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Energy consumption during Refractance Window® evaporation of selected berry juices

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SUMMARY

The Refractance Window® evaporator represents a novel concept in the design of evaporation systems for small food processing plants. In this system thermal energy from circulating hot water is transmitted through a plastic sheet to evaporate water from a liquid product flowing concurrently on the top surface of the plastic. The objectives of this study were to investigate the heat transfer characteristics of this evaporator, determine its energy consumption, and capacity at different tilt angles and product flow rates. The system performance was evaluated with tap water, raspberry juice, and blueberry juice and puree as feed. With a direct steam injection heating method, the steam economy ranged from 0.64 to 0.84, while the overall heat transfer coefficient (U) was 666 W m^{-2} °C^{-1}. Under this condition, the highest evaporation capacity was 27.1 kg h^{-1} m^{-2} for blueberry juice and 31.8 kg h^{-1} m^{-2} for blueberry puree. The energy consumption was 2492–2719 kJ kg^{-1} of water evaporated. Installation of a shell and tube heat exchanger with better temperature control minimized incidences of boiling and frequent discharge of condensate. The steam economy, highest evaporation rate and overall heat transfer coefficient increased to 0.99, 36.0 kg h^{-1} m^{-2} and 733 W m^{-2} °C^{-1}, respectively. Copyright © 2004 John Wiley & Sons, Ltd.

KEY WORDS: falling liquid film; steam economy; energy consumption; blueberry; raspberry; juice

1. INTRODUCTION

Intensive use of thermal energy and loss of product quality of heat sensitive liquid foods are two major concerns in the design and operation of evaporators used in the food industry. Multi-stage evaporators have been developed that operate at sub-atmospheric pressures to increase energy efficiency and reduce thermal degradation of products (APV CREPACO, 1992). Those systems are, however, very expensive and operate well only within a narrow range of fluid viscosity. The Refractance Window® (RW) evaporator, developed by MCD Technologies Inc., is relatively simple, inexpensive and can be used with a diverse range of products, including those with high sugar and pulp content. It basically uses hot water at normal atmospheric

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pressure to evaporate water from liquid foods. Small fruit and vegetable processors who cannot afford expensive capital investments in multi-effect vacuum evaporators may find this equipment very useful. In addition to concentrating food products, the RW evaporator is versatile and has potential for use in processing nutraceutical, pharmaceutical, biotechnology, and chemical products, as well as in waste recovery and remediation for a diverse range of industries.

Before a new type of food processing equipment is commercialized, it is important to conduct adequate experiments to evaluate its performance. For an evaporator, the measures of performance usually include steam economy (energy efficiency), amount of water it can evaporate per hour (i.e. its capacity), operating temperature range, and nature or type of products it can handle. The procedures for determining energy use and other performance indices for vacuum evaporators are outlined by Minton (1986), Rumsey (1986), and Chen and Hernandez (1997). Similar performance characteristics are needed for the RW evaporator in which the energy for evaporation is obtained from hot water.

In the present configuration of RW evaporator (Figure 1), the water that acts as the heat source is first heated by directly injecting steam into a water tank. The water is heated to the temperature required for the process, usually a few degrees below boiling point (96–98°C), and maintained at that level to avoid formation of bubbles that would reduce heat transfer to the product. The hot water is then circulated beneath a transparent plastic with the product flowing concurrently on the upper part of the inclined flat surface of the plastic. The energy from the hot water is transmitted through the plastic sheet for heating and evaporation of water from a liquid product that makes a number of passes through the evaporator until the desired concentration is reached. Compared to conventional falling film evaporators, the design of RW evaporator makes it possible to attain higher solids concentration without fouling of the evaporation surface. The plastic surface only needs cleaning at the end of the process or whenever a different product is to be processed.

The objective of this study was to determine the performance characteristics of the RW evaporator, and particularly to document its energy consumption per unit weight of water evaporated, heat transfer per unit of surface area, evaporation capacity at different tilt angles, and ways of improving the system.

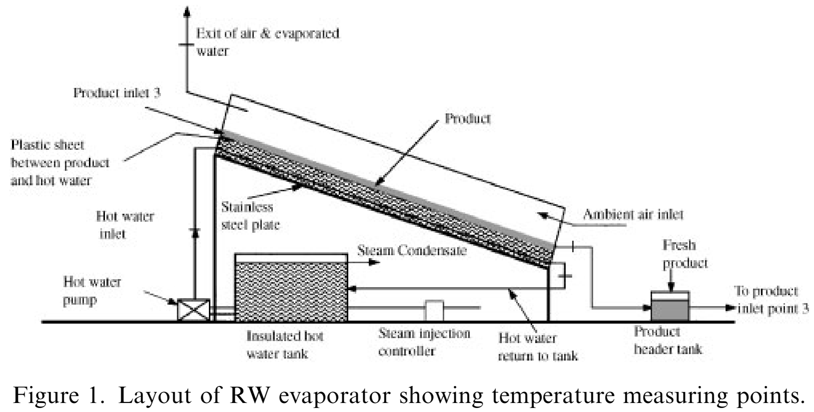

Figure 1. Layout of RW evaporator showing temperature measuring points.

2. THEORETICAL CONSIDERATIONS ON HEAT AND MASS BALANCES IN RW EVAPORATOR

The evaporator capacity, which is a measure of the amount of water evaporated per hour, can be estimated from the overall mass balance and dry matter balance:

$$\boldsymbol{m}_{\mathrm{p,in}} = \boldsymbol{m}_{\mathrm{p,out}} + \boldsymbol{m}_{\mathrm{w}} \tag{1}$$

$$\boldsymbol{m}_{\mathrm{p,in}} \times \boldsymbol{x}_{\mathrm{in}} = \boldsymbol{m}_{\mathrm{p,out}} \times \boldsymbol{x}_{\mathrm{out}} \tag{2}$$

where $\boldsymbol{m}_{\mathrm{p,in}}$ and $\boldsymbol{m}_{\mathrm{p,out}}$ are the mass flow rates ($\mathrm{kg\,s^{-1}}$) of fluid product at the inlet and outlet of evaporator, respectively; $\boldsymbol{m}_{\mathrm{w}}$ is evaporation rate ($\mathrm{kg\,s^{-1}}$); $\boldsymbol{x}_{\mathrm{in}}$ and $\boldsymbol{x}_{\mathrm{out}}$ are the solids concentration (decimal) at the inlet and outlet points, respectively.

Combining Equations (1) and (2) yields:

$$\boldsymbol{m}_{\mathrm{w}} = \boldsymbol{m}_{\mathrm{p,in}} \left(1 - \frac{\boldsymbol{x}_{\mathrm{in}}}{\boldsymbol{x}_{\mathrm{out}}}\right) \tag{3}$$

By measuring the solids content $\boldsymbol{x}_{\mathrm{out}}$ during the evaporation process, the evaporation capacity ($\boldsymbol{m}_{\mathrm{w}}$) at different elevations and product flow rates can be calculated (Figure 2).

The rate of heat transfer per unit surface area is an important parameter that is necessary for evaluating the performance of evaporators. To determine the heating rate, the overall heat transfer coefficient (U), the effective evaporation surface (A) and the temperature difference (LMTD) between the hot and cold fluids must be known. The relevant equation is

$$\boldsymbol{Q}_1 = \boldsymbol{U}\boldsymbol{A}(\mathrm{LMTD}) \quad \text{and} \quad \mathrm{LMTD} = \frac{\Delta \boldsymbol{T}_{\mathrm{in}} - \Delta \boldsymbol{T}_{\mathrm{out}}}{\ln(\Delta \boldsymbol{T}_{\mathrm{in}}/\Delta \boldsymbol{T}_{\mathrm{out}})} \tag{4}$$

where $\boldsymbol{Q}_1$ is the energy for evaporation and sensible heating of product (kW); $\Delta \boldsymbol{T}_{\mathrm{in}}$ the temperature difference between circulating hot water and liquid product at the inlet (°C); $\Delta \boldsymbol{T}_{\mathrm{out}}$ the temperature difference between circulating hot water and liquid product at the outlet (°C).

Apart from pre-heating of juice to evaporation temperature, thermal energy from steam condensing in the circulating hot water or within the heat exchanger is used mostly for evaporating water from the product. The energy supplied by saturated steam is the same as the amount released when the steam condenses. Therefore, for steady state conditions, the latent heat of condensation ($\boldsymbol{Q}_{\mathrm{s}}$) is utilized for heating the product to evaporation temperature ($\boldsymbol{Q}_{\mathrm{p}}$) and evaporating water from the product ($\boldsymbol{Q}_{\mathrm{w}}$), with the remainder constituting thermal losses to the surrounding ($\boldsymbol{Q}_{\mathrm{L}}$). Most of these losses occur through the bottom stainless-steel plate supporting the plastic. If the condensate that normally circulates back to the boiler hot well is

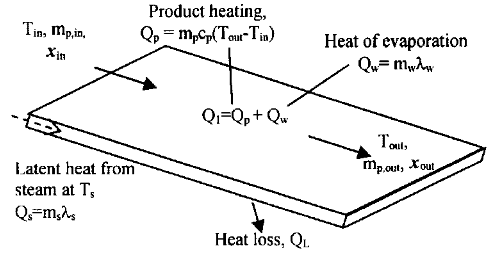

Figure 2. Representation of heat and mass transfer within the evaporator.

collected and weighed, then $\boldsymbol{Q}_{\mathrm{L}}$ also includes the enthalpy in discharged condensate. Therefore,

$$\boldsymbol{Q}_{\mathrm{s}} = \boldsymbol{Q}_{\mathrm{p}} + \boldsymbol{Q}_{\mathrm{w}} + \boldsymbol{Q}_{\mathrm{L}} \tag{5a}$$

where

$$\boldsymbol{Q}_{\mathrm{p}} = \boldsymbol{m}_{\mathrm{p}}\boldsymbol{c}_{\mathrm{p}}\Delta \boldsymbol{T}_{\mathrm{p}}, \;\; \boldsymbol{Q}_{\mathrm{w}} = \boldsymbol{m}_{\mathrm{w}}\lambda_{\mathrm{w}}, \;\text{ and }\; \Delta \boldsymbol{T}_{\mathrm{p}} = \boldsymbol{T}_{\mathrm{p,out}} - \boldsymbol{T}_{\mathrm{p,in}} \tag{5b}$$

For fruit juices, the specific heat capacity ($\boldsymbol{c}_{\mathrm{p}}$, kJ/kg°C) and latent heat of evaporation (λ_{w}, J/kg) at a known temperature $\boldsymbol{T}_{\mathrm{p}}$(°C) are given by the following equations (Rao and Vitali, 1999; Chen and Hernandez, 1997):

$$\boldsymbol{c}_{\mathrm{p}} = 4.187\{1 - \boldsymbol{x}_{\mathrm{in}}(0.57 - 0.0018(\boldsymbol{T}_{\mathrm{p}} - 20))\} \tag{5c}$$

$$\lambda_{\mathrm{w}} = 2499\ \mathrm{Exp}(-0.001016\boldsymbol{T}_{\mathrm{p}}) \tag{5d}$$

After determining $\boldsymbol{Q}_{\mathrm{p}}$ and $\boldsymbol{Q}_{\mathrm{w}}$, it is possible to calculate the heat transfer coefficient, U. The value of the product side heat transfer coefficient $\boldsymbol{h}_{\mathrm{p}}$ is a measure of how efficiently the latent heat from steam is transferred to the product being concentrated. Given that the temperature of the plastic surface on the hot water side is nearly the same as the bulk water temperature, the overall resistance to heat transfer between the circulating hot water and the vapor bulk is then provided by the plastic sheet and fluid food being concentrated. The value of $\boldsymbol{h}_{\mathrm{p}}$ depends on the superficial velocity of the liquid product and its physical and thermal properties such as viscosity and thermal conductivity.

$$\frac{1}{\boldsymbol{UA}} = \frac{1}{\boldsymbol{h}_{\mathrm{w}}\boldsymbol{A}} + \frac{\delta}{\boldsymbol{kA}} + \frac{1}{\boldsymbol{h}_{\mathrm{p}}\boldsymbol{A}} \tag{6}$$

The Mylar® plastic sheet used has a thermal conductivity of $0.155\,\mathrm{W\,mK^{-1}}$, a thickness of 0.2 mm, and covered a surface area of $6.64\,\mathrm{m}^2$. The resistance to heat transfer by conduction across the plastic sheet is fixed. Therefore, the magnitude of U depends on the combined radiation and convective heat transfer on both sides of the plastic sheet. From the above relationships, it is possible to calculate the ratio of energy used for evaporating water from product ($\boldsymbol{Q}_1$) to the net thermal energy supplied by condensing steam ($\boldsymbol{Q}_{\mathrm{s}}$). The evaporator performance can be expressed in terms of steam economy, i.e. the amount of water evaporated per kilogram of steam consumed, namely:

$$\text{Steam economy} = \frac{\boldsymbol{m}_{\mathrm{w}}}{\boldsymbol{m}_{\mathrm{s}}} = \frac{(\boldsymbol{m}_{\mathrm{p,in}} - \boldsymbol{m}_{\mathrm{p,out}})}{\boldsymbol{m}_{\mathrm{s}}} \tag{7}$$

3. MATERIALS AND METHODS

Experiments were conducted with water (to establish baseline parameters), raspberry juice, blueberry juice and blueberry puree after diluting the commercially available concentrates to about 10% solids content. The raspberry juice concentrate was supplied by Milne Fruit Products Inc. (Yakima, WA) while blueberry products were obtained from Valley Processing Inc. (Sunnyside, WA) courtesy of Overlake Foods Corp. (Olympia, WA). The berry juice concentrates and puree were shipped overnight to MCD Technologies Inc. (Tacoma, WA), allowed to thaw, then mixed and diluted as required for the RW evaporation tests.

Steam for heating the circulating water was supplied from a boiler (Steam generator model E-40, Clayton Industries, CA) and either directly injected into water contained in an insulated 200-gallon tank or passed through a shell and tube heat exchanger. The latter had a length of 0.95 m with shell diameter of 0.21 m. A PID controller that responded to the set water temperature in the tank was used to regulate the steam supply to both heating systems. For the direct steam injection, the water overflowing from an outlet pipe near the top of the tank was collected and weighed to determine the quantity of steam supplied. The condensate discharged from the shell and tube heat exchanger was measured similarly. The circulating water was heated to a temperature of about 97°C before starting product circulation. For easy adjustment of product flow rates, an electronic speed control was connected to a 3HP positive displacement rotary lobe pump (Waukesha Cherry-Burrell, Delavan, WI). During calibration, with speed settings at 184, 169 and 150 rpm, the pump respectively discharged 160, 147, and 130 kg of diluted product per minute. Besides the setting of feed pumping rates, experiments were done with the evaporator tray tilted at 24, 30, and 37° from the horizontal position. The rate of evaporation was measured by recording the decrease in the weight of product contained in a 200 kg capacity barrel, which was placed on a platform scale. Another small compressed air pump actuated by a float device in the evaporator header tank intermittently pumped more of the dilute feed product into the header tank to maintain the product level. To measure the evaporation rate, the decrease in volume in the supply tank was recorded until all the product was emptied into the header tank. At this point, the constant level in the header tank could no longer be sustained.

The total solids content of juice was measured using automatic temperature compensating type hand refractometers (Atago ATC-1E and ATC-2E for brix ranges $0 \sim 32\%$ and $28 \sim 62\%$, respectively). For higher total solids contents, concentrates were diluted before the measurement of brix. The brix number of the concentrates was then calculated from the dilution factor and measured brix of the diluted samples. For temperature measurements, type-T thermocouples were connected to a data logger (Model 21X, Campbell Scientific Inc., Logan, UT) and their output relayed to a computer to display the temperature profiles of product and circulating hot water at various points (Figure 1). The overall heat transfer coefficient, energy consumption and evaporator capacity were determined from the recorded temperature and flow rate data.

4. RESULTS AND DISCUSSION

4.1. *Baseline study with tap water as feedstock*

Table I shows the results of evaporation tests that were conducted on a prototype production machine using tap water as feedstock at a fixed evaporator tilt of 37°. Each of those experiments lasted 1.0 h. The quantity of water evaporated from the 6.64 m^2 surface ranged from 131 to 158 $\mathrm{kg\,h^{-1}}$ (19.7–23.8 $\mathrm{kg\,h^{-1}\,m^{-2}}$), while the energy consumption was from 2533–2787 $\mathrm{kJ\,kg^{-1}}$ of water evaporated. Modification of the air handling system and increase in circulating hot water flow rate increased the evaporation rate from 143 to 173 $\mathrm{kg\,h^{-1}}$ (21.5–26.0 $\mathrm{kg\,h^{-1}\,m^{-2}}$) (Tables I & II(a)). Further adjustment of the airflow pattern within the evaporator using a baffle reduced the splashing that occurred on the top cover and the inner walls. The amount of evaporated water condensing on the top cover and falling back onto the feedstock was also minimized. Since the flow direction of air is opposite to that of fluid, residence time of fluid product increased at higher airflow, leading to more evaporation. This countercurrent flow of air against the inclined evaporation surface created some beneficial turbulence and ripples on the fluid product. Zheng and Worek (1996) showed the positive influence of such ripples on heat and mass transfer in thin film evaporation by using equally spaced agitated glass rods on an inclined

Table I. Capacity and energy consumption of a prototype RW evaporator using water as feedstock (evaporator tilt angle: 37°, hot water flow: $0.9\,\text{kg}\,\text{s}^{-1}$).

			Heat transfer (kW)				
Feed circulation rate (rpm)	Evaporation rate ($\text{kg}\,\text{h}^{-1}$)	Condensate flow ($\text{kg}\,\text{h}^{-1}$)	Water evaporation	Product heating	Steam supply	Evaporation heat ($\text{kJ}\,\text{kg}^{-1}$)	Steam economy
184	145	195	94.5	12.1	112	2778	0.74
184	131	181	85.1	10.8	101	2787	0.72
184	137	186	89.0	11.1	105	2771	0.73
184	144	181	94.1	10.7	101	2533	0.79
184	158	200	103.3	12.3	112	2538	0.79
Mean	143 ± 7	189 ± 7	93 ± 5	11.4 ± 0.6	106 ± 5	2681 ± 117	0.76 ± 0.03

Table II. Capacity and energy consumption of RW evaporator after modification of airflow system (evaporator tilt angle: 37°, hot water flow: $6.8\,\text{kg}\,\text{s}^{-1}$).

			Heat transfer (kW)				
Feed circulation rate (rpm)	Evaporation rate ($\text{kg}\,\text{h}^{-1}$)	Condensate flow ($\text{kg}\,\text{h}^{-1}$)	Water evaporation	Product heating	Steam supply	Evaporation heat ($\text{kJ}\,\text{kg}^{-1}$)	Steam economy
(a) *Tap water as feedstock*							
184	169	225	109	20	147	2778	0.75
184	170	250	111	18	163	2727	0.68
184	174	234	114	22	152	2811	0.75
169	177	230	116	18	150	2828	0.77
Mean @184 rpm	173 ± 4	235 ± 8	113 ± 3	20 ± 2	153 ± 5	2746 ± 49	0.74 ± 0.03
(b) *Blueberry juice and blueberry puree as feedstock*							
184‡	139	194	90	15	127	2571	0.72
184‡	148	192	96	14	124	2557	0.77
169‡	135	185	88	13	121	2685	0.73
150‡	126	183	82	14	120	2729	0.69
172 ± 12‡	137 ± 7	189 ± 5	89 ± 4	14 ± 1	123 ± 3	2636 ± 72	0.73 ± 0.03
184§	211	274	137	19	179	2492	0.77
(c) *Raspberry juice as feedstock*							
184	132	169	85.8	18.4	95	2601	0.78
184	124	171	81.2	18.0	97	2796	0.73
184	154	212	100.7	15.7	118	2761	0.73
Mean	137 ± 12	184 ± 19	89.2 ± 7.6	17.4 ± 1.1	103 ± 10	2719 ± 79	0.75 ± 0.02

Feed type: ‡blueberry juice; §blueberry puree.

stainless-steel evaporation surface. The rods produced waves or eddies that increased the heat and mass transfer several times. For the RW evaporator, a balance is needed between fluid turbulence and the amount of splashing that is acceptable. Despite the good results of airflow adjustments, the direct steam injection system used during these baseline studies was not very efficient. The water temperature was difficult to control leading to supply of more steam than was necessary. The energy consumption therefore increased, causing a slight reduction in steam economy (Table II(a)). In later experiments, a new controller was installed and the direct steam injection system replaced with the shell and tube heat exchanger.

4.2. Evaporation of blueberry and raspberry juices

Blueberry and raspberry juices are usually concentrated to 65°Brix in commercial operations, while their puree counterparts are limited to about 28°Brix. The change in total solids content and viscosity of fruit juices and purees has an influence on the energy usage during evaporation. For the RW evaporator, the tilt angle, temperature and flow of both fluid product and circulating hot water, may also influence the evaporator performance. To evaluate the performance of the evaporator, temperatures at different points (Figures 3 and 4), evaporation rates (Figure 5), and steam economy (Figure 6) were plotted for different tray elevations and product flow rates.

Figures 3 and 4 are typical temperature profiles showing the effect of circulating water flow rate on fluid temperatures during evaporation of blueberry juice. At a water flow rate of $0.9\,\mathrm{kg\,s^{-1}}$, about 18–20°C temperature difference between hot water at inlet and outlet points was observed (Figure 3). However, when the water flow rate was increased to $6.8\,\mathrm{kg\,s^{-1}}$, the temperature of circulating hot water dropped only by an average of 3.2°C. For both cases of low and high water flow rates, the product temperature at the inlet and outlet differed by less than 1.2°C. As water in the product changes into vapor, it causes product cooling which prevents any significant product temperature rise between the inlet and outlet points. Tsay and Lin (1995) observed that liquid vaporization tends to be high at higher product inlet temperature. Since the product temperature in an evaporator operating at atmospheric conditions mainly depends on the heating medium temperature, it is advantageous to have the highest possible product temperature that does not lead to quality degradation.

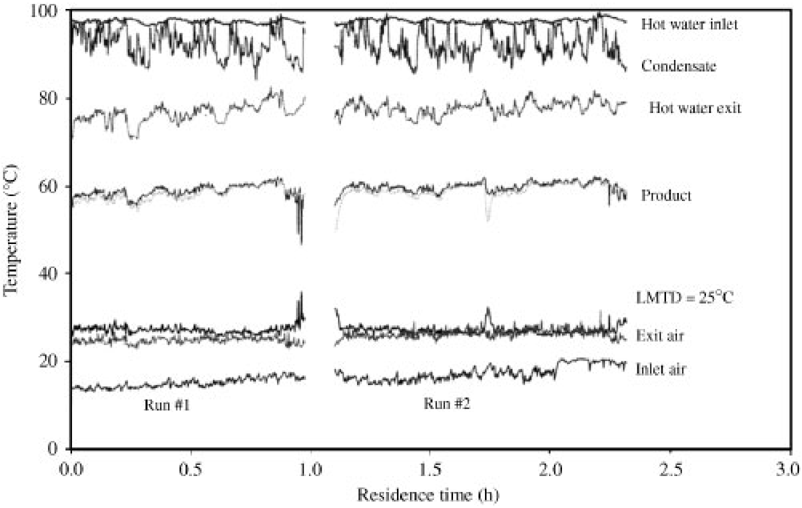

Figure 3. Typical temperature profiles during evaporation of blueberry juice under direct steam injection heating with water circulating at $0.9\,\mathrm{kg\,s^{-1}}$.

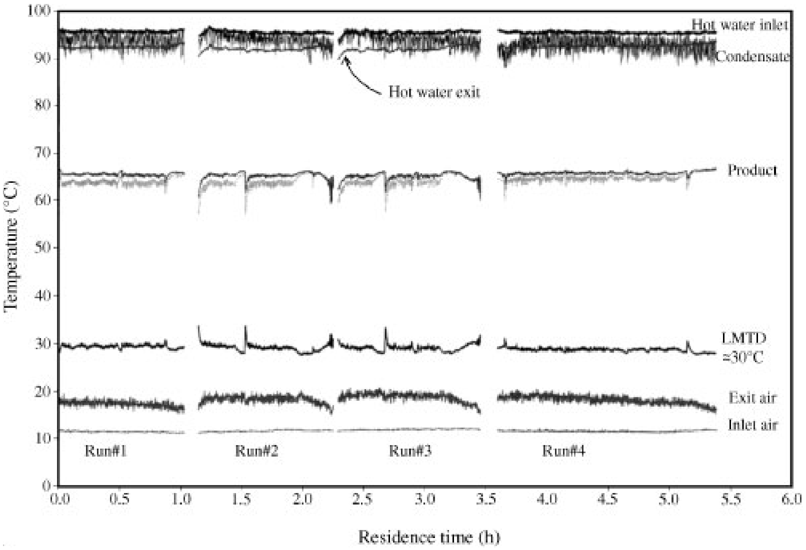

Figure 4. Typical temperature profiles during evaporation of blueberry juice under direct steam injection heating with water circulating at 6.8 $\mathrm{kg\,s^{-1}}$.

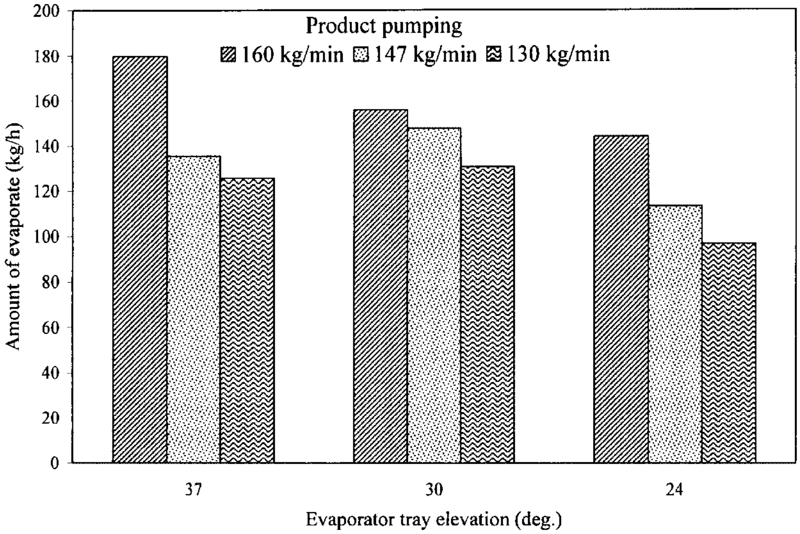

Figure 5. Evaporation rate for blueberry juice at different evaporator tilt angles and juice flow rates under direct steam injection heating.

With stable thermal input conditions, the log mean temperature difference (LMTD), defined in Equation (3) and plotted in Figures 3 and 4, can be used to monitor the performance of the evaporator. The temperature profiles indicate that LMTD increased from 25°C to about 30°C when the circulating hot water flow rate was increased from 0.9 to 6.8 $\mathrm{kg\,s^{-1}}$, respectively. This increase in LMTD as a result of increase in water circulation rate might have contributed to the high evaporation (Table I and II(a)). To maintain the high evaporation rate, the rest of the

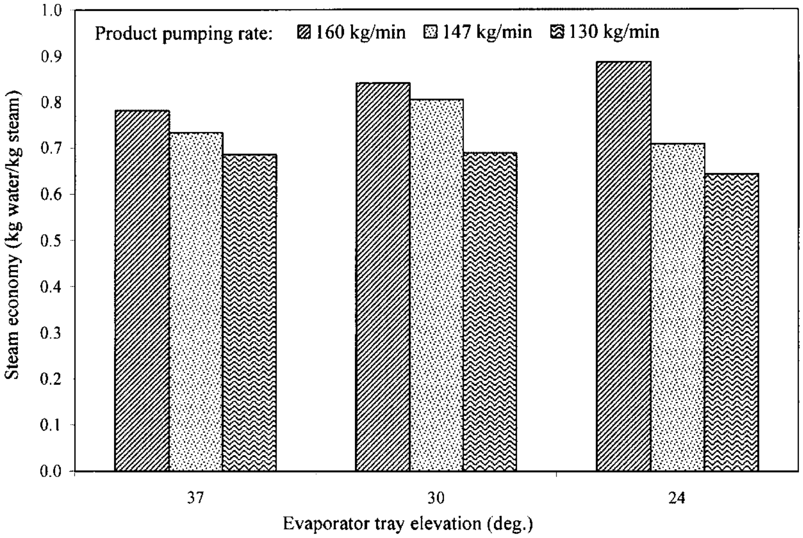

Figure 6. Quantity of water evaporated from blueberry juice per kilogram of steam used under direct steam injection heating.

experiments with berry products were conducted with hot water circulating at 6.8 kg s^{-1}. Yan and Soong (1995) investigated heat and mass transfer along an inclined heated plate with film evaporation and reported that a reduction in the inclined angle causes an increase in the air-liquid interfacial temperature, which in turn leads to a larger latent heat exchange. In the present study, the very high ratios of feed flow to evaporation rates might have masked the influence of tilt angle on the interfacial temperature. Apart from inclined angle, the viscosity and total solids content of fluid product, and the heat capacity rates of the fluids flowing on both sides of the plastic sheet might also influence the heat and mass transfer. For the three evaporator bed tilt angles used in this study, evaporation rates were higher at higher product circulating rates (Figure 5). The foregoing observations suggest that higher circulating water flow rate is preferable because it results in higher interfacial air–liquid temperature which increases the latent heat flux. The magnitude of the evaporative latent heat flux we obtained is several times that of the sensible heat flux (Table IV). This indicates efficient conversion of latent heat from condensing steam into heat flux for evaporation and agrees with conclusions made by Yan and Soong (1995).

When the evaporator tray was tilted at 37° from the horizontal with hot water circulating at 6.8 kg s^{-1}, the average energy required to evaporate 1 kg of water from blueberry juice, blueberry puree and raspberry juices were 2636, 2492 and 2719 kJ, respectively (Table II(b) and (c)). Commercial fruit juice concentrates are usually at higher brix than their puree counterparts, so the high amount of dissolved solids binds the water molecules more tightly resulting in a rise in boiling temperature. At a higher circulating water flow rate, the water temperature is nearly invariable and since the boiling point rise of the higher brix juice is more than that of puree, slightly more energy was used in the evaporation of blueberry juice to the higher brix than in the puree. With the direct steam injection heating method, the highest evaporation rates recorded during the concentration of blueberry juice and blueberry puree were 180 and 211 kg h^{-1} (i.e. 27.1 and 31.8 kg h^{-1} m^{-2}), respectively (Tables II(b) and III(a)). After the direct steam injection heating system was replaced with shell and tube heat exchanger, the highest evaporation rate for blueberry juice was 239 kg h^{-1} (38.0 kg h^{-1} m^{-2}) (Table III(b)).

The average mass of water evaporated per unit mass of steam used (steam economy) was 0.64–0.84 with direct steam injection heating method (Figure 6). These figures improved to 0.80–0.99 after the shell and tube heat exchanger was installed (Table III(b)). The direct steam injection is by design a more efficient heating method, but during this study it was difficult to control the temperature to avoid boiling of circulating water. The boiling led to enthalpy loss in discharged hot water that could have been used for the evaporation process. Therefore, replacement of the direct steam injection with the shell and tube heat exchanger reduced the wastage of circulating hot water and improved steam economy. The results of steam economy for both heating systems are very representative of the overall system performance since the data for each combination of tilt angle and product pumping speed were obtained for residence times of more than 1 h after reaching steady state operation condition. Aboabboud *et al.* (1996) analysed an atmospheric evaporator that included thermal energy recycling and obtained a steam economy of 2.83. Budin *et al.* (1998) reported a steam economy of 0.91 for a single effect vacuum evaporation process, while the 2- and 3-effect tomato paste evaporators investigated by Rumsey (1986) had economies between 1.38 and 2.60. Fellows (1988) reported steam economy values between 1.67 and 3.33 for one- to three-effect vacuum evaporators with vapor recompression, and 0.91 to 2.5 for those without vapor recompression. The steam economy obtained for the RW evaporator with shell and tube heat exchanger was from 0.80–0.99. Considering that the RW evaporator is operated at atmospheric conditions, its steam economy is comparable to 0.7–0.9 reported by Aboabboud *et al.* (1996) for evaporators without thermal energy recycling. Though operating it at higher inclines facilitates gravity flow of fluid product on the evaporation surface, steam economy values at 37 and 30° tilt angles did not appear different (Figure 6).

Table III.

Tilt angle	Feed circulation rate (rpm)	Evaporation rate (kg h^{-1})	Condensate flow (kg h^{-1})	Evaporation heat (kJ kg^{-1})	Steam economy	LMTD (°C)	Coeff. U (W m^{-2} °C^{-1})
(a) *Steam economy and energy for evaporation of blueberry juice at different tilt angles and feed pumping rates (direct steam injection system)*[‡]							
37°	184	180	230	2668	0.78	30.1	666
37°	169	136	185	2685	0.73	28.7	530
37°	150	126	183	2729	0.69	29.6	485
30°	184	156	186	2646	0.84	29.3	590
30°	169	148	184	2612	0.81	29.1	555
30°	150	131	190	2549	0.69	29.1	479
24°	184	144	163	2606	0.89	28.9	543
24°	169	114	161	2609	0.71	28.8	430
24°	150	97	151	2611	0.64	28.8	367
(b) *Performance of the evaporator after replacing steam injection system with shell and tube heat exchanger*[‡]							
37°	184	196	246	2718	0.80	36.0	619
37°	150	223	245	2587	0.91	35.3	685
30°	184	239	253	2598	0.95	35.4	733
30°	150	225	249	2618	0.91	35.3	699
24°	184	234	253	2624	0.92	35.2	728
24°	150	228	229	2624	0.99	34.8	722

[‡] Blueberry juice used as feedstock.

Table IV. Heat quantities used in calculation of heat transfer during evaporation of blueberry juice (Equations (5) and (6))[‡].

		Direct steam injection			Shell & tube heat exchanger		
Tilt angle	Feed circulation rate (rpm)	Product heating, Q_p	Evaporation Heat, Q_w	Heat from steam, Q_s	Product heating, Q_p	Evaporation Heat, Q_w	Heat from steam, Q_s
37°	184	16.4	116.7	150.0	19.4	128.4	114.5
37°	169	13.0	88.4	120.7	—	—	—
37°	150	13.6	81.7	119.7	14.3	146.1	116.8
30°	184	13.3	101.4	121.2	16.1	156.4	116.3
30°	169	11.2	96.0	119.8	—	—	—
30°	150	7.6	84.9	123.9	16.4	147.3	116.1
24°	184	10.7	93.8	106.2	17.4	152.7	116.0
24°	169	8.5	73.7	104.7	—	—	—
24°	150	7.3	62.9	98.3	17.0	148.9	114.1

[‡] Heating rates in kW.

The other measure of performance used in this study is overall heat transfer coefficient (*U*). which ranged from 367 to 666 $\mathrm{W\,m^{-2}\,°C^{-1}}$ and 619 to 733 $\mathrm{W\,m^{-2}\,°C^{-1}}$ for the direct steam injection and for the shell and tube heat exchanger, respectively (Table III). These values generally increased with product flow rate. However, the figures obtained are in the lower range when compared to 1930 $\mathrm{W\,m^{-2}\,°C^{-1}}$ reported for a 4-effect, 6-stage orange juice vacuum evaporator investigated by Rao and Vitali (1999). For a thin film scraped surface evaporator (TFSSE), Sangrame *et al.* (2000) obtained *U* values between 477 and 939 $\mathrm{W\,m^{-2}\,°C^{-1}}$ while using water as feed, and 626 and 911 $\mathrm{W\,m^{-2}\,°C^{-1}}$ with tomato pulp. The corresponding evaporation rates reported for the TFSSE were 14.7–30.7 and 13.2–33.7 $\mathrm{kg\,h^{-1}}$, respectively. This is an indication that the overall heat transfer coefficient of the RW evaporator is very good when compared to these vacuum systems.

5. CONCLUSION

The amount of water evaporated per unit weight of steam consumed ranged from 0.64 and 0.84, and the highest evaporation rate recorded with the direct steam injection method of heating was 211 $\mathrm{kg\,h^{-1}}$ (31.8 $\mathrm{kg\,h^{-1}\,m^{-2}}$). After the air volume and flow pattern in the system were improved, the average evaporation capacity increased from an average of 141 to 163 $\mathrm{kg\,h^{-1}}$ (21.2–24.5 $\mathrm{kg\,h^{-1}\,m^{-2}}$). Installation of a shell and tube heat exchanger further increased the capacity and steam economy to between 196 and 239 $\mathrm{kg\,h^{-1}}$ (29.5–36.0 $\mathrm{kg\,h^{-1}\,m^{-2}}$) and 0.80–0.99, respectively. During the concentration of blueberry puree, blueberry juice and raspberry juice, the average product temperature was 65°C, and 2492–2719 kJ of energy was used per kilogram of water evaporated. The overall heat transfer coefficient for the system increased with increase in circulating water flow rate, product flow rate, and evaporator tilt. Highest values of heat transfer coefficient were 666 $\mathrm{W\,m^{-2}\,°C^{-1}}$ with steam injection and 733 $\mathrm{W\,m^{-2}\,°C^{-1}}$ with the shell and tube heat exchanger. These were obtained with the evaporator tray inclined at 37° and hot water circulating at 6.8 $\mathrm{kg\,s^{-1}}$. Given that the Refractance Window® evaporator is much less expensive and operates at normal atmospheric conditions, the results obtained are very good when compared to other evaporators working under similar conditions.

NOMENCLATURE

A = effective evaporation surface area (m^2)
c = specific heat capacity (kJ kg^{-1} °C^{-1})
h = heat transfer coefficient (W m^{-2} °C^{-1})
k = thermal conductivity (W m^{-1} °C^{-1})
m = mass flow rate (kg s^{-1})
x = total solids content (g g^{-1} dry^{-1} wt^{-1})
U = overall heat transfer coefficient (W m^{-2} °C^{-1})
LMTD = logarithmic mean temperature difference (°C)
RW = Refractance Window®
T = temperature (°C)
Q = heat transfer rate (kW)
δ = thickness of Mylar® plastic sheet (m)
λ_s = latent heat of condensing steam (J kg^{-1})
λ_w = latent heat of evaporation of water from product (J kg^{-1})

Subscripts

p,in; p,out = product at inlet and outlet of the evaporator
w = water or vapour
s = steam

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