ENERGY CONSERVATION IN DRYING OF FRUITS IN TUNNEL DEHYDRATORS

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ABSTRACT

This study reports the energy use and thermal losses associated with tunnel dehydrators and discusses methods of increasing energy efficiency. These dehydrators can operate with an efficiency of water removal greater than 50%. It is shown that energy conservation techniques such as minimizing air leakage, increasing air recirculation, utilizing a furnace heat shield to prevent heat losses, and maximizing input can result in significant energy savings.

INTRODUCTION

Tunnel dehydrators are most widely used in artificial drying of fruits. Raisins, prunes and apples make up by far the bulk of the fruits dried in the USA and among these all the prune crop is artificially dried in tunnel dehydrators. Natural gas, propane or other fossil fuel sources are employed in supplying necessary thermal energy to accomplish dehydration. With the prospect of continuously rising cost and shortage of fuel supply, it is becoming increasingly important to economize the fuel consumption in this highly energy intensive

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operation. This study is based on the investigation of prune dehydrators but should be applicable to other fruit drying situations.

The first reported study on energy efficiency of dehydrators was conducted by Cruess and Christie (1921) when heated, forced-air dehydrators were introduced as a substitute for sun-drying of prunes. They indicated that countercurrent prune dehydratros should operate at an efficiency of at least 40%. They also recommended that energy could be saved by recirculating 75% or more of the air; preventing air from passing between the trays and walls and by-passing the fruit; and dipping the fresh fruit in lye to "check" the skin and increasing the drying rate. Subsequent reports dealt primarily with proper operation of dehydrators (Christie 1926; Christie and Ridley 1923; Kilpatrick *et al.* 1955; Perry 1944; Perry *et al.* 1946; Van Arsdel *et al.* 1973) and development design criteria with little or no specific mention of energy use except for indicating the value of recirculation. Recirculation was emphasized primarily to prevent case hardening. Case hardening is believed to be rapid drying of the surface of the fruit which restricts movement of the moisture from the interior of the fruit.

Most investigators agreed that relative humidities in the exhaust air of a countercurrent flow tunnel should be in the range of 35-40%. Guillou (1942) indicated that drying rate of prunes is not affected by relative humidity below 40%. Perry (1944) subsequently reported that relative humidity above 35% at 75°C (167°F) reduced drying rates. Mrak and Perry (1948) recommended countercurrent flow dehydrators could be operated at an exhaust end relative humidity of 60%, although the wet bulb temperature should never exceed 49°C (120°F).

Gentry *et al.* (1965) demonstrated that concurrent (parallel) flow operation of tunnels designed for traditional countercurrent flow operation would significantly increase fruit throughput. Initial tests revealed a 12% increase in heating energy consumption per ton of fruit dried for concurrent versus countercurrent flow. Since then, many of the older tunnels have been converted to concurrent flow operation and new tunnels are designed for this mode of operation. McBean *et al.* (1966) demonstrated that lye dipping of prunes was not effective in reducing drying time in concurrent flow tunnels.

A majority of the dehydrators were built when there was a cheap and unlimited supply of natural gas, and fuel efficiency of the dehydrators was not a major concern. Very little research focussed on energy conservation aspects of tunnel dehydrators has been reported. Groh (1978) suggested that increased recirculation and the use of heat exchangers would reduce energy use although he had no test data to support his suggestions. Carnegie (1980) has investigated the effectiveness of various heat exchange systems for recovering heat from exhaust air but has not discussed other areas of losses and means to reduce them.

The objectives of this study were to:

(1) Determine a heating energy budget for selected dehydrator types.
(2) Identify areas where heat losses could be minimized and energy conserved.
(3) Compare the energy consumption of concurrent versus countercurrent flow dehydrators.

PROCEDURE

Dehydrators

Three different types of dehydrators were selected. Distinct features of these dehydrators are listed in Table 1. Figures 1 and 2 are sketches of the dehydrators investigated. All dehydrators are concurrent flow, air recirculating tunnel dehydrators. They operate by removing a car of dry fruit from the cooler end of the tunnel and adding a car of fresh fruit to the other end, approximately every two hours, which is called a pull cycle. Approximately 18 h are required to dry a car of fruit.

Temperature Measurement

All the temperatures except the ambient air temperature were measured using copper-constantan thermocouples connected to a recording potentiometer. Temperatures measured were: (1) dry bulb and wet bulb temperatures of the drying air at various location in the

Table 1. Various tunnel dehydrators selected for testing

Location No.	Distinctive Features
1	Concrete tunnels with fan belt opening on the roof located downstream of fan. Partial recirculation of air. Tunnel originally designed for countercurrent flow.
2	Transite tunnels. Partial recirculation of air. Tunnel designed for concurrent flow.
3	"Miller" type tunnels made of cinder block. Partial recirculation of air.

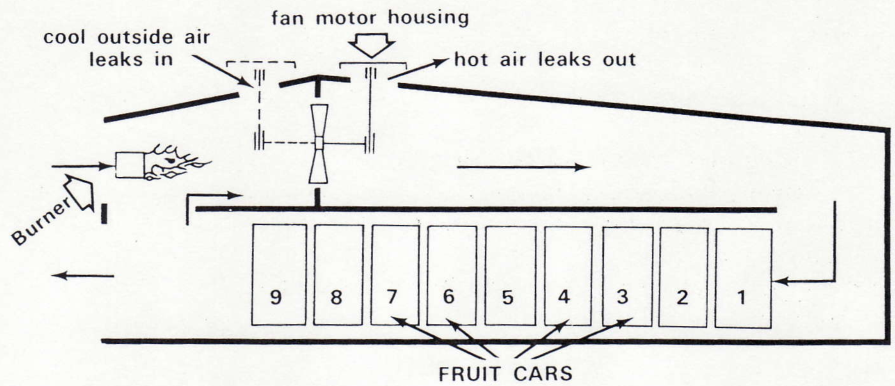

FIG. 1. TUNNEL DEHYDRATOR AT SITE 1 AND 2

At Location 1 the motor is downstream of the fan indicated by solid lines and at Location 2 motor is upstream of fan as indicated by dashed lines.

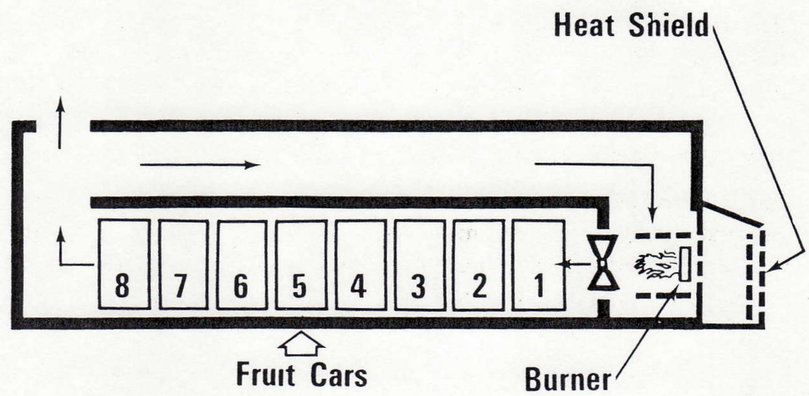

FIG. 2. MILLER TYPE DEHYDRATOR AT LOCATION 3 WITH SUGGESTED HEAT SHIELD

dehydrator, (2) temperatures of inside and outside surfaces of the dehydrator.

Wet bulb temperatures were measured by enveloping a thermocouple in a cotton wick supplied with water from a small water reservoir keeping the wick moist. Dry bulb temperature and relative humidity of the outside air were recorded by a mechanical hygrothermograph.

Air Flow Measurements

Air flow measurements were taken using a hot-wire anemometer or a vane anemometer. Measurements were made at various points in the cross-sections of interest and average flow calculated for use in energy balance computations.

Natural Gas Consumption

Five bellows-type in-line gas meters supplied by the Pacific Gas and Electric Company were installed to record the natural gas use at the burners. Each meter measured the gas consumption of one burner which supplied heat for a dual tunnel unit. Four of the meters were installed at location 1, and one at location 2 (Table 1). Gas flow measurements at location 3 were made using an orifice meter. Pressure gauges were installed in the gas supply line. Readings were taken at the end of each pull cycle.

Moisture Content

Samples were taken before and after the product was dried. The moisture content was determined using a vacuum oven (AOAC) for high moisture samples and calibrated conduction type meter for low moisture samples (DFA-AOAC). Net weight of dried product for each drying period was measured to compute the quantity of water removed in the dehydrator.

Test Conditions

In most of the tests the dry bulb temperature, humidity, air flow, initial moisture content, and the final moisture content of the prunes were not controlled by the investigators. These parameters were set by the management of the dehydrating units according to normal commercial operation.

Selected tunnels were modified as indicated to test various conservation techniques. Comparison of the energy consumption of concurrent versus counter flow was studied by analyzing three years of gas consumption data available from a drying cooperative (Dominik 1979).

RESULTS AND DISCUSSION

Table 2 lists an energy budget for the concrete, concurrent flow tunnel dehydrator at location 1. Fifty-three percent of energy is used for evaporating water. This represents a fairly high moisture removal efficiency compared to many types of other agricultural drying operations, especially such as nut and grain drying operations. However, this should be expected since the fruit enters the tunnel at about 70% moisture (wet basis) and energy use is relatively efficient at high moisture levels (Henderson and Perry 1966). The main areas of heat loss are in the exhaust air, burner inefficiency and air leaks. Heat lost by conduction through the wall and by hot fruit and trays leaving the tunnel is relatively small.

Table 2. Energy budget of a concrete, concurrent flow tunnel dehydrator for prunes at location 1

Thermal Energy Loss/Utilization	Percent of Total Thermal Energy Input
Moisture evaporation	58
Exhaust air	16
Burner and other losses	12
Air leaks (door, fan belt opening)	8
Walls and ceiling	3
Fruit and trays	3
Total	100

Pertinent data comparing performance of three dehydrator types studied are presented in Table 3. The wide range of energy efficiencies observed is due to factors such as tunnel design, level of maintenance, and operation procedure. This study revealed that energy use efficiency is affected by the following specific factors:
(1) Heat loss in exhaust air.
(2) Heat loss through air leaks.
(3) Amount of fruit dried per tunnel-day.
(4) Conductive and radiative heat loss through walls.
(5) Burner losses.

Table 3. Observed and calculated data showing comparison of various tunnel dehydrators

Location #	Heated Air: Temp, °C Dry Bulb t_d	Heated Air: Temp, °C Wet Bulb t_w	Heated Air: Flow[1] at t_d and t_w m^3/s	Moisture Content % (Wet Basis) Initial	Moisture Content % (Wet Basis) Final	Average Moisture Removed kg/h	EnergyQ Output kW	Energy Input From Fuel Consumption kW	Energy[3] Input MJ/kg of Water Removed	Efficiency[4] of Water Removed %
Col #: 1	2	3	4	5	6	7	8	9	10	11
1	84	46	14.01	77.1	21.2	590	418	715	4.54	58
2	87	46	12.41	69.4	17.2	520	368	706	4.88	52
3	82	46	9.02	71.1	20.5	380	269	686	6.50	39

[1]1 m^3/s = 2575 cfm implying flow rate at location 1 is 36050 cfm

[2]Enthalpy gain of the moisture in column 7 entering as part of the fresh fruit and discharged in exhaust. It refers to first heating the water to 74°C and then vaporizing at 74°C

[3]Computed from column 7 and 9 and using the conversion factor of 1 kW = 3.599 MJ/h

[4]Computed from column 8 and 9

Exhaust Air

The rate of energy lost in the exhaust air is determined by the amount of sensible plus latent heat (enthalpy) in the exhaust air and the quantity of air exhausted per unit time. Increasing recirculation will reduce the amount of exhaust air with a slight increase in enthalpy of the air. The net effect is a reduction in the amount of energy needed per unit time to keep the tunnel at operating temperature. The effect of increasing recirculation in a prune tunnel based on typical airflow and temperature conditions measured at all three dehydrator sites is illustrated in Fig. 3. The upper limit on the level of recirculation is a humidity above which it results in increased drying times. Perry (1944) indicated that 35% relative humidity at 74°C (165°F), or a wet bulb temperature of 52°C (125°F) at 74°C (165°F), to be this upper limit. At location 2 this effect was tested by comparing the seasonal energy use of a group of 18 tunnels under normal recirculation levels versus energy consumption of these tunnels with doors placed on the air exit of the tunnel to increase recirculation. Table 4 shows a 15% reduction in gas consumption can be achieved by

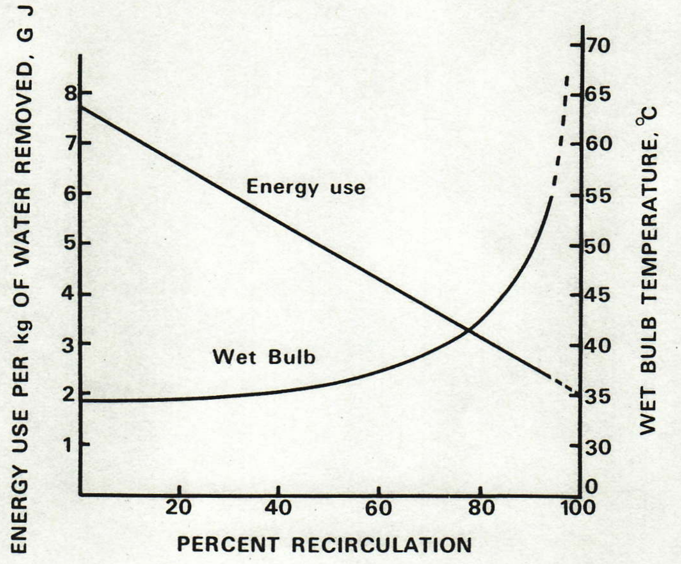

FIG. 3. EFFECT OF RECIRCULATION ON ENERGY USE AND HUMIDITY IN A CONCURRENT FLOW DEHYDRATOR

Table 4. Effect of increased air recirculation on energy usage in a prune tunnel operated at an exhaust dry bulb temperature of 68°C at location 2

	Wet Bulb Temp. (°C)	Seasonal Average Natural Gas Consumption (m^3/h)	Seasonal Average Existing Fruit Moisture (%)	Reduction in Gas Use (%)
Control tunnels	46	68	19.1	—
Tunnels with doors on air exhaust end	52	58	18.6	15%

increased recirculation. A graphic illustration of tunnel dehydrators operating under different modes with maximum, partial and no recirculation is presented on a skelton psychrometric chart in Fig. 4. The mixture (m) of fresh air (f) and some exhaust air (e) is heated from (m), to the desired hot air temperature (h). Now (m) to (h), the rise in dry bulb temperature required is less in the case of tunnel operating with the doors placed on the air exit than the conventional operating mode. Thus, resulting in significant energy use reduction.

For a two day period, one tunnel with doors was operated at a 60°C (140°F) wet bulb temperature. Although gas use for this tunnel could not be measured separately the exiting fruit moisture was not noticeably higher than that from neighboring tunnels with lower wet bulb temperatures. Wet bulb temperatures at this level require that outside air be ducted directly to the burner inside the tunnel. Without this, the burner will not remain lighted at these high levels of air recirculation.

Air Leakage

Air leakage was found to be a significant source of energy loss at location 1. This tunnel had been originally designed to operate in a counter-current mode. The tunnel was converted to concurrent flow by changing the direction of air flow. This resulted in the fan belt opening on the roof being on the positive pressure side of the fan, forcing 71°C (160°F) air out of the openings around the motor. Such air losses resulted in 8% of the energy requirements for the tunnel. This leakage can be prevented by sealing the openings. Tunnels designed with the opening on the negative pressure side of the fan do not have this problem but let too much cold air in unless the opening is reasonably well sealed. Air leakage around door seals and through holes was also seen to be a problem, although the magnitude of these losses were not measured and would vary from tunnel to tunnel.

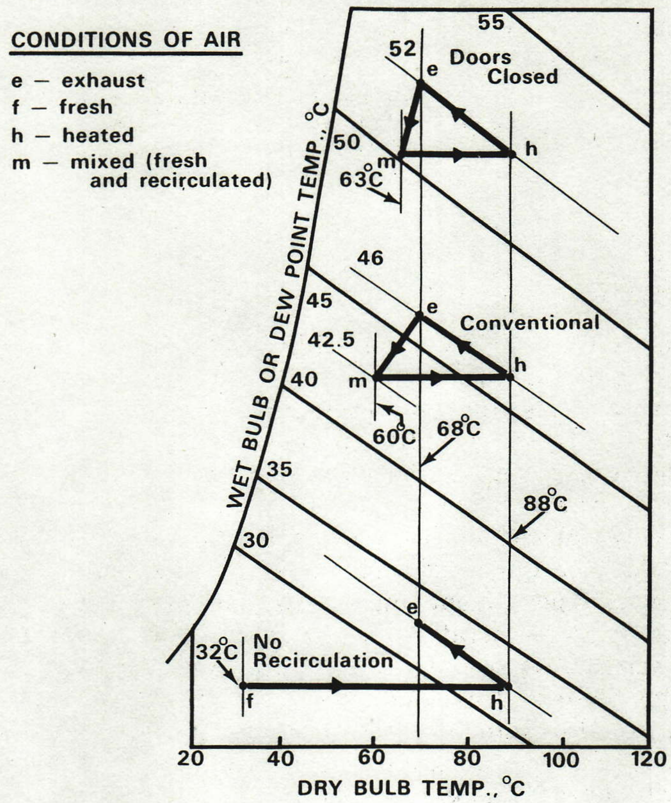

FIG. 4. PSYCHROMETRIC REPRESENTATION OF TUNNEL DEHYDRATORS AT LOCATION 2 UNDER DIFFERENT OPERATIONAL MODES

(a) no recirculation of air, (b) partial recirculation—conventional mode, (c) maximum recirculation with doors on air exhaust end closed.

Proper design and maintenance will reduce these losses to a minimum.

Tunnels at location 1 and 2 had canvas belt baffles installed in them. These baffles prevented hot and high velocity air from bypassing the fruit, traveling between the trays and the tunnel walls, and channelling out of the tunnel. The actual energy savings associated with properly installed baffles could not be calculated exactly because of difficulty in measuring the air flow between the tunnel walls and the tray, but rough estimates indicate savings of about

3-4%. Absence of such baffles at location 3 is one of several causes for low moisture removal efficiencies indicated in Table 3 (other causes for low efficiency at this location are discussed later in this section). The baffles would not be needed if doors were placed on the air exit end of the tunnel.

Fruit Dried Per Day

Fuel consumption and fruit output data revealed that fuel consumption per ton of fruit is directly affected by the quantity of fruit dried per unit time (Fig. 5). The data were collected for a three year period for 14 dehydrator locations each having a number of tunnels. The block effect for each location was removed, by subtracting the difference between the average of all the data and the average for an individual location. The linear regression equation indicates that for every additional ton per tunnel per day of fruit output the fuel use is reduced by equivalent of 1·93 MJ (183,000 Btu) of natural gas per tunnel-day (the relatively low R^2 value of 0.40 is expected since the fuel usage is a function of the various other factors that have been

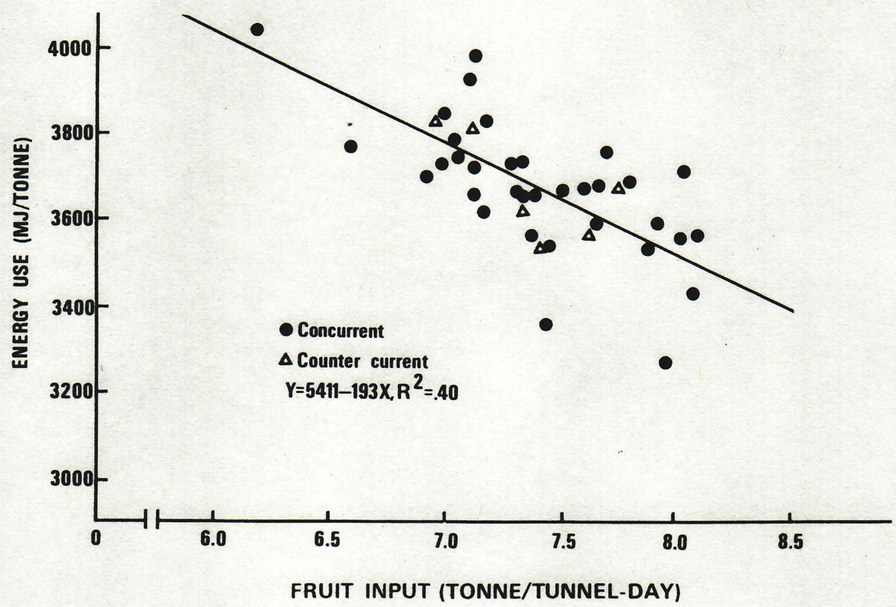

FIG. 5. EFFECT OF RATE OF FRUIT OUTPUT ON FUEL CONSUMPTION

mentioned). The range of fruit output for the cooperative was 4.3 to 9.5 tonne/tunnel-day (9480 to 20945 lb/tunnel-day) corresponding to fuel usages of 4849 MJ/tonne (2086 Btu/lb) and 3619 MJ/tonne (1557 Btu/lb). This variation is caused by varying amounts of fruit on the trays and in some cases by shutting down the dryer because of insufficient fruit deliveries to the drying facility. Bringing the output of the lowest up to the highest rate would result in an energy savings of 25%.

This effect can be explained by separating energy use that is associated with fruit output from energy use which is a function of time. The energy budget indicated that about one-half of the total energy use is for evaporating water. This use will increase as fruit output increases. Othe energy uses such as heat loss through walls, through heated trays and fruit leaving the tunnel, hot air leaving the tunnel and some burner losses are a function of hours of operation. Since increasing fruit output (primarily by increasing the amount of fruit on the drying trays) does not appreciably increase drying times, it will result in proportionately lower energy use for the time dependent energy uses.

Heat Loss Through Tunnel Surfaces

Heat lost through the walls and roof of a concrete tunnel was estimated as 18.5 KW (63,000 Btu/h). As indicated in Table 2 this is a small proportion of total heat input of 715 KW (Table 3). Tunnels (location 2, Table 1) constructed of transite (asbestos-cement board) have the potential of losing from 5% to 8% of the total energy consumption through the roof and walls. This heat loss can be reduced through the use of added insulation and by adding an extra layer of transite suspended at least a half inch below the roof in the area of the flame. This added layer of transite will prevent the flame from radiating heat to the roof and causing excessive heat loss.

In the 1930's and 1940's many "Miller" type prune dehydrators (Fig. 2) were built in California. These tunnels have the burner assembly located at the back end of the tunnel. The burner is located immediately behind a large steel plate which forms a portion of the rear wall. This steel plate gets very hot and becomes a large source of heat loss.

An experiment was performed on such a tunnel where a heat shield was placed behind the rear wall. The heat shield was constructed of three layers of expanded metal each separated by about an inch. This device shielded the rear steel plate from convective heat losses and absorbed radiant heat from the steel plate transferring it to the air that was passing through the shield into the dehydrator. With the shield in place the gas consumption was reducted by 10%.

Burner and other Losses

Table 2 indicates that 12% of the heat input was lost at the burner or was unaccounted for. Major proportion of such losses may be due to incomplete combustion of the gas, formation of water during combustion and radiant losses from a flame partially exposed to the outside. The efficiency data was generated by calculating the difference between the total heat input and all measured energy uses, the remainder was considered to be equal to the losses indicated here. It is to be noted that the total energy (heat) input into the system was calculated from the amount of fuel (natural gas) consumption and high/gross heat value of the fuel. Gross heat value includes the heat of formation of water during combustion. Gross and net heat values for methane are reported to be 4.581 MJ/m^3 (1013 Btu/ft^3) and 4.129 MJ/m^3 (913 Btu/ft^3) respectively (Perry and Chilton 1973). It is known that natural gas mainly consists of methane, thus, approximately 10% of the total energy input may be associated with the formation of water. As a result it is only 2% of the losses which were not accounted for. No measurements were made of the products of combustion (CO_2, CO, O_2, hydrocarbons), to indicate incomplete burning, because of the large amount of excess combustion air in the system. It is believed that properly installed and maintained burners should further reduce these losses.

Concurrent versus Countercurrent Operation

The data points in Fig. 4 indicated no significant difference between the two types of operation. This observation confirms the data reported by McRae (1951) which indicated that a selected group of countercurrent dehydrators in operation from 1935 to 1950 had an average efficiency of moisture removal of 45%. The average efficiency of moisture removal for the dehydrator cooperative during 1975-1977 was also equal to 45% with only 10% of the sampled tunnels operated in a countercurrent manner.

SUMMARY

The key to energy conservation is good management, proper maintenance of tunnels, and tray loading. The existing types of driers can operate efficiently if they are well maintained and properly operated. The list given below summarizes various techniques which can be employed to conserve energy in tunnel dehydrators and the anticipated fuel savings from those techniques:

Technique	*Fuel Savings*
Increased air recirculation (especially by adding doors to the exhaust end of the tunnel)	at least 15%
Fully loaded trays	0-25%
Use of a heat shield on "Miller" type tunnels	10%
Enclosure of motor well, sealing air leaks	8%
Properly installed and maintained burner	0-2%
Insulation of roof and use of radiant heat shield below roof on transite tunnels	3-5%
Properly installed baffles when doors are not installed on the exit.	0-4%

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