CHAPTER FIVE
DEVELOPMENT OF HEAT PUMP AIR DRIER (HPAD)

SUMMARY

Air drier of heat pump type was studied theoretically and experimentally. A one-ton capacity window type air conditioning set was modified to work as an air drier. Laboratory scale experiment was performed to work out for constraints and limitations of the adoption of air conditioning unit as an air drier. Condenser overheating is likely to occur unless an extra condenser coil is added to the unit. Bypass technique that might enhance dehumidification was studied, no significant effect was found, though. A full scale air drier unit of capacity 3.6 ton (refrigeration effect) was constructed and tested in two actual rubber smokings. The reduction of the smoking time was obvious in one test and not so obvious in the other. Explanation of the unfavorable result one was given based on higher loading density, lower air flow rate and moister rubber sheets.

5.1 INTRODUCTION

The previous chapters have concluded that energy saving measure, although economically feasible, is not practically acceptable by the factory owners. The smoking process can possibly be accelerated by employing dry firewood and dry inlet air. Dry firewood can be obtained by leaving the stock pile of firewood under sun light. Natural solar drying is usually sufficient. But for the rainy season keeping the stock pile undershed is required and may not be cost effective for such indoor storage since substantial capital cost is needed for a roof-covering space.

Moisture of inlet air was found to have an important role in rubber curing time (Chapter 4). Since nearly half of the water in the process comes from the inlet air humidity, development of a viable air drier is necessary then. Many air dehumidification methods were examined. Wet air can be treated chemically with water absorbent substances such as silica gel and lithium chloride. Preliminary study on water absorption capability of granular silica gel found that the gel could absorb water up to 27% of its own weight. Thus, substantial amount of gel is required if the gel is used in an air drier apparatus. Furthermore, heat is needed for the gel regeneration. The commercially available air dehumidification apparatus appears in a form of rotary sorption wheel, which is traversed by two air streams separated by seal as shown in Figure 5.1. The part of the wheel which is exposed to "process air" absorbs moisture from the air while the area exposed to "reactivation air" (hot air) releases the moisture into the reactivation air. More importantly, the unit cost is about 500,000 Bahts (U$ 20,000) (1992 price) which is unacceptably high. Problems in long-term maintenance and poor performance due to repeated regenerations and deposit of particulate on the gel are other factors that make the silica gel-based apparatus very doubtful and not so attractive. Lithium chloride is an interesting substance as it can absorb water up to three times of its own weight. But the price of the chemical is very high and problems similar to silica gel still exist.

Limitation of the chemical method has led to an idea that applies refrigeration principle; evolving an apparatus called "heat pump air drier" -HPAD. HPAD is an equipment that adopts (with modification) air conditioning cycle. The cycle consists of cooling, dehumidification and heat recovering (CDHR) processes.

5.2 PRINCIPLE OF HPAD

Principle of refrigeration cycle is employed to dehumidify the air by cooling the air to a temperature that the predetermined amount of water in the air is condensed. The cooled but dry air passes through the condenser to recover heat. The air drier is shown schematically by Figure 5.2. Refrigeration air drier is sometime known as heat pump air drier because, due to its main application in close-cycle drying process, it recovers heat and as a consequence dehumidifies the air as shown in Figure 5.3. The close-cycle system is very effective because it can extract more water from the incoming hot and humid air [23,24].

Close-cycle HPAD cannot be adopted to the rubber smoking room because of suspended particles in the smoke will deposit on the

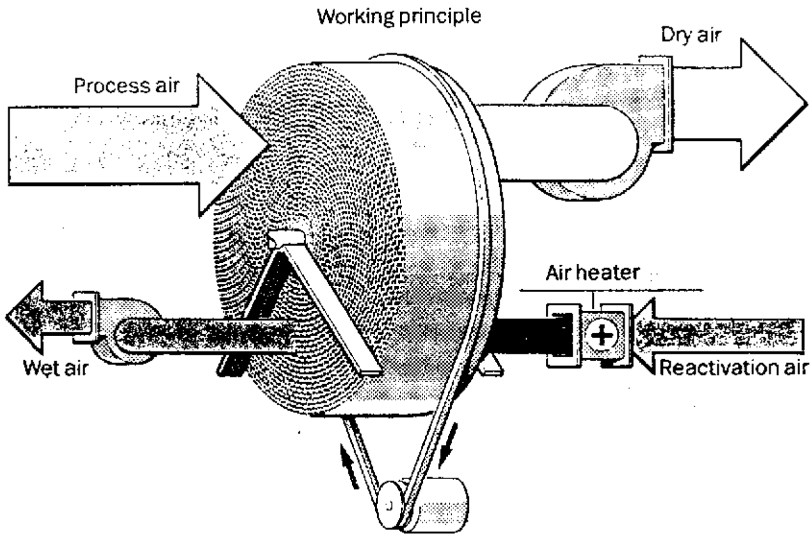


Figure 5.1 Rotary sorption wheel air drier

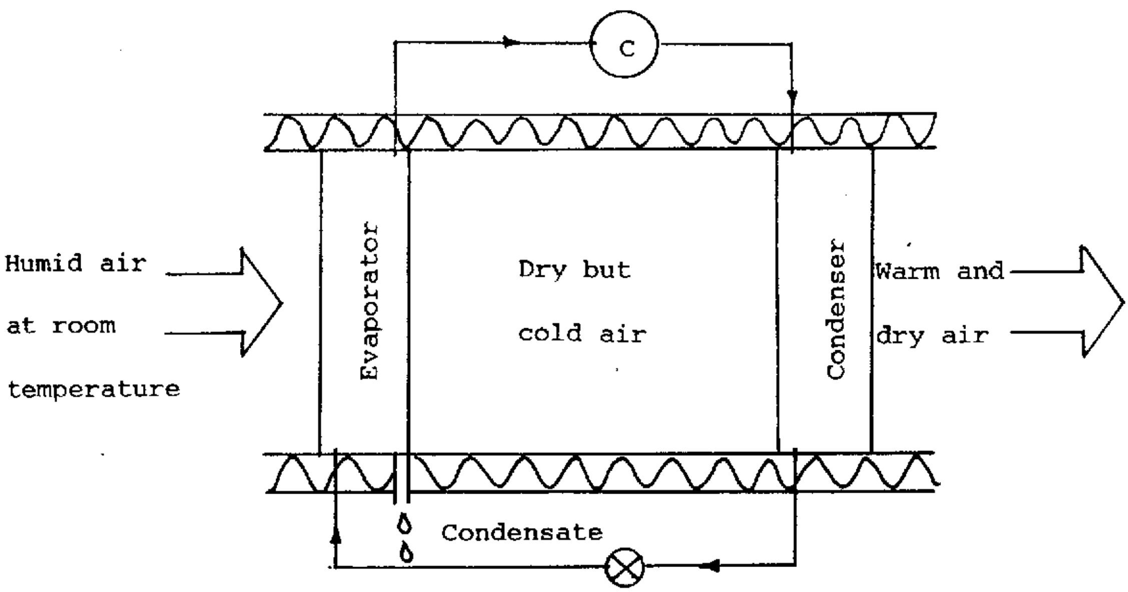


Figure 5.2 Principle of refrigeration cycle modified for air drier

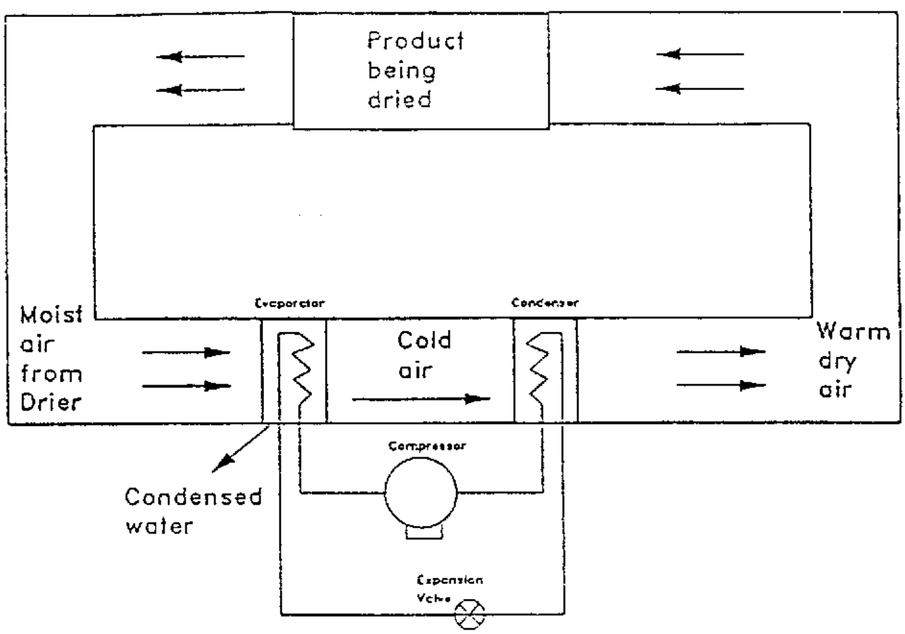


Figure 5.3 Close cycle heat pump air drier

components and reduce heat transfer effectiveness. Fin damaged by corrosion is another undesirable consequence. Therefore, an air drier implemented in this project was limited to an open-cycle type.

Air at ambient condition (typically 32-35 °C and 70-90% RH) is sucked through a series of evaporator, compressor and condenser. As the air passing through the evaporator, its temperature is reduced below dew point causing condensation. Air leaving the evaporator is at low-temperature saturation condition (100% RH but low moisture ratio). Its temperature is raised by recovering heat from compressor and condenser. The air leaving the system is, therefore, hot and dry (low RH and low moisture ratio).

5.3 LABORATORY-SCALE HPAD

5.3.1 Construction

A 1-ton (cooling capacity) window-type air conditioning unit, Figure 5.4, was modified to be a laboratory-scale HPAD. A covering hood coupled with a return duct and butterfly values, Figure 5.5 (a), was constructed. Air flow rate was measured by sets of pilot tube. Humidities at interesting locations were determined by wet bulb and dry bulb temperature technique. Figure 5.5 (b) gives details of measuring parameters and locations.

5.3.2 Theoretical Background of Refrigeration Cycle

The ideal cycle for vapor compression refrigeration is shown in Figure 5.6 as cycle 1-2-3-4-1. Saturated vapor at low pressure enters the compressor and undergoes a reversible adiabatic compression, 1-2. Heat is then rejected at constant pressure in process 2-3, and the working fluid leaves the condenser as saturated liquid. An adiabatic throttling process follows, process 3-4, and the working fluid is then evaporated at constant pressure, process 4-1, to complete the cycle. Heat absorbed by the evaporator and rejected at the condenser are calculated by equations (5.1)-(5.2), respectively.

$$q_L = h_1 - h_4 \qquad \text{(kJ/kg)} \qquad (5.1)$$

$$q_H = h_2 - h_3 \qquad \text{(kJ/kg)} \qquad (5.2)$$

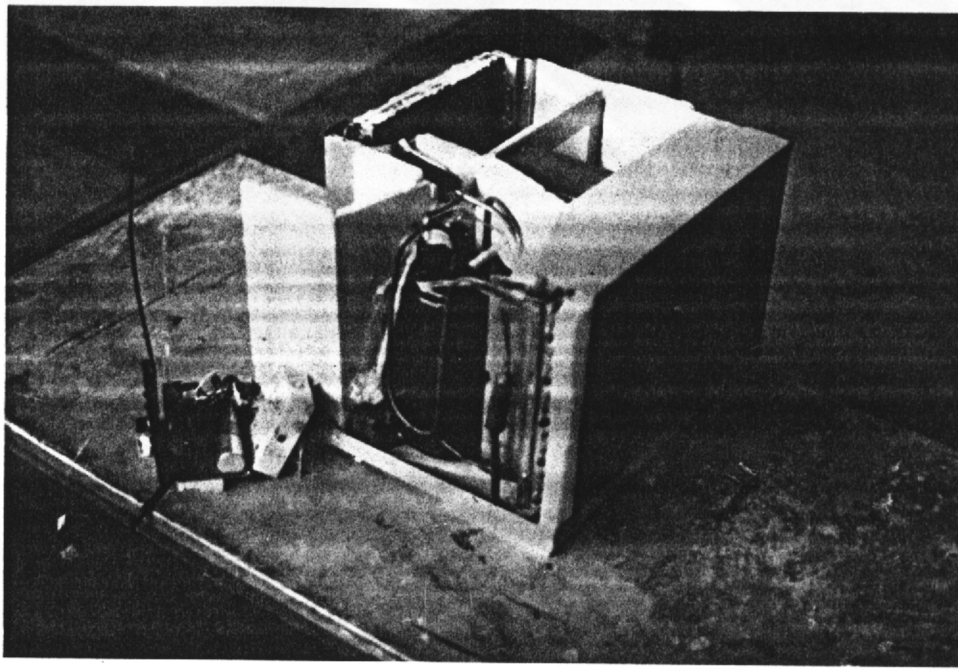
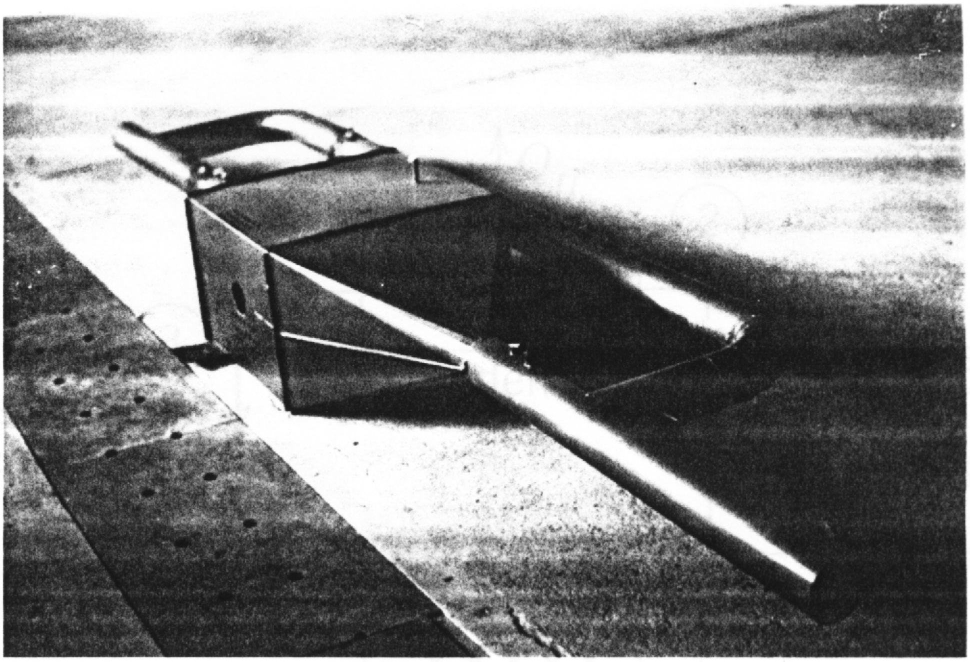
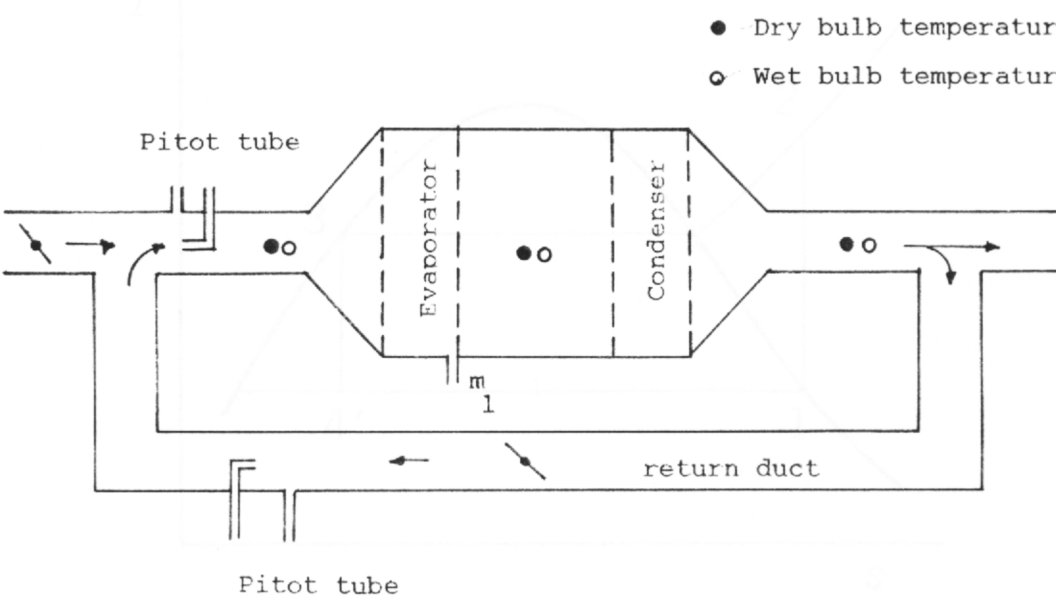


Figure 5.4 Laboratory-scale HPAD

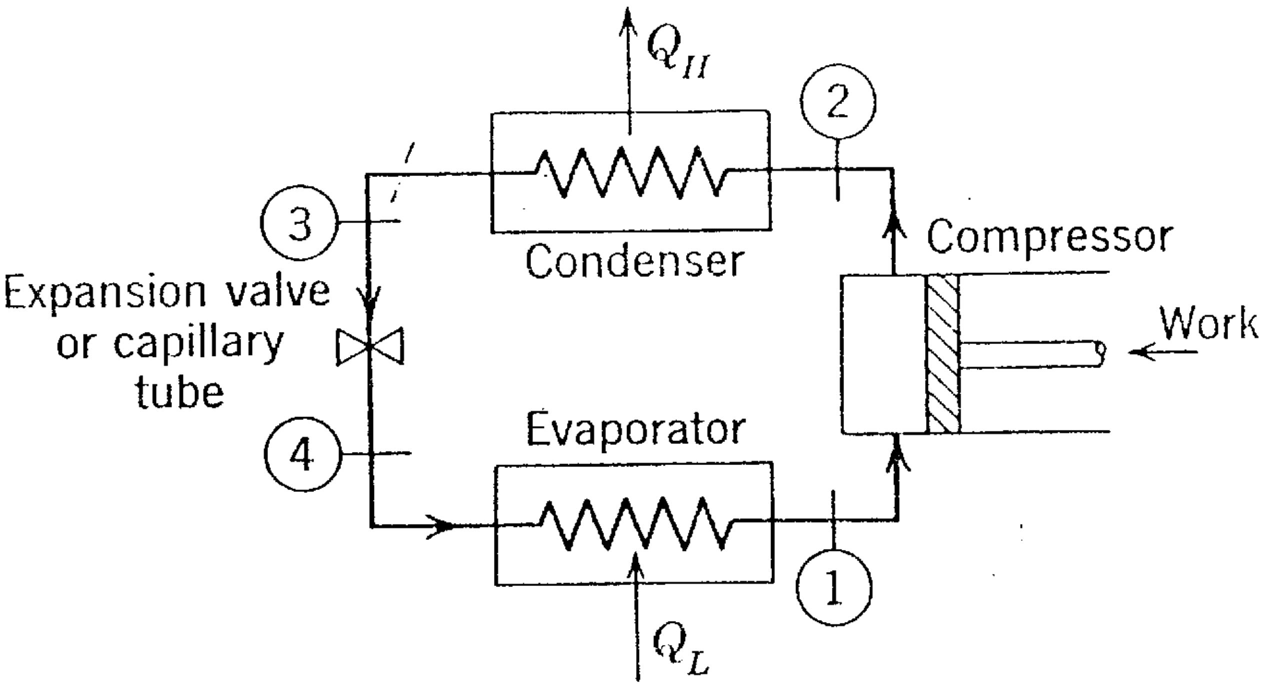


(a)



(b)

Figure 5.5 Covering hood for the laboratory-scale HPAD and parameters measured



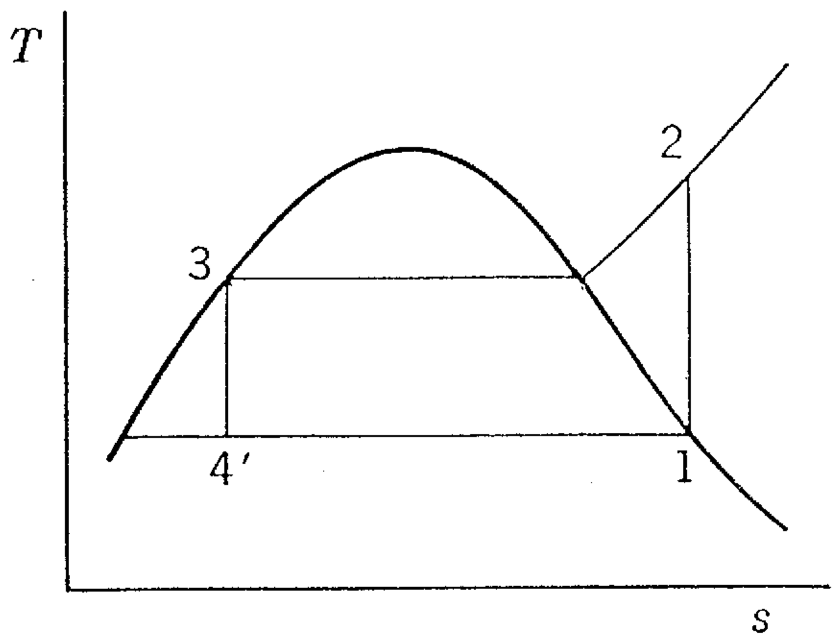


Figure 5.6 Vapor compression refrigeration cycle

The performance of a refrigeration cycle is given in terms of coefficient of performance, β , which is defined as

$$\beta = \frac{q_L}{w_c} = \frac{h_1 - h_4}{h_2 - h_1} \tag{5.3}$$

For the air (mixture of dry air and water vapor), the cooling through the evaporator can be shown by Figure 5.7. Energy equation by the first law is,

$$m_1h_1 + Q_L = m_2h_2 + m_l h_l \tag{5.4}$$

In dealing with air-water vapor mixture the changes in enthalpy of the water vapor can be found from the steam tables and the ideal gas relations can be applied to the air. Let us consider a steady-state, steady-flow process of Figure 5.7, the continuity equations for air and water are

$$\left.\begin{aligned} m_{a1} &= m_{a2} = m_a \\ m_{v1} &= m_{v2} + m_{l2} \end{aligned}\right\} \tag{5.5}$$

Equation (5.4) can be rewritten as,

$$Q_L + m_a h_{a1} + m_{v1}h_{v1} = m_a h_{a2} + m_{v2}h_{v2} + m_{l2}h_{l2}$$

or $$\frac{Q_L}{m_a} + h_{a1} + \omega_1 h_{v1} = h_{a2} + \omega_2 h_{v2} + (\omega_1 - \omega_2)\, h_{l2} \tag{5.6}$$

In the heat recovering process, Figure 5.8, the first law yields,

$$Q_H + m_2h_2 = m_3h_3 \tag{5.7}$$

and $$m_{a2} = m_{a3} = m_a \;, \quad m_{v2} = m_{v3}$$

Hence, $$Q_H + m_a h_{a2} + m_{v2}h_{v2} = m_a h_{a3} + m_{v2}h_{v3}$$

$$\frac{Q_H}{m_a} + h_{a2} + \omega_2 h_{v2} = h_{a3} + \omega_2 h_{v3} \tag{5.8}$$

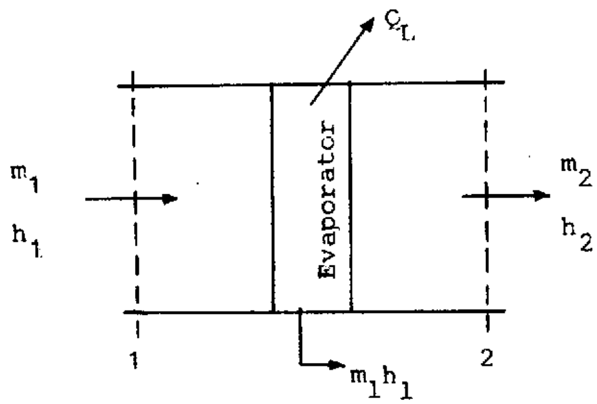


Figure 5.7 First law analysis of mixture cooled by an evaporator

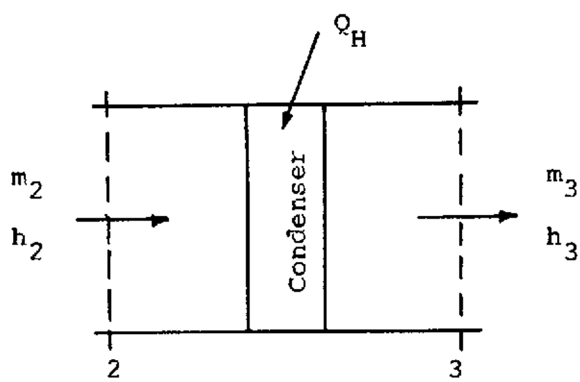


Figure 5.8 First law analysis of mixture heated by a condenser

5.3.3 Experimentation with Simple Air Drier

An experiment was carried out according to the set up shown in Figure 5.5. Inlet condition was varied by mixing the fresh air with the warm and dry return air. Amount of mass flew through the apparatus as well as the moisture content were examined. Results are given in Table 5.1.

The experiment was terminated at test 3 because the inlet humidity could not be further reduced. Humidity lower than 74% requires substantially high portion of return air (warm and dry). This resulted in high temperature of the inlet mixture and consequently raised the condenser temperature to the unacceptable level and caused electricity cut off. With R22 refrigerant, an upper limit of around 60°C applies [22]. In a conventional air conditioning unit, the air stream passing through the condenser is at a much higher flow rate in comparison to that of the evaporator. It seems that an extra condenser coil is needed if a conventional unit is to be adopted for dehumidification purpose. Theoretically, the cooling capacity of the system can be worked out by employing the first law of thermodynamics and principle of refrigeration cycle. Theoretical analysis is given in section 5.4.

5.3.4 Enhancing Dehumidification by Bypass Technique

Young [22] showed that while this basic system works in principle, its operation in practice can be far from effective or efficient. If, for example in the close cycle (Figure 5.3), the air is emerging from the drier at a temperature of around 50°C and a relative humidity of 50%, it will reach its dew point when cooled to about 37°C. If 20 kJ per kg dry air is removed from the air stream, final temperature would be 36°C, and only 2 grams of water per kg of dry air is removed. Most of the energy withdrawn from the air has been spent in removing the sensible heat from the air stream, and is wasted.

Young demonstrated that the heat pump air drier can significantly extract moisture if a bypass passage is provided. The drier in Young's report was installed to a drying cabin. Only fraction of warm and humid air exhausted from the cabin was passing through the evaporator. The rest was bypassed and mixed with the dry air

Table 5.1 Results of Experiment on 1-ton Air Drier.*

Test	Inlet		ω_i	ϕ_i (% at 30 °C)	outlet		ω'_e	ϕ_e (% at 30 °C)	m_l (kg/h)
	T_1(°C)	T_2(°C)			T_3(°C)	T_4(°C)			
1	29.6	26.7	.0212	78	44.6	27.2	.0160	60	1.56
2	30.7	26.4	.0204	76	53.9	27.5	.0135	50	0.56
3	32.8	26.6	.0198	74	56.7	28.9	.0140	52.5	0.46

* Parameter measurement is shown in Figure 5.5 (b)

behind the evaporator as schematically shown in Figure 5.9. Because of low volume flow rate through the evaporator, the after-evaporator air was at very low temperature. Consequently, substantial amount of water was extracted from the air. The mixture behind the evaporator was at lower humidity compared to the process without the bypass. Using the previous example, if 50% of the air was bypassed around the evaporator, the air passing through the evaporator would be cooled to 33°C, removing about 4.5 grams of water per kg of total drying air. Over twice the dehumidification effect has been achieved for the same rate of energy removal.

Even greater efficiencies can be achieved by the addition of conventional heat exchangers (often referred to as "recuperators"), around the evaporator as represented in Figure 5.10. These recuperators may either use recirculated water as the heat transfer medium, or be passive air to air heat exchangers. With the system properly designed, the recuperator before the evaporator can cool the air stream to or slightly below dew point, so that all the heat withdrawn by the evaporator is used for dehumidification.

5.3.5 Experimentation with Bypass Technique

A special passage was provided for bypass air to be mixed behind the evaporator. Air flow rates were measured by mean of orifices. Wet bulb and dry bulb temperatures of air entering and leaving the system were recorded as well as rate of condensation. Twenty one tests were conducted with percentage of bypass varying from 20 to 50. The amount of condensation was insignificantly affected by the amount of bypass. Average mass of condensation for 50% (5 tests), 40% (8 tests) 30% (5 tests) and 20% (3 tests) bypass were .0163 kg/min, .0171 kg/min, .0174 kg/min and .0194 kg/min, respectively. The difference in condensation might be the effect of difference in the inlet humidities (the 21 tests were run on different day and time). It is obvious that the bypass technique gave unfavorable result for inlet air at room temperature (30°C). However, the bypass technique was reported working well at high inlet temperature (50-55°C) [22]. This can be explained by the phenomenon that the absolute humidity (or humidity ratio) of near-saturated air varies nonlinearly with temperature as appeared in the psychrometric chart.

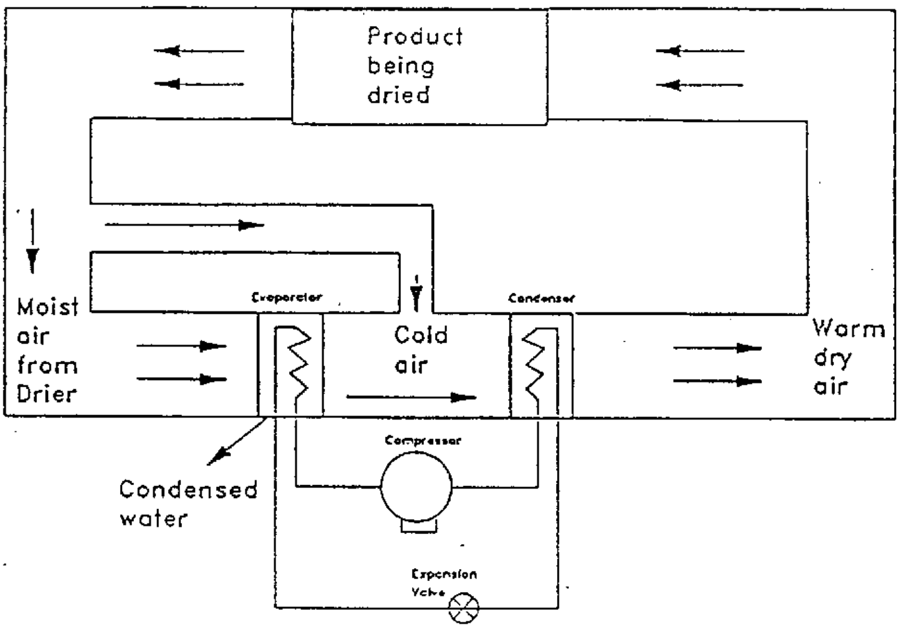


Figure 5.9 Enhancing air dehumidification by bypass technique

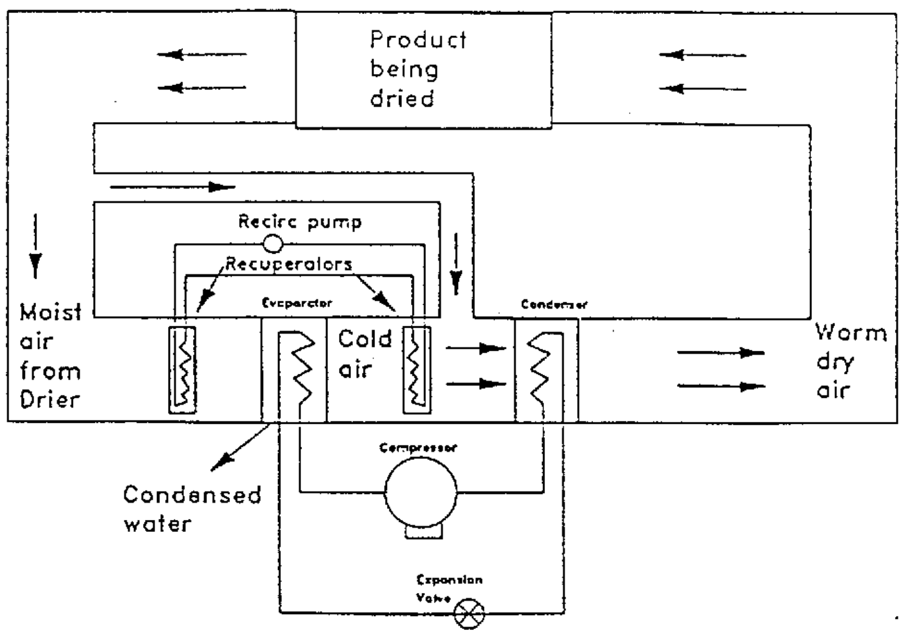


Figure 5.10 Air dehumidification with recuperator

5.4 DESIGN AND CONSTRUCTION OF FULL-SCALE HPAD

Results from the previous section concluded that it is possible to dehumidify the moist air by the cooling-dehumidifying-heat recovering process. Chapter 4 also revealed that dry air of relative humidity of 40% is desirable. Monitoring of rubber smoking process, Chapter 3, found that air flow rate into the smoking room was measured as 700 m^3/h. Hence, the full scale HPAD must be capable of dehumidifying 700 m^3/h of ambient (humid) air so that 40% RH (at ambient temperature) is achieved.

5.4.1 Determination of Cooling Capacity

The calculation was based on the following assumptions.

Ambient condition		$P = 0.1$ MPa, $T = 30\,^\circ C$, $\Phi = 80\%$
Final condition		$P = 0.1$ MPa, $T = 30\,^\circ C$, $\Phi = 40\%$
Air flow rate	=	700 m^3/h, air density = 1.12 kg/m^3

From steam table at 30 °C $P_g = 4.246$ kPa

$$\Phi = 0.80 = \frac{P_v}{P_g}$$

Therefore,
$$P_v = 0.80 \times 4.246$$
$$= 3.40 \quad \text{kPa}$$
$$P_a = P - P_v$$
$$= 100 - 3.40 = 96.6 \quad \text{kPa}$$
$$\omega_1 = 0.622\ \frac{P_v}{P_a}$$
$$= 0.622 \times \frac{3.4}{96.6} = 0.0219 \quad kg/kg_{air}$$

From psychrometric chart at 30 °C, Φ 40% yields $\omega_2 = 0.0108$

The air leaves the evaporator as saturated mixture which gives $P_g = P_v$. Hence,

$$0.0108 = 0.622 \times \frac{P_g}{100 - P_g}$$

Thus,
$$P_g = 1.705 \quad \text{kPa}$$

From steam table with P = 1.705 kPa, the saturated temperature just behind the evaporator is 15 °C. Water extracted from the air can be calculated as

$$m_l = (\omega_1 - \omega_2)\ m_a$$

$$= (0.0219 - 0.0108) \times 700 \times 1.12$$

$$= 8.7 \ \text{kg/h} \quad \text{at } 15\,^\circ\text{C}$$

Equation (5.6) gives, $\frac{Q_L}{m_a}$

$$\frac{Q_L}{m_a} = h_{a2} - h_{a1} + \omega_2 h_{v2} - \omega_1 h_{v1} + (\omega_1 - \omega_2)\ h_{l2}$$

$$= 1.0035(15-30) + 0.0108 \times 2528.9 - 0.0219 \times 2556.3 + (.0219-.0108) \times 62.99$$

$$= -43.02 \quad \text{kJ/kg}$$

m_a was estimated as 767 kg/h (ρ =1.12 kg/m^3, ω = 0.0219, flow = 700 m^3/h)

$$\therefore \quad Q_L = -32996 \quad \text{kJ/h}$$

$$= 31296.7 \quad \text{Btu/h}$$

$$= 2.6 \quad \text{ton (refrigeration effect)}$$

It is reasonable to assume that the cycle is operating at COP = 3.0, therefore work of the compressor is

$$w_c = \frac{32996}{3 \times 3600} = 3 \quad \text{kW}$$

In ideal condition (e.g. reversible heat transfer process) an air conditioning unit of 2.6 ton refrigeration capacity is required. Conventional air conditioner is not designed for air dehumidification and the air temperature of 15 °C (for 700 m^3/h) could not be possibly obtained for the 2.6 ton unit. It was decided that a unit of 3.6 ton capacity should be selected for this experiment.

5.4.2 Modification of Conventional Air Conditioner

Preliminary study in section 5.3 revealed that the outlet temperature from the condenser coil was very high due to limited flow rate of the air through the condenser. Therefore, the modified unit consisted of an extra condenser. Cooling fan of the additional condenser was operated only when the temperature of the condenser exceeded a set limit. Figure 5.11 shows the appearance of the modified air conditioner for air dehumidification purpose.

The air drier underwent trial test in the laboratory. Results of the test were summarized in Table 5.2. Calculation for refrigeration capacity based on equation (5.6) yielded a figure of 2.77 tons which is 77% of the quoted specification. Volumetric flow rate of the inlet air was 759 m^3/h (calculated from ω_1, ω_2, m_1 and associated temperatures) which is enough for the smoking process. Relative humidities of inlet and outlet were 77% (30 °C) and 19.5% (at 46 °C or 46% at 30 °C), respectively. It was considered that although improvement is needed for a better performance, this HPAD apparatus deserved trial tests in a real situation.

5.5 HPAD ASSISTED RUBBER SMOKING

5.5.1 Installation and Experiment Procedure

The unit was installed at the rear of the smoking room near the furnace intake. A flexible duct made from canvas was used to connect the HPAD outlet and the furnace intake hood as shown in Figure 5.12. A kW-h meter was equipped to the HPAD to record electrical energy consumption. Type-k thermocouple was used to monitor temperatures and humidity (wet bulb temperature) at inlet and outlet of the HPAD. Temperatures, air flow rate, amount of condensate and firewood consumption were recorded. Air velocity in the duct and hence the flow rate was determined by a vane type anemometer.

5.5.2 Results and Discussion.

There were two smoking experiments with different flow rate conducted with the aid of HPAD. Results are tabulated in Table 5.3.

Wet bulb and dry bulb temperatures were measured for inlet air to the HPAD, outlet of the HPAD (entering furnace) and exhaust from the smoking room. Analyzed result in shown in Table 5.4. Humidity ratios of air entering the HPAD, entering the furnace and leaving the room are presented in Figure 5.13. Unlike humidity ratios across the HPAD, the exhaust humidity ratio exhibited substantial fluctuation.

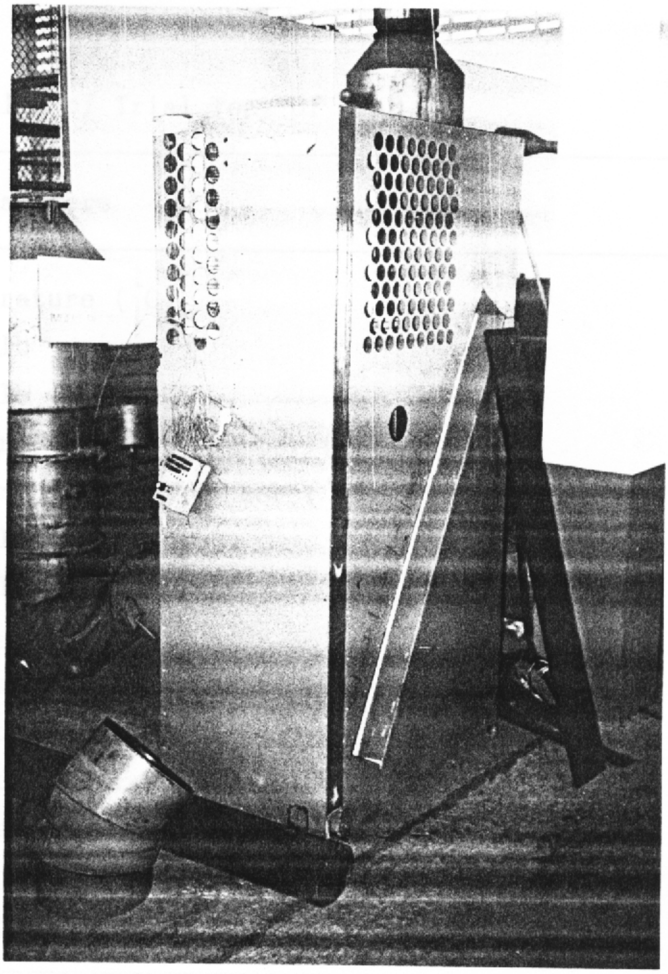


Figure 5.11 Air drier constructed for field trial

Table 5.2 Results of Trial Test of HPAD

Measuring parameters	
Ambient temperature (°C)	
- Wet bulb	26.5
- Dry bulb	30.0
Air temperature after evaporator (°C)	9.6
Air temperature after condenser (°C)	
- Wet bulb	25.5
- Dry bulb	46.0
Condensate (kg/h)	7.0
HPAD cycle temperature (°C)	
- at condenser entry	64.0
- at condenser exit	37.9
- at drier exit	24.1
- at evaporator entry	14.9
- at evaporator exit	7.9

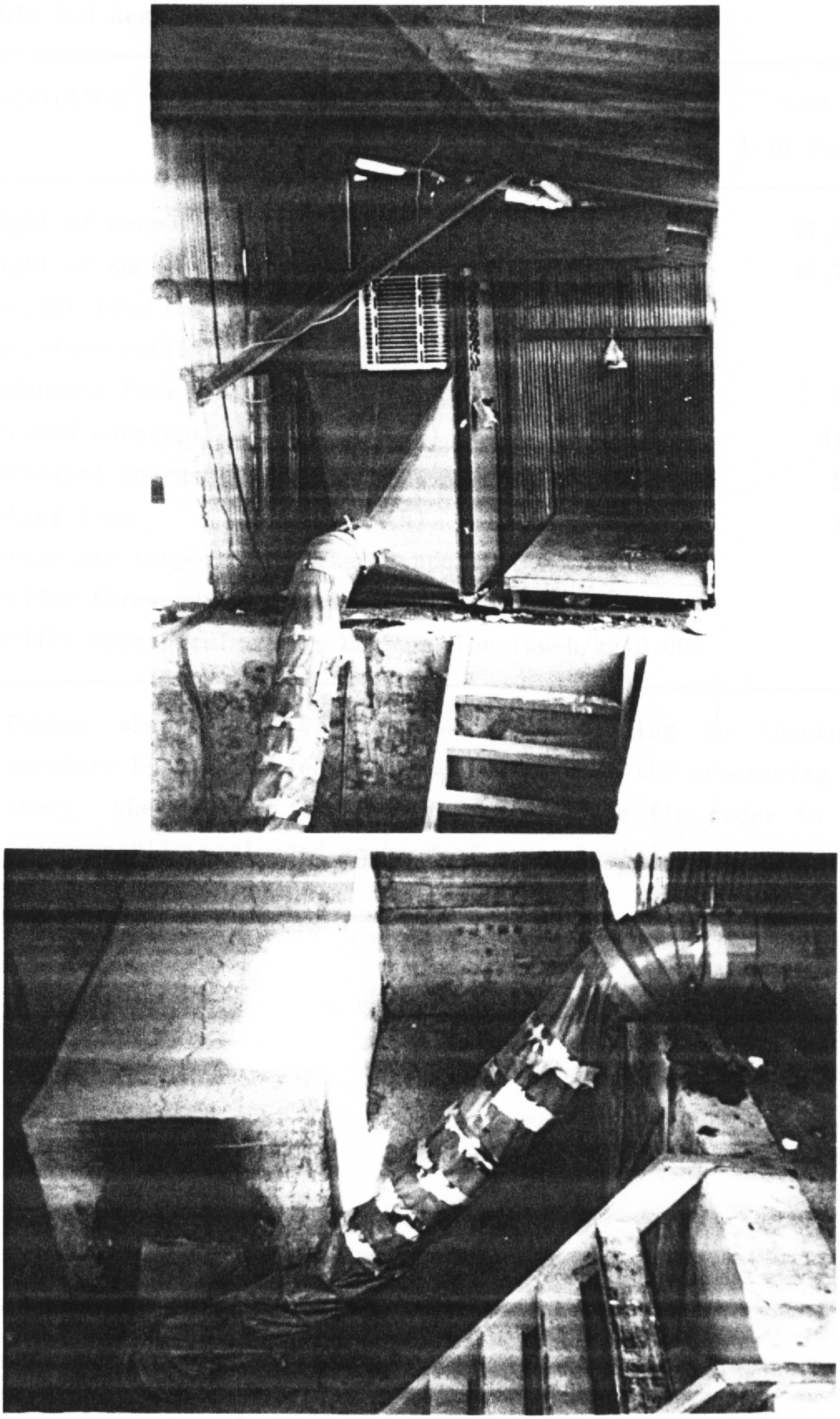


Figure 5.12 Connection of HPAD to the furnace

Table 5.3 Results of HPAD-Assisted Smoking

Description		Test 1 7-11 Aug. 1992	Test 2 4-10 Sept. 1992
Weight of unsmoked rubber (kg)		39,488	47,555
Weight of smoked rubber (kg)		38,338	45,744
% weight loss (moisture, dry basis)		3.00	3.96
% moisture estimated by factory*		2.90	3.90
Condensate from HPAD	(kg)	514.15	725.25
Firewood consumption	(kg)	2842.3	3413.5
Electrical energy consumption	(kW-h)	363	588
Smoking time	(h)	86	142
Average air temperature after evaporator (°C)		5.92	5.60
Specific firewood consumption (kg wood/kg rub)		0.074	0.075
Specific electrical energy consumption (kw-h/kg)		0.009	0.013

* Rubber sheets were visually sorted according to thickness and moisture by an experienced personnel so that the processing time of every sheets in the room was the same (in order to prevent unnecessarily prolonged smoking of some sheets).

Table 5.4 Analysed Result of HPAD-Assisted Smoking

Parameters @	Test 1	Test 2
Humidity ratio of (average value)		
- air entering HPAD	0.0194	0.0194
- air leaving HPAD	0.0133	0.0082
- exhaust from smoking room	0.0410	0.0360
Relative humidity of exhaust (%)	85	68
Air flow rate (m^3/h)	881	412
Condensate (kg/h)	5.91	5.07
Water inherent in air entering HPAD (kg)	1646	1270
Water inherent in air entering furnace (kg)	1129 (36%)*	537 (16%)*
Water emitted from firewood** (kg)	843 (27%)	1013 (30%)
Water extracted from rubber (kg)	1150 (37%)	1811 (54%)
Water consisted in exhaust (kg)	3122 (100%)	3361 (100%)

@ all calculation, otherwise indicated, are based on average humidity ratio

* figures in bracket represent percentage with respect to water in exhaust

** based on 42.2% moisture dry basis

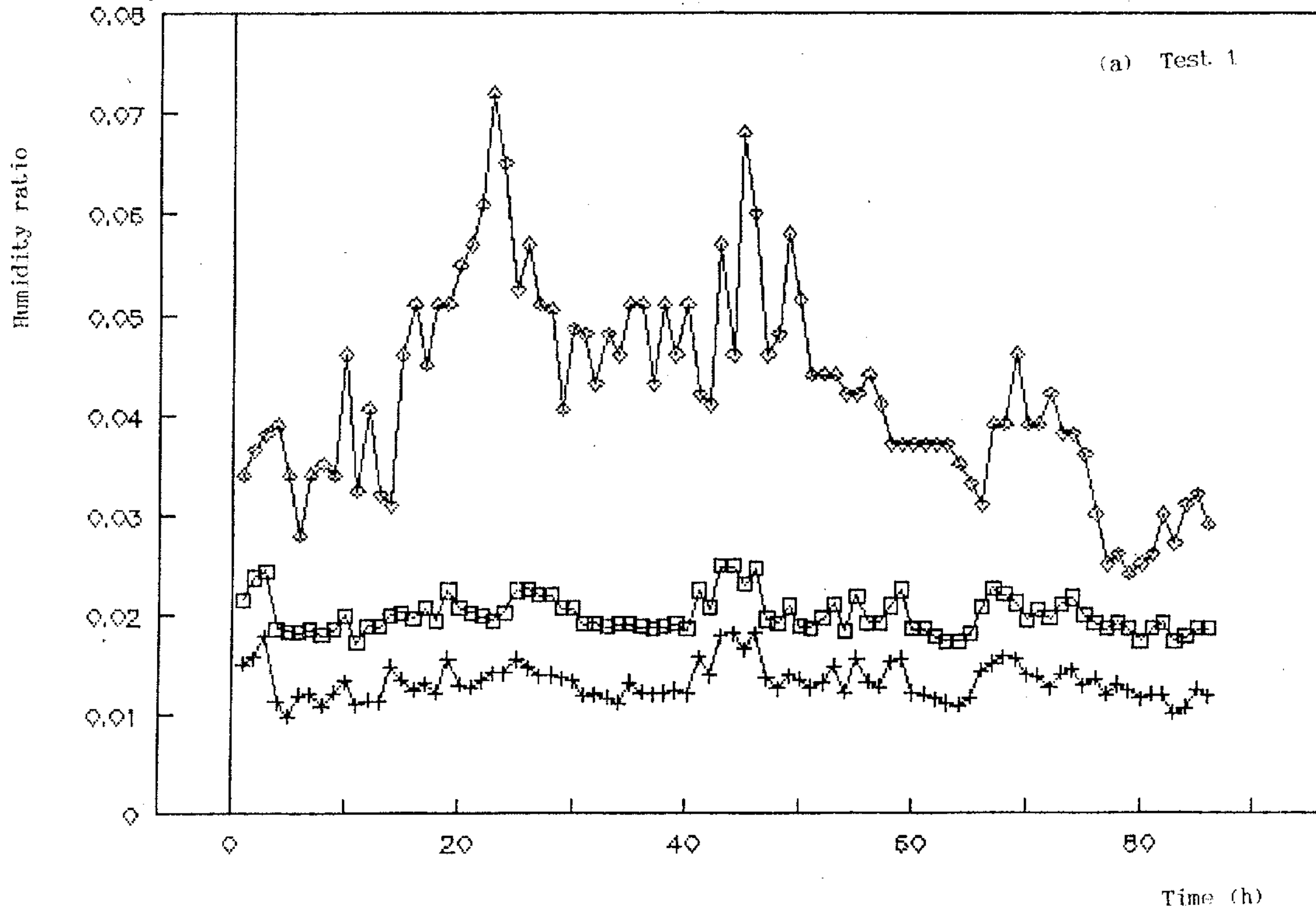


Figure 5.13 Humidity ratios of inlet air and exhaust

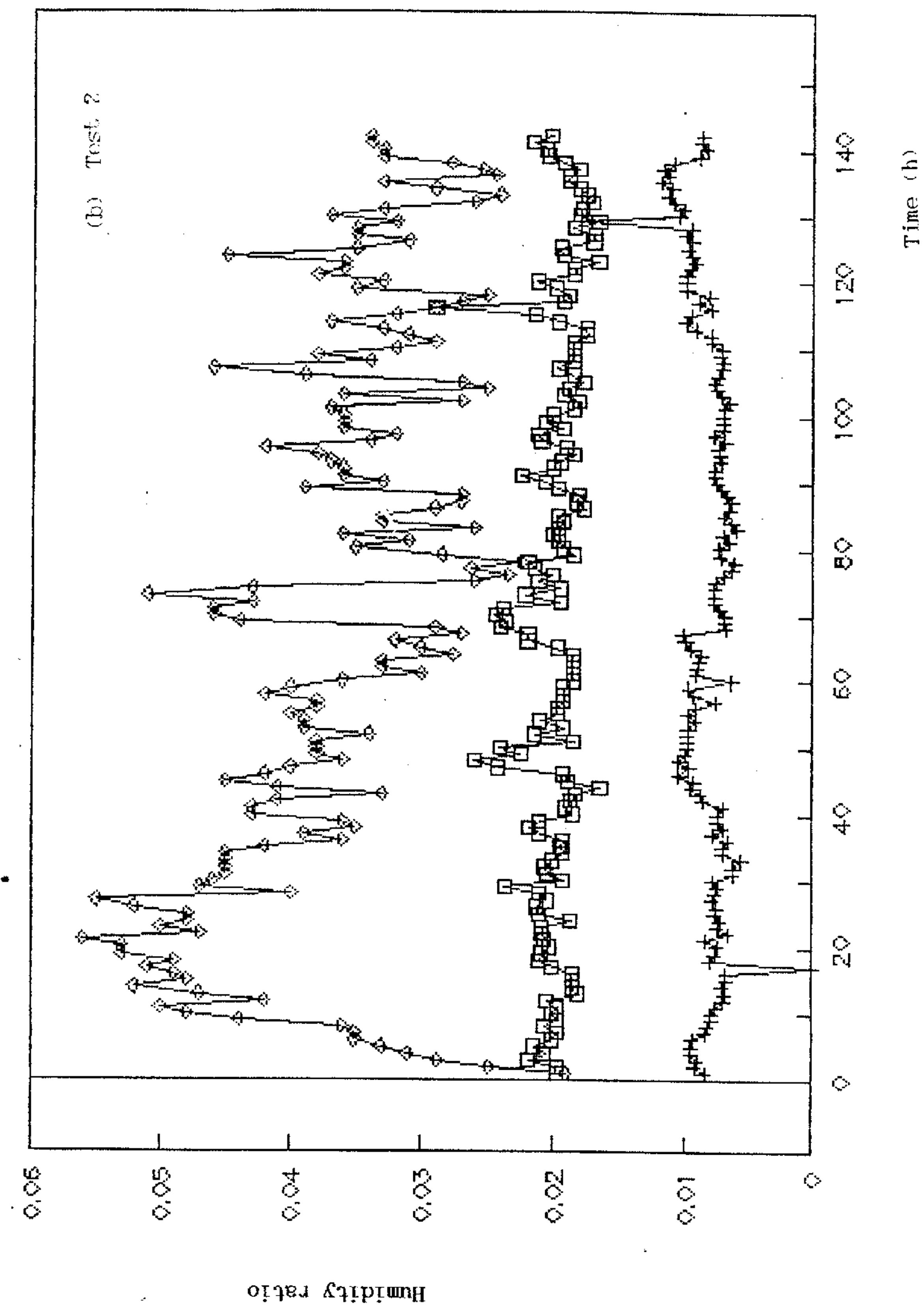


Figure 5.13 (cont.)

Since volume flow rate of the inlet air was relatively constant and sampling period was also constant (every 1 hour) throughout the experiment, average values of humidity ratios were used in the subsequent calculation with an acceptable error. Comparison of the role and involvement of water (moisture) when smoked the rubber with and without the aid of HPAD was shown in Table 5.5.

Generally, the furnace operator, based on his experience, is the one who estimates the smoking duration. It was estimated that the normal operation of Test 1 and Test 2 (the last 2 columns of Table 5.5) should take approximately 120 hours and 155 hours to finish the process, respectively. This implies that at least a significant reduction in processing time was achieved in Test 1. However, it was not clear that Test 2 processing time was affected by the application of the HPAD. It was possible that the HPAD in test 2 did not effectively decrease the smoking time. This could be the result of very high loading density in Test 2 (82.5 kg/m^3 compared to 68.5 kg/m^3 for Test 1). In addition, the rubber sheets in Test 2 contained more water (3.9%) in comparison with Test 1 (3.0%) while the air flow rate for Test 2 (412 m^3/h) was only half of that for Test 1 (881 m^3/h). Drier air (Test 2) was obtained with the expense of lower air flow rate. Although only 16% of water in the exhaust came from the inlet air (test 2, Table 5.1), the flow rate was probably too low to affect the smoking time. This conclusion seemed contradictory to the findings presented in Chapter 4. It must be borne in mind that the specific air flow rate in this case was only 0.0086 $m^3/h/kg$ which was lower than the minimum figure of 0.011 $m^3/h/kg$ quoted in Chapter 4 experiments. Furthermore, the physical size of the chamber and the absence of smoke (in Chapter 4) might play a certain role in the smoking process and make the results not comparable. However, it is obvious that water removing rate (from rubber) was significantly improved when the HPAD was employed.

Table 5.3 shows that the HPAD did not affect the specific firewood consumption. Specific electrical energy consumption required to run the HPAD was in the range of 0.009-0.013 kW-h/kg. Taking electricity cost of 1.80 Baht/kW-h, the specific electricity cost becomes

Table 5.5 Effects of HPAD on Rubber Smoking

	Without HPAD*			With HPAD	
	Test 1	Test 2	Test 3	Test 1	Test 2
Water					
- in exhaust (kg)	4460.9	4180.6	4200.5	3122	3361
- extracted from rubber (kg)	1311.9	1255.6	1096.2	1150	1811
- emitted from firewood (kg)	1154.0	1027.3	935.2	843	1013
- inherent in inlet air (kg)	1995.0	1897.7	2174.0	1129	537
Percent of water in rubber	2.9	2.9	2.3	3.0	3.96
Smoked rubber (kg)	45,526	43,850	47,322	38,338	47,555
Smoking time (h)	166.5	110.0	122.5	86	142
Inlet relative humidity at 30 °C (%)	Untreated (≃ 70-80%)			50	32
Water removing rate from rubber (kg/h)	7.88	11.4	8.9	13.37	12.75

* From Chapter 3.

0.016-0.023 Baht/kg. At present the specific firewood cost is in the range of 0.013 Baht/kg (6 m^3 for 45,000 kg of rubber and 100 Baht/m^3). Therefore, running cost for the HPAD-assisted smoking process is not financially competitive (HPAD-assisted method still requires firewood). However, if, in the foreseeable future, the firewood price increases due to increasing demand for consumer product manufacturing factories, e.g., furniture, toy etc., The running cost for the HPAD will be in a better position. It is interesting to note that the investment cost of a heavy duty 4 ton capacity HPAD is in the range of 70,000 Baht [24] which is only 17.5% of the cost for constructing a new smoking room (excluded land). Adding a new room means 100% increment in productivity. Whether the 17.5% investment for, say, 25-30% increment in productivity is acceptable or not is a question to be answered.

5.6 CONCLUSION

An attempt to reduce the smoking time was undertaken based on the principle that the process can be accelerated by introducing dry air into the furnace. A heat pump air drier that cools, dehumidifies and heats the inlet air was constructed from an air conditioning set. It was experimentally verified that the smoking time in actual practice could be reduced by HPAD assisted process. However, economic feasibility study of implementing the HPAD in the rubber smoking industry has to be carried out in full detail. Extensive research, development and engineering must be undertaken in order to obtain a viable air drier.