. The units operate 24 hours a day, six days a week. To indicate the potential energy savings, ambient conditions of 75°F dry bulb and 62°F wet bulb are selected for the following sample calculations.

Figure 3 shows typical dehydrator conditions as broken lines in the conventional operating mode, with an airflow rate of 50,000 cubic feet/minute (220,000 lbs of dry air per hour). The energy added by the burner is:

$$Q_B = \dot{m}(h_B - h_A)$$

$$Q_B = 220,000 \text{ lb/hr} (50.6 - 27.9) \text{ Btu/lb}$$

Fig. 3 PSYCHROMETRICS OF FRUIT DEHYDRATION

$$Q_B = 4,944,000 \text{ Btu/hr}$$

The water vapor added during the combustion process is approximately 75 pounds per million Btu for natural gas and approximately 50 pounds per million Btu for No. 2 fuel oil. These quantities are not significant in most dehydrator operations and total approximately 0.0017 lbs of water per pound of dry air increase in specific humidity in the above example. This represents 45 gallons of water per hour.

Assuming that the drying process is one of adiabatic saturation, the air will exit the tunnel at condition C, 130°F dry bulb. During this process, each pound of dry air will pick up 0.0065 lbs of water (0.0172 - 0.0107). At 220,000 lbs/hr, the water extraction rate will be 1,430 lbs/hr or 4,134 gallons per 24-hour day.

Now to demonstrate the energy savings potential proposed, let us assume that approximately 70% of the 224,000 lbs/hr airflow is recycled through a damper system such that the absolute humidity of the air returned to the burner and the tunnel does not exceed 200 grains of moisture per pound of dry air. This can be controlled by sensing air dew point temperature and increasing the return air when the dew point temperature falls below 87°F. Under these or similar conditions, it is proposed that the drying time will not increase greatly.

This recirculation rate will result in an air exhaust/make-up air quantity of only 66,000 lbs/hr. This airflow quantity is small enough to suggest the feasibility of exhaust recovery heat exchangers. The psychrometrics of this process are shown in Figure 3 in the solid lines.

Outside air will now be preheated to approximately 114°F (point D) with the exhaust air (point E), which will cool to about 97°F (point H). The energy balance on this process is:

$$\begin{aligned} Q_R &= \dot{m}(h_E - h_H) = \dot{m}(h_D - h_A) \\ &= 66,000(71.2 - 61.8) = 66,000(37.3 - 27.9) \\ Q_R &= 620,400 \text{ Btu/hr heat recovery} \end{aligned}$$

This energy recovery is about 30% of the total energy available in the exhaust and is considered conservative from the standpoint of hardware feasibility. Up to 50% of the total available heat may prove to be economically recoverable from the exhaust. This would increase Q_R up to 1,079,000 Btu/hr.

Preheated air at D would then be mixed with the recirculated air from E to provide an air supply to the burner at condition F. The burner would reheat the full air supply to 160°F (point G) where it would enter the fruit. The energy required by the burner would be:

$$\begin{aligned} Q_B &= \dot{m}(h_G - h_F) \\ &= 220,000(71.2 - 61.2) \\ Q_B &= 2,220,000 \text{ Btu/hr} \end{aligned}$$

which is 44% of that required in the first case. In this situation, water would be removed at the rate of:

$$\begin{aligned} \Delta G &= \dot{m}(W_E - W_G) \\ &= 220,000(0.0357 - 0.0290) \\ \Delta G &= 1,474 \text{ lbs/hr} \end{aligned}$$

or approximately the same rate as before.

The 56% reduction (1 - 44%) in thermal energy input will be

partially offset by an increase in fan horsepower necessary to overcome the pressure drop across the heat exchanger; and there may be some increase in drying time of the fruit because of the higher moisture level in the air. Nevertheless, a net energy reduction of 40% over conventional operation should be obtained.

IV. Experimental Program

The energy recovery heat exchanger was installed in September in the Superior Farming Company dehydrator facility (3). Its unmonitored operation commenced September 14, 1976. Due to unseasonal inclement weather in September, it was necessary to suspend work on the test tunnel. All tunnels were pressed into service by the Superior Farming Company in order to maximize product through-put.

Late in October, due to the aforementioned weather, fresh undehydrated products became unavailable. This released the test tunnel from service. Monitoring instrumentation was then installed. Total monitoring commenced on October 27, 1976. Partially dried stored product has been reprocessed during the winter of 1976-77. This operation yielded important initial data indicating that the energy conservation heat recovery heat exchanger was performing below specifications. Figure 4 shows the actual exchanger performance as compared with the predicted performance curves. Data acquisition was then terminated to resolve the heat exchanger problem. Appendix I shows the exchanger design and the original specifications for the heat exchanger.

Failure of the heat exchanger to perform according to original design specifications severely impacted the remainder of

Fig. 4

the experimental program. Because of the pressure drop through the heat exchanger, which reduced airflow through the dehydrator tunnel, which in turn reduced the product fruit through-put, data from the modified tunnel were of no value. The heat exchanger was by-passed so the experimental tunnel could be used for its normal fruit drying purpose. The remainder of the program was spent working with the heat exchanger manufacturer (4) to determine the cause of the substandard performance and agree on a suitable replacement unit. Correspondence pertaining to the problem is included in Appendix II.

A satisfactory replacement design was ultimately selected; however, at the close of the experimental program, the heat exchanger had neither been constructed, installed nor tested.

V. Heat Exchanger

A heat exchanger utilizing heat pipes was selected for this program because the proposed design met the pressure drop limitations and the energy savings goal. It had the advantage of requiring no pump and no antifreeze when compared to the two-coil run-around cycle. The intake-exhaust air ducts are a compatible configuration for use of this type exchanger.

As indicated in Appendix II, Hughes engineers found poor agreement between their tests and published literature or coil manufacturer's data. An alternate design, described in Appendix III, was selected which the manufacturer claims can meet the original specifications and fit the required envelope. Installation and testing must be done during a future program.

VI. Conclusions

A heat recovery heat exchanger, an exhaust air recirculation damper, and associated hardware and instrumentation were installed in one dehydrator tunnel of a 24-tunnel facility at the Superior Farming Company's Commodity Center in Bakersfield, California. The intent was to demonstrate an energy savings of up to 40% of that now used without exhaust gas recirculation and a heat recovery heat exchanger.

These goals were not met due to heat exchanger performance below design specifications. A new heat exchanger should be installed and tests continued.

Notes and References

1. "Marketing California Dried Fruits; Prunes, Raisins, Dried Apricots and Peaches," 1974, Federal-State Market News Service.
2. Van Arsdale, W. V., et al, 1973, Food Dehydration, Vol. 1, 2nd ed., and Vol. II, AVI Publishing Co., West Port, Connecticut.
3. Superior Farming Company, 3501 Stockdale Highway, Bakersfield, California 93309.
4. Hughes Aircraft Company, Electron Dynamics Division, Torrance, California 90509.

APPENDIX I

Original Hughes Heat Exchanger Design

TO: The University of Arizona

REF: Env. Research Lab

ORIGINATOR: 49.5x138x6R12P

DATE: 6-2-76

PRELIMINARY HEATBANK UNIT SPECIFICATIONS	Condition 1	Condition 2	
EXHAUST AIR FLOW (SCFM) (LB/HR)	51,640	30,000	
SUPPLY AIR FLOW (SCFM) (LB/HR)	51,640	30,000	
EXHAUST TEMPERATURE (°F)	130	155	
SUPPLY TEMPERATURE (°F)	75	85	
EXHAUST FACE VELOCITY (SFPM)	485	285	
SUPPLY FACE VELOCITY (SFPM)	485	285	
FACE AREA (H" X L")	49.5x138x15.5		
DIVIDER LOCATION (EXHAUST/SUPPLY)			
WEIGHT			
HEATBANK MATERIAL (PIPES)	Aluminum		
HEATBANK MATERIAL (FINS)	Aluminum		
ROWS DEEP	6		
FINS PER INCH (EXHAUST/SUPPLY)	12		
FIN STYLE	Plate		
EXHAUST PRESSURE DROP (IN./W.G. STD)	.43 (STD)	.22 (STD)	
INCOMING PRESSURE DROP (IN./W.G. STD)	.43 (STD)	.22 (STD)	
RECOVERY EFFICIENCY (%) (EXHAUST/SUPPLY)	61.9	72.2	
RECOVERY (BTUH)	423,300	365,800	
PRICE: FREIGHT ALLOWED ☒ YES ☐ NO	\$4848		
DELIVERY (WEEKS ARO)	8		
HEATBANK MODEL NUMBER	15-49.5x138-6R12P		

ADDITIONAL COMMENTS:

SHIPPING WEIGHT = 1450 LBS

ESTIMATED SHIPPING COST = \$139.00

HEAT PIPE WORKING FLUID TO BE FREON 12.

ADDITIONAL COMMENTS:

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APPENDIX II

Correspondence with Hughes on Heat Exchanger Problem

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09 May 1977

Dr. John Groh
University of Arizona
Environmental Research Laboratory
Tuscon International Airport
Tuscon, Arizona 85706

Dear Dr. Groh:

I have reviewed the history problems associated with the heat pipe heat recovery unit, and also the options that are available to solve the efficiency and pressure drop problems to our mutual satisfaction.

The thermal model designed to predict performance of the heat recovery unit was based on published technical literature and data supplied to us by coil manufacturers.

Tests at Superior Farms indicated that efficiency was considerably lower than predicted by the thermal model. Tests at EDD confirmed that the efficiency predictions were too optimistic, and the data on heat transfer areas and fin efficiency, supplied by fin tubing and coil manufacturers, did not agree with the experimental results.

Upon evaluation of all results, we proposed to replace the old unit at no cost to Environmental Research Laboratory with another unit that meets efficiency requirements. To meet efficiency requirements, the number of rows or the number of fins has to be increased. This would increase the pressure drop to a level outside specifications, and therefore would be unacceptable.

During the past month, we changed our method of manufacture in order to improve fin efficiency and heat transfer areas, and make performance more consistent from unit to unit. Our test bed was modified to increase accuracy of heat transfer measurements. We are planning to accumulate sufficient test data to upgrade our heat transfer model.

In view of the above, I like to propose that we take the following steps before a new unit is selected.

DR. JOHN GROH

09 MAY 1977

1. Upgrade the thermal model and run trade-offs in order to select the most suitable unit to meet your requirements.
2. Jointly review options available.
3. Select a design which best meets your requirements.
4. Fabricate a new heat recovery unit.

I estimate that the data acquisition and thermal model upgrading will be completed by June. At that time, we would like to discuss the available options.

Sincerely,

ELECTRON DYNAMICS DIVISION

(Handwritten signature)
A. Basulis
Assistant Manager
Thermal Products

AB:ds

cc: J. H. Johnson
W. R. Iriarte
A. G. Peifer

APPENDIX III

Performance of Hughes Supplied Replacement Heat Exchanger

(Revised February 1978)

HEAT EXCHANGER PERFORMANCE
EFFECTIVENESS VS. FACE VELOCITY

REPLACEMENT HEAT EXCHANGER