

A heat pipe heat recovery heat exchanger for a mini-drier

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Abstract

This paper considers the thermal design and the experimental testing of a heat pipe (thermosyphon) heat exchanger for a relatively small commercially available mini-drier. The purpose of the heat exchanger is to recover heat from the moist waste air stream to preheat the fresh incoming air. The working fluid used was R134a and the correlations are given for the evaporator and condenser inside heat transfer coefficients as well as for the maximum heat transfer rate. The theoretical model and computer simulation program used for the thermal design calculations are described. The validity of the as-designed and manufactured heat exchanger coupled to the drier is experimentally verified. The theoretical model accurately predicted the thermal performance and a significant energy savings and a reasonable payback period was achieved.

Keywords: Heat pipe heat exchangers, thermosyphons, air drying, energy savings

Nomenclature

A	Area, m ²
Bo	Bond number, $d / \sqrt{\sigma / g(\rho_l - \rho_v)}$
c_p	specific heat (at constant pressure), J/kg°C
d	diameter, m
g	gravitational constant, 9.81 m/s ²
h	heat transfer coefficient, W/m ² °C
h_{fg}	latent heat of vaporization, J/kg
Ja	Jacob number, $c_{pl}(T_w - T_{sat})/h_{fg}$
k	thermal conductivity, W/m°C
Ku	Kutateladze number, $\dot{Q} / \left(\rho_v h_{fg} \left(\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right) \right)$
L	length, m
\dot{m}	mass flow rate, kg/s
P	pressure, Pa
R	heat transfer rate, W
R	thermal resistance, °C/W
Re	Reynolds number, $Re = 4\dot{Q}/(\pi d \mu h_{fg})$
t	time, s or h

T	temperature, °C
T_{avg}	average temperature, °C

Subscripts and superscripts

c	condenser, cold
e	evaporator, exit
h	hot
hp	heat pipe
i	inlet, inside
l	liquid
max	maximum
new	new
o	outside
v	vapour
w	water

Greek symbols

ρ	density, kg/m ³
ν	kinematic viscosity, m ² /s
ϕ	relative humidity
μ	dynamic viscosity, kg/ms
σ	surface tension, N/m

1. Introduction

In the light of an ever increasing demand for energy, the need for energy savings has become an important economic consideration. One means of saving energy is to recover a portion of the energy in a warm waste stream and then to use the recovered energy to preheat another colder stream. A heat pipe heat exchanger (HPHE) is a device capable of salvaging energy in this way (Dunn and Reay, 1994). Commercial production of HPHE began in the mid-1970s and has since found many applications, particularly in process and agricultural air-drying and the heating ventilation and air conditioning industries (Russwurm, 1980). A characteristic of all these drying operations is the need for large quantities of energy for the evaporation of water from the product and the subsequent release of

large quantities of moist hot air back into the atmosphere.

A HPHE is a liquid coupled indirect heat transfer type heat exchanger and employs a number of individually-sealed or groups of sealed heat pipes or thermosyphons as the major heat transfer means from the high temperature to the low temperature fluid. Each heat pipe is lined with a wicking structure in which a small amount of working fluid is present and can be divided into an evaporator or heat addition section and a condenser or heat rejection section. When heat is added to the evaporator section, the working fluid present in the wicking structure is heated, vaporizes and flows to the cooler section, condenses and, in so doing, giving up its latent heat of vaporization. The capillary forces in the wicking structure then pump the liquid back to the evaporator (Dunn and Reay, 1994).

Thermosyphons on the other hand are essentially heat pipes but without the wicking structure. The difference between the two is that the thermosyphon uses gravity to transfer heat from a heat source that is located below the cold sink. As a result, the evaporator section is situated below the condenser section. The working fluid evaporates, condenses in the condenser section and flows back to the evaporator section under the influence of gravity. It has been shown that in the presence of gravity, thermosyphons are preferred to heat pipes owing to the fact that the wicks in heat pipes produce an additional resistance to the flow of condensate (Pioro and Pioro, 1997). Figure 1 illustrates the principal difference between the heat pipe and the thermosyphon.

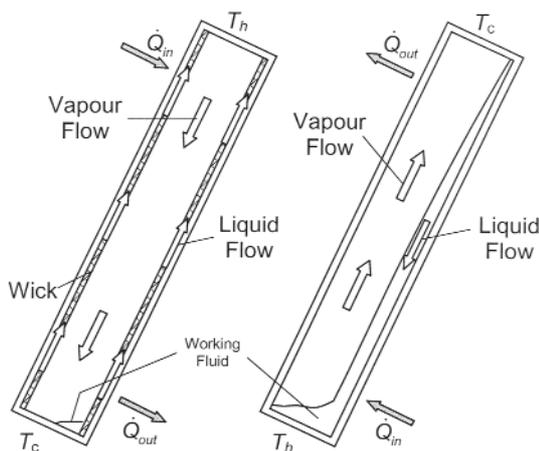


Figure 1: Principal difference between the thermosyphon and heat pipe

The use of heat pipes or thermosyphons in the heat exchanger configuration is ever-increasing in popularity and is attributed to thermosyphons having no moving parts and auxiliary working fluid pumping power requirements are therefore not needed. They can be used for gas-to-gas, gas-to-liquid and liquid-to-liquid heat exchange.

The hot and cold streams of the HPHE can also be completely isolated preventing cross-contamination of the fluids. Because of the individually sealed pipes, should one pipe fail the heat exchanger will still operate. Also, by eliminating the wick and using a commercial refrigerant as working fluid, existing standard HVAC technology is applicable. HPHEs may thus be viewed as being a commercially attractive option to a company in formulating its product range and product energy efficiency rating.

In this paper, the use of a HPHE utilising thermosyphons (instead of the more complicated wicked heat pipes) are considered to recover the waste heat from a relatively small commercially available air dryer and to use this heat to preheat the incoming cold air. (The drier is typically used for small scale drying of fruits, vegetables, herbs, meat and other products and its wet loading capacity is between 50 and 250 kg.) Heat transfer theory not available in the general literature is given with special emphasis on the inside evaporator and condenser heat transfer coefficients and the maximum heat transfer rate using R134a and Butane as working fluid. A computer program to predict the thermal performance of a HPHE is described. An economic evaluation of the potential energy savings is undertaken and experimentally validated.

2. Theory

2.1 Thermosyphon heat transfer theory

For a single two-phase closed thermosyphon, as shown in Figure 1, and for the thermal resistance diagram shown in Figure 2, heat is transferred from a heat source, through the evaporator wall, into the working fluid and then out through the condenser to the heat sink. This heat transfer rate may thus be conveniently expressed in terms of a temperature difference and the sum of a series of thermal resistances as:

$$\dot{Q} = \frac{\bar{T}_h - \bar{T}_c}{\sum R} = \dot{Q}_e = \frac{\bar{T}_h - \bar{T}_i}{\sum R_e} = \dot{Q}_c = \frac{\bar{T}_i - \bar{T}_c}{\sum R_c} \quad (1)$$

Where

$$\bar{T}_h = (T_{hi} + T_{ho})/2$$

$$\text{and } \bar{T}_c = (T_{ci} + T_{co})/2,$$

$$\sum R = R_e + R_c, \quad \sum R_c = R_{co} + R_{ew} + R_{ci}$$

$$\text{and } \sum R_c = R_{ci} + R_{cw} + R_{co}$$

$$R_{co} = \frac{1}{h_{co} A_{co}}, \quad R_{ew} = \frac{\ln(d_o/d_i)}{2\pi k L_e}$$

$$\text{and } R_{ci} = \frac{1}{h_{ci} A_{ci}}$$

$$R_{ci} = \frac{1}{h_{ci}A_{ci}}, R_{cw} = \frac{\ln(d_o/d_i)}{2\pi kL_c}$$

and $R_{co} = \frac{1}{h_{co}A_{co}}$

$$A_{co} = \pi d_o L_c, A_{ei} = \pi d_i L_c, A_{ci} = \pi d_i L_c$$

and $A_{co} = \pi d_o L_c$

Knowing the heating and cooling water inlet and outlet temperatures and the mass flow rates of the heating and cooling streams, the evaporator and condenser section heat transfer rates can be calculated in accordance with the conservation of energy as:

$$\dot{Q}_e = \dot{m}_c c_p (T_{hi} - T_{ho}) \pm \dot{Q}_{\text{loss/gain}} \quad (2)$$

and

$$\dot{Q}_c = \dot{m}_h c_p (T_{co} - T_{ci}) \pm \dot{Q}_{\text{loss/gain}} \quad (3)$$

The right hand terms $\dot{Q}_{\text{loss/gain}}$ in equations (2) and (3) account for the heat that is not transferred to the working fluid in the evaporator, and from the working fluid in the condenser, but that which is lost or gained from the environment through the heating/cooling jacket walls as well as through the structure supporting the thermosyphon.

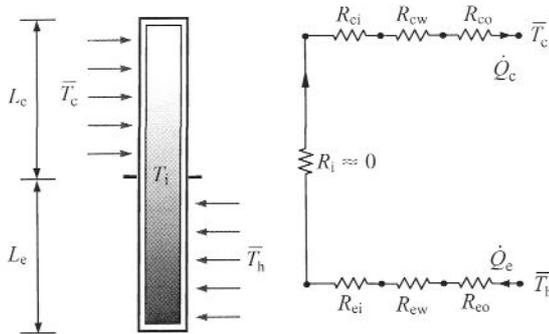


Figure 2: Thermal resistance diagram for a thermosyphon

2.2 Thermosyphon heat transfer coefficient correlations

The successful implementation in the design of a HPHE requires a detailed knowledge of the heat transfer characteristics. Correlations for the inside thermosyphon evaporator and condenser heat transfer coefficients and the maximum heat transfer rates are given by equations 4, 5 and 6 (Meyer and Dobson, 2005). These correlations were experimentally determined for inside pipe diameters varying from 15.0 to 32.0 mm and lengths of 2 to 6 m for a 50% liquid charge to evaporator section volume fill ratio, temperatures of 15 to 80°C and using R134a and a butane mixture commercially sold as

gas lighter fuel (50% n-butane, 25% iso-butane and 25% propane by mass) as working fluid.

Other heat exchanger parameters such as tube and fin spacing and outside heat transfer coefficients and pressure drops are readily available in the general heat transfer and heat exchanger literature (Mills, 1995 & Incropera and DeWitt, 2002) and hence are not given here. The maximum heat transfer rate increases significantly of about 40% (Meyer and Dobson, 2005) if a thermosyphon is inclined from the vertical. Although increasing significantly once inclined, the increase is relatively insensitive for inclination angles of 15 to 60° from the vertical. For this reason, correlations were generated only for the vertical and a 45° inclined orientation.

$$\phi = 90^\circ h_{ci} = 3.4516 \times 10^5 Ja^{0.855} Km^{1.344} \quad (4a)$$

$$\phi = 45^\circ h_{ci} = 1.4796 \times 10^5 Ja^{0.99355} Km^{1.300} \quad (4b)$$

$$\phi = 90^\circ h_{ci} = 4.61561 \times 10^9 Re_i^{0.364} \left[k_f / \left[\frac{V_f^2}{g} \left(\frac{\rho_f}{\rho_l - \rho_v} \right) \right]^{1/3} \right]^{2.05} \quad (5a)$$

$$\phi = 45^\circ h_{ci} = 3.7233 \times 10^5 Re_i^{0.136} \left[k_f / \left[\frac{V_f^2}{g} \left(\frac{\rho_f}{\rho_l - \rho_v} \right) \right]^{1/3} \right]^{1.916} \quad (5b)$$

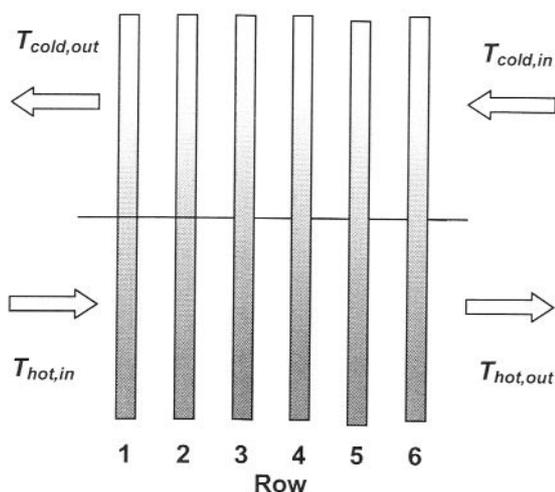
$$\phi = 90^\circ \dot{Q}_{\text{max}} = 1.6553 \times 10^6 Bo^{0.3156} Km^{1.6040} \quad (6a)$$

$$\phi = 45^\circ \dot{Q}_{\text{max}} = 7.4685 \times 10^6 Bo^{0.2101} Km^{1.9189} \quad (6b)$$

2.3 Heat exchanger thermal performance prediction procedure

Given or assuming the physical definition of the HPHE (for example, the dimensions given in Table 2), the heat transfer coefficients and thermal conductivities (h_{eo} , k_e , h_{ei} , h_{ci} , k_c , h_{co}), and the temperatures (T_{hi} , T_{ho} , T_{ci} and T_{co}), the internal temperature T_i can then be found by trial and error by guessing values for T_i in equation 1 such that ***. In a similar way, temperature and heat flux dependent variables may be taken into account as well.

The basic solution procedure requires the hot and cold stream inlet temperatures be specified. Starting from the 1st row (of the counter flow heat exchanger configuration shown in Figure 3, for example) cold stream outlet temperatures are estimated and then by 'marching' from one row to the next, the inlet cold stream temperature is calculated. This iteration procedure is repeated for different values of cold stream outlet until the calculated cold stream inlet temperature corresponds to the initially



specified value.

Figure 3: Basic design configuration of the HPHE

3.4 Computer program

The thermal design specifications need to be specified for the HPHE and include the atmospheric pressure, hot stream inlet temperature, cold stream inlet temperature, a desired cold stream outlet temperature and the hot and cold stream mass flow rates to be specified (Table 1).

Table 1: Drier user's required specification for the HPHE

Inlet hot temperature	40 – 60°C
Inlet cold temperature	Ambient air
Desired outlet temperature	Whatever is attainable
Mass flow of the air into the condenser section	0.72 kg/s
Mass flow of the air into the evaporator section	0.72 kg/s

Table 2: Design and manufactured definition of the HPHE tested

Working fluid	R134a
Tube bank configuration	Al-Plate and Cu-tube
Evaporator length	0.35 m
Condenser length	0.35 m
Number of tube rows	6
Number of tubes per row	11
Longitudinal pitch	0.0381 m
Transverse pitch	0.0381 m
Fin pitch	10 Fins/inch
Fin thickness	0.0002 m
Outside diameter of tubes	0.01588 m

Inside diameter of tubes 0.01490 m

The next step requires assumed heat exchanger geometry, including the pipe diameter length and evaporator to condenser length ratio, the number of rows and the number of tubes per row, and whether staggered or not, or whether finned or not (Table 2).

The solution now requires a series of nested trial and error iterations such that the evaporator heat transfer rate equals the condenser heat transfer rate, and whether the desired cold steam outlet condition has been met (for the assumed counter flow arrangement as assumed as shown in Figure 3). A flow diagram showing how the computer program (which was written in visual basic) is shown in Figure 4 (Meyer, 2005) and a converged solution is given in Table 3.

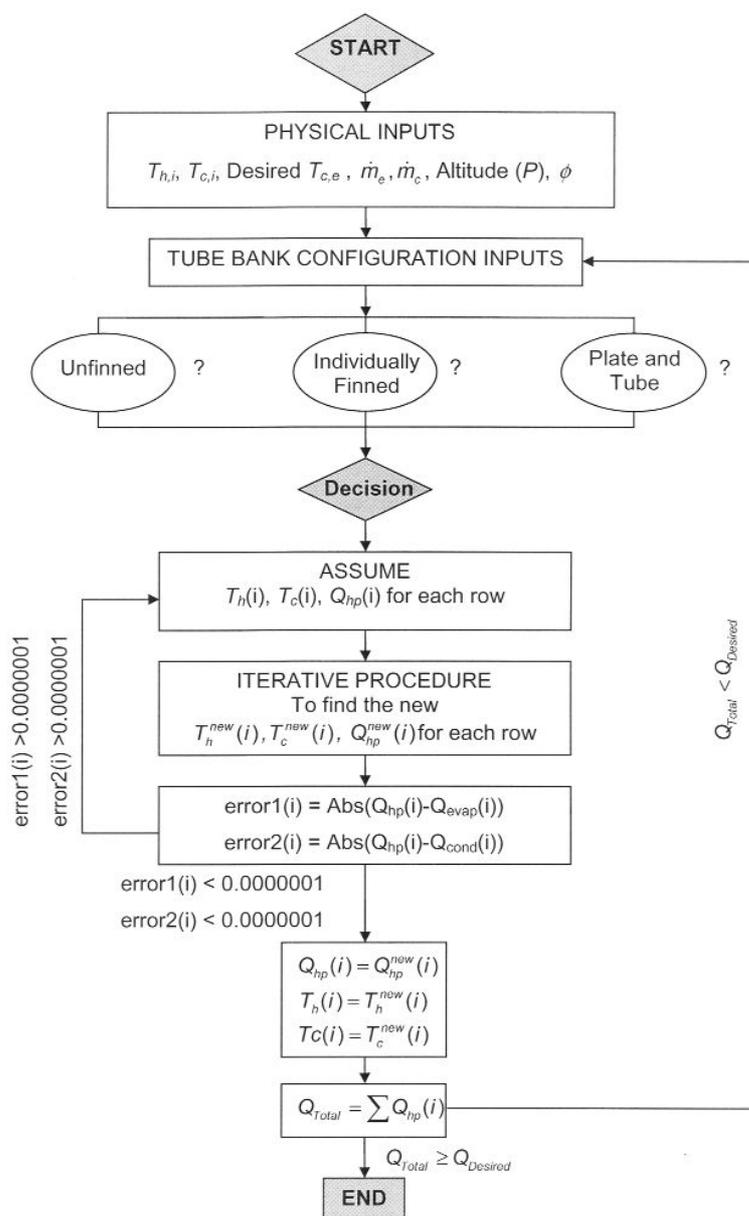


Figure 4: Flow diagram for the main iteration loop of the computer program

Table 3: Theoretically calculated temperatures, heat transfer rate and pressure drop for the HPHE defined in Table 2 (Row i-j refers to the temperature between the rows i and j)

	<i>Inlet, Outlet</i>	<i>Row 1-2</i>	<i>Row 2-3</i>	<i>Row 3-4</i>	<i>Row 4-5</i>	<i>Row 5-6</i>	<i>Outlet, Inlet</i>
Hot stream temperature [°C]	50.00	47.87	45.73	43.6	41.46	39.32	37.19
Cold stream temperature [°C]	34.82	32.69	30.55	28.41	26.28	24.14	22.00
Total Pressure Drop [Pa]				281.53			
Total heat load [W]				9297.95			

4 Experimental set-up

The experimental set-up consisted of the HPHE retrofitted to a standard drier using flexible ducting as shown in Figures 5 and 6. For the drier, its overall dimensions are 2.8 m long, 1.4 m wide and 1.9 m high. The overall dimensions of the HPHE are

shown in Figure 7, its detailed specifications are given in Table 2, and it was manufactured in accordance with standards and technology for copper pipe and aluminium plate finned heat exchangers as normally applied in the HVAC industry.

The air-drier typically evaporates water from the product being dried and exhausts this moist warm air into the atmosphere. With the HPHE installed,

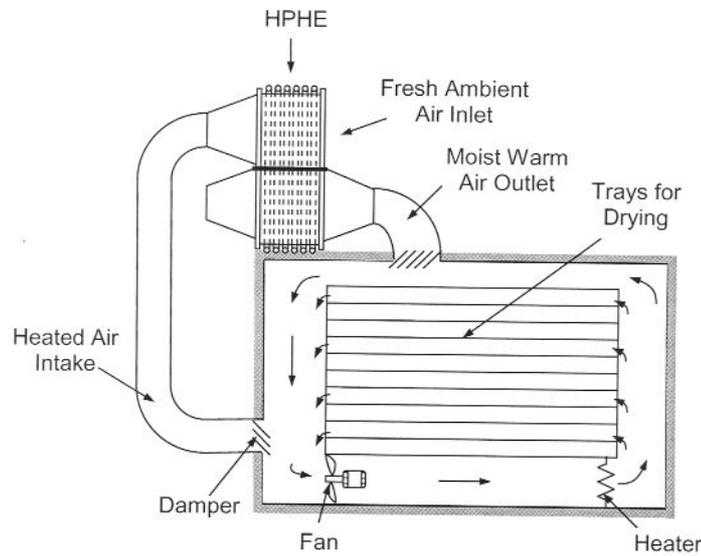


Figure 5: Schematic diagram of the HPHE retrofitted to the mini-drier

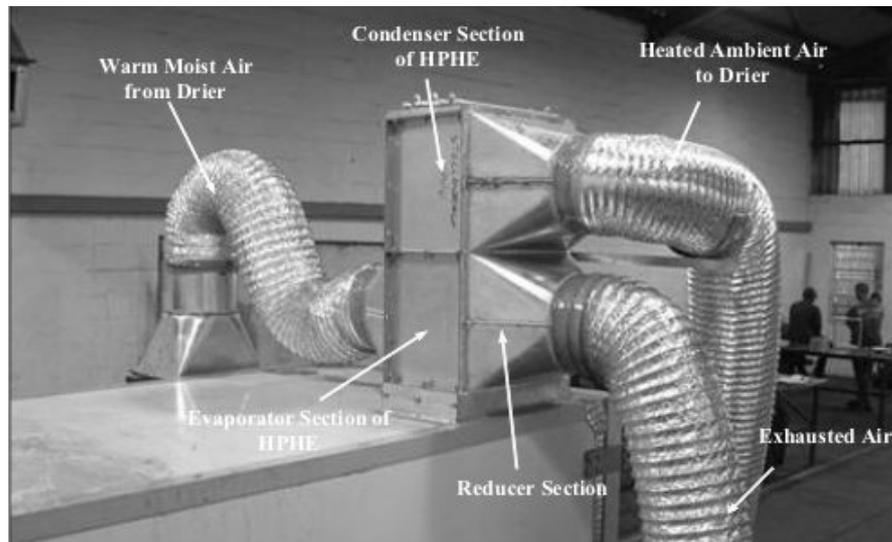


Figure 6: Image of the HPHE retrofitted to the mini-drier

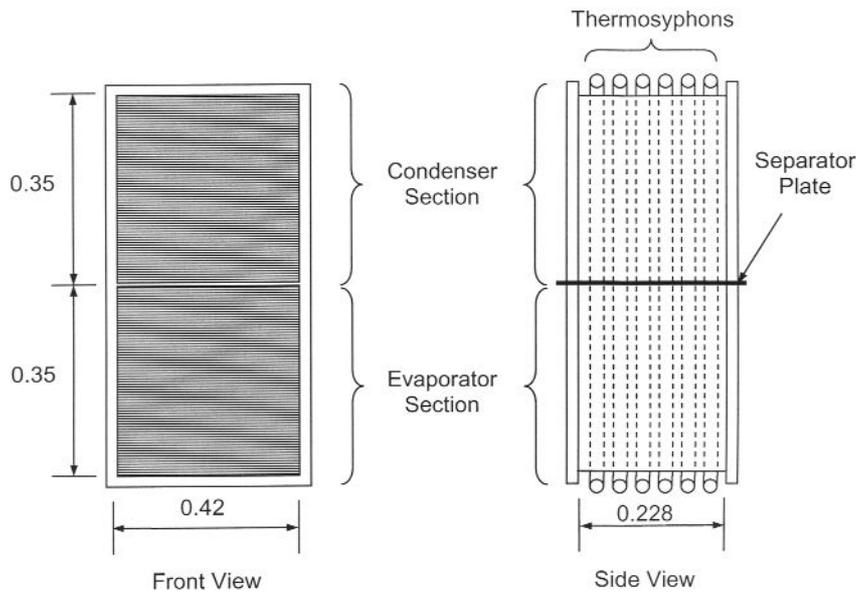


Figure 7: The as-designed and manufactured HPHE (also, see Table 2)

this warm moist air is then fed through the evaporator section of the HPHE. Fresh ambient air is then drawn through the condenser section of the HPHE, where it is heated up and is then fed back into the system thereby reducing the load on the heating elements inside the drier.

Temperature measurements were taken at the inlets and outlet of the respective hot and cold streams, and an anemometer was used to measure the flow velocities, from which the air mass flow rates could be calculated. A kWh-meter was used to measure the electrical energy consumption. To ensure accuracy and repeatable drying operations, the product to be dried was simulated using wet towels laid out on the drying racks. Tests runs with and without the HPHE could thus be compared.

5 Results and discussion

5.1 Drier-HPHE experimental results

The temperature and heat transfer rates for the first 45 minutes of operation of the drier (with the HPHE fitted) are given in Figure 8. After 35 min-

utes, the drier thermostat (set at a temperature of 50°C) starts to control the drier temperature and the heat transfer rate between the hot and cold streams varies between 7 700 and 9 000 W.

The drying period was continued for a total of 5 hours and the kWh-meter electrical power consumption readings at different times are given in Figure 9. With the HPHE installed, the total drier power consumption is 35.4 kWh and, for identical operation of the drier, but without the HPHE, the power consumption is 52.2 kWh.

5.2 Experimental verification of the theoretical HPHE design

The experimental heat transfer rates at different hot and cold stream temperature (as reflected in Figure 8) are compared with the theoretical predicted values in Figure 10. For temperature differences above 15°C, the mathematical model results compare reasonably well with the experimental values.

At the lower temperature differences, the correspondence becomes less favorable and erratic. This

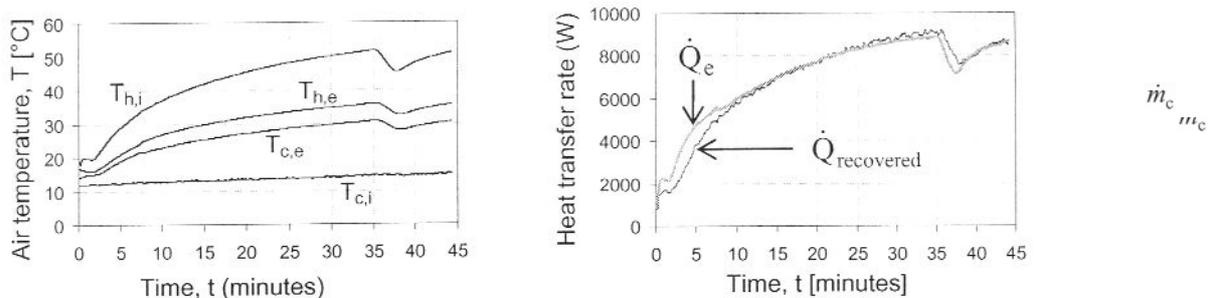


Figure 8: Experimental temperature and heat transfer rate as a function of time for the HPHE for $\dot{m}_c = 0.562 \text{ kg/s}$ and $\dot{m}_h = 0.571 \text{ kg/s}$

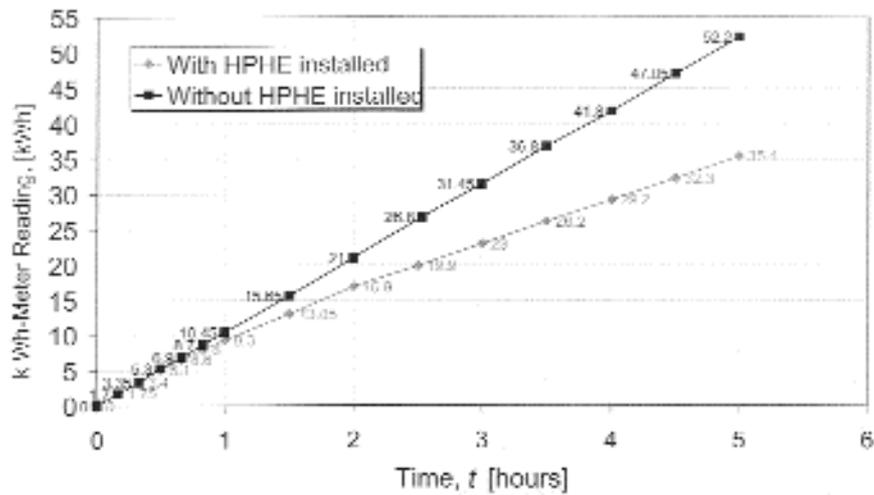


Figure 9: Energy consumption (kWh-meter readings) for the drier unit operation with and without the HPHE

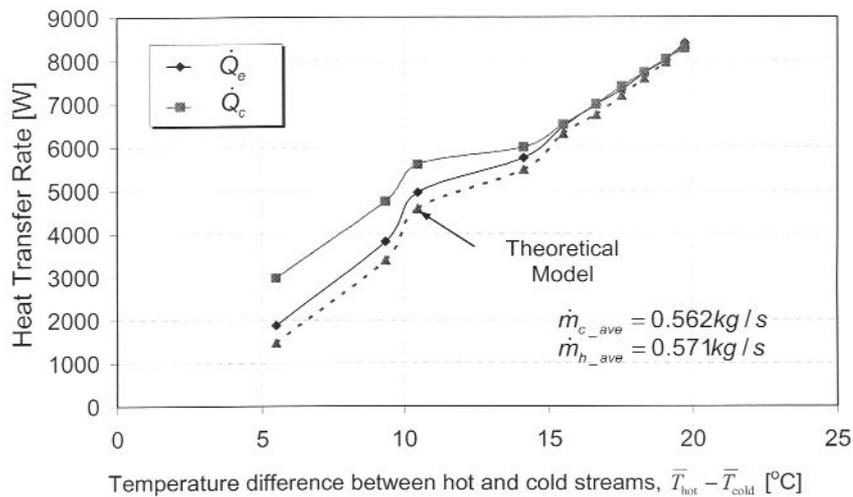


Figure 10: Theoretically predicted and experimentally determined HPHE heat transfer rates as a function of different average temperature differences between the hot and cold streams

is to be expected because at the lower temperature differences errors in temperature measurements become more significant. The smaller the temperature differences, the greater the heat exchange area needed to transfer the same amount of heat.

A larger heat exchange area implies a more expensive heat exchanger and, hence, to keep costs down it is preferred to operate heat exchangers at temperature differences greater than 15°C. The reason for the 'kink' at a temperature difference of 11°C is attributed to a discontinuity at the range crossover point from one heat transfer coefficient correlation to another used in the theoretical model.

5.3 Economic evaluation

The results of an economic evaluation of the energy saving using the HPHE are given in Table 4. The actual material and labour (costs including a nominal mark-up) incurred to retrofit the HPHE are

reflected in Table 4, which amounted to R7 469, and with the energy saving minus additional running costs of R2 321 yielded a simple payback period of 3.2 years. The anti-corrosion epoxy protective coating of R2231 constituted a significant additional cost.

The manufacturer of the HPHE claimed that their experience showed that it would not be necessary for typical agricultural produce drying. The manufacturer of the drier, on the other hand, insisted that the protective coating was necessary. Had this cost, however, been incurred, the payback period would have been 2.3 years. Had the HPHE been included in the design as a standard production feature, an even lower payback period of about 1.8 years is deemed possible?

6. Conclusions

The heat transfer rate between the hot and cold

Table 4: Economic evaluation (in 2004-Rands)

<i>Additional costs with HPHE:</i>	
Standard HPHE	R2660
Anti-corrosion epoxy coating	R2231
Variable speed fan	R1000
Ducting	R78
Reducer sections	R 500
Labour	R1000
1 Total HPHE installation cost	R7469
2 Additional HPHE running cost	R400/year
<i>Energy consumption without HPHE:</i>	
Electrical energy consumption	10.44kWh/h
No. of operating shifts per annum	250
Number of hours per shift	18
Cost of electricity	R0.18/kWh
3 Annual cost of energy without HPHE	R8456
<i>Energy consumption with HPHE:</i>	
Electrical energy consumption	7.08kW/h
No of operating shifts per annum	250
Number of hours per shift	18
Cost of electricity	R0.18/kWh
4 Annual cost of energy with HPHE	R5735
5 Saving per year (3 – 4 – 2)	R2321
6 Simple payback period (1/5)	3.2 years

streams of the heat pipe (thermosyphon) heat recovery heat exchanger is accurately predicted by the theoretical model for average temperature difference between the two streams of greater than 15°C.

The experimental evaluation of the heat recovery heat exchanger retrofitted to the mini-drier yielded a 32% saving of R2 321 per annum (item 5, Table 4) and for the total heat pipe installation cost of R7 469 (item 1, Table 4) a simple payback period of $7469/2321 = 3.2$ years was obtained. A significantly lower payback period is deemed possible if the heat exchanger be included in the design of the mini-drier as a standard production feature.

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